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Title: **Conceptual Design Analysis of an 8,000mt Crane for HMC's NSCV**

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Subject: **Conceptual Design Analysis of an 8,000mt Crane for HMC's NSCV**

Heerema Marine Contractors (HMC) transports, installs and removes all types of offshore facilities with three of the world's largest crane vessels. In the future a New Semi-submersible Crane Vessel (NSCV) will be added to the fleet, which has to be equipped with a dual lifting capacity of 16,000mt.

Up to now, no in-depth research has been performed by HMC before a certain crane type is chosen for a specific vessel. One of the main reasons is that at the time these vessels were developed (1980s) the variety of available crane types, capable of carrying out very heavy lifts, was limited. From that moment on, this variety has been enlarged.

At this moment only heavy lift cranes with lifting capacities up to 7,100mt are available. Therefore, besides the actual crane choice also attention has to be paid to scaling and upgrading heavy lift cranes to meet the required lifting capacity.

This Master Thesis shall provide a well thought out choice between the available crane types and scaling/upgrading possibilities. Therefore, the following questions have to be answered:

- Do market developments indicate extra requirements for heavy lift cranes in the future?
- Which types of heavy lift cranes are available and are suitable to be installed on the NSCV?
- What are the possibilities of enlarging the lifting capacity and what are the effects on the heavy lift cranes and the NSCV?

The assignment shall provide an advice which crane type(s) show most potential to be installed on the NSCV if a total dual lifting capacity of 16,000mt is required.

It is expected that is concluded with a recommendation for further research opportunities based on the results of this study.

This report should be arranged in such a way that all data is structurally presented in graphs, tables and lists with belonging descriptions and explanations in text.

The report should comply with the guidelines of the section. Details can be found on the website.

The professor,

Prof. G. Lodewijks

CONTENTS

1.0 INTRODUCTION

In the near future a New Semi-Submersible Crane Vessel (NSCV) will be added to the fleet of Heerema Marine Contractors (HMC). The NSCV has to be able to install and remove all types of offshore facilities that require lifting equipment for dual lifting operations up to 16,000mt. Currently, no offshore heavy lift crane is available with such lifting capability.

This Master Thesis investigates a possible scaling and upgrading of current available offshore cranes to meet the 16,000mt lift requirement to provide HMC with an advice which crane type shows most potential to be installed on the NSCV. Not only the type of crane is important, but also which crane characteristics are most optimal for the NSCV.

An existing crane type with maximum available lifting capacity of 7,100mt is the A-frame crane with a roller slew bearing. Its lifting capacity is close to the required 8,000mt, however it has some poor crane characteristics. Two other crane types, the A-frame crane with a bogie slew bearing and the mast crane, have some better crane characteristics than the A-frame crane with a roller slew bearing. However, their lifting capacity is currently 4,500mt respectively 5,000mt. These lifting capacities are far off the required 8,000mt, so crane scaling and upgrading requires a major redesign effort.

Firstly, the context is explored to find the boundaries and the constraints related to the semi-submersible vessel the cranes are installed on, and the constraints related to design and operational crane specifications. Secondly, the offshore market prospects and trends are assessed to search for possible new crane requirements that meet new market needs. In Chapter 4 a selection of crane types is made that are expected to be capable to meet the boundaries, constraints and requirements from potential new markets.

The three most promising crane types are traded off against the set of design, operation and performance criteria. These criteria are weighted against each other by a Multi-Criteria Analysis (MCA) in Chapter 6.

The MCA reveals that one of the main parameters is the air draft of the NSCV. A low air draft makes it possible to meet passage restrictions, thus shortening transit time and getting access to new markets. For A-frame cranes proven designs already exist that can reduce their air draft. For mast cranes such design is not yet available.

Since the air draft is a key factor for the successful scaling of the mast cranes, a conceptual design analysis is performed to investigate the possibilities and design features that reduce the remaining height of a mast crane. The design analysis of this so called “back-mast concept” is focused the technical feasibility and described in Chapter 8. Its main components are conceptually designed and the required adjustments to the mast crane investigated. An essential element in this concept is the so-called back-mast that has to be installed on the mast crane. The effects of having a back-mast on the MCA results are discussed in Chapter 9.

This report is concluded with an advice to HMC which crane type shows the most potential for the NSCV that meets the lifting capacity of 8,000mt. The advice also addresses the engineering effort and risks involved when existing cranes are scale and upgraded to meet the demands for the NSCV.

2.0 PROBLEM DEFINITION

This Master Thesis shall provide HMC with and advice which crane design is most optimal for the NSCV. The NSCV has to be equipped with two cranes able to carry out dual lifts (load suspended by two cranes) of 16,000mt. There are no cranes available on the market that meet the required lifting capacity. The maximum available lifting capacity at this moment is 7,100mt. Therefore, scale enlargement and upgrade possibilities should be assessed next to the trade-off to decide on the best suiting crane type for the specified purpose. The research question of this Master Thesis is formulated as:

Which crane type shows most potential, to be installed on the NSCV, when a dual lifting capacity of 16,000mt is required?

Besides the required lifting capacity also other aspects are relevant. Those are related to vessel characteristics, crane functionality, crane types and crane specifications (footprint, boom hinge point height and lifting height). Specifications related to slewing, luffing and hoisting speeds are not included because they are not typical for a particular crane type. The following section formulates the research boundaries to define the scope of the project.

Vessel characteristics

The vessel, on which the lifting equipment has to be installed, is of the type semi-submersible (semi-sub). Such semi-sub has larger stability than a monohull. Vessel stability is essential during heavy lifting and is defined as its sensitivity to heave, pitch and yaw motions in (extreme) weather conditions.

A semi-sub usually consists of a deck, two floaters and a number of columns between the deck and the floaters (Figure 1). During transit, the semi-sub can be ballasted to a shallow draft so only part of the floaters are submerged. For heavy lifting operations, ballast water is added to submerge the floaters and partly the columns, increasing vessel stability.

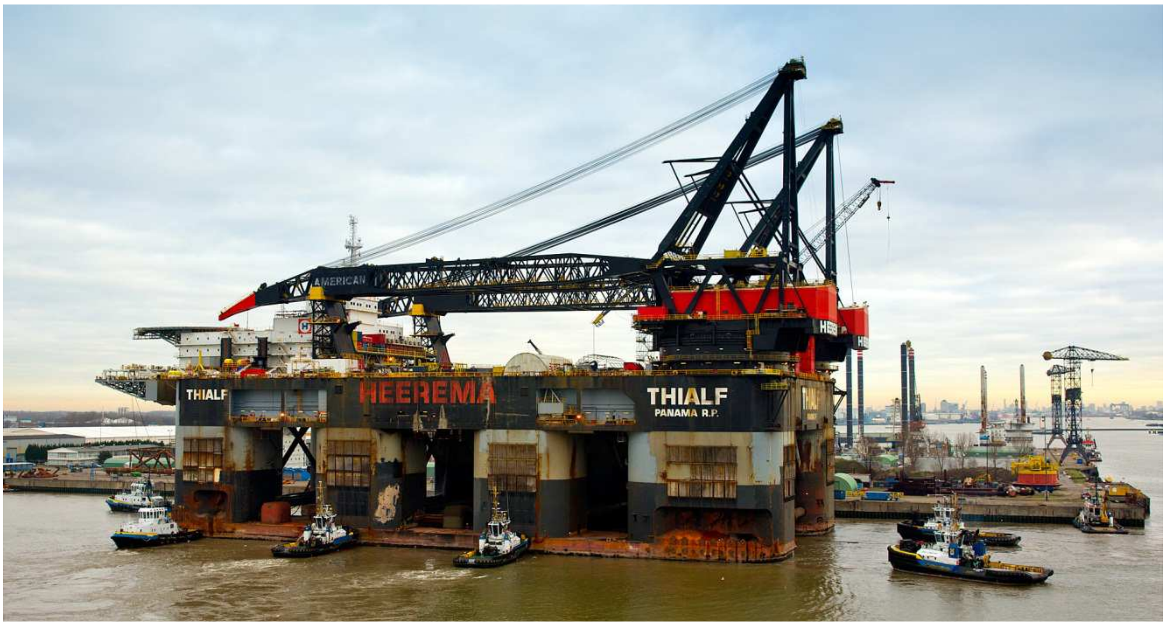


Figure 1: Thialf (Broekhoven)

An example of a semi-sub is the Thialf, shown in Figure 1. The lifting equipment requires a relatively large amount of buoyancy and the floaters of the semi-sub have most buoyancy at their bow. Therefore, the lifting equipment of the NSCV is placed at bow, and above the columns of the semi-sub for optimal load transfer between the two. The dimensions of the semi-sub partly determine the available area for the lifting equipment (width of the NSCV is 93m and the length is 170m).

The NSCV will be used as working island. Therefore a large deck space is required and the area between the floaters is completely covered with deck. This restricts the type of lifting equipment that can be installed. For example the type of lifting equipment on the Svanen (Figure 2) and Ostrea (Figure 3) is not suitable because it restricts the ability to maneuver the vessel around the load.

Figure 2: Svanen 10,000mt lifting capacity (LondonArray)

Figure 3: Ostrea 8,700mt lifting capacity (Vanhemelrijk)

Also the two types of lifting equipment on the Pieter Schelte cannot be implemented on the NSCV. On the U-shaped front of this vessel the legs of the topside (part of an offshore platform above sea level on which equipment is installed) are grabbed and the topside is lifted as a single piece (Figure 4). The full deck of the NSCV does not allow this lifting principle. After the topside is removed the jacket (the topside's supporting structure) is lifted by tilting lift beams as shown in Figure 5. These beams do not have the required crane functionality and cannot be implemented on the NSCV, as will be discussed in the following section.

Figure 4: Pieter Schelte, topside lifting system (Setcorp)

Figure 5: Pieter Schelte, jacket lifting system (Deltamarin)

Crane functionality

One of HMC's requirements is the ability of the NSCV to dual lift (Figure 6). Compared to a single lift, a dual lift has the advantages that the load positioning accuracy is improved, the load rotation can be suppressed more effectively and the stresses in the load structure can be reduced. A disadvantage of dual lifting is the complex operational interface with respect to the high degree of communication and coordination required. If one of the cranes fails to suspend the load as intended, one of the cranes can be severely overloaded.

Another functional requirement is the operating area of the lifting equipment. It has to be possible to slew the cranes relative to the vessel and to carry out lifting operations between the cranes from the own deck of the NSCV and the sea. An important aspect is, that lifting between the cranes is not required for the very heavy lifts because these are lifted from a barge. Thus not from the own deck of the NSCV. This is a consequence of the limited space between the cranes and the maximum allowable deck load of approximately 12,000mt.

Therefore, it is an option to temporary install a third crane when very heavy loads have to be lifted. However, transportation, installation and commissioning is a complex, costly and time consuming process. Assessing the feasibility of this option is a Master Thesis itself and falls out of the scope of this Master Thesis. Therefore, the number of cranes is restricted to two.

Figure 6: Thialf dual lift (HMC)

Crane types

On both the Balder and Hermod (two of HMC's semi-subs) two cranes are installed with different lifting capacities. At the moment the cranes were purchased it was the intention to place them on two monohulls. Later it was decided to place them on semi-subs together with a second crane.

It is a logical choice to install two similar crane types with equal lifting capacity to meet the required lifting capacity of 16,000mt. Installing different crane types is unfavorable with respect to the availability of spare parts, engineering cost, vessel stability etc. Therefore, this Master Thesis investigates a concept of equal cranes with equal lifting capacities only.

Crane specifications

To keep sufficient space available on deck of the NSCV and to match the cranes with the columns of the semi-sub, the dimensions of the location where the crane is integrated with the vessel (footprint) are not allowed to be larger than those currently occupied by the cranes on the semi-subs of HMC.

Since the cranes have to be able to slew over reels and mission equipment on deck, the height of the boom hinge point is another important crane specification. The boom hinge point has to be placed at 30m above the deck of the NSCV.

Another requirement is the heights at which the cranes have to be able to lift. Lifting at large heights is an important requirement for the installation of certain platform components (e.g. flare booms and drilling towers). The maximum lifting height of the Thialf is 95m (above deck) and is considered as a minimum for the cranes on the NSCV.

The maximum lowering capacities do not lead to new requirements on the cranes. The lowering capacity leads only to a requirement on the available wire rope length. It is independent of the crane type.

Figure 7: Drilling tower removal (Berghuis)

Summary of research boundaries

The constraints and requirements of this Master Thesis form the basis for the crane design choice and are summarized in Table 1.

Table 1: Main boundaries

Boundary type	Restriction
Dual lifting capacity	≥ 16,000mt
Semi-sub dimensions	170 x 93m
Crane location	At bow of NSCV above its columns
Slewing behavior of cranes	Able to slew relative to the deck
Working area cranes	Lifting possible between the cranes above deck
Crane types and lifting capacity	Equal for both cranes
Lifting height	≥ 95m
Footprint dimensions	≤ 30m
Boom hinge point above deck	30m

3.0 OFFSHORE MARKET PROSPECTS AND TRENDS

The offshore market prospects are assessed to determine if new crane requirements can be expected in the next decades. The lifting market is substantially oversupplied, driven by the crane vessels with relatively low lifting capacities. However, for the crane vessels with high lifting capacities (≥5000mt) the demand exceeds the supply for offshore oil and gas related lifting operations (Figure 8). With a total lifting capacity of 16,000mt the NSCV will belong to this latter part of the heavy lift market. The focus in the following sections lies on the expected changes in the heavy lifting market, most importantly how these changes will influence the crane requirements. The assessed areas of application are deep-water lifting and lowering, platform removal and offshore wind farms.

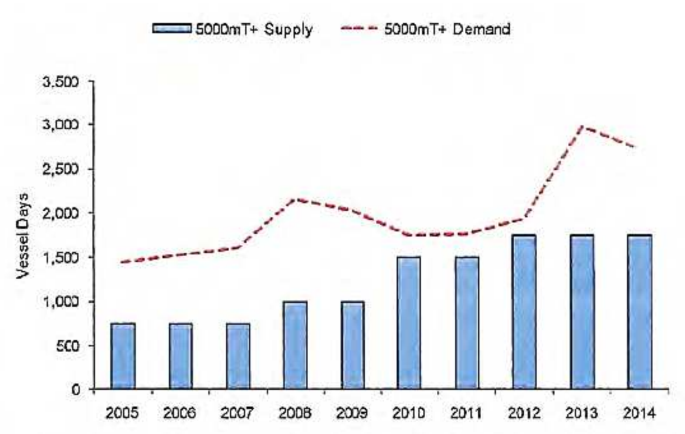


Figure 8: 5,000mt heavy lift supply and demand in vessel days (Infield 2010)

3.1 Deep-water lifting and lowering

At this moment, the supply significantly exceeds the demands for deep-water lifting and lowering. In the next decades the deep-water oil and gas production is expected to grow from 8 million barrels a day around the year 2012 to 10 million barrels a day in 2025 (MacKenzie). The number of projects related to this market is therefore expected to grow, possibly preferring other crane vessel requirements than found on current ones.

The most important crane component affected by deep-water lifting and lowering, is its wire rope. Steel wire rope can have free rotation under load, possibly resulting in damaged wire, or loss of the end termination. This especially occurs when the tension is removed and the rotation tries to unwind (Figure 9). Non-rotating wire ropes can have significant problems with increasing lengths. Free rotation effects of wire ropes can be suppressed by retaining the orientation of the hoisting block and keeping the wire ropes under tension. This can be achieved for instance by a Remotely Operated underwater Vehicle (ROV). However, the possibilities are limited. Therefore, the applied wire rope length for deep-water lifting and lowering has to be taken into consideration in the crane design.

Figure 9: Free rotation effect of wire ropes (HMC)

For very deep-water lifting operations, steel wire ropes with multi-fall lowering systems are limited in their application. As the depth increases the ratio of the weight of the wire rope to the Safe Working Load (SWL) becomes extreme. At 3,000m depth the weight of a 5” wire rope is about the same as its 170mt SWL. At a depth of 6,000m the SWL of the steel wire rope is even entirely used up by its own weight, leaving zero SWL (Rowe 2001).

The increasing wire rope weight with increasing length can be solved by applying synthetic ropes. Synthetic ropes are at least five times lighter than steel wire rope of an equal load rating and are either buoyant or close to neutral. Therefore, they have a small effect on its SWL. Synthetic ropes require less maintenance compared to steel wire ropes, which require frequent lubrication to maintain performance and prevent corrosion.

Synthetic rope usage does have a number of disadvantages. One of them is their lower resistance to bending fatigue than steel wire ropes, resulting in a shorter service life. Also most synthetic ropes degrade quicker than steel wire ropes because a significant amount of heat is generated with repetitive bending (e.g. in the case of active heave compensation). The durability and the service life of synthetic ropes are also questionable due to the low melting point of the materials, the little available track record and the potential problems related to stretch and creep. Current developments show possibilities to improve the service life of wire ropes.

Conventional drum winches are not suitable for synthetic ropes. The high inertia of the drum winches, the long rope lengths and the high tension may cause the wire ropes to become embedded in underlying layers. The maximum tension in the ropes decreases as the number of synthetic rope layers increases. Traction winches (Figure 10) provide a solution for this problem, but system design is critical, particularly the design of the grooves which grip the ropes to avoid slippage. An advantage is constant line pull, but the coordination is difficult for high speeds. The systems suitable for synthetic ropes are mechanically more complex. Therefore, the expected cost for synthetic rope usage are higher than that for steel wire ropes.

Figure 10: Synthetic rope on traction winch

Due to the perceived shortcomings of synthetic ropes there is a discrepancy in the factors of safety between steel wire ropes and synthetic ropes. Certification bodies recommend the use of various material factors to calculate the properties of synthetic ropes. The recommended factors of safety tend to be unnecessarily overcompensated. Research shows these safety factors can be reduced, making synthetic rope usage more attractive (Offshore-mag).

In recent years, synthetic ropes have been successfully used in both single-drum and traction winches and have replaced steel wire ropes in many applications such as heavy lift slings and deep-water mooring lines. In many cases, synthetic ropes will outlast and outperform wire rope. Unfortunately, synthetic ropes are (not yet) suitable to be used as hoisting wires in heavy lift cranes. Since it is also unknown how fast developments take place, no measures related to synthetic wire rope application are taken into consideration in the crane design.

3.2 Platform removal

In this section is assessed how the prospects in the platform removal market influence the crane requirements. The graph below shows the expected cease in production of UK fields in a given year. Forecasts have changed significantly over the three years by the changing field closure dates. The Cessation of Production (COP) predictions for 24 fields in 2011 were changed by five years or more against the figures given for the same fields in 2010 – 18 fields were extended and six shortened.

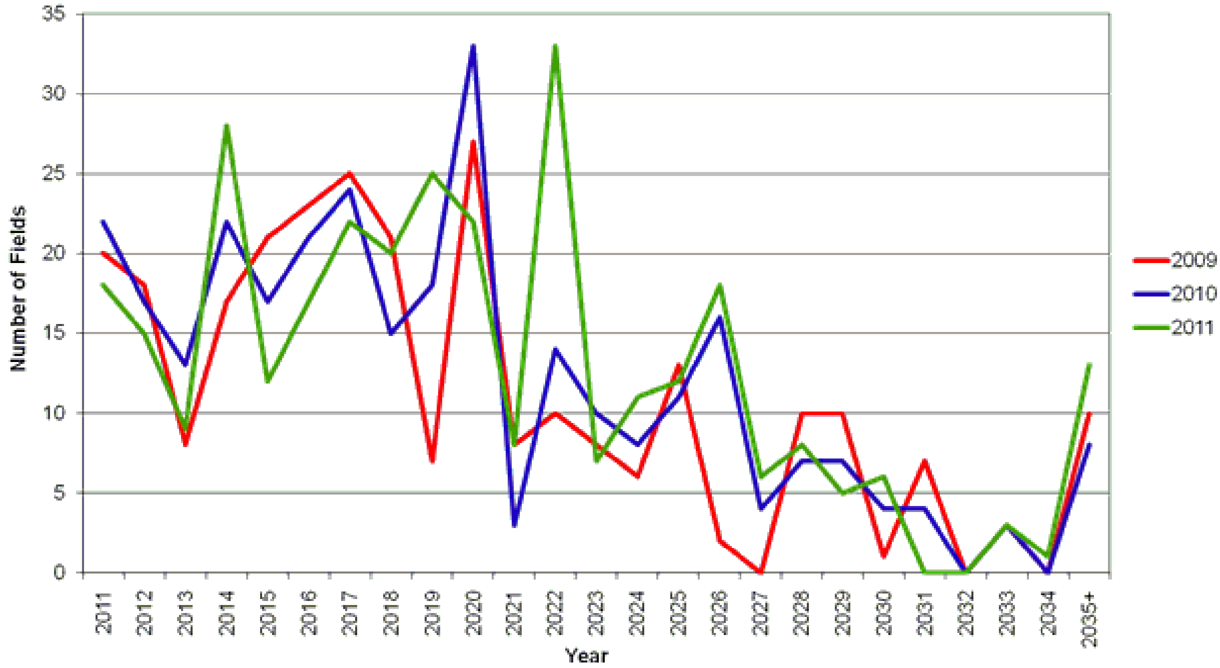


Figure 11: Forecasting the Cessation of Production (COP) of field removal (GovernmentUK)

Global research (Infield) shows a global demand of vessel days from 2005 – 2009 for platform removal of 19% and installation of 81%. The average number of removal projects between 2010 and 2014 is expected to increase to 26%.

The available options for platform removal are:

1. Removal as one piece in a single lift
2. Removal as original installation components in multiple lifts (reverse installation)
3. Removal as a combination of modules in multiple lifts

Option 1: Removing platform components (e.g. topsides and jackets) in a single lift is advantageous since the environmental risks are small because pipe cutting is reduced to a minimum. This is especially important for topside removal due to the presence of hydrocarbons. A disadvantage, but an interesting market for HMC, is the need for a crane vessel with large lifting capacity. Especially in the northern sector of the North Sea the platforms are too large/heavy to be removed in a single lift. Even the NSCV with a lifting capacity of 16,000mt will not be able to handle these lifts (Andresen).

Option 2: Another option is the removal in reverse order to installation. Before this can be started, components have to be surveyed (e.g. pipework, cabling, module structures) to determine the extent of the module preparation required prior to lifting. The structural integrity of the modules needs to be checked and if necessary, to be strengthened. The CoGs of the loads have to be determined, pad eyes to be (re)applied and lifting frames to be installed. Approximately the same amount of preparations is required as for single lift removal.

Figure 12: Jacket removal

Option 3: Lifting combined modules can be an effective option since fewer lifts are required compared to the reverse installation (in which modules are lifted individually). A disadvantage is that sequencing, surveying and the fabrication and lift point attachment as well as additional strengthening is required. Whether combined removal is advantageous compared to reverse installation depends on the module configuration and their weights (Manago).

Implementing single lifting capabilities of large topsides and jackets for the NSCV is not possible due to its closed deck. Instead of removing a section as a whole, it can be removed in modules. Only a few new vessel are developed for removal projects (e.g. Pieter Schelte). Therefore, heavy lift vessels as the NSCV vessels are expected to stay key players in the removal market for a long time.

For removal projects a large opening between the cranes is preferred for two reasons. The first reason is that the removed modules can be moved through the opening between the cranes, providing the possibility to place the modules on the own deck of the NSCV. The second reason is that for down-ending of removed jackets (changing the orientation from vertical to horizontal) a large opening between the cranes is required because otherwise the cranes would have to be slewed outwards to provide sufficient space between the boom tips.

How the platform removal market will develop is uncertain, but significant growth is expected in the next decades. Wishes for the NSCV related to platform removal are a large deck space and a large opening between the cranes.

3.3 Offshore wind farms

This section assesses how the prospects in the offshore wind energy market influence the crane choice. This market develops fast (Figure 13). In 2007 the European wind energy production was 119TWh (3% of total wind energy production). By 2030 the production is expected to reach 935TWh (50% of total wind energy production). The growth in wind energy production requires the construction of several large wind farms in Europe. Plans are formed to build farms with a capacity of 25 to 33GW (Decker). These farms require a very robust electrical transmission system with high availability and minimal maintenance. The power transmissions require High Voltage Alternating Current (HVAC) or High Voltage Direct Current (HVDC) substations.

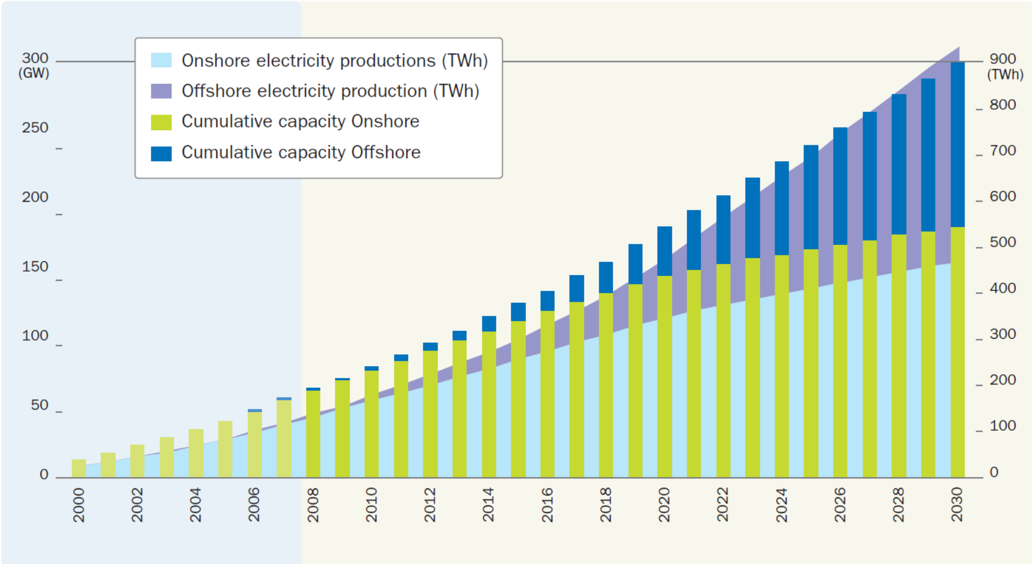


Figure 13: Electricity from wind up to 2030

For distances greater than 50 – 70km the capacity per unit length of submarine cables makes the use of AC cables impractical for transmitting large amount of power (Electricalreview 2010). DC transmission lines can be cost effective if the AC/DC and DC/AC conversion cost are less than the incremental losses of an AC transmission line. If this is the case more complex AC/DC and DC/AC converters are beneficial. The investment cost for this option are larger, but transmission losses are reduced.

Up to now offshore wind turbines are limited to shallow water (20 – 30m water depth). For HMC, the installation of substations is the only work related to the offshore wind farms. The wind turbines are usually installed by specialized crane vessels. HVAC and HVDC substations are placed offshore on symmetrical monopoles/bipoles and jackets. This placement method is expected to be used as the main substation support until 2020 (Decker). Innovative concepts, like far (>60km from shore) and deep (>60m water depth) offshore, will not significantly contribute before 2030 in Europe.

For the common substation installation method the topside and jacket weight (up to 9,000mt) and corresponding dimensions do not differ from the current platforms installed by HMC. Mentioned substation characteristics are for large wind farms and are not expected to increase in the next decade (Vågfelt 2011). The developments in de wind energy market do not provide additional requirements for the heavy lift cranes to be installed on the NSCV, but the installation of substations is a potential market for the NSCV.

Figure 14: Wind farm substation

3.4 Conclusion

In the deep-water oil and gas production and in the related offshore projects growth is expected in the next decades. The applied wire rope length for deep-water lifting and lowering has to be taken into consideration in the crane specifications. Since synthetic ropes are (not yet) suitable to be used as hoisting wires in heavy lift cranes, no measures related to synthetic wire rope application are taken into consideration in the crane design.

How the platform removal market will develop is uncertain, but growth is expected in the next decade up to 30% of all offshore projects. Related wishes for the NSCV are a large deck space and a large opening between the cranes.

Growth is also expected in the offshore wind farm market for the next decades. The expected characteristics of lifting operations related to these markets do not show changes indicating specific crane requirements for the NSCV that have to be taken into account.

4.0 AVAILABLE CRANES

From the moment heavy lifting went offshore, various types of heavy lift cranes have been installed on different vessel types (e.g. semi-subs, monohulls, barges, catamarans). Just as each vessel type has specific operability characteristics, each crane type has specific characteristics and specifications.

In this chapter the characteristics of commonly used offshore crane types are discussed. This inventory forms the reference set for the trade-off in Chapter 5 to find the most suitable crane that can be scaled up to meet the required lifting capacity of 8,000mt. The crane types are: A-frame cranes, mast cranes, slewing-mast cranes, sheerlegs, ringer multi-cranes, kingpost cranes, fixed boom cranes, knuckle boom cranes and telescopic boom cranes (Figure 15).

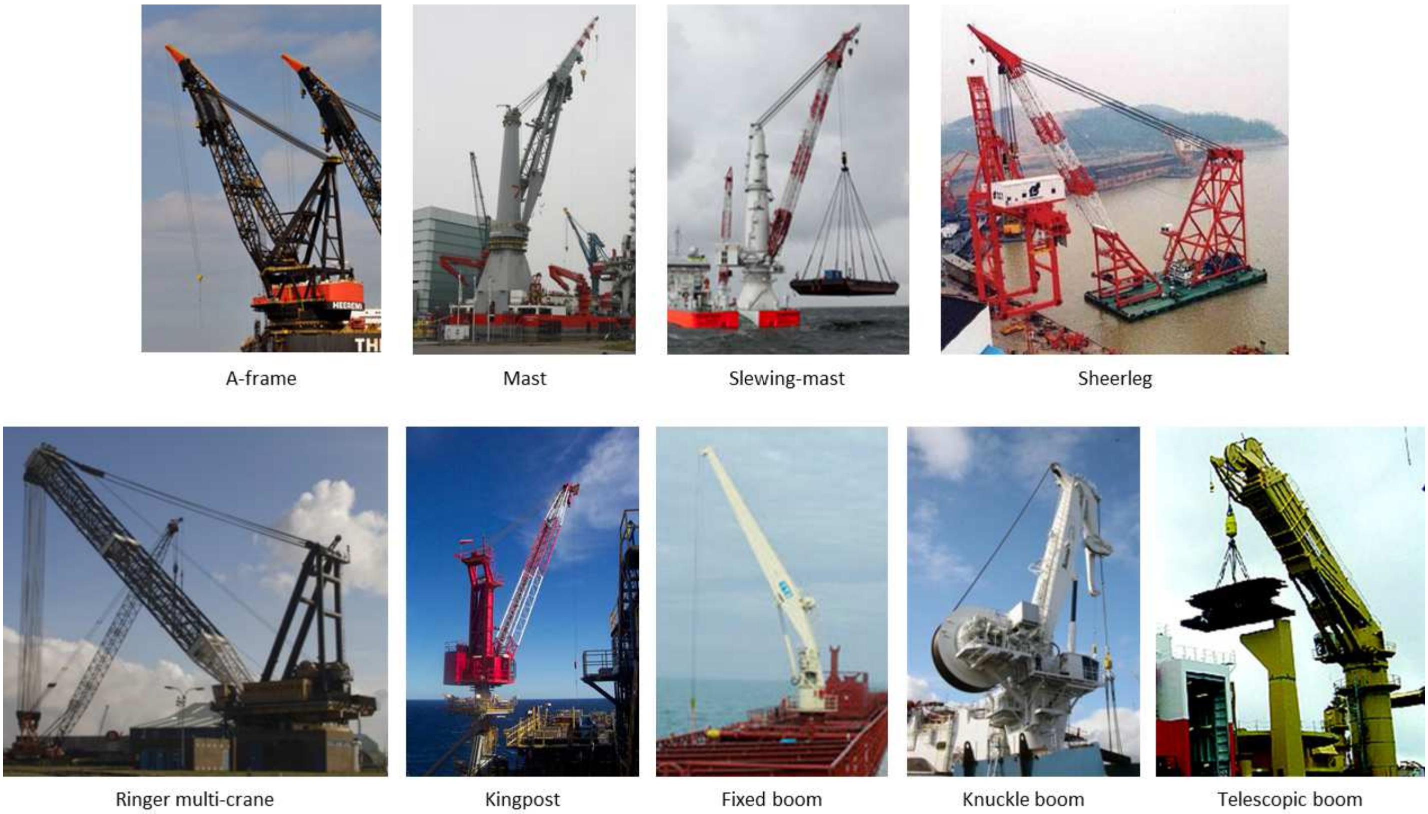


Figure 15: Available offshore crane types (NOV)

Some of the crane types shown in Figure 15 are not suitable for the NSCV since the demanded crane characteristics and specifications are not met:

1. Sheerlegs are unsuitable because they are unable to slew relative to the vessel;
2. Ringer multi-cranes have a large footprint, making it unfeasible to comply with the footprint restrictions;
3. Kingpost cranes and their typical way of installation makes them unsuitable for the NSCV;
4. Fixed boom, knuckle boom and telescopic boom cranes all have high shear forces acting in their booms because they are luffed relatively close to their boom hinge point.

These crane types will therefore be unfeasible to meet the required lifting capacity of 8,000mt (Appendix A). The characteristics of the remaining crane types (A-frame cranes, mast cranes and slewing-mast cranes) do not show initial restrictions to meet the demanded crane requirements and constraints and will be discussed in more detail in the following sections.

4.1 A-frame cranes

The terms pedestal, tub, or A-frame crane are all used for the same crane type. The terms pedestal and tub refer to the structure below the slew bearing of the crane. To avoid confusion the term A-frame crane is used in this report for the crane shown Figure 16. The typical A-frame shape is also found on other crane types (e.g. sheerlegs and ringer multi-cranes), but these crane types differ significantly from A-frame cranes and a clear distinction can be made.

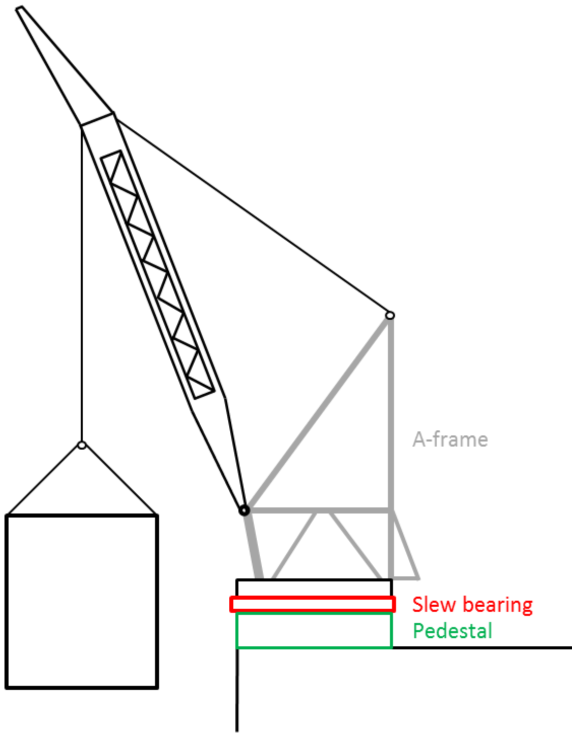


Figure 16: A-frame crane

The A-frame cranes consist of a boom and an A-frame installed on a fully revolving slew platform. The pedestal of the crane is integrated in the structure of the vessel. All crane components above the slew bearing are able to slew as a whole, relative to the pedestal of the crane. Because the boom and the A-frame are placed above the slew bearing, the overturning moment has to be transferred by the slew bearing of the crane. This requires a relatively large footprint.

When the boom is placed in the boom rest, the height of the crane is determined by its A-frame. This height is called the remaining height of a crane. The winches, drives and gear boxes are located above the slew bearing and require a significant amount of space. This configuration leads to extra slew bearing loading. The slewing range of an A-frame crane is not restricted because the power is provided by a slip ring system.

The presence of the counterweight compensates the overturning moment that is caused by boom weight, the blocks (depending which blocks are applied) and the wire ropes used for hoisting. The counterweight requires a large free deck space outside the footprint area of the crane. The presence of counterweight makes it possible to slew the crane without affecting vessel stability.

Lifting capacity

The heaviest payload ever lifted by a crane vessel (with A-frame cranes) was a dual lift of 12,150mt (Sabratha deck in Libya, Mediterranean Sea) by the Saipem 7000 in 2004. The Saipem 7000 is a similar crane vessel as the Thialf, put into service in 1988.

The three A-frame cranes with the largest maximum revolving lifting capacity of a single crane are:

1. Thialf SB and PS: 7,100mt (dual lift 14,200mt)
2. Saipem 7000 SB and PS: 7,000mt (dual lift 14,000mt)
3. Hermod SB: 5,000mt (dual lift 8,100mt)

The maximum lifting capacities of the A-frame cranes on the Thialf and Saipem 7000 are close to the required 8,000mt for the NSCV since only a lifting capacity increase of 13% is required. Because other crane types could be more optimal for the NSCV, the crane choice is certainly not a done deal.

Figure 17: Thialf

Figure 18: Saipem 7000

Figure 19: Hermod

4.2 Mast cranes

The typical component of a mast crane, as the name suggests, is the mast. Contrary to A-frame cranes, during crane slewing less components slew relative to the vessel since the mast is directly welded to the pedestal of the crane. The pedestal is integrated in the structure of the vessel.

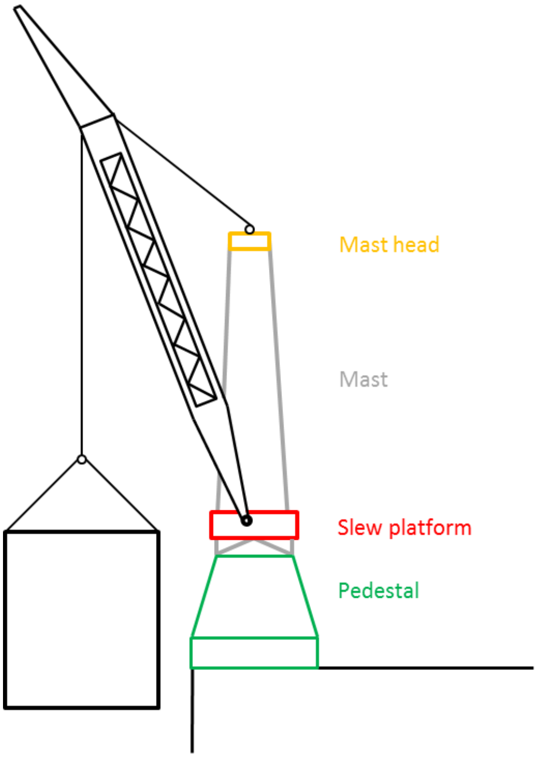


Figure 20: Mast crane

A mast crane consists of a boom whose hinge point is located on the slew platform. The slew platform is able to rotate around the fixed mast. At the top of the mast the mast head (orange part in Figure 20) is located. It is able to rotate relative to the mast. When the boom is slewed, the mast head follows this motion. In the pedestal the rotating winch frame is mounted (Figure 21). It also follows the motion of the mast head. The orientation of the winch frame relative to the mast head is controlled by electrical machines. Because for a mast crane the winch frame is located below the slew bearing, no space for these components above the slew bearing is required and the vertical loading on the slew bearing is reduced.

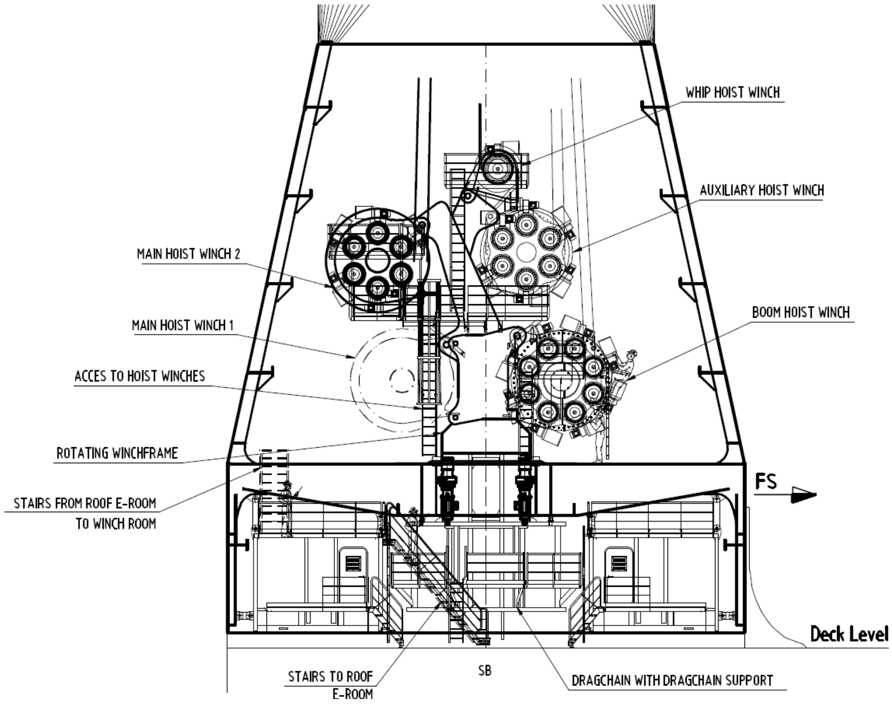


Figure 21: Pedestal with rotating winch frame (HMC)

In mast cranes the overturning moment is not transferred by its slew bearing, but by the mast. This causes relatively large radial forces in the slew bearing compared to A-frame cranes. Some mast cranes favour a certain slewing angle at maximum lifting capacity, not only due to ballasting limitations of the vessel, but also due to the fact that stiffening in the mast is only applied at a specific location. This stiffening is only most effective at a certain slewing angle.

