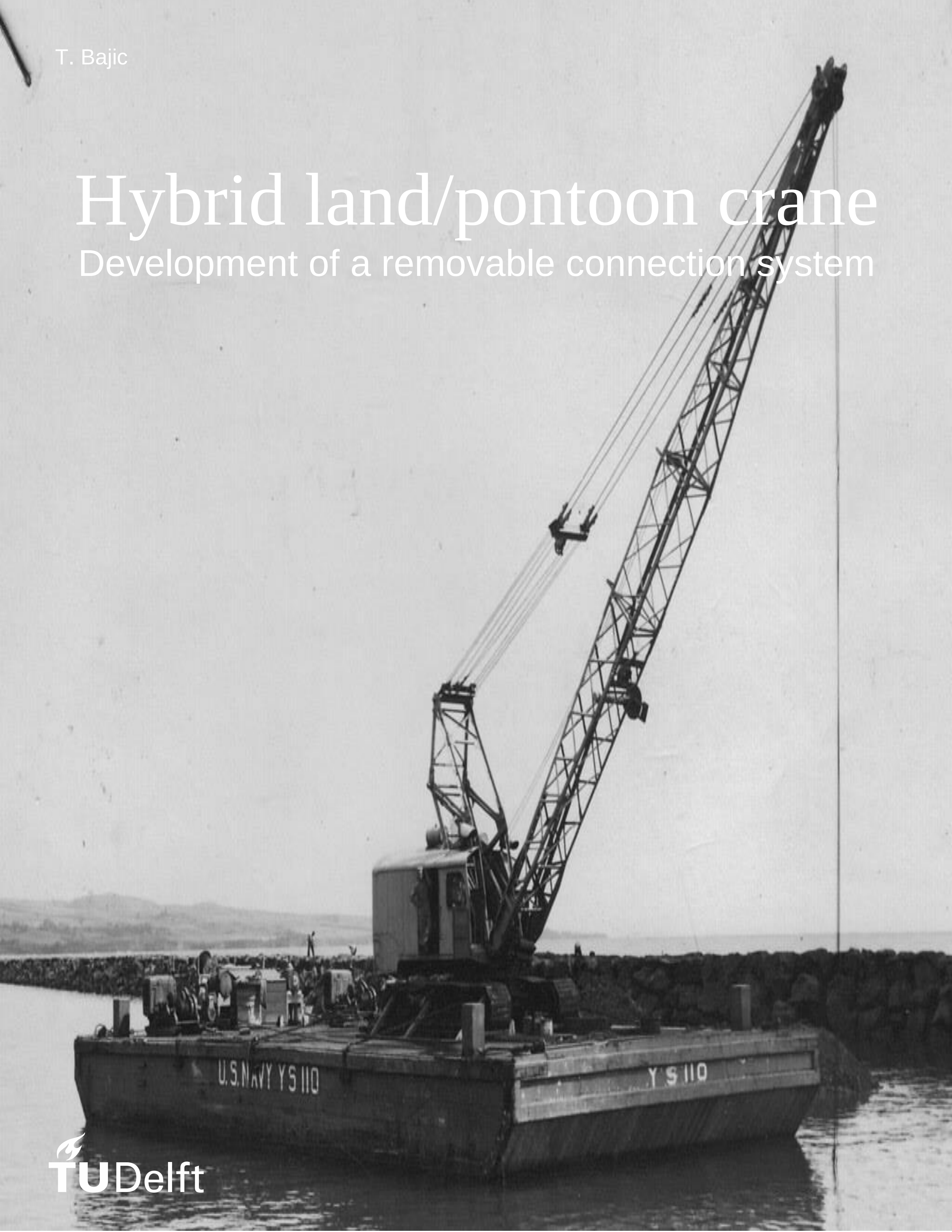
T. Bajic

Hybrid land/pontoon crane

Development of a removable connection system



TU Delft

Hybrid land/pontoon crane

Development of a removable connection system

By

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TU Delft

ABSTRACT

An unfamiliar subject to the contemporary connecting options for cranes on pontoons is removability; Various difficulties and obstructions related to the heavy-duty nature of the equipment lead to a limited amount of practiced conventional connecting options for this purpose, which all have the absence of removability in common. Furthermore, the consideration of various types of efficiencies and performance aspects has not been common practice within this field. The aim of developing a removable and efficient crane-to-pontoon connection system is therefore set.
An initial literature research is performed in order to obtain a variety of connecting options which have the potential of forming the basis of a new removable connection system, optimized for crane-to-pontoon configurations. Subsequently, creating a unique rating system, specifically for crane-to-pontoon connection systems, led to a substantiated selection process for the most feasible option among the potential connecting options. The turnbuckle option obtained the highest ranking and was therefore selected to proceed the design process with.
Developing the turnbuckle option into a complete connection system and accomplishing all defined aims led to an encounter with various engineering challenges. An integral design process led to the discovery of a proficient combination of components, which overcome the challenges and provide satisfaction with respect to the aims of the project.
The parameters of the developed conceptual design are finally quantified in order to prove feasibility and efficiency. Applicable design parameters are found which pass the safety requirements, while minimizing the material consumption. With these parameters, the removable design is compared to conventional real case connection systems in terms of cost-efficiency, which resulted in the observation that multiple €10^6 in long-term savings are anticipated per deployed crane due to the removability feature.

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GLOSSARY

Particular terms which might cause confusion are below defined for clarification.

Removable: Low time and effort required for connection and disconnection.

Connecting option: A basis principle of connecting used generally in engineering which can possibly be applied between crane and pontoon.

Connection system: The elaborated and adapted design of a connecting option and its elements integrated with the crane and pontoon.

Trim angle: A dynamical angle about the transversal axis (y-axis in Figure 1) of the floating body.

Heel angle: A dynamical angle about the longitudinal axis (x-axis in Figure 1) of the floating body (also known as list angle)

All other terms for the motions of the pontoon with crane are also directly adopted from standard ship motion definitions, as seen in Figure 1.

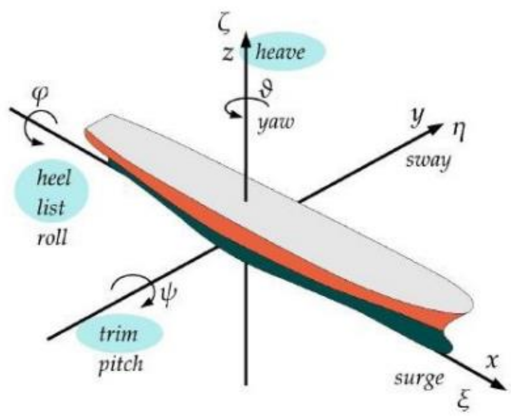


Figure 1: vessel motions [1]

1 INTRODUCTION

1.1 PROBLEM DESCRIPTION

In current engineering practice, cranes are designed specifically for their applications. However, occasionally it can occur that a land crane is deployed for operation on a floating pontoon. This can have various causes; A vessel might for instance not be able to dock the shore due to shallow water, or a particular shore might not be equipped with cranes [2]. Figure 2 shows a typical example of a land crane operating on a pontoon.

Figure 2: A land crane operating on a floating pontoon [3]

Due to the heavy-duty properties of these cranes and the environmental conditions on water, rotations (dynamical angles) are created on the floating body which have 5 main consequences for the crane with respect to stability and structural reliability. Figure 2 below illustrates the concept of these rotations. Note the distinction between the global axis (X,Y) and the local axis (x,y).

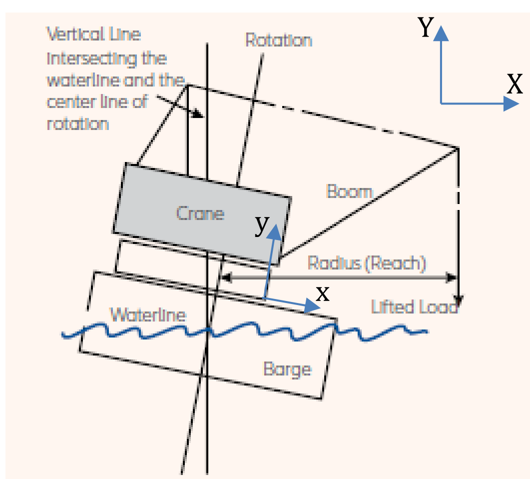


Figure 3: Rotation of the pontoon (barge) with crane on water [4]

The 5 consequences can be summarized in the following way [5]:

1. Lateral forces on the boom, superstructure and slewing mechanism are created.
2. The load radius with respect to the boom joints may increase, causing an increase in bending moments.
3. A gravitational force component on the crane in local horizontal direction (x-direction as indicated in Figure 3) is created, inducing the risk of crane shifts along the pontoon deck.
4. The center of gravity of the crane shifts in global horizontal direction (X-direction as indicated in Figure 3) while the support points of the understructure converge to each other on this axis, inducing the risk of tipping over of the crane.
5. The inertia of the crane during rotations can cause temporary misalignments in orientation between crane and pontoon, resulting in hits between the pontoon deck and the crane understructure, causing damage to crane components.

A number of measures are taken to mitigate the effects of these consequences [2] [5]. One of them is to implement a maximum permissible dynamical angle and adapting the pontoon design to this angle (particularly due to the 1st consequence). The manufacturer of the crane or a qualified engineer usually determines this permissible angle. Another important measure is to place friction material like wooden mats under the wheel tracks, limiting the crane's ability to shift. In order to ensure however that the crane moves along with the pontoon as one body, a sufficiently reliable connection between the crane and the pontoon is required. The following four options are available for this purpose (*United States Department of Labor*) [6]:

1. Direct rigid fixation between crane and pontoon using cables, chains, welding, bolting or similar.
2. Enclosing the crane area on the pontoon with barricade restraints that are rigidly fixed to the pontoon, which disable any movement of the crane in any direction.
3. Mounting the crane to a rail system on the pontoon with rail clamps and rail stops which can fully support the loads of the crane while enabling longitudinal repositioning if rail stops are removed.
4. A centerline wire rope system that can fully support the loads of the crane in all directions and that keeps the crane in the centerline of the pontoon while enabling longitudinal repositioning.

What these four connecting options have in common is that they cannot be simply removed and re-applied if necessary. This is a result of the magnitude of the forces associated with the dynamical angles. These connecting options are therefore generally applied in way that it can withstand high magnitude loads, which make them inconvenient to apply and to remove. The reasons for this are various; cables/ropes for instance require a long and intensive tensioning process in order to get them on a sufficient tension level while bolts in these situations are too large to handle manually and are furthermore not applicable on each crane part, requiring the addition of welding and/or barricading [2]. So, once a crane is deployed on a pontoon, it is generally considered as a permanent move. The crane will therefore not be available anymore for utilization on land when necessary, causing a major loss in the utilization value of the available equipment of the deployer. These conventional connecting options lack furthermore the science with respect to efficiency and simplicity.

1.2 AIM AND RESEARCH QUESTION

Considering the problem above, the aim of this thesis is to develop a crane-to-pontoon connection system that enables and facilitates a standard land crane to be utilized as a hybrid operational crane that can operate on a pontoon or a shore on demand, increasing the overall efficiency in port equipment utilization. This means that the list of the four conventional connecting options presented by the United States Department of Labor is aimed to be upgraded with a 5th option, which will be a generally removable connection system.
A corresponding main research question is formulated; **How can a removable crane-to-pontoon connection system be developed in order to turn a standard land crane into a hybrid operational crane that can operate on a pontoon or a shore on demand?** This question is divided in the following consecutive sub-questions:

1. What is the state of the art with respect to available removable connecting options which might be applied/adapted to the fixation of a land crane on a floating pontoon?
2. Which connecting option is most feasible and what are the associated criteria?
3. How can a conceptual design of the crane-to-pontoon connection system be created with the most feasible connecting option?
4. How to quantify design parameters?
5. How to quantify efficiency?

1.3 SCOPE

In order to prove feasibility and efficiency of the design of the connection system, the research is in collaboration with *IHC Engineering Croatia d.o.o.* [2] based on heavy-duty input sample data provided by the company. This includes crane data, environmental conditions and pontoon data, which confine the scope of this project. The details concerning this data can be found in section 5.4.

1.4 LAYOUT

The outline of the report starts, after the introductory chapter, with a chapter on the state of the art with respect to removable connecting options in the form of a literature research into potential connecting options. The subsequent chapter consists of an appointment of relevant performance aspects and indicators, an analysis into boundary requirements for the connecting options within these indicators and the final selection procedure for the most suitable option. The fourth chapter is dedicated to the conceptual model design of the new connection system. All of the main design parameters and characteristics are determined here and all mechanical design decisions and adaptations are here substantiated. Subsequently, the fifth chapter provides the structural analysis and safety evaluation of the system, including an input loads and motions analysis, leading to quantified design parameters which pass the safety requirements, followed by the sixth chapter, which gives the efficiency quantification and comparison of the design with respect to conventional real case connection systems. The report is thereafter concluded by reflecting on the gained knowledge with respect to the research questions and discussing the added scientific value with respect to efficiency of port equipment utilization due to the findings made throughout the research.

2 POTENTIAL CONNECTING OPTIONS

2.1 GENERAL

In order to answer the first research question, a literature research is performed to explore a variety of connecting options that have the potential of enabling simple connection and disconnection between crane and pontoon. The range of sources used purposefully exceeds the topic of cranes and pontoons only, in order to find new solutions which might be applied in this field. A provisionary analysis on the found options is performed with respect to their possibilities and potential difficulties without applying any ratings or rankings. All available information is laid down, so that a substantiated selection procedure can be performed in chapter 3.

2.2 FISH PLATE PIN CONFIGURATION

Fish plates can be designed in a way that allows them to be effective in multiple directions with simple pins. Figure 4 below shows a tower crane connection with its foundation on land.

Figure 4: Tower crane foundation connection with pins [7]

It can be seen that with two big pins and one smaller pin, the crane is prevented from shifting in any direction. A corresponding flange needs to be welded on a suitable place on the crane, which needs to have the holes configured in a way that allows the two big pins to be placed directly underneath each other while oriented perpendicularly to each other. With the smaller pin, the bigger pins can be connected to each other, which keeps them in place. In principle, there should be no external pressure on these pins when the crane is out of operation, so that they can be pulled in and out manually.

2.3 TURNBUCKLE WITH PIN

A turnbuckle is a device which is meant to be placed in between ropes or cables in order to apply the necessary tension on a particular fastening system, as can be seen in Figure 5 below.

Figure 5: A turnbuckle keeping two cables under tension [8]

It consists of an elongated frame with two threaded bolts going into both ends. The holes at these ends are threaded in opposite directions, so when the frame is rotated, both ends are tightened (or released if rotated in the other direction) [9].
Occasionally it can also occur that a turnbuckle is used for holding two pieces of equipment together, as seen in Figure 6 below.

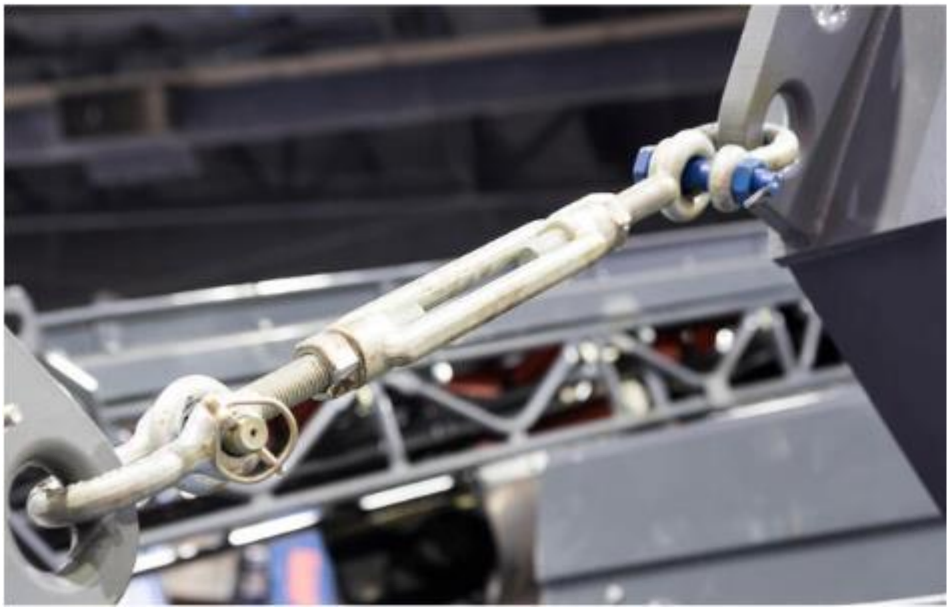


Figure 6: A turnbuckle used at a machine [10]

Turnbuckles are produced with various types of end fittings as seen in Figure 7 below.

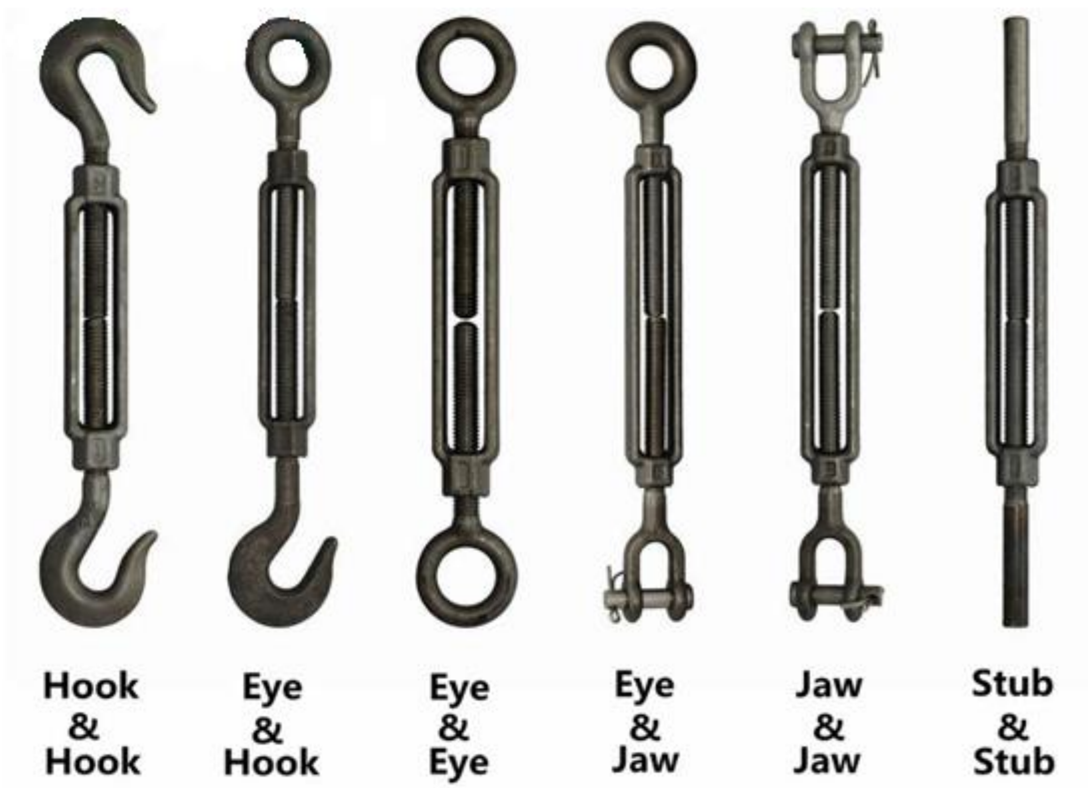


Figure 7: Different end fitting configurations [11]

The fitting that corresponds to the requirements of the potential crane to pontoon connection system would be the jaw & jaw fitting with cotter pins as seen in the figure, which would allow for a simple connection and disconnection. They are produced for any size and application type. Figure 8 shows an example of various sizes of turnbuckles with simple pins sold at *Rose brand* [12].

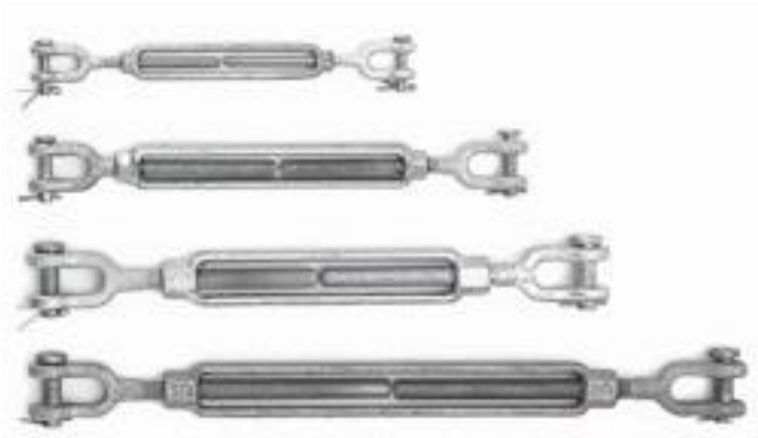


Figure 8: Turnbuckle with pin in different sizes [12]

A relevant advantage when applied for the new crane to pontoon connection system would be their variable length when rotating the frame. This means that it can be connected with the end of the pontoon to different cranes with different dimensions while not wasting any pontoon-space in between. This variable length also gives at the same time the opportunity for pre-tensioning the system while enabling comfortable placement.
As mentioned, turnbuckles are designed to take in tension stresses. If applied at the connection between crane and pontoon however, these devices will also be subject to compression stresses. Due to the slender frame that turnbuckles usually have, buckling is expected to arise as a problematic factor in this case. A compression test carried out on a turnbuckle (see figure 11) at the *UNC Charlotte* shows how it buckles relatively quickly with respect to its maximum tensile strength [13].

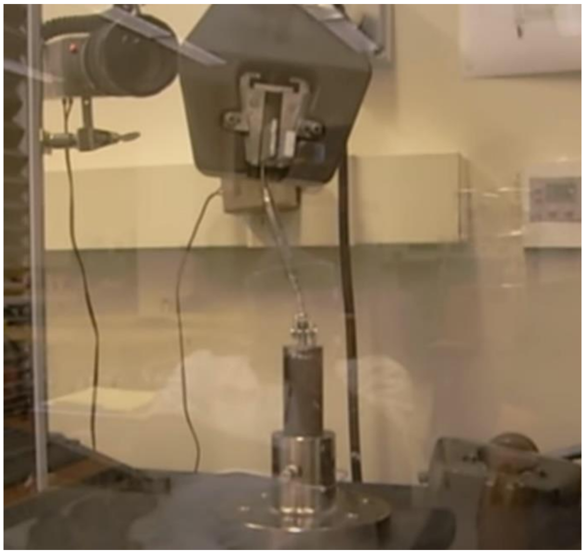


Figure 9: Turnbuckle compression test [13]

If turnbuckles are thus applied for the connection between crane and pontoon, a reinforcement against buckling will need to be found and implemented into the used turnbuckles.

2.4 SPRING CLIP TECHNOLOGY

Spring clips are mainly elementary constructions, consisting of spring material and a form of clamps. They are produced in a number of variants and are predominantly used in the millimeter domain. Various types of spring clips are displayed in Figure 10 below.

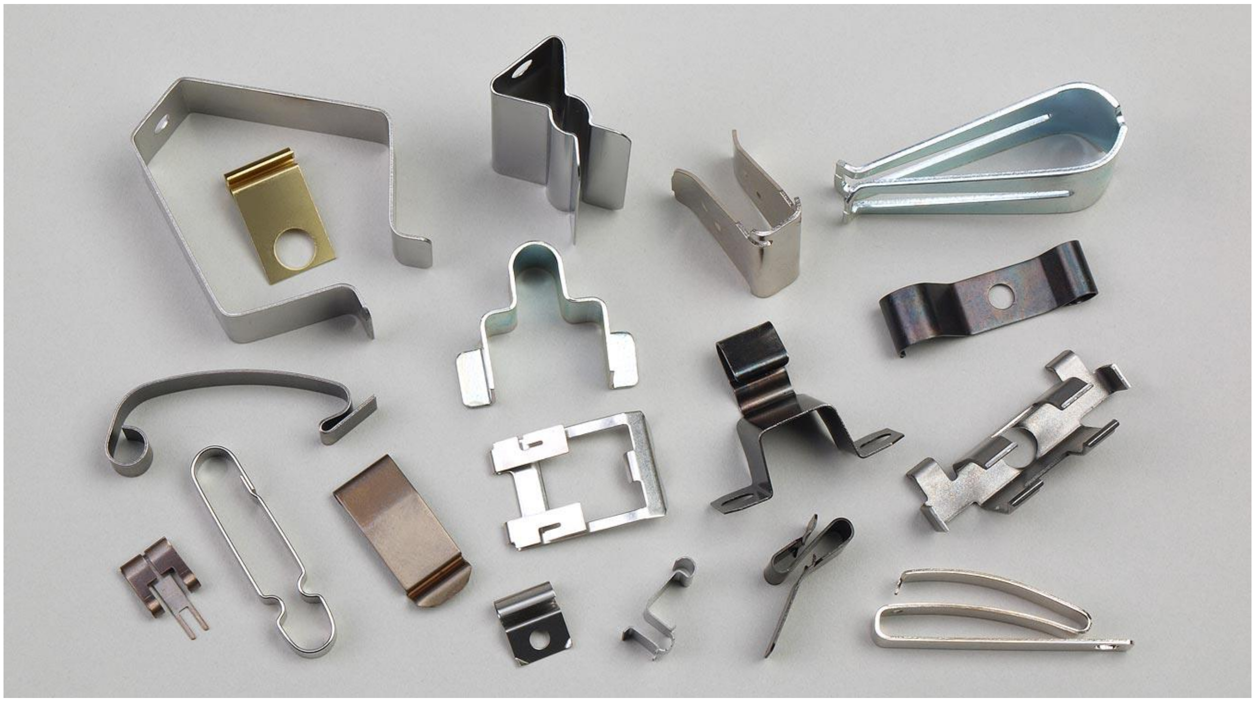


Figure 10: Various types of spring clips [14]

The main principle of any spring clip is based on a particular spring force which needs to be overcome in order to click the desired object onto the clip. Once the object is clicked, the clip returns to its initial position, blocking the object from moving away unless the same force is applied on the spring again. The shapes of these clips are usually designed in angular ways that allows spring force to be applied by simply pushing the object onto the clip. The shape can also decide whether pushing has no effect on the reverse side, ensuring a higher reliability fixation.
Böllhoff group [15], specialized in fastening technology, has been able to develop an ''innovative and high-strength quick fastening'' solution in number of variants, using spring clip technology. Various clicking combinations have been made available by the company, enabling ''permanent'' and ''removable'' spring clip connections. Figure 11 below shows one variant of these spring clips by Böllhoff. It has a black frame which encloses two separate connection sides and each side has its own steel clip element which can be pushed into the frame. As noted in the description, this variant features a removable as well as a permanent connection side. This can be seen by the shapes of the steel connection clips. The upper clip has a one-side angle, so once the hill has been surpassed, the plate cannot return in the same way. The lower clip has a two-sided angle, enabling simple disconnection in the same way as it has been connected. Even though this removable connecting principle appears to have potential for the new crane-to-pontoon connection system, it can be read that its usage is intended for small-scale applications within these products. Also, the given configurations would require a pontoon flange immediately next to the crane, resulting in the same space-related issues as pointed out for the fish plate pin configuration.

BÖLLHOFF

DST Double fastener 30x10

Features

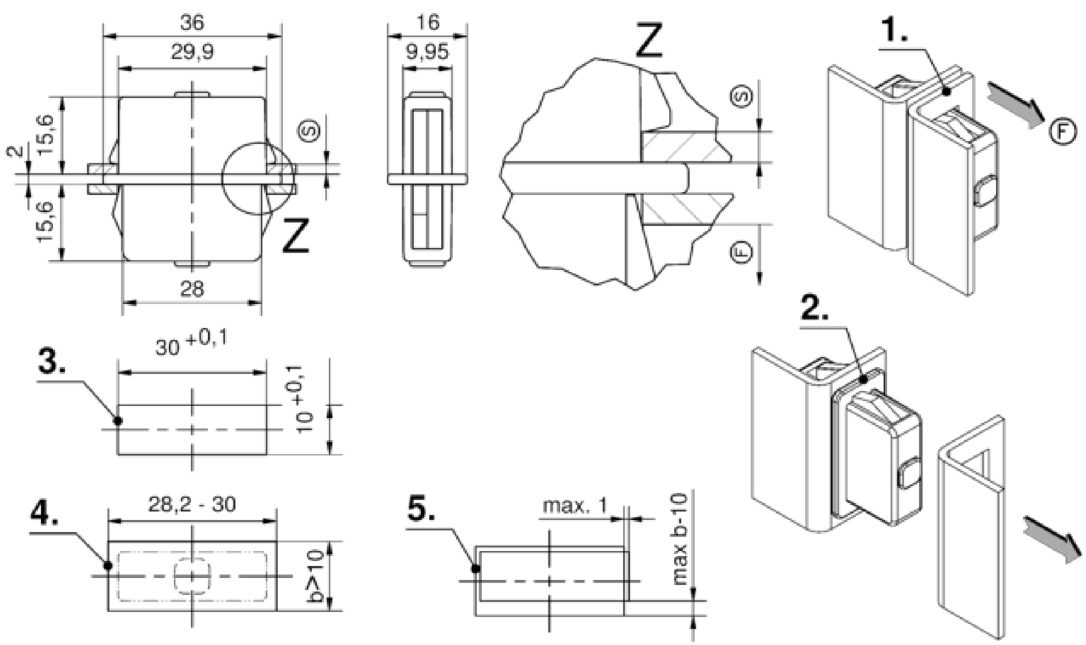
- Double fastener with snap technology for a one-sided fixed and a one-sided removable connection of two sheet metal panels.
- Tool-less assembly.
- Use on double-wall or canted non-sealed doors and sidewalls.
- Double fasteners hold door in place and prevent "rattling".
- The "fixed" installation opening is 30x10 mm.
- For the removable connection, the installation opening can be chosen from 28x10 mm to 30x16 mm.

DST

RoHS
WEEE

Materials

- Double fasteners: PA, black or GDZn, black
- Clip elements: sintered steel



Notes

1. Mounted
2. Pulled off
3. Installation opening for fixed side
4. Installation opening for removable side / tolerance compensation
5. Possible offset of the two installation openings

The sheet thickness (S) can be designed differently

F = Pull-off force

Fastener, GDZn, black

Product number	Design	Pulling force	Clamping range	Vibration resistant	Mounting type
D29590013315005	One-sided pull-off	~50N	2,0 – 2,3 mm	Yes	toolless

Figure 11: DST double fastener by Böllhoff [15]

Figure 12 below shows a disassemble of the DST fastener, showing the spring that is integrated with the steel clip inside the frame.

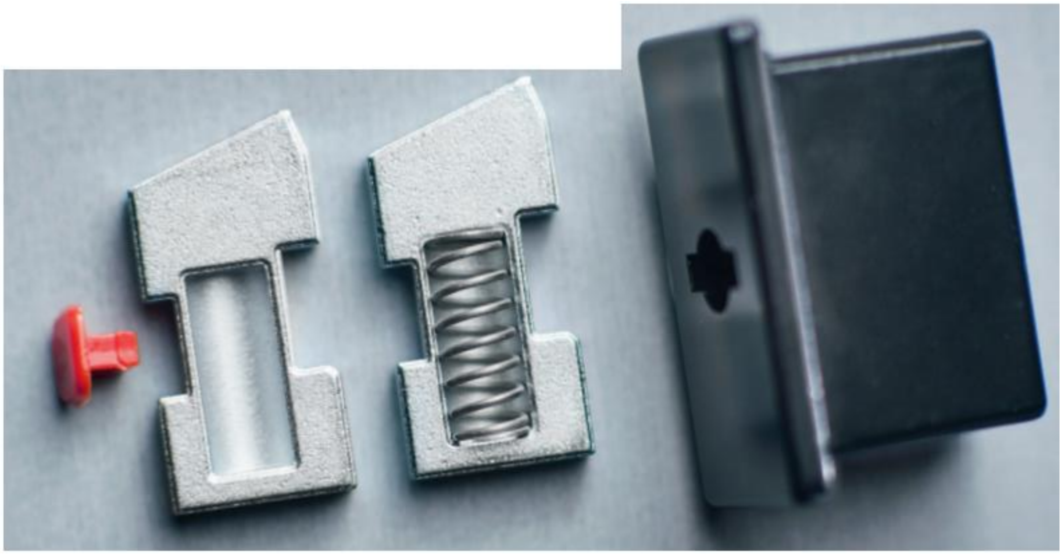


Figure 12: Disassemble of DST fastener [15]

2.5 SPRING CLAMPS

More variants of spring-based connections are possible. Mat clamps used for automotive purposes are another example as shown in Figure 13 below.

Figure 13: Automotive mat clamps [16]

Figure 14 shows how the spring is integrated inside the frame.

Figure 14: Spring inside mat clamp [16]

Their advantage would be that their pedal-like shape would allow for the spring to be engaged by feet, if they would be fixed to the pontoon deck. This would be helpful due to the heavy-duty properties that the spring would have if applied in the large-scale domain as with the crane. As, with the DST fastener, these pedals would have to be placed close to the pontoon area. Steel rods for clamping would need to be welded on the crane, which must not significantly increase the total crane width. Unfortunately, there are no examples to be found of these type of spring clamps applied at that scale. Other spring clamps however can be found at heavy-duty lifting applications. Figure 15 shows an example of heavy-duty lifting clamps. A linear spring is visible on the top part of the frame.

Figure 15: Heavy-duty lifting clamps [17]

The spring in these clamps is typically engaged simply with a lever. Locking the lever clamps the object firmly in all directions within the working capacity. The object is released again by simply reversing the lever. Some lifting clamps connect by simply pushing the object into the jaws, which can be seen as a form of spring clip technology [18]. Figure 16 shows in detail a lifting clamp with its locking lever and the black frictional surface clamp connected to the spring.

Figure 16: Lifting clamp with locking lever [19]

Since these clamps are designed for lifting purposes, it can be assumed that their capacity is rated only for outwards forces. It is therefore unknown which structural response can be anticipated when inwards forces are applied (which would be the compression forces if applied between crane and pontoon). Unfortunately, their frames resemble in most cases a circular design which have relatively short lengths from end to end without much variation, which would require a pontoon flange nearby the crane area that would be usable for single crane dimensions.

2.6 Bar clamps

Bar clamps are generally used at workspaces and are able to secure any piece of material in between two surfaces. Figure 17 below shows a typical example of bar clamps.

Figure 17: Bar clamp example [20]

As displayed in the figure, it consists out of a long steel frame with two movable surfaces (blue). One of the surfaces is movable simply by hand and can be pinned down at the desired position on the frame. The other surface is connected to a threaded rod which goes through a threaded hole. This surface is thus moved by rotating the rod. If an object is placed in between the surfaces, a clamping force can be exerted on the object by rotating the rod as far as possible. As with the spring clamps, the crane would also need to be equipped with an additional welded rod to its frame for connection with bar clamps.
Various configurations of this connecting principle are available. The surfaces can for instance be made in custom shapes, giving a more advantageous clamping, as can be seen with the golf club clamp in Figure 18, designed for circular rods.

Figure 18: Golf club clamp [21]

The movement of the surface is realized in this case with a handle configuration that converts rotation about a fixed point into linear movement of the clamping surface.

Even though clamping an object with these tools would go relatively comfortably, this type of connection strategy would involve a number of issues if applied between crane and pontoon. Analogously with previously described connecting options, the tool would have to be fixed on the pontoon immediately next to the crane, which would obstruct future placement of bigger equipment. Also, since these tools are in principle not designed to take on external stresses, it is questionable for which type of loads they are suitable. Considering the presence of a longitudinal rod, buckling might as well arise here as a problem. Furthermore, the tool consists out of a relatively large number of components and each of them would have to be separately analyzed and verified with respect to this stress suitability.

2.7 SCISSOR LINKAGE

Scissor mechanisms are widely used due to the functionality of longitudinal extension. Figure 19 shows a typical scissor mechanism application at a wall light, enabling the lamp to be moved at a desired distance from the wall.

Figure 19: Scissor mechanism applied at a wall light [22]

The mechanism consists out of a number of adjacent identical members that cross each other. At each intersection, pin connections are applied, allowing the members to rotate with respect to each other and thus the total length of the mechanism to be adjusted with simple axial force. That feature would not be suitable when used between crane and pontoon. However, a potential option would be to replace the pin connections with weld connections, converting it into a rigid scissor construction and connecting it with simple cotter pins at both ends. As with the turnbuckle option, the advantage of this type of linkage would be that it can connect the crane directly with a distant point on the pontoon, leaving the pontoon-space unedited in between. In addition, its structure has a form of protection against buckling in contrary to the turnbuckle, because the diagonal orientation of the members reduces axial compression forces and the intermediate connections reduce the effective buckling lengths. Moreover, the mass is distributed distantly from the axial center of mass, giving a higher radius of gyration. This makes it at the same time lightweight in comparison to a conventional beam. It also has better manual gripping possibilities and easier connectivity with pins at its ends.

Scissor structures are commonly used at various heavy-duty applications as well. Figure 20 below shows a scissor mechanism applied at a car lift.
The disadvantage of the scissor with respect to the turnbuckle would be the lack of a finished off pin connection fitting at the ends. Simply drilling holes through the endpoints of the structure and placing pins through it could lead to excessive stress concentrations.

Figure 20: Scissor car lift [23]

2.8 SUMMARY

A collection of 6 connecting options are found which have characteristics that can potentially satisfy the aims of this project. Each of the options has its unique essence and its downsides. The different variants within each option are discussed as well in order to find the most suitable products that are currently available. In the upcoming chapter, a more extensive analysis is performed over each of the options to find out which one has the optimal suitability to base the connection system design on.

3 CONNECTING OPTION SELECTION

3.1 GENERAL

The subsequent task after completion of the potential connecting option analysis with respect to the first research question is to construct a selection procedure, with respect to the second research question, for the most feasible connecting option to design the new crane to pontoon connection system with. The first step is to find all relevant aspects regarding this system. Thereafter, the performance indicators within these aspects are defined. These are used to assess the potential options individually and to subsequently compare them against each other in the relevant aspects. Before a sensible assessment can be done however, an investigation has to be made on criteria within these performance indicators and a grading scheme has to be defined. Each connecting option can then receive an independent assessment for each of its respective indicators based on the found criteria. Afterwards, a final ranking and selection can be made with the acquired data.

3.2 RELEVANT ASPECTS AND INDICATORS

By analyzing various connecting options in chapter 2, provisionary insight is gained in important factors for a new connection system. These are discussed already in that chapter and are included into the list of relevant aspects. All other aspects associated with the main objectives and scientific purpose of the assignment are added, which leads to a final list of relevant aspects which is discussed in the sections below.
The state of proficiency within the requirements of the aspects is indicated with the variables expressed below each aspect. These are the dotted items, which represent the performance indicators, followed by a brief description after the minus sign.