The footprint of mast cranes is small compared to A-frame cranes because the overturning moment is not transferred by the mast and not by the slew bearing. Because the dimensions of the slew platform on the back of the mast crane are small and no counterweight is applied, the free space required on the deck outside the footprint of the crane is much smaller than for A-frame cranes.

Because the mast is integrated in the structure of the vessel, the remaining height of the crane is large (when the boom is placed in the boom rest). The relative movement between the vessel and the winch frame require the power and the control signals to the winches have to be transferred by a system allowing relative movement. On the Sapura 3000 the slewing range is not restricted, because wireless data transfer is applied. On the Aegir the slew range is restricted (to 280° both ways from the zero line) because conventional cabling is placed in a drag chain to guarantee reliable data transfer.

The absence of counterweight means that the overturning moment caused by the boom, blocks etc. is not compensated. Therefore, the effects on vessel stability are larger for mast cranes than for A-frame cranes.

Lifting capacity

The maximum lifting capacity of a mast crane is 5,000mt over stern of the Borealis mast crane (Figure 22). The maximum lifting capacity of this crane when slewing is 4,000mt. The Aegir mast crane has a maximum lifting capacity when slewing of 4,000mt. Its maximum lifting capacity over stern is not higher because no stiffening is applied in the mast and the tackles are only designed for a lifting capacity of 4,000mt.

The three mast cranes with the largest maximum lift capacity of a single mast crane are:

1. Seven Borealis: 5,000mt over stern and 4,000mt revolving
2. Aegir: 4,000mt revolving
3. Sapura 3000: 3,000mt revolving

The maximum lifting capacity of the mast crane on the Seven Borealis is 5,000mt and is far off the required 8,000mt for the NSCV since a lifting capacity increase of 60% is required. Therefore, probably the engineering effort and related risks are larger for mast cranes than for A-frame cranes.

Figure 22: Seven Borealis

Figure 23: Aegir

Figure 24: Sapura 3000

4.3 Slewing-mast cranes

Just as for mast cranes, the typical component for a slewing-mast crane (Figure 25) is the mast, although in this case the slew platform is not placed around the mast but the mast is placed on the slew platform. The mast is able to slew relative to the pedestal of the crane and the vessel the crane is installed on. Therefore, the name slewing-mast crane will be used in this report. Another name used for this crane type is an A-frame crane with a mast as A-frame.

Figure 25: Slewing-mast crane (Liebherr)

A slewing-mast crane consists of a mast and a boom whose hinge point is located on the slew platform. On the back of this slew platform the winches and drives are located. The mast, slew platform and boom are slewed as a whole relative to the pedestal of the crane. Therefore, during slewing the top of the mast follows the boom tip automatically and the mast head does not rotate relative to the mast.

The slewing-mast principle causes that the overturning moment is transferred by the slew bearing and not by the fixed mast, as is the case for mast cranes. This crane design causes small radial loading, but large axial loading in the slew bearing compared to mast cranes. Therefore, the footprint of a slewing-mast crane is relatively large compared to mast cranes. The remaining height of a slewing-mast crane is similar to that of a mast crane.

The slewing range of a slewing-mast crane is not restricted since power is supplied by slip rings and the cabling between the winches and crane driver cabin undergo the same slewing motion. Because the drives and winches are placed on the slew platform and not on a rotating winch frame in the pedestal of the crane, the CoG of a slewing-mast crane is relatively high compared to that of a mast crane. The tail swing of this crane type is large, but the weight on the back of the slew platform reduces the overturning moment acting on the slew bearing. The smaller ballast system requirements compared to mast cranes, but permanent vertical loading on the slew bearing is increased.

Lifting capacity

The maximum revolving lifting capacity of a slewing-mast crane is 2,000mt (Liebherr MTC 78000) and is only a quarter of the required 8,000mt for the NSCV. Therefore, the expected engineering effort and development risk for this crane type is large.

4.4 Crane type overview

In previous sections a number of crane- and vessel characteristics are discussed for the three crane types: footprint, slewing range, tail swing, remaining height, ballast system requirements and the maximum lifting capacity (Table 2).

Table 2: Potential crane types

Crane type	**Footprint**	**Slewing range**	**Tail swing**	**Remaining height**	**Ballast system requirements**	**Max. lifting capacity** [mt]
A-frame	Large	∞	Large	Medium	Small	7,100
Mast	Medium	∞ or 2*280̊	Small	Large	High	5,000
Slewing-mast	Medium	∞	Large	Large	Medium/High	2,000

The footprints of mast cranes are relatively small compared to A-frame cranes. When a slewing-mast crane is scaled to 8,000mt its footprint is expected to be equal to that of an A-frame crane since the overturning moment is transferred by the slew bearing. Of all crane types the slewing range of all discussed crane types can be unrestricted. However, for some mast cranes the slewing range is restricted. For mast cranes the tail swing is relatively small compared to the other crane types due to the absence of counterweight and since the drives and winches are placed inside the pedestal. The remaining height of mast cranes and slewing-mast cranes is larger than that of A-frame cranes. The ballast system requirements are small for A-frame cranes, whereas the requirements for mast cranes and slewing-mast cranes are higher.

A-frame cranes seem to be most suitable to meet the required lifting capacity of 8,000mt. For mast cranes more engineering is expected and for slewing-mast cranes even more because their maximum available lifting capacities are far off the required 8,000mt. Scaling a slewing-mast crane to 8,000mt results in such disadvantageous crane characteristics that this crane type is not included in the crane types trade-off as a potential crane type for the NSCV. However, this crane type is included in the MCA to validate the strength of the criteria.

In the previous sections the most obvious crane- and vessel characteristics are discussed in a nutshell, necessary to show the potential of the cranes for the NSCV and to understand their designs. In the next chapter a number of crane characteristics are added to obtain a complete overview of criteria, needed for the crane types trade-off.

5.0 CRANE TYPES TRADE-OFF

The conclusion in Section 4.4 was that two crane types show the potential to be installed on the NSCV: A-frame cranes and mast cranes. In this chapter firstly criteria for the crane types trade-off are discussed. Secondly these criteria are weighted in a Multi-Criteria Analysis (MCA) to determine the most promising crane type for the NSCV.

5.1 Crane characteristics

The criteria are related to the most important crane characteristics. Figure 26 shows a general crane lay-out with the boom of the crane above sea. This crane orientation is the only one feasible because only then the maximum dual lifting capacity (16,000mt) of the NSCV can be used. Any other crane orientation would require the lifting of 16,000mt from the deck of the NSCV, and this would exceed the maximum allowable deck load. A load has to be lifted from a barge, moored alongside the bow of the semi-sub. Also the dimensions of very heavy loads generally not allow a load to be suspended between the two cranes since the distance between the cranes is insufficient (Section 2.0).

When lifting a load several crane characteristics are important (Figure 26). Depending on how the position of a vessel is maintained (anchored or by dynamic positioning) minimum clearances between both the load and the boom, and between the load and the vessel have to be maintained (Table 3). The boom outreach from the centerline of the crane, the lifting height of the main block, the height of the boom hinge point, above deck and the distance between the boom hinge point and the edge of the vessel are important crane characteristics to be considered in the trade-off.

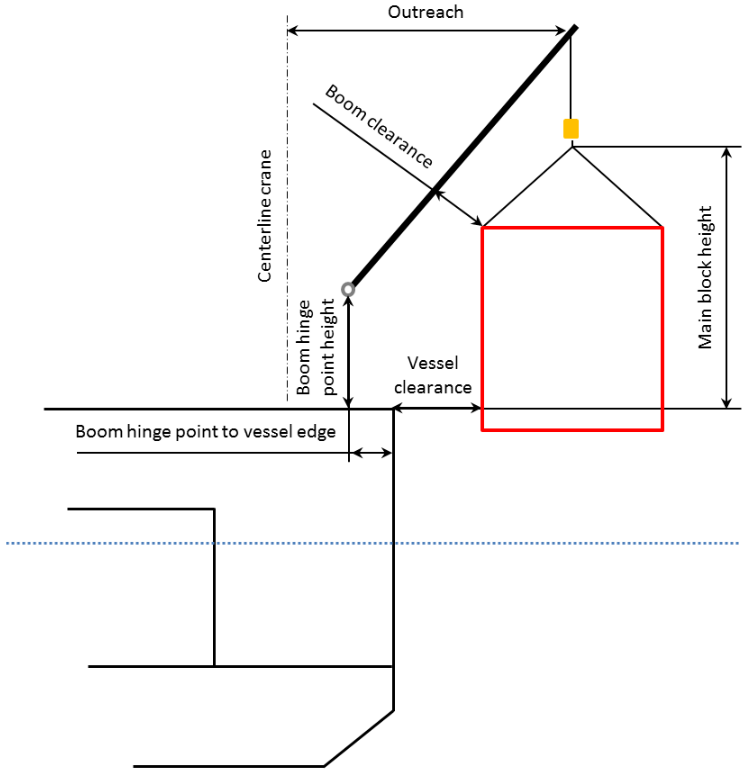


Figure 26: Crane characteristics

Table 3: Minimum clearances

	Crane Driver cabin	**Crane** Other	**Hull** Above waterline	**Hull** Below waterline
Lift off (from barge) [m]	5.0	3.0	3.0	n/a
Installation anchored [m]	5.0	3.0	3.0	8.0
Installation dynamic positioning [m]	5.0[1)]/8.0[2)]	3.0[1)]/8.0[2)]	3.0[1)]/8.0[2)]	10.0

1) Clearance of vessel to lifted object
2) Clearance of vessel to fixed structure

The crane design and the location of the crane on the semi-sub is a compromise of lifting very heavy loads and lifting loads with large dimensions. Lifting between the opening of the cranes is mainly determined by the distance between the centerlines of the cranes. An optimal configuration is found of the following crane characteristics: footprint, boom hinge point location and minimum outreach. Furthermore, it is important to keep in mind that stability of the semi-sub, the usable deck space, the ability to lift loads between the cranes and to lift loads close to the centerlines of the cranes are not compromised.

Not only the crane characteristics view as in Figure 26 is important, but also the distance between the cranes. For light loads with large dimensions a large distance between the cranes is preferred, while for heavy loads with small dimensions a small distance is preferred. Data of past lifting operations carried out by the Thialf shows that its cranes only lifted a few times close to their maximum lifting capacities (Appendix B). Therefore, in the crane trade-off a large distance between the cranes is considered more optimal than a small distance between the cranes.

5.2 Criteria

In this section the relevant criteria and their weight factors for the MCA are argued by an analysis of the consequences of the crane design by changing specific criteria. A high weight factor means the criteria is important in the crane trade-off, whereas a low weight factor is less important. Also the importance of criteria for A-frame cranes and mast cranes is assessed. At the end of each section, the typical characteristics for A-frame cranes (with a roller slew bearing and a bogie slew bearing) and mast cranes will be discussed. The incorporated criteria are those briefly discussed in Section 4.4 and other criteria, being important design parameters of the NSCV:

1. Lifting capacity up to 8,000mt
2. Lifting capacity >8,000mt
3. Footprint
4. Boom hinge point location and load curve
5. Tail swing
6. Slewing range
7. Air draft
8. Vessel stability
9. Load handling
10. Deep-water lifting and lowering
11. Maintenance and environmental sensitivity
12. Design maturity and reliability
13. Operating cost
14. Cost cranes

The cranes for the NSCV have not yet been designed and will be specifically designed for their application. Therefore, crane characteristics such as slewing, luffing and hoisting speeds are insensitive to the choices of the basic crane design, and cannot be used for crane type comparison. These criteria, just as many others, can be adjusted to the wishes of the customer and are independent of the crane type. This criteria discussed in the following sections include data the cranes on HMC’s vessels and of two 8,000mt crane proposals. The two proposals are a mast crane design by Huisman and an A-frame crane design by Amclyde. With this data the two crane types are compared.

5.2.1 Lifting capacity up to 8,000mt

Scale enlarging and upgrading of the cranes is required to increase their lifting capacities to the required 8,000mt. If the current maximum available lifting capacity for a certain crane type is far off the 8,000mt, it is likely the engineering effort and development risks for a crane upgrade will be high. When increasing the lifting capacities of mast- and A-frame cranes the components requiring the largest engineering effort, or even restricting the possibilities of scale enlargement are called the critical components of the cranes. The critical components of the two crane types are assessed in this section.

Less critical crane components are the boom, the wire ropes and the sheaves. The boom structure can be scaled relatively simple. For the wire ropes and sheaves the applied number can be enlarged. Critical components are found between moving pars. However, not between all moving parts when sufficient space for scale enlargement is available (e.g the boom hinge point) and no secondary effects occur. On the other hand, increasing the load transfer area in the slew bearing, required to increase a crane's lifting capacity up to 8,000mt, is critical. Because the diameter of the crane's footprint should not be larger than 30m (as discussed in the problem definition).

So far, A-frame cranes are seen as a single crane type. In the rest of the report A-frame types will be distinguished by their slew bearing type: the bogie slew bearing (Figure 27 and Figure 28) implemented on the Balder and the Hermod and the roller slew bearing (Figure 30 and Figure 30) implemented on the Thialf. This distinction is made because these two slew bearing types affect the scale enlargement possibilities of the cranes and other trade-off criteria (e.g. boom hinge point location and operating cost).

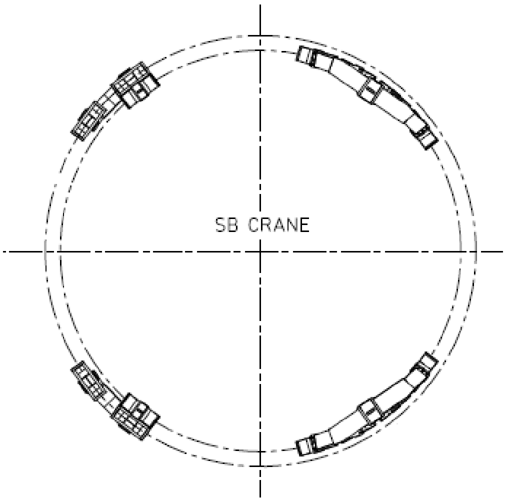


Figure 27: Bogie set arrangement, boom side on the right, tail side on the left (HMC)

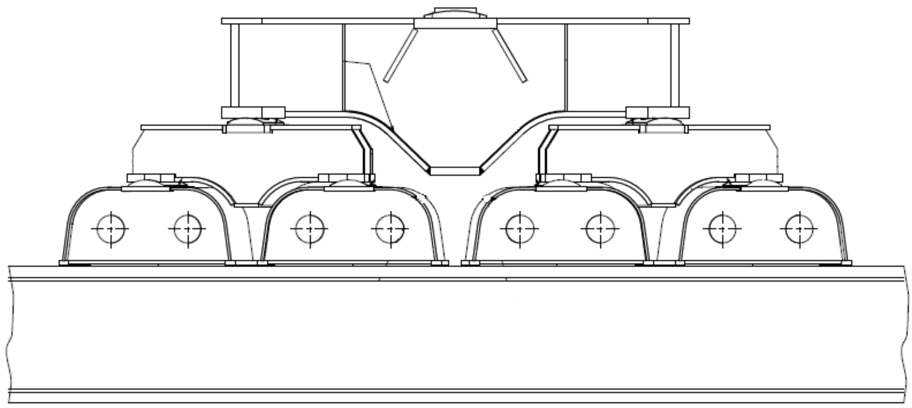


Figure 28: Bogie set (HMC)

For *A-frame cranes with a bogie slew bearing* the critical component is the slew bearing when its maximum lifting capacity of 4,500mt (revolving) has to be enlarged to the required 8,000mt. This slew bearing type is critical for upscaling since the available space for additional bogie wheels is limited by the diameter of the rail to keep the footprint dimensions of the crane acceptable.

Figure 27 shows the circumference of the top rail is only filled half with bogie wheels. It provides the possibility to double the number of bogies (keeping in mind that the top of a bogie set has to be located under a boom hinge point for optimal load transfer in the crane design). This seems sufficient to increase the lifting capacity of the crane from 4,500mt to 8,000mt since the load transfer area is hereby also doubled.

By filling the total circumference with bogies it should be kept in mind that also the total height of the whole bogie-set increases. This is not critical since the height of the boom hinge point has to be enlarged compared to that of a 4,500mt crane. Also the number of bogies placed under the rail, required to handle the overturning moment, needs to be enlarged when the same amount of counter weight is applied. This does not cause any scale enlargement issues since sufficient space available is on this rail. Little development risks and low engineering effort are the result.

The maximum lifting capacity of *A-frame cranes with a roller slew bearing* is currently 7,100mt (revolving). Figure 29 shows the possibility of adding top roller rings to increase the lifting capacity up to 8,000mt. Adding roller rings is also possible at the bottom rings, which handle the overturning moment of the crane.

Figure 29: Top view slew bearing rollers Thialf

Figure 30: Outer slew bearing rollers Thialf

Mast cranes have the mast as additional critical component compared to A-frame cranes. This crane design causes large radial forces perpendicular to the mast (Section 4.2). These forces are transferred from the boom hinge point, via the slew platform and the slew bearing to the mast, resulting in high slew bearing load.

The implemented slew bearing types in mast cranes have changed over time when the maximum lifting capacity and the associated mast diameter increased. For mast cranes with a lifting capacity up to 800st, ball bearings were used. When the lifting capacity went up to 3,000st this bearing type could no longer transfer the increased bearing loading and roller bearings were applied. Once the mast dimensions and loading on the slew bearing increased to the currently available maximum lifting capacity of 5,000mt the radial load could no longer be transferred by roller bearings and were replaced by plain bearings. The axial bearing loading could still be transferred by roller bearings (Figure 31). To determine if upscaling of a mast crane's slew bearing to 8,000mt becomes critical, the slew bearing design is assessed.

The plain bearing surfaces of the Aegir's mast crane consist of multiple Orkot bearing pads shown in Figure 32 (Orkot-Marine-Bearings). These pads are made of fiber reinforced plastic with a top layer of Teflon, to reduce the friction. The Orkot pads are greased to increase the allowable dynamic contact pressure.

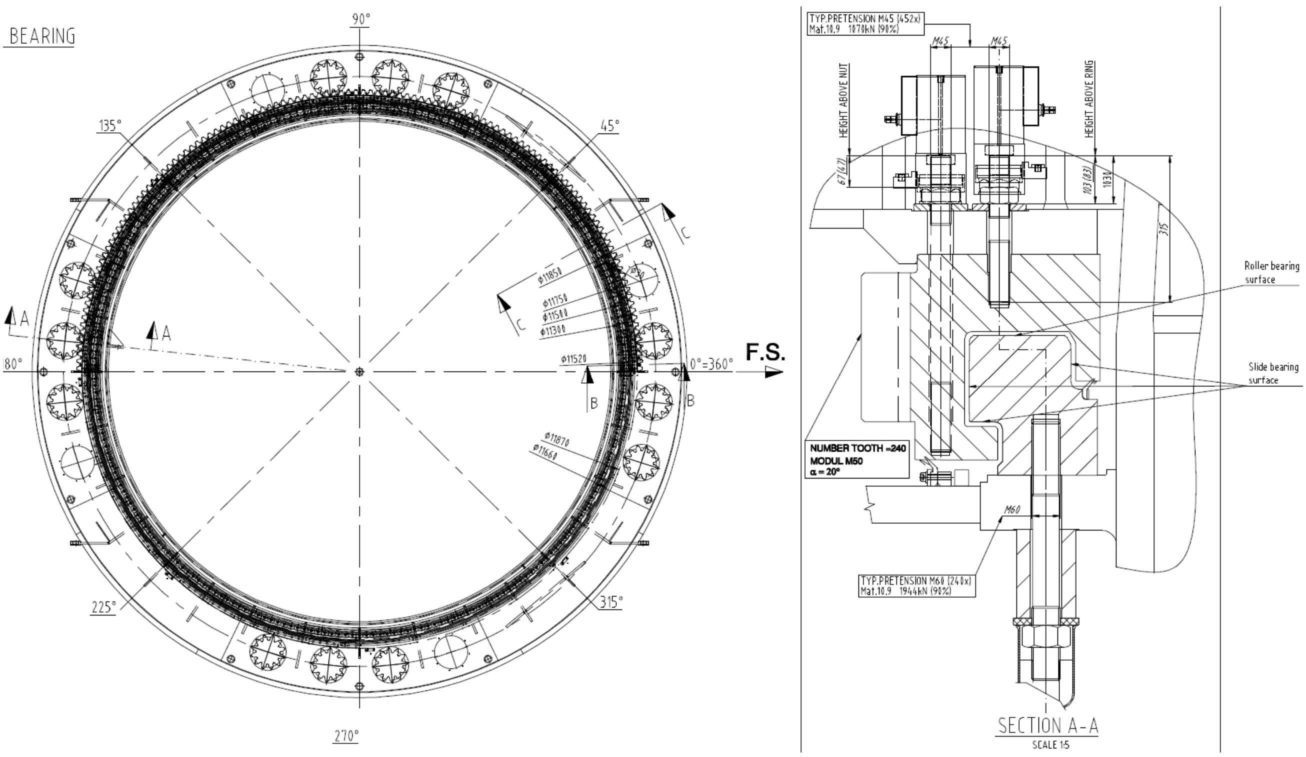


Figure 31: Top view of the Aegir's slew bearing (left) and the cross sectional area of the bearing (right) (HMC)

The roller bearing transfers axial bearing loading in the slew bearing of Aegir's mast crane and consists out of multiple cage sections. These cage sections prevent the rollers from making contact with each other, each containing 12 rollers (Figure 32). In smaller bearings, the races and cages are manufactured of one piece of material. This is not feasible for the bearing dimensions (diameter approximately 9m) of a 5,000mt mast crane considering the manufacturability, maintenance and a possible replacement of bearing components. The rollers travel between race plates made of Hardox 550. These plates are applied to withstand the high local loading of the rollers and are wear resistant due to its high toughness, good bendability and weldability.

Figure 32: Slew bearing jacked up, showing the Orkot pads, race plates and cage sections (HMC)

Figure 33: Aegir's slew bearing (HMC)

A critical issue related to the slew bearing is the deformation of the whole bearing caused by mast loading. Once the deformation is such that the clearance between the inner and outer ring of the slew bearing is reduced to zero, the stiffness of the outer ring contributes in maintaining the circular bearing shape.

The development of bearings over time indicates that increasing the lifting capacity of a mast crane up to 8,000mt is expected to cause major design challenges. These are related to stick-slip, teeth matching and reliability. Stick-slip is caused by plain bearing surfaces alternating between sticking to each other and sliding over each other, with a corresponding change in friction force. Transferring radial loading in the slew bearing by a plain bearing instead of a roller bearing causes a significant stick-slip increase. For a slow-speed plain bearing the stick-slip is larger than for a roller bearing.

Teeth matching, called evolvente, between the teeth of the bearing's outer ring and the teeth on the gear will cause design challenges when increasing the bearing diameter from 9m for a 5,000mt mast crane to 14m for an 8,000mt crane. This is caused, when the diameter of the bearing is enlarged, in a radial play increase between the inner and outer ring from 2 to 3mm. Radial play is caused by the component tolerances which are linearly related with the bearing diameter and the production tool accuracy. A larger radial play causes teeth matching issues.

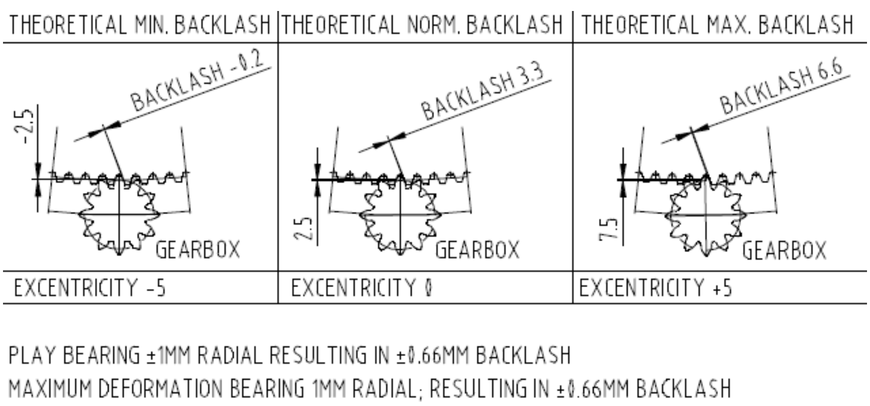


Figure 34: Backlash slew bearing Aegir (HMC)

Bearing diameter enlargement also causes design challenges related to backlash (Figure 34), which is the amount of clearance between mated gear teeth, when both gears are at nominal center distances. Intolerable gear performance (Beek) in a mast crane can be introduced such as sudden impulse or shock load, vibrations and noise, excessive heat, and other wear mechanisms such as a form of surface fatigue (pitting) or as adhesive wear (scuffing).

The expected service life of the slew bearing on the mast crane of the Aegir is 20 years. By increasing the mast diameter and keeping the slewing behavior similar, from a geometrical point of view the distance passed between the bearing surfaces is increased by ±30%. This decreases the service life of the gears, slew bearing teeth but also of the plain bearing surfaces. Only earlier replacement of some components and increasing maintenance cost can be expected. Upscaling will not become critical when enlarging its lifting capacity up to 8,000mt, but testing is required to provide accurate service life and maintenance predictions.

To keep the (local) stresses in the slew bearing and mast deformations at acceptable levels, the material thickness of the Aegir's mast is 80mm (Figure 35). The deformation and mast design has to ensure that the desired service life of the slew bearing is obtained. Increasing mast crane loading from 4,000mt up to 8,000mt requires the mast diameter to be enlarged from 9 to 14m. This could compromise the financial feasibility and the manufacturability of the mast in the slew bearing area and of the slew bearing itself. The reason is the required material dimensions have to be increased significantly to obtain the required slew bearing service life. The small amount of stiffening in the slew bearing area indicates scale enlarging is possible, but engineering effort and related risks are significant.

Figure 35: Ring sections of mast Aegir (HMC)

Increase of radial play is also expected to reduce the slew bearing service life when local stresses in the slew system increase. The contact pressures in the slew bearing of an 8,000mt mast crane are expected to be equal or less than those in the slew bearing of the 4,000mt mast crane on the Aegir. This is likely since the load transfer area of the slew bearing increases with the bearing diameter. If the design is changed such that the contact pressures in the slew bearing remain unchanged, no wear and fatigue increase is expected. Bearing service life will decrease when enlarging its diameter.

In summary, for both A-frame crane types scale enlargement effects are expected when enlarging the lifting capacity up to 8,000mt. Of a mast crane the bearing service life is expected decrease by increasing its mast diameter. This is not the case for A-frame cranes since their slew bearing diameter remains constant. More detailed research is required towards backlash and mast deformation in the slew bearing area. Increasing the lifting capacity of mast cranes will cause more design challenges than for A-frame cranes.

Increasing the lifting capacity of the cranes to the demanded 8,000mt is essential. Together with the expected engineering efforts for some crane types, the weight factor of these criteria in the MCA should be high.

5.2.2 Lifting capacity >8,000mt

Future market demands might require to further enlarge the lifting capacity of the cranes, beyond the 8,000mt. This can be realized without major reconfigurations to the crane. Three options are currently available on *A-frame cranes*. Increasing the amount of counterweight on the crane, the tie-back mode where the boom is tied-back to the deck (Figure 36) and the tie-back mode where the counterweight is tied-back to the deck.

Figure 36: Boom tie-back mode

Figure 37: Counterweight tie-back mode

Applying these modes is advantageous since the maximum lifting capacity of the crane is increased, or a certain load can be lifted at a larger outreach. By increasing the amount of counterweight on the crane (1,365mt on the SB crane of the Hermod, Table 4) the crane is still capable to slew. However, the service life of the bearing is decreased by the permanent increased loading of the slew bearing and the pedestal. In case the boom is put in tie-back mode, loading on all crane components is reduced. However, the crane is no longer capable to slew. For the counterweight tie-back mode, not on all crane components the loading is reduced. Only on the slew bearing and the pedestal. Therefore, the boom and A-frame still have to be designed for the maximum load moment. An advantage of this tie-back mode is that it can easily be applied at multiple slew angles and that it can be applied relatively quick.

Table 4: Temporary lifting capacity enlargement of A-frame cranes

	Balder PS Bogie slew bearing	**Balder SB** Bogie slew bearing	**Hermod PS** Bogie slew bearing	**Hermod SB** Bogie slew bearing	**Saipem 7000** Roller slew bearing
Lifting capacity enlargement mode	Boom tie-back	Boom tie-back	Boom tie-back	Additional counterweight	Counterweight tie-back
Original lifting capacity [mt]	2,200	3,300	3,000	5,000	7,000
Enlarged lifting capacity [mt]	3,000	4,000	4,000	-	-
Original outreach [m]	26 - 27.5	24 – 33.5	26 – 30.5	24 - 29	40
Enlarged outreach (if applicable) [m]	26 - 33.5	24 – 37.5	26 - 39	24 - 32	41

When the boom is tied-back to the deck, the largest maximum lifting capacity increase of 33%, in combination with an outreach enlargement of 28%, is obtained on the Hermod SB crane (Table 4). Load curve improvement by the counterweight tie-back mode is generally less effective. For the Saipem 7000 the maximum lifting capacity is not enlarged, but only the outreach by 1% at maximum lifting capacity. At other parts of the load curve this tie-back mode is more beneficial: the outreach at a lifting capacity of 6,000mt is enlarged by 11% (from 45 to 50m).

Applying the boom tie-back mode is a time consuming process. On the Balder, three days are scheduled for the reconfiguration of the crane. The first step of applying the tie-back mode consists of placing a temporary support between the boom and the A-frame (Figure 38). Then a part of the sheave nest is lowered to the deck, secured with wire ropes, and then the wire ropes between the pad eyes on deck and the boom are tensioned. At last, the temporary boom support is removed so that the boom is fully suspended in its tie-back mode.

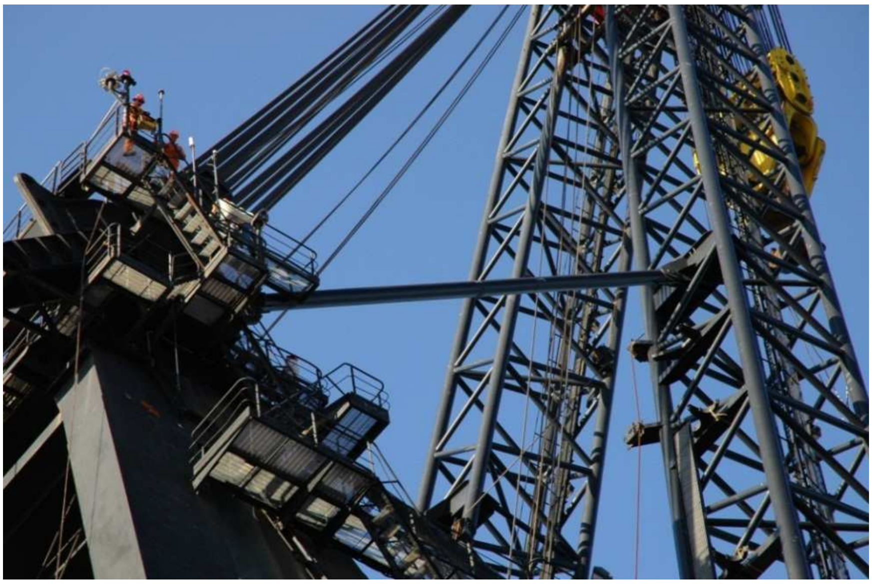


Figure 38: Temporary boom support (HMC)

So far, all discussed measures to improve the load curves of the cranes are for A-frame cranes. Assessing the possibilities for putting the boom of a *mast crane* in tie-back mode shows complications due to the relatively large height of the mast compared to that of an A-frame. It is likely the mast would be in the path of the tie-back ropes when the boom is not fully luffed. This can be solved by placing the tie-back ropes along the left and right side of the mast, but then the offlead angle of the wire ropes could become critical. Also the booms have to be slewed outwards to accommodate the connection of the wire ropes to the deck.

To reduce the loads acting on the slew bearing and pedestal of a mast crane, tying the slew platform to the deck of the vessel is an option. Also applying temporary counterweight is a possibility, but will have the same disadvantages as mentioned for A-frame cranes. The feasibility of these concepts needs further research

To conclude, it is more complicated to temporary increase the lifting capacity of mast cranes beyond the 8,000mt than for A-frame cranes. Because on both the Balder and Hermod methods are implemented to enlarge the lifting capacities of the cranes it is conceivable that lifting capacity enlargement is also required on the NSCV in the future. The weight factor of this criteria in the MCA should be medium.

5.2.3 Footprint

The footprint of a crane is the mounting interface between the crane and the vessel (Figure 39 and Figure 40). The important footprint characteristics are the dimensions and the shape. Because the NSCV will be used as a working island the less deck space the cranes take, the more useful deck space remains available to place equipment and modules on.

Figure 39: Circular footprint of an A-frame crane with roller slew bearing (HMC)

All *A-frame cranes* (roller slew bearing and bogie slew bearing) of HMC are placed on semi-subs and have a circular footprint. Because this footprint shape does not meet the rectangular column shape of a semi-sub above which the crane is placed (Figure 1), the integratability of the crane with the semi-sub is not optimal. A shape transition in the crane is required to match its footprint with the semi-sub's column, or the structural requirements in the semi-sub below the crane are larger to transfer the loads from the circular footprint to the rectangular column.

Mast cranes have generally a rectangular footprint since this is incorporated in the design to match its footprint to the shape of a vessel. The shape transition from the circular slew bearing to the square footprint is realized between the slew platform and the deck. Therefore, the higher the slew platform is located above the deck, the more space for this shape transition is available.

The slew platform is located just below the boom hinge point for a mast crane (Figure 39). For A-frame cranes the boom hinge point is located much higher than the slew bearing (Figure 40). This leaves less space for the shape transition, assuming the boom hinge points are at equal height above deck, than for mast cranes. Therefore, the integratability of mast cranes is generally better than that of A-frame cranes.

Figure 40: Square footprint of a mast crane (HMC)

Besides the footprint shape also its dimensions are important. Mast cranes have generally smaller dimensions than A-frame cranes because the overturning moment for a mast crane is transferred through the fixed mast and not through the slew bearing as is the case for A-frame cranes. The transfer of the overturning moment through the slew bearing (roller and bogie) requires a larger load transfer area than for the fixed mast. The relatively large radial forces in the slew bearing of a mast crane do not require a slew bearing as large as that of A-frame cranes. The smaller footprint of mast cranes is considered beneficial since more deck space remains available than for A-frame cranes.

Table 5 presents the maximum lifting capacities of A-frame cranes and mast cranes with corresponding footprint characteristics.

Table 5: Footprint cranes

	A-frame crane Roller slew bearing Thialf	**A-frame crane** Bogie slew bearing Hermod	**Mast crane** Seven Borealis
Lifting capacity revolving [mt]	7,100	4,500	5,000
Footprint dimensions [m]	Ø 28.4	Ø 29.0	16.8 x 16.8

As discussed in Section 2.0 the maximum footprint dimension for an 8,000mt crane is a diameter of 28m. The footprint dimensions for an 8,000mt mast crane are 22.0m, when the material thickness and the material quality are assumed not to change (Table 6). For A-frame cranes it is expected that the diameter of the footprint can be maintained at the current footprint diameter (28.0m) as discussed in the lifting capacity up to 8,000mt section.

The cranes will be placed above the columns of the NSCV and as close to the edge of the semi-sub as possible. Therefore, a smaller footprint will enlarge the distance between the pedestals of the cranes and the useful deck space. Table 6 shows that the occupied deck space for 8,000mt A-frame cranes is 21% larger than for mast cranes. To put the footprint area of the cranes into perspective with the total deck area, the deck area of the Thialf (9,290m^2) is taken as reference.

Table 6: Footprint 8,000mt cranes

	A-frame crane	Mast crane
Footprint shape	Circular	Square
Footprint dimensions [m]	Ø 28.0	22.0 x 22.0
Footprint area [m^2]	616	484
Distance between pedestals [m]	37	49

The footprint shape and dimensions determine the distance between the centerlines of the cranes (Figure 41). As discussed in Section 5.1, a large distance between the cranes is considered more optimal than a small distance between the cranes. The centerline of a mast crane can be placed closer to the corner of the semi-sub than of an A-frame crane, causing a larger distance between the cranes. Therefore, a mast crane is placed closer the bow to the vessel and the outreach of the crane is relatively small. A small outreach leaves a larger lifting capacity and available lifting height than for a large outreach.

How the effect described in previous paragraph affects the required outreach of the cranes is assessed with a lift for which the location of the main blocks of the crane have to be positioned as shown in Figure 41 (a typical topside and jacket lift, Appendix B) and the pedestal characteristics are as in Table 6. In this case, the outreach of a mast crane is 42.3m and of an A-frame crane 44.6m. Thus the larger distance between the centerlines of the cranes is compensated by their location closer to the bow of the vessel.

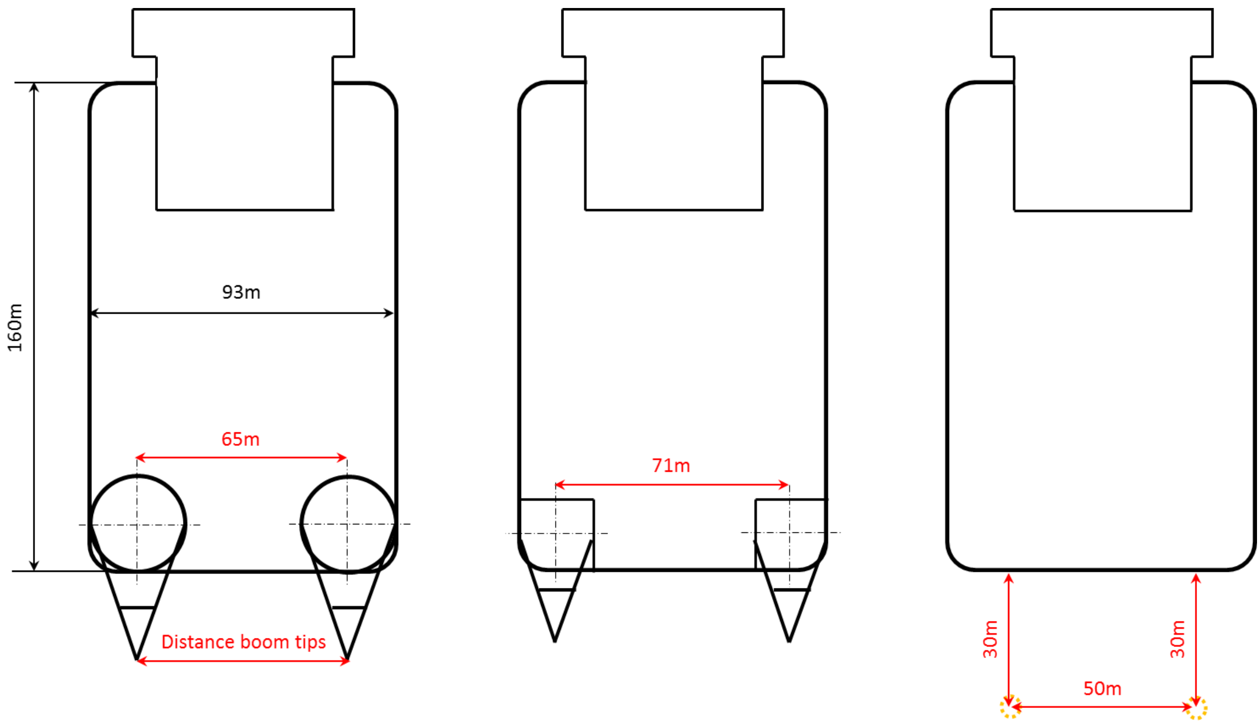


Figure 41: Locations crane centerlines of A-frame cranes (left) and of mast cranes (right)

To conclude, the integratability of mast cranes is better than A-frame cranes because more space is available for the shape transition. Also the useful deck space is larger for mast cranes than for A-frame cranes because mast cranes have a smaller footprint. Mast cranes are located closer to the bow of the vessel, reducing the required outreach to suspend a load at a certain distance from the bow of the NSCV. A smaller outreach means a larger available lifting capacity and lifting height. The weight factor of this criteria in the MCA should be high.

5.2.4 Boom hinge point location and loadcurve

Another crane characteristic is the location of the boom hinge point. This location makes sure optimal load transfer in the crane is guaranteed and depends on the slew bearing type. For an A-frame crane with a bogie slew bearing the tops of the bogie sets are located under the boom hinge points. A-frame cranes with a roller slew bearing have a larger distance between the centerline of the crane and the boom hinge point. The distance for mast cranes is small to prevent an additional moment in the slew bearing at maximum vertical loading on the boom hinge point.

The boom hinge point location theoretically determines the distance between a load and the crane at small lifting heights (Figure 42). However, other crane components fill the gap between the pedestal of the crane and its boom. Therefore, the location of the boom hinge point does not have effect on the load dimensions that can be lifted and is not included in the MCA.

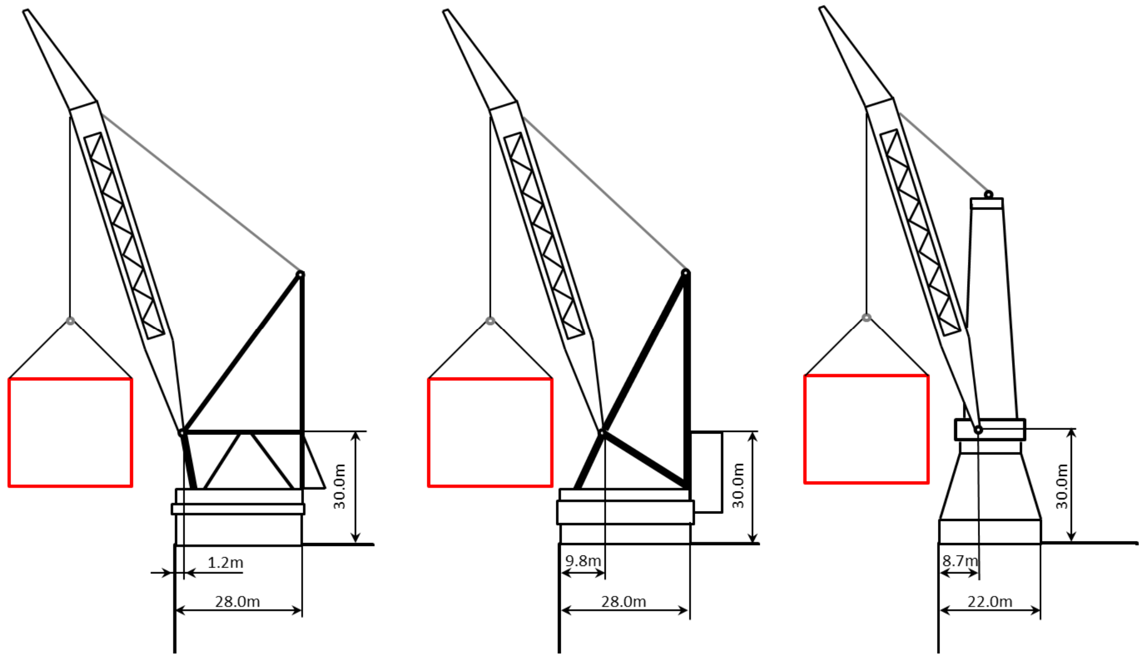


Figure 42: Boom hinge point locations, roller slew bearing A-frame crane (left) bogie slew bearing crane (center) and mast crane (right)

On the other hand, the minimum outreach of a crane is important since it determines how close a load can be suspended to the crane's centerline and to the semi-sub's edge. This crane characteristic is important when lifting loads between the cranes above deck (e.g. piles shown in Figure 43). A small minimum outreach (Table 7), in combination with a small footprint, provides the possibility to position the main blocks at large distance from each other above deck. This enlarges the range where loads can be suspended above deck. Also when the minimum radius of the crane is small, loads can be suspended close to the edge of the semi-sub.

Table 7: Minimum outreach of cranes

	Balder PS A-frame bogie slew bearing	**Balder SB** A-frame bogie slew bearing	**Hermod PS** A-frame bogie slew bearing	**Hermod SB** A-frame bogie slew bearing	**Thialf** A-frame roller slew bearing	**Aegir** Mast
Original lifting capacity [mt]	2,200	3,300	3,000	5,000	7,100	4,000
Boom length (hinge point to main falls) [m]	86.5	75.6	86.5	75.6	85.3	77.1
Minimum outreach [m]	22.9	24.0	26.0	24.0	31.2	18.0
Boom hinge point from centerline [m]	3.5	4.4	3.5	4.4	12.8	2.5

Figure 43: Upending of piles (HMC)

The boom lengths of all HMC's cranes in Table 7 are approximately equal. The boom hinge point of the Thialf cranes with roller slew bearing lies far from the centerline of the crane compared to the other cranes, causing a large minimum outreach. Smallest minimum outreach, considered beneficial, is obtained with mast cranes.

The minimum outreach of a crane is displayed in its load curve, being one of the main criteria for the crane selection process. It shows which Safe Working Load (SWL) is lifted at a certain outreach. The shape of the load curve is determined by the crane components. By adjusting these components, the shape can be modified such that the requirements of the customer are met. As an example, Figure 44 shows the load curve of a the Aegir's mast crane. The topping tackle is the wire rope connection between the boom and the mast and the hoisting tackle the connection between the boom and the load.

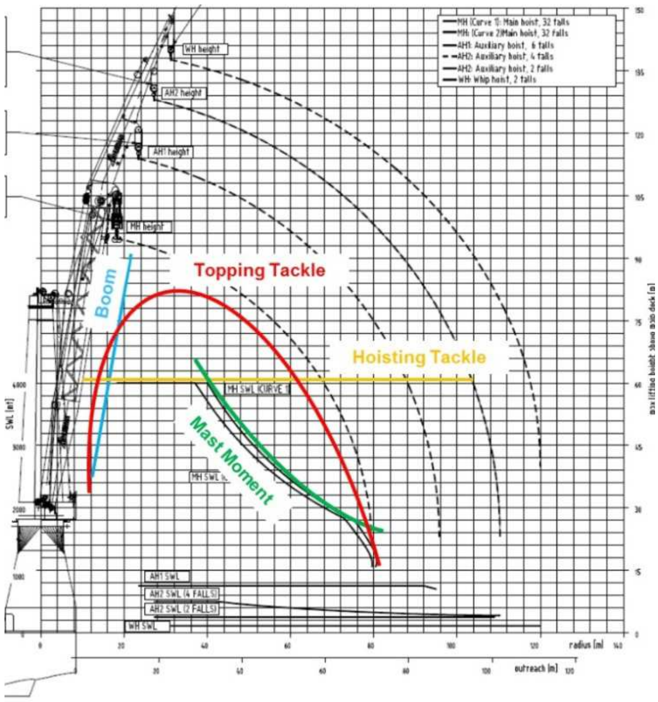


Figure 44: Load curve explanation

The load curves can be used to compare the crane performance with respect to different criteria:

- At which outreach range the maximum lifting capacity is available;
- The slope of the load curve at which the maximum load capacity decreases when the outreach of the crane is increased.

The slope of the load curve is preferably as small as possible to remain the lifting capacity of the crane at increasing outreach. High structural cost of certain crane components are involved (mast/A-frame) when the slope has to be kept as small as possible. Load curves differ per crane type, Figure 45 shows how.
This figure includes three crane types: A-frame cranes with roller slew bearing and bogie slew bearing, mast cranes and sheerlegs. Because the figure shows cranes with different lifting capacities only attention has to be paid to shape of the load and the minimum radii.

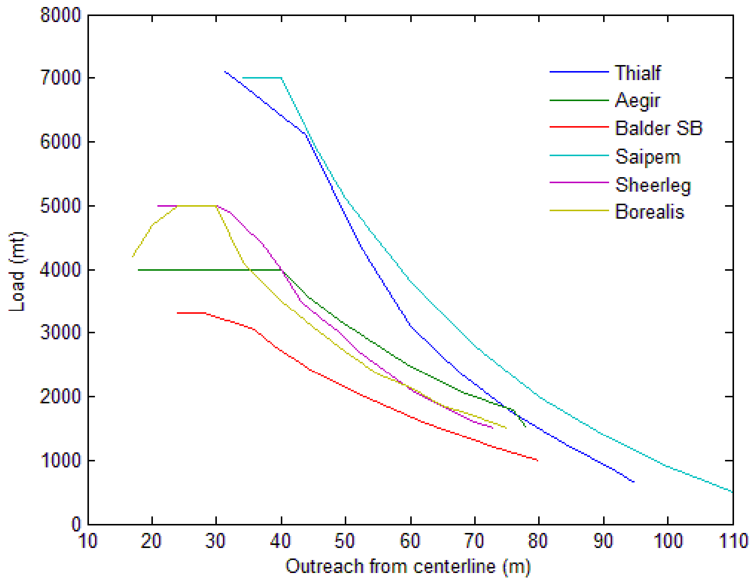


Figure 45: Load curve comparison

A comparison of the A-frame cranes with roller slew bearing on the Thialf with those on the Saipem shows that at a large boom radius the maximum lifting capacity for the Thialf is smaller than expected. It also shows clear difference between the load curves of the mast cranes of the Aegir and Seven Borealis. This is caused by the reduced number of falls in the hoisting tackle of the Aegir and the smaller moment of area of the Aegir's mast.

The maximum lifting capacity for the Aegir is available over the total slew range, but for the Seven Borealis it is only available in a certain slew range. The reason is that the extra applied material in the mast is only effective in one direction.

The curve also shows that the minimum radius of A-frame cranes with roller slew bearing is significantly larger than that of A-frame cranes with roller slew bearing, mast cranes and sheerlegs. This is mainly caused by the large footprint of A-frame cranes with roller slew bearing in combination with the location of their boom hinge point.

The location of the boom hinge point is not included in MCA since no significant effects on the crane characteristics are found. The minimum radius of a crane is important when lifting above the own deck of the NSCV. How smaller the minimum range, the larger the operating area of the cranes above deck. Therefore, criteria D is contributed with a high weight factor in the MCA.

5.2.5 Tail swing

Besides the direct space occupied by the footprint of the crane, also the space above deck, required for the slewing motion of the cranes is important (indicated by the red circle in Figure 46). The maximum tail swing of a crane is defined as the most outward point of the slew platform (excluding the boom).

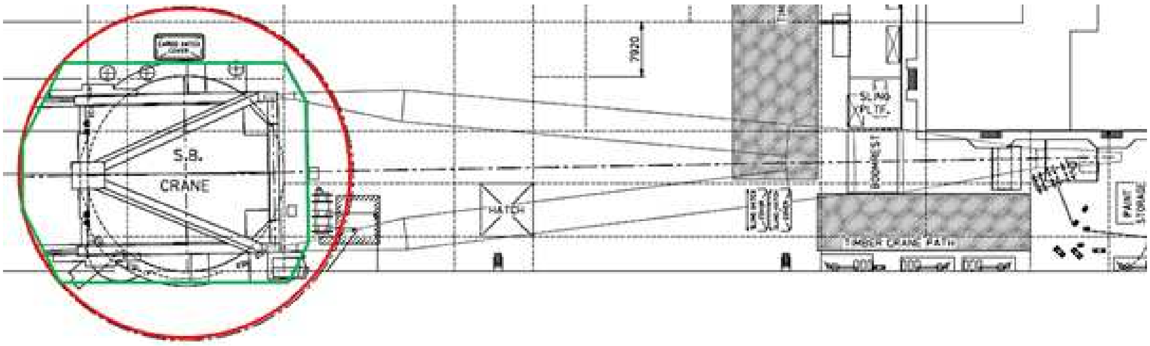


Figure 46: Tail swing of the A-frame cranes on the Thialf (HMC)

Not only the projection of the tail swing on the deck is important, but also its height above the deck. A tail swing high above deck is preferred above a tail swing close to the deck because in the first case less interference with the mission equipment on deck is expected. Because the tail of a crane is located above sea when lifting operations are carried out above the deck, the ability to lift loads between the opening of the cranes is not affected.

Cranes with counterweight (*A-frame cranes*) have generally a large tail. Reducing the tail swing of a crane with counterweight makes the counter weight less effective, requiring an increase of counter weight and larger permanent loading of the slew bearing, or resulting in a larger overturning moment acting on the pedestal of the crane.

An overview of all HMC's heavy lift cranes and the proposed 8,000mt cranes are presented in Table 8. To make a valid comparison, the tail swing height above deck is obtained by locating all the boom hinge points at 30m by increasing the height of the crane's pedestal.

Table 8: Tail swing of crane types

Vessel	**Crane type**	**Counterweight**	**Lifting capacity** [mt]	**Tail swing** [m]	**Tail swing above deck** [m]
Thialf	A-frame roller slew bearing	Yes	7,100	24.3	14.4
Balder	A-frame SB bogie slew bearing	Yes	3,300	19.8	15.7
Balder	A-frame PS bogie slew bearing	Yes	2,200	16.8	17.2
Hermod	A-frame SB bogie slew bearing	Yes	5,000	21.0	15.7
Hermod	A-frame PS bogie slew bearing	Yes	3,000	17.8	19.6
Aegir	Mast crane	No	4,000	13.4	27.0
-	Mast crane	No	8,000	18.5	27.0
-	A-frame roller slew bearing	No	8,000	17.7	12.6

The table shows that when the counterweight of an A-frame crane is removed, the tail swing is approximately equal to that of a mast crane with equal lifting capacity. A significant advantage of mast cranes compared to A-frame cranes is that the tail swing is located ±10m higher above deck. Therefore, placing equipment closer to the cranes is possible. Especially for monohulls, with limited deck space, a small tail swing prevents interference with mission equipment on deck. For semi-subs, designed as crane vessels, more deck space is available making the tail swing less critical. Because the cranes of the NSCV are installed on a semi-sub and the tail swing of the cranes is high above the deck, the weight factor of this criteria in the MCA should be medium/low.

5.2.6 Slewing range

The slewing range of a crane is defined as the angle the crane is able to slew. For *A-frame cranes* the slewing angle is not restricted because the power to the crane is supplied by slip rings and there is no relative slewing motions between the driver's cabin and the winches. The slewing angle of the Aegir's *mast crane* is restricted because the driver's cabin and the rotating winch frame undergo relative slewing motion, which requires data transfer between the two. For the Aegir's mast crane the power and data is supplied via a drag chain. It is also possible to implement wireless data transfer and supply the power by slip rings, resulting in an unrestricted slewing range. Reliability issues and involved cost resulted in the choice to apply a mast crane with a restricted slewing range (2*280̊ from the zero line, Figure 47).

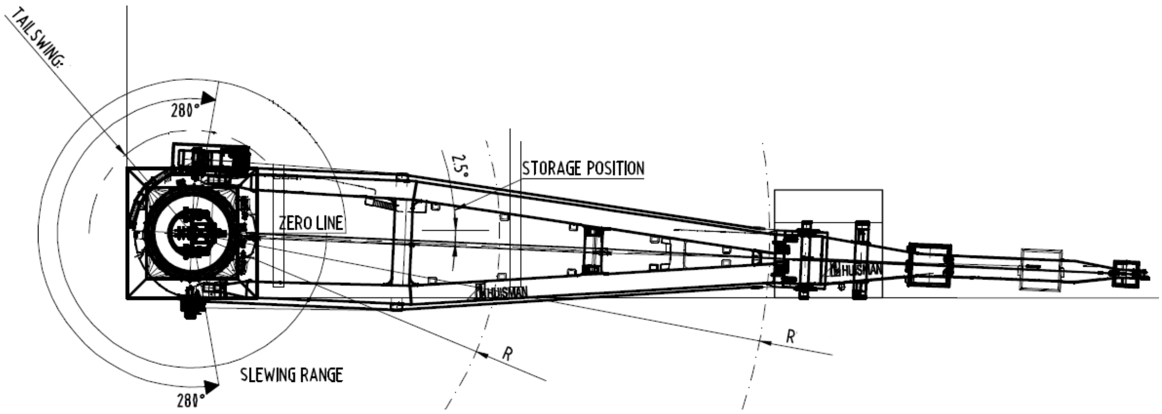


Figure 47: Slewing range Aegir (HMC)

Cranes with an unrestricted slewing range are preferred above to the ones with a restricted slewing range: the crane driver does not have to pay attention to this factor. The average time required for a lifting operation (cycle time) is less for a crane with an unrestricted slewing range than for a crane with a restricted slewing range.

How much the slewing range restrictions affect the average cycle time of lifting operations depends on their characteristics, the vessel types, vessel layout and if one or two cranes are installed. If a single mast crane with a restricted slewing range is installed on a monohull the lifting operations can be carried out in the whole 360° slew range. Thus the average cycle time of the lifting operations is enlarged.

Two mast cranes with a restricted slewing range complicate the average cycle time. The arrangement of the zero line and corresponding slewing range restrictions has to be such that all types of lifting operations can be carried out. This is schematically presented in Figure 48.

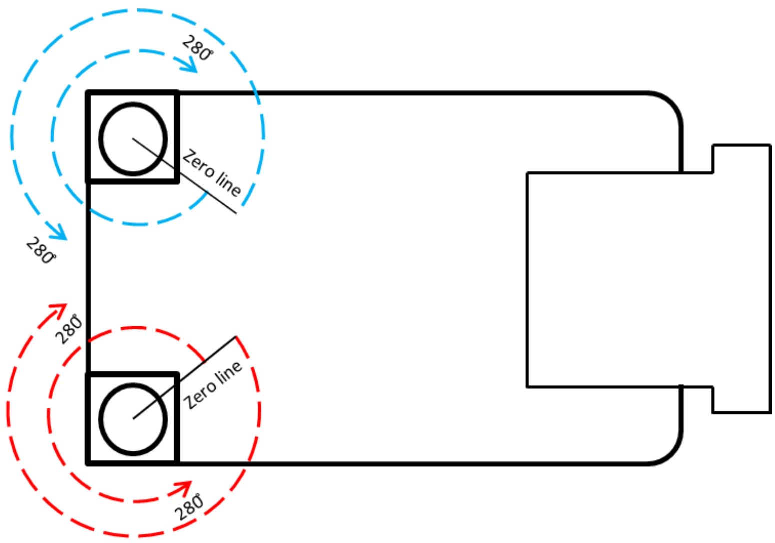


Figure 48: Arrangement of the slewing range restriction on the NSCV

In case two cranes would be installed on the NSCV it is quicker to slew over a larger angle (above sea) with a small boom angle than to slew over a smaller angle between the cranes above deck. Increasing the boom angle would be required to prevent the crane booms colliding when lifting operations are carried out in between the cranes above deck.

Two mast cranes with a restricted slewing range can carry out all possible lifting operations. To reduce the average cycle times of the lifting operations and prevent that slewing is forced to be stopped when the slewing limit is reached, the crane driver has to be fully aware at which slewing angle the crane is relative from its zero line. Thus a restricted slewing angle will not cause restrictions, but only an increase of average cycle time. This, in combination with the fact that a mast crane on the Sapura with an unlimited slewing range does not experience data transmission issues, causes a low weight factor of the slewing range criteria in the MCA.

Table 9: Slewing range

	A-frame crane	Mast crane
Slewing range	∞	∞ or 2*280°

5.2.7 Air draft

The air draft of the NSCV is determined by the air draft of the semi-sub at maximum draft (if allowable) and the remaining height of the cranes. By submerging, the draft of the semi-sub is enlarged, whereas its air draft is reduced. The remaining height of the cranes is the highest point above deck, when the boom is placed in its boom rest. For an A-frame crane this is determined by the A-frame (Figure 49) and for a mast crane by the mast. Adding the height of the remaining structure to the distance between the deck and the water level at a certain draft provides the total air draft of the NSCV.

Worldwide draft restrictions of passages apply. None of the current HMC vessels meets the minimum requirements of the passages presented in Table 10. Therefore, HMC cannot carry out projects at the Black- and Baltic sea and the transit time of the vessels can be reduced when the restrictions of passages are met.

Table 10: Draft restrictions of passages

	Panama Canal New Panamax	**Suez Canal** Suezmax	**Bosphorus**	**Baltic Sea** Storebaelt
Draft [m]	15.2 (in tropical fresh water)	18.9	n/a	n/a
Air draft [m]	57.91	68	58	65

The remaining height of A-frame cranes is smaller than that of mast cranes, but its remaining height is still too large to meet the passage restrictions. The remaining height of an A-frame crane can be reduced by applying a foldable A-frame. With folded A-frames, the NSCV is able to meet the restrictions of passages (Figure 49).

Figure 49: Saipem 7000 with folded A-frames (Safetye-managementuk)

For a mast crane the option to lower or to remove the mast is not yet available. Developing a method capable to reduce the remaining height of a mast crane is expected to require large engineering effort and to cause development risk. Development is complicated due to the large mast dimensions, the loads acting in the mast (caused by the overturning moment) and the wire ropes running in the mast. However, the A-frame folding process is also a complicated and time consuming process. Because for A-frame cranes it is already possible to reduce their remaining height and for mast cranes it is not, A-frame cranes score higher in the MCA.

Further research is needed to determine to which extent the NSCV design cost, related to the required adjustments to meet the passage restrictions, are outweighed by the economic benefits. It is expected the benefits are quite large due to the high day rates that of these crane vessels. When transit times are reduced by weeks, the saved cost are large. Therefore, this criteria has a medium/high weight factor in the MCA.

5.2.8 Vessel stability

Cranes influence the stability of the semi-sub by their weight and their CoG location. A-frame cranes have counterweight at the tail of the crane to compensate the overturning moment of the boom. Mast cranes are generally not equipped with counterweight. The counterweight affects both the Horizontal CoG (HCG) and the Vertical CoG (VCG) of the cranes.

To make a valid comparison between the CoG locations of the crane types, the boom hinge points have to located at equal heights above deck since the boom weight is ±20% of the total crane weight. Therefore, the boom weight significantly influences the VCG of the total crane. The CoG locations of the cranes are not expected to change when increasing the lifting capacities up to 8,000mt since the CoG locations of the crane components (e.g. A-frame, mast and winches) are not expected to change. Therefore, the CoG locations of the cranes in Table 11 can be used directly for crane comparison. With the CoG comparison (weight and location) is assessed to which extent the crane type affects the vessel stability.

Table 11: CoG comparison

	A-frame Thialf Roller slew bearing	**A-frame** Balder SB Bogie slew bearing	**A-frame** Hermod PS Bogie slew bearing	**Mast** Aegir
Lifting capacity revolving [mt]	7,100	3,300	3,000	4,000
Hinge point of boom to deck [m]	24.4	19.9	19.9	30.0
Weight (excl. counterweight) [mt]	5,813	4,175	3,371	4,330
Counterweight [mt]	1,100	1,000	793	-
Total weight [mt]	6,913	5,175	4,164	4,330
Original VCG above deck [m]	19.16	14.1	17.09	29.0
Compensated VCG above deck [m]	24.76	24.2	27.19	28.7
CoG from centerline, in boom rest [m]	12.36	17.37	21.17	18.0

The total weight of A-frame cranes with roller slew bearing and mast cranes is approximately equal to their revolving lifting capacities. For A-frame cranes with bogie slew bearing this is larger. Likely caused by the large crane dimensions compared to their maximum lifting capacities. Increasing the lifting capacity up to 8,000mt will result in approximately equal crane weights.

To assess if and how significant the CoG the a cranes can affects vessel stability, the largest weight and VCG differences are used for the calculations. This is the case when comparing an A-frame crane with roller slew bearing and counter weight, with a mast crane without counterweight.

For HMC's semi-subs, counterweight is less important compared to other heavy-lift companies because the semi-subs of the HMC fleet have greater stability, indicated by the GM-value (Figure 50). The GM-value, called the metacentric height, is the distance between the CoG and the metacenter (determined by a ratio of the inertia resistance of the semi-sub and its volume). A larger GM-value means a larger righting arm and greater stability, causing the roll or pitch angles by crane slewing to be small.

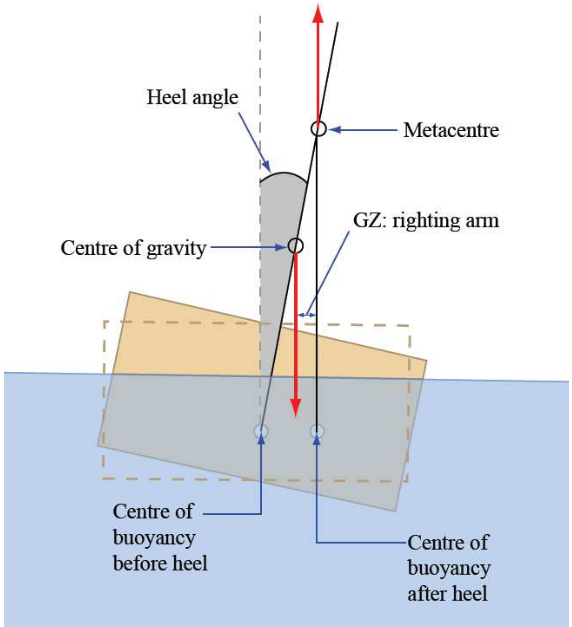


Figure 50: Vessel stability

The design of the NSCV is for a large part determined by the HMC minimum requirement for stability, given by the transverse GM-value:

$$GM_T \geq 116 * \frac{load}{\Delta} \tag{1}$$

When a total load of 16,000mt and the NSCV displacement (Δ) of 240,000m^3 are taken into account, the GM_T has to be at least 7.7m. The cranes influence the GM_T by their weight and CoG location. When the height of the CoG above deck is considered, the influence on the GM_T is given by ΔGM_T as function of the difference in the height of their CoGs (dZ) and the total weight of the cranes:

$$\Delta GM_T = \frac{dZ * weight}{\Delta} \tag{2}$$

As seen in Table 11, the VCG of a mast crane (28.7m) and of an A-frame crane with a roller slew bearing (24.8m) differs significantly. The GM_T difference caused by this difference is 0.23m and means that the Capital Expenditures (CAPEX) of the NSCV are decreased or the maximum allowable lifting height of the cranes can be enlarged. The analysis shows that the maximum lifting height of A-frame cranes can be increased with 3.5m compared to mast cranes.

Also the distance between the HCG and the centerline of the crane differs for an A-frame crane and a mast crane. The effects of this distance on the roll of a vessel is most critical and is assessed for the most extreme case: one crane is in its boom rest and the other crane has its boom fully luffed at an angle of 70̊. The boom weights of the mast crane and the A-frame crane with a lifting capacity of 8,000mt are ±1,300mt.

For a mast crane the distance between its HCG and centerline is zero and for an A-frame crane 4.8m towards the tail of the crane (when its boom is left out of the calculations). The weight of a mast crane with a lifting capacity of 8,000mt is approximately 7,000mt with boom and 5,700mt without boom. For a mast crane the total weight is 8,200mt and without boom 6,900mt.

Now the roll angle caused by crane slewing, changing the HCG of the cranes, will be calculated with the NSCV at maximum draft and without ballast water compensation. The righting moment should be equal to the overturning moment caused by the CoG from the centerline of the crane. For the righting moment, up to 10̊ of roll according standards, the righting arm is given by:

$$GZ = \frac{righting\ moment}{\Delta} = GM * sin(\varphi) \tag{3}$$

The NSCV will have a GM value of ±20m at maximum draft. When the crane slewing case is calculated as described in previous paragraphs, the following results are obtained: the roll angle for the NSCV with A-frame cranes is 0.13° and with mast cranes 0.37°. A roll angle of 0.5° is considered as maximum acceptable by HMC. This value is small because besides the lifting activities the semi-sub is used as a working island. Thus, for an A-frame crane a larger load moment can be slewed compared to a mast crane before the maximum roll angle is exceeded.

The calculations, comparing A-frame cranes with mast cranes show that a load can be lifted 3.5m higher above deck in case A-frame cranes are installed instead of mast cranes. Also the CAPEX of the NSCV can be decreased when A-frame cranes are installed instead of mast cranes. Last result is that a heavier load can be suspended by an A-frame crane than by a mast crane before the roll angle of the vessel is compromised. The effects of the crane's weight and CoG location on the vessel stability result in a low/medium weight factor in the MCA.

5.2.9 Load handling

During lifting operations several load tuggers are secured to the load to reduce its swinging and/or to control its orientation. The tugger wire arrangement and tension in the wire ropes influences the motions of the load. The most common way load tuggers are applied is by placing them on the slew platform and guide the tugger wire to a fairlead on the boom to the load (preventing lateral motion, but can rotate radially).

Usage of a fairlead is advantageous because the angle of the sheave in the fair lead is optimal to the load. When a load pin is present in the fairlead, accurate information about the tugging force on the load is provided. Placing the fairlead on the boom instead of on the slew platform itself, causes that the tugger wire length adjustments, needed during boom luffing is reduced.

To control the orientation of the load more effective when the tugger wires are cross-linked it is beneficial to place the tuggers as far from the centerline of the boom as possible. Crosslinking means that the tugger wire on the left side of the crane's boom is secured to the right side of the load and vice versa.

For A-frame cranes the load tuggers can be placed at larger distance from the centerline of the boom than for mast cranes. Simply because the width of the boom is larger. With a larger boom width, the orientation of the load can be controlled more effectively. Figure 51 shows the tugger wire arrangement on a mast crane. The red arrows indicate the tuggers to the blocks and the yellow arrows indicate the load tuggers to the load.

Figure 51: Load tuggers on Seven Borealis (HMC)

Load tuggers are essential for load handling. When equal load tuggers are installed on A-frame cranes they are more effective in load handling than on mast cranes. To obtain the same load handling capabilities, load tuggers with larger pulling capacity have to be installed on a mast- than on an A-frame crane. Because only a small cost difference is caused compared to the total cost of the crane this criteria is not listed in the MCA.

5.2.10 Deep-water lifting and lowering

Deep-water lifting and lowering operations cause an amplified vertical motion of the load, even under a mild sea state, due to axial resonance of the wire ropes, caused by the elasticity. A heave compensation system is used to control the vertical motion of the load and to reduce the dynamic loads in the hoisting system.

The main requirement for the cranes related to deep-water lifting and lowering is the space required for the heave compensation system (Figure 52) and the storage winch. Therefore, this space requirement for the equipment is included as parameter in the MCA. Since growth is expected since the offshore exploration and production head to increasing water depths even beyond the 3,000m, also the redesign effort and complexity to adjust the cranes to growth in deepwater lifting and lowering capabilities is considered in the MCA.

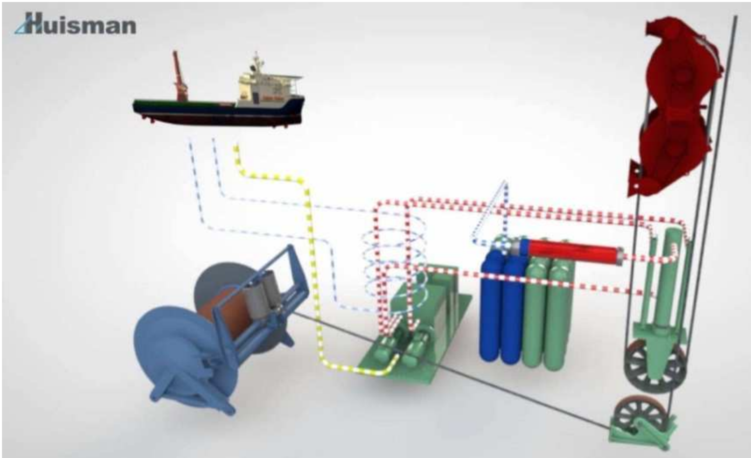


Figure 52: Active heave compensation

Generally more space is available in A-frame cranes than in mast cranes if the heave compensation system has to be installed in the crane and the Abandonment and Recovery (A&R) wire, used for pipelines and the installation of infrastructure on the sea floor, has to run through the crane.

A-frame cranes provide an easier accommodation for the heave compensation system than mast cranes, which have less space is available in its pedestal. For mast cranes the extra space required can be created by placing the heave compensation system in the hull of the semi-sub or by placing it on deck. Although the pedestal of the crane is a load bearing structure no complications are expected.

Should the heave compensation system be installed in a mast crane that way, its slewing range becomes restricted. In A-frame cranes the heave compensation system could be placed relatively simple, leaving the unrestricted slewing angle intact.

The criteria in the MCA for deep-water lifting and lowering concerns the redesign effort and complexity to accommodate the heavy compensation system and the flexibility of the design to accommodate changes in future demands. In Section 3.1 is concluded that the deep-water lifting and lowering market and related lifting operations will grow. This illustrates why a low weight factor should be added to this criteria in the MCA.

5.2.11 Maintenance and environmental sensitivity

The cost related to maintenance are significant during the long service life heavy lift cranes are designed for. The maintenance cost are primarily related to rotating components (e.g. bearings and gears) but in the longer term also paint and replacement of structural components due to corrosion are significant. Cranes have therefore to be designed for maintenance. Components that require frequent maintenance have to be easily accessible and possibly removable.

The main difference between A-frame and mast cranes is their different bearing type. A-frame cranes only have a slew bearing, whereas mast cranes have a slew bearing, mast head bearing and rotating winch frame bearing. A disadvantage of A-frame cranes is that both the roller slew bearing and the bogie slew bearing are open systems. For mast cranes all bearings are sealed and thus protected from environmental influences (Figure 53).

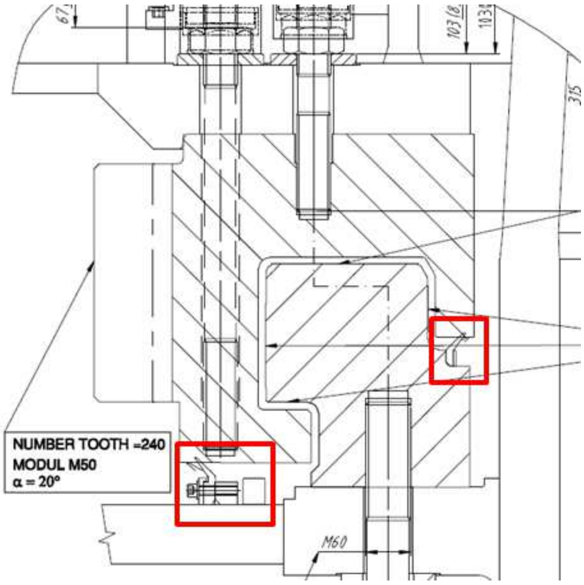


Figure 53: Sealed slew bearing of a mast crane (HMC)

Although a sealed slew bearing seems beneficial because it provides protection from the harsh environmental conditions (e.g. high salt levels, high humidity, etc.) it does not have significant influence on the crane trade-off. The not-sealed slew bearings are functioning properly on the Balder, Hermod and Thialf. There is no evidence that sealed bearings are less sensitive to environmental impact. Therefore, this design aspect is disregarded in the MCA. Exact cost related to slew bearing maintenance are not available, only indications can be provided. Therefore, maintenance cost are not included in the MCA.

5.2.12 Design maturity and reliability

The reliability of the cranes is of high importance during offshore projects. The involved cost when projects are delayed by downtime of the cranes are high. A-frame cranes capable of carrying out very heavy lifts are already in use for a long period of time, whereas the very heavy lifting market is relatively new for mast cranes. This is illustrated by the track record of the lifting capacity development of the two potential crane types (A-frame and mast cranes) for the NSCV, shown in Figure 53. Of the cranes having lost their slewing ability by temporary design adaptations to enlarge their lifting capacities, only maximum slewing lifting capacities are included.

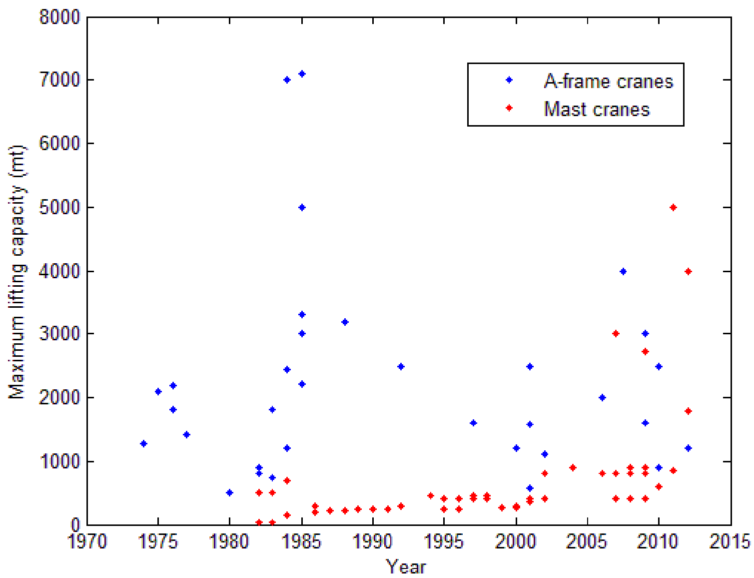


Figure 54: Maximum lifting capacity track record

Some cranes in the track record have been upgraded. Because the focus of this research lies on the maximum available lifting capacity of cranes, only the maximum lifting capacities after an upgrade have been included (at the year of the upgrade). It would be illogical to include the lifting capacity of an old crane which is significantly increased by the upgrade. A disadvantage is that if only a small upgrade is recently applied to an old crane, the crane is listed in the track record as a relatively new crane. This is justified by the focus of this research on the state-of-the-art lifting capacities of the cranes.

Figure 53 shows that A-frame cranes have the largest available maximum lifting capacity (7,100mt) and reached it decades ago at the moment the maximum capacity of mast cranes was significantly lower (700mt). The two different A-frame cranes with the largest lifting capacity (Thialf and Saipem 7000) are still performing well under heavy usage. After the mid-eighties no more cranes with such high lifting capacity are manufactured.

The lifting capacity track record of mast cranes shows that in the past years the maximum lifting capacity increased by a factor of five (from 900mt in 2004 to 5,000mt in 2011). The need for larger lifting capacity of mast cranes increases. However, the gap between the available 5,000mt and the required 8,000mt for the NSCV is still significant.

For the two crane types in the track record is it difficult to draw conclusions about their potential of increasing their maximum lifting capacities. Nevertheless, it is concluded that A-frame cranes with a lifting capacity of 7,100mt will likely perform well after a small lifting capacity increase to 8,000mt. The reliability of mast cranes still has to be proven since this crane type is relatively new to the heavy lifting market, mainly because use of plain bearings for these bearing diameters is new. The two cranes in which this type of bearing is applied (found on the Seven Borealis and Aegir) are not yet in full operation. Therefore, long term predictions about reliability are uncertain and not verified.

The design maturity of A-frame cranes is high and for mast cranes low. The involved development risks when enlarging the lifting capacity of a mast crane up to 8,000mt are also larger than for A-frame cranes. Therefore, the design maturity and reliability of the cranes are included in the MCA with a medium/high weight factor.

5.2.13 Operating cost

The main operating cost of the cranes are related to their power consumption used for slewing, luffing and hoisting operations, the ballast system and personnel cost. Also cost related to changing crane configuration, during usage have to be taken into account. For example putting the crane in tie-back mode, re-reeving the crane, changing blocks, etc. Of all factors contributing to the total operating cost, only the slewing power and the power consumption by the ballast system are suitable for comparing the two crane types, the other factors are equal.

The mast of a mast crane is static, whereas the A-frame of an A-frame crane undergoes the same slewing motion as whole crane. Based on this, it is expected mast cranes require less installed slewing power than A-frame cranes. However, another factor influencing the required slewing power is the slew bearing type in combination with bearing loading.

For a mast crane the radial loads in the slew bearing are large and are transferred by a plain bearing. A disadvantage of this bearing type is the large friction between the surfaces as discussed in Criterion 1 (stick-slip and dynamic). For A-frame cranes the slew bearing consists out of rollers, causing less slewing friction than mast cranes. Therefore, the slewing power for a mast crane is expected to be larger than that of an A-frame crane with equal lifting capacity. This expectation is confirmed by the installed slewing power of the cranes, shown in Table 12.

Table 12: Installed slewing power

	Aegir 4,000mt	**Thialf** 7,100mt	**Balder SB** 3,000mt
Slewing power [kW]	2,800 (slew platform) 90 (winch frame)	3,132	1,050
Slewing speed [rpm]	0.16 (full load) 0.4 (no load)	0.25 (full load) 0.5 (no load)	0.3
Total weight [mt]	4,300 (total) 1,300 (pedestal and mast)	6,913	5,175

The installed slewing power of the 4,000mt Aegir mast crane is approximately the same of the 7,100mt Thialf cranes. Scale enlarging up to 8,000mt would mean the installed slewing power for a mast crane is approximately double to that of an A-frame crane with roller slew bearing. For an A-frame crane with bogie slew bearing the installed slewing power is expected to be 20% smaller than for a roller slew bearing A-frame crane.

The installed slewing power of the cranes has to be put into perspective with the total installed power of the NSCV. It is expected approximately equal installed electrical power as on the Thialf is applied: 56,400kW. The installed slewing power of an 8,000mt mast crane would then be ±10% and of an A-frame crane ±5% of the total installed electrical power. Because the total electrical power is only used during transit, these percentages are expected to be even larger. However, the cranes are slewed a fraction of the time when working offshore.

The ballast system of the Thialf has a transfer capacity of 20,800m^3/hr (6*2,600m^3/hr). The total installed power (Dragonpump) is only 1% of the total installed electrical power and the cranes have a small effect on vessel stability (Criterion 8). Therefore, the influence of the total operating cost of the NSCV is negligible.

To conclude, the power consumed by crane slewing and the ballast system is expected to be a fraction of the total operating cost of the NSCV. Therefore, this criteria is not included in the MCA.