3.2.1 Safety

It is of highest importance that the developed system is capable of ensuring the necessary structural reliability for safe and durable utilization. It must be therefore be assessed whether the potential connecting option will be able to carry out the main purpose of the connection system, which is ensuring that the crane and pontoon move as one body at all times. It must be checked for each connecting option if it can potentially be designed for the load magnitudes associated with crane action on a pontoon. All potential failure risks must be taken into consideration in order to make a preliminary safety assessment of the options. Connection elements might fail because they are not designed for the occurring force levels or for the occurring stress states in this problem. Also, particular connecting options might not restrict all motions in every degree of freedom, which would lead to crane movement and damage. Furthermore, because the occurring forces in this problem are of dynamic nature, excessive oscillations of the stress levels could occur on the connection points. The clearances at these points can therefore be reason for accelerations, hits and damages which could ultimately lead to failure.
If these risks are eliminated, the crane will move along with the pontoon as one body at all times and the connection system will remain unharmed, which are the conditions for safety. This leads to the three performance indicators below.

- *Force resistance (kN)*

  - The achievable force resistance of the removable connecting element, based on a variant with a feasibly applicable size.

- *DoF restriction (yes/no)*

  - Whether all degrees of freedom restricted by the connecting option.

- *Preload (yes/no)*

  - Whether the connection can be properly preloaded in order to eliminate clearances.

3.2.2 Space-efficiency

The necessary contribution in practice by the new connection system requires that the consumption of space is optimized. The system must not obstruct any future placement of other equipment once the connected crane has been deboarded. This means that ideally the pontoon connection point elements are placed at the edges of the pontoon, so that the entire pontoon deck surface remains available at all times. Also the crane must remain deployable in the same way as it was before being used on the pontoon. The increase in total crane area due to the crane connection point elements must therefore be limited to a minimum.

- *Area of pontoon elements (m^2)*

  - The total pontoon deck surface area permanently consumed by the connection system.

- *Maximum available area (m^2)*

  - The maximum rectangularly shaped area on the pontoon that is deployable after installation of the pontoon elements.

- *Width of crane elements (m)*

  - The width of the crane connection point elements which are measurable from the top-view of the crane.

3.2.3 Cost-efficiency

The new connection system does not deliver a valuable contribution in practice if the costs of the system exceed particular boundaries. The aim of the project is to develop a system that is applicable widely by as many deployers as possible. There should ideally therefore not be any deployer financially restrained in purchasing and utilizing the system.

- *Price (€)*

  - Preliminary estimate of total costs for purchasing the connection elements.

3.2.4 Manageability

The purpose of the project is to enable simple connection and disconnection, ideally by hand. This would maximize its practical contribution and scientific value. The final system must therefore not lack in the aspect of manageability. This means that it needs to be assessed how comfortable and time-consuming the connection and disconnection will be with each option. Also, the mass and size of the final removable elements is relevant here, since it influences the amount of effort necessary for applying and transferring the device. Furthermore, it needs to be checked if the option can be applied manually or additional tools will be required.

- *Time of application (t)*

  - The accumulated time needed for applying the connection and to remove it.

- *Number of personnel (-)*

  - The amount of construction workers needed to successfully establish the connection.

3.2.5 Adaptability

After the deficiencies of each option have been defined in various aspects, the question remains whether the connecting option can easily be mechanically adapted or modified in order to eliminate these deficiencies. An option with a relatively big amount of deficiencies, but relatively high adaptability might therefore be a better choice than an option with less deficiencies but low adaptability. This aspect is rated by the anticipated level of engineering complexity needed to make the design satisfy all requirements and also the final fabrication complexity in practice of the anticipated adaptation.

- *Engineering complexity (-)*

  - A custom rating for the engineering complexity of the necessary mechanical adaptations to eliminate the deficiencies of the connecting option.

- *Fabrication complexity (-)*

  - A custom rating for the fabrication complexity of the anticipated adaptations.

3.3 CRITERIA

In order to be able to use the performance indicators as defined in section 3.2 for assessment of the connecting options, particular criteria need to be found. After an indicator has received a certain value, it is compared against the corresponding criteria in order to assign a rating to the indicator for each connecting option.
There are four possible ratings that the value of an indicator can receive, which are graded with points in the following way:

Rating	Points
Satisfactory	3
Acceptable	2
Marginal	1
Unacceptable	0

There are 4-levelled and 2-levelled rating classes used. 4-levelled classes are applied at the indicators which allow for compromise at their rating and when there is made use of estimations. 2-levelled classes are used at particular safety-related indicators which do not have the possibility of compromise in their rating. This is further elaborated per indicator in the sections below.
The next task is to substantiate the boundary values for each indicator that determine which rating should be assigned to a particular indicator value. By adding up the points received for each indicator, a total amount of points can be determined for each connecting option and a final ranking of the options can be established.
These values are based on preliminary estimations at this stage of the research for the purpose of finding the optimal approach for the new connection system design.

3.3.1 Reference configurations

The following 5 reference crane-to-pontoon configurations are found which can be used to extract reference data when necessary.

3.3.1.1 Reference 1 (crane barge Marinos)

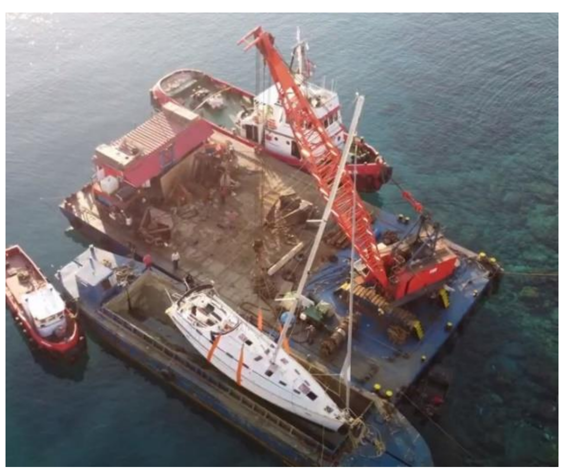


Figure 21: Crane barge Marinos [24]

This crane barge has been secured to the pontoon with yellow brackets at the ends of both track wheels as seen in Figure 21.

3.3.1.2 Reference 2 (ICOP SpA)

Figure 22: Liebherr crane on pontoon [25]

A Liebherr crane utilized on a pontoon by the Italian construction company ICOP SpA has been secured with barricades at each side of both track wheels, as well as with sets of three cables at the side of each track wheel as seen in Figure 22.

3.3.1.3 Reference 3 (Hitachi Sumitomo)

Figure 23: Hitachi Sumitomo pontoon crane [26]

A Hitachi Sumitomo crane has been secured to a pontoon with the help of transversal and longitudinal barricades as seen in Figure 23.

3.3.1.4 Reference 4 (TOMH crane Lima)

Figure 24: TOMH crane Lima [27]

The crane depicted in Figure 24 is connected to the barge similarly as in example 3. However, there are no barricades in front of the crane, which means that the front trim has been considered neglectable because loads take place for the most part at the aft part of the pontoon when the crane is positioned there. The barricades placed behind the crane are therefore relatively big if compared to previous examples. The transversal barricades are in single piece here from one side to the other, lowering the pontoon area consumption.

3.3.1.5 Reference 5 (floating crane Amfitriti)

Figure 25: Floating crane Amfitriti [28]

This crane, seen in Figure 25, has been secured with big barricades at the back and smaller barricades in front. In addition, four chains at each side are applied for restriction in transversal direction. A connection flange is applied on the pontoon surface for each of the chains separately.

3.3.2 Force resistance

To determine boundary values for the necessary force resistance, an estimation on the maximum force levels occurring in the given problem is made.
In order to find the exact values of the maximum occurring loads on the connection system with the given input data, an input loads analysis is carried out in detail in chapter 5. However, the approximate scale-size of these loads can be determined in advance by simplifying the problem. Instead of considering each 5 of the consequences listed in section 1.1, only consequence 3 is calculated with, which is related to the gravitational force component on the crane in the direction along the pontoon (local x-direction in Figure 26) due to the dynamical angle. The 5 consequences are listed once more below.

1. Lateral forces on the boom, superstructure and slewing mechanism are created.
2. The load radius with respect to the boom joints may increase, causing an increase in bending moments.
3. **A gravitational force component on the crane in local horizontal direction (x-direction as indicated in Figure 3) is created, inducing the risk of crane shifts along the pontoon deck.**
4. The center of gravity of the crane shifts in global horizontal direction (X-direction as indicated in Figure 3) while the support points of the understructure converge to each other on this axis, inducing the risk of tipping over of the crane.
5. The inertia of the crane during rotations can cause temporary misalignments in orientation between crane and pontoon, resulting in hits between the pontoon deck and the crane understructure, causing damage to crane components.

Generally, wooden mats are placed below the crane tracks in order to counteract this force with friction. However, as the pontoon floats on water, these mats are likely to get wet, which reduces the friction coefficient. Also, the level of surface smoothness/wear of the crane track steel might cause a further drop in friction. The effective friction coefficient might in practice therefore be relatively low. Since the exact value is uncertain, the calculation is performed with a friction coefficient of zero as a safety factor.
Connecting options that are not able to resist this force level receive an unacceptable safety rating. A marginal rating can be assigned if the maximum resistance is around the same value as the approximate force level and acceptable if it is significantly higher than the force level. A margin of 5% will be implemented for this purpose, since this is a standardly applied percentage for tolerances used at vessel strength calculations at IHC [2]. A satisfactory rating will be given if there is no particular limit to the maximum stress resistance of the connecting option.
All the relevant stress states must be considered for each option when analyzing this resistance. In principle, there are 6 different stress states possible on the crane to pontoon connection system. These are: tension stress, compression stress, shear stress, bending stresses (about both axis) and torsion. The combination of occurring stress states needs to be identified firstly for each connecting option. The occurring stress state which is resisted the least by the analyzed option is therefore the rated stress state in this indicator.
The necessary data for this simplified load calculation is the maximum dynamical angle and the total weight of the loaded crane. The angle according to the input data (see section 5.4.1) is not higher than 4° in operation and the weight of the crane including its maximum capacity is (350t+16t=366t)*9.81 = 3590.46kN. The sine of this angle is multiplied with the weight to obtain

the induced gravitational force component in local x-direction, resulting in: sin(4)*3590.46kN ≈ 250kN. Figure 26 below is used to conceptually illustrate this loading case.

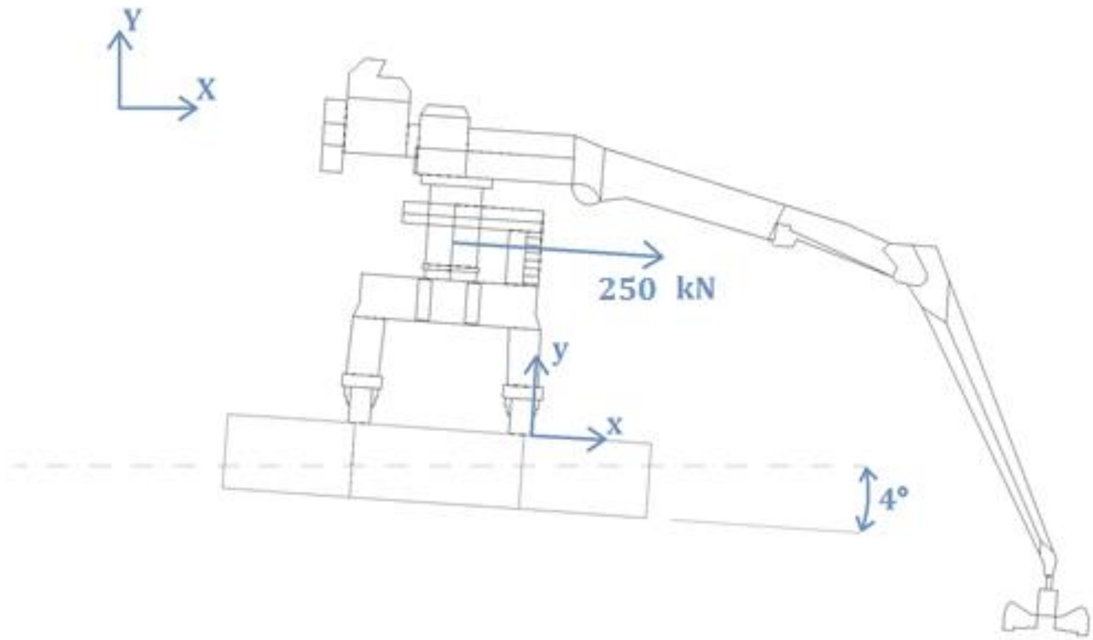


Figure 26: Sketch of gravitational force component on crane

The final table for the boundary values of the stress resistance indicator can thus be defined as follows:

Rating	**Maximum force resistance**
Satisfactory	>> 250kN
Acceptable	> 250kN + 5%
Marginal	250kN ± 5%
Unacceptable	< 250kN -5%

Table 1: Force resistance boundary value scheme

3.3.3 DoF restriction

The newly designed connection system must ensure that the crane moves along with the pontoon as one body. The forces and motions in operation and navigation can lead to translations and rotations relative to the pontoon in all possible directions. This means that all 6 degrees of freedom need to be restricted by the connection system. Only if this is the case, the system can comply to the necessary safety requirements. There is therefore no distinction in rating between a system that restricts 5 degrees of freedom or only one degree of freedom, since both are simply deficient. The boundary value scheme is therefore limited to only a satisfactory or an unacceptable rating.

Rating	**Amount of restricted DoF**
Satisfactory	6
Unacceptable	< 6

Table 2: DoF restriction boundary value scheme

3.3.4 Preload

Since the connection system is intended to be applied repeatedly and manually, a clearance fit is assumed at the connection points [29]. Once connected, this clearance is preferably taken away in order to eliminate uncontrolled collisions/damage between connecting elements at the connection points when loads start to occur on the crane and to ensure rigidity, such that the crane moves along with the deformation of the connection elements only and is prevented from gaining any momentum relative to the elements, which would lead to higher stresses on the system. This can be achieved by applying preload on the system. It is therefore investigated for each connecting option whether it is possible to apply preload with it. An option in which preload application is enabled receives a satisfactory rating. An option that does not provide this possibility is not immediately considered unacceptable, since a direct safety risk is not necessarily induced by it. A marginal rating is rather given, leading to the boundary value scheme below.

Rating	**Preload applicable**
Satisfactory	yes
Marginal	no

Table 3: Preload boundary value scheme

3.3.5 Area of pontoon elements

After a connecting option has been selected and applied, a design of the connection system will be made including the connection elements that will be permanent part of the pontoon. This could be in the form of a flange, a bracket or similar. The original total available area of the pontoon will thus be reduced permanently to a particular extent by the connection system. The aim is obviously to limit this reduction as much as possible.

The exact value of the reduction in area cannot be determined before a final design has been made. Preliminary estimations for each connecting option therefore need to be established in order to rate the options in this performance indicator.

In order to find the right balance for the boundary value scheme, the amount of area occupied by the crane with respect to the total pontoon area needs to be taken as a relevant factor, because a crane that consumes a relatively big portion of the pontoon area is generally in need of a connection system that also consumes a relatively big portion and vice-versa; the magnitude of the forces is related to the crane-size and also the pontoon size, because a bigger pontoon generally has lower dynamical angles.

According to the input data, the area of a track wheel of the crane is 9.5x1.0m and the distance between the two tracks 7.6m. These dimensions are illustrated together with the known pontoon dimension in Figure 27 and Figure 28 below which show the front-view and side-view respectively of the crane and pontoon.

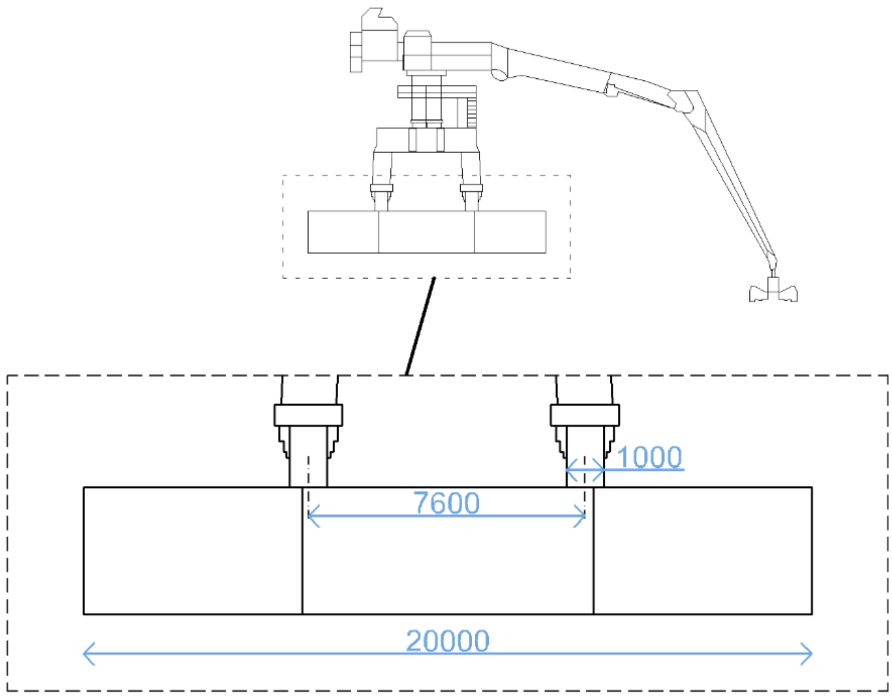


Figure 27: Front-view of crane and pontoon with track wheel area detail

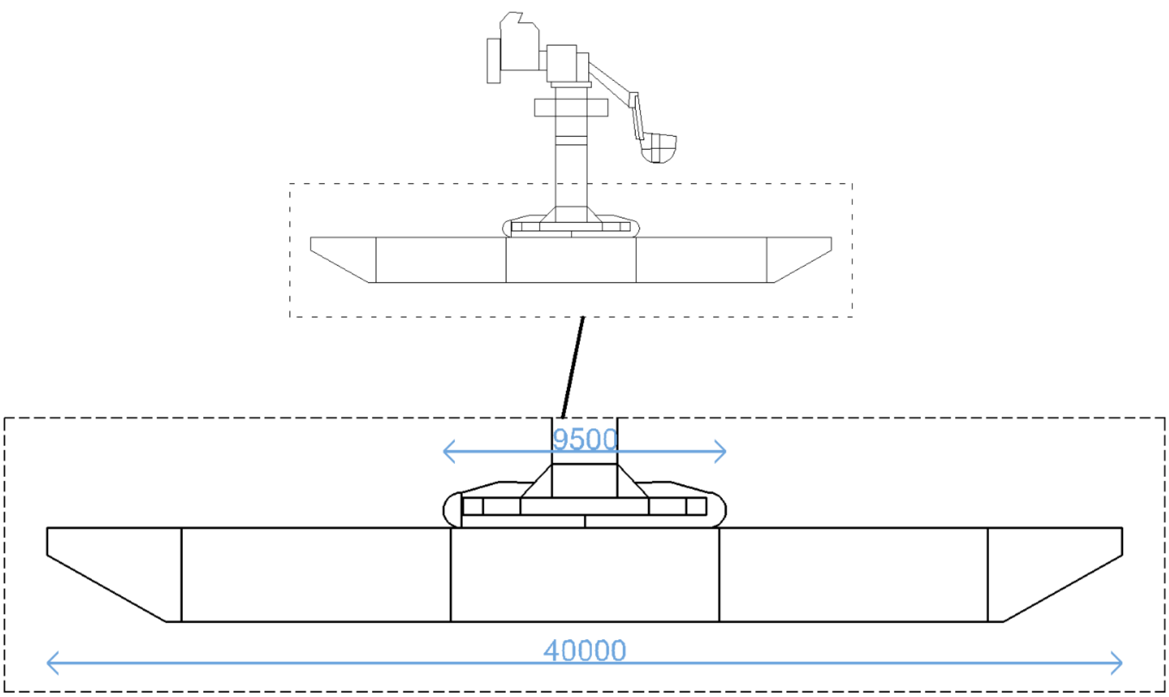


Figure 28: Side-view of crane and pontoon with track wheel area detail

The total pontoon area consumed by the crane can thus be deduced by taking the distance between crane tracks and adding half of the track width on the left and the right side (see Figure 27), which will give the total width = 7.6m + 2*($^1/_2$* 1m) = 8.6m. Multiplying this with the length gives the total

area = 8.6m * 9.5m = 81.7m^2. The total pontoon area is equal to 20m * 40m = 800m^2. This means that the crane occupies roughly 10% of the pontoon area in this case.
The other ratio which needs to be defined is the area of pontoon elements relative to the crane-size. These ratios will be used with the following formula to find a sensible area consumption ratio which needs to be minimized:

$$\frac{{pontoon\ elements\ area}/{crane\ area}}{{crane\ area}/{pontoon\ area}} =..\ specific\ area\ consumption \quad (1)$$

It can be seen that a bigger crane area scales the ratio to the second order. This is applicable because a bigger crane size means bigger loads which require bigger connection elements, but also bigger dynamic angles, which further increase the loads on the elements. A smaller pontoon size would in principle only increase the loads due to bigger dynamical angles, so that it scales the ratio only to the first order.
Since this is a custom formula, there is no data available yet for judgement of the specific area consumption values. In order to define sensible goals for this consumption, the reference configurations are used and their dimensional proportions. The boundary value scheme for this performance indicator is based on the approximate specific area consumption values in these examples.

3.3.5.1 Reference 1 (crane barge Marinos)

The yellow brackets (see Figure 21) are fixed to the pontoon and are therefore considered as the pontoon elements in this configuration. It can furthermore be assumed that these brackets are rigidly welded to the crane, since there are no other connection elements to be found in this configuration which constrain the crane in transversal direction. The specific area consumption in this example is calculated with formula (1). The two relevant ratios of this formula are determined through measurements on the available footage.

$$specific\ area\ consumption = \frac{{pontoon\ elements\ area}/{crane\ area}}{{crane\ area}/{pontoon\ area}} \approx \frac{0.075}{0.0625} = 1.2$$

3.3.5.2 Reference 2 (ICOP SpA)

The pontoon flanges belonging to the cables as well as the barricade brackets (see Figure 22) are taken into account as the pontoon elements in this case. The following values are obtained with the measurements:

$$specific\ area\ consumption = \frac{{pontoon\ elements\ area}/{crane\ area}}{{crane\ area}/{pontoon\ area}} \approx \frac{0.08}{0.075} = 1.067$$

3.3.5.3 Reference 3 (Hitachi Sumitomo)

The pontoon area taken up by all barricades added together (see Figure 23) is considered here as the pontoon elements area. The pontoon size in this example is relatively small compared to the previous examples (see [26] for full footage), leading to the following values for formula (1):

$$specific\ area\ consumption = \frac{pontoon\ elements\ area/_{crane\ area}}{crane\ area/_{pontoon\ area}} \approx \frac{0.084}{0.111} = 0.76$$

3.3.5.4 Reference 4 (TOMH crane Lima)

The following values are obtained by inspection of the barricades seen in Figure 24:

$$specific\ area\ consumption = \frac{pontoon\ elements\ area/_{crane\ area}}{crane\ area/_{pontoon\ area}} \approx \frac{0.0683}{0.16} = 0.43$$

3.3.5.5 Reference 5 (floating crane Amfitriti)

The big barricades, small barricades and the chain connection flanges as seen in Figure 25 are taken into account here to determine the consumption.

$$specific\ area\ consumption = \frac{pontoon\ elements\ area/_{crane\ area}}{crane\ area/_{pontoon\ area}} \approx \frac{0.1365}{0.075} = 1.82$$

The found range for the specific area consumption within these 5 reference examples is {0.43, 1.82} with a mean value of 1.0554. Since one of the project-based goals is that the utilized pontoon remains usable for any other application besides operation with the currently applied crane, the aim for the area consumption is a lower value than the average found from these examples in practice. A value around this average is therefore considered marginal for this project. The satisfactory rating is based on the lower bound of the found range of values, which leads to the boundary value scheme below for the specific area consumption. Since the crane and pontoon area have already been determined with the used input sample data at the start of this section, the only remaining variable is the performance indicator, the area of pontoon elements, whose corresponding value is displayed in the third column, with the crane area of 81.7 m^2 and the crane to pontoon ratio of 0.1 inserted in formula (1).

Rating	Specific area consumption	area of pontoon elements
Satisfactory	≤ 0.4	$\leq 3.3\ m^2$
Acceptable	$< 1.05 \ \wedge\ > 0.4$	$< 8.6\ m^2 - 5\% \ \wedge\ > 3.3\ m^2$
Marginal	1.05	$8.6\ m^2 \pm 5\%$
Unacceptable	> 1.05	$> 8.6\ m^2 + 5\%$

Table 4: Area of pontoon elements boundary value scheme

3.3.6 Maximum available area

Another aim with respect to space-efficiency is that the deployable area on the pontoon after installation of the pontoon elements is maximized. A rectangularly shaped area is considered as deployable since most equipment have rectangularly shaped undercarriages. The biggest empty rectangle on the pontoon after installation of the elements is thus considered here as the maximum available area.
Ideally, the pontoon elements are placed at the edges of the pontoon with the width of the elements close to zero, so that the biggest possible rectangle is close to the pontoon size. In the worst case, there is no empty space left on the pontoon after installation of the elements and placement of the crane, so that the smallest possible available rectangle is close to the crane size. A linearly distributed rating scheme can be applied in between these two extremes ({81.7m^2, 800m^2}), leading to the following boundary values:

Rating	**Maximum available area**
Satisfactory	≥ 620 m^2
Acceptable	≥ 440 m^2 ∧ < 620 m^2
Marginal	≥ 260 m^2 ∧ < 440 m^2
Unacceptable	< 260 m^2

Table 5: Maximum available area boundary value scheme

3.3.7 Width of crane elements

The deployed crane will need to be permanently equipped/expanded with particular elements, so that it can be linked with the connecting option to the pontoon elements. The relative increase of the width of the crane due to these elements is considered relevant. The elements are assumed to be fixed to the steel sides of the track wheels, so that the total length of the crane remains unchanged.
This increase is acceptable up to the point where the cranes applicability at other future occasions is compromised. These boundary values need to be estimated here; If the total width of the crane increases with 25% due to these elements, it can definitely be considered as unacceptable. Any increase less than 5% on the other hand can be considered as satisfactory. The boundary values below will be used, given the input crane width of 8.6m. The percentages relative to the crane width are given in the parentheses next to the values.

Rating	**Width of crane elements**
Satisfactory	≤ 0.43m (5%)
Acceptable	≤ 1.29m (15%) ∧ > 0.43m (5%)
Marginal	≤ 1.72m (20%) ∧ > 1.29m (15%)
Unacceptable	> 1.72m (20%)

Table 6: Width of crane elements boundary value scheme

3.3.8 Price

A provisionary estimate of the necessary budget for purchasing the materials for the system will be determined for each connecting option. The aim for the price of the elements is set to be less than 1% of the estimated crane price. If the costs exceed 5% of the crane price, it is considered as

unacceptable. The price of the reference crane must therefore be estimated firstly. According to *Charter Capital* (Equipment financing consultant) the prices for crawler cranes vary within the range from $ 1,000,000 - $ 5,000,000 [30]. In order to find a suitable point within this range for the used reference crane, a linear correlation between size and price is assumed. The approximate maximum and minimum crawler crane sizes is linked to the upper and lower bound respectively of the found price range. Through interpolation a price value for the reference crane is estimated.
The company *Liebherr* [31] presents its LR 13000 as ''the most powerful conventional crawler crane in the world''. The area of the wheel track is listed as 21.9*16.4 = 359.16m^2.
''The smallest telescopic crawler crane in North America'' according to the company *Maeda USA* [32] is their CC1485 mini-crane. The area of the wheel track is listed as 3.61*2.49 = 8.99m^2. A track wheel area range of {8.99m^2, 359.16m^2} for cranes in practice is therefore defined that linearly correlates with the price range {$1,000,000, $5,000,000}. The track wheel area of 81.7m^2 of the reference crane corresponds in this case with a price of approximately $1,830,000.
The boundary value scheme below is used, with the percentages relative to the crane costs given in the parentheses next to the values. Note that the tolerance of 5% is calculated with respect to the price of the connection elements.

Rating	**Price**
Satisfactory	≤ $18,300 (1%)
Acceptable	< $ 91,500 (5%) - 5% ∧ > $18,300 (1%)
Marginal	$ 91,500 (5%) ± 5%
Unacceptable	> $ 91,500 (5%) + 5%

Table 7: Price boundary value scheme

3.3.9 Time of application

The provisionary amount of time necessary for applying and removing the link between the crane and pontoon is assessed. There is no exact data or criteria available for the time needed in general for connecting a crane to a pontoon. There is therefore no way to construct a sensible boundary value scheme for this performance indicator. Each connecting option therefore receives a substantiated custom rating within the four possible ratings for this indicator.

3.3.10 Number of personnel

The amount of people needed in cooperation in order to apply the connection is another relevant indicator for the option selection. The ultimate goal for the system is that it is easy to handle to the extent that a single person can apply the connection completely. If two persons are needed, it is considered as acceptable and three as marginal. Investigation is performed to find the most approximate number of personnel for each connecting option in order to use the boundary value scheme below.

Rating	**Number of personnel**
Satisfactory	1
Acceptable	2
Marginal	3
Unacceptable	> 3

Table 8: Number of personnel boundary value scheme

3.3.11 Engineering complexity

Each connecting option is rated with respect to the engineering complexity associated with the necessary mechanical adaptations to its already existing configuration in order to eliminate unacceptable (and preferably also marginal) aspects. As already indicated in section 3.2.5, a custom rating for this indicator is assigned for each connecting option, after investigating the potential modifications and the corresponding difficulties. Of the four possible ratings, the most applicable is given.

3.3.12 Fabrication complexity

Similarly to the engineering complexity, a custom rating is assigned for the fabrication complexity of the potential mechanical adaptations.

3.4 Ratings

Each of the connecting options found in chapter 2 receive in this section ratings in each of the separate performance indicators. An approximate value for each indicator is determined and subsequently assessed with the criteria and boundary values found in section 3.3. A matrix is found below which displays the awarded points per indicator for each option. The indicators are listed in the first column. The connecting options are listed in the first row with their respective abbreviations of their names. The indicators are abbreviated as well where necessary. For clarity, the order of items in the matrix follows the same sequence as the analysis in this chapter. The total amount of points with respect to all relevant design aspect (safety, space-efficiency, cost-efficiency, manageability and adaptability) are noted in the final row for each option.
Detailed calculations and argumentations for each rating can be found in Appendix B.

Aspects	FPPC	TBWP	SCT	SC	BC	SL
Force r.	2	0	0	0	0	3
DoF rest.	3	3	3	3	0	3
Preload	1	3	1	3	3	1
AOPE	3	3	3	3	3	3
MAA	1	2	1	1	3	2
WOCE	1	3	3	3	1	3
Price	3	3	2	3	3	3
Time of ap.	2	3	2	3	2	2
Nr. of pers.	2	2	2	1	2	2
EC	2	3	3	2	2	1
FC	1	2	1	1	2	1
Total	21	**27**	21	23	21	24

Table 9: Connecting option ratings

With 27 points, the turnbuckle with pin option is rated the highest among all options. Second in place comes the scissor link, with 24 points. The crucial properties that cause the turnbuckle to prevail over the scissor link are its preloading ability, time of application and its adaptation simplicity. This result is further verified by altering the point system at the binary indicators.

These are the DoF rest. and Preload, which have only two possible rating outcomes. The turnbuckle scores maximally in these two indicators. It is therefore considered using 1 point as the high score and 0 as the low score for these indicators, which lowers the scoring discrepancy between the turnbuckle and the other options. It is however found that the final outcome is not influenced by this.
It can be concluded that the turnbuckle with pin option is the most advantageous option to base the design of the crane to pontoon connection system on. A conceptual design for this system is created in the next chapter with the use of turnbuckles.

3.5 SUMMARY

The second research question is answered with the taken steps in the sub-sections of this chapter. 11 relevant performance indicators for connecting options are derived from the defined relevant performance aspects of safety, space-efficiency, cost-efficiency, manageability and adaptability. Each of these indicators can be quantified with a particular value. However, various quantification methods must be applied throughout the indicators, as explained in section 3.2. Corresponding criteria for assessment of the indicators is subsequently found. A combination of reference sources and custom assessment values are used in order to define the criteria for each indicator as explained in section 3.3. A four-leveled point assignment system is introduced here in order to provide high, low and intermediate rating possibilities for the indicators. The potential connecting options are thereafter rated with the constructed rating system, leading to the final scoring table as seen in section 3.4. Since the highest amount of points is assigned to the turnbuckle with pin, it is selected as the connecting option to proceed the design process with.

4 DESIGN PHASE I

4.1 GENERAL

As concluded in chapter 3 for the second research question, turnbuckles with jaw & jaw end fittings with removable pin (fifth turnbuckle variant in Figure 7 in section 2.3) are used to proceed the project with. The subsequent step is to design a conceptual model of the system with this connecting option, which answers the third research question. This is phase I of the design process where a parametric conceptual design is determined.
As discussed in section 2.3, the main deficiency of a turnbuckle connection with respect to crane-to-pontoon utilization is that compression forces are not allowed on the turnbuckle. The main scientific objective of this research question is thus to develop a turnbuckle connection system which eliminates compression forces on the turnbuckle, regardless of input force direction. The engineering challenges encountered within this objective and the initial thesis aims are discussed in section 4.3. Subsequently, the selected design components in correspondence to the challenges are discussed in section 4.4. The overview of the parametric conceptual design is found in section 4.2 below.

4.2 OVERVIEW

The overview of the designed parametric concept can be seen in the figures below.

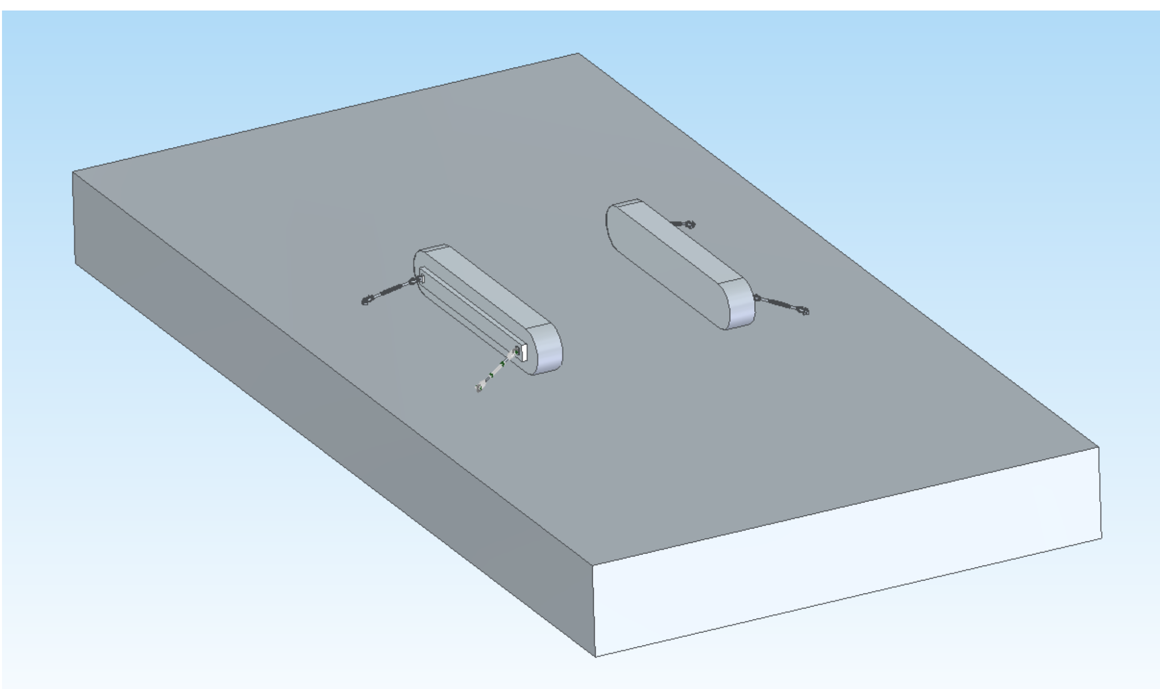


Figure 29: overall model (isometric)

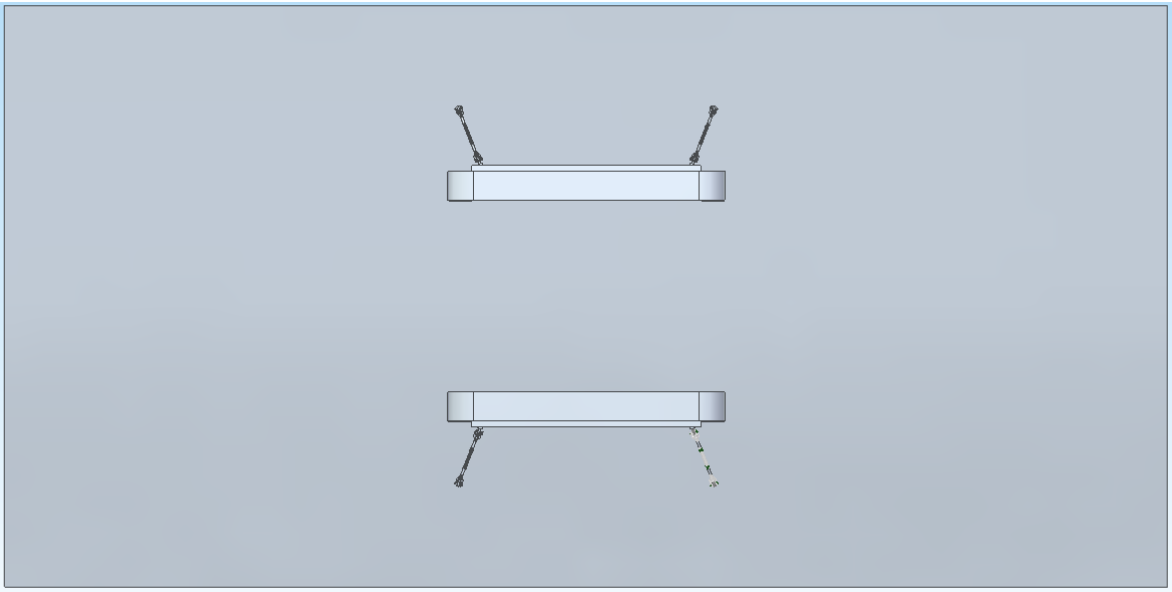


Figure 30: overall model (top-view)

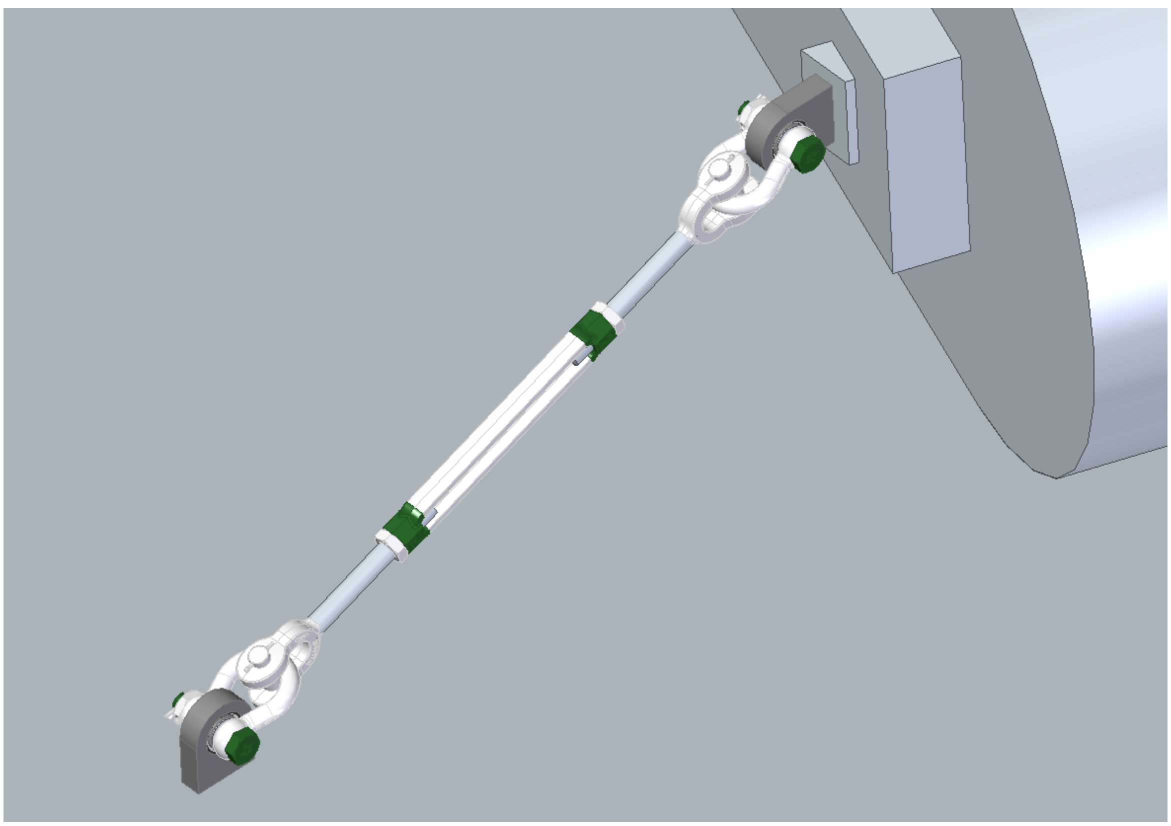


Figure 31: connection system per crane corner

An overview is provided in Table 10 below of the components that the design consists of per crane corner, together with their main design parameter(s) which is governing for the dimensions and properties of the component.