5.2.14 Cost cranes

The total cost of the cranes needed to fulfill the required lifting demands of the NSCV are an important criteria in the crane trade-off. It has to be kept in mind that if application of certain crane types has extra requirements on the semi-sub, these extra cost also have to be included. Unfortunately no specific data are available about the crane cost installed on the semi-subs of HMC. Also it is not valid to compare the cost of cranes manufactured 30 years ago with the cost of the Aegir mast crane manufactured in 2012.

The cost related to transport of the crane, installation and commissioning will not be taken into account in this Master Thesis because this highly depends on the location the crane is built. Also the cost related to testing and decommissioning of the crane will not be taken into account because this will provide a significant difference between the crane types. This makes the MCA a technical trade-off with only the operating cost taken into consideration.

6.0 MULTI-CRITERIA ANALYSIS OF AVAILABLE CRANE TYPES

The criteria in Section 5.2 cover the different aspects in the crane trade-off. An MCA is applied to integrate the information and trade-off the cranes in a transparent and systematic way. The criteria weights are determined by a pair-wise comparison of the criteria. By a pair-wise comparison the preferences of relative importance are thought out well due to its systematic method.

The weight factors are applied to four crane types: A-frame cranes (the distinction is made between the roller and bogie slew bearing), mast cranes and slewing-mast cranes. The slewing-mast crane is included to validate the strength of the MCA. Beforehand it is already known this crane type is unfavourable.

The used weight factors vary between one (equally important weight factors) and nine (more important or less important). In case the criteria in the row is preferred to the criteria in the column, a rating larger than one is given. Vice versa, when the criteria in the column is preferred to the criteria in the row, the reciprocal value of the rating is given.

Table 13: Weighing factors MCA

More importance than	Equal	Less importance than
9 8 7 6 5 4 3 2	1	2 3 4 5 6 7 8 9

In Section 5.2 an indication is given about the weights to be given to the criteria. In this section the actual criteria weights for the MCA are determined by means of a pair-wise comparison (Table 14). The criteria in the rows are compared with the criteria in the columns. For example the criteria 'Lifting capacity up to 8,000mt' is much more important than the criteria 'Deep-water lifting and lowering'. Therefore, a value of nine is given, shown in Table 14.

Table 14: Pair-wise comparison

Random criteria ranking		C1	C2	C3	C4	C5	C6	C7	C8	C9	C10
Lifting capacity up to 8,000mt	C1	1,00	5,00	3,00	2,00	5,00	6,00	2,00	6,00	9,00	6,00
Lifting capacity >8,000mt	C2	0,20	1,00	0,50	0,17	4,00	3,00	0,33	2,00	3,00	0,33
Footprint	C3	0,33	2,00	1,00	2,00	4,00	8,00	2,00	6,00	7,00	2,00
Boom hinge point location and load curve	C4	0,50	6,00	0,50	1,00	4,00	6,00	1,00	2,00	7,00	3,00
Tail swing	C5	0,20	0,25	0,25	0,25	1,00	4,00	0,25	0,50	6,00	0,20
Slewing range	C6	0,17	0,33	0,13	0,17	0,25	1,00	0,20	0,33	0,50	0,14
Air draft	C7	0,50	3,00	0,50	1,00	4,00	5,00	1,00	3,00	5,00	7,00
Vessel stability	C8	0,17	0,50	0,17	0,50	2,00	3,00	0,33	1,00	3,00	0,33
Deep-water lifting and lowering	C9	0,11	0,33	0,14	0,14	0,17	2,00	0,20	0,33	1,00	0,20
Design maturity and reliability	C10	0,17	3,00	0,50	0,33	5,00	7,00	0,14	3,00	5,00	1,00
	Total	3,34	21,42	6,68	7,56	29,42	45,00	7,46	24,17	46,50	20,21

With the sum of the columns shown in Table 14, the matrix is normalized to find the average of the rows. These averages are the weights of the criteria. The weights of all the criteria sum to one and are shown in Table 15.

Table 15: Normalized matrix

Random criteria ranking		C1	C2	C3	C4	C5	C6	C7	C8	C9	C10	Weight
Lifting capacity up to 8,000mt	C1	0,30	0,23	0,45	0,26	0,17	0,13	0,27	0,25	0,19	0,30	0,26
Lifting capacity >8,000mt	C2	0,06	0,05	0,07	0,02	0,14	0,07	0,04	0,08	0,06	0,02	0,06
Footprint	C3	0,10	0,09	0,15	0,26	0,14	0,18	0,27	0,25	0,15	0,10	0,17
Boom hinge point location and load curve	C4	0,15	0,28	0,07	0,13	0,14	0,13	0,13	0,08	0,15	0,15	0,14
Tail swing	C5	0,06	0,01	0,04	0,03	0,03	0,09	0,03	0,02	0,13	0,01	0,05
Slewing range	C6	0,05	0,02	0,02	0,02	0,01	0,02	0,03	0,01	0,01	0,01	0,02
Air draft	C7	0,15	0,14	0,07	0,13	0,14	0,11	0,13	0,12	0,11	0,35	0,15
Vessel stability	C8	0,05	0,02	0,02	0,07	0,07	0,07	0,04	0,04	0,06	0,02	0,05
Deep-water lifting and lowering	C9	0,03	0,02	0,02	0,02	0,01	0,04	0,03	0,01	0,02	0,01	0,02
Design maturity and reliability	C10	0,05	0,14	0,07	0,04	0,17	0,16	0,02	0,12	0,11	0,05	0,09
	Total	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00

The three most important criteria are: lifting capacity up to 8,000mt, footprint and the air draft. The data in Table 15 are used to calculate the consistency. This ratio measures the extent to which the weights are given consistent to the criteria in the pair-wise comparison. In this case the consistency ratio is 0.05, where 0.1 is the maximum allowable (Dalalah 2010). Thus the weights are given consistent to the criteria.

To rate the four crane types, scoring is given of each crane type on each criterion. The scores are multiplied with the weights. For a crane type the sums of these multiplications indicate how well it scores on the criteria, shown in Table 16.

Table 16: Crane type ratings

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

As expected, the sum for the slewing-mast crane is low. The score differences for the two A-frame crane types and the mast crane are small. Therefore, based on current comparison, no convincing advice can be provided to HMC which crane type is most suitable for the NSCV.

Most of the trade-off criteria depend on the original crane characteristics and the effects on the semi-sub. However, the score of the air draft criterion depends on the possibility to implement an additional crane functionality, providing the possibility to reduce the remaining height of the crane.

On A-frame cranes foldable A-frames can be implemented, but for mast cranes such an option does not exist. Therefore, mast cranes score lower on this criterion. In case a method is found to reduce the remaining height of a mast crane, a more convincing crane choice can be provided to HMC. Then the mast crane will score higher, whereas the scores of the other crane types do not change. The score of a mast crane on the air draft criterion is expected to change from 2 to 3, increasing its total score from 3.43 to 3.58. Therefore, a detailed research, regarding the possibilities to reduce the remaining height of a mast crane, is carried out in the following chapter.

7.0 REMAINING HEIGHT REDUCTION

The remaining height of a crane is an important criterion in the crane type trade-off, as shown in Chapters 5 and 6. A foldable A-frame is able to reduce the remaining height of A-frame cranes (Saipem). At this moment it is not possible to reduce the remaining height of mast cranes. In this chapter it is shown which options are available to temporarily reduce the remaining height of a mast crane.

7.1 Boundaries

When assessing the options to reduce the remaining height of mast cranes, boundaries have to be taken into account. These boundaries are related to which extent the remaining height has to be reduced and which crane design restrictions have to be taken into account.

7.1.1 Air draft

The remaining height of a crane, being the highest point of a crane above deck when the boom is placed in its boom rest, affects the air draft of the NSCV. The (air) draft is the driving factor for transit through certain passages.

The worldwide (air) draft restrictions of passage are presented in Table 17. In this table the beam of a vessel is the measured width of the broadest part of a vessel.

Table 17: Restrictions of passages

	NSCV	**Panama Canal** New Panamax (DNV)	**Suez Canal** Suezmax (Lethsuez)	**Bosphorus** Black sea (Bosphorusstrait)	**Storebaelt** Baltic Sea (Storebaelt)
Length [m]	180	366	n/a	n/a	n/a
Beam [m]	93	49	n/a[2)]	>93m	>93m
Draft [m]	10-30	15.2[1)]	18.9	>30m	>30m
Air draft [m]	-	57.9	68	58	65

[1)] = In tropical fresh water

[2)] = Vessels with beam over 74.75m may be allowed to transit the canal under special request

Transit through the Panama Canal with the NSCV is not possible because its 93m beam exceeds the restriction of 49m, its draft is therefore not relevant. The beam of the NSCV is no restriction for the Suez Canal, Bosphorus and Storebealt. For these passages the air draft is generally governing. However, for the Suez Canal also the maximum draft of 18.9m has to be taken into account.

7.1.2 Remaining height and height boom hinge point

The total height, from the keel to the deck of the semi-sub, is 50m. With the NSCV's maximum draft of 30m and the maximum allowable (air) draft of the passage restrictions, the maximum allowable air draft above its deck is calculated, shown in Table 18.

Table 18: Allowable air draft above deck

	Suez Canal Suezmax	**Black sea** Bosphorus	**Baltic Sea** Storebaelt
Max. remaining height structure [m]	36.9	38	45

Since all passage restrictions have to be met, the maximum allowable height above deck and the remaining height of the cranes is 36.9m. This can only be achieved when the complete structure of the mast crane above its slew platform is removed (Figure 55). This removable mast section has a weight of 1,256mt and a height of 50m. The boom hinge point height above the deck (30m) is not allowed to be changed because it is a main boundary, shown in Table 1.

Figure 55: Mast section to be removed (HMC)

7.1.3 Functional requirements

The functional requirements, on the type of connection that has to be applied between the mast sections and on the system to reduce the remaining height, are mainly determined by the easy of usage, the frequency and the time demand. It is expected that the remaining height of the cranes only has to be reduced once every few years. Therefore, as folding the A-frame, a quite complicated and time demanding activity in the other of days, may be acceptable. Another requirement is that the NSCV has to be able to reduce its remaining height by its own resources.

7.1.4 Re-reeving

In the mast several wire ropes are routed for the topping tackle and the hoisting tackles (the main, aux and whip hoists). Removing and re-reeving these wire ropes is a time consuming process, taking days. An important constraint is therefore that the wire ropes have to stay in place when reducing the remaining height of the crane.


Figure 56: Wire ropes running in the mast (HMC)

7.1.5 Load testing

Reducing the remaining height and restoring the crane in its original configuration has to be realized without harming the main structure of the crane. If the main structure is harmed a load test is required, which is a costly and time consuming process. Cutting the mast and re-welding it after the restrictions are passed is therefore not an option. The only option is to apply a dedicated design for the connection between the two mast sections (e.g. a bolted flange joint) that does not harm the main structure of the crane.

7.2 Crane to reduce remaining height

Two options exist to reduce the remaining height of the two cranes installed on the NSCV. The first option is to redesign crane one such that its remaining height can be reduced by crane two, and to redesign crane two such that it is able to reduce its remaining height by its own resources. The second option is to redesign both cranes to make them able to reduce their remaining height by their own resources. It is assumed the first option is most optimal in cost and time point of view. Further details are given at the end of Chapter 8 because it depends on the required redesign of the cranes and the easy of usage of the concept.

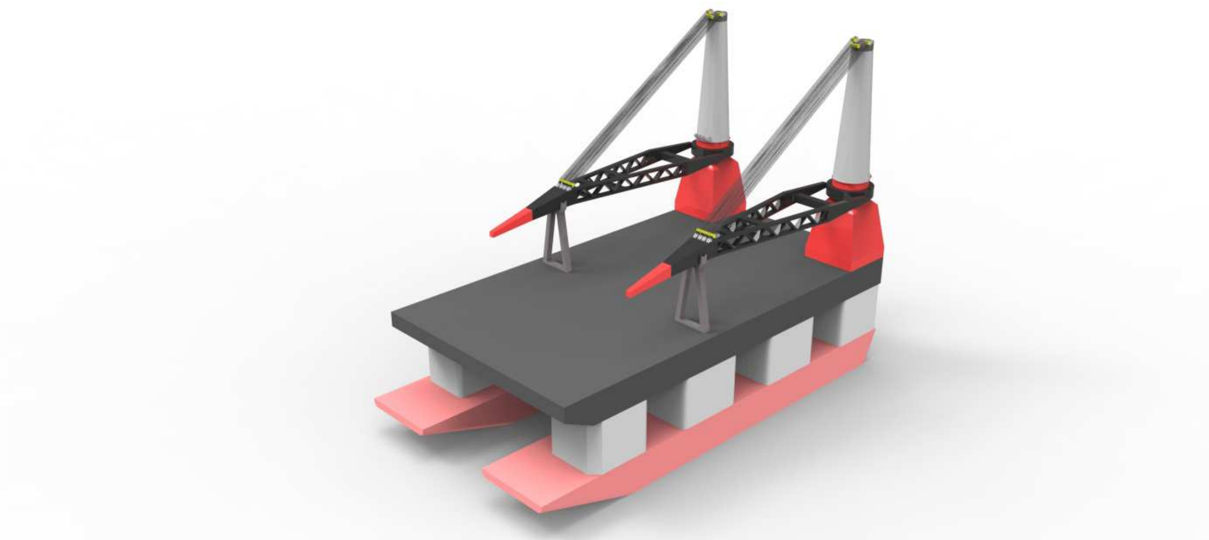


Figure 57: Cranes in original configuration

For the first option, before crane two can reduce the remaining height of crane one, the topping tackle of crane one needs to be relocated by crane two (Figure 58). Relocation is required to prevent re-reeving of the wire ropes (discussed in Section 7.1.5) and is done by relocating the sheave nests, together with the wire ropes. The sheave nests are moved from the boom’s tip to the bottom of the removable mast section. When the sheave nests are relocated, the mast section is removed and placed on the deck of the NSCV by crane two. Besides relocating the sheave nests, also a connection needs to be applied between the two mast sections.

Figure 58: Sheave nests on crane one relocated

The feasibility of reducing the remaining height of crane one by crane two, depends on the load curve of the cranes, the distance between the centerlines of the cranes (72m) and the height at which the crane hook(s) are secured to the removable mast section. Changing the orientation of the removable mast section from vertical to horizontal, before the mast section can be laid down on deck, requires the use of two hoisting blocks. One block is secured to the top and the other block to the bottom of the removable mast section.

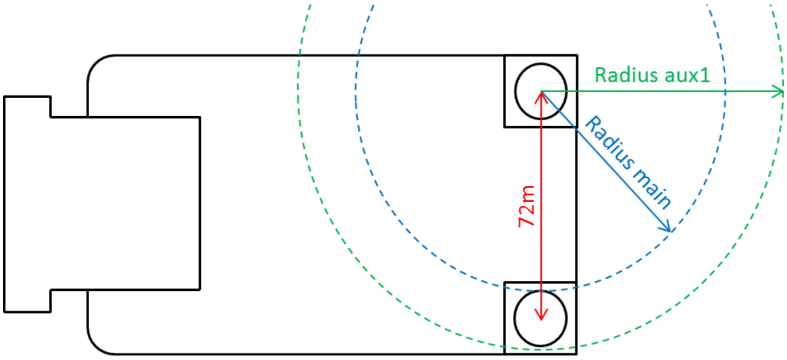


Figure 59: Distance between cranes and lifting radii of the hoists

Figure 60 shows the maximum lifting heights and the lifting radii of the main and aux hoisting blocks of the Thialf cranes. These load curves are used as a reference since no load curves of 8,000mt cranes are available and the load curves of the mast cranes on the NSCV will be equal or better than those. This is expected because by enlarging the lifting capacity of a crane, generally its load curve shape remains unchanged. The red lines in Figure 60 indicate the distance between the centerlines of the cranes and the height of the removable mast section's top (80m above the deck).

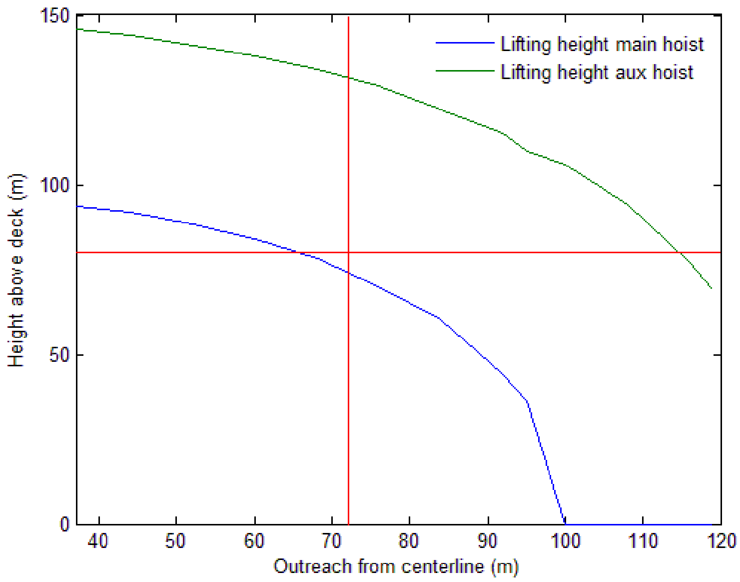


Figure 60: Required and available radius and lifting height

The figure shows it is geometrically possible to reach the mast head with the aux1 hoist, but not with the main hoist. Therefore, in this case the aux1 hoist is secured to the mast head and the main hoist to the mast bottom. Securing the hook blocks to the removable mast section has to be taken into consideration in the design of the crane because pad eyes need to be applied

It is likely the load curve of an 8,000mt mast crane is better than that of the Thialf cranes. Therefore, for this research it is assumed that crane one is able to lay down the removable mast section of crane two on the deck of the NSCV. In this case, the difficulties described in previous paragraph are avoided. It is advised that reducing the remaining height of one crane with the other is considered in the design of the cranes.

The removable mast section is placed on the deck of the NSCV between the pedestals of the cranes (Figure 61). The available space between the pedestals is 48m, taking a semi-sub width of 93m and a pedestal width of 22m into account. This is sufficient since the mast diameter of an 8,000mt crane is only ±14m. When the mast is removed, measures have to be taken protecting the wire ropes between the removable mast section and the winches, located in the remaining structure of the crane.

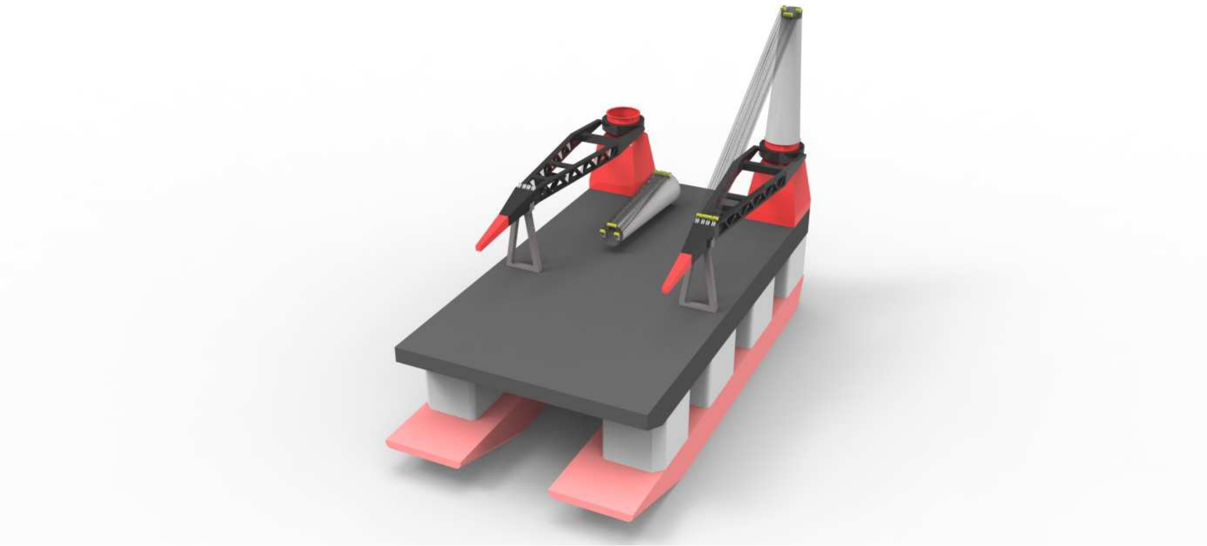


Figure 61: Removable mast section placed on the deck of the NSCV

7.3 Concepts to reduce remaining height

The result of the approach in Section 7.2 is that one of the cranes on the NSCV has still its original remaining height after it laid down the mast of the other crane on its deck. There are two options, capable of reducing the second crane, without the help of another crane (vessel): a telescopic mast (Figure 62) and overturning the mast. Overturning the mast can be done in two directions: towards the boom of the crane (Figure 63) or towards the tail of the crane (Figure 64).

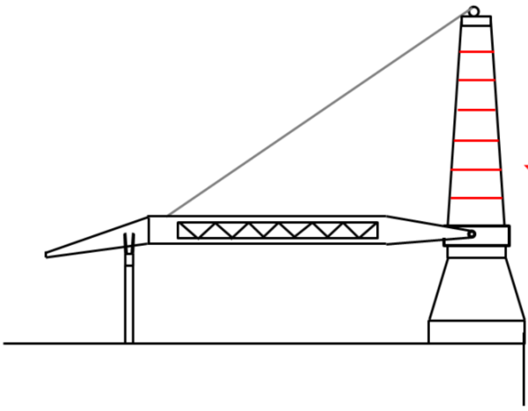


Figure 62: Telescopic mast

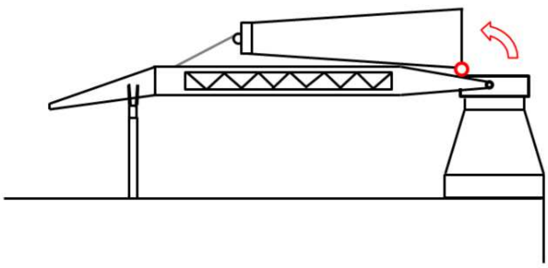


Figure 63: Mast overturned on the boom

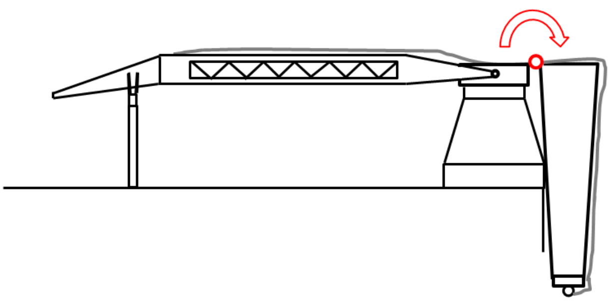


Figure 64: Mast overturned to the tail of the crane

The overturning moment of a mast crane is transferred by the mast, causing a large bending moment and large shear forces in the mast. Therefore, for the telescopic mast concept, engineering the bearings between the mast sections and securing them is complicated, and high risks are involved. This is the main reason the telescopic mast concept is not recommended. This choice is justified since Huisman also stopped the development of this concept for these reasons.

In case the mast is overturned on the boom, the boom hinge point has to be lowered to meet the maximum allowable remaining height (Table 18). To comply with the main boundaries in Table 1 this is not allowable and causes the concept in which the mast is overturned on the boom to be infeasible.

The only option left is to overturn the mast towards the tail of the crane (Figure 65). Overturning in this direction is only possible when the crane is installed with its tail to the edge of the vessel and sufficient height is available between the hinge point of the overturning mast and the waterline. When the mast is fully overturned, the mast head is hanging just above sea. For mast overturning an additional component is required, called the back-mast, shown in Figure 65. This component is needed to suspend the overturning mast. Therefore, this concept is called the “Back-mast concept”. The basic design of this concept is discussed in Chapter 8.

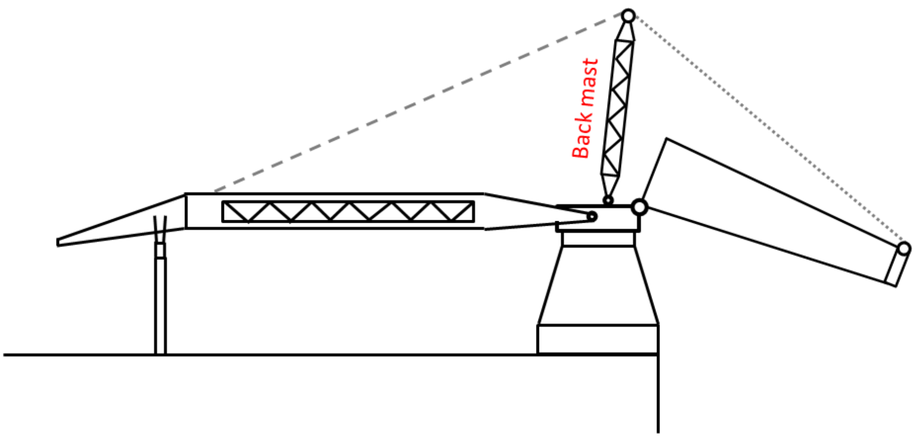


Figure 65: Back-mast concept

8.0 BACK-MAST CONCEPT

For the back-mast concept, the part of the mast above the crane's slew platform is overturned towards the tail of the crane. This chapter discusses the basic design of the back-mast and the connection between the overturned mast section and the mast section that is welded to the pedestal of the crane. Other design adjustments, required to make this concept feasible, are also discussed.

When the mast is in its original position, the back-mast (the blue component shown in Figure 66) is located between the boom and the mast. The tackles between the boom, the top of the back-mast, and the mast head consist of the original topping tackle of the crane. How this is realized is discussed in Section 8.3. Once the mast is overturning, the angle of the back-mast has to change with the overturning mast to prevent the tackles from clashing with the mast bottom, as shown in Figure 67. This figure shows the fully overturned position of the mast. The hinge points of the back-mast are placed on the slew platform of the crane.

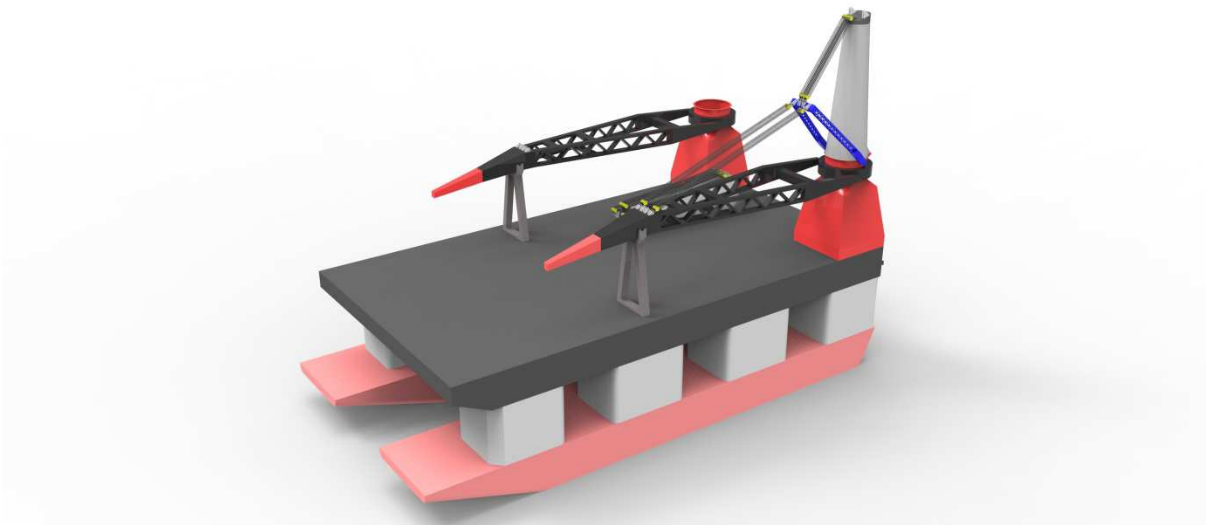


Figure 66: Back-mast concept, with the mast in original position

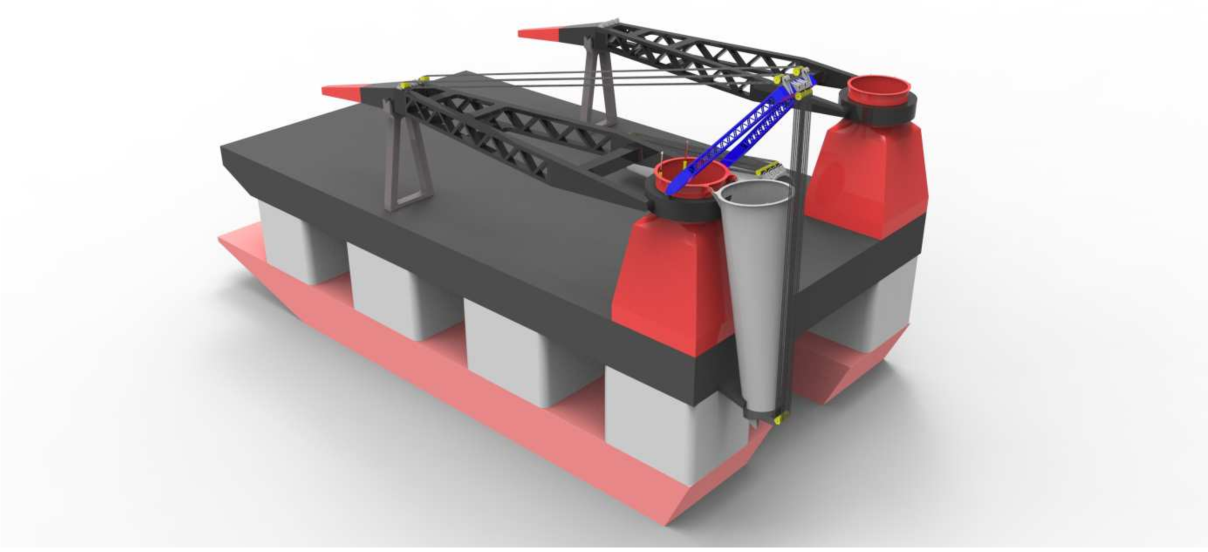


Figure 67: Back-mast concept, with the mast fully overturned

8.1 Bolted flange joint

For the two cranes on the NSCV a connection has to be applied between the two mast sections, which has to be removed when the mast is overturned. Re-welding the mast sections is not an option because it requires a load test before the crane can be put into operation (Section 7.1). A bolted flange joint is the only removable connection that is able to effectively transfer the acting loads between the two mast sections. In this section the design of the bolted flange joint is realized in five steps:

1. Determine the loads acting on the bolted flange joint;
2. Determine the number of bolts;
3. The method and time required to apply and remove the bolts;
4. Fatigue life estimation of the bolted flange joint;
5. Design of the flange.

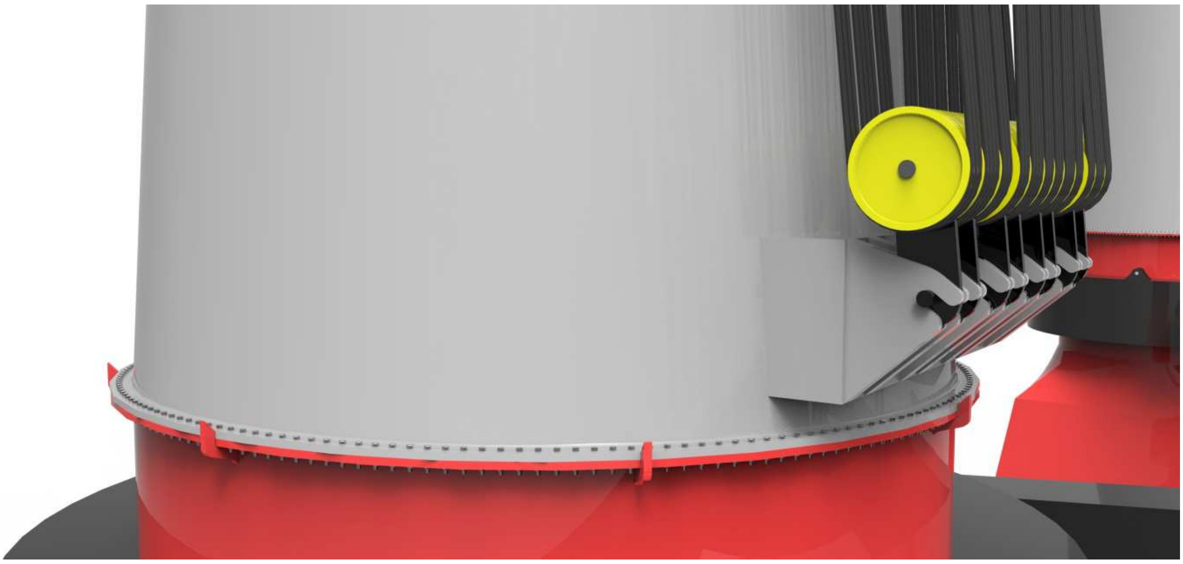


Figure 68: Bolted flange joint

8.1.1 Acting loads

The bolted flange joint has to be designed for the operating conditions of an 8,000mt mast crane. Loading of the bolted joint is caused by the loads in the topping tackle (between the mast head and the boom) and by the weight of the crane components located above the bolted flange joint (Figure 69).

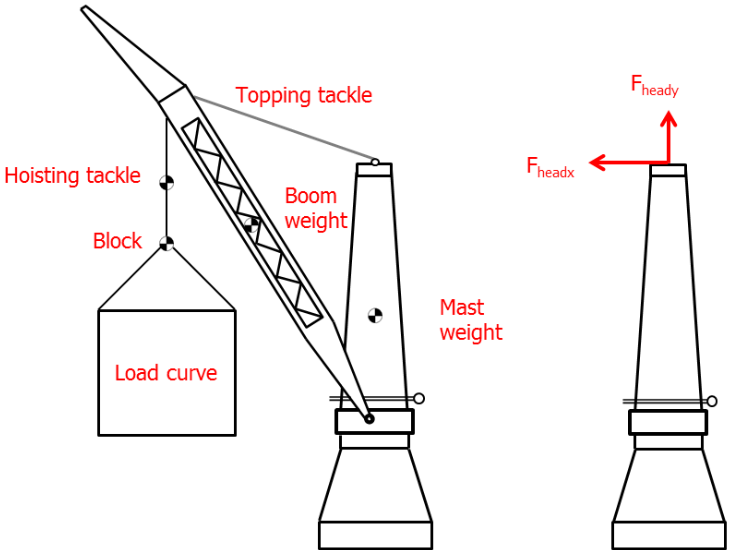


Figure 69: Loads on the mast head

The loads in the topping tackle (Figure 69) are determined by scaling the load curve of the Thialf cranes from 7,100mt to 8,000mt and taking the weight of the boom (1,400mt) and the hoisting block weight plus hoisting tackle weight (300mt) into account. Also a Dynamic Factor (DF) of 1.1 on the live load is applied. This value is also applied to the Aegir's mast crane design. With these loads the total overturning moment and the loads acting in the topping tackle ($F_{toppingtackle}$) and on the mast head (F_{headx} and F_{heady}) are calculated as a function of the crane's outreach from the centerline.

In Figure 70 the vertical loading on the mast head changes from positive (tension) to negative (compression). This is caused by the location of the boom sheaves, that fall below the mast head when the outreach of the crane is large. With the maximum loads on the mast head, and the own weight of the mast (1,250mt), the loads acting in the bolted flange joint are determined. The maximum positive vertical and horizontal forces act at the same moment when the crane is lifting at maximum capacity. These loads determine the bolt size and number of bolts. The maximum forces on the mast head are:

- F_{headx} = 4,149mt
- F_{heady} = 1,425mt (tension)
- F_{heady} = -1,533mt (compression)

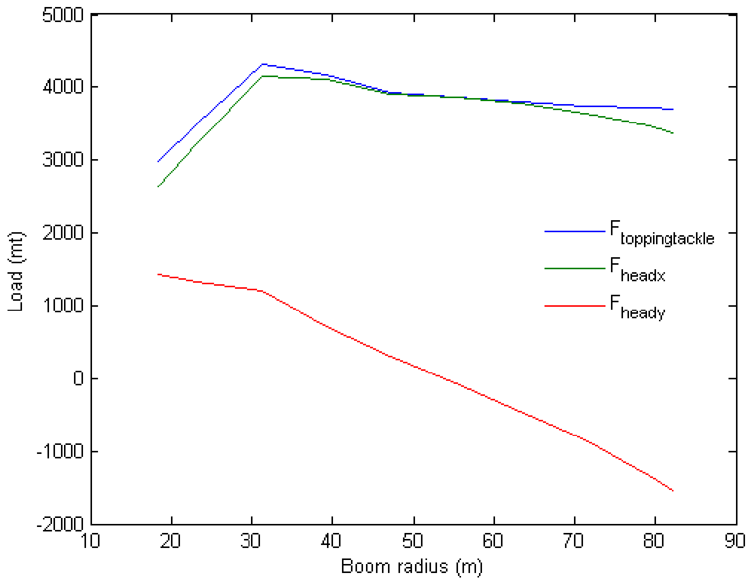


Figure 70: Loads on 8,000mt mast crane components

In the following paragraphs is discussed how these loads have to be taken into account when designing the bolted flange joint. The maximum loads in the bolted flange joint only occur a few times in the crane's lifetime. Therefore, fatigue does not play a role. Because the load resultant of the topping tackle acts at the centreline of the mast head it is assumed the vertical load acting on the mast head is equally divided over the bolted flange joint. The horizontal load acting on the mast head causes a bending moment and shear loading.

The maximum shear loading in the flange acts at the neutral axis of the bending moment. On one side of the neutral axis, the bending moment enlarges the bolt length, increasing the stress level. On the other side, the bending moment reduces the bolt length, reducing the stress level. Therefore, the average preload is present in the bolted joint on the neutral axis of the bending moment.

Shear loading with a bending moment could cause movement between the two flanges when the crane is in operation. Movement can be prevented by ensuring a minimum preload in the bolted flange joint, which is able to transfer the acting shear loading. Other options are to apply a key(way) or resin between the flanges. The first option is chosen, because only a small number of extra bolts is required. The tensile loading by F_{heady} on the bolted joint affects the average preload and is taken into account in the bolted flange joint design.

The bending moment causes that one side of the flange is under compression and the other side of the flange is under tension. Crane slewing over 180° causes, the loads acting on the bolted flange by the bending moment, changing sign. How this dynamic loading is taken into account is discussed in the following section.

8.1.2 Number of bolts

The bolted flange joint is designed such that the clamping force between the flanges is sufficient to transfer the maximum loads acting simultaneously, as discussed in Section 8.1.1. The design steps are calculating the allowable static stresses in the applied bolt type, the acting loads on the bolted flange joint and herewith the number of bolts required to stay below the maximum allowable static stress levels of the bolts.

The applied bolt type is M60-10.9 bolts. This bolt type is chosen since similar bolts are used for slew bearing securing and the bolt weight and the required hydraulic tensioning devices (Figure 72) are still manageable for personnel, whereas the time required for bolt removal and tightening is acceptable. The maximum stresses in the bolts are shown in Table 19.

Table 19: Allowable bolt stresses

	Material quality 10.9
Minimum tensile stress [kN/mm^2]	1.0
Yield stress [kN/mm^2]	0.9
Tensile stress [kN/mm^2]	0.36

The tensile stress amplitude (σ_a) of 0.36kN/mm^2 is taken into account. For a single M60 bolt with a stress area (A_s) of 2,362mm^2 the maximum load amplitude (F_a) is calculated with:

$$F_a = \sigma_a * A_s \tag{4}$$

The allowable load amplitude on a single bolt is in this case 850.3kN. The higher the stiffness factor between the clamped material and the bolt, the smaller the fraction of the external load is taken by the bolt. With an axial stiffness of the clamped material of three times the bolt stiffness, only one fourth of the external load is taken by the bolt. This is a common value in a flange design and is assumed for further calculations. In this case, the allowed load fluctuation acting on the flange is calculated with:

$$\Delta F_m = 4 * F_a \tag{5}$$

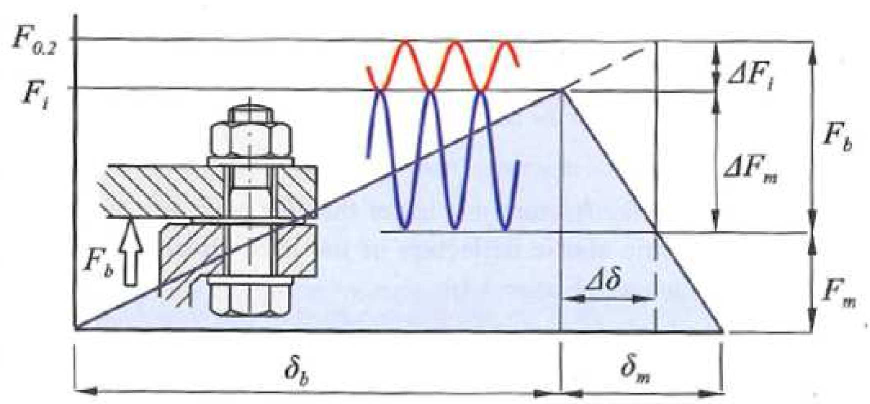


Figure 71: Load-deformation curves of the bolt and the clamped material (Beek 2006)

The allowed load fluctuation acting on the flange is in this case 3,401kN. The maximum load acting in the bolts is determined by adding the tensile loads caused by the bending moment (F_{headx}) to the tensile loads (F_{heady}) and the shear loads (F_{headx}). The sum is given by F_b and determines the dynamic loading on the clamped material (ΔF_m) and the bolt (ΔF_i) as shown in Figure 71. The clamped material is preloaded with F_i.