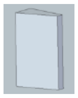
Quantity		Component	Main design parameter(s)
1x		Accessory	Inclination angle β (degrees) Placement height h (mm)
2x		Rod end with spherical bearing	Bore diameter d (mm)
2x		Bore adapter	Interference fit tightness ⦸ (-)
2x		Bow shackle	Working load limit t_1 (ton) Bow length ℓ (mm)
1x		Turnbuckle	Working load limit t_2 (ton)

Table 10: Component overview

4.3 Encountered challenges

With the jaw & jaw turnbuckle with removable pin (see Figure 32 below) as the main connector, a crane-to-pontoon connection system is designed. The turnbuckle is in principle not suitable for these type of applications due to the presence of compressive forces. An innovative connection system is thus to be developed which is free from any compression stress at all times. At the same time, the aims of simplicity and overall efficiency is to be incorporated in the design while maintaining the effectivity of a crane-to-pontoon connection system. Various engineering challenges are therefore encountered, which are discussed in the sub-sections below.

Figure 32: Jaw & jaw turnbuckle with removable pin

4.3.1 Multi-axial loading/compression

The input forces that the crane exerts on the connection system which originate from pontoon motions on water and the environmental conditions are not exclusive to any particular direction. The problem must thus be considered as three-dimensional and the system must be designed for forces occurring in any direction. If a solution is thus found for one particular direction, the problem remains present in the other directions and the found solution might not be applicable to the other directions. At the component selection process, in must therefore be ensured that 3-dimensional protection is enabled. Furthermore, in order to achieve a design which is in particular free from any compression stress, all potential causes for compression need to be considered. Compression stresses on the turnbuckle are not necessarily induced only by axial/normal forces

that act trough the center of the body. Shear forces, lateral forces, bending stresses, frictional forces, off-centered forces or partially axial forces all can be inductive to compression stresses.
A reference example below shows how complete elimination of compression stresses from the system is not achieved by simply implementing a tolerance at the turnbuckle connection point. Figure 33 below shows a turnbuckle with a tolerance at its connection points.

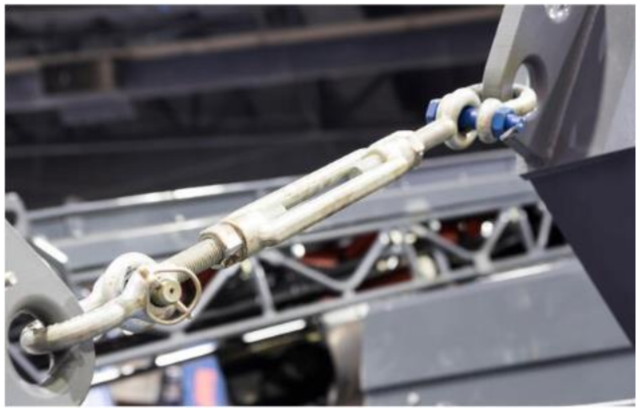


Figure 33: A turnbuckle used at a machine [10]

If the machine parts start acting compressive towards the turnbuckle, there is no instantaneous compression stress due to the tolerance at the connection points. This tolerance alone does not solve the problem however, because as soon as the distance between the machine pieces decreases, gravity will adjust the turnbuckle position until a new supportive contact point between the turnbuckle ends and the eye openings is established. This new contact point can be inductive for compression stresses. This concept is schematically illustrated in Figure 34.

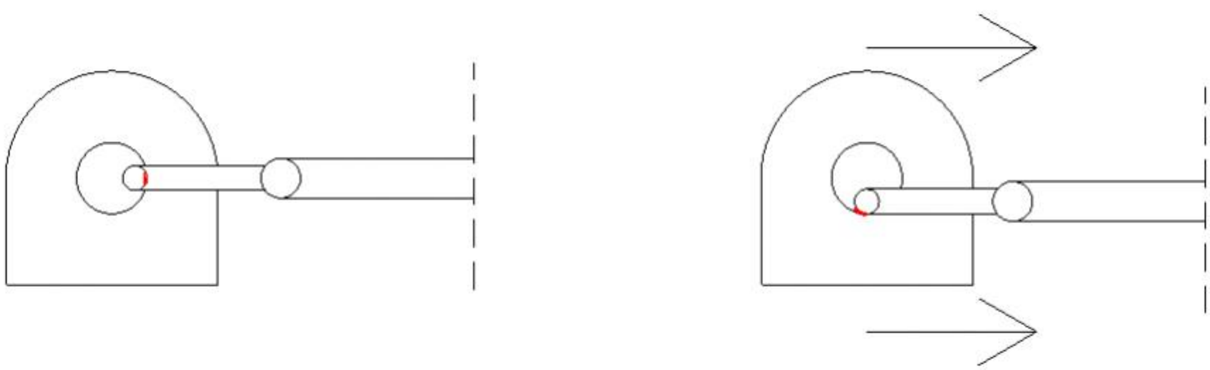


Figure 34: Alteration of contact surface due to inward displacement (side-view)

On the left-side image it is shown how the contact surface between a turnbuckle and an eye opening, indicated with red, is located at the inner half of the circular end fitting of the turnbuckle at the starting position. At the right-side image, it is shown how this contact surface shifts towards the outer half of this circular end fitting once the lashing eye displaces towards the turnbuckle, which subjects the turnbuckle to possible compression stresses due to shear and axial stresses, even though the tolerance of the opening is not (fully) consumed yet. This concept not only applies to gravitational effects, but also to any other motions and directions that alters the contact points in the connection system.

Compression scenarios are encountered in the form of bending stresses as well. Consequently, bending stresses can be dissected into different categories with different causes. The primary applicable category is a cantilever-type internal moment on the turnbuckle potentially caused by overly restricted turnbuckle end fittings. Secondarily, an internal moment can be caused due to frictional torque at (rotationally) unrestricted end-fittings.
All of the potential compression stress scenarios stated above shall need to be eliminated and absent from the final design.

4.3.2 Flexibility and tightness

In order to eliminate all potential compression stresses as discussed in section 4.3.1, the system needs to be able to freely (without friction) adapt its orientation to the directions of the forces and its resulting strain, such that the stress on the turnbuckle remains purely axial and outward at all times. At the same time however, the heavy-duty nature of the occurring forces requires that all conjunction points within and between components are tightly pressed against each other at all times, such that clearances are absent, which obstructs the necessary flexibility. The flexibility and the tightness are thus two counteracting properties here. A method needs to be found however to fully incorporate both properties into the design.

4.3.3 Asymmetry of system

Because the system is at one end connected to the side of the crane and at the other end to the pontoon deck, the supporting surfaces are perpendicular to each other, so the stress states at one end differ from the stress states at the other end. Components that are effective at one end do therefore not necessarily suffice at the other end. Maximizing the similarity and compatibility between both ends of the system for the purpose of simplicity requires thus particular attention at the design of components.

4.3.4 Pre-fabricated components

In order to maximize the simplicity for the end-user, the system needs to be built up by pre-fabricated components as much as possible. However, finding directly available components that satisfy the requirements presents a challenge on itself. Furthermore, the desired components are not necessarily designed to be used in conjunction with each other. Adaptations need to be found therefore to make the components compatible with each other. Some components simply do not fit with each other and can therefore not be combined.

4.4 COMPONENTS

A combination of components is found that overcomes the challenges stated in section 4.3 with the potentiality to optimize the connection system performance indicators (these components are listed in the overview before in Table 10). Relevant information about the components and their function is found in the sections below. Furthermore, it is explained how the components are linked to the encountered challenges.

4.4.1 Overview

An overview is provided in Table 11 below of the relations between the challenges and the used components in the design. It can be found here which challenges are tackled by which component.

Engineering Challenge	Accessory	Rod end	Bore adapter	Bow shackle	Turnbuckle
Multi-axial loading/compression	X	X		X	
Flexibility and tightness		X	X	X	
Asymmetry of system	X	X			
Pre-fabricated components		X	X	X	X

Table 11: Component relations to challenges

The detailed explanation about each and component and their relation to the corresponding challenges is found in the sections below.

4.4.2 Bow shackle

In order to tackle the problem illustrated in Figure 34, a bow shackle is implemented as seen in Figure 35 below. The shackle ensures that the original contact point at the turnbuckle end remains intact at gravitational repositioning of the turnbuckles at the compressed corners of the system during a loading scenario. The creation of a compressive contact surface due to gravitational effects is therefore excluded. The shackle thus partially solves the challenge related to ''versatility of compression stresses''. It is furthermore fully pre-fabricated.

Figure 35: Shackle example [33]

Another issue associated with the contact point shift illustrated in Figure 34 (section 4.3.1) is the created clearance. A relatively big discrepancy is present between the starting configuration and the compressed configuration, so once the compression stops, the lashing eye has the opportunity to gain relatively high momentum when returning back to the starting position, which results in high slamming forces on the connection points. The shackle solves this problem as well, as it rotates freely about its pin at the joint when tension disappears and adapts its orientation to gravity automatically like the turnbuckle does. This way the original contact surface on the turnbuckle can be preserved at all times, regardless of gravity. This concept is schematically illustrated in Figure 36 below with a jaw & jaw turnbuckle. The shackle thus partially solves the challenge related to ''flexibility and tightness'' as well.

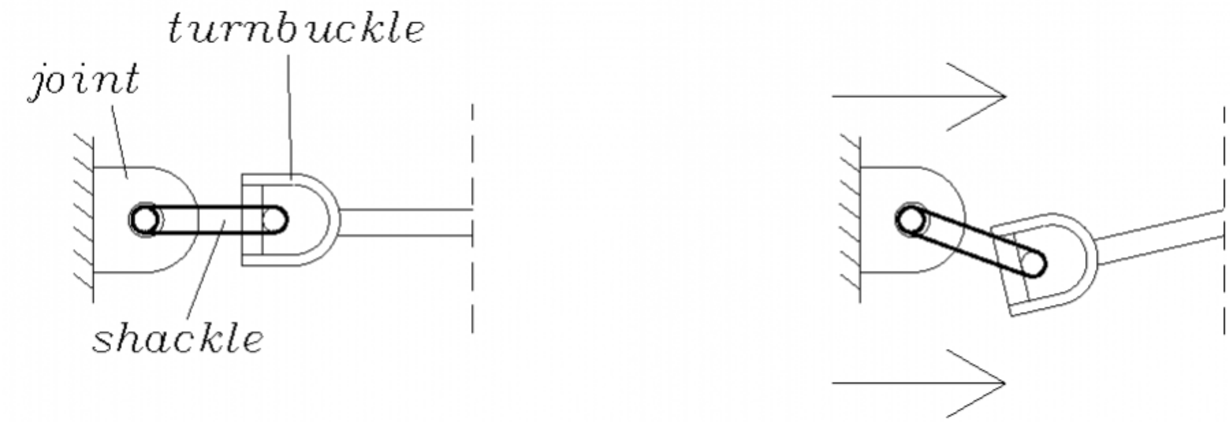


Figure 36: shackle function (side-view)

The left-side image shows the preloaded starting position of the configuration. The shackle and the turnbuckle are aligned parallel due to the tension of the preload. The right-side image shows what happens if the joint displaces towards the turnbuckle. Loss of tension causes both the shackle and turnbuckle to rotate downwards simultaneously due to gravity until a stable equilibrium is reached. This preserves the contact between the turnbuckle pin and the shackle bow at the same spot regardless of gravity. The point of contact also remains at the correct spot, such that the horizontal fork-part of the turnbuckle jaw is not sheared (if the turnbuckle is preloaded correctly, at the midpoint of the jaw pin).

Apart from the working load limit, the secondary limiting factor of the shackle is the tolerance it provides between the turnbuckle and the joint. As seen in Figure 36, the minimum radial distance between the turnbuckle and the joint decreases when the deformation takes place. This tolerance must be sufficient, such that the turnbuckle and the joint do not touch at the extreme scenario of deformation. For this purpose, shackle variants are available that have a larger bow length than ordinarily. Two main design parameters are therefore applicable to this component, which are the working load limit t_1 and the bow length ℓ.

A compilation of potential dimensions sets for the applied shackles is used as provided by manufacturer *Greenpin* [34] [35]. Within this compilation, dimension sets for shackles with standard bow are included as well as ''BigMouth'' shackles with a bigger bow than usual . The two shackle variants are displayed in Figure 37 below. The dimension sets for both variants are found in Appendix C. Here it can be seen that for most of the working load limit options, two options are available for the bow length ℓ (standard or BigMouth).

Figure 37: Standard bow shackle vs. BigMouth shackle (GreenPin) [35] [34]

4.4.3 Rod end with spherical bearing

Due to various challenges stated in section 4.3, inserting the shackle into a simple joint as illustrated in Figure 36 does not satisfy the design requirements. Referring back to ''flexibility and tightness'', orientation changes need to be free from friction in order to eliminate all compression stresses, while at the same time a tightly pressed contact surface is required at the connection points. The joint thus needs to be incorporated with a bearing. In addition, referring back to the ''multi-axial loading'', the joint must provide freedom of rotation about each axis. In order to analyze this aspect, the situation in Figure 36 is illustrated from a top-view perspective in Figure 38 below. The case is sketched where the joint displaces diagonally towards the turnbuckle. It can be seen at the right-side image that, even though the shackle orientation is adjusted accordingly due to gravity, a contact surface emerges at the side/front of the turnbuckle pin (indicated with the red spot) before fully consuming the tolerance, leading to an undesired state of shear/compression on the turnbuckle. Note that a similar mechanism is assumed at the opposite (pontoon) side of the turnbuckle, such that equal and opposite forces arise at the opposite side, creating these stresses. Furthermore, because a press fit is necessary between the shackle pin and the joint (see section 4.4.4 for more information), lateral movements (axially to the shackle pin) of the shackle are restricted, such that the stresses emerge instantly when the lateral contact occurs.

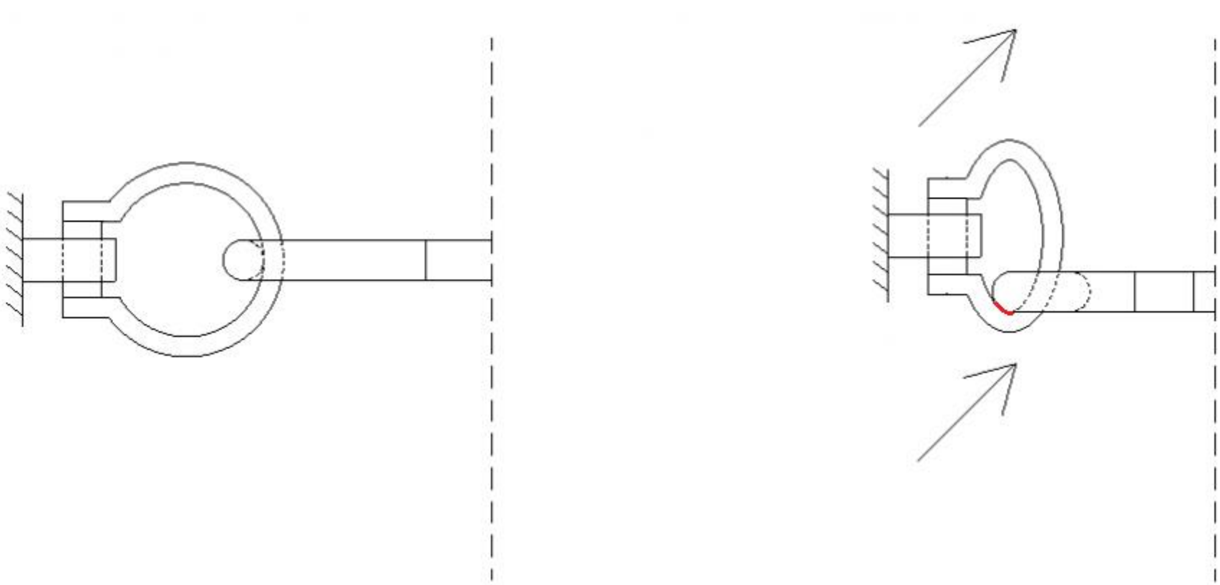


Figure 38: Shackle causing shear/compression (top-view)

The joint should thus provide freedom to rotate in the top-view plane as well, such that the shackle also properly adjusts its orientation with respect to this plane. This concept is illustrated in Figure 39 below.

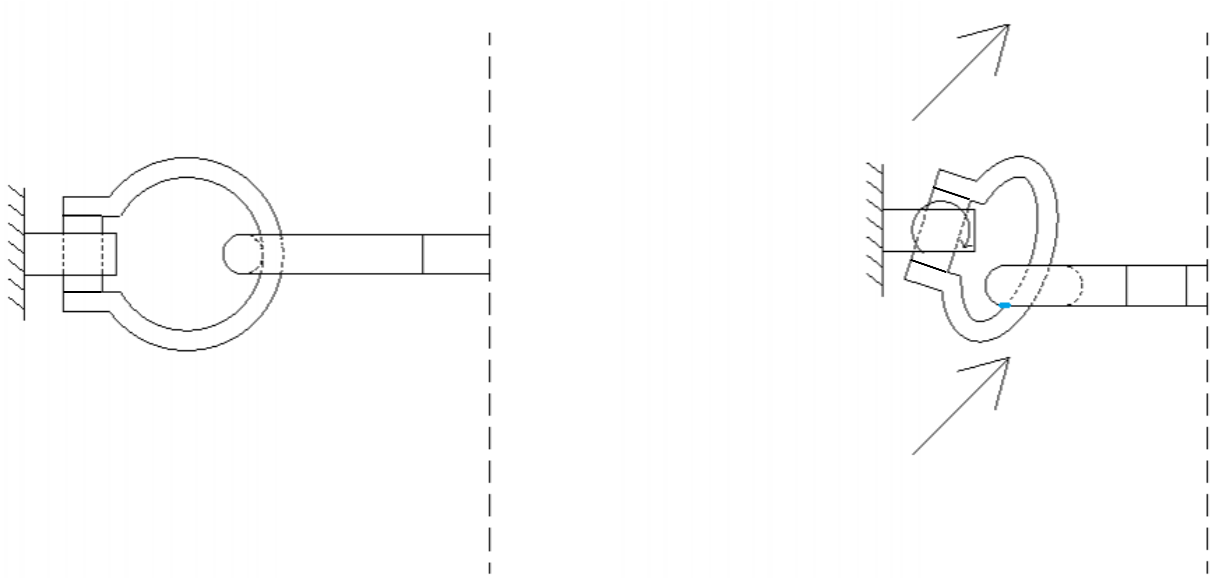


Figure 39: shackle with dynamic hinge (top-view)

By implementing this multi-axial pivot point with negligible friction to the system, any possible forces at the side/front of the turnbuckle pin as described above are absorbed and converted to rotating moments about this pivot point before any traction can be gained on the turnbuckle pin that could lead to shear/compression.
Having this pivot point is not only essential for the compression cases, but also for the tension cases. A tension stress is applied in a direction that is never entirely parallel to the turnbuckles. A slight orientation change needs therefore to take place on the turnbuckle in order for its deformation to remain purely axial as required. Because the forces exerted by the shackles on the turnbuckle pins are high in the cases of tension, high frictional torque would counterpose the orientation change of the turnbuckle if it would simply rotate about its pins, leading to significant bending stresses and thus compression stresses. By implementing the pivot point with negligible resistance on the shackle joint, bending stresses (and thus compression stresses) due to friction are prevented on the turnbuckle.
Referring back to the ''multi-axial loading'', the applied tension stresses can have a vertical component as well (perpendicular to the pontoon surface), meaning that orientation change of the turnbuckle must be facilitated with respect to friction in this plane as well. The shackle joint must therefore also provide smoothness when rotating axially about the shackle pin.
The suitable pre-fabricated component meeting all requirements mentioned in this section is a ''hydraulic rod end with rectangular welding face'', as seen in Figure 40. Its eye opening is equipped with a 'spherical plain bearing', allowing free rotation about each axis. It enables thus smooth axial rotation while freely tolerating angular misalignments. It is primarily designed to withstand forces in radial direction, but can take up forces in axial direction as well in smaller magnitudes. These characteristics provide the necessary potential to be used in the design.

Figure 40: Hydraulic rod end (Schaeffler) [36]

This component is supplied for a wide range of force magnitudes and scale sizes. The bore diameter correlates with the load capacity and it determines furthermore the compatibility with the other components. Each diameter size is correlated to a single set of component dimensions. The bore diameter *d* is therefore the single design parameter for this component. The dimension sets applied to this project are taken from the rod ends supplied by manufacturer *Schaeffler* [36] and are found in Appendix D.
Schaeffler prescribes furthermore a maximum allowable tilt angle for the spherical bearing inside the rod ends, which is denoted with the α symbol in Figure 41 below. This maximum angle applies to any axis radial to the bearing.

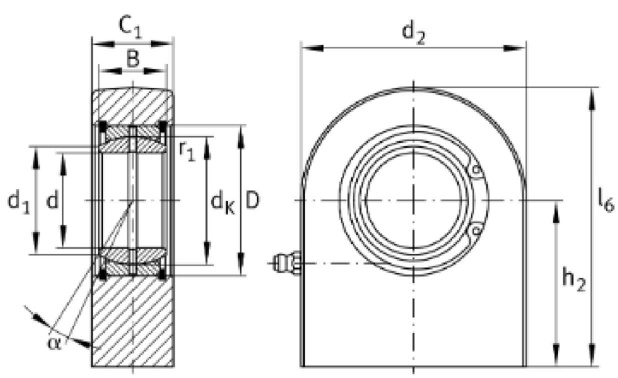


Figure 41: Hydraulic rod end with parametrical dimensions (Schaeffler) [36]

Because of this requirement, a parallel alignment between the rod ends on both sides of the design is ideal, giving a starting tilt angle of zero degrees as displayed in Figure 42 below. When the system is strained on its maximum, the resulting tilt angle should not be higher than the allowable tilt angle. If this angle is exceeded, loss of contact surface area emerges between the inner and outer rings of the spherical bearing, which makes the prescribed maximum allowable dynamic and static radial load values invalid [37].

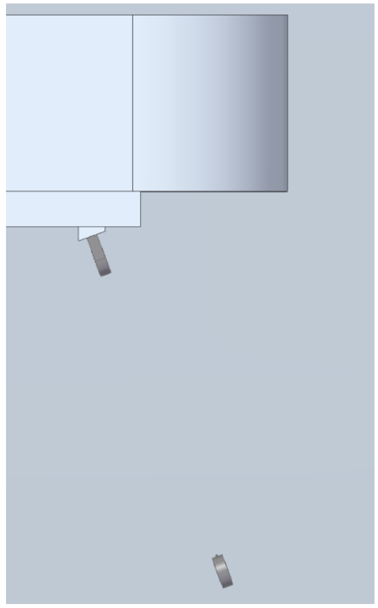


Figure 42: Rod ends alignment (top-view)

4.4.4 Bore adapter

The shackle is intended to be inserted with its pin into the rod end bore. However, the load capacity of a shackle with a corresponding pin diameter exceeds greatly the load capacity of the rod end. In order to ensure comfort when connecting the turnbuckle and to maximize the overall efficiency of the design, it is aimed to apply the smallest and lightest shackle as possible. This means that the bore diameter needs to be adapted to a smaller diameter. This adapter is envisioned as a solid steel cylinder as seen in Figure 43 below. It has an outer diameter that matches the bore diameter of the rod end, and it has an inner diameter that matches the pin diameter of the shackle. This component thus provides the compatibility factor with respect to the ''pre-fabricated components'' challenge. Furthermore, it is fabricated with the correct interference fit diameters, providing the necessary tightness to the compound with regards to the ''flexibility and tightness'' challenge.

Figure 43: Bore adapter (Terre) [38]

What can be seen with this reference bore adapter by *Terre* [38] is that the edges are slightly trimmed, which allows for smoother assembly. The first portion of the adapter at the edge can

therefore be simply slotted into the eye of the rod end, so that the rest of the adapter can be pressed into the eye with more security and comfort. The inner circular edge is trimmed as well, so that the same principle applies to the insertion of the shackle pin in the eye of the adapter.
These type of adapters are not widely available and are not supplied at a large scale-size. Because of their simplicity however, they can be simply fabricated at a custom size.
The nominal outer diameter is always equal to the rod end bore diameter *d* and the nominal inner diameter is always equal to the shackle pin diameter (dimension b in Appendix C). The length of the cylinder is always equal to the bearing width (dimension B in Appendix D). The specific design parameter of the bore adapter is the interference fit tightness at the inner surface with the shackle pin. By tuning this parameter, the appropriate amount of interference fit pressure can be regulated. This parameter is quantified as the interference fit tightness ⃠ between the nominal inner diameter and the actual inner diameter. Standardized fit tightness parameters are used for ⃠, which give a different interference value in mm for each diameter size. The difference between the nominal outer diameter and the actual outer diameter is predetermined, since *Schaeffler* [37] prescribes an m6 fit tightness for the shaft insertion into the rod end bearing.
The inner diameter contracts at the insertion of the bore adapter in the rod end bearing due to the m6 interference fit pressure on the outer surface of the bore adapter. In order to obtain the desired fit at the inner diameter for the shackle insertion, the adapter is fabricated with an inner diameter slightly smaller ($\approx$ 1mm for sufficient margin) than nominal. After the bore adapter is inserted into the rod end bearing, the inner diameter is finished by increasing it up until the corresponding value for ⃠ is obtained. A visualization of the bore adapter placement inside the rod end bearing is shown in Figure 44 below.

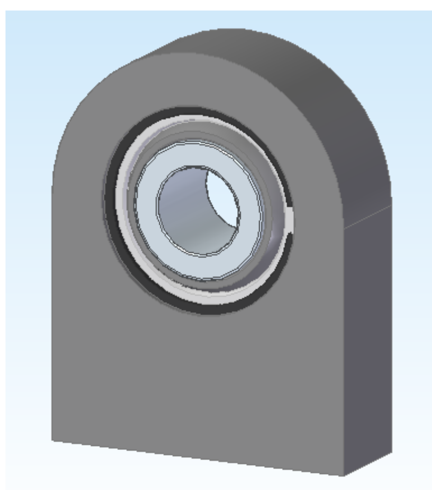


Figure 44: Bore adapter placement in rod end

4.4.5 Accessory

In order to provide protection against the ‘’multi-axial loading’’ with the utilization of a single turnbuckle per crane corner, the turnbuckles should be aligned in a way that contains directional vectors in each axis (X, Y and Z). This is however not directly possible due to the ‘’asymmetry of system’’, since the crane track wheels are aligned parallel to the longitudinal pontoon axis (X axis) at all times. Welding the rod end directly on the track wheels would thus provide turnbuckle stiffness only perpendicular to this axis. By adding an inclination angle on the crane connection point, the stiffness can be distributed over multiple axis. This is envisioned by a trapezium-shaped accessory as an additional steel component that is welded to the side of the track wheel, as seen in Figure 45 below with the accessory encircled.

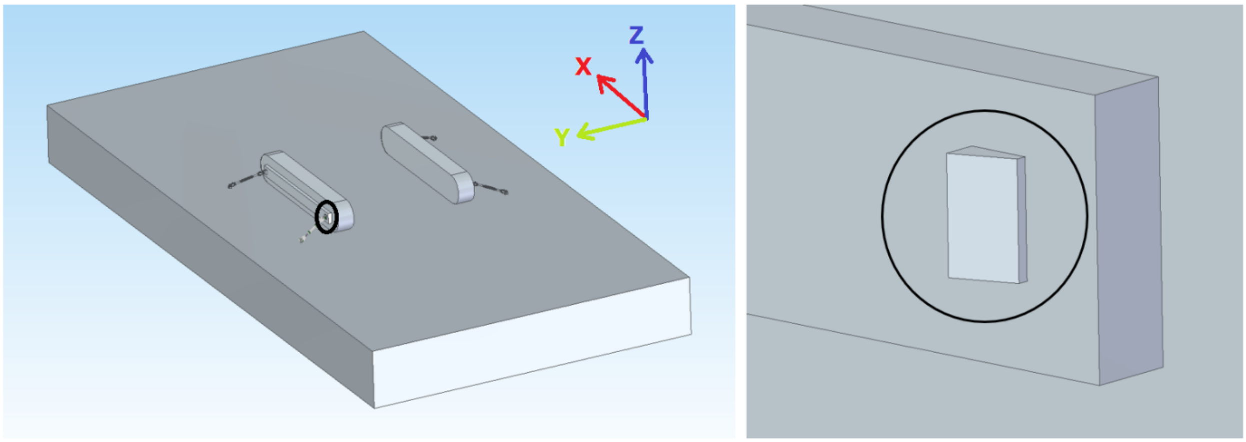


Figure 45: Accessory (global and detail view)

In general, the interior of crane track wheels is filled with steel, as seen with the *Liebherr* crane in Figure 46 below. This steel has a weldable outer surface, which is suitable for the addition of a steel trapezium-shaped accessory.

Figure 46: Liebherr crane track wheels [39]

The inclination angle β of the trapezium determines the amount of turnbuckle stiffness transferred from the X-axis to the Y-axis. The placement *h* determines the amount of stiffness transferred to the Z-axis, since it directly correlates with the upward inclination angle of the turnbuckle. These are therefore the two main design parameters of this component. Figure 47 below shows a schematic visualization of these parameters.

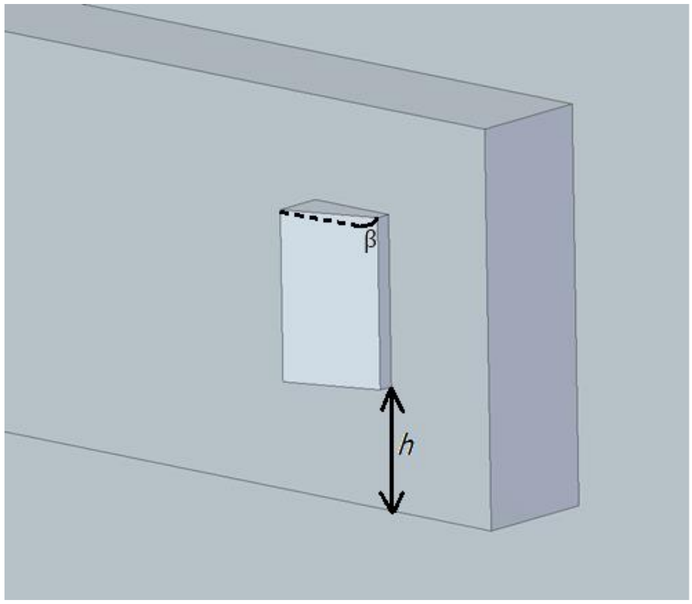


Figure 47: accessory with main design parameters

4.4.6 Turnbuckle

Once all crane and pontoon components are installed, the configuration as displayed in Figure 48 is obtained. As can be seen in this figure, the crane is placed on the pontoon without a connection yet. The shackle therefore hang loosely downwards.

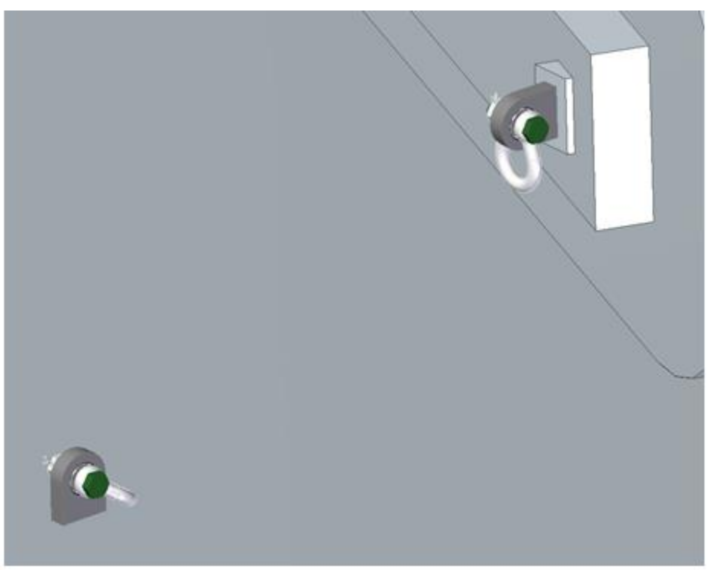


Figure 48: Crane and pontoon components (disconnected)

As discussed before, the main connector of the system is the turnbuckle with jaw & jaw end fittings with removable pins. For connection, the pins are placed through the shackle loops at crane and pontoon side. A small cotter is pushed through the end of the pin, which locks the pin inside the turnbuckle jaw as seen in Figure 49 below.

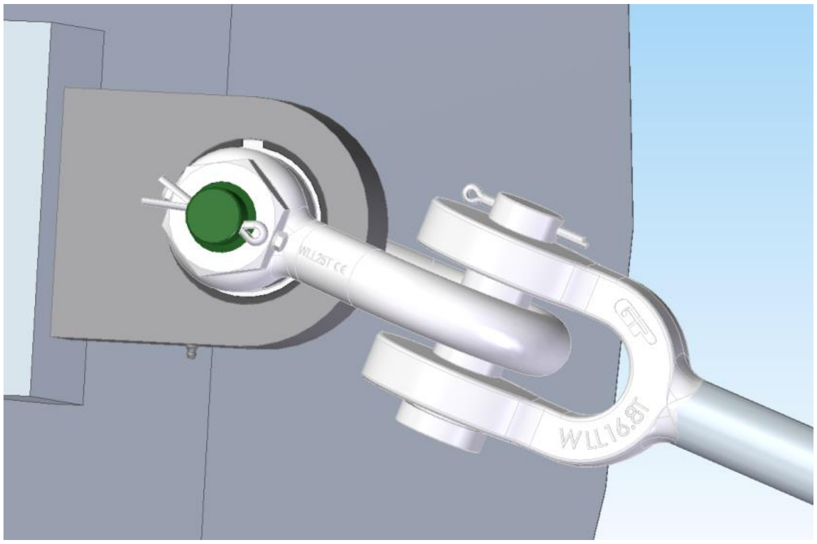


Figure 49: Turnbuckle connection to shackle

The frame of the turnbuckle is subsequently rotated until a stiff angular alignment is reached between both shackles, which establishes the necessary preload on the system and finalizes the connection as seen in Figure 50. For removal of the connection, this process is simply reversed.

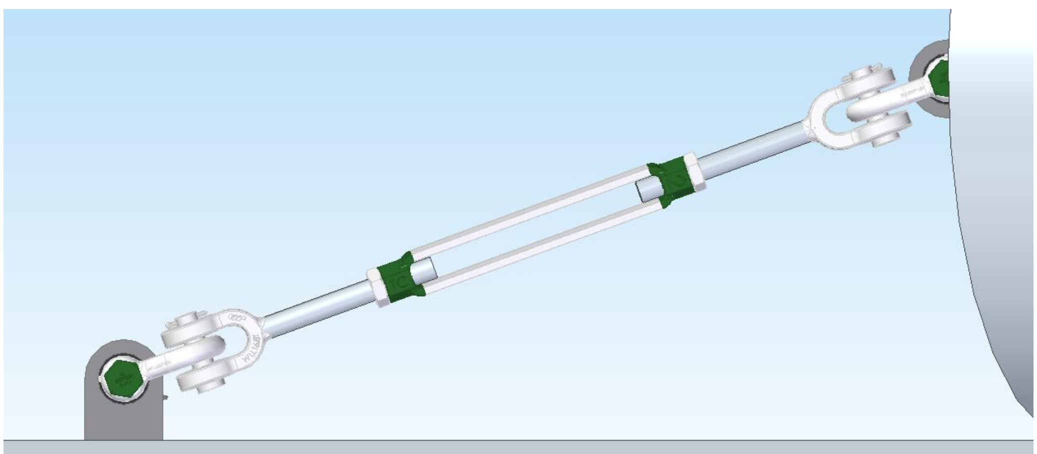


Figure 50: Preloaded and finalized turnbuckle connection

The applied type of turnbuckle is fully pre-fabricated. Its main design parameter is the working load limit t_2. The corresponding dimension sets are adopted from manufacturer *Greenpin*, which can be found in Appendix E.

4.5 Summary

The third research question is answered with the found combination of components which overcome the encountered engineering challenges. The scientific objective of creating a turnbuckle application which filters out any occurring compression forces is therefore reached. The design provides furthermore the necessary features, which are the multi-axial loading possibilities, tightness at the interconnections and no alteration or loss of contact surfaces.

The majority of the applied components are pre-fabricated and the remaining components are simple steel shapes. The majority of the components are furthermore recurring in two identical counterparts. This makes the aim of end-user simplicity achieved as well. Design decisions in order to maximize overall efficiency are furthermore implemented. The next chapter comprises phase II of the design process where the design parameters are quantified.

5 DESIGN PHASE II

5.1 GENERAL

With phase I of the design completed, phase II of the design is initiated. This part of the research exhibits that the design is practicable and feasible in terms of construction time and effort, availability of components and material consumption. This means that the smallest possible component sizes that pass the safety evaluation are found here and the design parameters are quantified accordingly, which answers the fourth research question.
The first step of this design phase is to determine and clarify the safety requirements of the designed connection system. The next step is to obtain suitable input sample reference data for the parameters surrounding the connection system. This includes crane data, pontoon data and environmental conditions. The third step is to quantify the load-cases corresponding to this input data. Subsequently, suitable design parameter values corresponding to these load-cases are determined by performing calculations and collecting component manufacturer data.

5.2 OVERVIEW

Table 12 below provides an overview of the used values for each design parameter. This combination of values thus satisfies all strength and deformation requirements, while applying a minimization from a material consumption perspective. it is found that these parameter values lie within the range of manufacturer availability and within the range of feasible construction time and effort, meaning that the size and weight of each used component can be handled manually by the cooperation of two construction workers.

	Value
$\boldsymbol{\beta}$	20°
$\boldsymbol{h}$	71mm
$\boldsymbol{d}$	80mm
$\varnothing$	j6
$\boldsymbol{t_1}$	25t
$\boldsymbol{\ell}$	178mm
$\boldsymbol{t_2}$	16.8t

Table 12: design parameter values

Detailed information about the determination of these design parameter values is found in the sections below.

5.3 SAFETY REQUIREMENTS

In order to perform the safety evaluation to determine suitable design parameter values, the safety requirements which must be satisfied are determined firstly. These requirements are directly related to prevention of mechanical failure of the connection system and can be categorized in two types; strength requirements and deformation requirements. The sections below give further explanation on this topic. All requirements are numbered for reference purposes.

5.3.1 Strength requirements

Within the strength requirements, each component individually must be able to withstand the maximum stresses related to the load-cases. This is assessed differently per component as explained in the sections below. The requirements are noted after the dots at the end of each section.

5.3.1.1 Accessory

The Von-Mises stress criterion is applied for evaluating the stress on the accessory. The applied equivalent stress value is multiplied with the Eurocode safety factor corresponding to variable loads, which is equal to 1.5 [40]. The steel grade with the lowest yield strength that is applied within the Eurocode standard EN 1993-1-8 [41] is used as the grade for assessment, which is S235 with a yield strength of 235 MPa.
The accessory is fixed on the track wheel of the crane by welding. The strength of these welds needs to be proven sufficient. The maximum area available for the size of the welds is limited by the surface area of the steel frame within the track wheel. The maximum available welding area must thus be bigger than the necessary welding area. The necessary area is determined in accordance with Eurocode standard EN 1993-1-8, which can be found in Appendix F.

- $1.5 \times \sigma_{vm} < \sigma_y$ (1)
- $A_{weld\text{-}available} > A_{weld\text{-}minimum}$ (2)

5.3.1.2 Rod end with spherical bearing

The rod end consists out a steel frame with an incorporated spherical plain bearing. It needs to be verified with respect to bearing-specific rules prescribed by the manufacturer *Schaeffler*, as well as to the steel stress on the frame separately [37].
In order to find the correct dynamic load value on the bearing, the applied radial load is multiplied by the axial load factor X. This product must be lower than the allowable dynamic load C_r. The value of X depends on the ratio between the applied axial and radial bearing loads and can be determined with the graph shown in Figure 51 below. This ratio must not be higher than 0.3, meaning that the amount of allowable axial force depends on the amount of radial force applied. With higher radial stress, the bearing is protected better against axial forces.

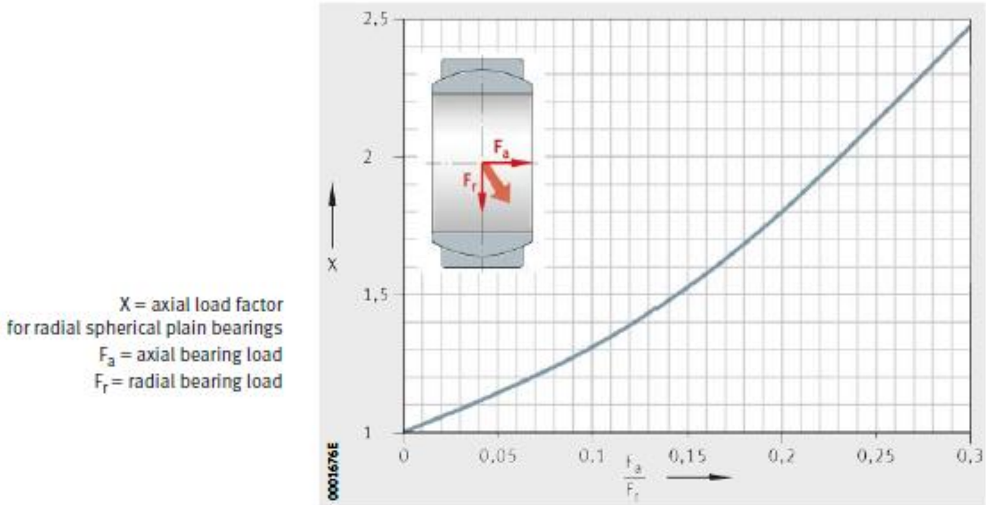


Figure 51: Axial load factor for plain bearings with combined loading [37]

Because the turnbuckles are in their default setting aligned fully radially to the rod ends, the partial axial load vector arises due to deformation only. The allowable tilt angle limit however (explained in section 5.3.2.2) implies that the deformations never reach the level where $F_a = 0.3F_r$. This requirement is thus already overlapped by requirement (13).

In order to find the correct static load value, the applied radial load is multiplied by the factor for a pulsating load frequency according to *Schaeffler*, which is 2.75 [37]. This product must be lower than the allowable static load C_{0r}.

The Von-Mises stress criterion is applied for evaluation of the stress on the rod end frame. The applied equivalent stress value is multiplied with the Eurocode safety factor of 1.5. The steel grade used for these rod ends is S355 [37]. The yield strength of this grade depends on the steel thickness according to DIN EN 10025 [42].

Furthermore, the strength of the welding area is assessed with Eurocode standard EN 1993-1-8 (Appendix F). The available welding area for the rod end is limited by the surface area of the accessory. The maximum available welding area must thus be bigger than the necessary welding area.

- $XF_r < C_r$ (3)
- $2.75F_r < C_{0r}$ (4)
- $1.5 \times \sigma_{vm\text{-}frame} < \sigma_y$ (5)
- $A_{weld\text{-}available} > A_{weld\text{-}minimum}$ (6)

5.3.1.3 Bore adapter

As the bore adapter is pressed inside the bearing, it is subject to interference fit stress immediately and at all times. The interference fit pressure on the outer surface is to be calculated according to the m6 fit parameter in addition with the radial deflection caused by the shackle pin interference. The inner surface is subject to the interference fit pressure caused by the shackle pin interference.

The stress created by the input force exerted through the shackle on the bore adapter when the loads are applied is superposed on the interference pressure in order to obtain the total radial pressure on each surface. Furthermore, circumferential stress in created due to the radial pressures. The Von-Mises criterion is therefore applied with these stresses and the Eurocode safety factor to evaluate the strength.

- $1.5 \times \sigma_{vm} < \sigma_y$ (7)

5.3.1.4 Bow shackle

The bow shackle manufacturer *Greenpin* [35] prescribes a working load limit for the each shackle size, which is likewise the design parameter t_1 as discussed in chapter 4. A safety factor of 5.0 is incorporated in this working load limit, so that no additional safety factors need to be superposed on this load limit. The force applied on the bow must at all times be lower than the load limit.
The pin of the shackle is loaded differently than the bow, since the interference fit force due to the interference ⃠ with the inner bore adapter diameter is applied only at the pin. Apart from radial stress, this interference fit gives the pin a constant circumferential stress. The working load limit prescribed by *Greenpin* does not account for the circumferential stresses. The pin is thus additionally verified with the Von-Mises criterion as a safety check.
The bow of the shackle is loaded purely by the turnbuckle force, which are transferred through the turnbuckle pin. Bow-to-pin contact points are allowed according to *Greenpin* [43], as long as the pin diameter is larger than the bow diameter in order to limit the stress concentrations.

- $F_{bow} < t_1$ (8)
- $1.5 \times \sigma_{vm\text{-}pin} < \sigma_y$ (9)
- $d_{bow} < d_{turnbuckle\text{-}pin}$ (10)

5.3.1.5 Turnbuckle

The turnbuckle is loaded only axially and in tension. This force depends on the magnitude and direction of the input forces as well as the inclination angles of the turnbuckles. The only requirement is that this force is below the working load limit prescribed by *Greenpin* [44] for each turnbuckle size, which is likewise the design parameter t_2 as discussed in chapter 4. A safety factor of 5.0 is incorporated in the working load limit.

- $F_{turnbuckle} < t_2$ (11)

5.3.2 Deformation requirements

Particular requirements related to deformation of the base configuration of the design are implemented. These are derived from various limitations and potential safety risks with regards to mechanical failure of the system as discussed in the sections below.

5.3.2.1 Tolerance

The starting space in between the turnbuckle and the rod end is limited as seen in Figure 52. The crane displaces towards the turnbuckles at the retracted corners of the configuration when the loads are applied due to the elongation of the turnbuckles at the extended corners. Even though the system is designed in a way that prevents compression forces on the turnbuckles at the retracted corners, if the available tolerance is exceeded the turnbuckle and rod end will touch, leading to compression forces and damage.

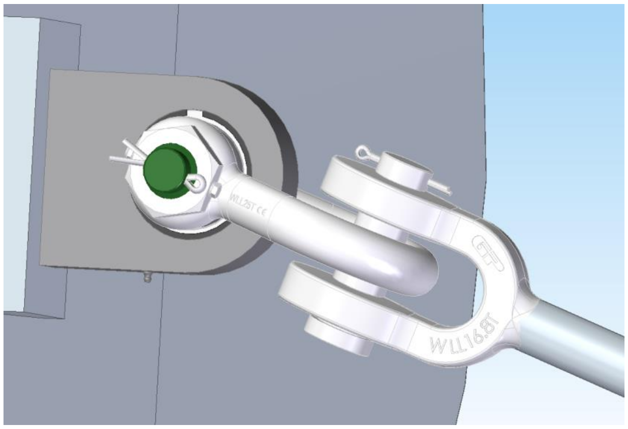


Figure 52: Turnbuckle tolerance

The deformation of the system must thus be limited, such that the tolerance between the turnbuckle and rod end remains positive at all times. Figure 53 below shows schematically the turnbuckle deformation as a result of a particular force F. ΔTB is the resulting displacement of the system in the direction of the force F.

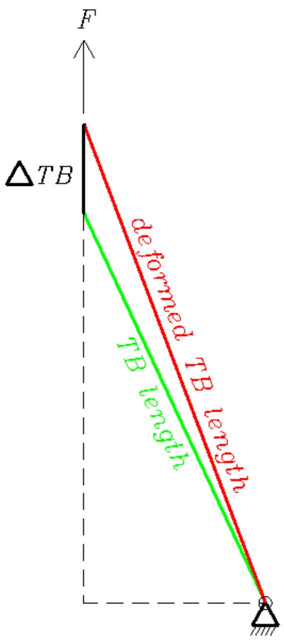


Figure 53: Turnbuckle deformation

At the opposing turnbuckle, the rod end displaces thus towards the turnbuckle with a distance of ΔTB in the direction of the applied force. The partial vector of ΔTB in the direction of the turnbuckle alignment decreases the spacing between the rod end and the turnbuckle. ΔTB is therefore used as an upper limit value of the occurring space reduction between turnbuckle and rod end. The starting tolerance, which must at all times be higher than the space reduction, is expressed in terms of component parameters. The bow length ℓ of the shackle is here the parameter which provides the tolerance. It is reduced by default however by the protrusions in the shackle bow by the turnbuckle jaw and the rod end. As seen in Appendix E, the total turnbuckle length is denoted with D and the length between pins with B. The outer frame diameter of the rod end is denoted with d_2 as seen in Appendix D. The diameter of the shackle pin is denoted with b as seen in Appendix C.

- $\Delta TB < \ell - \frac{1}{2}(D - B) - \frac{1}{2}d_2 + \frac{1}{2}b$ (12)

5.3.2.2 Tilt angle

Allowable rotation of the turnbuckles with respect to their starting position is limited due to a maximum allowable tilt angle of the spherical bearing inside the rod end, which is denoted with the α in Figure 54 below. This tilt angle is defined as the angle about any axis radial to the bearing with respect to the starting position shown in Figure 54. *Schaeffler* prescribes a different maximum allowable tilt angle (within the range of 5° to 9°) for each rod end size, which can be found in Appendix D. Exceeding this limit causes loss of contact surface between the inner and outer ring of the bearing, such that the prescribed load capacity values are not valid anymore [37].
This tilt angle α can be determined with the original and deformed turnbuckle lengths as seen in Figure 55. Note that only the angle about the radial bearing axis is measured here.

- Tilt angle < α (13)

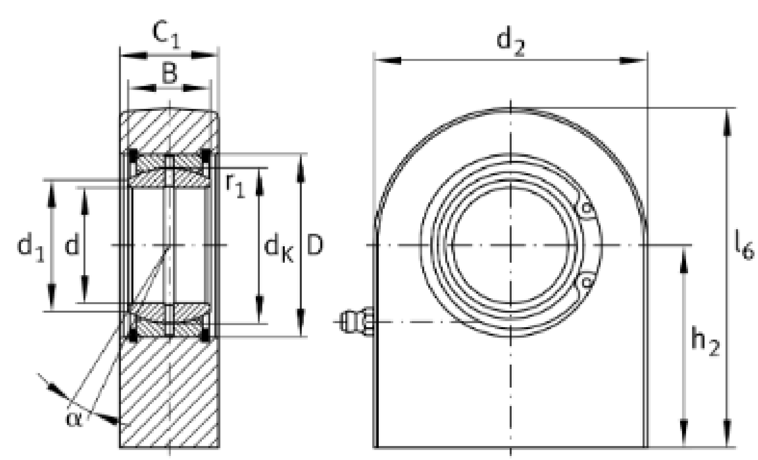


Figure 54: Hydraulic rod end with parametrical dimensions (Schaeffler) [36]

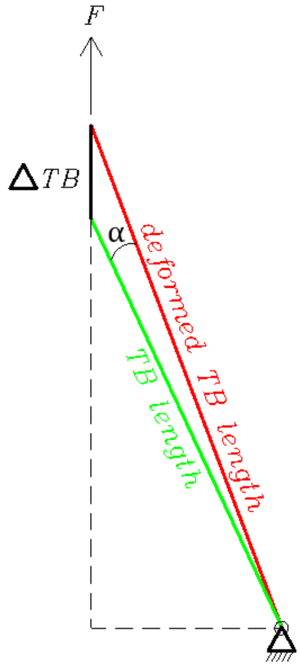


Figure 55: tilt angle measurement (top-view)

5.3.2.3 Radial internal clearance

There is a small radial internal clearance present between the inner and outer ring of the spherical plain bearings in the rod ends at their initial unloaded state. This clearance is reduced by the interference fit between the bore adapter and the bearing. Ideally, the clearance is reduced as much as possible by the fit. However, it is not desired that this clearance gets below zero, since this causes a preload on the bearing. The operational clearance of the bearing must thus remain positive.

- Operational clearance > 0 (14)

5.4 INPUT DATA

The IHC input data for a heavy-duty crane application is found in the sub-sections below. This includes crane data, environmental conditions and pontoon data [2]. This data is used to obtain force magnitudes, corresponding to heavy-duty utilization, which must be suitable to the conceptual design of the connection system.

5.4.1 Crane

A generic reference crane with heavy-duty properties as shown in Figure 56 below is given as input by IHC;

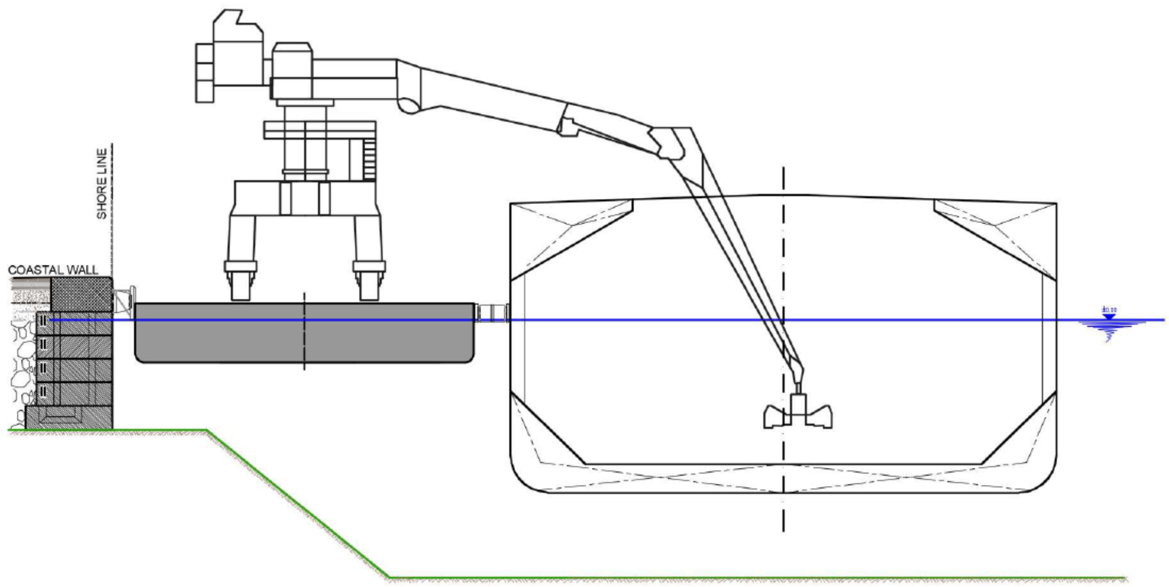


Figure 56: Front-view of the reference crane on a pontoon in between a shore and a vessel [2]

The following properties and parameters are determined for the reference crane [2]:

Height (until boom): 11.9 m
Weight: 350t
Capacity: 16t
Arm reach: 35m
Track wheel area: 9.5x1.0m
Distance between track wheels: 7.6m
Maximum permissible dynamical angle: 4°

5.4.2 Environmental conditions

IHC has furthermore performed a study on environmental conditions in areas along the Croatian coast in order to retrieve sensible values for the environmental parameters for a theoretical location of reference, which have led to the following input values [2]:

Wind speed = 40m/s (78kn)
Wave length = 6.24m
Wave height = 0.57m

5.4.3 Pontoon

IHC has been able to determine preliminary pontoon dimensions with these crane properties and environmental data, that are valid when the crane is at the amidships position on the pontoon by using board stability calculation software. With the following pontoon dimensions, the applied hydrodynamical stability criteria were satisfied while the maximum dynamical angle in operation will be equal to the permissible dynamical angle of 4° [2]:

Length = 40m
Width = 20m
Height = 3.5m

5.5 LOAD CASES

In order to determine the corresponding load cases for the safety evaluation of the connection system, all the occurring scenarios associated with crane-to-pontoon utilization are determined firstly. By analyzing these scenarios, the corresponding forces to each scenario can be determined. The load cases can subsequently be determined with these forces.
IHC defines three possible scenarios in which a crane-to-pontoon configuration can be situated and which cover all possible actions that a floating crane can perform. These three scenarios are: onboarding, operation and navigation. Within these three scenarios, all of the potential loading cases and risks are thus covered [2]. The safety evaluation is therefore confined within the scope of these scenarios. In the sections below, the scenarios and their respective consequences are discussed. The maximum loads are defined at the end of each section for each scenario. The load cases are finally split up in terms of these scenarios and their maximum loads in the final sub-section.
As explained in section 4.4.6, once the turnbuckles are correctly connected, a preloaded condition is established on the system, which leads to the following two simplifications to be taken into consideration in these calculations;

- The crane and pontoon move as one body, except for the small deformations occurring in the connection system. This allows a quasi-static approach to be taken.

- The turnbuckles are loaded simultaneously, because every turnbuckle is tightened and preloaded at the moment when the loads occur. It is therefore taken that the loads are distributed proportionally along the loaded turnbuckles during each loading process.

For these simplifications to remain valid, it is imperative that inspection of the preload is regulated throughout the loading cycles. The standardized safety factors applied in the safety requirements in section 5.3 are furthermore considered sufficient to account for any disturbances occurring with respect to the preload tightness.

5.5.1 Operation

At the scenario of operation, the crane is fixed with the connection system at the amidships position of the pontoon and performs lifting operations by moving bulk from one side to the other in between the coastal wall and a vessel (as illustrated in Figure 56) while the floating body remains stationary. Within this scenario, the crane arm will be stretched out and loaded, causing a center of gravity shift which results in a dynamical angle of heeling (note the front-view in Figure 56). As mentioned in section 5.4.1, the maximum value for this angle is 4° (including the effects from the environmental conditions) as determined by IHC [2]. In order to keep the crane static, a particular reaction force R_y and potentially a reaction moment M_x are caused by the connection system, which is schematically drawn with the red lines in Figure 57 below.

After the arm has been loaded or unloaded, it retracts itself back towards the center of the crane, so when the upper part of the crane rotates from one side to the other, the arm remains close to the center of gravity of the crane and pontoon, so that the trimming angle during operation can remain relatively small [2]. Because the length of the pontoon is much larger than the width, the trim angle in operation appears after stability analysis to be approximately 0° in this case [2], which means that the reaction force R_x and the reaction moment M_y are also zero. Only the forces and moments resulting from the 4° heeling angle (R_y and M_x) are thus to be considered in this scenario. Because of this, and due to the symmetric distribution of R_y and M_x by the connection system along the x-axis, the load case can be simplified to a 2d-situation as seen in Figure 57 below, which shows the global loading case on the crane for this scenario.

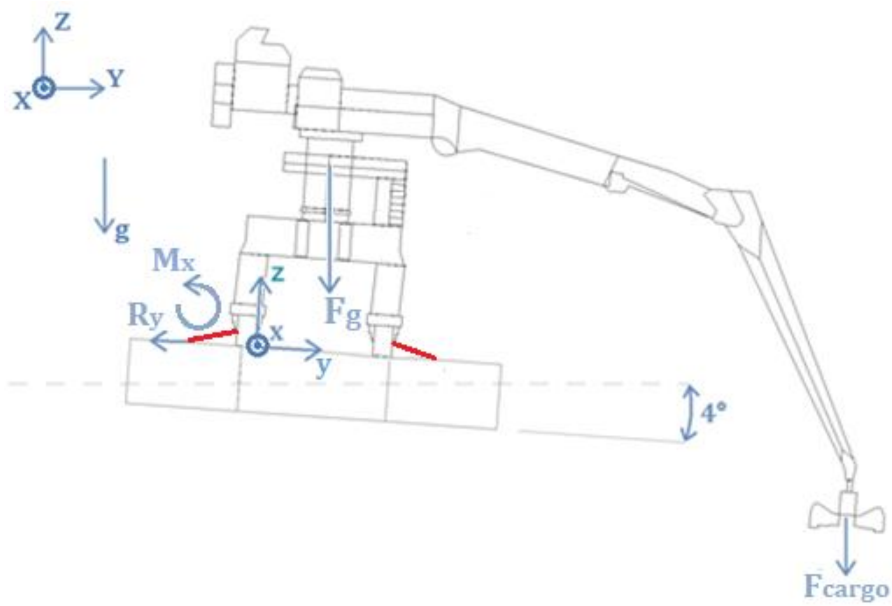


Figure 57: Schematic drawing of loading case in operation (front-view)

A calculation is performed in order to find the reaction force R_y and reaction moment M_x. The front-view of the crane as seen in Figure 57 is modelled for this purpose in structural analysis software ''MultiFrame'' in 2d. The correct weight distribution of all crane components including the counterweight is implemented along the ZY-plane, together with the cargo load of 16 tons at the end of the boom and the 4° heeling angle. The contact surfaces between the crane and pontoon are modelled as two fixed points in this analysis, which simulates the effect of the connection system. Each of these two points thus correspond to an entire track wheel of the crane. The reaction force and moment at each of these points is thus the total force and moment applied by the connection system on the respective track wheel, in order to keep the crane static in this situation. Figure 58 below shows the modelled crane in this configuration.

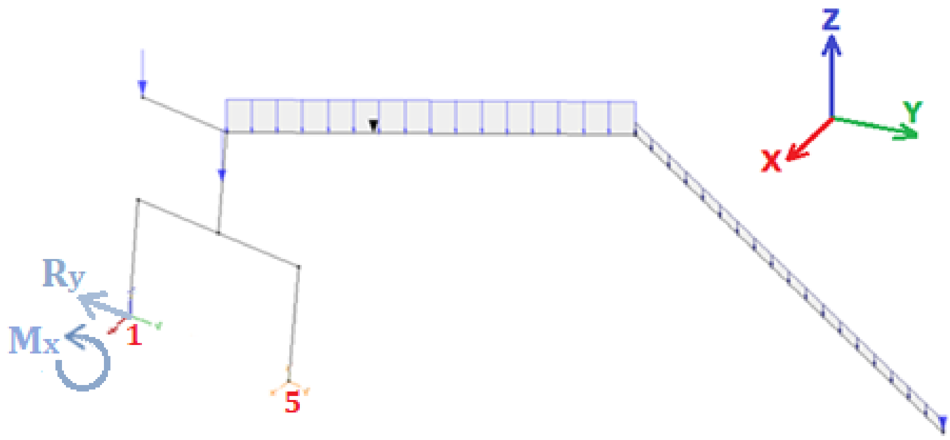


Figure 58: MultiFrame analysis model

The obtained results of the analysis are shown in Table 13 below. The joints listed in the table are the end points of each line in the crane model seen in Figure 58 (Overlapping end points of multiple lines are taken as single joints). The relevant joints here are the fixed joints at the pontoon surface, which are joints 1 and 5 (indicated with the red numbers in Figure 58). The reaction forces and moments at these joints are shown in this table. Since the turnbuckle connection system is designed to only pull (and not press), the reactions R_y and M_x are fully concentrated on joint 1. Joint 5 is therefore constrained only in the z-direction.

	Joint	Label	Ry' kN	Rz' kN	Rx' kN	Mx' kN-m	My' kN-m	Mz' kN-m
1	1		-212,602	50,830	0,000	0,000	0,000	-0,000
2	2		0,000	-0,000	0,000	0,000	0,000	0,000
3	3		-0,000	-0,000	0,000	0,000	0,000	-0,000
4	4		-0,000	-0,000	0,000	0,000	0,000	0,000
5	5		-0,000	2989,524	0,000	0,000	0,000	-0,000
6	6		-0,000	0,000	0,000	0,000	0,000	-0,000
7	7		0,000	-0,000	0,000	0,000	0,000	0,000
8	8		-0,000	0,000	0,000	0,000	0,000	-0,000
9	9		0,000	-0,000	0,000	0,000	0,000	0,000

Table 13: MultiFrame analysis results

The reaction force R_y with a magnitude of nearly 213kN is the gravitational force component generated along the pontoon surface due to the sine of the heeling angle. Furthermore, the angle causes the two contact surfaces on the ground to converge on the global Y-axis. The center of gravity of the loaded crane must at all times be in between the two contact surfaces with respect to the global Y-axis. If the angle causes both contact surfaces to come in front or behind the center of gravity on this axis, a tipping moment is created on the crane, requiring a reaction moment M_x. As seen in the table, the normal forces R_z are still positive on both contact surfaces (joint 1 and joint 5) and the reaction M_x equal to zero, meaning that the tipping moment is not reached yet with the 4° angle. Only the 213kN force in the y-direction is thus considered relevant for the operation load case.

5.5.2 Navigation

In the scenario of navigation, the crane is being transferred on sea while being fixed on the pontoon on the same position and with the same connection system as with operation. This can be for the purpose of longitudinal movements along the coastal wall in order to reach all vessel compartments, as indicated with the arrows in the top-view in Figure 59, or for transferring the crane to a different port. The crane is free from any cargo load during navigation.

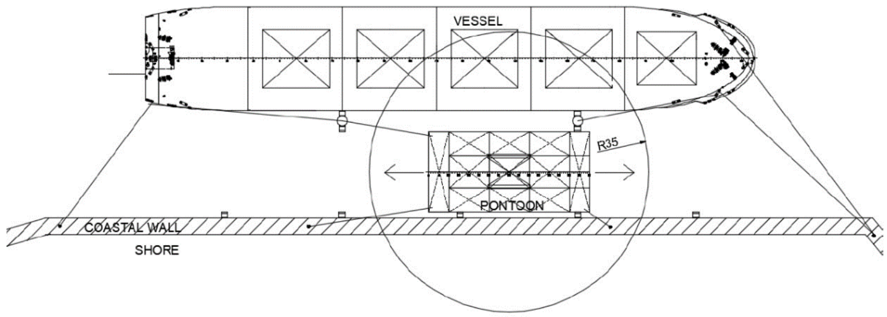


Figure 59: Top-view of the pontoon with crane in between the theoretical vessel and coastal wall [2]

Even though there are no operational center of gravity shifts in this scenario, the forward motion in conjunction with the wind and wave effects due to the environmental conditions cause particular dynamical angles and forces. The first loading distinction from the operation load case is the presence of acceleration force at navigation. The connection system must ensure that the crane accelerates steadily along with the pontoon. A reference vessel acceleration value of 0.1 m/s^2 provided by *OrcaFlex* [45] is used for the calculation. With the crane weight of 350 tons, an acceleration force of 35kN is generated. This force is directed always longitudinally to the pontoon in the x-direction.

The effect of the environmental conditions needs to be analyzed for this scenario in isolation from the cargo lifting dynamical angle. For that purpose, an analysis is run in board stability calculation software *DelftShip* [2]. The crane is modelled here with the extreme environmental conditions applied, but without any cargo load. The results give a 0.4° heeling angle, which leads to 24kN force in the y-direction. Because a tipping moment remains absent at a bigger angle of 4° analyzed in operation, it is concluded that a tipping moment remains absent in this scenario as well. The resulting trimming angle is negligibly small, so that no relevant force is considered in the x-direction due to environmental conditions.

Because the acceleration force and the environmental force are two independently acting forces, three different load cases are possible within this scenario; only acceleration force, only environmental force or both forces at the same time. At the case of only environmental force, only a force in the y-direction is considered. This case can thus be neglected since the operation case also is a force only in the y-direction, but with a higher magnitude, which means that it overlaps the case of only environmental force. The acceleration only case and the combined load case are thus added to the safety evaluation.

5.5.3 Onboarding

The final scenario to be evaluated in order to fully cover all load cases is the boarding process of the crane on the pontoon. The crane is placed from the coast to the stern of the pontoon, indicated with the blue arrow in the side-view in Figure 60. The crane is then moved towards the center of the pontoon. Before the center is reached however, a moment is caused on the pontoon due to the crane weight, resulting in a dynamical angle of trimming. This angle however can completely be eliminated by fully filling one ballast tank in the fore of the pontoon, indicated with the red arrow, so that here the trimming angle can be neglected [2].

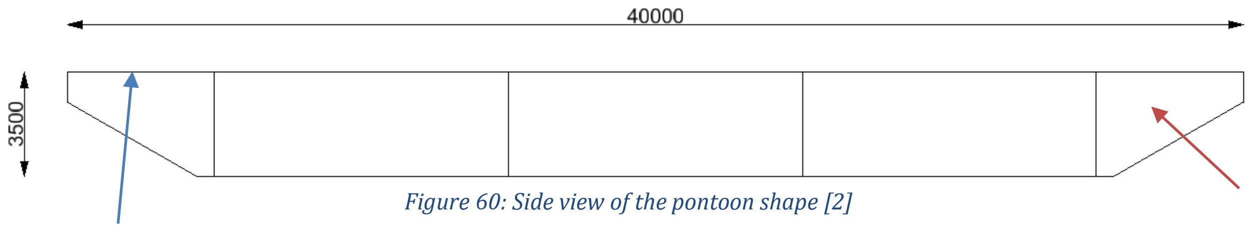


Figure 60: Side view of the pontoon shape [2]

By making the crane follow a straight line from the stern to the center of the pontoon, no heeling angles are created. The entire boarding process can thus be completed without dynamical angles on the floating body, so that no input forces need to be considered in this scenario and no connection is needed during the process.

5.5.4 Summarization

The load cases are summarized in Table 14 below with the forces on the crane as determined in the sections above. The total force values are expressed in the local crane coordinate system (x-y-z). Because no relevant input forces are found for the onboarding scenario, only the operation and navigation load cases are considered.

	Operation	Navigation (1)	Navigation (2)
$\sum F_x$	0	35kN	35kN
$\sum F_y$	213kN	24kN	0
$\sum F_z$	0	0	0

Table 14: Summarization of load cases

The translation of these loads from the crane to the connection system is displayed in the sub-sections below for the operation and the two navigation load cases.

5.5.4.1 Operation

A 3-dimensional schematic view of a free-body diagram of the integral connection system at one of the four crane corners for the load case of operation is shown in Figure 61 below. The local crane directions are shown in the upper left corner.

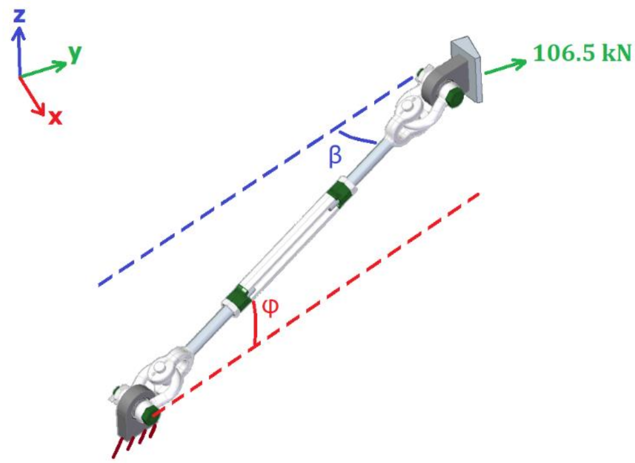


Figure 61: Operation load case

Since the tension is spread over two crane corners, the applied force per tensioned corner is 0.5×213 kN = 106.5 kN in the local y-direction of the crane. The force applied at the accessory and is transmitted through the turnbuckle to the rod end which is fixed on the pontoon. Once the turnbuckle is stretched by the force, a static situation is reached where the turnbuckle applies a force that is coaxial to its longitudinal axis which cancels the crane force. The turnbuckle force is always higher than the crane force due to the misalignment between turnbuckle and crane force. This misalignment is caused by the inclination angle β and the upward angle φ. The discrepancy angles between the turnbuckle and the y-axis is angle β about the z-axis and angle φ about the x-axis. The cosines of these angles are factored into the crane force in order to obtain the turnbuckle force. The magnitude and direction of the turnbuckle force is applied to each of the components in the system.

5.5.4.2 Navigation (1)

The body diagram for the load case of navigation (1) is displayed in Figure 62 below. The combined crane forces in the x and y directions cause particular resultant forces and deformations on the turnbuckle which are different from the other two load cases.

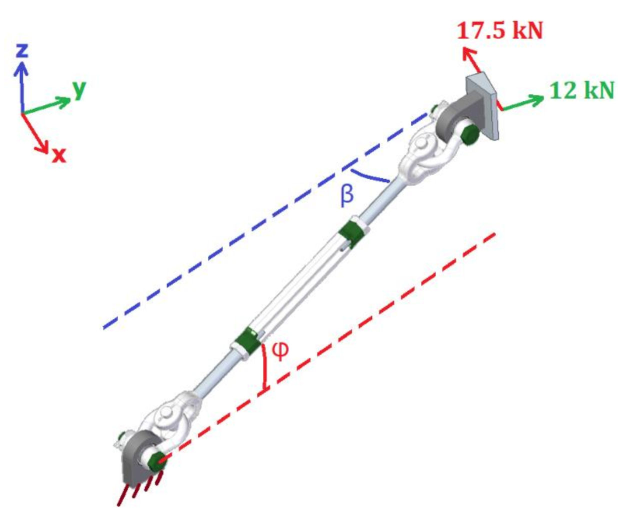


Figure 62: Navigation (1) load case

5.5.4.3 Navigation (2)

The body diagram for the load case of navigation (2) is displayed in Figure 63 below. Due to the single force in the x-direction, the deformations can have larger values here than in the other two load cases, depending on the value for angle β.

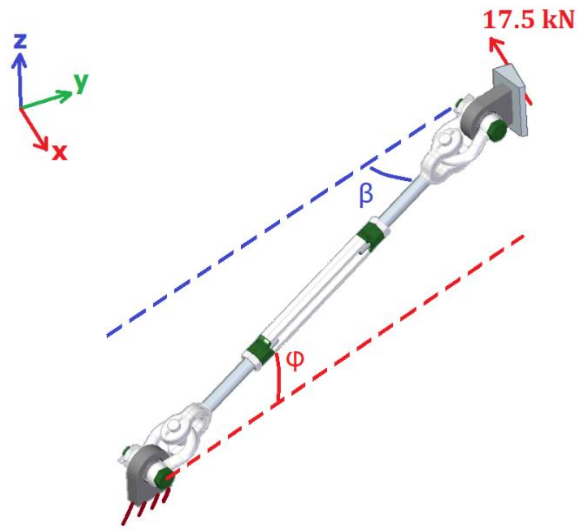


Figure 63: Navigation (2) load case

5.6 SAFETY EVALUATION

The safety requirements as determined in section 5.3 are evaluated in this section to the load cases as determined in section 5.5. The calculation method for each requirement is shown in the subsections below. The design parameters values corresponding to the minimum component size for passing the safety requirements are determined in these sections. These parameter values are implemented to finalize design phase II.