The minimum preload value to prevent separation of the bolted flange joint is determined by the summed total of the loading on the clamped material and the preload which always has to be present (F_m). In this case F_m is zero and therefore the minimum preload value is ΔF_m, as shown in Figure 71.

With the maximum loading on the bolted flange joint, the number of bolts in the flange joint are calculated. The bolted flange joint loading consists of the bending moment and the shear loading caused by the horizontal mast head loading and the tensile loading caused by the vertical mast head loading.

Firstly, the contribution of the bending moment (M_B) is calculated (Muhs 2003). The load amplitude by the bending moment is doubled to determine the total stress range because the bending moment is on one side of the flange positive and on the other side negative:

$$F_{bm} = \frac{2 * M_B * r_{max}}{\sum r^2} = \frac{2 * F_{headx} * mast_{height} * r_{max}}{\sum r^2} \tag{6}$$

To this, the load amplitude caused by shear loading is added. Shear loading is transferred between the two steel surfaces of the flange. The static friction coefficient (μ) between the two greasy steel surfaces is taken into account as 0.2 (Engineeringtoolbox).

$$F_s = \frac{F_{headx}}{\mu * \#_{bolts}} \tag{7}$$

The maximum bending moment acts when the mast head is under tension. The load range caused by the vertical load acting on the mast head (F_t) is added, calculated with:

$$F_t = \frac{F_{heady}}{\#_{bolts}} \tag{8}$$

The total stress range acting on the bolted flange joint (F_{max}) is calculated by adding all the discussed stress components (F_{bm}, F_s and F_t):

$$F_{max} = \frac{2 * F_{headx} * mast_{height} * r_{max}}{\sum r^2} + \frac{F_{heady}}{\#_{bolts}} + \frac{F_{headx}}{\mu * \#_{bolts}} \tag{9}$$

When the maximum allowable bolt loading is kept below 850.3kN, the minimum number of bolts applied in the flange joint is calculated as 406. For this number of bolts $\sum r^2 = 9996 m^2$.

8.1.3 Applying and removing the bolts

In this section is shown that it is possible to apply the required preload of the 406 bolts, without exceeding their yield stress. For this application and bolt size it is common practice to preload the bolts with hydraulic bolt tensioners (Figure 72).

Figure 72: Hydraulic bolt pre-tensioning

For this tensioning method an annular hydraulic jack is placed around the screw, stretching it axially. When the required stress level is reached, the nut is tightened snugly and the pressure released, resulting in a preloaded bolt without any frictional or torsional stresses.

Despite the use of thread lubricants when hydraulic bolt tensioners are applied a preload error of approximately 10% is common, depending on the bolt length. For this design analysis a 10% error seems realistic (Lifetime-reliability). Assuming this error is applied on all the bolts, a larger number bolts could be required to ensure the bolt stresses remain below their maxima. That this is not the case is shown in the following paragraphs.

The maximum preload to prevent yielding of the bolts, is given by:

$$F_{i,max} = F_{0.2} - F_a \tag{10}$$

As discussed before, the maximum allowable stress range in the flange is 3,401kN. Only one fourth of the total stress range is taken by the bolt. Therefore, the load amplitude F_a is 425kN. For an M60-10.9 bolt: $F_{0.2.}$ is 2,093kN. With the formula above $F_{i,max}$ is calculated as 1,701kN (80% of the bolt yield stress).

Besides the maximum preload also the minimum preload is determined to ensure that separation of the bolted joint is prevented. The minimum preload is given by:

$$F_{i,min} = F_m + \Delta F_m \tag{11}$$

In this case F_m is zero and ΔF_m is three times the allowable load amplitude. Then $F_{i,min}$ = 1,275kN (60% of the bolt yield stress). When the mean value of the maximum and minimum preload is taken as set point for the applied preload and the preload error of 10% is taken into account, the maximum applied preload is 1,637kN and the minimum is 1,339kN. These values are on the safe side of the values discussed before. Thus separation of the bolted joint is prevented and when the 8,000mt crane is lifting at maximum capacity the yield stress of the bolts cannot be exceeded. Therefore, the applied number of 406 bolts is sufficient.

Each M60 bolt weighs approximately 10kg. Once the bolts are removed, they are placed in crates on the gangway. Placing the hydraulic bolt tensioner, waiting until the bolt is pre-tensioned and the bolt is applied, and moving the system to the next takes approximately twelve minutes.

Bolt tensioners are used to achieve an accurate and pre-determined bolt loading. In the ideal situation all bolts in the joint would be tensioned simultaneously (100%), but in practice 50%, 33% or even 25% simultaneous tensioning is often carried out. Then two, three or four tensioning passes, by moving around the bolts in diametrically opposed fashion, are required (Metal). Partial tensioning takes longer, but enables the user to optimize between equipment cost and available time.

For the conceptual design analysis an example of the bolt tensioning time is given. For a 20% simultaneous tensioning procedure, 82 tensioners are required. The time required for all the five tension passes, is five hours when the cycle time of 12 minutes is taken into account. More in-depth research is required to optimize the cost of the equipment and the day rate of the NSCV.

8.1.4 Fatigue

In the previous calculations is assumed that the required number of bolts in the flange joint is determined by the maximum mast crane loading. However, fatigue can also be a critical factor in the design of the bolted flange joint. Fatigue could decrease the service life of the bolts significantly. In case the calculated service life of the bolts is unacceptable, enlarging the bolt number or the bolt dimensions could be necessary to increase the service life of the bolts. Another option is to take early bolt replacement into account. Therefore, bolt fatigue, caused by offshore conditions and hoisting operations, is considered in this section.

To calculate the design life of the bolts, the number of load cycles is determined by interpolating between VDI 2230 and Eurocode 3. Eurocode 3 is too conservative whereas VDI 2230 is too less conservative for a survival probability of 97.7% (Schaumann 2009). Since the bolt diameter influences the fatigue strength of the bolts, a correction factor is applied, as shown in Figure 74.

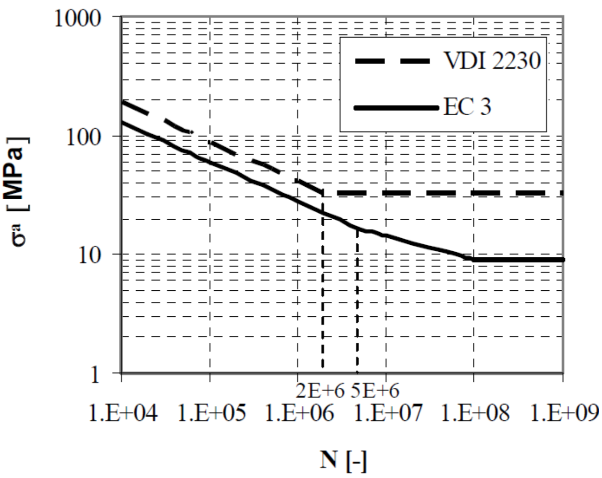


Figure 73: Normative S/N-curves of Eurocode 3 and VDI 2230

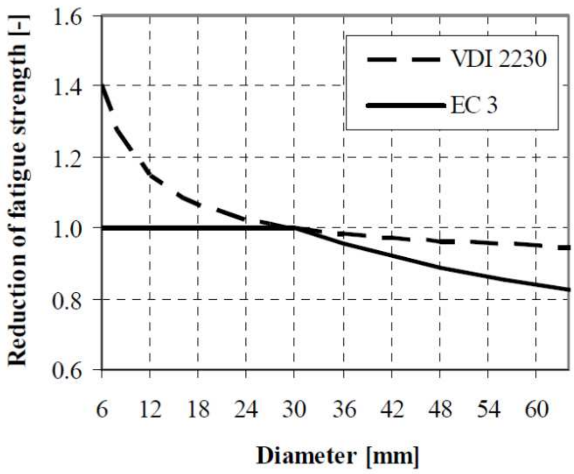


Figure 74: Influence of bolt diameter on fatigue strength

This section consists of two parts: fatigue caused by offshore conditions and fatigue caused by hoisting operations. Firstly, fatigue caused by offshore conditions is assessed. The design life of the bolted joint has to be infinite for offshore conditions due to the large number of load cycles caused by the roll and pitch motion of the NSCV. The maximum allowable stress amplitude for infinite life is 30MPa (Figure 73).

According to HMC standards, roll (ϕ) and pitch angles (ψ) of 5˚, each with a period (T_r and T_p) of 10s have to be taken into account in the design of the NSCV. When the mast cranes on the NSCV are subjected to these environmental conditions, their booms are placed in the boom rests. Therefore, it is assumed the load acting on the bolted joint is solely formed by the weight of the crane components above the flange (W = 1,250mt).

The load normal to the deck due to roll (F_{r_normal}) where y is the transverse distance parallel to deck from the center of roll motion to the CoG of the cranes (y = 30m) is given by (Lloyds 2009):

$$F_{r\_normal} = 0.07024 * W * \frac{\phi}{{T_r}^2} * y \qquad (12)$$

The maximum loads normal to the deck due to roll are 132.3mt acting in the CoG of the cranes.

The load normal to the deck due to pitch (F_{p_normal}) where x is the longitudinal distance parallel to deck from center of pitch motion to the CoG of cranes (x = 70m) is given by:

$$F_{p\_normal} = 0.07024 * W * \frac{\psi}{{T_p}^2} * x \qquad (13)$$

The maximum loads normal to the deck due to pitch are 308.8mt acting in the CoG of the cranes.

The load parallel to the deck due to roll ($F_{r\_parallel}$) where z_r is the distance normal to deck from center of pitching motion to CoG of the (z_r = 42m) is given by:

$$F_{r\_parallel} = 0.07024 * W * \frac{\phi}{{T_r}^2} * z_r \qquad (14)$$

The maximum loads parallel to the deck due to roll are 185.2mt acting in the CoG of the cranes.

The load parallel to the deck due to pitch ($F_{p\_parallel}$) where z_p is the distance normal to deck from center of pitching motion to CoG of the (z_p = 73m) is given by:

$$F_{p\_parallel} = 0.07024 * W * \frac{\psi}{T_p^{\ 2}} * z_p \qquad (15)$$

The maximum loads parallel to the deck due to pitch are 322.1mt, acting in the CoG of the cranes.

The loads caused by offshore conditions are shown in Table 20. The pitch loads, parallel to the deck are governing and causes, when a Vertical CoG (VCG) of 20m is assumed, a maximum bending moment acting on the bolted flange joint of 6.30E4kNm.

Table 20: Bolted joint loads by offshore conditions

	Load [mt]
Roll load, normal to deck	132.3
Pitch load, normal to deck	308.8
Roll load, parallel to deck	185.2
Pitch load, parallel to deck	322.1

With 406 bolts the maximum load on the flange connection is determined by taking the envelope of the maximum loads occurring at the same time. These are caused by pitch motion of the NSCV and are the normal and parallel loads. Maximum loading on the bolted flange joint due to sea conditions is based on equation (9) and is given by:

$$F_{offshoreconditions} = \frac{2 * F_{p\_parallel} * VCG * r_{max}}{\sum r^2} + \frac{F_{p\_normal}}{\#_{bolts}} + \frac{F_{p\_parallel}}{\mu * \#_{bolts}} \qquad (16)$$

The maximum force on the bolted flange joint is calculated at 134.9kN. The maximum allowable force to guarantee infinite life is 510.2kN. Therefore, when only these sea conditions are considered, the number of bolts does not have to be enlarged to guarantee the service life of the bolts. However, besides fatigue caused by sea conditions, also fatigue caused by hoisting operations has to be assessed.

The main contributors of fatigue caused by hoisting operations are aux hoist and whip hoist usage. Therefore, only these two are assessed. Firstly aux hoist usage at its maximum lifting capacity (900mt). The loads on the mast head are shown in Figure 75.

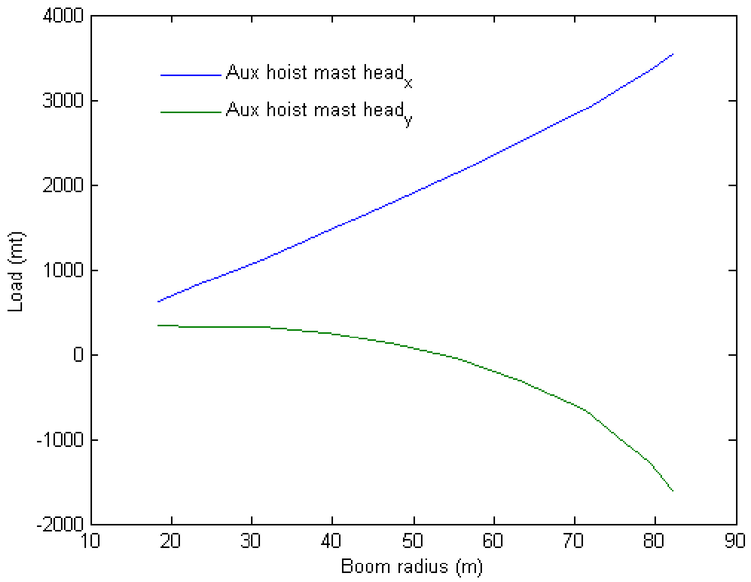


Figure 75: Aux hoist loads on mast head

With the maximum load on the bolted flange joint of 2,909kN, the maximum stress in the bolts is calculated with equation (9) as 171.0MPa. Figure 71 shows that for this stress level, the maximum allowable number of load cycles is 10,000. Not every lifting operation causes a full load cycle in all bolts. Therefore, is assumed that only a quarter of the lifting operations cause a full load cycle of a bolt. For only this case it is expected that 40,000 lifting operations can be carried out before the maximum allowable number of load cycles is exceeded.

This complies with the HMC specification of cranes, in which it is assumed the aux hoist is to be designed for 200 maximum capacity lifts each year and that a service life of 20 years is required. With the service life distribution of the cranes according HMC standards, the expected fatigue life of the bolts is calculated:

- Transit: 11%
- Maintenance: 9%
- Working offshore: 80%

Besides fatigue by aux hoist usage, also fatigue by whip hoist usage is considered. The HMC specifications for the new cranes of the NSCV show that the whip hoist has to be designed at its maximum capacity (200mt) for 25 times a day when working offshore. The total number of lifting operations in its design life is then 146,000, causing 36,500 load cycles for the bolts. The maximum loads on the mast head by whip hoist usage are shown in Figure 76.

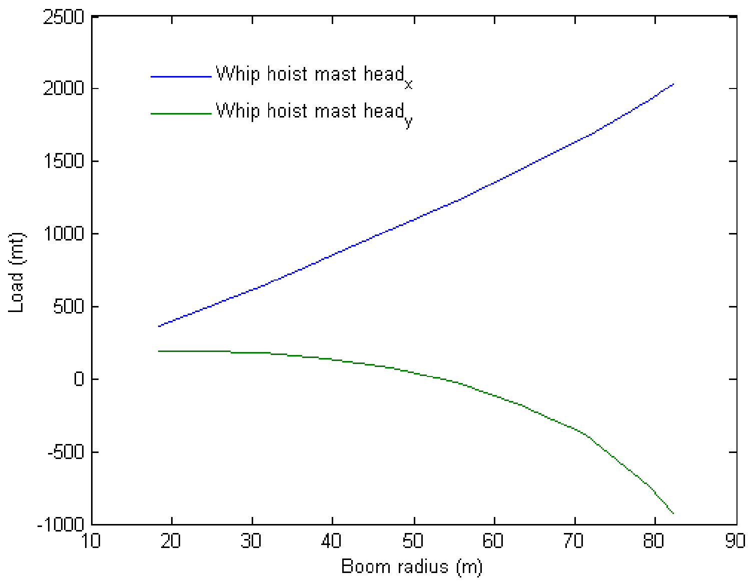


Figure 76: Whip hoist loads on mast head

In this case, the maximum force in the bolts is 1,681kN, as calculated by equation (9) The stress in the M60 bolts is then 98.9MPa. According Figure 71, the allowable number of load cycles is then 70,000. Thus the total number of lifting operations in the design life of the crane is 280,000. This is significantly lower than the total number of expected load cycles on the bolts (36,500).

Up to now, all fatigue cases are assessed individually, but actually all fatigue cases contribute to the total fatigue of the bolted joint. Miner's cumulative damage theory is applied, in which the contributions of the separate fatigue cases are accumulated. In this theory the expected number of stress cycles (n) are divided by the allowable number of stress cycles (N). The sum total has to stay below one to prevent material failure of the bolts.

$$D = \sum_{i=1}^{j} \frac{n_i}{N_i} = \frac{n_{offshore}}{N_{offshore}} + \frac{n_{aux}}{N_{aux}} + \frac{n_{whip}}{N_{whip}} \qquad (17)$$

For the offshore conditions and the two hoisting operation cases, the summed total of the calculations is 0.25. This value is below one, so the expected service life of 20 years is met with the assessed bolted joint design (406 M60-10.9 bolts). Regular inspection of the bolted joint is essential to guarantee its service life, especially because the crane usage expectations are based on simplified user schemes.

8.1.5 Flange

The next step is the design of the flange. In Section 8.1.4 a bolt diameter (d) of 60mm and a joint stiffness factor (C_m) of 0.25 are assumed. With these parameter values, the minimum required flange thickness (l_m) is calculated for a cone angle (ϕ) of 30° and the assumption that the bolt and flange are manufactured of the same material. The cone angle indicates the stressed material shape (orange in Figure 77) of the flange around the bolts.

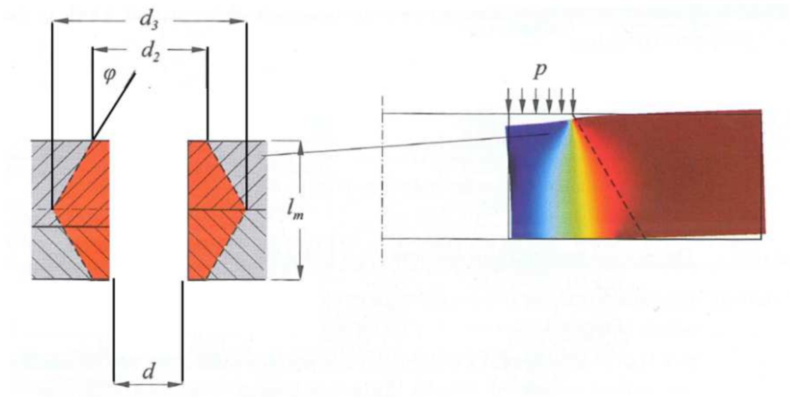


Figure 77: Stress distribution in flange (Beek)

The joint stiffness factor is given by (Beek):

$$C_m = \frac{1}{1.5 + 0.289\left(\frac{l_m}{d}\right)^2} \quad (18)$$

For this design, the minimum flange thickness is 177mm.

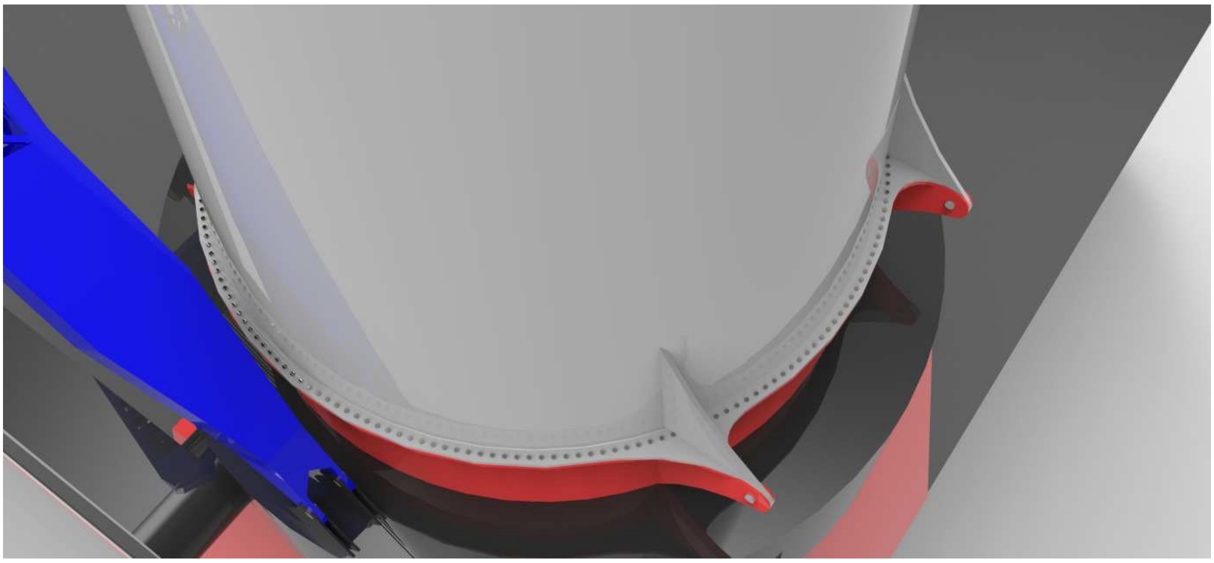


Figure 78: Basic flange design lower mast section

For load transfer between the flange and the mast a total flange thickness of 180mm is appropriate since the mast thickness is 80mm. The number of bolt rows that are required to prevent bolt loosening when the bolts are preloaded is calculated with the minimum bolt spacing (Beek):

$$d_3 \geq 1.5 * d + l_m * tan(\varphi) \quad (19)$$

The minimal required bolt spacing (d_3) is 194mm. Bolt spacing is generally determined by this value and not by the space required to place the hydraulic tensioning devices because they are compact tools. To keep the elastic deformation and stresses in the flange as low as possible, it is beneficial to place one row of bolts on the inside of the mast and on the outside.

With an average mast diameter and corresponding flange diameter of 14m, the required number of bolt rows is 1.7. In the bolted flange joint design, the 406 bolts are divided over two rows. In the calculations shown in Section 8.1.2 is assumed that the number of bolt rows has no influence on the maximum bolt loading. This is allowable since the distance between the bolt rows is small compared to the mast and flange diameter.

The next step in the flange design, is calculating the minimum distance of the bolts from the edge of the flange. This distance is given by (Beek):

$$Edge\ distance \geq 0.5 * d_3 \tag{20}$$

The minimum edge distance is 97mm. The total flange width with two bolt rows is found by two times the bolt diameter (2*60mm) and corresponding edge distances to two times the mast rounding (2*80mm) and the thickness of the mast (80mm). In total the minimum flange width is 554mm.

Another design aspect is to apply guides, ensuring that the two flanges and the bolt holes are properly aligned with each other when the overturned mast section is put back in its original position. How the guides are placed in the design differs for the two cranes on the NSCV.

The guides of the crane, whose mast section is lifted by the other crane, have to be placed around the total circumference of the flange to align the two mast sections with each other. How these guides are located is shown in Figure 79.

Figure 79: Guides on crane one

For the crane with the overturning mast, no guides have to be placed close to the hinge points of the crane. The hinge points make sure the two mast sections are properly aligned with each other. Therefore, it is assumed that only two guides are placed on opposite directions of the hinge points, as shown in Figure 80.

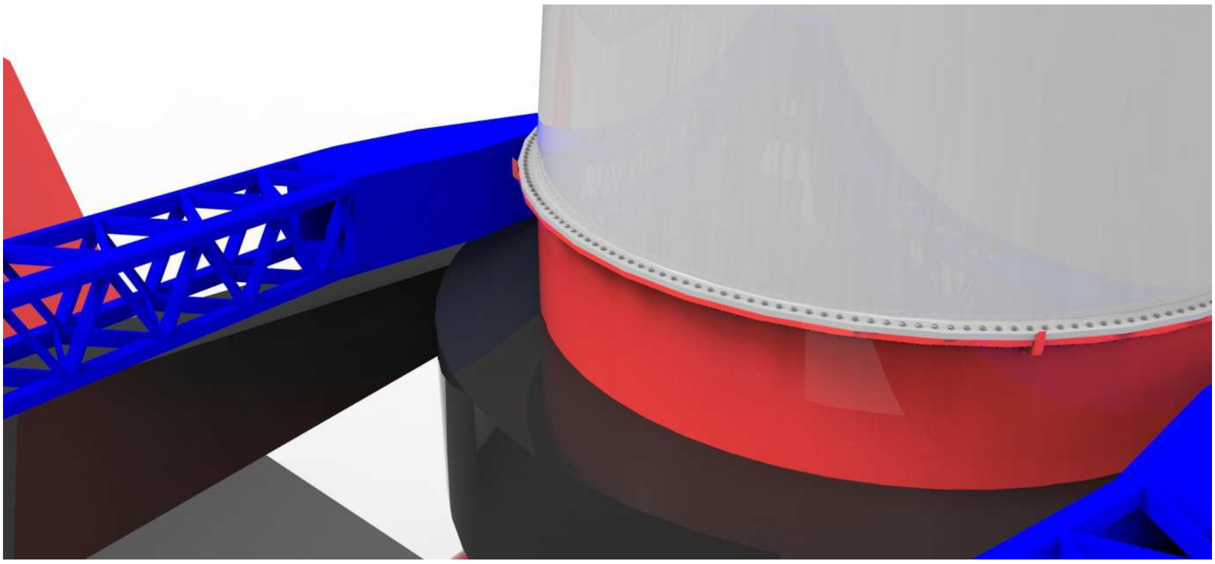


Figure 80: Guides on crane two

8.2 Overturning activation

For the back-mast concept a system is required that brings the Horizontal CoG (HCG) of the overturning mast to the other side of the mast hinge point, as is shown in Figure 81. Once the HCG is brought to the other side of the mast hinge point, the overturning mast is suspended by the tackles and the back-mast. The overturning angle (α) of the mast is shown in Figure 82.

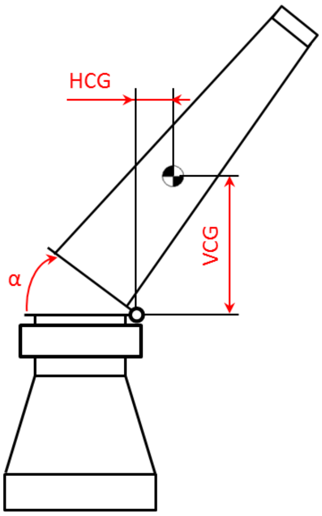


Figure 81: Overturning mast

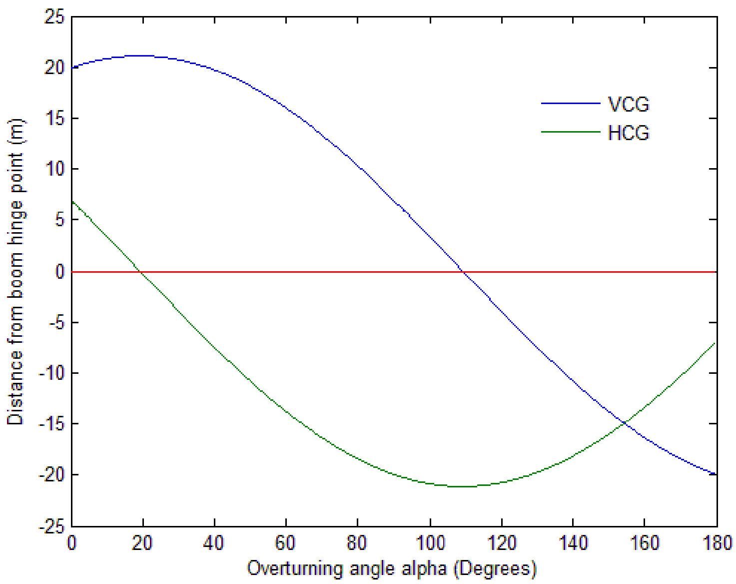


Figure 82: VCG and HCG positions depending on overturning angle

Before the HCG is zero, an angle of 19° has to be overcome (Figure 82). Besides bringing the HCG to the other side of the mast hinge point, also the mast needs to be supported when the mast is put back at its original position. Two options are assessed that can realize these two requirements:

1. Installation of hydraulic cylinders in the mast;
2. Placing a hoisting system on the slew platform.

Hydraulic cylinders can be installed in the mast section that is welded to the pedestal of the crane (red component shown in Figure 83). The hydraulic cylinders are extended until the overturning angle of 19° is reached. When the overturning mast is suspended by the tackles and the back-mast, the hydraulic cylinders need to be detached from the overturning mast.

When the overturning mast is put back at its original location, the hydraulic cylinders have to be secured to the overturning mast to gently put down the overturning mast on its support. The hydraulic cylinders are retracted until the overturning mast is fully supported by the flange.

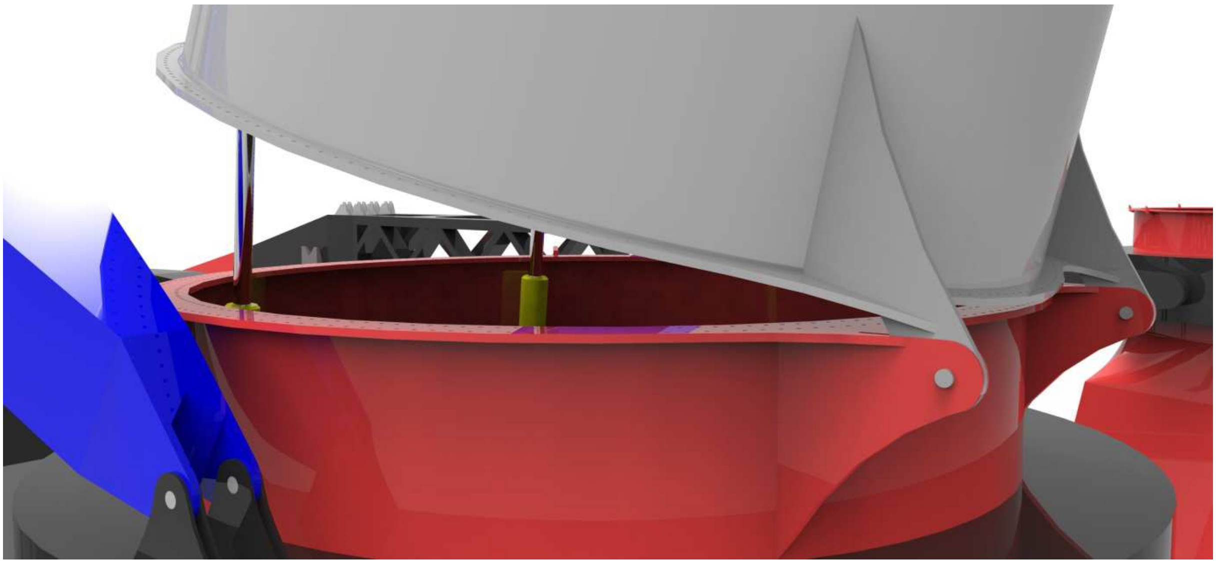


Figure 83: Hydraulic cylinders (yellow components)

Placing the hydraulic cylinders further from the hinge point of the overturning mast reduces the required cylinder diameter, but enlarges the required stroke. How large the required cylinder diameter is depends on the hydraulic pressure. In the case that a common maximum hydraulic pressure of 250bar is used and that two hydraulic cylinders are applied at the in the middle of the mast, then the minimum cylinder diameter is 560mm and the stroke 2400mm. Another option is to place the cylinders as far from the mast hinge point as possible. In that case the minimum cylinder diameter is reduced to 400mm, but the stroke is enlarged to 4800mm.

Alternatively, a hoisting system could be temporarily placed on, and secured to, the slew platform of the mast crane. The available space for the hoisting system is indicated in yellow in Figure 84. By lifting the mast bottom with the hoisting system, the overturning angle of 19˚ can be reached. The location at which the hoisting system needs to be placed on the slew platform is currently occupied by winches. The hoisting tackle has to be detached from the overturning mast, to prevent interference between the hoisting tackle and the overturning mast when the overturning angle of the mast becomes large.

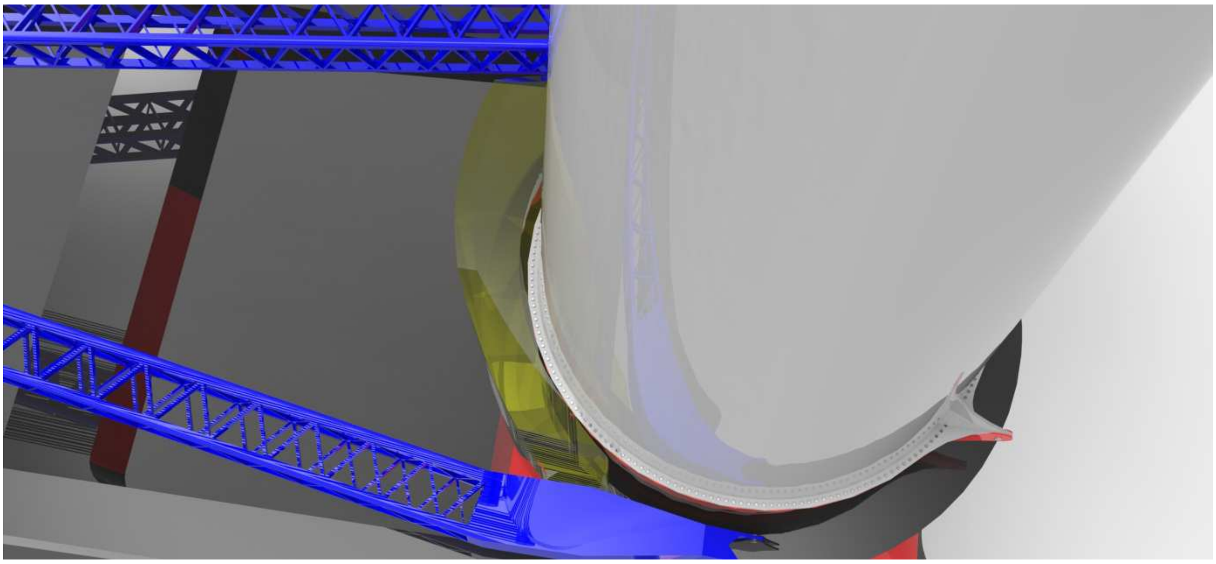


Figure 84: Available space for hoisting system (yellow)

The limited available space to place the hoisting system on the slew platform, but moreover the small space between the mast and the hoisting system requires large engineering effort if this concept has to be implemented. The hoisting system also has to be stored, whereas the hydraulic cylinders can remain present in the mast. Based on this, the advice is given to apply hydraulic cylinders, but further research to the technical and economic feasibility is necessary.

8.3 Relocating the sheave nests

For the back-mast concept two independent tackles are required to suspend the overturning mast and to adjust the back-mast angle when the mast is overturning. The angle of the back-mast has to change with the overturning mast to prevent the tackles from clashing with the mast bottom (Figure 85).

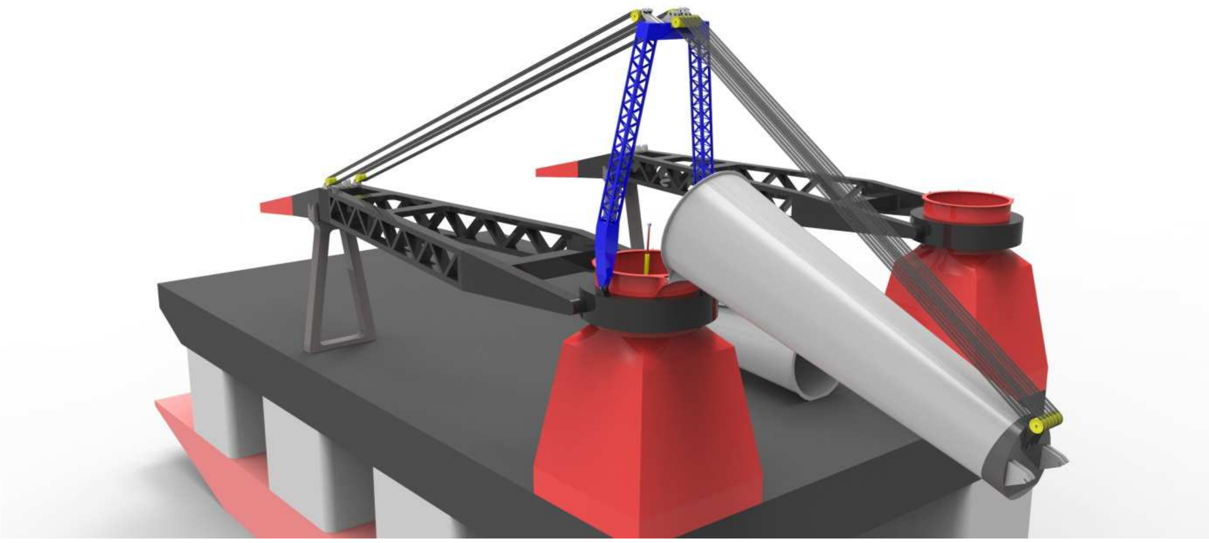


Figure 85: Back-mast concept with two tackles

One tackle has to be placed between the boom and the back-mast top and one tackle between the back-mast top and the mast head. To make the back-mast concept economically feasible, the original topping tackle of the mast crane and the original sheave nests (Figure 86) are reused and therefore have to be relocated.

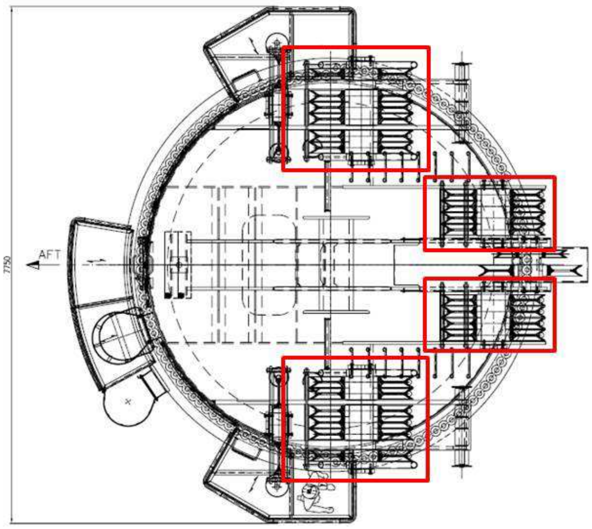


Figure 86: Top view of the Aegir's mast head with sheave nests

The A-frame crane on the Balder follows this approach when its boom is put in tie-back mode (Figure 87), as discussed in Section 5.2. After the sheave nests are relocated, in this case disconnected from the A-frame and secured to the deck, the tension in the tackle is reapplied.

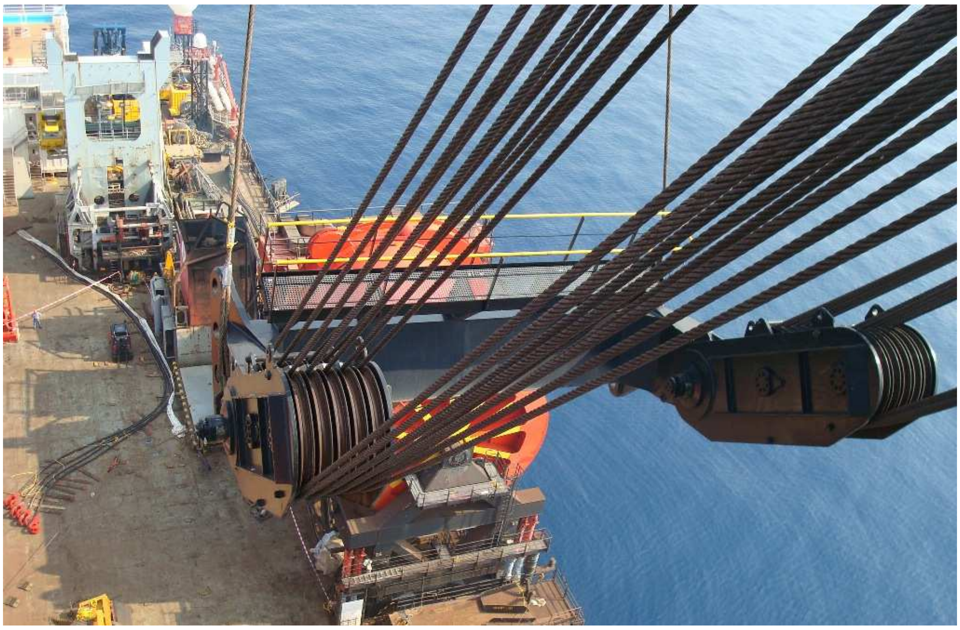


Figure 87: Lowering the sheave nests on the Balder

For the back-mast concept, the eight sheave nests on the 8,000mt mast crane have to be relocated. In the original crane configuration, four sheave nests are located on the mast head and four on the boom. For the back-mast concept, half of the sheave nests on the mast head have to be moved to the top of the back-mast, and half of the sheave nests on the boom to the top of the back-mast (Figure 88).