5.6.1 Strength requirements evaluation

The strength requirements are evaluated in the subsections below in the most suitable sequence, meaning that the design parameters of lowest interdependence are determined firstly and the parameters of highest interdependence lastly.

5.6.1.1 Turnbuckle size and inclination

The smallest turnbuckle size with corresponding inclination angles are found in this section. This means that the design parameters t_2 and β are found by evaluating the corresponding strength requirements. With the corresponding upward angle, design parameter h can be determined in a later stage.

The applied turnbuckle force $F_{turnbuckle}$ is equal to the input force(s) divided by the partial vector(s) of the turnbuckle alignment in the direction of the input force(s). These partial vectors are directly related to the inclination angle β and the upward angle φ as described in section 5.5.4.1. The minimal turnbuckle working load limit t_2 within component availability (Appendix E) with corresponding angles that pass requirement (11) for all load cases are listed in Table 15 below. Iterative manual goniometric calculations are performed for this purpose.

	Value
t_2	12.7t
β	20°
φ	20°

Table 15: Parameter values for t_2, β and φ

5.6.1.2 Rod end size

In order to determine the necessary rod end size, the safety requirements regarding the spherical bearing and the steel frame are evaluated. As a result, the minimal design parameter *d* within component availability (Appendix D) is obtained.
The steel frame is evaluated firstly (requirement 5), in order to see what frame size is necessary at minimum. Frame sizes are modelled iteratively in the rod end dimensions from Appendix D in FEMAP according to the sub-sections below. This is done in conjunction with requirement (6) in order to know which weld size needs to be applied to each frame size. Appendix F is used to determine the corresponding weld size for each modelled frame size.
In order to prevent bending of the square bottom surface of the frame and therefore mitigate the bending stresses in the profile, a square groove weld (full penetration) is envisioned for restriction of the bottom surface. In addition, fillets welds are placed on side of the square groove weld in order to ensure a smooth transition of the stress (as seen in Figure 64 below). The fillet welds are dimensioned here as sufficient to carry the entire load. The thickness of the square groove weld can then be kept relatively small to facilitate the welding process and is taken as half of the fillet weld size.

5.6.1.2.1 Sub FEA model reasonability

Because the maximum turnbuckle force and its corresponding direction is determined in section 5.6.1.1, the maximum externally applied force on the individual components of the connection system is known, which means that these individual components can be evaluated individually. For the frame of the rod end, no particular working load limits or usage guidelines are given, which means that a separate load analysis must be carried out for it, as stated by the manufacturer [37]. Manually solving for the combined stress flow in the profile is too complex. However, a sub FEA model of this component can be deployed for this purpose in FEMAP if the geometry, loads and boundary conditions are modelled correctly.

5.6.1.2.2 Geometry and elements

The rod end frame is modelled as a single body with the corresponding weld connection. Figure 64 below indicates which portions of the single body represent the welds and which portion represents the frame. Because the cross-section of the rod end frame and the weld combination varies only along its length and width and is constant along its thickness, plate elements are deployed with the implementation of the corresponding thickness of the frame.
Note that the welds are modelled in a schematic way, only for the purpose to simulate the correct circumstances for evaluation of the rod end frame. The Eurocode formulas (appendix F) remain governing for evaluation of the welds.

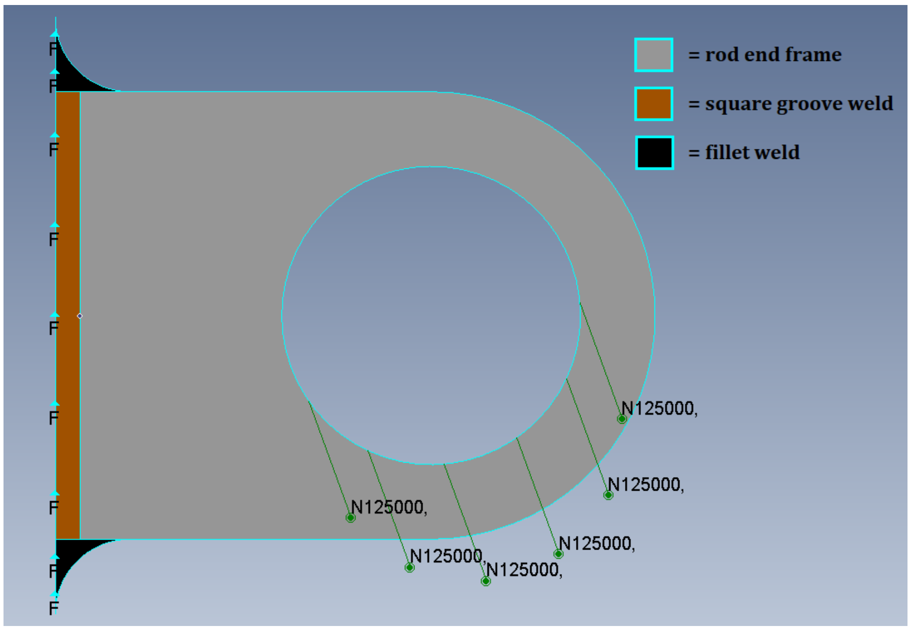


Figure 64: FEMAP model of rod end frame (sample of d=80mm)

5.6.1.2.3 Boundary conditions

The single body of the frame/weld combination is constrained in all 6 degrees of freedom with the ''fixed'' boundary condition at the bottom surface of the welds, as indicated with the ''F'' along the bottom curves as seen in Figure 64. This represents the fixation of the body to the parent surface, which can be either the pontoon surface or the accessory.

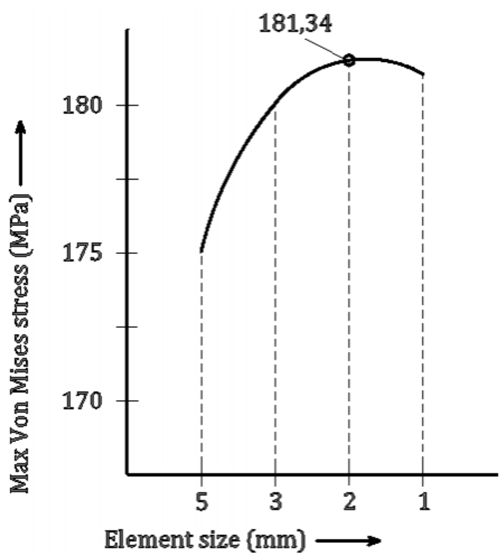
5.6.1.2.4 Loads

In order to simulate the force exerted by the embedded bearing on the frame, the ''bearing force'' option is selected and applied on the corresponding half circle of the hole in the frame (indicated with N125000 as seen in Figure 64). A magnitude of 125 kN is applied in order to match the working load limit of the applied turnbuckle in section 5.6.1.1. Because this component is placed in two perpendicular orientations on the two ends of the connection system, the direction of this force is varied in two ways; the direction corresponding to the rod end on pontoon surface and the direction corresponding to the rod end on the crane side.

As long as requirement (14) regarding the internal bearing clearance is satisfied, no additional pressures with respect to interference fits need to be implemented in the model.

5.6.1.2.5 Mesh and results

The loading direction on the rod end frame on pontoon surface appears to be more critical than on the crane side. Frames of rod ends with *d* = 80mm are able to resist this load on pontoon side with a safety factor slightly above the required 1.5, which makes it the optimal size.
A mesh sensitivity analysis shows that the model is free from stress singularities and that a stress convergence along the entire profile is reached around the mesh size of 1 mm. Graph 1 below shows the maximum Von Mises stress values on the profile of the frame located at the pontoon surface versus the implemented element size. A peak value of 181.34 MPa is found, corresponding to an element size of 2mm (at *d* = 80mm).



Graph 1: Peak stress vs. Element size (d=80mm)

Figure 65: Stress flow along profile with 2mm elements (d=80 mm)Figure 65 below shows the stress flow along the profile for *d* = 80mm. The critical stress values are located at around the hole of the frame.

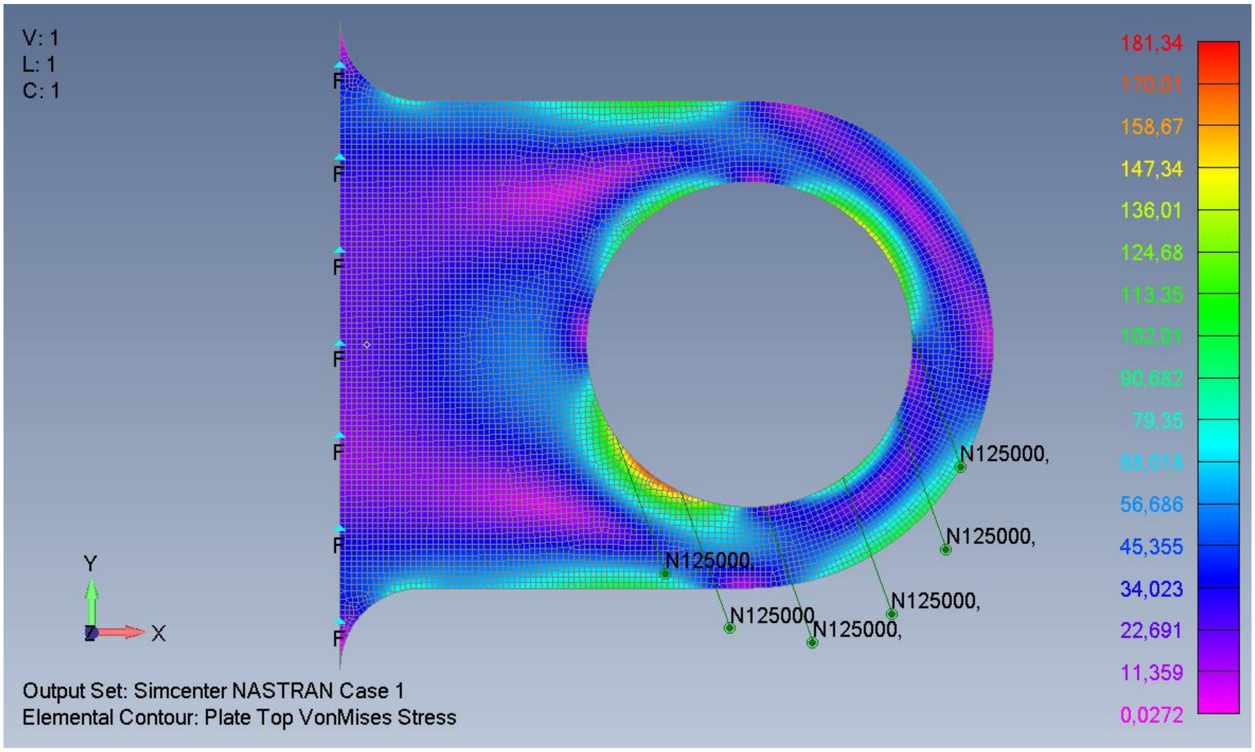


Figure 65: Stress flow along profile with 2mm elements (d=80 mm)

2cm (along the thickness of the frame only) is considered sufficient according to the performed Eurocode calculations for the size of the fillets of the welds at this frame size (this weld size is also applied to the FEMAP model).
The two bearing requirements, (3) and (4), are thus evaluated for d = 80mm and satisfied as well. The axial load factor X at the maximum allowable tilt angle multiplied by the radial bearing force is for every load case smaller than the dynamic load rating C_r and the pulsating load factor 2.75 multiplied by the radial bearing force in for every load case smaller than the static load rating C_{0r}. It is thus concluded that the suitable rod end size is d = 80mm.

	Value
d	80mm

Table 16: parameter value for d

5.6.1.3 Preliminary shackle parameters

Before finding the suitable dimensions for the bore adapter, a preliminary shackle size is found firstly, since the nominal inner diameter of the bore adapter is equal to the shackle pin diameter, by which the strength calculations for the bore adapter must be based on.
The preliminary shackle size found is this section consists of the available manufactured sizes that satisfy requirements (9) and (11).
The bow forces in correspondence to the input forces of the load cases require shackles with a minimum working load limit of t_1 = 13.5t to satisfy requirement (9).
The width of the rod ends within the determined value for d is 63mm. The shackle dimension ''width inside'' (e in Appendix C) must at least be bigger than 63mm in order to be suitable with the

applied range of rod ends, so the corresponding minimum value for the bow length design parameter ℓ is 178mm.
These values for the shackle design parameters are to be fine-tuned with shackle sizes that pass also requirement (10). For determining the Von-Mises stress on the shackle pin however, knowledge of the interference fit stress interaction between shackle pin and bore adapter is necessary, which is determined in section 5.6.1.4 below.

5.6.1.4 Bore adapter/shackle pin interaction

The nominal outer diameter of the bore adapter is equal to the rod end bore diameter *d*. This outer diameter is slightly increased with the m6 interference margin (0.028 mm for d = 80mm) as prescribed by rod end manufacturer *Schaeffler* [37]. The nominal inner diameter is equal to the pin diameter of the used shackle. In this section, the possible amount of applicable interference δ between the shackle pin and inner bore adapter diameter without reaching the yield stress on either the shackle pin or bore adapter is determined, which corresponds to requirements (8) and (10). The equivalent Von-Mises stresses on the shackle pin and the bore adapter as a result of the interference fit stresses in conjunction with the load cases is determined for this purpose by using the preliminary parameter value for t_1 (as determined in the previous section) for the shackle pin dimensions.
The amount of interference δ is varied in the calculations in accordance with standard ISO 286-2 for shaft tolerances [46]. These standardized fit sizes are used as increments for the value for δ, which is iteratively implemented in formula (2) below in order to find the amount of interference pressure (radial) on the shackle pin surface.
The interference fit pressure is calculated with the contact surface interference fit pressure formula as given by bearing manufacturer *GGB Timken* [47];

$$p = \frac{\delta}{\frac{d}{E_o}\left(\frac{d_o^2 + d^2}{d_o^2 - d^2} + v_o\right) + \frac{d}{E_i}\left(\frac{d^2 + d_i^2}{d^2 - d_i^2} - v_i\right)} \tag{2}$$

With;

p = interference fit pressure (Pa)
δ = interference margin (m)
d = nominal shaft/bore size (m)
d_o = outer diameter housing (m)
d_i = inner diameter shaft (m)
E_o = Young's modulus housing (Pa)
E_i = Young's modulus shaft (Pa)
v_o = Poisson ratio housing
v_i = Poisson ratio shaft

In order to obtain the radial stress on the pin due to the input loads, the radial force value in the load cases is divided by the loaded rectangular mid-surface of the pin (diameter x length of bore adapter) in accordance with the ''bearing stress'' rule, which states that the reaction force in cylindrical contact (without clearance) is distributed equally and radially around the pressed side

of the cylinder (half of the cylinder) [48]. The radial input stress is superposed on the interference pressure to obtain the total radial stress on the pin.
The pin has a full solid diameter, which means that the circumferential stress is equal to the total radial stress [49].
Because the radial input load acts transversally to the cross-section of the pin, bending moments are created, leading to an axial stress component on the pin. Properties of the circular cross-section and length of the pin are used for obtaining the axial stress value. The transversal force furthermore induces a shear stress, which is determined with the maximum shear stress formula for circular cross-sections (formula (9) in section 10.1.1).
These radial, circumferential, axial and shear stress values are used for determining the equivalent Von-Mises stress on the critical locations on the shackle pin.
Because the bore adapter has a hollow profile, it is modelled as a thick-walled pressure vessel with a combined external and internal pressure. The radial and circumferential stresses are determined with the pressure vessel formulas below;

$$\sigma_c = \frac{r_i^2 p_i - r_o^2 p_o - r_i^2 r_o^2 (p_o - p_i)/r^2}{\left(r_o^2 - r_i^2\right)} \tag{3}$$

$$\sigma_r = \frac{r_i^2 p_i - r_o^2 p_o + r_i^2 r_o^2 (p_o - p_i)/r^2}{\left(r_o^2 - r_i^2\right)} \tag{4}$$

With;

σ_c = circumferential stress
σ_r = radial stress
r_i = inner diameter
r_o = outer diameter
r = diameter of stress evaluation
p_i = internal pressure
p_o = external pressure

By determining the internal and external pressures p_i and p_o, the maximum stresses in radial and circumferential direction can be found.
The external pressure p_o is composed out of a superposition of the following two pressures:

- The interference fit pressure between the bore adapter and the rod end bearing
- The pressure on the outer surface due to the radial input force in the load cases

The bore adapter is inserted in the rod end with an m6 fit. This interference between bore adapter and rod end is further magnified once the shackle pin is fitted inside the bore adapter with interference ⃠. The maximum theoretical radial deflection of the bore adapter due to the pin

interference is thus equal to the δ between pin diameter and inner bore adapter diameter. The final value used for the amount of interference between the bore adapter and the rod end bearing is thus $\delta_{m6} + \delta_{\oslash}$. This interference value is used in formula (2) to obtain the total external interference pressure on the bore adapter. The radial pressure due to the load cases is determined with the ''bearing stress'' principle (as discussed above). The loaded rectangular surface area is taken here as the outer diameter multiplied by the length. This radial pressure is added to the interference pressure to obtain the total external pressure p_o.
The internal pressure p_i on the inner bore adapter surface is equal in magnitude to the radial pressure as determined for the shackle pin surface. With both p_i and p_o known, the circumferential and radial stresses σ_c and σ_r can be found for various values of r.
These radial, circumferential, and shear stress values are used for determining the equivalent Von-Mises stress on the inner and outer surface (extreme stress locations) of the bore adapter.
The performed calculations lead to the conclusion for the preliminary shackle working load limit t_1 and bow length ℓ no ISO interference fit between the bore adapter and shackle pin is possible without reaching the yield stress on the pin. By implementing the subsequently higher working load limit of t_1 = 25t, the tightest possible ISO interference fit is j6. The final values for design parameters t_1, ℓ and $\oslash$ is found in Table 17 below. Due to requirement (10), the range for t_2 is updated as well, in order to match with the range for t_1.

	Value
t_1	25t
ℓ	178mm
$\oslash$	j6
t_2	16.8t

Table 17: Parameter values for t_1, ℓ, $\oslash$ and t_2

5.6.1.5 Accessory and welding area

In order to evaluate the feasibility of the accessory, the required size of this component is determined. This size, including the corresponding welds, must be suitable with the available height (765mm) of the steel beam inside the track wheel of the crane in order to satisfy requirements (1) and (2). The placement in height (design parameter h) of the accessory must be conform the determined upward angle of the turnbuckle, which is $\varphi = 20°$.
The same combination of the square groove weld in conjunction with the fillet welds as applied with the rod end is applied here as well.
Investigation into the lowest material-consuming dimensions (smallest volume) for the accessory which satisfy the requirements is performed with FEMAP according to the sub-sections below.

5.6.1.5.1 Sub FEA model reasonability

Due to the trapezium shape of the component, the multi-axial loading direction and unknown stress concentration factor at the connection with the track wheel frame, a sub FEA model of the accessory is necessary to solve for the stress flow throughout the profile. By integrating the accessory and track wheel frame into a single body, the actual stress concentrations on the flat surface of the accessory are simulated. The stress concentration factor at the inclined surface due to the rod end connection must also correctly be simulated. The rod end (including welds) is therefore modelled here as well and integrated with the accessory as a single body at the corresponding weld connection. The force applied to the rod end is known and can be implemented in the model. This force is transferred from the rod end to the accessory due to the solid connection. The determined weld area for the rod end in section 5.6.1.2 is used here.

5.6.1.5.2 Geometry and elements

In order to minimize the material consumption, the inclined surface of the trapezium (where the rod end is connected) is ideally dimensioned as equal to the bottom surface of the rod end including its welds (smallest possible surface). The stress concentration factors on the stress results obtained for the rod end are furthermore also based on the usage of this minimized connection surface. The downside of minimizing this profile is that the stress concentration factor on the flat surface of the accessory is maximized due to the big area difference with the corresponding track wheel frame surface. It is thus investigated whether these ideal dimensions can be applied to the accessory. The track wheel frame (marked in brown in Figure 66) is modelled in its given dimensions as a rectangular bar. However, because the bar thickness is irrelevant for this calculation, only a small portion of the frame thickness is modelled in order to reduce the computational time. It is integrated as one body with the accessory, which is attached nearly at the end of the bar surface as seen in Figure 67 below.

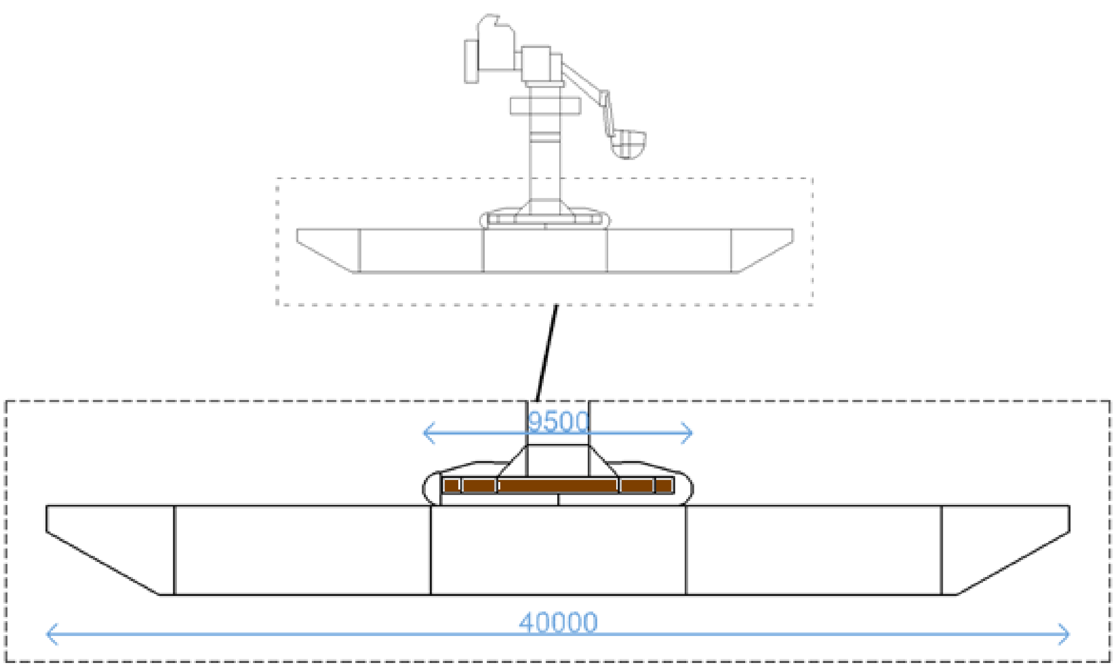


Figure 66: Side-view of crane and pontoon with track wheel area detail

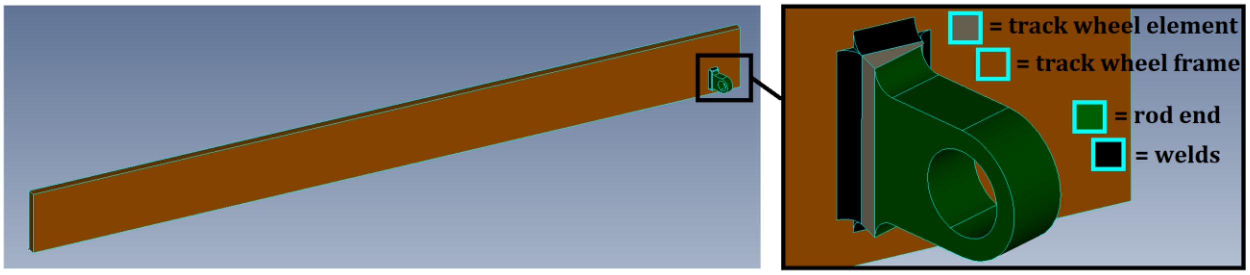


Figure 67: FEMAP model of accessory and frame

The determined inclination angle of β = 20° (section 5.6.1.1) is applied to the shape of the accessory here. Also, the determined upward angle of φ = 20° is applied, which results in a placement height of 71mm for the design parameter *h*.
Because the mid-surfaces do not coincide in this configuration, solid elements are used instead of plate elements. Since the results on the track wheel frame are irrelevant, a relatively rough element size (50mm) is used for the frame part. A relatively fine element size (≤ 5mm) is used on and around the accessory.

5.6.1.5.3 Boundary conditions

The modelled track wheel frame is in essence a cut-off section from a thicker steel beam. The entire back-surface of the modelled frame is therefore restricted in all 6 degrees of freedom with the ''fixed'' condition as seen in Figure 68 below. Note that this is a back-view of the body, as opposed to the front-view seen in Figure 67.
The connection of the accessory to the frame is envisioned as the same square groove weld and fillet weld combination as applied to the rod end. This means that the accessory including the welds can simply be modelled as a single solid body with the frame and no further constraints need to be implemented.

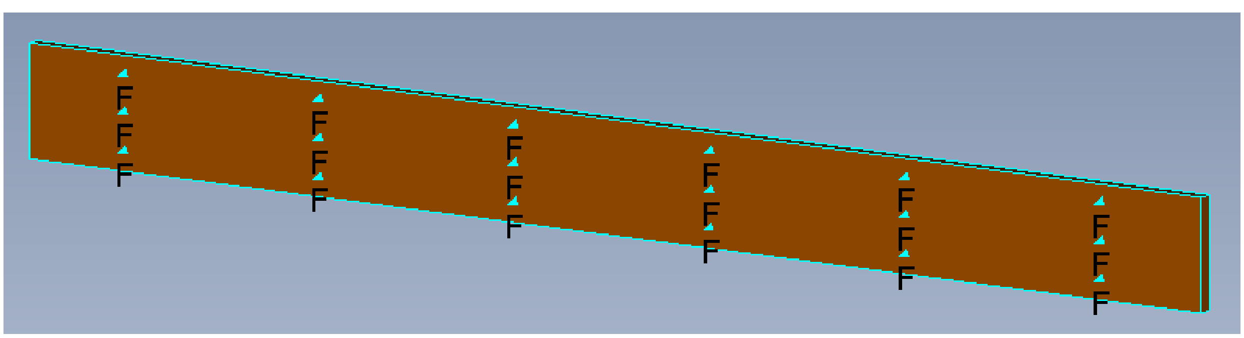


Figure 68: Fixed surface of track wheel frame

5.6.1.5.4 Loads

The turnbuckle force of 125kN is applied to the rod end with the ''bearing force'' option as performed in section 5.6.1.2. An inner half-cylindrical surface, directed 20° downwards, is defined to apply the force on as seen in Figure 69 below.

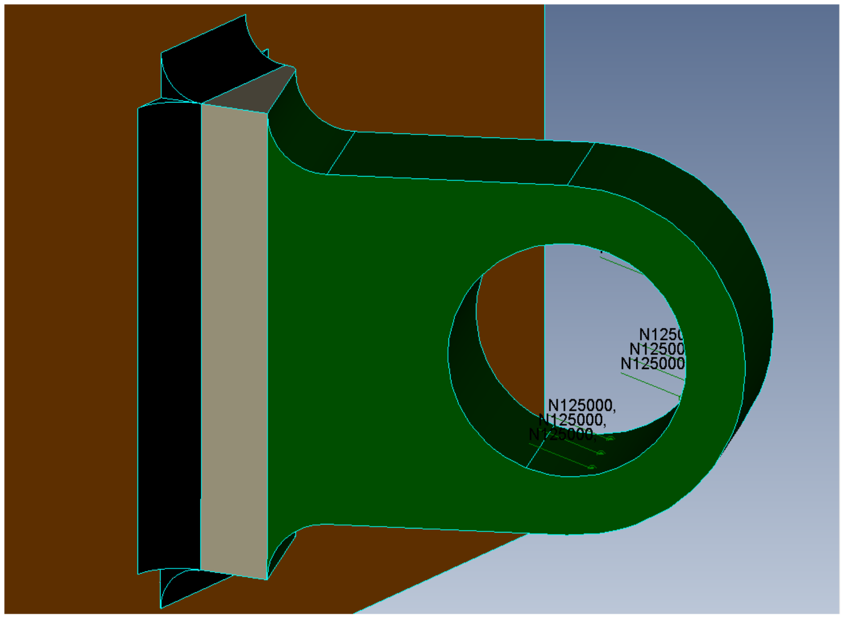


Figure 69: Applied load on accessory

5.6.1.5.5 Mesh and results

The mesh size on the track wheel frame and on the rod end has been kept constant at 50mm and 3mm respectively. The mesh size on the accessory is set at 3mm for the first iteration and 1.5mm for a second iteration, which resulted in nearly identical stress results of approximately 30 MPa (± 2.3%) at the critical area of the accessory. Considering these obtained results, it is concluded that the yield stress including safety factor (156,7 MPa if steel grade S235 is used) is not reached on the accessory if the dimensions for minimization of the material consumption are used. Figure 70 below shows the found stress flow along the profile.

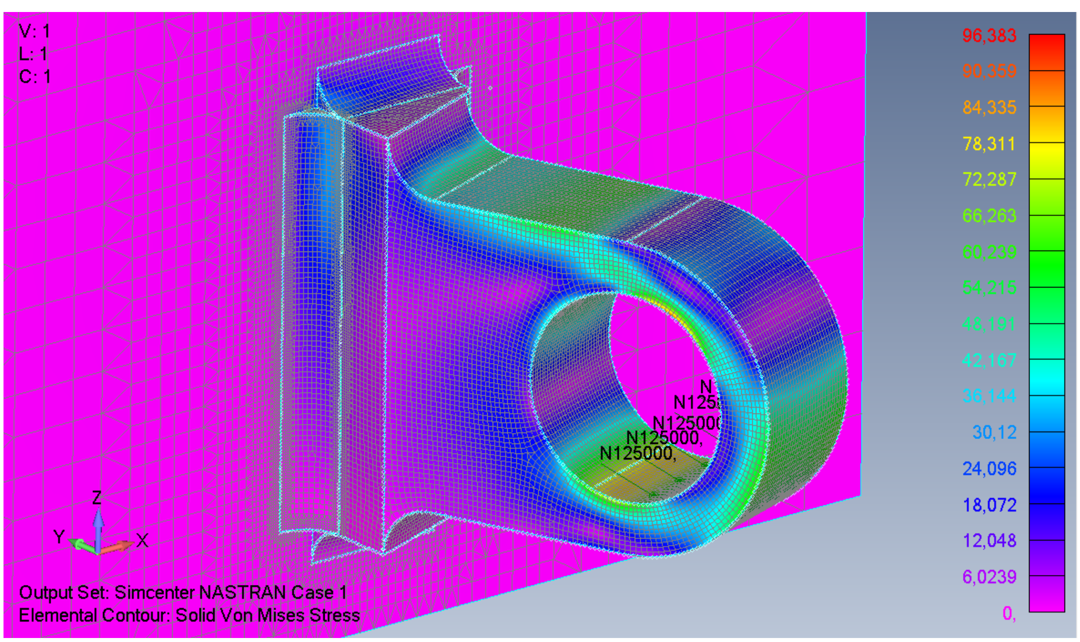


Figure 70: Stress flow along profile

With the use of these dimensions, the corresponding design parameter value for the placement height *h* is thus 71mm.

	Value
h	71mm

Table 18: Parameter value for h

5.6.2 Deformation requirements evaluation

With the strength requirements verified, it is evaluated it this section whether the determined values of the design parameters corresponding to the strength requirements must be narrowed down further in order to comply also with the deformation requirements.

5.6.2.1 Tolerance

It is evaluated whether the deformations in the load cases exceed the allowable tolerance, which is the free space in between the rod ends and turnbuckles as explained in section 5.3.2.1. It is analyzed for this purpose in FEMAP what the extreme turnbuckle elongation is.

5.6.2.1.1 Sub FEA model reasonability

Due to the large discrepancy in length between the turnbuckle and the rest of the components, the elongation of the rest of the components is considered negligible compared to the elongation of the turnbuckle. By modelling a conservative representation of the turnbuckle and subjecting it to the maximum turnbuckle force, an upper limit value for the turnbuckle elongation can be extracted, which can be used to determine the maximum deformation of the system. A sub FEA model of the turnbuckle is therefore considered appropriate for this purpose.

5.6.2.1.2 Geometry and elements

The geometry of the turnbuckle is simplified to the use of rectangular and cylindrical shapes comprised into a single body as seen in Figure 71 below.

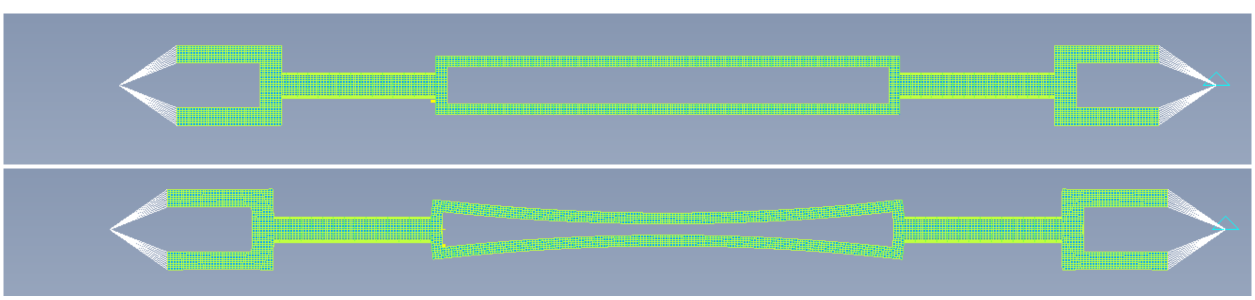


Figure 71: Elongation analysis of turnbuckle in FEMAP

The upper image of the figure shows the turnbuckle in the undeformed state and the lower image shows the deformed state (with a scale factor implemented). It is ensured that applied adaptations with respect to the actual geometry do cause a more conservative end result for the elongation of the body. Due to the use of cylindrical shapes, solid elements are used to build up the model.

5.6.2.1.3 Boundary conditions

The turnbuckle is at one end (right side of Figure 71) constrained in all degrees of freedom with the ''fixed'' boundary conditions. This is done by applying this constraint to a ''rigid element'', which is linked to all nodes at this end of the turnbuckle.

5.6.2.1.4 Loads

The maximum turnbuckle force of 125 kN is applied at the other end of the turnbuckle (left side of Figure 71) in axial direction. This is done by applying the load to a rigid element, which is linked to all nodes at this end of the turnbuckle.

5.6.2.1.5 Mesh and results

The mesh size of the solid elements is varied between 3mm and 5mm. It is obtained from the analysis that the elongation on the turnbuckles under extreme loading conditions is not larger than 2mm. By drawing deformation triangles in Autocad with the original and deformed turnbuckle length (as depicted in Figure 55 in section 5.3.2.2), The deformation distance ΔTB is deduced with measurements. It is found that the extreme values for ΔTB do not exceed 10mm. With this value for ΔTB, the determined values for the design parameters can be used for satisfying the equation of requirement (12).

5.6.2.2 Tilt angle

Measurements of the tilt angle (α in Figure 55) within the drawn deformation triangles give an upper limit of 0.5° for the value of the tilt angle, which is below the allowable tilt angle α of the rod end. Requirement (13) is thus fully satisfied within the determined values of the design parameters.

5.6.2.3 Radial internal clearance

The expansion of the inner ring of bearings within the range of d = 80mm is according to *Schaeffler* [37] equal to 0.72 multiplied with the amount of interference δ of the inserted shaft. δ is conservatively counted here as the sum between the amount of interference at inner and the outer diameter of the bore adapter. The interference at the outer diameter is calculated according to the m6 tightness as prescribed for the shaft insertion in the bearing. The interference at the inner diameter depends on the shackle pin diameter and the used tightness ⃠. The performed calculations lead to the conclusion that the determined combination of design parameter values do not lead to an exceedance of the radial internal clearance of the rod end bearing. Requirement (14) is thus satisfied.

5.7 Summary

The dimensions of the conceptual design from design phase I are fully defined with the quantified design parameters in design phase II. It is found in this phase that the designed innovations with respect to removability in crane-to-pontoon connections and compression-free turnbuckle utilization are realizable within component manufacturer availability while passing all strength and deformation requirements. With the applied minimization of material consumption, it is furthermore obtained that these parameter values lie within the range of feasible construction time and effort, meaning that the size and weight of each used component can be handled manually by the cooperation of two construction workers.

These results correspond to the applied set of input data, which is chosen to generally represent heavy-duty crane utilization on a pontoon. It can thus be concluded at the end of this research question that in general the aim of developing a removable crane-to-pontoon connection system and to upgrade the official connecting option list by the United States Department of Labor (section 1.1) with a removable fifth option is reached. However, it is not known yet what the gains in terms of efficiency in port equipment utilization are of the removable design in comparison to the conventional connection systems listed by the United States Department of Labor. In order to fully achieve the aim of the thesis, the efficiency increase needs to be confirmed, which is elaborated by the next research question in the next chapter.

6 EFFICIENCY QUANTIFICATION

6.1 GENERAL

With both design phases completed and the design parameters quantified, a final investigation is carried out in order to answer the fifth research question: How to quantify the efficiency of the design? In other words; what is the added value of the removability feature of the new design relatively to the conventional connection systems? This can be quantified by the amount of time saved at connection and disconnection, which is convertible to the amount of money saved, as elaborated in the sections below. A long-term projection of the saved costs is thereafter made, which is anticipated after multiple removals/reconnections with the removable system.

6.2 QUANTIFICATION OF SAVED TIME

In order to quantify the amount of time saved with the removable connection system relatively to the conventional connection systems, the two following assumptions are made;

- The four conventional connecting options, listed by the United States Department of Labor, consume at least a full day to complete all the labor to establish the connection and disconnection. There is no reference data available to find the exact times needed for these connections. However, the assumption is based on the approximate time consumption of tensioning of cables and ropes, cutting and welding of barricade parts and the production/placement of rails, which are prescribed by these four options. (section 3.3.1 provides reference configurations which envision the conventional connection systems)

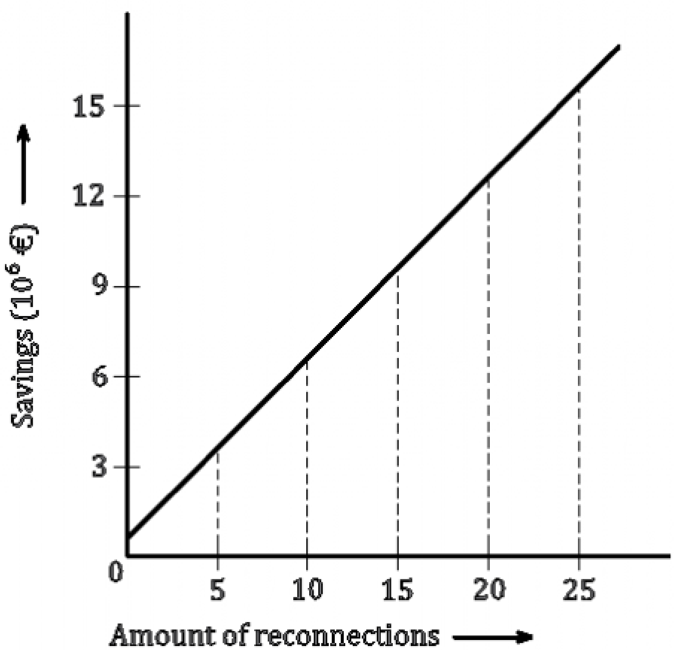
- Because the turnbuckle system is removable and thus reusable, it is connected and disconnected by simply removing the turnbuckles. With the determined design parameter values in the previous research question, it is found that the applied turnbuckles and other components are of feasible size to be handled simply by hand, which gives the assumption that the connection and disconnection time for the turnbuckle system is negligibly low compared to the conventional systems.

Resulting from these two assumptions, it is obtained that at least a full day is saved per connection with the removable system relatively to the conventional systems.

6.3 QUANTIFICATION OF SAVED COSTS

The amount of time saved on connecting is considered as additional time that is made available for crane operation. The amount of potential revenue generated within the time span of the additional operational time of one full day is of relevance here. The realistic revenue generated on a daily basis by operation with the type of heavy-duty crane as applied in the input data (section 5.4.1) with a capacity of 16t is in the range of €600000-€700000, according to data retrieved by IHC [2]. The

material expenses for the removable design need to be deducted from this, and are determined in section 6.3.1 below.
As seen in that section, the total costs for the system are nearly €14000. The material costs for the removable design are thus considered negligible and immediately recovered at the day of implementation, since €14000 << €600000.
Moreover, an additional day of operation is gained at each subsequent reconnection of the crane to the pontoon with the removable connection system. The total acquired savings accumulate thus over time and depend on the alternation frequency between land and pontoon operation. Graph 2 below shows a schematic representation of the relation between the potential savings and the projected amount of reconnections when the removable system is used at the reference (heavy-duty) scale-size. It can be seen that with a relative high amount of projected reconnections, the potential long-term savings reach heights of 10+ million €. (Note: a single removable connection system can in principle only be re-used with the same crane. When a different crane is used, different design parameter values apply to the connection system, requiring the re-calculation and re-purchase of connection system components).



Graph 2: Savings versus amount of reconnections

Significant profit is thus anticipated when applying the removable connection system instead of a conventional connection system when multiple removals/reconnections are projected. Even if only a one-time connection (0 reconnections) of the crane to the pontoon is desired, the removable design is the more profitable and efficient option over a conventional crane-to-pontoon connection system.

6.3.1 Price

The largest expense of the removable connection system design are the turnbuckles. The size of 16.8t can be purchased for €1662.50 per turnbuckle [50]. A total of four turnbuckles costs therefore €6650. The second largest expense are the rod ends. The size of d=80mm can be purchased for €560 per rod end [51]. A total of 8 rod ends costs therefore €4480. Furthermore, shackles of size 25t cost €289 per shackle [52]. A total of 8 shackles costs therefore €2312. The price for the additional steel needed to create the accessories and the bore adapters costs €505.62 (*Parkersteel* [53]). The total price for the connection system is thus summed up to €13948.

6.4 SUMMARY

A quantification of the efficiency of the removable design is performed in terms of time saved relatively to the conventional connection systems. Based on the various constructional actions required by the conventional connection systems, it is assumed that one full day is saved with the removable design per connection. This amount of time is converted to the amount of money saved, by taking the expected revenue of one operational day, which is found to be in the range of €600000-€700000 at the reference crane size and capacity.

The material price of the removable design is found to be negligible in proportion to the gained time-related savings that the removable design makes (€14000 : €600000). It is therefore concluded that the removable design provides instant profit with respect to a conventional system. An additional €600000-€700000 is saved with each subsequent reconnection, so with a high amount of projected reconnections, multiple millions of saved € are accumulated over time.

7 CONCLUSION

7.1 REFLECTION

It can be concluded that the aim of developing a crane-to-pontoon connection system that enables and facilitates a standard land crane to be utilized as a hybrid operational crane that can operate on a pontoon or a shore on demand, increasing the overall efficiency in port equipment utilization, is accomplished. The developed design provides the desired removability as well as simplicity for the end-user. It is furthermore found at the quantification of design parameters that construction time and effort are within the desired economical range. At the efficiency quantification, it found that €600000-€700000 is gained per reconnection, so that it potentially accumulates up to 10+ million € if a high amount of removals/reconnections are projected for a particular crane.
Considering the findings made throughout the sub research questions, the main research question (How can a removable crane-to-pontoon connection system be developed in order to turn a standard land crane into a hybrid operational crane that can operate on a pontoon or a shore on demand?) obtains the following answer; *by applying removable turnbuckles in conjunction with components that eliminate compression stresses, account for the multi-axial loading and asymmetry of the system, incorporate tight interconnections while maintaining flexibility and are prefabricated for the most part. These components are the bow shackle, rod end with spherical bearing, bore adapter and angular crane accessory.*
The scientific value of the performed research can primarily be defined as the addition of removability in crane-to-pontoon connection system, which creates the potential for significant long-term savings and increased operational efficiency. Furthermore, scientific progress is made through the development of a mechanical system which cancels out compression stresses in a turnbuckle, which is known as the primary weakness of turnbuckles. The designed system allows turnbuckles to be used in applications where compressive motions are present, where the use of turnbuckles would normally be avoided.

7.2 LIMITATIONS

The positive outcome of the research as elaborated in the reflection is subject to potential limitations, which are discussed in this section.
The relevant simplifications taken in design phase II, related to the preloaded condition of the turnbuckles, imply that the crane and pontoon move as one body at all times and that the loads on the turnbuckles are proportionally distributed at all times. The reasonability for this assumption is strengthened by mandating repetitive inspection of the preloaded condition on all four turnbuckles and the implementation of safety factors. A particular unknown margin of error is however not excluded within this preloaded state.
The quantification of parameters is limited to a single set of input data, which are given as a general representation of heavy-duty crane utilization on pontoons, which provided the proof of general effectivity of the developed system. However, it is unknown what the exact outcome of parameters would be with a less favorable set of input data and how it would impact the effectiveness of the system.

The height of the projected cost-savings in section 6.3 are based on the assumption that the connection time for a conventional connection system takes at least a full day, which resulted in the provision that at least a full day of operation is gained due to the negligible connection time of the removable design. In reality however, the exact duration is unknown due to unavailable reference data on this topic and dependency on multiple company-related factors. The projected savings are therefore to be taken as preliminary. Nevertheless, the positive trend as shown in Graph 2 and the anticipation of instant profit are valid in any case.

7.3 FOLLOW-UP RESEARCH

A follow-up research in correspondence to the described limitations is suggested. This includes a deeper investigation into the preloaded condition and the margin of error of its corresponding assumptions. Another relevant topic would be the impact on the effectiveness and efficiency of the system when altering the input data. This would provide a clearer spectrum of crane scale-sizes which are suitable for utilization with the removable connection system. Furthermore, an investigation into connection times of conventional connection systems is suggested. By scientifically determining boundaries for these duration values, the total cost-savings of the removable design can be projected with more accuracy.

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9 APPENDIX A (SCIENTIFIC RESEARCH PAPER)

Paper starts on next page

Hybrid Land/Pontoon Crane: Development of a Removable Connection System

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***Abstract* – An unfamiliar subject to the contemporary connecting options for cranes on pontoons is removability; Various difficulties and obstructions related to the heavy-duty nature of the equipment lead to a limited amount of practiced conventional connecting options for this purpose, which all have the absence of removability in common. Furthermore, the consideration of various types of efficiencies and performance aspects has not been common practice within this field. The aim of developing a removable and efficient crane-to-pontoon connection system is therefore set. The turnbuckle option obtained the highest rating among potential connecting options and is therefore selected to proceed the design process with. Developing the turnbuckle option into a complete connection system and accomplishing all defined aims led to an encounter with various engineering challenges. An integral design process led to the discovery of a proficient combination of components, which overcome the challenges and provide satisfaction with respect to the aims of the project. The parameters of the developed conceptual design are finally quantified in order to prove feasibility and efficiency. Applicable design parameters are found which pass the safety requirements, while minimizing the material consumption. With these parameters, the removable design is compared to conventional real case connection systems in terms of cost-efficiency, which resulted in the observation that multiple €10^6 in long-term savings are anticipated per deployed crane due to the removability feature.**

I. INTRODUCTION

In current engineering practice, cranes are designed specifically for their applications. However, occasionally it can occur that a land crane is deployed for operation on a floating pontoon. This can have various causes; A vessel might for instance not be able to dock the shore due to shallow water, or a particular shore might not be equipped with cranes [1]. What the contemporary connecting options, as prescribed by the *United States Department of Labor* [2], have in common is that they cannot be simply removed and re-applied if necessary. This is a result of the magnitude of the forces associated with the dynamical angles of the floating body. These connecting options are therefore generally applied in way that it can withstand high magnitude loads, which make them inconvenient to apply and to remove.

Considering the problem above, the aim of this thesis is to develop a crane-to-pontoon connection system that enables and facilitates a standard land crane to be utilized as a hybrid operational crane that can operate on a pontoon or a shore on demand, increasing the overall efficiency in port equipment utilization.

The outline of the research starts with a chapter on the state of the art with respect to removable connecting options in the form of a literature research into potential connecting options. The subsequent chapter consists of an appointment of relevant performance aspects and indicators, an analysis into boundary requirements for the connecting options within these indicators and the final selection procedure for the most suitable option. The fourth chapter is dedicated to the conceptual model design of the new connection system. All of the main design parameters and characteristics are determined. Subsequently, the fifth chapter provides the structural analysis and safety evaluation of the system, including an input loads and motions analysis, leading to quantified design parameters which pass the safety requirements, followed by the sixth chapter, which gives the efficiency quantification and comparison of the design with respect to conventional connection systems.

II. CONNECTING OPTIONS LITERATURE

A literature research is performed to explore a variety of connecting options that have the potential of enabling simple connection and disconnection between crane and pontoon. The range of sources used purposefully exceeds the topic of cranes and pontoons only, in order to find new solutions which might be applied in this field. A collection of 6 connecting options are found which have characteristics that can potentially satisfy the aims of this project. The essence and downsides of each option is discussed below.

1. Fish plate pin configuration

Fish plates can be designed in a way that allows them to be effective in multiple directions with simple pins. Figure 1 below shows a tower crane connection with its foundation on land.

Fig. 1: Tower crane connection with pins [3]

It can be seen that with two big pins and one smaller pin, the crane is prevented from shifting in any direction. In principle, there should be no external pressure on these pins when the crane is out of operation, so that they can be pulled in and out manually.

2. Turnbuckle with pin

A turnbuckle is a device which is meant to be placed in between ropes or cables in order to apply the necessary tension on a particular fastening system. Turnbuckles are produced with various types of end fittings as seen in figure 2.

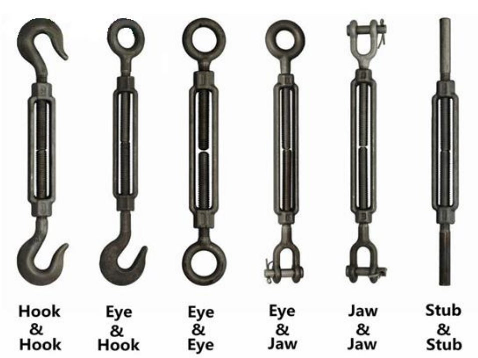


Fig. 2: Different end fitting configurations [4]

The fitting that corresponds to the requirements of the potential crane to pontoon connection system would be the jaw & jaw fitting with cotter pins as seen in the figure, which would allow for a simple connection and disconnection. A relevant advantage if applied for the new crane to pontoon connection system is the variable length and the ability for pre-tensioning the system.
As mentioned, turnbuckles are designed to take in tension stresses. If applied at the connection between crane and pontoon however, these devices will also be subject to compression stresses. Due to the slender frame that turnbuckles usually have, buckling is expected to arise as a problematic factor in this case.

3. Spring clip technology

The main principle of any spring clip is based on a particular spring force which needs to be overcome in order to click the desired object onto the clip. Once the object is clicked, the clip returns to its initial position, blocking the object from moving away unless the same force is applied on the spring again. *Böllhoff group* [5], specialized in fastening technology, has been able to develop an ''innovative and high-strength quick fastening'' solution in number of variants, using spring clip technology. below shows one variant of these spring clips by Böllhoff. It has a black frame which encloses two separate connection sides and each side has its own steel clip element which can be pushed into the frame.

Fig. 3: DST double fastener by Böllhoff [5]

Even though this removable connecting principle possesses the factor of simplicity, its usage is originally intended for small-scale applications within these products.

4. Spring clamps

More variants of spring-based connections are possible. spring clamps can be found at heavy-duty lifting applications. The spring in these clamps is typically engaged simply with a lever. Locking the lever clamps the object firmly in all directions within the working capacity. The object is released again by simply reversing the lever. Figure 4 shows a lifting clamp with its locking lever and the black frictional surface clamp connected to the spring.

Fig. 4: Lifting clamp with locking lever [6]

Since these clamps are designed for lifting purposes, it can be assumed that their capacity is rated only for outwards forces. It is therefore unknown which structural response can be anticipated when inwards forces are applied (which would be the compression forces if applied between crane and pontoon). Unfortunately, their frames resemble in most cases a circular design which have relatively short lengths from end to end without much variation.

5. Bar clamps

Bar clamps are generally used at workspaces and are able to secure any piece of material in between two surfaces. The surfaces can for instance be made in custom shapes, giving a more advantageous clamping, as can be seen with the bar clamp in figure 5. The movement of the surface is realized in this case with a handle.

Fig 5: Bar clamp [7]

Even though clamping an object with these tools would go relatively comfortably, this type of connection strategy would involve a number of issues if applied between crane and pontoon. Since these tools are in principle not designed to take on external stresses, It is questionable for which type of loads they are suitable. Furthermore, the tool consists out of a relatively large number of components and each of them would have to be separately analyzed and verified with respect to this stress suitability.

6. Scissor linkage

Scissor mechanisms are widely used due to the functionality of longitudinal extension. The mechanism consists out of a number of adjacent identical members that cross each other, as seen in figure 6 below. As with the turnbuckle option, the advantage of this type of linkage would be that it can connect the crane directly with a distant point on the pontoon, which makes it space-efficient. In addition, its structure has a form of protection against buckling in contrary to the turnbuckle.

Fig. 6: Scissor mechanism [8]

The disadvantage of the scissor with respect to the turnbuckle is the lack of a finished off pin connection fitting at the ends. Drilling holes through the endpoints of the structure and placing pins through it could lead to excessive stress concentrations.

III. CONNECTING OPTION SELECTION

The subsequent task after the potential connecting option analysis is to construct a selection procedure for the most feasible connecting option to design the new crane to pontoon connection system with. The first step is to find all relevant aspects regarding this system. Thereafter, the performance indicators within these aspects are defined. These are used to assess the potential options individually and to compare them against each other in various aspects. An overview of the relevant aspects and corresponding indicators is found below.

1. Safety

- Force resistance (kN)
- DoF restriction (yes/no)
- Preload (yes/no)

2. Space-efficiency

- Area of pontoon elements (m^2)
- Maximum available area (m^2)
- Width of crane element (m)

3. Cost-efficiency

- Price (€)

4. Manageability

- Time of application (t)
- Number of personnel (-)

5. Adaptability

- Engineering complexity (-)
- Fabrication complexity (-)

Each connecting option receives one of the four possible ratings seen in table 1 below.

Rating	**Points**
Satisfactory	3
Acceptable	2
Marginal	1
Unacceptable	0

Table 1: Rating/points scheme

After evaluation of the performance indicators for each connecting option, a final rating is obtained as seen in table 2.

Indicators	FPPC	TBWP	SCT	SC	BC	SL
Force r.	2	0	0	0	0	3
DoF rest.	3	3	3	3	0	3
Preload	1	3	1	3	3	1
AOPE	3	3	3	3	3	3
MAA	1	2	1	1	3	2
WOCE	1	3	3	3	1	3
Price	3	3	2	3	3	3
Time of ap.	2	3	2	3	2	2
Nr. of pers.	2	2	2	1	2	2
EC	2	3	3	2	2	1
FC	1	2	1	1	2	1
Total	21	**27**	21	23	21	24

Table 2: Connecting option ratings

With 27 points, the turnbuckle with pin option is rated the highest among all options. Second in place comes the scissor link, with 24 points. The crucial properties that cause the turnbuckle to prevail over the scissor link are its preloading ability, time of application and its adaptation simplicity. The validity of this result is further investigated by checking the effect of altering the point system at the binary indicators. These are the DoF rest. and Preload, which have only two possible rating outcomes. The turnbuckle scores maximally in these two indicators. It is therefore considered using 1 point as the high score and 0 as the low score for these indicators, which lowers the scoring discrepancy between the turnbuckle and the other options. It is however found that the final outcome is not influenced by this.

IV. DESIGN PHASE I

As concluded in the previous chapter, turnbuckles with jaw & jaw end fittings with removable pin (fifth turnbuckle variant in figure 2 in chapter II) are used to proceed the project with. The subsequent step is to design a conceptual model of the system with this connecting option. This is phase I of the design process where a parametric conceptual design is determined.
As discussed in chapter II, the main deficiency of a turnbuckle connection with respect to crane-to-pontoon utilization is that compression forces are not allowed on the turnbuckle. The main scientific objective of this design phase is thus to develop a turnbuckle connection system which eliminates compression forces on the turnbuckle, regardless of input force direction. At the same time, the aim of simplicity is incorporated in the design while maintaining the effectivity of a crane-to-pontoon connection system. Design decisions in order to maximize overall efficiency are furthermore implemented.

The overview of the designed parametric concept can be seen in the figures below.

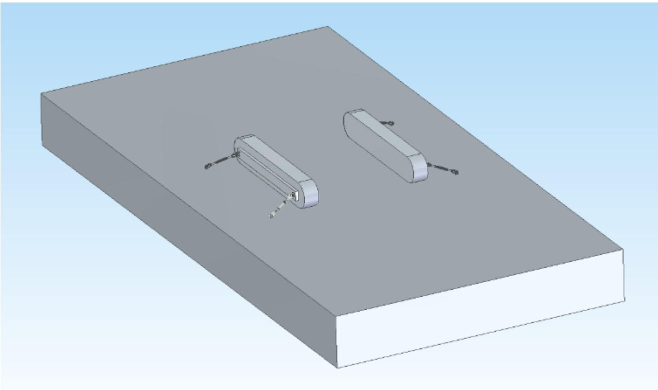


Fig. 7: Overall model (isometric)

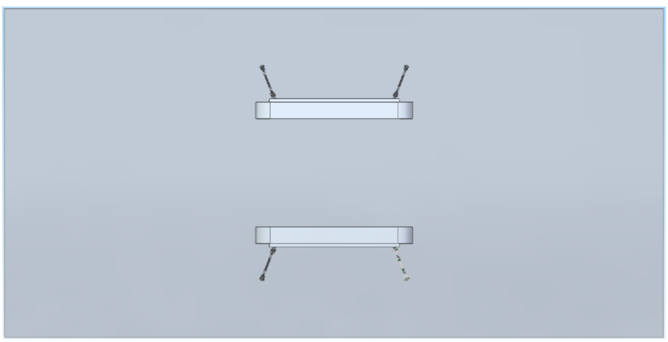


Fig 8: Overall model (top-view)

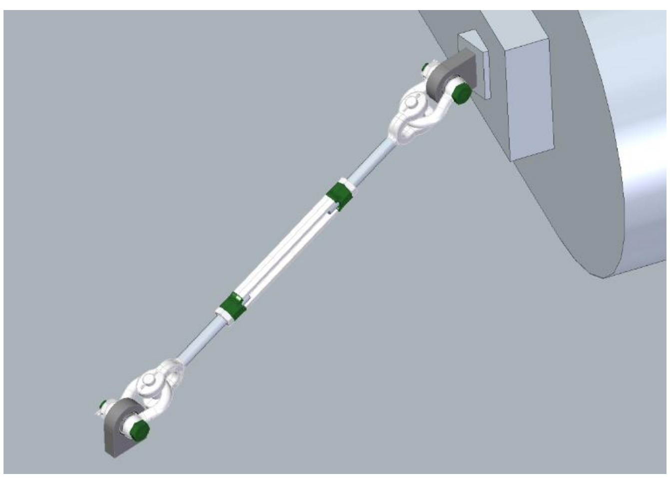


Fig 9: Connection system per crane corner

An overview is provided in below of the components that the design consists of per crane corner, together with their main design parameter(s) which is governing for the dimensions and properties of the component.

Quantity	Component	Main design parameter(s)
1x	Accessory	Inclination angle β (degrees) Placement height h (mm)
2x	Rod end with spherical bearing	Bore diameter d (mm)
2x	Bore adapter	Interference fit tightness ⦰ (-)
2x	Bow shackle	Working load limit t_1 (ton) Bow length ℓ (mm)
1x	Turnbuckle	Working load limit t_2 (ton)

Table 3: Component overview

V. DESIGN PHASE II

With phase I of the design completed, phase II of the design is initiated. This part of the research exhibits that the design is practicable and feasible in terms of construction effort, availability of components and material consumption. This means that the smallest possible components passing the safety evaluation are found here and the design parameters are quantified accordingly. Input sample data for the crane, pontoon and environment are provided by *IHC Engineering* [1].
Table 4 below provides an overview of the determined final values for each design parameter. This combination of values thus satisfies all strength and deformation requirements, while applying a minimization from a material consumption perspective.

	Value
β	20°
h	71mm
d	80mm
⦰	j6
t_1	25t
ℓ	178mm
t_2	16.8t

Table 4: Design parameter values

1. Crane data

The data corresponding to the applied reference crane is found below.

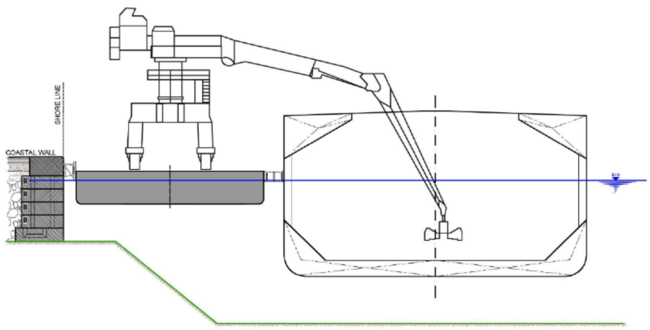


Fig. 10: Front-view of the reference crane on a pontoon

Height (until boom): 11.9 m
Weight: 350t
Capacity: 16t
Arm reach: 35m
Track wheel area: 9.5x1.0m
Distance between track wheels: 7.6m
Maximum permissible dynamical angle: 4°

2. Environmental conditions

IHC has furthermore performed a study on environmental conditions in areas along the Croatian coast which have led to the input values below.

Wind speed = 40m/s (78kn)
Wave length = 6.24m
Wave height = 0.57m

3. Pontoon data

With the following pontoon dimensions determined by IHC, the applied hydrodynamical stability criteria were satisfied while the maximum dynamical angle in operation will be equal to the permissible dynamical angle of 4°.

Length = 40m
Width = 20m
Height = 3.5m

4. Load cases

IHC defines three possible scenarios in which a crane-to-pontoon configuration can be situated. These three scenarios are: onboarding, operation and navigation [1]. The input data is used to derive the force values corresponding to these scenarios. The preloaded condition of the turnbuckles enables in these calculations the simplifications that the crane and pontoon move as one body at all times and that the pulling turnbuckles are loaded simultaneously and proportionally.
The calculated load cases are summarized in table 5 below. The total force values are expressed in the local crane coordinate system (x-y-z). Because no relevant input forces are found for the onboarding scenario, only the operation and navigation load cases are considered. The secondary navigation load case is considered due to higher deformations in x-direction than the first navigation case.

	Operation	**Navigation (1)**	**Navigation (2)**
$\sum \mathbf{F_x}$	0	35kN	35kN
$\sum \mathbf{F_y}$	213kN	24kN	0
$\sum \mathbf{F_z}$	0	0	0

Table 5: Summarization of load cases

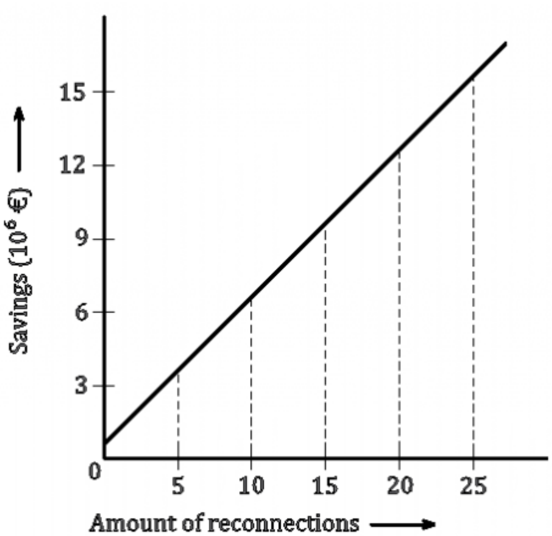
The necessary design parameter values for the connection system as seen in table 4 are determined according to these load cases. It is found with these values that the designed innovations with respect to removability in crane-to-pontoon connections and compression-free turnbuckle utilization are realizable within component manufacturer availability while passing all strength and deformation requirements. With the applied minimization of material consumption, it is furthermore obtained that these parameter values lie within the range of feasible construction time and effort, meaning that the size and weight of each used component can be handled manually by the cooperation of two construction workers.

VI. EFFICIENCY

With both design phases completed and the design parameters quantified, a final investigation is carried out in order to quantify the efficiency of the design. In other words; finding the added value of the removability feature of the new design relatively to the conventional connection systems. This can be quantified by the amount of time saved at connection and disconnection, which is convertible to the amount of money saved.
Based on the various constructional actions required by the conventional connection systems [2] and the simplicity of connecting the turnbuckles, it is assumed that at least a full day is saved per connection with the removable system relatively to the conventional systems,

resulting into an additional day that is gained for crane operation. The revenue generated on a daily basis by operation with the type of heavy-duty crane as applied in the input data with a capacity of 16t is estimated in the range of €600000-€700000 [1]. The total material costs for the removable design are found to be nearly €14000 and are thus considered negligible and immediately recovered at the day of implementation, since €14000 << €600000.

Moreover, an additional day of operation is gained at each subsequent reconnection of the crane to the pontoon with the removable connection system. The total acquired savings accumulate thus over time and depend on the alternation frequency between land and pontoon operation. Graph 1 below shows a schematic representation of the relation between the potential savings versus the projected amount of reconnections when the removable system is used at the reference (heavy-duty) scale-size. It can be seen that with a relative high amount of projected reconnections, the long-term potential savings reach heights of 10+ million €.



Graph 1: Savings versus amount of reconnections

Significant profit is thus anticipated when applying the removable system instead of a conventional system when multiple removals/reconnections are projected. Even if only a one-time connection (0 reconnections) of the crane to the pontoon is desired, the removable design is the more profitable and efficient option over a conventional crane-to-pontoon connection system.

VII. CONCLUSION

1. reflection

It can be concluded that the aim of developing a crane-to-pontoon connection system that enables and facilitates a standard land crane to be utilized as a hybrid operational crane that can operate on a pontoon or a shore on demand, increasing the overall efficiency in port equipment utilization, is accomplished. The developed design provides the desired removability as well as simplicity for the end-user. It is furthermore found at the quantification of design parameters that construction time and effort are within the desired economical range.

At the efficiency quantification, it found that €600000-€700000 is gained per reconnection, so that it potentially accumulates up to 10+ million € if a high amount of removals/reconnections are projected for a particular crane.

The scientific value of the performed research can primarily be defined as the addition of removability in crane-to-pontoon connection system, which creates the potential for significant long-term savings and increased operational efficiency. Furthermore, scientific progress is made through the development of a mechanical system which cancels out compression stresses in a turnbuckle, which is known as the primary weakness of turnbuckles. The designed system allows turnbuckles to be used in applications where compressive motions are present, where the use of turnbuckles would normally be avoided.

2. Limitations

The positive outcome of the research as elaborated in the reflection is subject to potential limitations, which are discussed in this section.

The relevant simplifications taken in design phase II, related to the preloaded condition of the turnbuckles, imply that the crane and pontoon move as one body at all times and that the loads on the turnbuckles are proportionally distributed at all times. The reasonability for this assumption is strengthened by mandating repetitive inspection of the preloaded condition on all four turnbuckles and the implementation of safety factors. A particular unknown margin of error is however not excluded within this preloaded state.

The quantification of parameters is limited to a single set of input data, which are given as a general

representation of heavy-duty crane utilization on pontoons, which provided the proof of general effectivity of the developed system. However, it is unknown what the exact outcome of parameters would be with a less favorable set of input data and how it would impact the effectiveness of the system.
The height of the projected cost-savings in section 6.3 are based on the assumption that the connection time for a conventional connection system takes at least a full day, which resulted in the provision that at least a full day of operation is gained due to the negligible connection time of the removable design. In reality however, the exact duration is unknown due to unavailable reference data on this topic and dependency on multiple company-related factors. The projected savings are therefore to be taken as preliminary. Nevertheless, the positive trend as shown in graph 1 and the anticipation of instant profit are valid in any case.

3. Follow-up research

A follow-up research in correspondence to the described limitations is suggested. This includes a deeper investigation into the preloaded condition and the uncertainty concerning its distribution of loads. Another relevant topic would be the impact on the effectiveness and efficiency of the system when altering the input data. This would provide a clearer spectrum of crane scale-sizes which are suitable for utilization with the removable connection system. Furthermore, an investigation into connection times of conventional connection systems is suggested. By scientifically determining boundaries for these duration values, the total cost-savings of the removable design can be projected with more accuracy.

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10 Appendix B (connecting option rating)

The rating process with detailed argumentations and calculations for each performance indicator of each connecting option can be found below.

10.1 Fish plate pin configuration

10.1.1 Force resistance

The connection elements for this option consist only out of simple pins and fish plates. These pins are in essence steel bars with a closed circular cross-section. They can therefore be ordered at almost any size at steel production companies. *Wuxi Ruizhen Stainless Steel Products Co.,Ltd* [54] for instance offers circular steel bars with customized size up to a diameter of 0.4m.
By taking a closer look at the configuration of these pins as seen in Figure 72 (originally from section 2.2), the occurring stress states on these pins can be discovered if applied with the crane to pontoon connection. The pins will at none of the potential impulses for crane translation or rotation be axially loaded. The only occurring stresses which are considerable on the pins in this case are the lateral compression stress and the shear stress, because of the crane plate and pontoon plate pushing in opposite directions. The bending stresses are also negligible, since the pontoon plate and the crane plate are placed directly next to each other, so that the arm of couple is negligibly small.

Figure 72: Tower crane foundation connection with pins [7]

A calculation is performed to explore the approximate amount of stresses potentially exerted on these pins if applied. The pin size used in the reference from Figure 72, which is a moderate size for utilization, is used to base the calculation on. The pin diameter seen in the picture is estimated at 5 cm, which gives a cross-sectional area of $\pi * 0.025^2 \approx 0.002$ m^2. The length of the pins is estimated at 15 cm. When the simplified reference force of 250kN (see section 3.3.2) is applied, only one of the two pins is loaded. This pin configuration is furthermore assumed to be installed at each of the four

crane corners. The force of 250kN is therefore divided over four total pins. A single pin will thus be assessed on F = 62.5kN.
Considering that the pins are subject to single-plane compression and shear stress in the simplified loading case, the *von Mises* formula for plane stress will be used for the equivalent stress σ' [48]:

$$\sigma' = \left(\sigma_x^2 - \sigma_x\sigma_y + \sigma_y^2 + 3\tau_{xy}^2\right)^{1/2} \qquad (5)$$

Since the applied force is one-dimensional, the formula can be further simplified to:

$$\sigma' = \left(\sigma^2 + 3\tau^2\right)^{1/2} \qquad (6)$$

In order to determine the pure compression stress σ, the *Hertzian* stress formula for cylindrical contact is used. The half-width b of the pressurized surface area is given by the formula below [48]:

$$b = \sqrt{\frac{2F}{\pi l}\frac{(1-v_1^2)/E_1 + (1-v_2^2)/E_2}{1/d_1 + 1/d_2}} \qquad (7)$$

Since the secondary contact surface is a cylindrical hole rather than another cylindrical object, d_2 is a negative value here. In order to find a fitting value for the diameter of this hole, the standard hole-size for clearance fits for pins with a diameter of 50 mm as found in the *ANSI standard b4.2* is used [29]. The found value is 50.16 mm, which is inserted as a negative value for d_2. Assuming that steel is the material for both contact surfaces, the half-width b of the contact surface can be determined with formula (4). The value b=1.9 mm is obtained. Subsequently, the maximum pressure can be found by inserting this value into the formula below [48]:

$$p_{max} = \frac{2F}{\pi b l} \qquad (8)$$

A value for the maximum pressure is found of p_{max} = 139.6 MPa. This value is therefore inserted as the compression stress σ in formula (3). This maximum pressure is located throughout the centerline of the pin [48]. Since the maximum shear stress is also located at the center of the cross-section [55], these two maxima coincide at the center-point of the cross-section. Therefore, the shear stress used in formula (3) to obtain the maximum equivalent compression stress has to be determined with the formula for the maximum shear stress for circular cross-sections, with the shear force denoted as V (equal to F) and the cross-sectional area as A [55];

$$\tau_{max} = \frac{4V}{3A} \qquad (9)$$

The found value for the maximum shear stress is τ_{max} = 42.5 MPa. This value is inserted as the shear stress τ in formula (3). Subsequently, a maximum equivalent stress of σ' = 157.8 MPa is obtained, which is lower than the yield stress σ_y = 235 MPa by a factor of 1.49.
Considering the factor of 1.49, a force higher than 250 kN + 5% is easily resisted by the analyzed pin size, but not to the extent that it can be considered as a comfortably safe resistance. An acceptable rating is therefore assigned to this indicator.

10.1.2 DoF restriction

The configuration as displayed in Figure 4 restricts translations and rotations with respect to each axis, so that all 6 degrees of freedom are restricted. This indicator yields therefore a satisfactory rating.

10.1.3 Preload

Once the pins have been inserted into the plate holes with the clearance fit, there is no possibility to apply a maintainable preload on the system in order to remove the clearance. A marginal rating is therefore assigned.

10.1.4 Area of pontoon elements

Figure 73 below shows a schematic view of the pontoon element in this connecting option, which is a fish plate with a 90° angle.

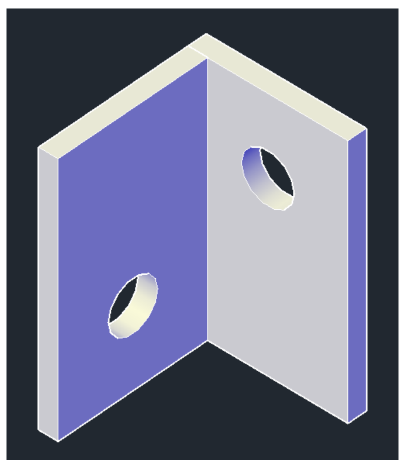


Figure 73: Pontoon element of fish plate pin configuration

Figure 74 shows the schematic top-view of the potential pontoon element.

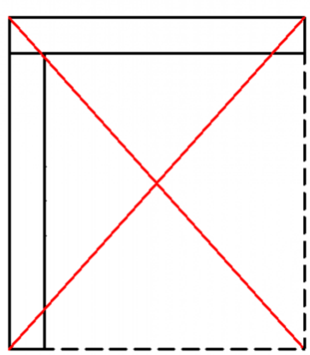


Figure 74: Top-view of pontoon element area

The pontoon area denoted with the red cross will be considered unusable for other purposes after placement of this fish plate. The area of pontoon elements will therefore be counted as a square, considering that the two perpendicular plate sections are of identical size.
Figure 75 below shows a schematic body diagram of the fishplate in the simplified loading situation from a front-view perspective. The case where the upper pin, rather than the lower pin, is pressurized is the more critical situation due to the higher arm of couple and is therefore the analyzed case here. The plate is assumed to be welded on the pontoon deck, which delivers a force (denoted with F_{weld}) which counteracts the force exerted by the pin on the plate (denoted with F_{pin}). These forces cause an axial stress due to bending stress as well as a shear stress on the plate. There is no torsion included in the shear stress, because the applied pin force goes through the shear center of the cross-section [56].

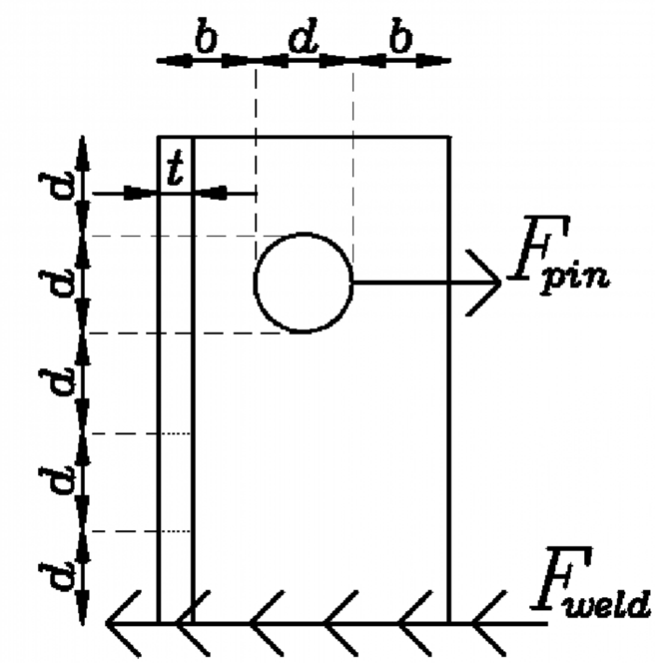


Figure 75: Schematic body diagram of fish plate

In order to limit the induced bending stress, the height of the plate is limited to 5 times the pin diameter *d*, such that each pin hole is surrounded at least with one diameter of plate material and so that the distance between both pins is at least one pin diameter. Furthermore, a standardly applied thickness for steel plates of 10mm is used for the plate thickness *t*. The required distance *b* is therefore the only variable that needs to be found in order to determine the total length of the plate in order to estimate the consumed pontoon area as indicated in Figure 74.
The relevant formula for determining the equivalent axial stress is the same as formula (6):

$$\sigma' = (\sigma^2 + 3\tau^2)^{1/2} \qquad (6)$$

The pure axial stress σ originates here from bending alone and its magnitude along the height (5 times d) of the plate depends on the moment of inertia of the cross-section and the magnitude of the moment which both vary along this height (the moment of inertia varies due to the holes), see formula below.

$$\sigma = \frac{My}{I} \tag{10}$$

After inspection of these two variables (M/I), the bending stress appears to be the highest at the point of the highest moment, at the bottom of the plate where the welds are applied. The cross-section here is a standard L-shape, as seen in Figure 76 below.