Relocating the sheave nests has to make sure that two independent tackles are formed. Therefore, the tackles between the boom and the back-mast top and between the back-mast top and the mast head have to be placed on separate winches. The original topping tackle of an 8,000mt mast crane is able to suspend the weight of the boom (1,400mt) and the maximum load. Tackle loading caused by mast overturning is relatively low, as is shown in Section 8.6.

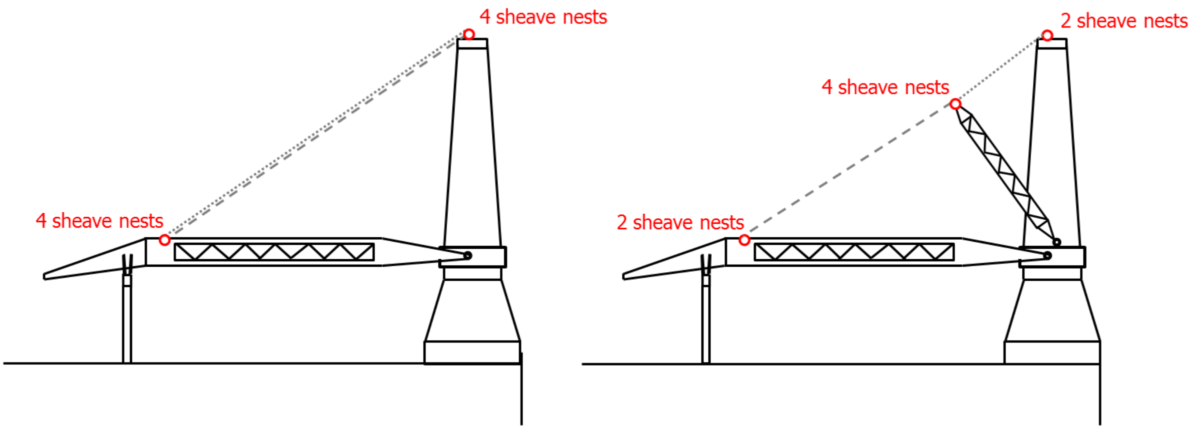


Figure 88: Relocating the sheave nests (left the original crane configuration and right the back-mast concept with relocated sheave nests)

Another attention point of the back-mast concept is that support is required for the wire ropes running in the mast and over the back-mast for the tackles when overturning the mast. Additional sheaves have to be placed in the mast and on the back-mast top. The main, aux and whip hoist wire ropes that are present in the mast, but not running in the mast, only need support.

The remaining sheave nests at the center of the mast head have to be moved more to the side of the mast head than their current locations (Figure 86). By moving these sheave nests, it is prevented that interferes with the mast head when the mast is overturning. The distance the sheave nests have to be moved is small. Therefore, is assumed this can be adjusted in the mast crane design. In this case, these sheave nests do not have to be relocated.

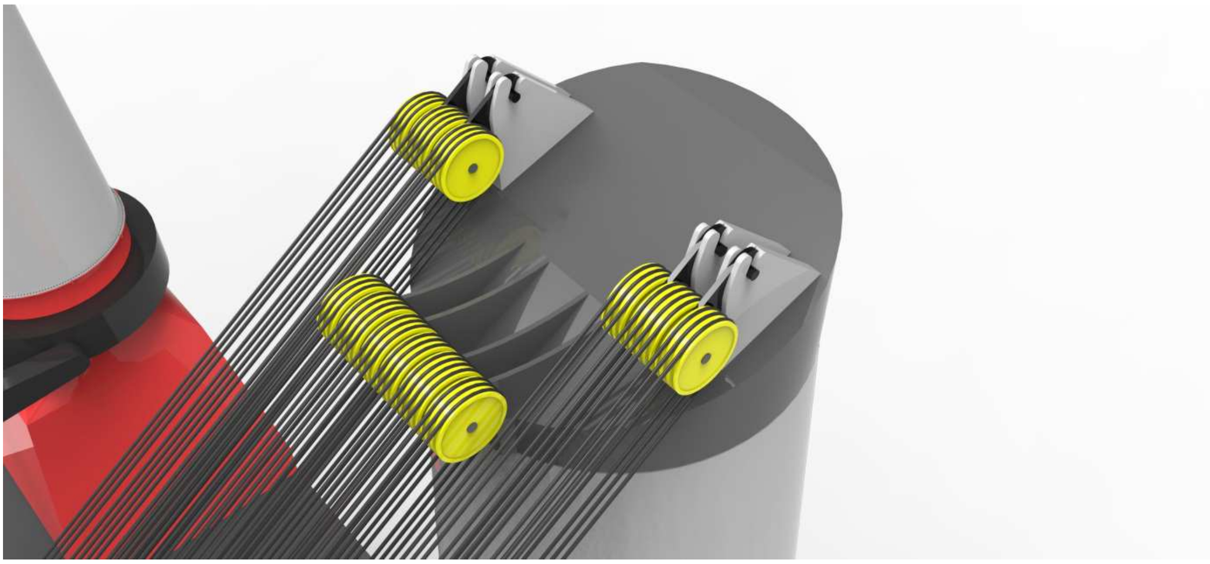


Figure 89: Sheave nests on the mast head

Relocating half of the boom sheave nests to the top of the back-mast can be carried out by a small crane on the NSCV with a lifting capacity of 35mt (the same as the Kobelco crane on the Thialf). The load curve of this crane is suitable because the height of the nests is small and the weight of a single sheave nest with wire ropes is approximately 30mt. Because the sheave nests are located at 80m above the deck, this crane cannot be used for relocating the mast head sheave nests to the top of the back-mast due to a lack of available lifting height. On the mast head of the Aegir crane a service crane is installed. Service cranes will also be installed on the NSCV, capable to relocate the sheave nests on the mast head.

The changing angles of the back-mast and the overturning mast section could cause interference between the sheave nests and the back-mast top. Therefore, the steel structure where the sheave nests are secured to, on the top of the back-mast, is designed as shown in Figure 90 (grey components).

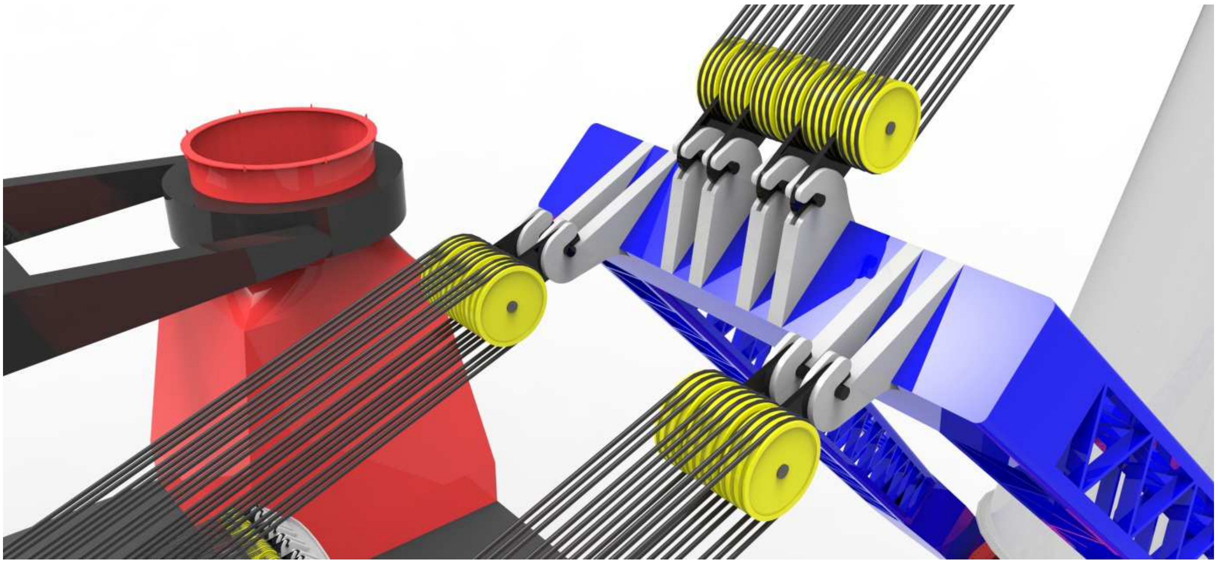


Figure 90: Sheave nests on the top of the back-mast

8.4 Mast overturning characteristics

In the back-mast concept the boom acts as a counterweight for the overturning mast. Therefore, the moment around the mast hinge point, that is induced by the weight of the overturning mast, must be smaller than the moment available by the weight of the boom. Otherwise, the boom is luffed when the mast is overturned. The moment induced by the hinged mast depends on its overturning angle and is shown in Figure 91.

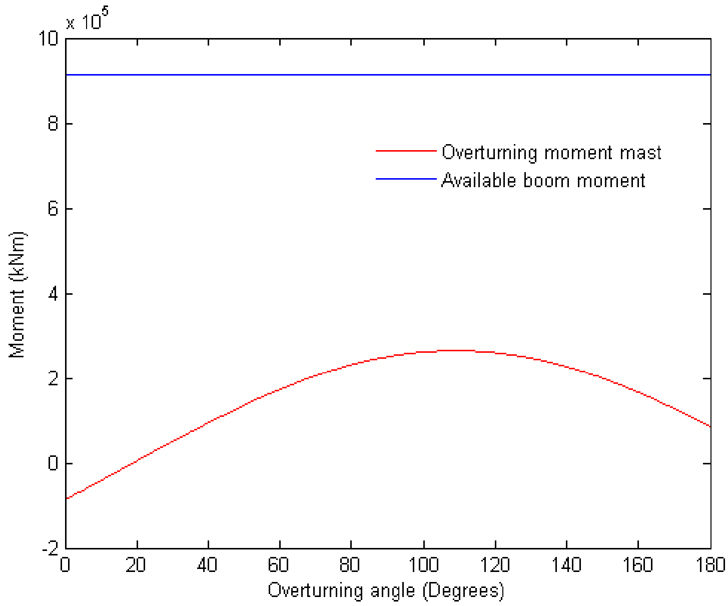


Figure 91: Overturning moments

The moment is maximum (2.60E5kNm) when the HCG of the overturning mast is at maximum distance from the mast hinge point, as is shown in Figure 92. This maximum value is significantly smaller than the moment needed to hold the overturning mast (9.13E5kNm). Therefore, accidental boom luffing by mast is prevented.

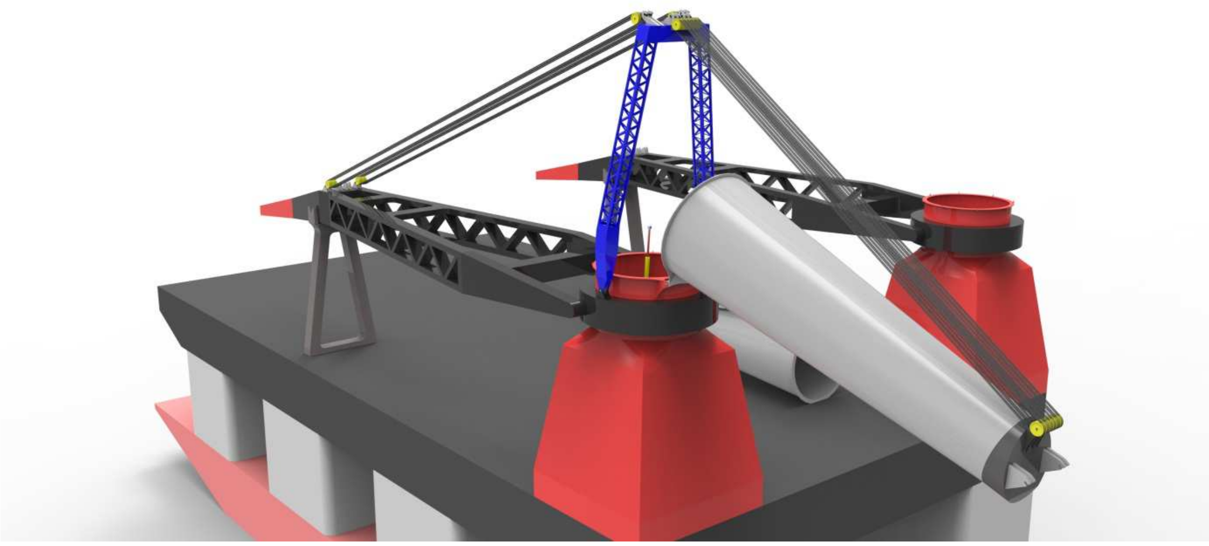


Figure 92: Maximum overturning moment around the mast hinge point

The overturning mast is held by the tackles and the back-mast. The back-mast angle is important for preventing tackle interference with the bottom of the overturning mast and for minimizing the compression acting on the back-mast. By optimizing the back-mast angle for a certain overturning angle of the mast, the back-mast compression is minimized. Minimizing the back-mast compression is done for two design cases:

1. At the maximum overturning moment around the mast hinge point;
2. At the maximum overturning angle (mast fully overturned).

The optimal back-mast angle for the maximum overturning moment is 84° and for the maximum overturning angle 48°. Another parameter that can be optimized is the back-mast length. For the two optimal back-mast angles, is the back-mast length varied between 15m and 55m. By varying the back-mast length is assessed what the effects are on the acting back-mast compression, as shown in Figure 93 and Figure 94.

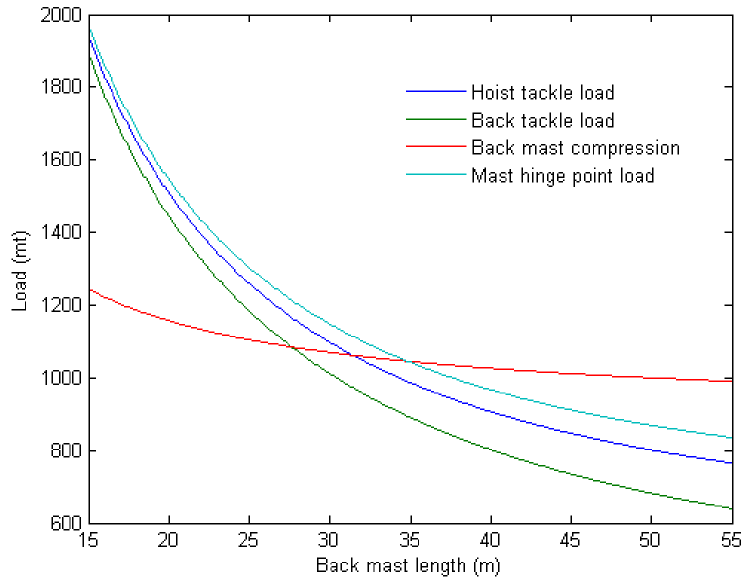


Figure 93: Loads at maximum overturning moment (back-mast angle of 84˚)

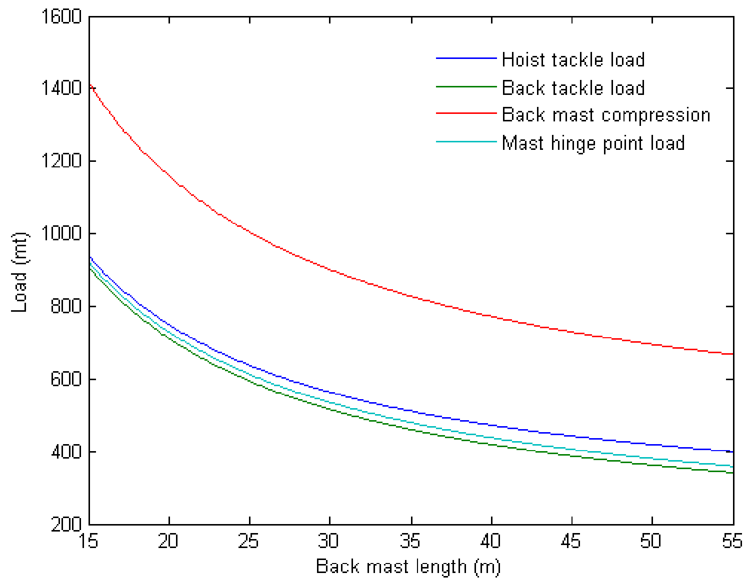


Figure 94: Loads at maximum overturning angle (back-mast angle of 48˚)

Above figures show that increasing the back-mast length reduces back-mast compression. For the maximum overturning angle case, the back-mast compression decreases more rapidly when increasing the back-mast length than for the maximum overturning angle case. An optimum is not evident.

The minimum back-mast length, necessary to prevent the tackle between the mast head and back-mast top from interfering with the mast bottom when the mast is fully overturned is considered as optimal. This outreach is 22m, determined by the dimensions of the mast bottom, the location of the back-mast hinge point and the clearance required between the tackle and the mast bottom. For this case the back-mast length is calculated as 33m.

The load transferred through the back-mast and its hinge points is transferred through the slew bearing of the mast crane. An additional moment and resulting bearing loading in the slew bearing has to be prevented. Therefore, the back-mast hinge point has to be located such that at a back-mast angle of 84˚, when maximum back-mast loading is acting, the back-mast line intersects the slew bearing's centerline, shown in Figure 95.

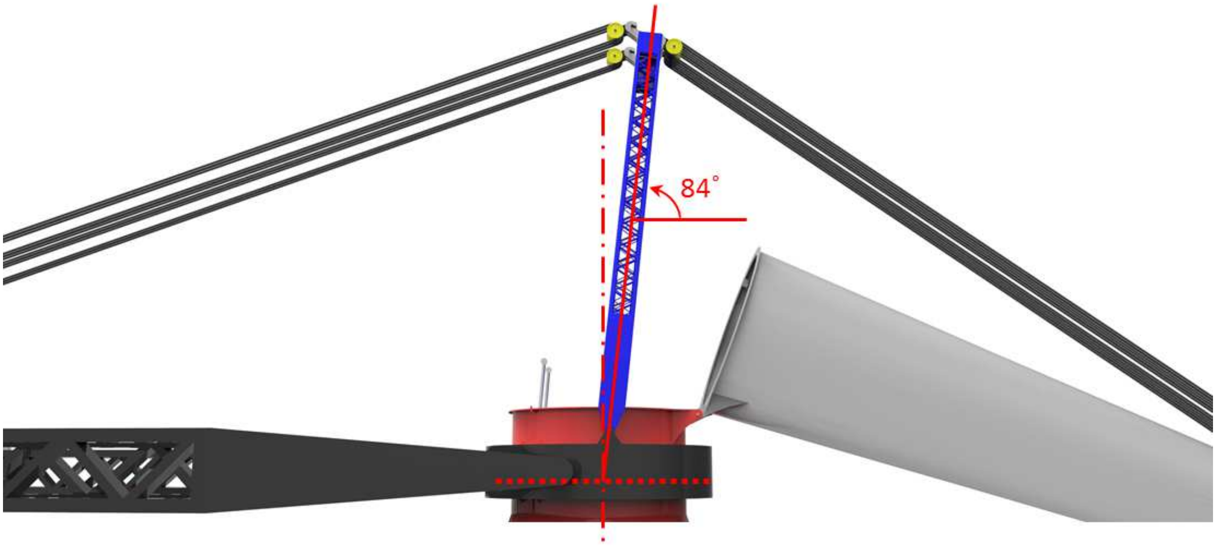


Figure 95: Optimal back-mast angle under maximum compression

8.5 Hinge points

For the design of the back-mast and of the mast hinge points are the maximum loads on the main components calculated. The loads depend on the overturning angle of the mast (α) and the angle of the back-mast (β). The optimal back-mast angle is determined for two cases at 84˚ and 48˚. Another important case is when actual overturning of the mast starts (α is 19˚ as discussed in Section 8.2). At this point the loading on the tackles and on the back-mast by the weight of the overturning mast is started.

For this mast angle, the optimal back-mast angle is achieved when the tackles between the boom and the mast head are as in line with each other. This is achieved for a back-mast angle of 120˚. Between the three back-mast angles is a linearly interpolation applied so that all angles are known for the whole overturning process of the mast (Figure 97). Herewith, the loads on the main components of the concept are determined.

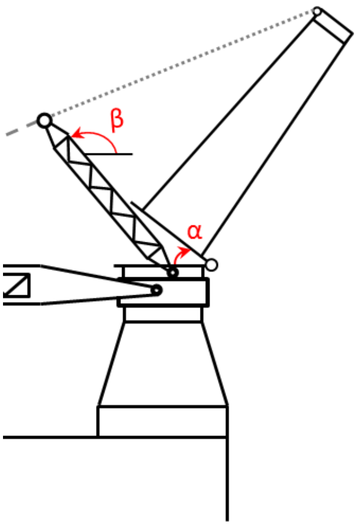


Figure 96: Overturning angle (α) and back-mast angle (β)

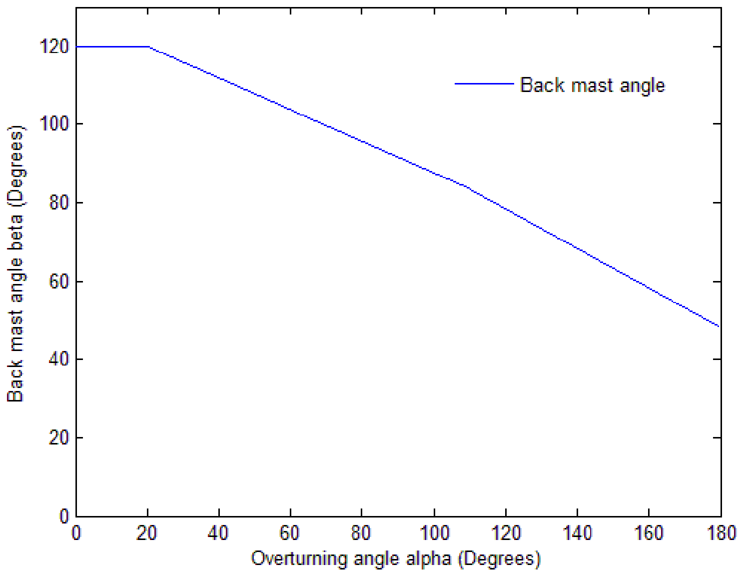


Figure 97: Back-mast angle (β)

The load on the mast hinge point, when the mast is overturning, is a combination of the tackle pulling on the mast head and of the weight of the overturning mast. In the calculations, the weight of the tackles is neglected. The mast hinge point has to be located towards the tail of the crane to prevent the overturning mast from interfering with the slew platform (Figure 98). The width of the slew platform at the tail of the crane is 2.1m.

Thus the centerline of the mast hinge point has to be located approximately 1.2m towards the tail of the crane to prevent interference. Then a clearance of 300mm is obtained between the slew platform and the overturning mast when the mast is fully overturned. Two mast hinge points are applied to effectively handle eccentric loading. No crane components interfere at the tail of the crane and sufficient space is available to accommodate the two hinge points, shown in Figure 98.

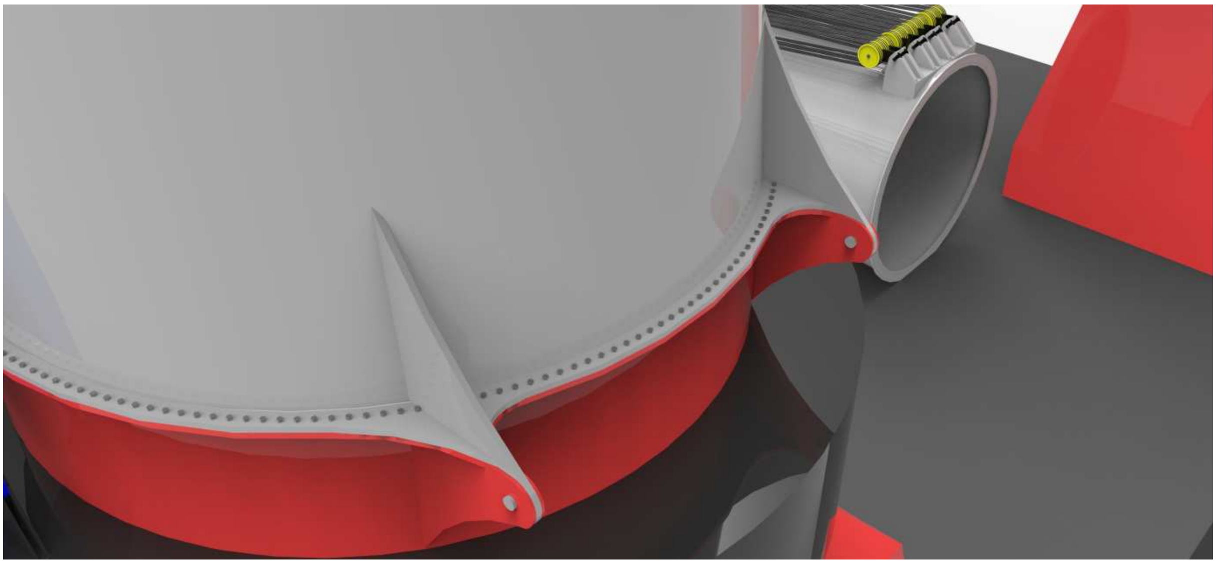


Figure 98: Mast hinge points

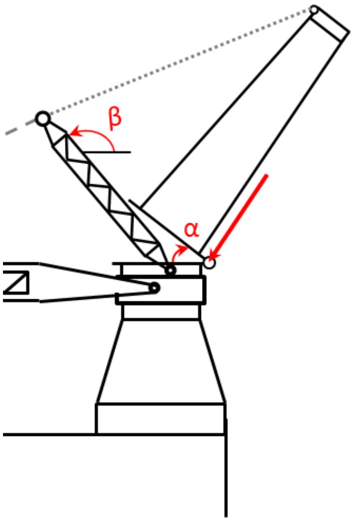


Figure 99: Mast hinge point load scheme

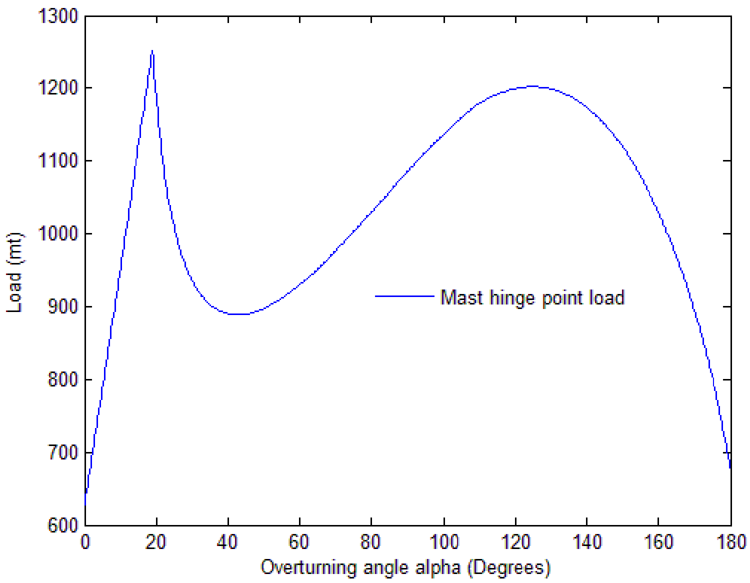


Figure 100: Mast hinge point load

In the first part of the graph (α from 0˚ to 19˚) the load on the mast hinge point increases from half the mast weight to the full mast weight (1,250mt). Latter is the case when the HCG of the overturning mast is above the mast hinge point. This is the maximum loading on the mast hinge point in the total overturning process of the mast. Beyond a mast overturning angle of 19˚ the overturning moment and the tackle load increase. The load on the mast hinge point decreases to half the mast weight when the mast is fully overturned.

When the mast is supported by the flange, the load acting on the mast hinge point has to be zero to prevent permanent loading of the hinge point. Therefore, no standard circular shaped hinge point can be applied. A solution is an oval shaped hinge point. Since an oval shaped hinge point causes a less optimal load transfer, it is assumed in the calculations that the loads on the hinge point increase 30%.

The required dimensions of the mast hinge point (Figure 98) are calculated by applying the Bleich method (Appendix E). The maximum yield stress of the steel used for the hinge points is 345MPa. Assuming the load is equally divided over the two hinge points and a safety factor of 1.1 is applied to include vessel and hoist accelerations, the force per hinge point is 8.80MN. To keep the stresses in the mast hinge point below the allowable values, a combination of the various hinge point parameters is determined for each hinge point of the overturning mast, shown in Figure 105 and Figure 101.

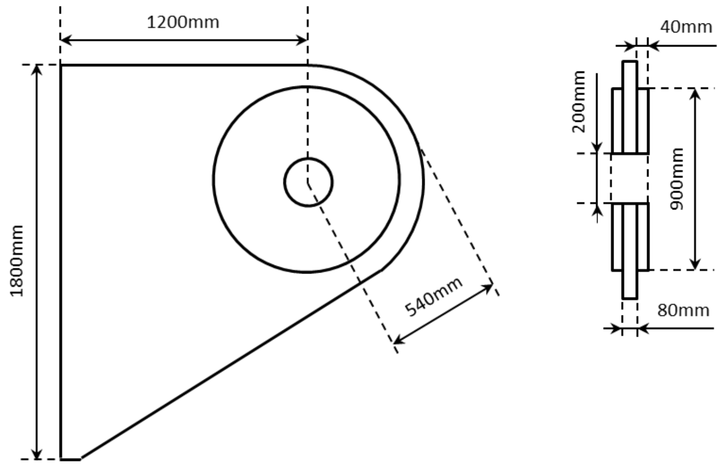


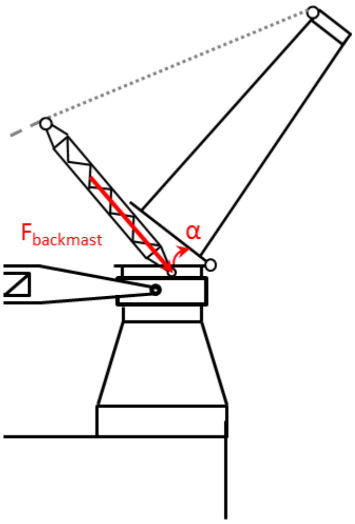
Figure 101: Mast hinge point (2x)

The critical stresses occur on the inside and outside of the eyes. They are calculated by combining the average stress with the bending moment. The maximum values are 66% the yield strength. Also the contact pressure between the surfaces has to stay below the maximum value of 90% the yield strength of the material (310.5MPa). The length of the joint (1,800mm) between the hinge point and the mast is calculated with the forces caused by the bending moment and the shear.

It is assumed that the flange on the lower mast section (of the bolted flange joint) provides sufficient stiffness at the top of the hinge point in preventing mast deflection when the mast is overturned. This is valid since the load transfer from the boom hinge point to the pedestal of the crane is similar and also in this area. However, a detailed assessment is needed to validate this presumption.

The next step is to design the hinge points of the back-mast by the Bleich method. The total external load on the back-mast is formed by the compression of the two tackles and has to be transferred to the slew platform of the crane. For the back-mast hinge point design, the load acting on the back-mast and the load caused by the own weight (100mt) of the back-mast are summed, shown in Figure 104.

Figure 102: Back-mast hinge points



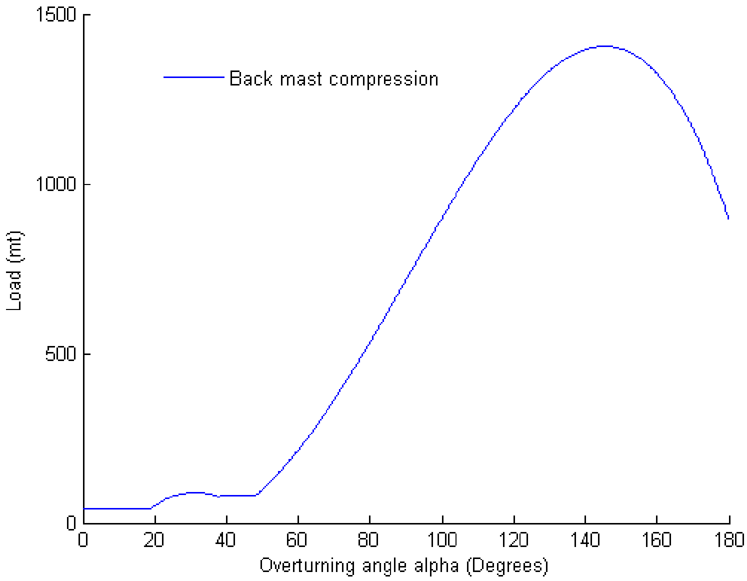


Figure 103: Back-mast hinge point load scheme

Figure 104: Back-mast hinge point load

The maximum back-mast compression that is induced by the tackles is 1,405mt. As described in Section 8.0, the hinge points of the back-mast are located opposite from each other on the slew platform of the crane. It is assumed the loads are equally divided over the back-mast hinge points. For this basic design are the wind loads neglected. Taking a safety factor of 1.1 into account, the maximum back-mast hinge point force is calculated at 7.7MN. The same steel is used for the back-mast as for the mast hinge point. To keep the stresses below the maximum values a combination of hinge point design parameters is determined and shown in Figure 105.

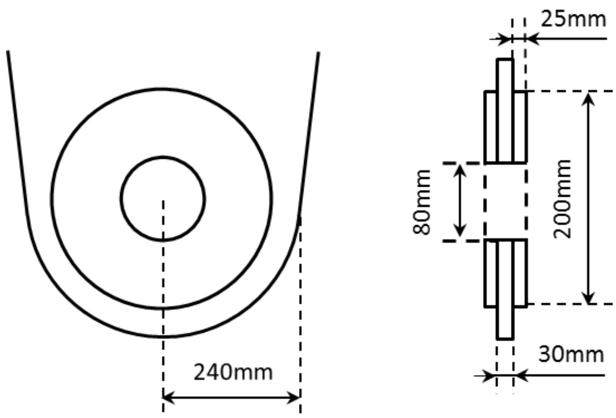


Figure 105: Back-mast hinge point (4x)

8.6 Tackles

So far it is assumed that the load carrying capacity of the tackles is sufficient after they have been relocated. In this section is calculated what the actual loading is in the tackles. For these calculations are the back-mast characteristics as shown in the previous sections used.

The original topping tackle of an 8,000mt mast crane consists of 40 falls. A fall is a wire rope loop, and each fall has in this case a load carrying capacity of 250mt. Thus the maximum load in the topping tackle of the mast crane in its original configuration is 10,000mt. For the back-mast concept the tackle is split by relocating the sheave nests. In this case, the maximum load carrying capacity of the tackles between the boom and the top of the back-mast, and of the tackle between the mast head and the top of the back-mast is 5,000mt. This is sufficient when comparing this value with the actual loading in the tackles, as shown in Figure 107.

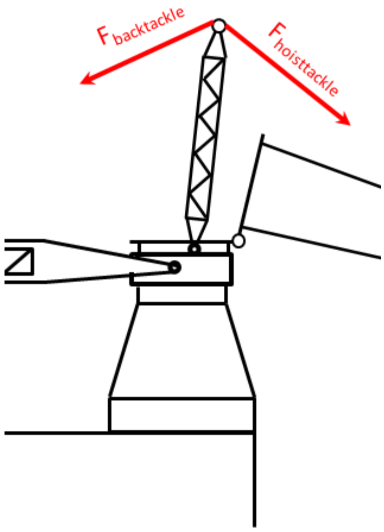


Figure 106: Tackle load scheme

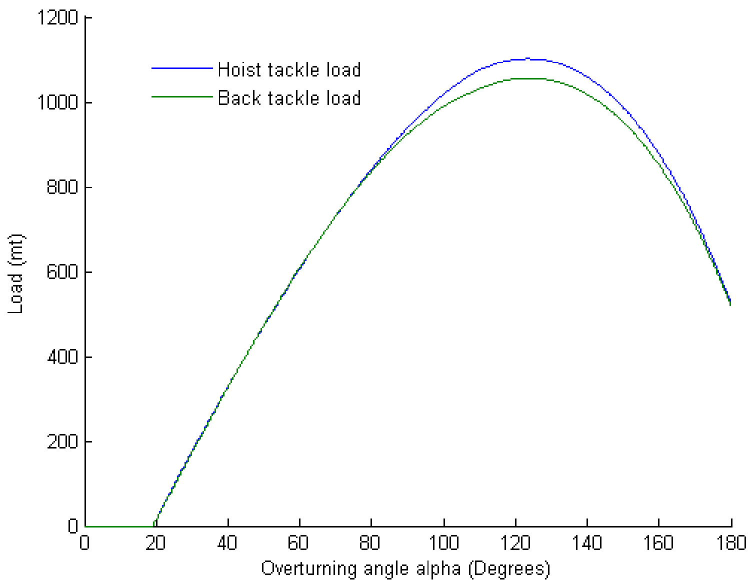


Figure 107: Tackle loads

The load in the tackles is assumed to be zero until the HCG of the overturning mast passes the mast hinge point. However, in reality a load will be present in the tackles due to their own weight and the weight of the back-mast. Determining this load is complicated and irrelevant in this part of the graph. Once the HCG of the overturning mast passes the mast hinge point, this load is added to the tackle loads by the weight of the overturning mast.

The maximum hoist tackle and back tackle loads are respectively 1,102mt and 1,056mt. These loads are only a quarter of the maximum load carrying capacities of the tackles. Therefore, no further calculations are carried out and is it assumed that the tackles have sufficient load carrying capacity to be used for the back-mast concept.

8.7 Back-mast design

The back-mast consists of two laced compression members and is designed for a maximum compression of 1,405mt (Section 8.5). The laced compression members (columns) are constructed from four chords. In between the chords lacing is applied on all column planes. This section shows the basic design calculations of the back-mast.

The total length of the back-mast, from the hinge points to the sheave nests is 33m (Section 8.4). The width of the back-mast bottom is 15m, determined by the slew platform dimensions. The width of the back-mast top is 8m to accommodate sufficient space for the sheave nests, without compromising the off-lead angle of the wire ropes. Thus the basic design of the back-mast is tapered. When the angle, under which the columns are placed, is taken into account, the force on a single column is 7,669kN. The design force for a column with four identical chords is calculated by Eurocode 3:

$$N_{ch,Ed} = 0.25N_{Ed} + \frac{M_{Ed} * h_0 * A_{ch}}{4I_{eff}} \tag{21}$$

$N_{Ed} = design\ value\ of\ the\ compression\ force\ to\ the\ built\ up\ member$

$M_{Ed} = design\ value\ of\ the\ maximum\ moment\ in\ the\ middle\ of\ the\ built\ up\ member$
$\qquad - considering\ second\ order\ effects$

$h_0 = distance\ between\ the\ centroids\ of\ the\ chords$

$A_{ch} = cross\ sectional\ area\ of\ one\ chord$

$I_{eff} = effective\ second\ moment\ of\ inertia\ of\ the\ built\ up\ member$

The design force for a column is calculated as 2,304kN and the maximum allowable material stress 207.0MPa. The complete calculation is available in Appendix F. The required cross sectional area per tube is calculated as 111.3cm^2. Also the laces between the chords are basically designed. The shear forces acting in the laces between the chords are given by:

$$V_{Ed} = \pi * \frac{M_{Ed}}{L} \tag{22}$$

The maximum shear forces for a moment off the centerline of the back-mast (M_{Ed}) of 3,123kNm and a chord length (L) of 31.5m is 320.4kN.

Also an analysis is done to the following three buckling modes (Figure 108):

1. Failure of the column as a whole;
2. Failure of one of the main component segments;
3. Failure of lacing between the main components.

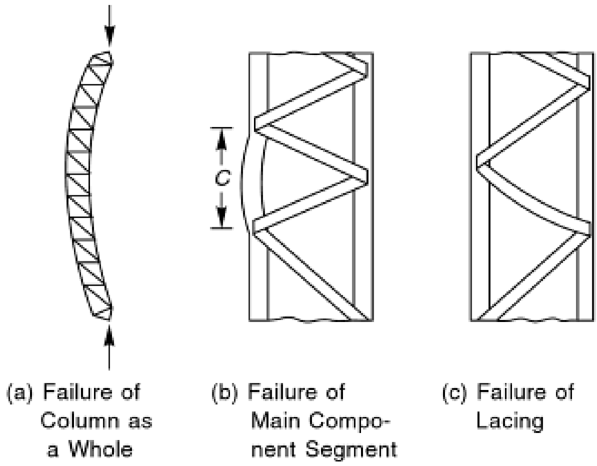


Figure 108: Failure modes of a latticed column (Shukla 2009)

For these calculations an angle of inclination (θ) of the lacing bars with the longitudinal axis of the component member is taken as 60° without further optimization (should be kept between 40° and 70°). The angle of inclination is a trade-off between required lacing dimensions and lacing buckling.