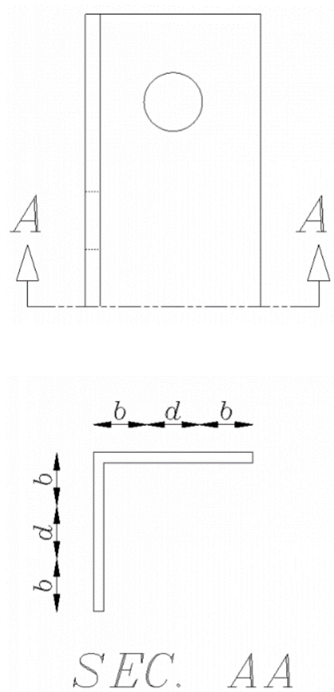


Figure 76: cross-section at bottom of plate

The magnitude of the pure axial stress σ depends furthermore on the distance y of the analyzed fiber with respect to the neutral axis of the cross-section, which obviously maximizes at the most distant fiber from the neutral axis. The shear stress τ conversely, when analyzed along the height of the plate, maximizes at the points where the cross-section is the smallest, thus at the points where the holes are located. Furthermore, given the L-shape section, the magnitude of the shear stress varies along the cross-section differently than the bending stress σ, see shear formula below for thin-walled open sections, with the first moment of area denoted with Q and the wall thickness denoted with t [56].

$$\tau = \frac{VQ}{It} \tag{11}$$

The maximum shear stress along this type of cross-section maximizes at the neutral axis and decreases parabolically to a slightly lower value at the fiber most distant from the neutral axis as seen in the example (with arbitrary stress-values) in Figure 77 below.

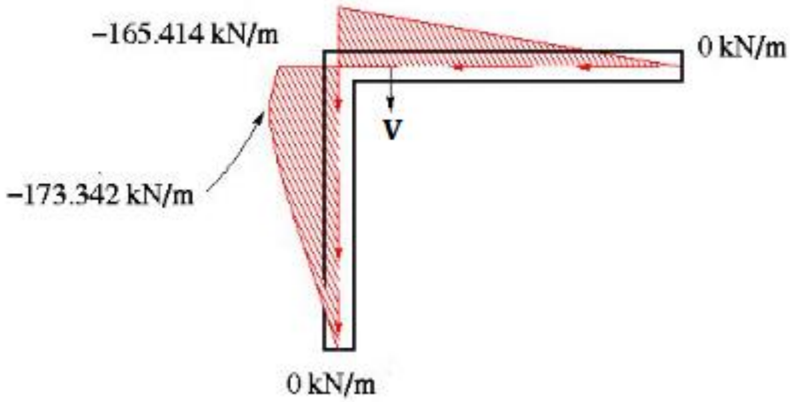


Figure 77: shear stress distribution in L-shaped section [57]

The maxima of the bending stress and the shear stress therefore do not coincide along the height of the plate nor along the cross-section. In order to obtain the exact value for the highest possible equivalent axial stress σ', a mathematical analysis is required. For the purpose of estimating the area of the pontoon elements however, additional simplifications will be applied. Assumed will be that the highest equivalent stress will be located at the bottom of the plate, where the bending stress is the highest. The cross-section shown in Figure 76 can therefore be used. The maximum equivalent stress along this cross-section is assumed to be at the top fiber of the section, because the bending stress is there maximized and the shear stress is not much lower than its maximum. The bending stress and shear stress are thus determined with the formulas (8) and (9) at the top fiber of the section and subsequently inserted into formula (3). The variable b is set equal to d, which results in an equivalent stress value of $\sigma' \approx 1720$ MPa, which does not suffice. Hence, b is set equal to $3d$ for the second iteration, resulting in $\sigma' \approx 43.5$ MPa. This value for σ' is considered conservative for design purposes. A first check however for the pontoon elements area can therefore be performed with this value.

With $b = 3d$, all parameters are known for determining the area of the pontoon elements, which is equal to four times $7d * 7d$ (see Figure 74 and Figure 76). With d = 5cm as determined in section 10.1.1, the total consumed area is equal to $0.49m^2$. Referring back to the set criteria for this performance indicator in section 3.3.5, a satisfactory rating is already achieved with this value, so there is no need for further iterations with a smaller value of b.

10.1.5 Maximum available area

With the area of the pontoon elements estimated, the maximum available rectangularly shaped area on the pontoon after installation of the elements can be estimated as well. It is known that these fishplate plate elements must be placed directly (or in the near vicinity) next to the crane so that the pin connection can be established. The plates are therefore modelled here as $7d$ * $7d$ squares at each corner of the reference crane are. This situation from the top-view of the pontoon surface can be seen in Figure 78 below.

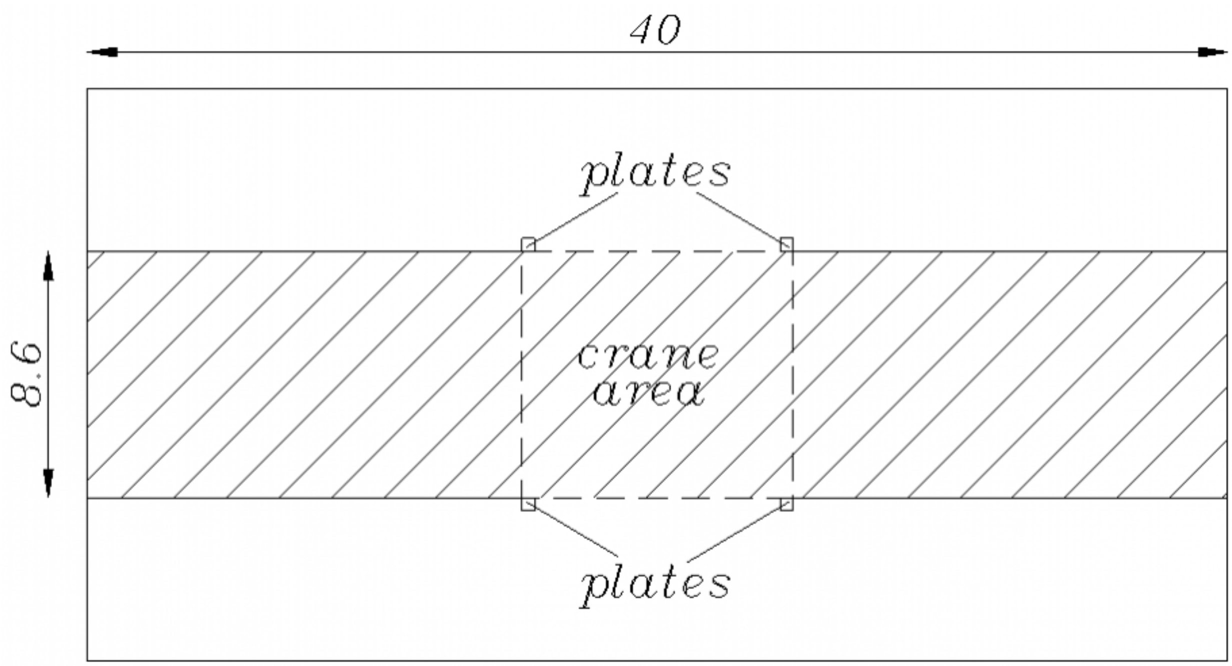


Figure 78: Maximum available rectangular area after fishplate installation (top-view)

The area where the crane would be placed is indicated with the dashed rectangle in the amidships position of the pontoon. The fish plates would therefore be placed adjacently to this rectangle and are indicated with the squares at the four corners of the area. Even though these squares look relatively small, they obstruct placement of other possible equipment on the pontoon when the crane is absent. The rectangle indicated with the hatched area gives the biggest possible rectangular area for utilization after installation of the permanent fish plate pontoon elements. Any equipment bigger than this size will therefore not be deployable on this pontoon anymore. The size of the rectangle is 40*8.6m = 344m^2. According to the rating criteria for this indicator in section 3.3.6, this value yields a marginal rating.

10.1.6 Width of crane elements

As seen in the reference in Figure 72, the crane is equipped with a flange similar to the fish plate to establish the pin connection. The forces exerted by the pin on these elements are equal to those on the pontoon plate, but the support of the flange is welded perpendicularly with respect to the pontoon surface, which makes the stress states differ. Nevertheless, the width estimated for the fish plate in section 10.1.4 is taken for the crane flange as well, under the reservation that a bigger plate thickness might be applied if necessary. A bigger plate thickness does however not affect the width of the crane elements.

The pin is assumed to be placed for optimal balance with its midpoint at the juncture of the fishplate with the crane flange, so that an equal amount of pin length sticks out on each side of the juncture. The pin length used to determine the force resistance needs to be increased due to the plate dimensions used in section 10.1.4, so that a proper connection between the two pins can be established with the smaller pin. The new pin length is therefore 40cm, which makes the pin edges precisely overlap (this increase would not affect the rating for the force resistance). Considering that the crane flange and the fishplate have a combined thickness of 2cm, 40cm-2cm=38cm is divided by 2 in order to find the amount of pin length sticking out on each side, which is equal to 19cm. The crane flanges can therefore be mirrored with respect to their orientation in the reference in Figure 72. This would leave a distance of one plate-side (7*d = 35cm) in between the crane and the juncture. Even though the force magnitudes on these crane flanges are equal to those on the pontoon plates, the loading situation differs due to the change in orientation. However, the only dimensions that possibly need to be adjusted due to this are the thickness of the flange plate perpendicular to the crane and its height (parallel to the crane height). The width of the element with respect to the crane can remain constant.

Another issue to address is that the angular nature of these elements cause obstructions during boarding of the crane to the crane area on the pontoon. Two situations are schematically drawn in Figure 79 below.

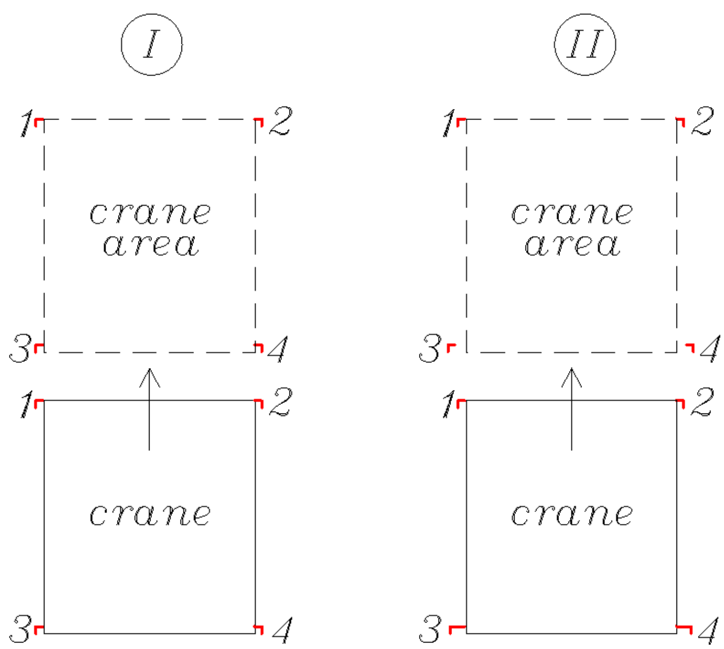


Figure 79: pontoon plate and crane flange configurations (top-view)

The pontoon plate and crane flanges are schematically indicated with the red color. The numbering indicates which crane flange is to be connected with which pontoon plate for both situations. For situation I, it can be observed that when the crane is driven towards its supposed area on the pontoon, the crane flanges 1 and 2 will be blocked by the pontoon plates 3 and 4, preventing the crane to reach its destination on the pontoon. This issue can be solved only by adding space in between pontoon plates 3 and 4, as drawn in situation II. That way, crane flanges 1 and 2 can reach their respective pontoon plates. However, for the crane flanges 3 and 4 to align with their respective pontoon plates in this situation, a longer plate-piece between the flange and the crane is required, which is drawn for flanges 3 and 4 in situation II. These two plate-pieces thus need to be extended by at least an additional pontoon plate width of $7d$ = 35cm. The final increase of the crane width is thus approximately equal to 0.35 + 0.35 = 0.70m on both sides. Accumulating the increase from both sides gives a total increase of 1.40m, which corresponds to a marginal rating according to the boundary value scheme for this indicator.

10.1.7 Price

The components of this connection system can be summarized as 4 angular steel plates for the pontoon, 4 angular steel plates for the crane, 8 big pins (2 for each corner) and 4 smaller pins. The price for 8 big pins with the diameter as estimated in section 10.1.1 (d = 5 cm and *length* = 60 cm) is analyzed firstly. Since this diameter is bigger than the usually sold cotter pins on the market, a steel manufacturer is consulted. *ParkerSteel* [58] reports a total price of £522.16 for 8 steel cylinders of that size. Converted to dollars gives a price of approximately $637.
4 angular steel plates with the dimensions as estimated in section 10.1.4 ($7d$ x $7d$ with t = 1cm and *length* = $5d$) must be composed of 8 flat plates ($7d$ x $5d$ with t = 1cm). They are offered by the same manufacturer for £261.82 [53], which converts to approximately $317. For the crane flanges, the calculation is performed with bigger angular plates, as seen in section 10.1.6 These might need to be manually composed with steel plates with a bigger thickness. In order to apply an upper limit, the price of a 3 x 1.5 m steel plate with a thickness of 3 cm is taken, which costs £2011.06 = $2477. This plate is considered sufficient to assemble four crane flanges of sufficient size.
These products cost thus combined around $3000. Since the price for the smaller pins is lower than the bigger pins, it can already be concluded that the total price is lower than $18,300. A satisfactory is thus assigned.
A check is performed to confirm that the plate thickness of 3 cm is able to withstand the forces. Because the reference force acts in the direction perpendicular to the track wheel, the stress can be taken as a normal stress acting on the welded surface. A value of $\sigma' \approx 8.3$ MPa is obtained. The satisfactory rating for this indicator remains thus valid.

10.1.8 Time of application

In principle, a minimum amount of time should be required for the placement of the big pins in the holes with the clearance fits, if assumed that the installation of the plates as well as the placement of the pontoon is done accurately. However, it can be expected at every boarding process that there will be significant time lost in the exact alignment of the crane with its flange holes to the holes of the fishplate on the pontoon, because it has to be done with high precision in order to be able to implement the pins. The holes in these pins thereafter need to be aligned so that the smaller pin can

be pushed through them (which can also be a clearance fit). The final step is the placement of the cotter through the smaller pin, or the cotter can be preset in the smaller pin at all times.
The time lost during aligning is significant, but is partially compensated by the quick pin placements. Furthermore, there will be no aligning at the removal of the connection, which reduces the accumulated time of applying and removing the connection. An acceptable rating will thus be assigned to this indicator.

10.1.9 Number of personnel

The weight of a single pin with the dimensions determined in the sections above is around 6 kg, which can easily be handled by a single person. The application of this connecting option might however not necessarily be performable by a single person due to the aligning process as described in the section above. One person might be needed for positioning the crane simultaneously while another person is continuously checking whether the plates and flanges are correctly aligned. Two persons would thus be working together in this case. An acceptable rating will therefore be assigned according to the boundary value scheme.

10.1.10 Engineering complexity

There are no unacceptable ratings assigned to the previously rated indicators of this connecting option. There are however three marginally rated indicators which have room for improvement. These are the *preload*, *maximum available area* and *width of crane elements*. The engineering complexity for developing the adaptations to improve these three indicators will be rated here.
In order to improve the preload indicator, a method needs to be developed for removing the clearance between the pins and the plate holes after placement. This can be achieved by pressing the pins firmly against the sides of the holes. A tool like a high-strength screw clamp as depicted in Figure 80 or similar could be used for this purpose.

Figure 80: screw clamp example [59]

This would require two additional flanges for clamping on the fishplates per pin as schematically shown in Figure 81 and Figure 82 below. The red crosses in Figure 81 indicate the clamping spots. This type of configuration however would press the pin against one side of the fishplate hole, such that the clearance at the other side of the hole doubles. This means that the pins function is eliminated if a crane force is applied in the same direction as the clamping force. So, instead of four engaged pins at a time (one for each corner), only two pins are engaged at a time from a single side. This means that pins of bigger size would have to be used in this scenario, which is feasible but undesirable at the same time.

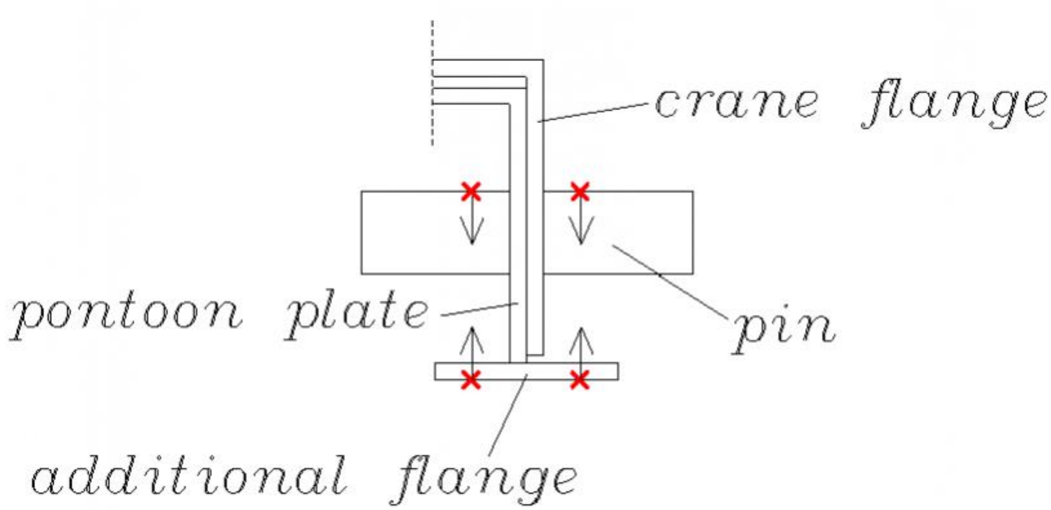


Figure 81: Pontoon plate with additional flange (top-view)

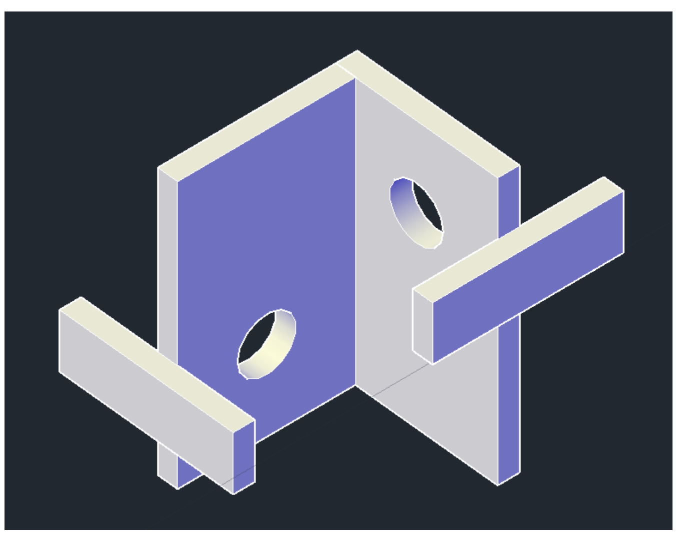


Figure 82: Pontoon plate with additional flange (isometric)

An adaptation that would fully solve the issue with preload without negative impact would require considerably more engineering complexity, like plate holes with a variable diameter. Improving the maximum available area on the other hand is also a challenging task. It requires the pontoon plates to be installed more distantly from the crane, which seems unfeasible without negatively impacting

the other performance indicators. The crane flanges would need to be widened even more to establish this, which is undesirable. A solution would be to implement crane flanges that have the ability to extend and retract with respect to the crane. This way the increase of the crane width would only be present when the crane operates on the pontoon and could be limited while the crane is on land. It would bring considerable engineering complexity for correctly designing every sub-component of the retractable mechanism. However, it is a type of mechanism that has been used often times in practice before and it could be a simple, manually movable mechanism.
The width of crane elements can be simply solved with a more detailed structural analysis which result into more beneficial steel dimensions.
Altogether, an acceptable rating is most suitable for this indicator.

10.1.11 Fabrication complexity

The fabrication complexity of the possible adaptations as discussed in the section above are rated in this section. The adaptations related to the preload do not bring along any particular difficulties in the workshop. A potential mechanism for increasing the pontoon plate distance on the other hand would require much more precision, time and materials to smoothly assemble the sub-components together, which fall short of the aims set for simplicity for this project. A marginal rating is therefore considered most suitable for this indicator.

10.2 TURNBUCKLE WITH PIN

10.2.1 Force resistance

Turnbuckles, as elaborated in section 2.3, are not designed to take up compression stresses. Since forces from each direction are considered in the structural analysis of the problem, compression forces will be present, in small amounts at least, if a standard jaw to lashing eye connection is applied as seen in Figure 83.

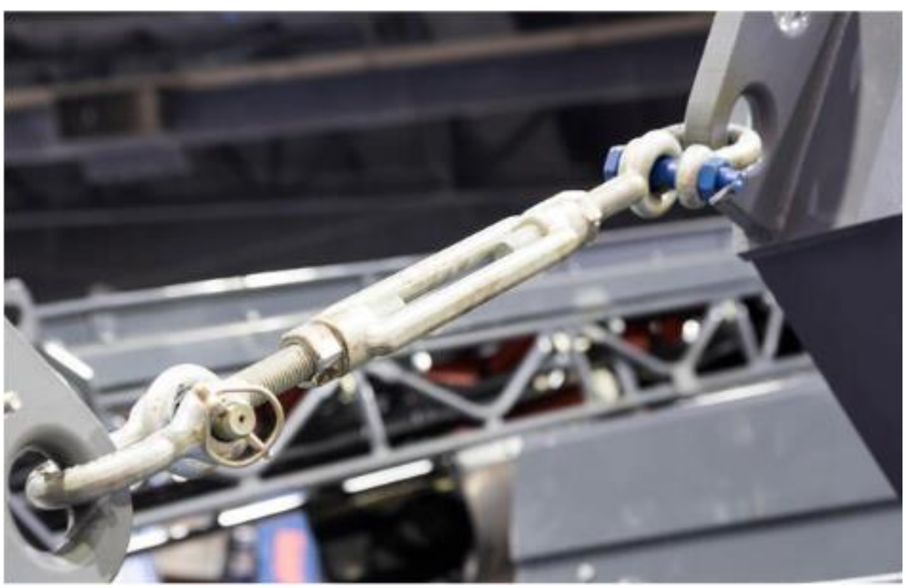


Figure 83: Turnbuckle jaw to lashing eye connection

If a force is applied in the direction from the crane towards the turnbuckle, the lashing eye of the crane moves inwards/towards the turnbuckle as much as the deformation/elongation of the turnbuckle under tension on the opposite side. This movement causes a particular amount of distortion at the connection point which induces a risk of compression forces occurring due to shear, bending or even axial forces on the turnbuckle on the compressed side. Figure 84 below schematically illustrates this potential turnbuckle configuration with 8 total turnbuckles in order to fully secure the crane in each direction. If a force F is applied in a particular direction, one set of turnbuckles will be under tension while the opposite set will be subject to the above described danger of compression. Also the turnbuckles at the sides will be subject to the risks of compression due to possible bending.

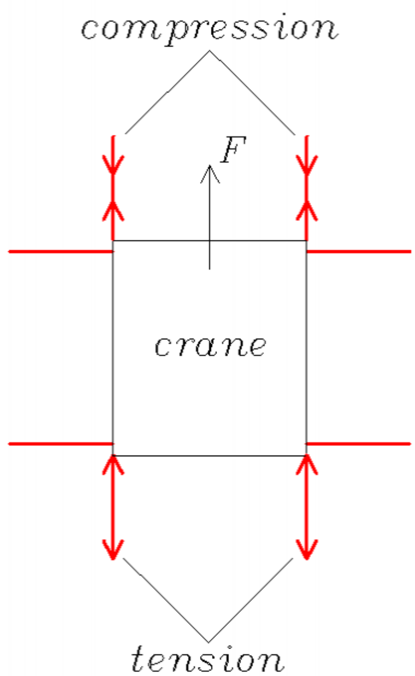


Figure 84: Schematic turnbuckle configuration (top-view)

It is unclear in advance to which extent this compression can be problematic, which means that an unacceptable rating must be given to this indicator. Nevertheless, a force resistance calculation is performed to find out which turnbuckle dimensions are suitable for assessing the remainder of the indicators with.
Since the simplified reference force is one-dimensional, turnbuckles may theoretically be placed at the side tension only. One turnbuckle is assumed per crane corner, so that two turnbuckles must take up the entire force in tension. The situation is schematically illustrated in Figure 85, with the turnbuckles denoted with TB 1 and TB 2. Since the reference force of 250 kN is distributed over two turnbuckles, 125 kN should be resisted per turnbuckle.

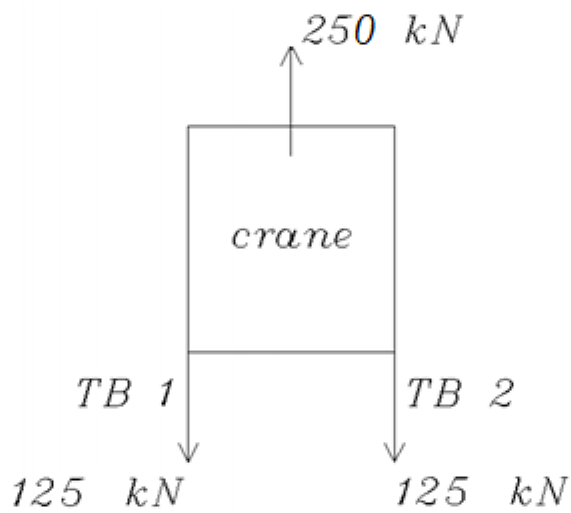


Figure 85: Schematic turnbuckle force distribution (top-view)

The question is to which extent a turnbuckle of feasible size can resist a force of 125 kN. In order to find a moderate size for manual utilization, a reference from a similar application type will be used. Figure 86 below shows the sea fastening of a 150-ton reel on board of a vessel.

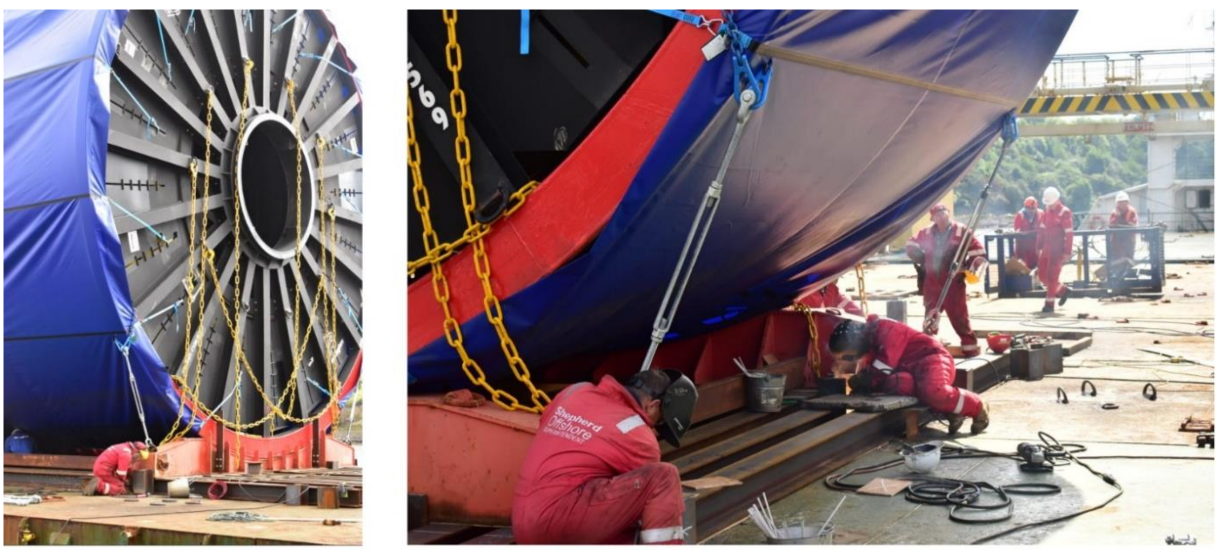


Figure 86: sea fastening of a 150-ton reel [60]

It can be observed that turnbuckles are used for tightening the fastening system. The total length of these turnbuckles in their unscrewed position can be estimated from the picture at approximately 170 cm. The rating of the turnbuckle option in this paragraph will therefore be based on that reference size.

Distributor *Greenpin* [44] is consulted for technical specifications of available jaw & jaw turnbuckles with simple pins at both ends. A turnbuckle with a length of 167.1 cm has a rated capacity of 164.8 kN, which is a suitable capacity for the potential design, considering the reference force of 125 kN per turnbuckle. These dimensions are therefore used for the rating of the remainder of the indicators.

10.2.2 DoF restriction

If turnbuckles are placed around the crane as illustrated in Figure 84, the crane is secured from translation in any direction, because there is a set of turnbuckles in tension for every direction on the plane of the pontoon surface. Also, the hypothetical cases of upward translation and/or rotation about any axis are associated with tension at a particular set of turnbuckles which prevents these motions. A satisfactory rating can therefore be assigned to this indicator.

10.2.3 Preload

Turnbuckles are inherently equipped with the possibility of preloading the connection, since it is their original main function. A satisfactory rating is given therefore.

10.2.4 Area of pontoon elements

The turnbuckles are to be connected to a particular form of lashing eyes on the pontoon. These lashings are to be welded to the pontoon deck. The amount of surface area consumed by these lashings, accumulated over all turnbuckles, is assessed here. The simplified reference force gave in the previous sections that 125 kN of tension has to be resisted by a single turnbuckle. This means that the corresponding lashing eye needs to be able to resist the same amount of force. A schematic body diagram of the pontoon lashing element is drawn in Figure 87.

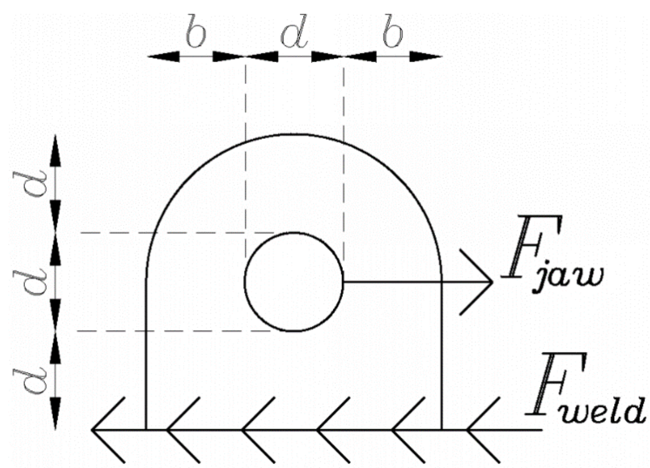


Figure 87: Schematic body diagram of lashing eye

The same assumptions and principles area applied as for the pin calculations in section 10.1.1. The *ANSI standard b4.2* is used to determine the hole size *d* for a clearance fit. So, instead of using a large eye opening as seen in the example in Figure 83, the size of the opening is limited as if it is a pin hole. This is mechanically advantageous, because a smaller opening gives a bigger half-width of the Hertzian stress surface area as seen in formula (7) and hence a smaller maximum pressure. The force exerted by the pin of the jaw of the turnbuckle is indicated with F_{jaw}. The chosen direction of this force is the worst-case scenario of application. A common pin diameter of a turnbuckle with the reference size and capacity as determined in 10.2.1 is 51 mm [44]. The corresponding hole size for a clearance fit according to the *ANSI* table is 51.16 mm [29]. Subsequently, as done with the fishplate in the previous section, the hole size d can be rounded to simply the pin diameter for estimation purposes, so that d = 51 mm. Also, the height of the element is limited to 3 times d, similarly to the fishplate, to limit the bending moment. Furthermore, the cross-section is consistently doubly symmetric with the forces going through the centroid, so that the risk of torsion is absent [56]. The standard plate thickness t = 10mm is applied, so that the only remaining variable for determining the consumed pontoon area is the distance b, with the total area being equal to (b + d + b)*t.

With the fishplate, the assumption is made that the combination of shear and axial stresses is the highest at the bottom of the element, where the bending moment is the highest. With this eye lashing however, the arm of couple is considerably lower (1.5d), which means that the shear stress might be the more dominant stress in this case. The evaluated cross-section will firstly be at the bottom, thus a simple rectangle with dimensions (b + d + b)*t. The same formula for determining the equivalent stress is used;

$$\sigma' = (\sigma^2 + 3\tau^2)^{1/2} \qquad (6)$$

The axial stress is determined again with the formula for bending stress;

$$\sigma = \frac{My}{I} \qquad (10)$$

This stress starts with zero at the center of the rectangular cross-section and maximizes linearly at the top fiber of the section. The shear stress has different distribution, depicted in Figure 88. It maximizes at the center and decays parabolically to zero at the top fiber. The maxima of the axial and shear stresses therefore do not coincide and cannot be combined together in formula (6).

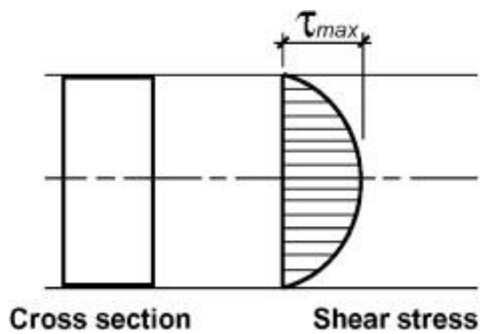


Figure 88: Shear flow rectangle [61]

The implemented simplification here is to base the calculation either on the centroid, where the shear stress τ is maximized, or on the top fiber, where the axial stress σ is maximized, depending on which of the two stresses is more critical. The maximum shear stress τ_{max} for a rectangular cross-section is determined with formula (12) below [56];

$$\tau_{max} = \frac{3V}{2A} \qquad (12)$$

The first iteration is performed with $b = d$ (and $F_{jaw} = V$ = 125 kN), which gives a maximum axial stress of $\sigma_{max} \approx$ 244 MPa and a maximum shear stress of $\tau_{max} \approx$ 123 MPa. Since this stress-state does

not suffice, a second iteration is performed with b = 1.5d, which gives a maximum axial stress of σ_{max} ≈ 104.5 MPa and a maximum shear stress of τ_{max} ≈ 91.9 MPa. The situation of σ = 0 MPa and τ = 91.9 MPa gives a higher equivalent stress σ' in formula (6) than the situation of σ = 104.5 MPa and τ = 0 MPa (σ' = 159.2 MPa versus σ' = 104.5 MPa), which means that the location of the centroid is considered more critical than the top fiber.
For the cross-section at the location of the eye opening, there is no bending stress, but the shear stress is significantly higher. The distribution at this type of cross-section is illustrated in Figure 89 below.

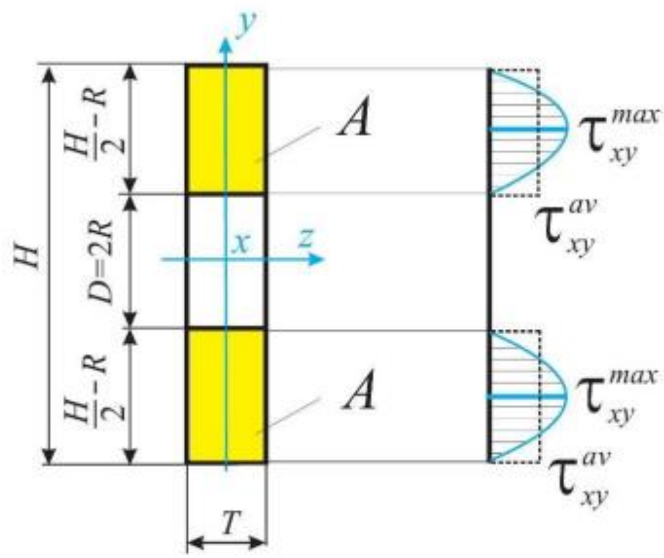


Figure 89: shear flow rectangular cross-section with hole [62]

The maximum shear stress τ_{max} can be found with formula (13) below [62], with A equal to the area of one of two rectangles as indicated in Figure 89;

$$\tau_{max} = \frac{3V}{4A} \qquad (13)$$

The maximum stress state and this juncture of the lashing is thus σ = 0 MPa and τ = 122.6 MPa with formula (13). This results in an equivalent stress of σ' = 212.3 MPa, which is higher than the stress state estimated at the bottom of the element. The most critical location is thus assumed to be at the eye opening. The following stress calculation iterations are thus based on that location.
Because 212.3 MPa is slightly lower than the yield stress of 235 MPa, this value of b = 1.5d is considered an adequate design value. The consumed pontoon area of one element is therefore equal to 4d*t = 0.00204 m^2. It is furthermore assumed that a total of 8 turnbuckles would be necessary as elaborated in section 10.2.1. The total estimated area is thus equal to 8*0.00204 = 0.01632 m^2, which yields a satisfactory rating.

10.2.5 Maximum available area

Besides the area of the pontoon elements, the distance between the pontoon elements and the crane when connected with turnbuckles is necessary for determining the maximum available area

indicator. The length of the reference turnbuckle can as mentioned be increased up to 167.1 cm if the thread is screwed outward, according to the datasheet provided by Greenpin [44]. 150 cm however will be calculated with, in order to conserve the buffer of variable length.
The available area can in theory be maximized if the turnbuckles are placed diagonally from crane. Fewer turnbuckles are necessary in that case, because the tension force on a single turnbuckle can then be activated in two directions instead of one. Furthermore, the turnbuckles are ideally inclined more towards the side where the loads are the highest, in order to limit the maximum crane displacement due to turnbuckle strain. A 60°/30° angle proportion is considered for this purpose. The side with the highest load is assumed to be in the direction of the reference force (see section 3.3.2 and Figure 26). Figure 90 illustrates this concept, with the turnbuckles indicated with the red crosses and the maximum available rectangular area hatched.