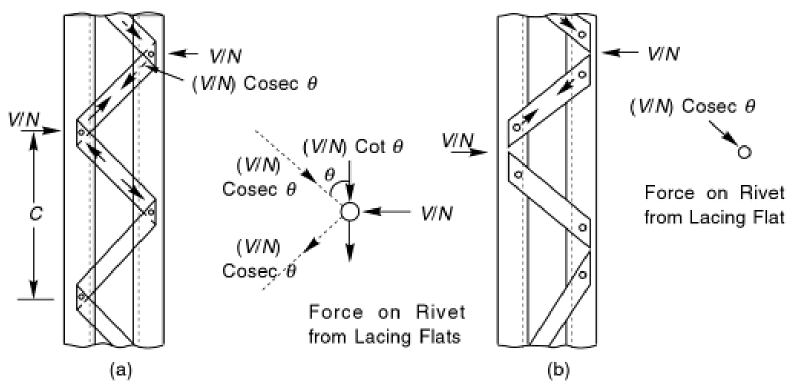


Figure 109: Force in lacing (Shukla 2009)

The columns consist of four tubes, each with a diameter of 300mm and a wall thickness of 5mm. For a single lacing system, in opposite faces of the built-up member with two parallel laced planes, the lacing should be arranged so that one is the shadow of the other. The number of lacings on a single side of the column is 31. The basic design of the back-mast, obtained in this section, is viable to show the technical feasibility of the back-mast concept, but further research to the back-mast design is required.

8.8 Back-mast installation and usage

The first step needed to start installation of the back-mast is transit of the NSCV to allowable sea conditions. These sea conditions are not yet defined and have to be determined by further research. Then the back-mast is installed on the slew platform of the crane. Installation is done with the service cranes of the NSCV. These cranes are also used for relocating the sheave nests (Section 8.3). When HMC decides to implement the back-mast concept on the NSCV, it must be ensured that the service cranes on the NSCV are able to place the back-mast. During installation, the hinge points of the back-mast are pinned to those on the slew platform, and the back-mast is placed on the boom of the mast crane (Figure 110).

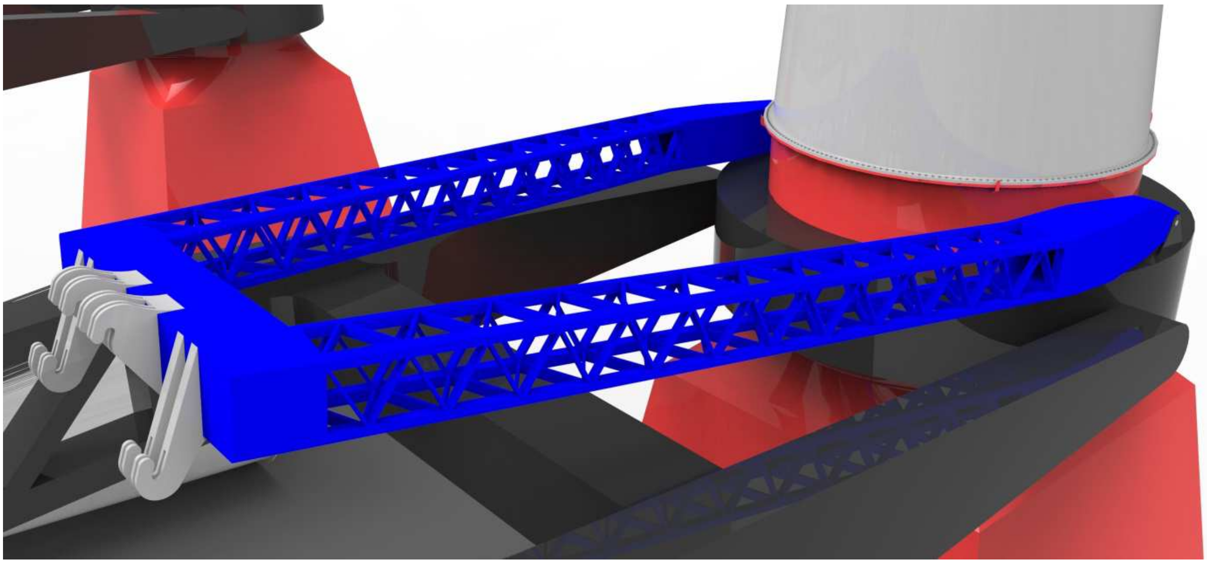


Figure 110: Back-mast installed and placed on the boom

Once the back-mast is installed, are the sheave nests relocated by the service cranes on the NSCV. The two center sheave nests on the mast head are moved to the center of the back-mast. The two sheave nests on the sides of the boom are moved to the sides of the back-mast. After the sheave nests have been relocated, are the sheave nests secured to the back-mast top (Figure 111).

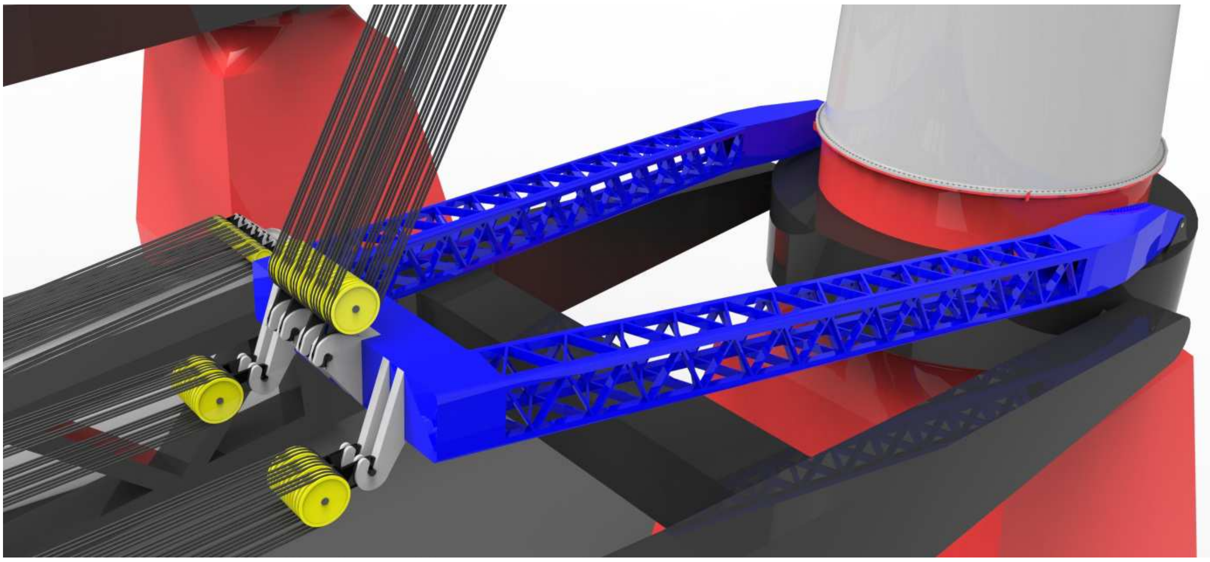


Figure 111: Back-mast installed and the sheave nests relocated

After securing the sheave nests, the back-mast can be hoisted to its original position (120˚). The angle of the back-mast is controlled by elongating and shortening the tackles. Overturning of the mast can be started once the bolts are removed between the mast sections and the hydraulic cylinders expand.

The completely overturned mast needs support at the mast head by a frame that is placed on the hull of the semi-sub. This frame has to be covered with wooden fenders to prevent damage to the mast and the semi-sub. To prevent slamming between the overturned mast and the support frame, is it advised to implement a securing system between the two.

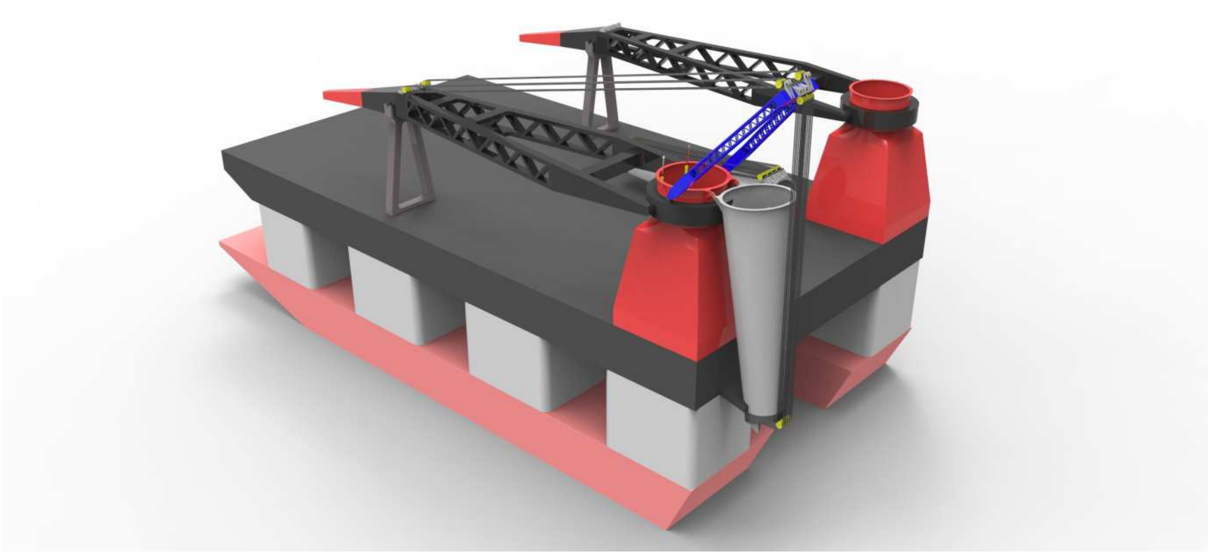


Figure 112: Mast fully overturned

When the mast is secured to the frame, it is not necessary to remove the back-mast in order to meet the passage restrictions. Lowering the back-mast, by elongating the tackle between the boom and back-mast top, until the back-mast is almost horizontal is sufficient, shown in Figure 112. It is important that the back-mast is not lowered to a complete horizontal position, since it is then no longer possible to erect the back-mast.

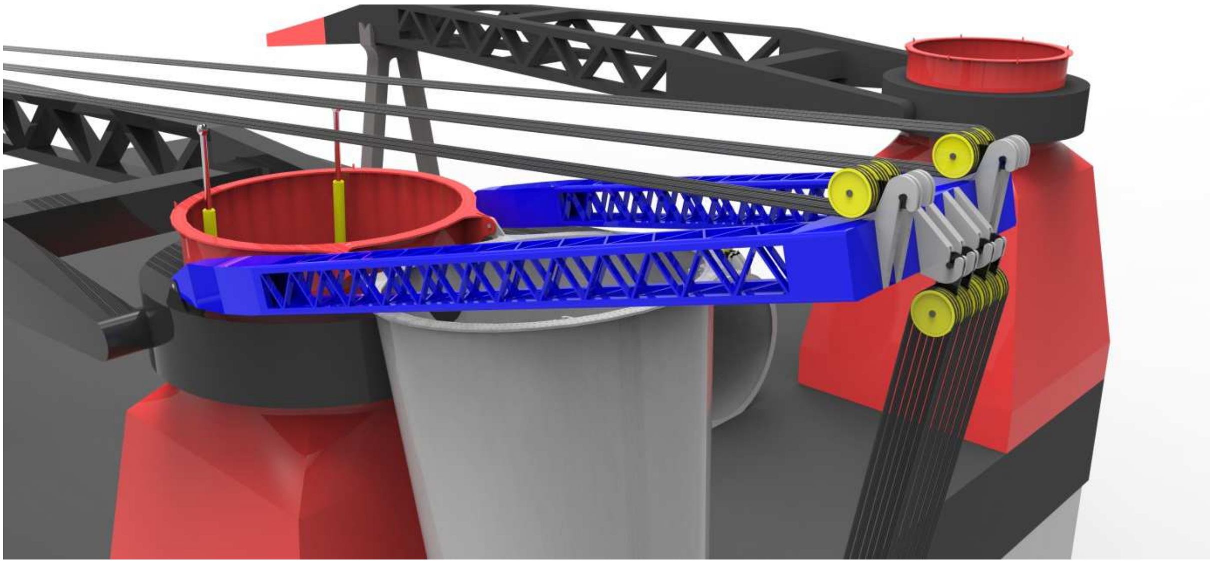


Figure 113: Back-mast lowered

Chapters 7 and 8 show that the required redesign of the mast crane is large for implementation of the back-mast concept. New components such as the back-mast, hydraulic cylinders and hinge points are needed. The required redesign is significantly less for implementation of the concept in which only the mast is placed on the deck of the NSCV (the right crane in Figure 113). Only a bolted flange joint and removable sheave nests have to be applied. Therefore, is implementation of the back-mast concept on only one of the two cranes on the NSCV, the most optimal choice. This choice is based on the required engineering effort and the involved cost and time. The time involved in reducing the remaining height of a mast crane versus an A-frame crane requires further research. Both methods are considered as time-consuming but very valuable when possible.

9.0 MULTI-CRITERIA ANALYSIS WITH BACK-MAST CONCEPT

Before a new multi-criteria analysis is shown with the back-mast concept, are the score differences of the multi-criteria analysis of Chapter 6 showed:

- A-frame crane with a roller slew bearing: 3.37
- A-frame crane with a bogie slew bearing: 3.30
- Mast crane (without the back-mast concept): 3.43

Above score differences are small. Therefore, it was difficult to provide HMC with a convincing advice which crane type is the most suitable for the NSCV.

In Chapter 8 a conceptual design analysis of the back-mast concept is conducted. The results show that a feasible concept is developed which is able to reduce the remaining height of a mast crane. Therefore, with respect to the air draft criteria, a mast crane becomes more competitive with an A-frame crane. Both cranes are now able to meet the restrictions of passages shown in Table 17.

In Table 21 the scores of the crane types are shown, but now for a mast crane with the back-mast concept implemented.

Table 21: Crane type rating with back-mast

		Scores				Weights*Scores			
Criteria	**Weight**	**A-frame** Roller	**A-frame** Bogie	**Mast**	**Slewing-mast**	**A-frame** Roller	**A-frame** Bogie	**Mast** Back-mast	**Slewing-mast**
Lifting capacity up to 8,000mt	0,26	5	4	3	1	1,28	1,02	0,77	0,26
Lifting capacity >8,000mt	0,06	4	4	2	2	0,25	0,25	0,12	0,12
Footprint	0,17	2	2	5	3	0,34	0,34	0,84	0,51
Boom hinge point location and load curve	0,14	1	3	5	3	0,14	0,43	0,71	0,43
Tail swing	0,05	1	1	5	2	0,05	0,05	0,23	0,09
Slewing range	0,02	5	5	4	5	0,10	0,10	0,08	0,10
Air draft	0,15	3	3	3	2	0,44	0,44	0,44	0,29
Vessel stability	0,05	5	5	3	3	0,23	0,23	0,14	0,14
Deep-water lifting and lowering	0,02	4	4	3	3	0,08	0,08	0,06	0,06
Design maturity and reliability	0,09	5	4	2	1	0,47	0,37	0,19	0,09
	1,00					3,37	3,30	3,58	2,09

The highest score in the table above is for a mast crane with the ability to lower its remaining height when the back-mast concept is implemented. This is mainly caused by the small footprint of a mast crane, the small minimum radius and the small tail swing compared to the other crane types. Now a mast crane is able to reduce its remaining height, which was seen as one of the main disadvantages compared to A-frame cranes, the mast crane scores highest in the MCA, shown in Table 21.

Although the advice, to implement the back-mast concept, is mainly based on its technical feasibility, a valuable new method is found to reduce the remaining height of a mast crane in a semi-sub. Further research is required to the economic feasibility of the concept.

CONCLUSION AND RECCOMMENDATIONS

The assessment of the offshore market prospects and trends indicate that growth is expected in the deep-water oil and gas production. For this market, the implementation of a heave compensator has to be taken into account in the crane design. The platform removal market is expected to grow up to 30% in the next decade of all offshore projects. For this market a large deck space and a large opening between the cranes of the NSCV is essential.

The research on the scaling and upgrading of currently available offshore cranes shows that A-frame cranes with a roller slew bearing will require the least amount of engineering effort to obtain the required lifting capacity of 8,000mt, while complying with the boundaries. Also A-frame cranes with a bogie slew bearing can be upgraded and scaled such that they meet the requirement of the lifting capacity. This will require more engineering effort. Mast cranes will require even more redesign effort because the mast diameter and corresponding slew bearing diameter have to be enlarged. By enlarging the slew bearing diameter, the tolerances between the gears and the slew bearing teeth become a critical parameter and have to be monitored closely, to guarantee bearing service life.

Besides the lifting capacity of 8,000mt and the growth potential beyond 8,000mt, the air draft and footprint of the cranes, the load curve characteristics and tail swing, and the design maturity are the most important trade-off criteria. The multi-criteria analysis shows that mast cranes score better than A-frame cranes, despite their design maturity is lower due to the engineering effort. The main advantages of mast cranes are their small footprint, the large available space between the cranes and the small tail swing. Mast cranes affect vessel stability more than A-frame cranes due to their lack of counterweight, but this is outweighed by its advantages.

Another main disadvantage of mast cranes is the air draft. For A-frame cranes it has already been demonstrated that their A-frame can be folded.

The conceptual design analysis demonstrates a concept that reduces the remaining height of a mast crane, installed on a semi-sub. This is realized by separating the mast in two sections, overturning the top section towards the tail of the crane by applying a back-mast.

In the back-mast concept a back-mast is installed on the slew platform of the crane. The bolted flange joint between the two mast sections consists of 406 M60bolts. The original topping tackle is reused by relocating the sheave nests. To minimize back-mast compression and prevent tackle interference with the bottom of the overturned mast, the back-mast angle is changed when overturning the mast. The back-mast is lowered when the NSCV passes the height restriction.

The conceptual design analysis shows that the back-mast concept is technically feasible. Further engineering is required on the following aspects due to their immaturity: The first one is the support of the wire ropes present in the overturning mast and the wire ropes running over the top of the back mast. The second one is the deformation and stresses in the mast and in the slew bearing when the mast is overturned.

To meet the required dual lifting capacity of 16,000mt, HMC is advised to install two mast cranes on the NSCV, in combination with a back-mast to reduce the remaining height of the cranes. On the first mast crane on the NSCV only a bolted flange joint between the two mast sections has to be applied and the possibility to relocate the sheave nests has to be enabled. The mast of this crane is lifted by the second mast crane and placed on the deck. For the second mast crane also the back-mast concept needs to be applied because the remaining height has to be reduced by its own resources and the other crane can no longer assist.

This advice is mainly technically based. Obviously the development lead time and the cost of the concept are the decisive factors and require further research.

APPENDIX A: UNSUITABLE CRANE TYPES

A: Sheerlegs

A sheerleg is not able to rotate relative to the vessel it is installed on. Therefore, the required crane functionality as described in Chapter 2 is not obtained, causing this crane type to be unsuitable for the NSCV.

Three sheerleg types are common, distinguished by the method used to transfer the load moment to the back of the vessel it is installed on. The first method is a back-mast (Figure 114), the second method a back frame (Figure 115) and a third (less common option) is securing the boom directly to pad eyes integrated in the structure to the vessel.

Figure 114: Sheerleg with back-mast

Figure 115: Sheerleg with back frame (Diytrade)

On sheerlegs no counterweight is applied to compensate the weight of the boom, blocks and wire ropes. However, some compensation is obtained by placing the drives and winches at the stern of the vessel. Sheerlegs are generally unable to lift loads of the own deck of the vessel they are installed on.

Sheerlegs are used for lifting operations in narrow, shallow and calm water due to the limited draft of the vessel. Ship salvage, installing cranes on quays, installing civil structure, loading and unloading of large cargo into ships and bridge building are common lifting operations.

The three sheerlegs with the largest maximum lift capacity of a single crane are:

1. Asian Hercules III: 5,000mt (being built)
2. HL 5000: 5,000mt
3. Zhenhu 7: 4,000mt

The maximum lifting capacities are found on the Asian Hercules III and HL 5000 (Figure 116) are 5,000mt.

Figure 116: HL 5000

B: Ringer multi-cranes

A ringer multi-crane is a mobile offshore crane and has the possibility to change its boom configuration and can be put in multiple lifting modes: pedestal, fixed and ringer. That is why this crane type is called a ringer multi-crane (Figure 117).

Figure 117: Ringer multi-crane with a long single boom (left), a heavy duty boom (center) and a ringer attachment (right) (Zwagerman)

A ringer multi-crane consists of an A-frame and a boom mounted to a slew platform. The boom sections can be installed as a long single boom or as a heavy duty double boom. For both boom configurations the A-frame can be put in fixed position. Another option is apply the ringer attachment (counterweights traveling on rails). Changing the boom configuration is a time consuming process.

The three lifting modes of the crane are the standard pedestal slewing mode, the fixed mode and applying the ringer attachment. When the crane is in the pedestal slewing mode, it can handle the smallest load moment compared to the other modes. The load moment can be increased by applying the ringer arrangement. In fixed position the top of the A-frame is secured by wire ropes to the deck of the vessel. In this configuration the crane can handle the largest load moment, but loses its slewing capabilities. In the pedestal slewing mode the slew bearing has to transfer the load moment. With the ringer attachment applied and when the crane is in the fixed mode, the load moment on the slew bearing is decreased.

The tail swing of this crane type is large due to the large slew platform. The weight on the back of the slew platform reduces the load moment on the slew bearing. However, the permanent vertical load on the slew bearing is increased due to the extra weight of the drives and winches. The foundation of this crane type can be placed without having to make adjustments on the vessel. A related disadvantage is that the footprint of the crane type is very large. This makes ringer multi-cranes unsuitable for the NSCV.

The maximum revolving lifting capacity of a ringer multi-crane in pedestal slewing mode is 1,000mt. In fixed position this is 1,600mt at 40m and when the ringer attachment is applied it is 1,600mt at 22m. This value does not come close to the required 8,000mt for the NSCV. Mammoet has built a similar crane type, which is not designed as offshore crane, with a maximum lifting capacity is 3,200mt.

C: Kingpost cranes

Typical about a kingpost crane, as the name suggests, is the kingpost. The kingpost is integrated in the vessel structure. A similar crane type, not discussed in this report, is the Crane Around the Leg (CAL) typically found on a jack-up platform.

Figure 118: Kingpost crane (Seatrax)

A kingpost consists of a boom that can be slewed around a stationary kingpost. The overturning moment has to be transferred from the revolving structure of the crane to its king post.

Kingpost cranes have a small footprint because the overturning moment is resolved by means of two vertically-spaced radial bearing assemblies around the kingpost. The limited footprint and tail swing makes this crane type suitable in situations where limited space is available. All vertical loads are carried into the stationary kingpost by means of a concentric thrust bearing (Figure 119).

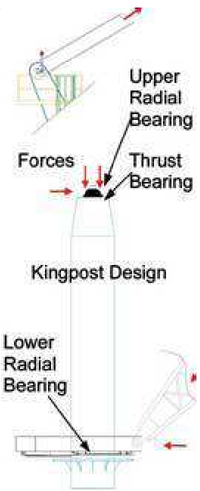


Figure 119: Loads on kingpost

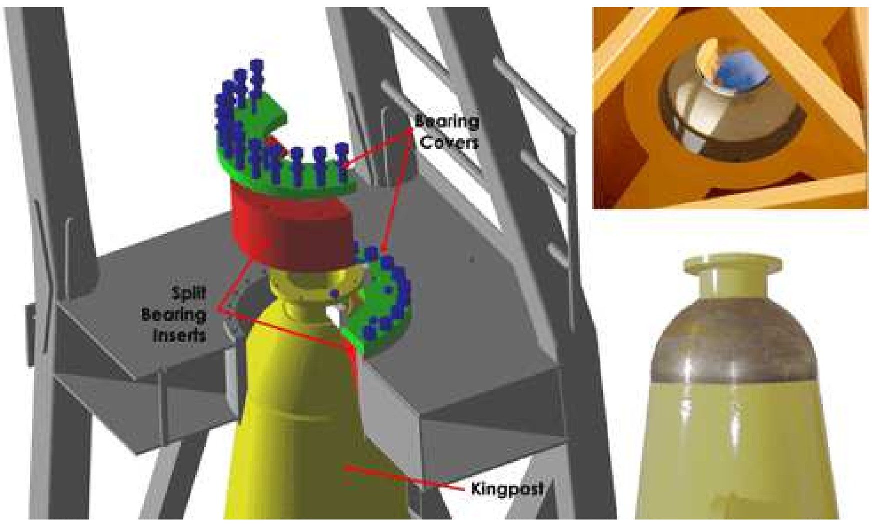


Figure 120: Thrust and upper radial bearing (Seatrax)

On kingpost cranes no counterweight is applied, resulting in a small tail swing. Kingpost cranes are often compared with mast cranes because the kingpost can be seen as the mast of a mast crane. In a kingpost crane a secondary frame is placed around the kingpost causing high material usage and large weight of the crane. Scale enlargement of the thrust bearing is difficult due to tolerance and deformation issues, limiting the maximum lifting of this crane type and causing this crane type to be unsuitable for the NSCV.

The kingpost cranes are most commonly used for small lifting operations on vessels and oil and gas platforms. The maximum lifting capacity is 150mt (Offshore-technology). This value does not come close to the required 8,000mt for the NSCV.

D: Fixed boom cranes

Typical about a fixed boom crane is the single box girder of the boom, luffed by hydraulic cylinders. Because the fixed boom is luffed by hydraulic cylinders there are no winches required for boom luffing, but only for hoisting.

Figure 121: Fixed boom crane

The boom of a fixed boom crane can be rotated relative to the vessel. The crane design results in a low CoG and a low remaining height. However due to the usage of hydraulic cylinders the shear forces in this crane type are relatively large, resulting in a relatively heavy boom. The long boom length causes the minimum radius to be large. The maximum allowable hydraulic pressure and the large shear forces in the boom causes this crane type to be infeasible for the NSCV.

The simple design keeps maintenance and crane cost low. On a fixed boom crane in general no counterweight is applied. The footprint and tail swing of this crane type are small.

The fixed boom crane is suitable for external lifts over the side of the vessel and internal lifts on the deck. The reliable and sturdy design make it especially suitable for rough load handling. However, this type of crane is not suitable if frequently the radii of the boom has to be changed during a lifting operation while keeping the load at the same height above deck. The simplicity of the cranes offer a cost-effective alternative to knuckle boom crane, which will be discussed in the following section.

The maximum lifting capacity of a fixed boom crane is about 350mt . This value does not lie close the required 8,000mt for the NSCV.

E: Knuckle boom cranes

The typical knuckle shape of the boom of this crane type has given its name. The boom of a Knuckle Boom Crane (KBC) consists of two parts.

Figure 122: Knuckle boom crane with drum at the tail (Huisman)

A KBC's boom can be rotated relative to the vessel and is hinged around the middle. Hydraulic cylinders control the position of the boom sections (main boom and knuckle boom). A boom in two sections is beneficial because the pendulum effects, which can be induced on a suspended load by vessel motions can be substantially reduced because the boom tip can be kept very close to the lifting point of the load, keeping the wire length to a minimum. This makes the crane safe and efficient, but results in a crane with a complex load curve and a large minimum radius. Another typical point of a KBC is that on the boom a double set of sheaves is required. When the boom is knuckled, the wire rope runs over one of the sheaves and when the booms are in line the wire rope runs over the other sheave.

Very heavy lifting with this crane type is infeasible because use of hydraulic cylinders supporting the boom causes large shear forces in the boom. Another reason is that the maximum allowable hydraulic pressure in the cylinders is restricted causing that a very large cylindrical area is required.

The KBCs are suitable for light lifting operations at constant height above deck, thus not high above deck. The limited footprint, tail swing and remaining height makes this crane type suitable in situations where limited space is available. There has to be kept in mind that the minimum radius is large. A large minimum radius of a KBC causes limitations when very tall items have to be lifted or loads have to be lifted at high levels at a short radius.

The maximum lift capacity of a KBC is about 250mt (Huisman). This value does not come close to the required 8,000mt for the NSCV.

F: Telescopic boom cranes

The telescopic boom crane does not differ that much with the fixed boom crane. The only difference is that of a telescopic boom crane the boom can be extended and retracted.

Figure 123: Telescopic boom crane

The sections of a telescopic boom crane have a rectangular, trapezoidal, or other shape of symmetrically cross-sectional segments fitting into each other. These segments can be hydraulically extended and retracted. The telescopic boom makes this crane type a space-saving, compact crane with an optional outreach compared to a general fixed boom crane. Because the outreach is optional the relatively heavy load lifts can be carried out with the boom retracted.

The telescopic boom crane makes working in areas possible that would be impossible without the possibility to extend and retract the boom. Also this crane type can lift loads at different radii when the space above the crane is limited which is not possible with a fixed boom crane.

The maximum lifting capacity of a telescopic boom crane is about 500mt (NOV). These value do not come close to the required 8,000mt.

Table 22: Available main crane types

	Crane type	Slew range	Footprint	Tail swing	Remaining height	Sheer forces in boom	Max cap. [mt]
1	A-frame	∞	Large	Large	Medium	Small	7,100
2	Mast	Variable	Medium	Small	Large	Small	5,000
3	Slewing-mast	∞	Medium	Medium	Large	Small	2,000
4	Ringer multi-cranes	Variable	Large	Large	Medium	Small	1,600
5	Sheerlegs	-	Large	-	Medium	Small	5,000

Table 23: Available secondary crane types

	Crane type	Slew range	Footprint	Tail swing	Remaining height	Sheer forces in boom	Max cap. [mt]
6	Kingpost	∞	Small	Medium	Medium	Small	150
7	Fixed boom	∞	Small	Small	Small	Large	350
8	Knuckle boom	∞	Small	Variable	Small	Large	250
9	Telescopic boom	∞	Small	Small	Small	Large	500

APPENDIX B: LIFTING DATA THIALF

The NSCV is expected to carry out similar lifting operations as the Thialf. Therefore, studying the lifting data of the Thialf will provide a general idea about the expected lifting projects characteristics of the NSCV. Because the Thialf was for maintenance in a dry-dock in 2011 there are no data available for that period. In general, for the year 2012 not sufficient data is (yet) available to be used for future expectations. Also for other years about some lifting characteristics not sufficient data is available, discussed in the specific section.

When assessing the data an important point to keep in mind is that the Thialf has the largest lifting capacity available on the heavy lifting market. Thus it is not possible to look to lifting projects that went to competitive companies if the Thialf would have had a larger lifting capacity. In this market segment the loads are designed for the crane vessel. During the design phase of the load it is really important to take the capacity of the crane vessel into account because otherwise placement of the component could become impossible.

Type of lifting operations

The assessed lifting operations are only the so called paid lifts and do not include the lifts of personnel on/off the vessel and the 'coffee' lifts. In total 489 registered lifting operations have been carried out by the Thialf. Of the 489 registered paid lifts the following ratio between the type is shown:

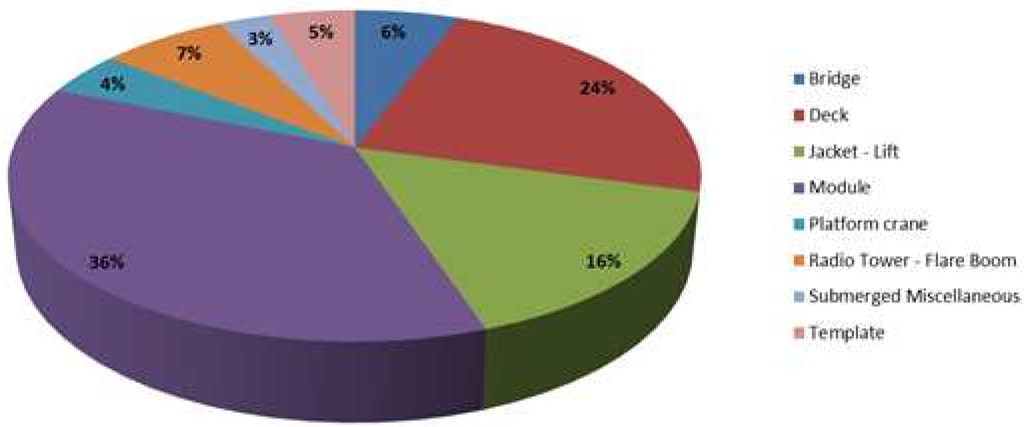


Figure 124: Type of lifting operations Thialf

Lifting weight

Only a few projects are carried out by a crane vessel per year it is not possible to see if the average or maximum lifted weight by the Thialf increases. Also the lifted weight strongly depends on the type of project.

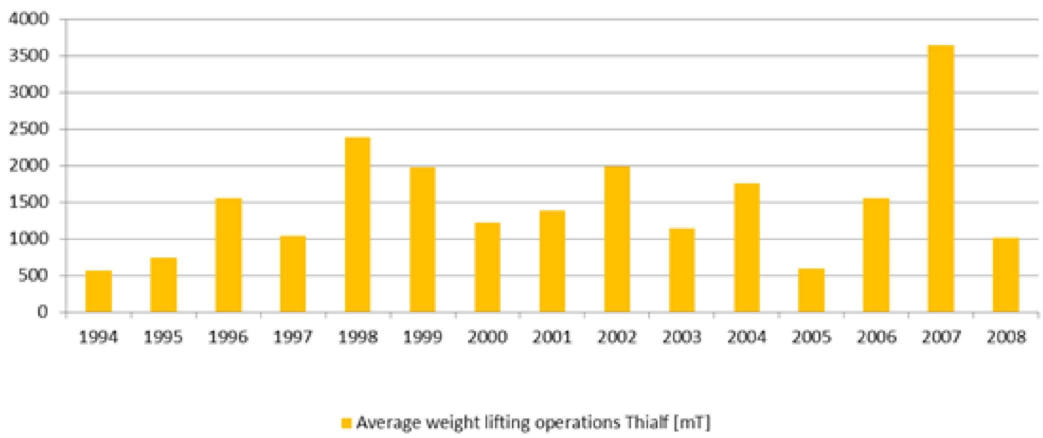


Figure 125: Average weight lifting operations Thialf

The average weight of all paid lifts carried out by a single crane on the Thialf is 1,479mt. This means that when a dual lift was carried out, the lifted weight was divided by two. Of all recorded paid lifting operations only 9% are very heavy lifts (>5,000mt). Because clients are loath to approach the maximum capacity of a crane this maximum is preferable not used. The heaviest dual lifting operation carried out by the Thialf was 11,660mt.

Load dimensions

The available data about the load dimensions lifted by the Thialf does not clarify in which configuration the load was lifted. This means it is not clear how the width or the length of the load is lifted relative to the crane and vessel. Also the available data do not reveal what the height of the load is.

Of the 497 lifting operations recorded only 54 times the lifting height was critical. The lifting height is critical if the block cannot be lifted higher or the boom clearance with the load is critical. Of the most common paid lifting operations the largest load dimensions are shown in Table 24 to get a general overview of the load dimensions lifted by the Thialf.

Table 24: Maximum load dimensions lifted by Thialf

Load type	Project nr.	Length [m]	Width [m]	Weight [mt]
Module	I/0246	63	70	4,194
Jacket - lifted	I/0225	37	73.5	3,489
Deck	I/0310	76	71	5,570
Deck	I/0176	45	107	11,200

Single, partly or dual

In this section is assessed in which ratio the lifting operations were carried out as single, partly dual or dual lifts. A single lifting operation is carried out by only one crane and a dual by two. A partly dual lifting operation means that during the lifting operation the load is only partly suspended by two cranes. In the studied timespan no trends in single, partly, or dual lifting operations is found. Only a slight increase in dual lifting operations is seen and about 23% of the lifting operations are dual lifts, 4% partly dual lifts and 73% single lifts.

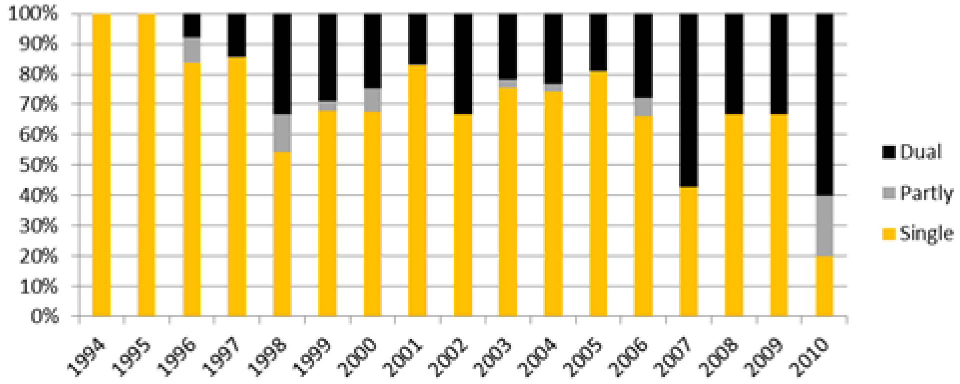


Figure 126: Single, partly or dual lifts Thialf

Removal or installation

Of a total of 540 heavy lifting operations it was recorded if the action performed was installation or removal. With this information the ratio between removal and installation of lifts is studied. No trends in the ratio between removal and installation are shown. In the last years, most of the lifting operations were installation (about 80-90%). In the total time span, 24% of the lifting operations were removal and 76% installation It can be expected that in the future the ration between installation and removal projects will shift.

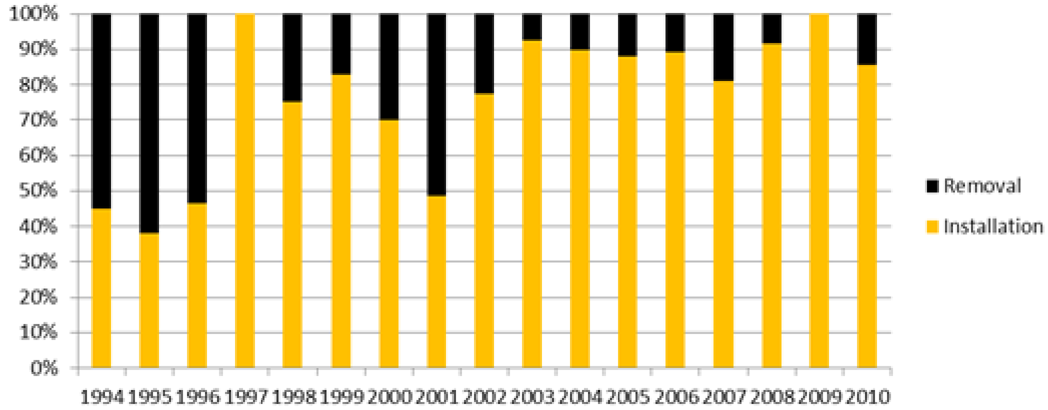


Figure 127: Removal and installation projects Thialf

APPENDIX C: MATLAB

Back-mast and tackle loads

```
%NewLoadbasedonvariablebackmastangle.m

%overturning angle of 180 degrees
    x=0:0.005:pi;
    x_degrees=x*180/pi;
    xx_degrees=...
      (120) .* (x_degrees<20) +...
      (120-36/89*(x_degrees-20)) .* (x_degrees>=20)+...
      (-36/360*(x_degrees-109)) .* (x_degrees>=109); %84 and 48
    xx=xx_degrees/180*pi;

%mast
    mast_height=50;
    mast_bottom=14;
    mast_top=8;
    mast_thickness=0.075;
    mast_weight=1.1*(7.850*mast_height*pi*(0.5*mast_bottom+0.5*mast_top)*mast_thickness) %+10%
for stairs etc.
    head_weight=96.6*sqrt(8/4)

%back-mast
    back_mast_length=30;

%boom
    boom_length=77.075;
    boom_weight=1400;
    boom_pivot_to_cl=2.5;
    boom_pivot_to_cog=57.01;
    boom_moment=9.81*boom_weight*(boom_pivot_to_cl+mast_bottom/2+boom_pivot_to_cog);

%location CoG
    CG_y_start=(head_weight*mast_height+mast_weight*mast_height/3)/(head_weight+mast_weight);
    CG_x_start=mast_bottom/2;
    CG_r=sqrt(CG_y_start^2+CG_x_start^2)
    angle_CG_r=acos(mast_bottom/2/CG_r);
    head_r=sqrt(mast_height^2+(mast_bottom/2+mast_top/2)^2)
    angle_head_r=asin(mast_height/head_r)
    VCG=sin(angle_CG_r+x)*CG_r;
    HCG=-cos(angle_CG_r+x)*CG_r;
    moment=-HCG*9.81*(mast_weight+head_weight);
    mast_head_x=-head_r*cos(x+angle_head_r);
    mast_head_y=head_r*sin(x+angle_head_r);
    back_mast_x=back_mast_length*cos(xx);
    back_mast_y=back_mast_length*sin(xx);

figure(1)
    plot(x*180/pi,VCG)
    hold all
    plot(x*180/pi,HCG)
    xlabel('Overturning angle (Radian)')
    ylabel('Distance (m)')
    legend('VCG distance','HCG distance')
    set(gcf,'Name','Locations')

figure(2);
    plot(x*180/pi,-moment,'r')
    hold all
    plot(x*180/pi,boom_moment,'b')
    xlabel('Overturning angle (Degrees)')
    ylabel('Moment (kNm)')
    legend('Overturning moment mast','Available boom moment')
    set(gcf,'Name','Moment')

figure(3)
    plot(x*180/pi,back_mast_x)
    hold all
    plot(x*180/pi,back_mast_y)
    plot(x*180/pi,mast_head_x)
    plot(x*180/pi,mast_head_y)
    xlabel('Overturning angle (Degrees)')
    ylabel('Back-mast top location (m)')
    legend('Back-mast x','Back-mast y','Mast head x','Mast head y')
    set(gcf,'Name','Mast locations')
```

```
figure(4)
    plot(x_degrees,xx_degrees)
    axis([0 180 0 130])
    xlabel('Overturning angle alpha (Degrees)')
    ylabel('Back-mast angle beta (Degrees)')
    legend('Back-mast angle')
    set(gcf,'Name','Back-mast angle')

figure(5)
    alpha=-(pi/2-angle_head_r-x);
    beta1=atan(abs(mast_head_y-back_mast_y)./abs(mast_head_x-back_mast_x));
    beta2=pi/2+atan((back_mast_x-mast_head_x)./(back_mast_y-mast_head_y));
    beta=...
        (beta1) .* (x*180/pi<161)+...
        (beta2) .* (x*180/pi>=161);
    gamma1=alpha+beta;
    gamma2=alpha-beta;
    gamma=...
         (gamma1).* (mast_head_y>back_mast_y)+...
         (gamma2) .* (mast_head_y<=back_mast_y);

    teta=...
        (abs(beta-(xx-pi/2))) .* (mast_head_y>=back_mast_y)+...
        (abs(beta-(pi/2-xx))) .* (mast_head_y<back_mast_y);
    phi=atan(back_mast_y./(back_mast_x+boom_length));
        zeta1=xx-pi/2-phi;
        zeta2=pi/2-xx+phi;
    zeta=max(zeta1,zeta2);

    plot(x*180/pi,alpha*180/pi)
    hold all
    plot(x*180/pi,beta*180/pi)
    plot(x*180/pi,gamma*180/pi)
    plot(x*180/pi,teta*180/pi)
    plot(x*180/pi,phi*180/pi)
    plot(x*180/pi,zeta*180/pi)
    xlabel('Overturning angle (Degrees)')
    ylabel('Angles (Degrees)')
    legend('Alpha','Beta','Gamma','Teta','Phi','Zeta')
    set(gcf,'Name','Angles')

figure(6)
    head_force_trans=-moment./mast_height/9.81;
    hoist_tackle_load=...
      (0) .* (x_degrees<19.03) +...
      (head_force_trans./cos(gamma)) .* (x_degrees>=19.03);
        max_hoist_tackle_load=max(hoist_tackle_load)
    back_mast_trans=hoist_tackle_load.*cos(teta);
    back_tackle_load=...
        (0) .* (x_degrees<19.03) +...
        (back_mast_trans./cos(zeta) .* (x_degrees>=19.03));
            max_back_tackle_load=max(back_tackle_load)
    hoist_tackle_comp=hoist_tackle_load.*sin(teta);
    back_tackle_comp=back_tackle_load.*sin(zeta);
    back_mast_comp=50*sin(xx)+hoist_tackle_comp+back_tackle_comp;
        max_back_mast_comp=max(back_mast_comp)

    hold all
    plot(x*180/pi,hoist_tackle_load)
    plot(x*180/pi,back_tackle_load)
    %plot(x*180/pi, back_mast_trans)
    xlabel('Overturning angle alpha (Degrees)')
    ylabel('Load (mt)')
    legend('Hoist tackle load','Back tackle load')
    set(gcf,'Name','Component forces')

figure(7)
    %plot(x*180/pi,hoist_tackle_comp)
    hold all
    %plot(x*180/pi,back_tackle_comp)
    plot(x*180/pi,back_mast_comp)
    xlabel('Overturning angle alpha (Degrees)')
    ylabel('Load (mt)')
    legend('Back-mast compression')
    set(gcf,'Name','Component forces')
```

```
figure(8)
    mast_hinge_load=...
       ((mast_weight+head_weight)+(mast_weight+head_weight)*HCG/mast_bottom) .*
       (x_degrees<19.03) +...
       (sqrt((cos(pi/2-gamma).*hoist_tackle_load).^2+((mast_weight+head_weight)-
       (mast_weight+head_weight)*HCG./mast_head_x).^2) .* (x_degrees>=19.03));
    plot(x*180/pi,mast_hinge_load)
    xlabel('Overturning angle alpha (Degrees)')
    ylabel('Load (mt)')
    legend('Mast hinge point load')
    set(gcf,'Name','Mast hinge point load')
```

Load curve and component forces 8,000mt crane

```
%NewLoadcurveComponentforces.m

fhoist=1.1;
fscale=8/7.1;

%mast
    mast_height=55; %above hinge point boom
    mast_bottom=14;
    mast_top=8;
    mast_thickness=0.075;
    mast_weight=1.1*(7.850*mast_height*pi*(0.5*mast_bottom+0.5*mast_top)*mast_thickness)
       %+10% for stairs etc.
    head_weight=96.6*sqrt(8/4)
    hoist=300;

%boom
    boom_length=85.3;
    boom_weight=1400;
    boom_pivot_to_cl=12.8;
    boom_pivot_to_deck=24.4;
    boom_pivot_to_cog=57.01;

figure(1)
    %Thialf load curve
    radius=                 [31.2 36 44 52 60 68 76 84 92 95]; %100 108 116 119
    height_main_above_wl=   [119.2 117 115 111.2 107 101 93 83 67 59]; % 22.9 22.9 22.9 22.9
    height_aux1=            [22.9 146 144 141 138 134 129 122 115 110]; % 106 94 77 69
    main_capacity=          [7100 6750 6100 4400 3100 2350 1750 1250 825 650 ];   %0 0 0 0
    height_main_above_deck=height_main_above_wl-22.9;
    height_main_above_pivot=height_main_above_wl-22.9-boom_pivot_to_deck;
    radius_from_pivot=radius-boom_pivot_to_cl;
    boom_angle=atan(height_main_above_pivot./radius_from_pivot);
    overturning_moment=main_capacity.*radius_from_pivot;
    overturning_moment_incl=
       fscale*fhoist*overturning_moment+boom_weight*boom_pivot_to_cog*cos(boom_angle)+hoist*ra
       dius_from_pivot;
    [max_overturning_moment_incl, Index]=max(overturning_moment_incl)

    plot(radius_from_pivot, overturning_moment)
    hold all
    plot(radius_from_pivot, overturning_moment_incl)
    xlabel('Boom radius (m)')
    ylabel('Overturning moment (mtm)')
    legend('Overturning moment load curve','Total overturning moment')
    set(gcf,'Name','Load curve 8,000mt')

%max forces TT

figure(2)
    boom_transv=overturning_moment_incl./boom_length;
    %main hoist height > mast height
    alpha=boom_angle;
    beta=pi/2-alpha;
    gamma=atan((height_main_above_pivot-mast_height)./(radius_from_pivot+boom_pivot_to_cl));
    teta=atan((mast_height-height_main_above_pivot)./(boom_pivot_to_cl+radius_from_pivot));
    %main hoist height <= mast height
    zeta=pi/2-teta-alpha;

    FTT=...
        (boom_transv./cos(beta+gamma)) .* (height_main_above_pivot > mast_height) +...
        (boom_transv./cos(zeta)) .* (height_main_above_pivot <= mast_height);

    head_x=FTT.*cos(gamma);
    head_y=FTT.*sin(gamma);
```

```
    max_head_load_x=max(head_x)
    max_head_load_y=max(head_y)
    min_head_load_y=min(head_y)

    plot(radius_from_pivot,FTT)
    hold all
    plot(radius_from_pivot,head_x)
    plot(radius_from_pivot,head_y)
    xlabel('Boom radius (m)')
    ylabel('Load (mt)')
    legend('F_t_o_p_p_i_n_g_t_a_c_k_l_e','F_h_e_a_d_x','F_h_e_a_d_y')
    set(gcf,'Name','Topping tackle and mast head forces')

%Infinite life
figure(3)
    overturning_moment_inf= ...
        (overturning_moment_incl/1.2) .* (radius_from_pivot<=70)+...
        (overturning_moment_incl) .* (radius_from_pivot>70);
    plot(radius_from_pivot,overturning_moment_inf./radius_from_pivot)

figure(4)
    plot(radius_from_pivot,overturning_moment_inf)
    xlabel('Boom radius (m)')
    ylabel('Overturning moment (mtm)')
    legend('Overturning moment infinite life')
    set(gcf,'Name','Load curve 1,000mt')

figure(4)
    boom_transv_inf=overturning_moment_inf./boom_length;
    %main hoist height > mast height
    alpha=boom_angle;
    beta=pi/2-alpha;
    gamma=atan((height_main_above_pivot-mast_height)./(radius_from_pivot+boom_pivot_to_cl));
    teta=atan((mast_height-height_main_above_pivot)./(boom_pivot_to_cl+radius_from_pivot));
    %main hoist height <= mast height
    zeta=pi/2-teta-alpha;

    FTT_inf=...
        (boom_transv_inf./cos(beta+gamma)) .* (height_main_above_pivot > mast_height) +...
        (boom_transv_inf./cos(zeta)) .* (height_main_above_pivot <= mast_height);

    head_x_inf=FTT_inf.*cos(gamma);
    head_y_inf=FTT_inf.*sin(gamma);

    max_head_load_x_inf=max(head_x_inf)
    max_head_load_y_inf=max(head_y_inf)
    min_head_load_y_inf=min(head_y_inf)

    %plot(radius_from_pivot,boom_transv_inf)
    hold all
    plot(radius_from_pivot,FTT_inf)
    plot(radius_from_pivot,head_x_inf)
    plot(radius_from_pivot,head_y_inf)
    xlabel('Boom radius (m)')
    ylabel('Load (mt)')
    legend('Topping tackle','Mast head_x','Mast head_y')
    set(gcf,'Name','Topping tackle and mast head forces')

figure(5)

overturning_moment_whip=boom_weight*boom_pivot_to_cog*cos(boom_angle)+900*(boom_length+26)*cos
(boom_angle);
    boom_transv_aux=overturning_moment_whip./boom_length;
    FTT_whip=...
        (boom_transv_aux./cos(beta+gamma)) .* (height_main_above_pivot > mast_height) +...
        (boom_transv_aux./cos(zeta)) .* (height_main_above_pivot <= mast_height);

    head_x_whip=FTT_whip.*cos(gamma);
    head_y_whip=FTT_whip.*sin(gamma);

    max_head_load_x_aux=max(head_x_whip)
    max_head_load_y_aux=max(head_y_whip)
    min_head_load_y_aux=min(head_y_whip)

    plot(radius_from_pivot,head_x_whip)
    hold all
    plot(radius_from_pivot,head_y_whip)
    xlabel('Boom radius (m)')
    ylabel('Load (mt)')
    legend('Aux hoist mast head_x','Aux hoist mast head_y')
    set(gcf,'Name','Topping tackle and mast head forces for inf aux useage')
```

```
 figure(6)
overturning_moment_whip=boom_weight*boom_pivot_to_cog*cos(boom_angle)+200*(boom_length+35)*cos
(boom_angle);
    boom_transv_whip=overturning_moment_whip./boom_length;
    FTT_whip=...
        (boom_transv_whip./cos(beta+gamma)) .* (height_main_above_pivot > mast_height) +...
        (boom_transv_whip./cos(zeta)) .* (height_main_above_pivot <= mast_height);

    head_x_whip=FTT_whip.*cos(gamma);
    head_y_whip=FTT_whip.*sin(gamma);

    max_head_load_x_whip=max(head_x_whip)
    max_head_load_y_whip=max(head_y_whip)
    min_head_load_y_whip=min(head_y_whip)

    plot(radius_from_pivot,head_x_whip)
    hold all
    plot(radius_from_pivot,head_y_whip)
    xlabel('Boom radius (m)')
    ylabel('Load (mt)')
    legend('Whip hoist mast head_x','Whip hoist mast head_y')
    set(gcf,'Name','Topping tackle and mast head forces for inf whip useage')

 figure(7)
    radius_thialf=[31.2 36 44 52 60 68 76 84 92 95];
    capacity_thialf=[7100 6750 6100 4400 3100 2350 1750 1250 825 650 ];

    radius_aegir=[18 40 44 49 60 68 76 78];
    capacity_aegir=[4000 4000 3600 3200 2470 2070 1800 1500];

    radius_baldersb=[24 28 36 39 45 53 62 73 80];
    capacity_baldersb=[3300 3300 3050 2800 2400 2000 1600 1200 1000];

    radius_saipem=[34 40 45 50 60 70 80 90 100 110];
    capacity_saipem=[7000 7000 6000 5100 3800 2800 2000 1400 900 500];

    radius_sheerleg=[21 24 27 30 32 34 37 40 43 46 49 52 55 58 61 64 67 70 73];
    capacity_sheerleg=[5000 5000 5000 5000 4900 4700 4400 4000 3500 3250 3000 2700 2500 2250
       2050 1900 1750 1600 1500];

    radius_borealis=[17 20 24 30 34 40 45 50 55 60 65 70 75];
    capacity_borealis=[4200 4700 5000 5000 4100 3500 3100 2700 2350 2150 1850 1700 1500];

    plot(radius_thialf, capacity_thialf)
    hold all
    plot(radius_aegir, capacity_aegir)
    plot(radius_baldersb, capacity_baldersb)
    plot(radius_saipem, capacity_saipem)
    plot(radius_sheerleg, capacity_sheerleg)
    plot(radius_borealis, capacity_borealis)
    xlabel('Outreach from centerline (m)')
    ylabel('Load (mt)')
    legend('Thialf','Aegir','Balder SB','Saipem','Sheerleg','Borealis')
    set(gcf,'Name','Loadcurve comparison')
```

Thialf load curve and lifting mast with other crane

```
% NewPositionsbackmastandmoments

close all;
clc;
clear;

%mast
    mast_height=50;
    mast_bottom=14;
    mast_top=8;
    mast_thickness=0.075;
    mast_weight=1.1*(7.850*mast_height*pi*(0.5*mast_bottom+0.5*mast_top)*mast_thickness)
       %+10% for stairs etc.
    head_weight=96.6*sqrt(8/4)

%boom
    boom_length=77.075;
    boom_weight=1400;
    boom_pivot_to_cl=2.5;
    boom_pivot_to_cog=57.01;
```

```
%overturning angle of 180 degrees
    parts=1000;
    x=0:1/parts:pi;

%location CoG
    CG_y_start=(head_weight*mast_height+mast_weight*mast_height/3)/(head_weight+mast_weight);
    CG_x_start=mast_bottom/2;
    CG_r=sqrt(CG_y_start^2+(CG_x_start-1.2)^2)

    angle_CG_r=acos(mast_bottom/2/CG_r);
    head_r=sqrt(mast_height^2+(mast_bottom/2)^2);
    angle_head_r=atan(mast_height/(mast_bottom/2));

    head_x=head_r*cos(x+angle_head_r);
    head_y=head_r*sin(x+angle_head_r);

figure(1);
    VCG=sin(angle_CG_r+x)*CG_r;
    HCG=cos(angle_CG_r+x)*CG_r;
    zeroline=0*x;

    plot(x*180/pi,VCG)
    hold all
    plot(x*180/pi,HCG)
    plot(x*180/pi,zeroline)

    xlabel('Overturning angle alpha (Degrees)')
    ylabel('Distance from boom hinge point (m)')
    legend('VCG','HCG')
    set(gcf,'Name','Locations')

%boom moment and overturning moment
    boom_moment=9.81*boom_weight*(boom_pivot_to_cl+mast_bottom/2+boom_pivot_to_cog);
    moment=-HCG*9.81*(mast_weight+head_weight);
    [max_moment,index_moment_max]=max(moment)
    angle_moment_max_degrees=index_moment_max/parts*180/pi

figure(2);
    plot(x*180/pi,moment,'r')
    hold all
    plot(x*180/pi,boom_moment,'b')
    xlabel('Overturning angle (Degrees)')
    ylabel('Moment (kNm)')
    legend('Overturning moment mast','Available boom moment')
    set(gcf,'Name','Moments')

%push
figure(3);
    moment_push = ...
        (0) .* (HCG < 0) + ...
        (-moment) .* (HCG>=0);
    push_angle_min=asin(mast_bottom/2/CG_r)
    push_arm_min=mast_bottom/2;
    push_arm_max=mast_bottom;
    push_force_min_perp=moment_push/push_arm_max;
    push_force_max_perp=moment_push/push_arm_min;
    max_push_force_min_perp=max(push_force_min_perp)
    max_push_force_max_perp=max(push_force_max_perp)

    plot(x*180/pi,push_force_min_perp)
    axis([0 20 0 14000]);
    hold all
    plot(x*180/pi,push_force_max_perp)
    xlabel('Overtuning angle (Degrees)')
    ylabel('Force (kN)')
    legend('Lifting at mast bottom edge','Lifting at mast CL')
    set(gcf,'Name','Perpendicular force')

    max_area=max_push_force_max_perp/25000;       %if force in kN
    min_area=max_push_force_min_perp/25000;

    cyl_diam_max=sqrt(max_area/(pi/4)/2)           %2 cylinders
    cyl_diam_min=sqrt(min_area/(pi/4)/2)           %2 cylinders

    min_stroke=sin(push_angle_min)*(push_arm_max/2)
    max_stroke=sin(push_angle_min)*push_arm_max
```

```
figure(4)
    %Thialf load curve
    boom_length_thialf=85;

    radius=                  [31 36 44 52 60 68 76 84 92 95 100 108 116 119];
    height_main_above_wl=    [119.2 117 115 111.2 107 101 93 83 67 59 22.9 22.9 22.9 22.9];
    height_main_above_deck=height_main_above_wl-22.9;
    height_main_above_pivot=height_main_above_wl-22.9-25;
    height_aux1=             [22.9 146 144 141 138 134 129 122 115 110 106 94 77 69];

    plot(radius, height_main_above_deck)
    axis([37 120 0 150]);
    hold all
    plot(radius, height_aux1)

    xL = get(gca,'XLim');
    line(xL,[80 80],'Color','r');
    yL = get(gca,'YLim');
    line([72 72],yL,'Color','r');

    xlabel('Outreach from centerline (m)')
    ylabel('Height above deck (m)')
    legend('Lifting height main hoist','Lifting height aux hoist')
    set(gcf,'Name','Thialf load curve')

figure(5)
    fhoist=1.3;
    hoist=200;
    main_capacity=           [7100 6750 6100 4400 3100 2350 1750 1250 825 650 0 0 0 0];

    boom_angle=atan(height_main_above_pivot/radius);
    overturning_moment=main_capacity.*radius;
    overturning_moment_incl=
       overturning_moment*1.3+boom_weight*boom_pivot_to_cog*cos(boom_angle)+hoist*radius;
    [max_overturning_moment_incl, Index]=max(overturning_moment_incl)

    plot(radius, overturning_moment)
    axis([30 100 0 4.5E5]);
    hold all
    plot(radius, overturning_moment_incl)

    xlabel('Radius (m)')
    ylabel('Overturning moment (mtm)')
    legend('Load curve','Total overturning moment')
    set(gcf,'Name','Thialf load curve')
```

Determine optimal angle and length of back-mast

```
% NewMaxoverturningangledetermineangleandlength.m

%back-mast
    back_mast_length=[15:0.1:55];

%overturning angle of 180 degrees
    x=180/180*pi %109 or 180
    xx_degrees=48;
    xx=xx_degrees/180*pi
    moment=-8.64E4;

%mast
    mast_height=50;
    mast_bottom=14;
    mast_top=8;
    mast_thickness=0.075;
    mast_weight=1.1*(7.850*mast_height*pi*(0.5*mast_bottom+0.5*mast_top)*mast_thickness) %+10%
for stairs etc.
    head_weight=96.6*sqrt(8/4)

%boom
    boom_length=77.075;
    boom_weight=1400;
    boom_pivot_to_cl=2.5;
    boom_pivot_to_cog=57.01;
    boom_moment=9.81*boom_weight*(boom_pivot_to_cl+mast_bottom/2+boom_pivot_to_cog);
```

```
%location CoG
    CG_y_start=(head_weight*mast_height+mast_weight*mast_height/3)/(head_weight+mast_weight);
    CG_x_start=mast_bottom/2;
    CG_r=sqrt(CG_y_start^2+CG_x_start^2)
    angle_CG_r=acos(mast_bottom/2/CG_r);
    head_r=sqrt(mast_height^2+(mast_bottom/2+mast_top/2)^2);
    angle_head_r=atan(mast_height/(mast_bottom/2));
    VCG=sin(angle_CG_r+x)*CG_r;
    HCG=-cos(angle_CG_r+x)*CG_r;

    mast_head_x=7;
    mast_head_y=-50;
    back_mast_x=back_mast_length*cos(xx);
    back_mast_y=back_mast_length*sin(xx);

%angles
    alpha=pi/2+angle_head_r;
    beta=pi/2+atan((back_mast_x-mast_head_x)./(back_mast_y-mast_head_y));
    gamma=alpha-beta;
    teta=beta-(pi/2-xx);
    phi=atan(back_mast_y./(back_mast_x+boom_length));
    head_force_trans=-moment/mast_height/9.81;
    hoist_tackle_load=head_force_trans./cos(gamma);
    back_mast_trans=hoist_tackle_load.*cos(teta);
    zeta=pi/2-xx+phi;
    back_tackle_load=back_mast_trans./cos(zeta);
    hoist_tackle_comp=hoist_tackle_load.*sin(teta);
    back_tackle_comp=back_tackle_load.*sin(zeta);
    back_mast_comp=hoist_tackle_comp+back_tackle_comp;
        max_back_mast_comp=max(back_mast_comp)


figure(1)
    plot(back_mast_length,zeta)
    hold all
    plot(back_mast_length,beta)
    plot(back_mast_length,gamma)
    plot(back_mast_length,teta)
    plot(back_mast_length,phi)
    plot(back_mast_length,alpha)
    xlabel('Back-mast length (m)')
    ylabel('Angle (Radian)')
    legend('Zeta','Beta','Gamma','Teta','Phi','Alpha')
    set(gcf,'Name','Component loads max moment')

figure(2)
    plot(back_mast_length,hoist_tackle_load)
    hold all
    plot(back_mast_length,back_tackle_load)
    plot(back_mast_length,back_mast_comp)

    mast_hinge_load=sqrt((cos(pi/2-gamma).*hoist_tackle_load).^2+(mast_weight-
mast_weight*HCG./mast_head_x).^2);
    max_mast_hinge_load=max(mast_hinge_load)

    plot(back_mast_length,mast_hinge_load)
    xlabel('Back-mast length (m)')
    ylabel('Load (mt)')
    legend('Hoist tackle load','Back tackle load','Back-mast compression','Mast hinge point
load')
    set(gcf,'Name','Component loads max overturning angle')
```

Bleich mast hinge point

```
P=8.80E6;
yield=345;

a=0.35;
d=0.20;
t=0.20; %twice mast thickness
r=(a+d)/2
v=a/2
A=a*t
z=r^2*t*(r*log((2*r+a)/(2*r-a))-a)
beta=-r^2/z*(4/pi^2-1/8)/(2*(1/A+r^2/z))
N=P*(0.25-beta)
M=P*r*(0.25+beta)
```

```
shear=N/A/1000000
shear_allowable=0.4*yield
sigma1=(N/A+M*v/z*r/(r-v))/1000000
sigma2=(N/A-M*v/z*r/(r+v))/1000000
sigma_allowable=0.66*yield
contact_pressure=P/(d*t)/1000000
contact_pressure_allowable=0.9*yield
```

Bleich back-mast hinge point

```
P=7669E3/4
yield=345;

a=0.20;
d=0.08;
t=0.08; %twice mast thickness
r=(a+d)/2
v=a/2
A=a*t
z=r^2*t*(r*log((2*r+a)/(2*r-a))-a)
beta=-r^2/z*(4/pi^2-1/8)/(2*(1/A+r^2/z))
N=P*(0.25-beta)
M=P*r*(0.25+beta)

shear=N/A/1000000
shear_allowable=0.4*yield
sigma1=(N/A+M*v/z*r/(r-v))/1000000
sigma2=(N/A-M*v/z*r/(r+v))/1000000
sigma_allowable=0.66*yield
contact_pressure=P/(d*t)/1000000
contact_pressure_allowable=0.9*yield
```

APPENDIX D: HINGE POINTS

Calculating the hinge point dimensions is done with the Bleich method. When the total load is equally divided over the mast hinge point the maximum design load on each of the two hinge points is:

$$F_{hinge} = 0.5 * 1.1 * 1.3 * 1{,}246 * 9.81 = 8.80MN$$

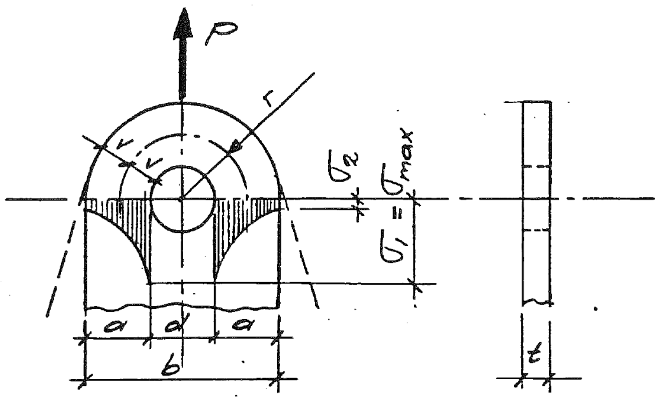


Figure 128: Maximum hinge point stresses

To keep the stresses in the mast hinge point below the maximum values a combination of the various hinge point parameters is calculated shown in Figure 128. After a number of iterations these parameters are obtained:

$$d = 0.20m$$
$$a = 0.35m$$
$$t = 0.20m$$

In the calculations below is shown that with these parameters the maximum occuring stresses stay below maximum stress values. The following material properties are assumed:

$$\sigma_{yield} \leq 345.0MPa$$

The maximum allowable stress by the combined forces acting on the mast hinge point:

$$\sigma_e = 0.66 * 345 = 227.7\text{MPa}$$

The maximum allowable shear stress:

$$\sigma_s = 0.4 * 345 = 138.0MPa$$

The radius of the material surrounding the hinge pin:

$$r = \frac{a + d}{2} = \frac{0.35 + 0.20}{2} = 0.275m$$

Half the width of the material radius:

$$v = 0.5 * a = 0.5 * 0.35 = 0.175m$$

The surface area of the material:

$$A = a * t = 0.35 * 0.20 = 0.07m^2$$

The dimensionless parameters required for the calculations:

$$z = r^2 * t\left(r * \ln\left(\frac{2r + a}{2r - a}\right) - a\right) = 9.62E - 4$$

$$\beta = -\frac{\frac{r^2}{z}\left(\frac{4}{\pi^2} - \frac{1}{8}\right)}{2\left(\frac{1}{A} + \frac{r^2}{z}\right)} = -0.12$$

The load (N) and bending moment (M):

$$N = P\left(\frac{1}{4} - \beta\right) = 3{,}244kN$$

$$M = P * r\left(\frac{1}{4} + \beta\right) = 318.0kNm$$

Now the maximum stresses on the inside and outside of the eyes can be calculated by combining the average stress with the bending moment.

The average stress:

$$\frac{N}{A} = 46.3\text{MPa}$$

Combined inside stress of the pad eye:

$$\sigma_{max} = \sigma_1 = \frac{N}{A} + \frac{M * v}{z} * \frac{r}{r - v} = 205.4\text{MPa}$$

Combined outside stress of the pad eye:

$$\sigma_2 = \frac{N}{A} - \frac{M * v}{z} * \frac{r}{r + v} = 11.0Mpa$$

The maximum stress values stay below the allowable stress of 227.7MPa. Also the contact pressure between the surfaces is verified:

$$\sigma_{contact} = \frac{P}{d * t} \leq 0.9 * \sigma_{yield}$$

$$\frac{8{,}800E3}{0.20 * 0.20} \leq 0.9 * 345.0E6$$

$$220.0MPa \leq 310.5MPa$$

The maximum stresses stay below their limits. Now the rest of the mast hinge point is basically designed. The cross section of the plate in which the eye is placed must have a minimum cross sectional area. The length of this cross section is for a conservative calculation:

$$l = d + 2a = 0.90m$$

Then the minimum plate thickness (t_m in Figure 129) in which the pad eye has to be placed and to which it has to be connected to the mast is calculated by:

$$\frac{\mathrm{P}}{\mathrm{l} * \mathrm{t}_m} = \frac{8{,}800\mathrm{E}3}{0.80 * \mathrm{t}_m} \leq 0.4 * 345.0\mathrm{E}6$$

$$t_m \geq 80mm$$

$$t_m + 2 * t_c \geq 200mm$$

$$t_c \geq 40mm$$

The minimum radius of the main plate can be calculated with a rule of thumb:

$$r_m = 1.2 * r_c = 1.2 * 400 = 540mm$$

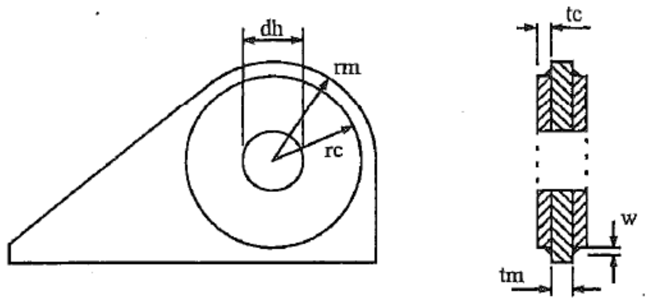


Figure 129: Pad eye

The maximum loading (1,246mt) between the hinge point and the mast is formed by a bending moment and shear forces. The total stress is:

$$\sigma_{max} = \sqrt{\sigma_{bm}{}^2 + 3\tau^2} \leq 227.7\mathrm{MPa}$$

$$\sigma_{bm} = \frac{M * y}{I} = \frac{0.5 * 1{,}246E3 * 9.81 * 1.2 * \frac{L}{2}}{0.065 * \frac{L^3}{12}} = \frac{677.0E6}{L^2}\mathrm{MPa}$$

$$\tau = \frac{0.5 * 1{,}246E3 * 9.81}{L * 0.065} = \frac{94E6}{L}\mathrm{MPa}$$

$$\sigma_{max} = \sqrt{\left(\frac{677.0E6}{L^2}\right)^2 + 3\left(\frac{94E6}{L}\right)^2} \leq 228\mathrm{MPa}$$

$$\mathrm{L} \geq 1{,}799\mathrm{mm}$$

The maximal back-mast compression is 1,360mt. It is assumed the loads are equally divided over the two back-mast hinge points and wind loading on the back-mast is neglected. The hinge points are located opposite from each other on the slew platform of the crane, close to the centerline of the mast. Just as for the mast hinge point calculations a safety factor of 1.1 is taken into account, resulting in a load of 680mt per hinge point. When also the own weight of the back-mast (50mt) and the angle of the two masts of the back-mast are taken into account the loads on the two hinge points are:

$$F_{hinge1,2} = 7{,}669kN$$

Calculating the back-mast hinge points is done with the Bleich method and the same steps are carried out as in the mast hinge point calculations. The following hinge point parameters are obtained:

$$a = 0.20m$$
$$d = 0.08m$$
$$t = 0.08m$$

In the hinge point material the combined inside stress are the maximum stresses have to stay below the maximum stress limit, just as the contact stresses between the pin and eye.

$$\sigma_{max} = \sigma_1 = \frac{N}{A} + \frac{M * v}{z} * \frac{r}{r - v} = \frac{2.83E6}{0.016} + \frac{2.77E5 * 0.100}{7.97E - 5} * \frac{0.140}{0.140 - 0.100} = 206.3 \leq 227.7\text{MPa}$$

$$\sigma_{contact} = \frac{P}{d * t} = \frac{7{,}669}{0.08 * 0.08} = 299.6 \leq 310.5MPa$$

Thus both the maximum stress in the hinge point material and between the contact areas are allowable. With this given, the rest of the hinge point dimensions can be determined.

$$\frac{\mathrm{P}}{\mathrm{l} * \mathrm{t}_m} = \frac{0.25 * 7{,}669\mathrm{E}3}{0.48 * \mathrm{t}_m} \leq 0.4 * 345\mathrm{E}6$$

$$t_m \geq 29mm$$

$$t_m + 2 * t_c \geq 80mm$$

$$t_c \geq 26mm$$

$$r_m = 1.2 * r_c = 1.2 * 200 = 240mm$$

APPENDIX E: BACK-MAST STRUCTURE

When the angle under which the mast is placed is taken into account, the load on a single mast is 7,669kN. For the back-mast design Eurocode 3 is applied. First the columns are designed based on the allowable stresses in the back-mast material. After that a buckling analysis will be carried out.

The failure modes that are assessed in the buckling analysis are failure of the column as a whole, failure of one of the main component segments and failure of lacing between the main (Figure 130).

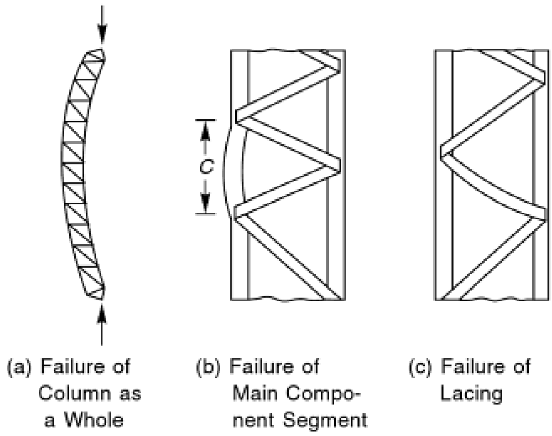


Figure 130: Failure modes of a latticed column (Shukla 2009)

The column is a compression member, but the shear forces also have to be taken into account, given by:

$$V_{Ed} = \pi * \frac{M_{Ed}}{L} = \pi * \frac{3{,}123}{31.5} = 320.4kN$$

The angle of inclination (θ) of the lacing bars with the longitudinal axis of the component member should be kept between 40° and 70°. The angle of inclination is a trade-off between required lacing dimensions and lacing buckling. For this case an angle of inclination is used without detailed optimization towards welding and material cost (θ = 60°).

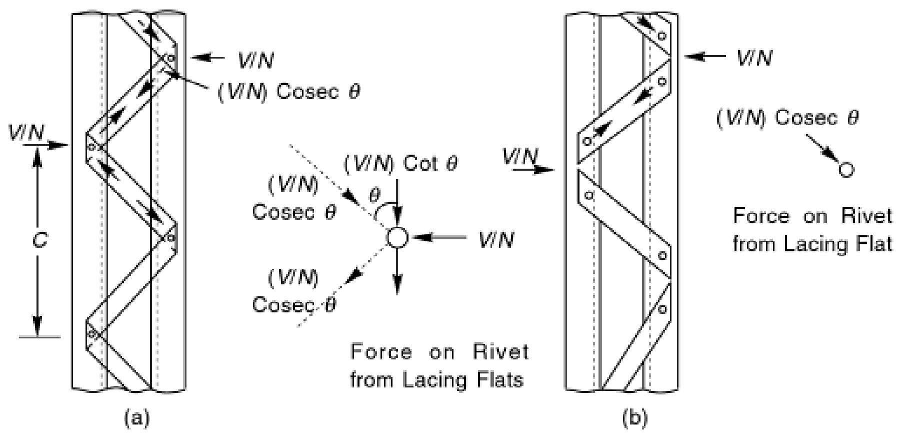


Figure 131: Force in lacing

The allowable stress in the back-mast material when Fe345 is used:

$$\sigma_{compression} \leq 0.6 * \sigma_{yield} = 0.6 * 345.0 = 207.0MPa$$

The compressive force in the lacing is for a single lacing system (N = 2):

$$F_{lacing} = \frac{V_{Ed}}{N * \sin(\theta)} = \frac{320.4}{2 * \sin(60^{\circ})} = 185.0kN$$

$$A_{lacing} \geq \frac{185.0E3}{207.0} = 893.6mm^2$$

As lacing tubes with a diameter of 80mm and wall thickness of 4mm is therefore sufficient (A = 955.0mm^2). For the lacing also has to be checked if buckling could occur (Figure 108).

The buckling load according to Euler when the buckling factor (K) is one for this particular case:

$$I = \frac{\pi\left({d_2}^4 - {d_1}^4\right)}{64} = \frac{\pi(80^4 - 72^4)}{64} = 69.1cm^4$$

$$P_{kr} = \frac{\pi^2 * E * I}{(K * L)^2} = \frac{\pi^2 * 2.1E11 * 69.1E - 8}{(1 * 2.31)^2} = 268.6kN$$

The maximum acting load is smaller than the critical load, failure of lacing is prevented. Now the chords will be designed. The design force $N_{ch,Ed}$ for a column with four identical chords should be determined from:

$$N_{ch,Ed} = 0.25N_{Ed} + \frac{M_{Ed} * h_0 * A_{ch}}{4I_{eff}} = 0.25 * 7{,}669 + \frac{3{,}089 * 2 * 0.0109}{4 * 0.0434} = 2{,}304kN$$

$M_{Ed} =$ *design value of the maximum moment in the middle of the built up member − considering second order effects*
$h_0 =$ *distance between the centroids of chords*
$A_{ch} =$ *cross sectional area of one chord*
$I_{eff} =$ *the effective second moment of inertia of the built − up member*

First the design value of the maximum moment in the middle of the built-up member considering second order effects has to be calculated.

$$M_{Ed} = \frac{N_{Ed} * e_0 + M^I{}_{Ed}}{1 - \frac{N_{Ed}}{N_{cr}} - \frac{N_{Ed}}{S_v}} = \frac{7{,}669 * \frac{31.5}{500} + \frac{7{,}669}{3.5}}{1 - \frac{7{,}669}{91{,}654} - \frac{7{,}669}{151{,}645}} = 3{,}089kNm$$

$N_{Ed} =$ *the desing value of the compression force to the built up member*
$e_0 = \frac{L}{500}$
$M^I{}_{Ed} =$ *design value of the maximum moment in the middle of the built up member − without second order effects*
$N_{cr} =$ *the effective critical force of the built up member*
$S_v =$ *the linear stiffness of the lacings*

The effective critical force on the built up member is:

$$N_{cr} = \frac{\pi^2 * E * I_{eff}}{L^2} = \frac{\pi^2 * 2.1E11 * 0.00225}{31.5^2} = 4{,}691kN$$

$$I_{eff} = 0.5 * {h_0}^2 * A_{ch} = 0.5 * 2{,}000^2 * 112.3E4 = 2.25mm^4$$

$$S_v = \frac{n * E * A_l * a * {h_0}^2}{2 * d^3} = \frac{2 * 2.1E11 * 955.0E - 6 * 2.3 * 2^2}{2 * 2.3^3} = 151{,}645$$

After a number of iterations for each tube the cross sectional area per tube is found:

$$A_{chord} \geq \frac{2{,}304E3}{207.0} = 11{,}130mm^2 = 111.3cm^2$$

To keep the stresses at allowable level the cross surface of each tube has to be at least 111.3cm^2. Tubes with a diameter of 310mm and a wall thickness of 12mm will be used (112.3cm^2, NEN 2323).

Buckling of column as a whole where the buckling factor (K) is two:

$$P_{kr} = \frac{\pi^2 * E * I_{eff}}{(K * L)^2} = \frac{\pi^2 * 2.1E11 * 2.25E - 3}{(2 * 31.5)^2} = 11{,}750kN$$

The critical buckling load is smaller than the loads acting on a column (7,669kN). The last buckling check that has to be performed is failure of a main component segment.

$$I = \frac{\pi(310^4 - 286^4)}{64} = 12{,}491\text{cm}^4$$

$$P_{kr} = \frac{\pi^2 * E * I}{(K * L)^2} = \frac{\pi^2 * 2.1\text{E}5 * 12{,}491\text{E}4}{(1 * 2{,}310)^2} = 48{,}517kN$$

This value is larger than the maximum load on a single tube (2,304kN). Thus no buckling of the main component segment will occur.

For a single lacing system in opposite faces of the built-up member with two parallel laced planes the lacing should be arranged so that one is the shadow of the other. The number of lacings on a single side of the column with a length of 31.5m is 28.

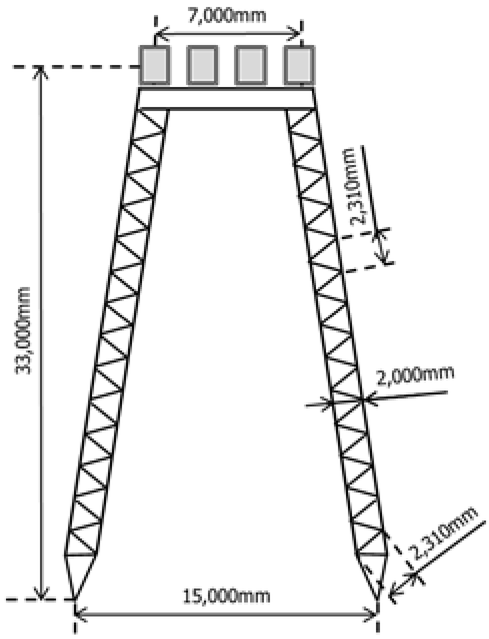


Figure 132: Back-mast

The own weight of the two masts and top section of 5m width of the back-mast with 20mt sheave nests is approximately:

$$8 * 7{,}850\big((31.5 + 5) * 111.3E - 4 + 28 * 2.31 * 8.93E - 4\big) + 20E3 = 49.1mt$$

Thus the estimated own weight of the back-mast (50mt) added to the maximum back-mast compression is close to the actual weight of the back-mast. Extra sheaves and stiffeners will be added, thus the actual weight will be close to the assumed 50mt.

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