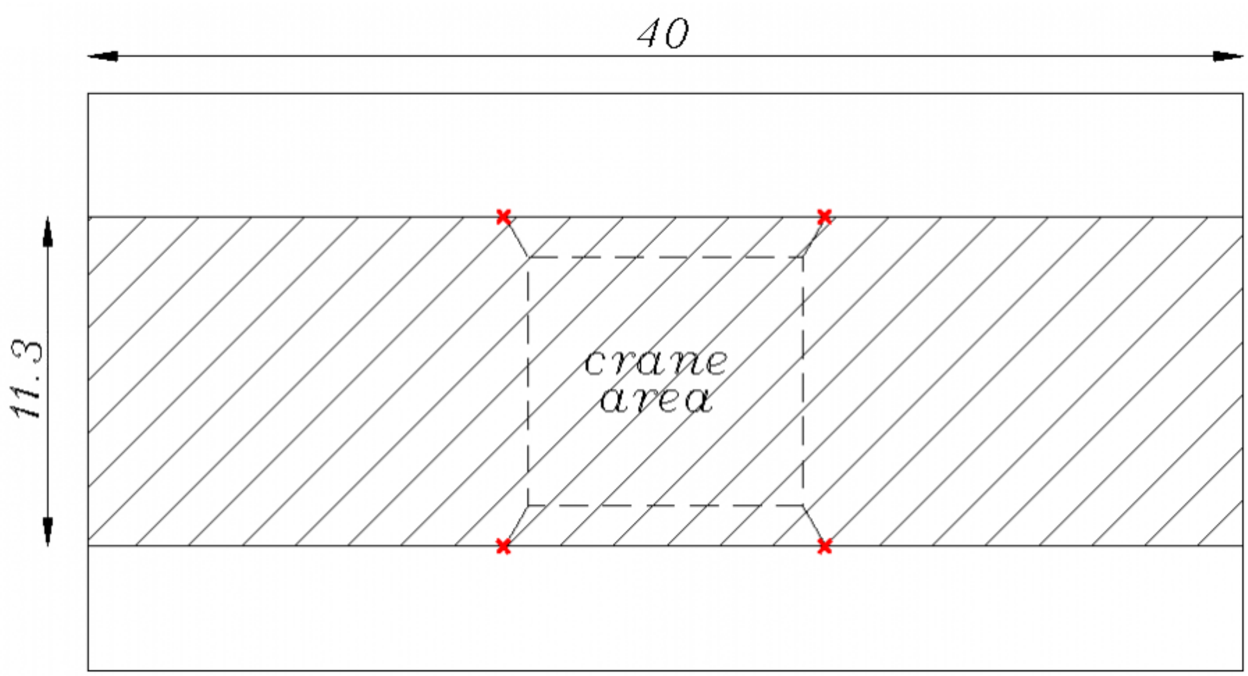


Figure 90: Maximum available area with turnbuckles

The maximum available area with turnbuckles can thus be estimated at 40*11.3 = 452 m^2, which means that an acceptable rating will be assigned to this indicator.

10.2.6 Width of crane elements

Because the crane elements are loaded with an equal force as the pontoon elements and the pontoon elements have already been assessed in the worst-case scenario of application, the dimensions for the pontoon elements will suffice for the crane elements as well. The difference is that this lashing eye is welded with its bottom to the side of the crane wheel track. The height of the lashing as determined in 10.2.4 is therefore the width of the lashing when applied as the crane element, which is 3*d = 0.153 m. Because it is applied at both sides, this width is multiplied by 2 in order to obtain the total increase of the crane width, which is 0.306 m. A satisfactory rating is therefore assigned to this indicator.

10.2.7 Price

Turnbuckles with the reference capacity and length used in the previous sections are sold for $1753.86 apiece [63]. Because a configuration with 4 of these turnbuckles is assumed to be feasible, the total price for the turnbuckles is 4*1753.86 = $7015.44.
Adding the total price for buying the steel for the lashing eyes and to customize them to the desired dimensions will make the total price not exceed the satisfactory threshold of $18,300. A satisfactory rating is thus assigned.

10.2.8 Time of application

Because of the variable length of the turnbuckles, there needs no time to be wasted for aligning the crane at an exact spot. After the crane is placed at the spot on the pontoon that is considered approximately the amidships position, the turnbuckles will adapt to the necessary length to reach the connection from the pontoon elements to the crane elements. The connection is expected to be established quickly due to the pin fittings at the ends. Because there are no particular time-consuming actions to be associated with connecting and disconnecting the turnbuckles, a satisfactory rating can be assigned.

10.2.9 Number of personnel

The connection can in principle comfortably be completed by a single person. However, the weight of a reference turnbuckle is 49 kg. It is therefore likely that an assistant is necessary when carrying a turnbuckle to its connection point. An acceptable rating is thus most suitable to this indicator.

10.2.10 Engineering complexity

There is only one unacceptable rating given to this connecting option, which is the force resistance. There are furthermore no marginal ratings. For the engineering complexity will therefore be looked at possibilities to solve the problem regarding the compression forces.
There are two options to solve the problem. One option is to design a reinforcement on the turnbuckle, which makes it resistant to the compression forces. Because this problem is related to buckling, adaptations need to be looked at which decrease the body's slenderness ratio by increasing the radius of gyration or decreasing the effective buckling length. Possibilities for increasing the radius of gyration are for instance adding a welded T-stiffener at each side of the turnbuckle frame. A solution thereafter needs to be found to conserve the possibility of manual gripping of the turnbuckle. This could be in the form of a gap or interval at the stiffener. Stress concentrations at this gap might be the cause for some engineering difficulties, but are feasible to calculate and solve. Alternatively, the effective buckling length can for instance be decreased by welding a sheet of steel at the center of the frame, in between the two screws. These two possible solutions are schematically sketched in Figure 91.

Figure 91: potential reinforcements for turnbuckles against buckling

The second option to solve the problem is to design the connection elements in a way that the turnbuckles are compression-free at all times. This requires a precise calculation of the shape and size of the eye openings, deformations and clearances. The design must ensure that the compressed side is free from any shear or axial contact with the crane elements at those occasions. Altogether, the implementation of these discussed options are realistic engineering projects that do not require the use of unknown or peculiar science and offer a good solution to the problem. A satisfactory rating is therefore given to this indicator.

10.2.11 Fabrication complexity

The mentioned potential adaptations to the turnbuckle are workshop jobs that are commonly carried out, but not without difficulties, depending on the designed gap-shape. Implementing the centerpiece would be an easier fabrication, but could cause difficulties when welding at the inside of the frame. The second adaptation option with respect to the elimination of compression would require also effort, but can be accomplished with the use of simple steel pieces, which would require basic modifications and customization to fabricate them. An acceptable rating is considered most suitable for this indicator.

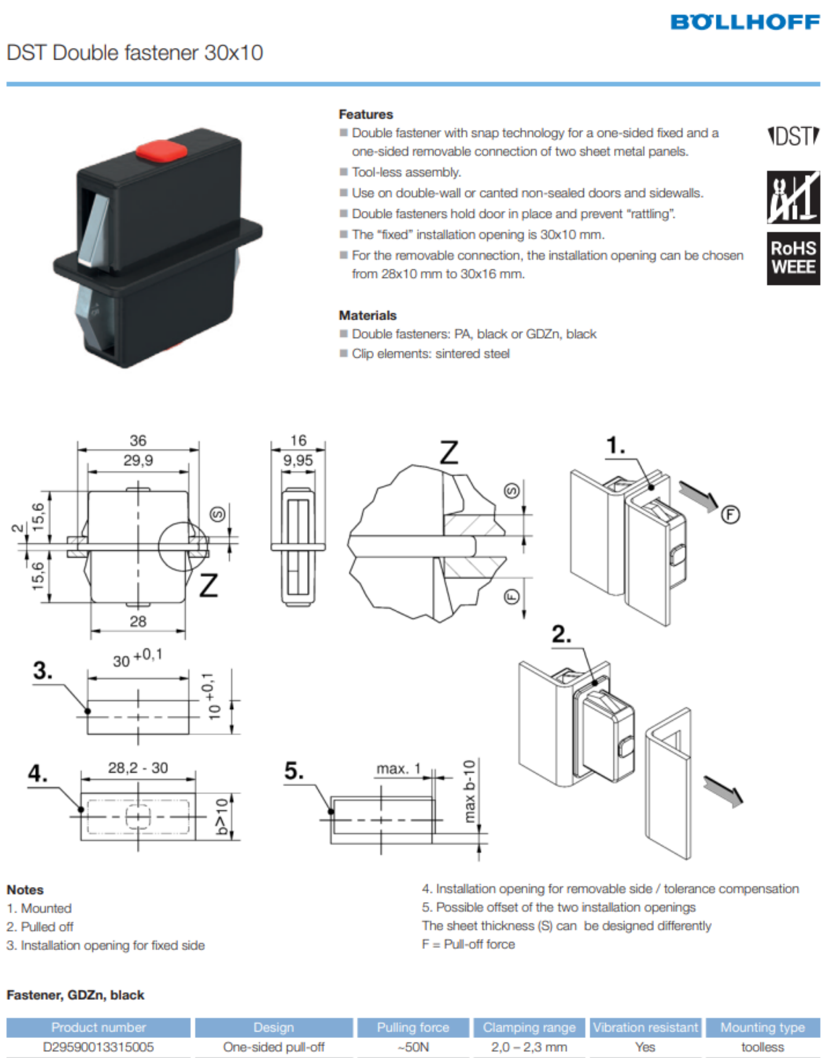
10.3 SPRING CLIP TECHNOLOGY

10.3.1 Force resistance

Spring clips are usually designed for the small-scale domain. The datasheet of the example shown in Figure 11 in section 2.4 (or Figure 92 below) indicates that an approximate pulling force of 50N is expected with these devices. It is thus not intended for large-scale applications, which means that an unacceptable rating must be assigned to this indicator.

10.3.2 DoF restriction

Figure 11 is pasted once more below as Figure 92 for better inspection. By observing the sketches of the system, it can be concluded that this type of clip joins the two flanges with respect to every degree of freedom, so that a satisfactory rating can be assigned here.



Product number	Design	Pulling force	Clamping range	Vibration resistant	Mounting type
D29590013315005	One-sided pull-off	~50N	2,0 – 2,3 mm	Yes	toolless

Figure 92: DST double fastener by Böllhoff [15]

10.3.3 Preload

A preloaded connection with these clips could in theory be possible if the plates (indicated with 1. In Figure 92) are designed thicker than the distance between the metal clip and the centerpiece of the frame. This way the clip would push with a force in the direction denoted with F in Figure 92, preloading the system. It is unclear however how comfortable it would be to manually establish this type of tight connection with the clip. A marginal rating is therefore assigned.

10.3.4 Area of pontoon elements

Because the spring clips from the reference are not strong enough to be used, the size of the potential pontoon elements cannot be determined with the reference size of the clips. In order to determine an approximate equivalent value for the pontoon elements is spring clips would be used, the size of the reference spring clip need to be scaled up. Assuming that a total of four clips are used (one for each corner), the reference force of 250kN can be divided by four, which gives an amount of 62.5kN per clip. Because, the pulling force from the reference is 50N, the desired force resistance is higher by a factor 1250. Equivalent spring clips that would be strong enough are assumed therefore to be bigger in size by the same factor of 1250. If the upscale is assumed to be equally distributed over all three dimensions, $1250^{1/3} = 10.8$ is the factor of increase in each direction. The used dimensions are therefore (with respect to the dimensions found in 10.3.1) equal to 32.4x35.9x10.8 cm.
The pontoon element is modelled for this estimation as an angular plate as indicated with 1. In Figure 92. This plate therefore has a rectangular gap of 32.4x10.8 cm. The reference force is assumed to operate in the same direction as the force F in the figure. This gives a combination of shear forces, bending moments and even torsion due to the applied force not being aligned with the shear center of the plate. However, a similar stress state as with the fishplates in section 10.1.4 is expected, where the bending stress is the more dominant stress, because of the relatively large arm of couple between the bottom welding and the top of the rectangular gap, where the spring clips apply their force. Similar plate dimensions as with the fishplates can therefore be assumed, which yields a satisfactory rating.

10.3.5 Maximum available area

Because these clips connect the pontoon plate with the crane flange from one plate side only, in contrary to the fishplate pin configuration, which needs to be connected at both plate sides, the pontoon plate can here be placed further from the crane. It can be placed adjacent to the crane flange as seen in Figure 92 instead of overlapping with the crane flange. However, in order to receive an acceptable rating for this indicator ($440m^2$), the distance from the crane needs to be increased by at least 1.2m at both sides, as can be seen in the figure in section 10.1.5. If crane flange dimensions similar to those of the pontoon plates are assumed, this distance is not reached. The same marginal rating as for the fishplates is therefore assigned.

10.3.6 Width of crane elements

The spring clip is assumed here to connect to the crane by means of an angular plate, as also seen in Figure 92. One half of the spring clip must stick out on both connection sides, so that the width of a plate side must be at least equal to half of the spring clip length so that it fits in between the crane and the pontoon plate, which is approximately 18 cm. Because the weld of the crane flange is applied at the side of the plate instead of at the bottom, any additions to this width do not benefit the stress states. It could moreover only increase the bending moments. The thickness of the plate can be adjusted for this purpose.
Because in the case of spring clips the crane flange and pontoon plate do not overlap, the two crane flanges at the back do not need to be widened as was needed for the pin configuration. The total increase in width is here thus only equal to 18 cm at both sides. A total increase of 0.36m is thus assumed, which yields a satisfactory rating.

10.3.7 Price

Four spring clips are assumed to be necessary. The price of the reference spring clip from Böllhoff is around $8 apiece [64]. Because a scale up by a factor of 1250 was assumed with the previous indicators, the same factor is used to scale up the theoretical price of a large-scale spring clip, which results in an approximate price of $40000 dollars in total for the clips. Because steel plates of similar size as for the fishplate pin configuration are used, the calculated price of $640 for those plates corresponds approximately to the price for the necessary plates for the spring clips, which means that the acceptable threshold of $91.500 – 5% will not be exceeded. An acceptable rating is thus assigned here.

10.3.8 Time of application

The spring clip connection is established quickly by simply inserting the clip through the gaps of the plates. However, as explained with the pin configuration, time will be lost in the exact aligning of the crane flanges with the pontoon plates. An acceptable rating will therefore be assigned here as well.

10.3.9 Number of personnel

The same argumentation used for the pin configuration applies here as well; The aligning process may require two persons cooperating, even though the spring clips are attachable by a single person. An acceptable rating is thus assigned.

10.3.10 Engineering complexity

This connecting has one unacceptable rating, related to the force resistance, and two marginal ratings, related to the preload and maximum available area.

To solve the problem with the force resistance, the spring clip can be scaled up without modifying much of the mechanical concept, which keeps the engineering complexity low. A solution for the preload may be feasible as well. This could be in the form of thicker plates, but with chamfered edges at the gaps, allowing a sliding motion of the clip from its original state to the preloaded state. This concept is schematically illustrated in Figure 93 below. The clip can be forced into the gap of a plate that is slightly thicker than the available space in between the two metal clips by pushing onto the angular edges of the upper side of the gap.

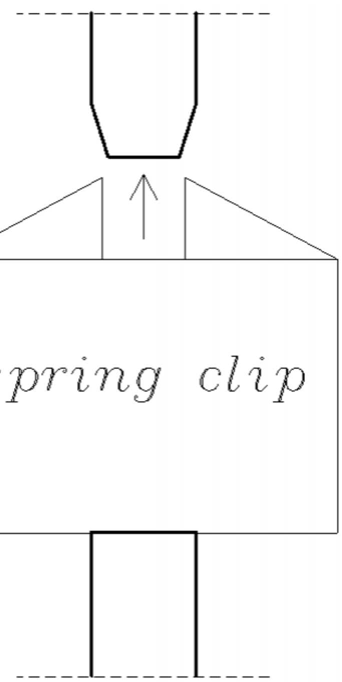


Figure 93: Preloading spring clip (side-view)

The maximum available area could be improved as well by simply extending the frame of the spring clip. Figure 92 shows that with the centerpiece on the frame (indicated with 2.), the two metal clips can work independently from each other and can be placed distantly from each other.
These adaptations are all achievable with low engineering complexity, such that a satisfactory rating can be assigned to this indicator.

10.3.11 Fabrication complexity

Even though these adaptations are not complicated to engineer, it is a big task to completely reconstruct the spring clip in a bigger form, which is not desirable for this project. A marginal rating is assigned here.

10.4 SPRING CLAMPS

10.4.1 Force resistance

The *Terrier* heavy-duty lifting clamp with locking lever found at the *Van Gool* [65] warehouse with a capacity of 166.8 kN is the reference spring clamp for assessment in this section. It is displayed in Figure 94.

Figure 94: Lifting clamp with locking lever [19]

As already addressed in section 2.5, These lifting clamps are designed to take up the gravitational force by pulling the object upwards and not by pushing the object upwards. In other words, their capacity is rated for tension and not compression. An animation created by the lifting clamp manufacturer *Pfeifer* [66] exhibits the inner structure and mechanism of a lifting clamp with locking lever. It shows that the locking mechanism is an asymmetrical mechanism that does not block inward translation of the toothed clamping piece in the same way as it blocks the outward translation. Inward loading could therefore result in an unstable loading situation on the entire frame with unknown consequences, which means that the force resistance indicator needs to be rated as unacceptable for this option.
In order to find out if the rated capacity of the reference clamps is suitable to the magnitude of the problem, the force resistance is composed out of tension forces only due to the deficiency elaborated above. This means that, similarly to the turnbuckle analysis, the entire reference force of 250kN needs to be resisted by a single side of the crane. If one clamp per crane corner is assumed, the force is divided over two clamps, which gives 125kN per clamp. The reference clamps with a capacity of 166.8 kN per clamps are thus of suitable scale to proceed the assessment with.

10.4.2 DoF restriction

The clamps are designed to resist the gravitational force which operates in principle always in the same direction, axially to the clamps. The teeth of the clamping piece are therefore not expected to prevent movements laterally to the clamp. However, if the clamps are placed in a diagonal way, similarly to the turnbuckle configuration, the force will always be partially axially to the clamps, which means that the clamping force will be activated in each direction. Restricting each of the six degrees of freedom is thus a feasible goal with these spring clamps, so that a satisfactory rating can be given here.

10.4.3 Preload

The locking mechanism ensures that the clamping piece is pressed tightly against the object without clearances, which means that instant tension is exerted on the object once the external load is applied. The ring at the other side of the frame however must be connected in a different way to the pontoon element. This requires a pin shackle as the mediator between the pontoon element and the lift clamp. The locking mechanism can be applied to the crane element once the ring is pressed firmly against the shackle, in the direction towards the crane. Even though it will not provide much tension, this situation can in theory be considered as a preloaded system, because any clearances are eliminated from the system. A satisfactory rating is thus assigned here.

10.4.4 Area of pontoon elements

The pin shackle which needs to be placed in between the lift clamp and the pontoon element can be modelled in the same way as the jaw end fitting of the turnbuckle. Considering that the reference force per clamp is equal to the reference force per turnbuckle, the same pontoon elements with the same area as for the turnbuckles can be applied here with the clamps, which yields a satisfactory rating.

10.4.5 Maximum available area

The length of the reference clamp is approximately 81 cm [67] and a diagonal connection with the four corner points is assumed as with the turnbuckles in order to maximize the maximum available area. As highlighted earlier, in order to obtain an acceptable rating for this indicator, the distance between the crane and the pontoon elements needs to be at least 1.2m at each side, which is out of reach for these clamps. Because the pontoon elements with these clamps can be placed further away from the crane than with the fishplate pin configuration, it can already be concluded that the rating is marginal for this indicator.

10.4.6 Width of crane elements

Because these clamps are able to clamp onto a rectangular block, the crane elements can be made of simple rectangles that are just as wide as the depth of the clamping slot, which is 20.9 cm [67] . Any element width larger than this depth would not add any benefits and would only increase the induced bending stresses. The elements can be strengthened by increasing its length along the crane as much as needed. With a 20.9 cm increase on both sides, a satisfactory rating is assigned to this indicator.

10.4.7 Price

The most expensive elements in this system are the lift clamps, which are sold for €2396 apiece [65], which is equivalent to approximately $2558. A total of four clamps would thus cost $10,232. The additional shackles, lashings and steel sheets are not as expensive to increase the total price over $18,300 as was already seen with the price rating of the previous options. A satisfactory rating is thus assigned.

10.4.8 Time of application

The clamps are quickly connected with pin shackles to the pontoon and with the locking lever to the crane. Because of the clamping slot depth, slight variation is enabled for the connection distance between the crane and pontoon, so that no precise alignment is needed when boarding the crane on the pontoon. A satisfactory rating can thus be assigned here.

10.4.9 Number of personnel

The weight of a single reference clamp is 71 kg [68]. It is unlikely that a single person can pull the clamp tightly against the shackle, as was described for the preloading process, and at the same time close the locking lever. Due to the weight of the clamp and lack of an advantageous gripping possibility, assumed is that two construction workers need both hands to hold the clamp in the preloaded position. Another person is therefore needed to close the locking lever. A total of three persons is therefore assumed, which yields a marginal rating.

10.4.10 Engineering complexity

The indicators which require improvement at this connecting option are the force resistance, the maximum available area and the number of personnel.
The force resistance requires an adaptation to the inner mechanism of the spring clamp which allows double-sided loading, so that it can take up the compression forces, which is not an engineering project without difficulties, but is executable nevertheless. The alternative is to design the pontoon elements such that compression forces are eliminated from the system. A special type of shackles needs to be designed for this, which allow for the necessary compressional tolerances due to the deformation/elongation at the other side. Compression due to shear needs to be taken into account here as well.
The second indicator which needs to be improved is the maximum available area. The only way to achieve this is to design a special type of shackles with a big length, which extends the distance between the pontoon elements and the crane. Special attention needs to be spent here that the long

body is designed stiff enough, so that the preloaded condition can still be achieved without the weight of the body causing too much deformation.

The third indicator can be tackled by improving the weight-efficiency of the design of the clamps. Even though it is a feasible task, considerable complexity might occur due to the probability that the designers of the clamp have already invested a decent amount of engineering into lowering the weight of the object as much as possible. Adapting the design to the extent that its weight can be comfortably handled by a single person might be troublesome.

Altogether, these adaptions can be considered complex but feasible, so that an acceptable rating can be assigned here.

10.4.11 Fabrication complexity

Performing a modification to the inner mechanism of the clamp without damaging any components is considered too complex of a fabrication task for this project. Fabricating a special type of shackle for solving the problems with respect to compression as well as the maximum available area also does not come without effort, but is considered acceptable, given that the shape and dimensions are predetermined.

Reducing the weight of the clamp can require exchange of various components of the inner structure and frame, which is requires an undesirable amount of time and precision.

Altogether, a marginal rating can be given here.

10.5 BAR CLAMPS

10.5.1 Force resistance

A reference heavy-duty bar clamp from the manufacturer *Pony Jorgensen* [69] as shown in Figure 95 is analyzed in this section for the assessment of the bar clamps connecting option. This variant is chosen because it can join two simple plates together by using it overhead. It has a rated tensile strength of approximately 6.7kN. Bar clamps with a higher capacity than this are not (widely) available. Because both the pontoon plate and the crane flange will be located in between the same clamps, the only possible stress on the bar will be tension, so that the compression strength does not need to be considered here.

If the reference force of 250kN is divided over four bar clamps (one for each corner), 62.5kN of resistance is needed per bar clamp. Since 6.7kN is under the 5% tolerance implemented in the boundary value scheme for the force resistance indicator, an unacceptable rating is assigned here.

Figure 95: Heavy-duty bar clamps [70]

10.5.2 DoF restriction

The main deficiency of this connecting option is that it offers a firm fixation in just a single dimension, which is parallel to the bar. The connected plates are perpendicularly to the bar limited to slide with respect to each other by friction, but translations are not prevented in these directions and a stable system is not established. An unacceptable rating is given to this indicator.

10.5.3 Preload

The screw handle allows for affirmation of the joining between the pontoon plate and crane flange, so that a preloaded situation can be created. A satisfactory is thus assigned here.

10.5.4 Area of pontoon elements

The pontoon elements require no holes in the case of bar clamps, which means that simple plates can be used as the elements. The clamping position is ideally located at the bottom of the elements, immediately above the welded connection between the element and the pontoon. The arm of couple can this way be limited to a minimum, so that bending moment can be neglected. The only relevant force on the element will then be the shear force. This is however not entirely executable, due to the crane flange. The situation is schematically drawn in Figure 96 below.

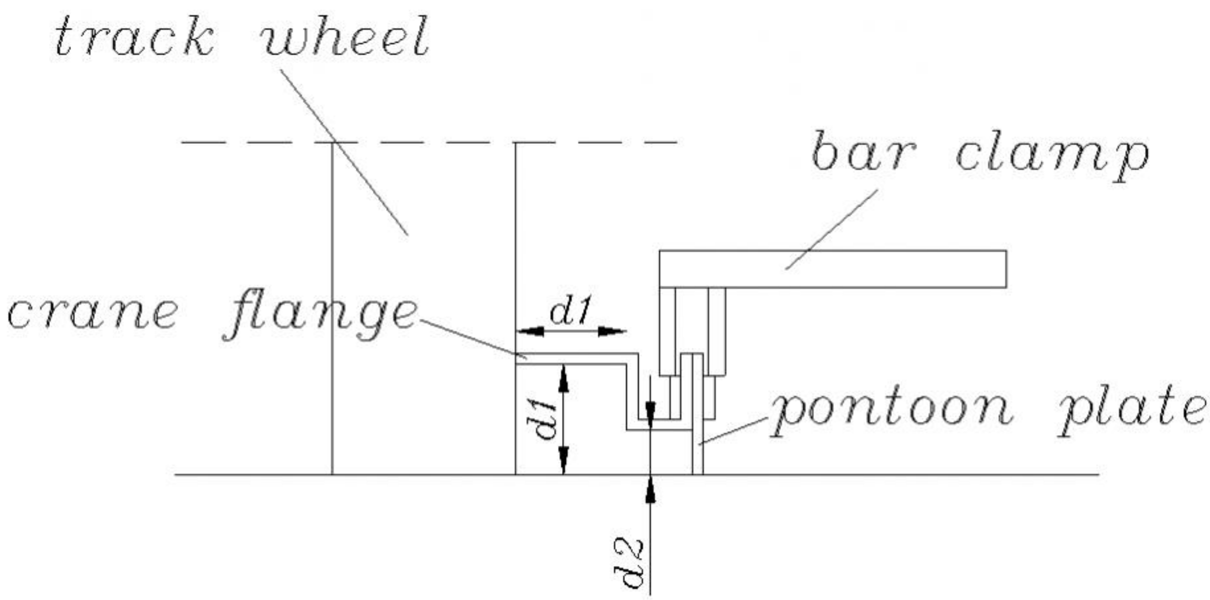


Figure 96: Schematic bar clamp system (front-view)

The crane flange cannot be connected to the lower corner of the track wheel, because the steel frame does not start immediately at the bottom. Also, space is required so that the weld connection can be applied from below. A rectangular area of d1 x d1 is therefore assumed to be left open below the connection. Furthermore, a clearance of d2 needs to be left open between the lowest point of the flange and the pontoon surface. Values of d1 = 10 cm and d2 = 4 cm are considered sufficient for this purpose. If a plate thickness of 1 cm is assumed, the clamping force is thus applied at an approximate height of 6.5 cm. This assumption is valid for the loading direction when the crane flange moves away from the pontoon plate. However, in the loading direction when the crane flange moves towards the pontoon plate, the pontoon plate operates as a barricade and the bar clamp provides no function at that instance. This can be considered as the worst-case loading situation for the pontoon elements, because the force between the plate and the flange is now distributed over their entire contact surface, so that the resultant force will be at half of the contact surface height, resulting in a bigger bending moment. By limiting however, the height of this contact surface such that it aligns with the clamping surface, the same height of 6.5 cm is applied. This value can therefore be used to estimate the maximum moment.

A calculation is performed to find out the necessary width of a plate with a 2 cm plate thickness and a height that is limited to 10 cm. The maximum bending moment is then equal to 62500*0.065=4062.5Nm. This value can then be inserted in formula (10) to obtain the maximum bending stress, which is located at the top fiber of the cross-section (1 cm from the neutral axis). The moment of inertia of a rectangular cross-section with a width of 10 cm is taken for the first iteration. A value of σ = 609.4 MPa is obtained.

The plate is thickened to 3 cm for the second iteration, resulting in $\sigma \approx$ 270 MPa.

A third is performed with a plate thickness of 3 cm and width of 20 cm, giving $\sigma \approx$ 135 MPa.

The shear stress, which maximizes at the neutral axis for a rectangular cross-section, is obtained with formula (12), which gives $\tau_{max} \approx$ 16 MPa with these dimensions. This results in $\sigma' \approx$ 137.8 MPa.

An area of 0.2x0.03 m per plate is thus an adequate value to proceed with. Considering a total of four plates, the total area of the pontoon elements is equal to 0.024 m^2, which yields a satisfactory rating.

10.5.5 Width of crane elements

The relevant parameters for determining the total width of the crane elements is the distance d1 together with the space left open in between the U-shape for applying the clamp (see Figure 96). This space must at least be equal to the width of left orange part of the frame of the bar clamp. The height of the steel bar is given from the data as 3.3 cm [71]. From the proportions seen in Figure 95, the maximum width of the left orange part can be estimated at 7 cm. Adding two times the plate thickness from the vertical parts of the U-section and the distance d1 = 10 cm, a total width of approximately 20 cm is obtained. Because the configuration is applied at both crane sides, the total width is taken as 40 cm, which is within the range for a satisfactory rating.

10.5.6 Maximum available area

The clamps require that the pontoon plates are located adjacent to the crane flanges. The distance from the crane to the pontoon elements is thus equal to the width of the crane elements, which is 20 cm. As seen in the previous ratings for this indicator, this value corresponds to a marginal rating.

10.5.7 Price

The total price for four reference bar clamps is approximately $110. However, because the rated capacity is approximately tenfold lower than needed, this price is multiplied by 10 to receive a more suitable price-range for the object if it would be scaled up, giving $440. Furthermore, from Figure 96 it can be observed that the bending moments on the crane flanges are slightly smaller than the bending moments on the pontoon plates. A plate thickness of 3 cm suffices therefore also for the crane flanges. The stress due to axial force on the flange is only around 10 MPa with that plate thickness and 20 cm plate width. These plate dimensions can thus be applied for the crane flanges as well as the pontoon plates.
From previous price analysis for previous connecting options can already been concluded that the purchasing the necessary steel with these dimensions will not make the total price exceed $18.300, which means that a satisfactory rating is assigned.

10.5.8 Time of application

The clamps are applied by placing the clamps on the plates and are connected with simple rotation of the handle until the maximum is reached, which is not expected to consume significant time. At the boarding process, some time is lost by aligning, because the pontoon plates are located immediately next to the crane flanges in this system. The crane therefore needs to be transversally centered with more precision. Because there are no plate holes in this system and the width of the plates is significantly bigger than the width of the clamps, longitudinal precision is less relevant. An acceptable rating can be assigned to this indicator.

10.5.9 Number of personnel

The weight of the object is 3.5 kg and it has a bar length of 76 cm [69]. It furthermore has single-handed gripping possibility. In a corresponding scale-size however, the weight would approximately be 35 kg. 2 cooperating workers are therefore assumed for carrying out the connection. Because the aligning process is limited to a single axis with as highlighted in the section above, it is assumed that no additional person is needed for this purpose. An acceptable rating is therefore assigned here.

10.5.10 Engineering complexity

The potential adaptations to this connecting option need to be designed with respect to the force resistance, the DoF restriction and maximum available area.
The force resistance can be solved by simply scaling up the design of the clamps.
The DoF restriction can potentially be resolved by simply adding steel pieces to the frame that precisely enclose the connected plate and flange when applied.
The maximum available area, or in other words the distance between the crane flange and the pontoon plate, is more difficult to increase without negatively affecting other performance indicators. An easy solution would be to extend the bar and implement a mirrored version of the clamp at the other end of the bar as schematically visualized in Figure 97. This would however introduce compression stresses on the bar, which would most likely be problematic because of its length. Reinforcements for this are designable as well though, as discussed with the turnbuckle option. This would require the engineering of a stiffener with a particular type of gap in order to conserve the gripping possibility. The stress concentrations here will need to be considered. An opportunity to reduce the effective buckling length is not available here. The alternative option to design the crane- and pontoon elements in a way that eliminates compression stresses from the system is more difficult to design here than at the turnbuckle, because of the lack of rotation possibilities at the clamping point.
All problems appear in advance to be realistically solvable, but not without complexities, so that an acceptable rating can be assigned to this indicator.

Figure 97: Potential bar clamp adaptation

10.5.11 Fabrication complexity

If a scale-up is carried out, an entire frame needs to be newly fabricated. Elementary shapes can be used for this, which do not necessarily match the shape of the reference clamps in order to conserve simplicity.
The DoF restriction can be solved with the simple welding of additional pieces of steel onto the existing frame. The extension and mirroring of the bar clamp is also not a complex job. If two clamps are used and one end is cut off from both clamps, these ends can be welded together, which results in a symmetric bar clamp with a bigger length at the same time. The manufacturing and welding of a stiffener with a proper gripping gap or specific connection elements for eliminating compression stresses could potentially cause more difficulties, as already discussed for the turnbuckle. Altogether, an acceptable rating is most suitable.

10.6 SCISSOR LINKAGE

10.6.1 Force resistance

The scissor linkage, as discussed in section 2.7, is a general connecting principle found in various applications and products. They can be simply built up with steel slabs with the desired dimensions. Instead of pins at each intersection of the slabs, rigidly welded connections at each intersection is necessary for this project. In order to rate the force resistance, a scissor linkage of feasible size needs to be analyzed as the reference. A mini scissor lift is used for this purpose, as shown in Figure 98 below. The composition of the scissor can be considered as an assembly of three identical adjacent and double-sided diamond shapes with a total length of approximately 1.7m as estimated from the figure.
In order to estimate the force resistance, the necessary force to make the cross-section yield under simple axial stress is calculated. The cross-section of one slab is assumed to be a thin-walled hollow rectangular profile with dimensions estimated as: height = 8 cm, width = 4 cm, wall thickness = 0.3 cm, which is a standard set of dimensions for this shape [72]. This is not the cross-section however that can be calculated with. Because the connection points need to be custom made with this option, pin holes need to be added at the ends of the first and last slab of the profile. The stress will be therefore be concentrated the most at these ends, where the cross-section is reduced due to the holes.
The pin size used for the jaw of the reference turnbuckle analyzed in section 10.2 is valid for this application as well, which is 51 mm, with a hole size of 51.16 mm for a clearance fit. The cross-sectional area of the profile is calculated at this hole, which results in a value of approximately 0.00055 m^2. Because the cross-section exists out of four of these slabs, this profile area is multiplied by four, giving the following cross-sectional area of 0.0022 m^2. This area is multiplied with the yield stress for steel of 235 MPa, resulting in a force resistance of approximately 518kN, which is significantly higher than the reference force, which is 62.5kN if one linkage per corner is assumed. Because the four slabs are placed at big distances from each other and are interconnected at each half-length of a slab, the slenderness of the profile is kept low, so that compression/buckling is assumed not to be an issue with this connecting option. A satisfactory rating can thus be assigned to this indicator.

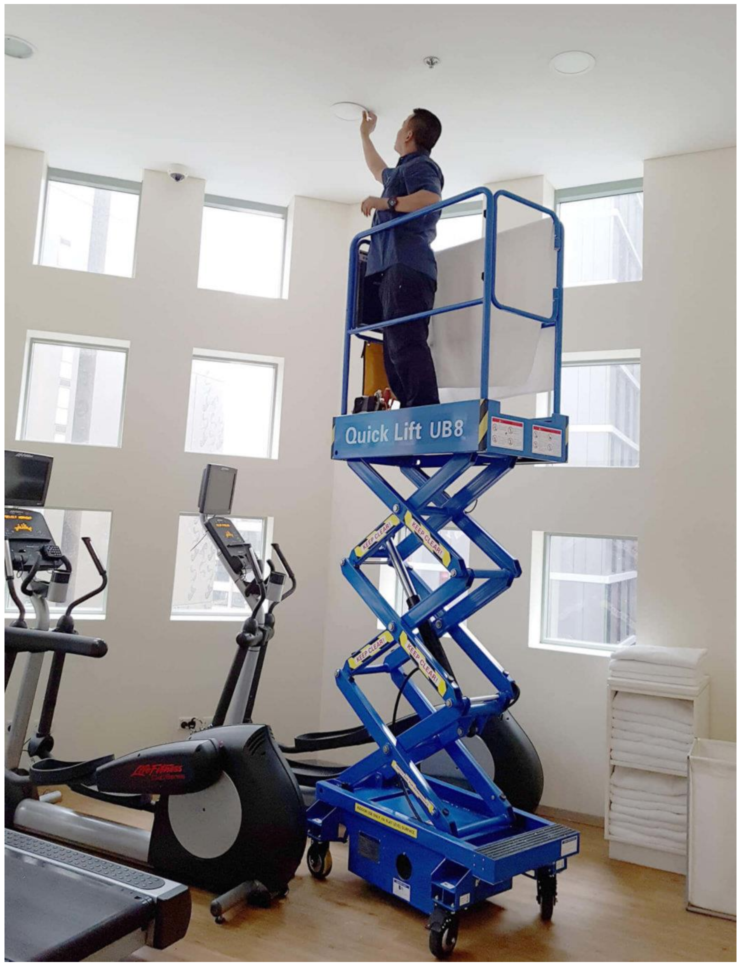


Figure 98: Mini scissor lift [73]

10.6.2 DoF restriction

The pin connections at the ends of the profile can be configured in way that restricts all degrees of freedom, as already discussed at the assessment of the turnbuckle. A satisfactory rating can thus be assigned here.

10.6.3 Preload

A clearance is present at the holes of connection points. The scissor mechanism does not offer the possibility of preloading the system, so that a marginal rating is assigned here.

10.6.4 Area of pontoon elements

The connection points and stress states can be modelled as similar to the turnbuckle analysis, so that a satisfactory rating is applicable for the area of pontoon elements for this option as well.

10.6.5 Maximum available area

The length of the reference scissor linkage is estimated at 1.7m. With a diagonal configuration with a 60°/30° angle proportion, the pontoon elements can be placed at a transversal distance of approximately 1.47 m, corresponding to a maximum available area of 462 m^2. This yields an acceptable rating for this indicator.

10.6.6 Width of crane elements

As stated in section 10.6.4, similar elements as for the turnbuckle analysis can be used, so that a satisfactory rating can be assigned here as well.

10.6.7 Price

The reference scissor linkage exists out of 3 adjacent 45° diamonds with a total length of 1.7 and is double-sided. This gives a total of 24 thin-walled hollow rectangular steel slabs with the dimensions as determined above that are necessary for this system. Parkersteel reports a price of £471.84 ≈ $585 [72]. Four scissor linkages give a total price of approximately $2340. Considering that the crane- and pontoon elements are modelled as similar to the turnbuckle analysis, the accumulated price of the connection system cannot exceed $18.300, so that a satisfactory rating is given.

10.6.8 Time of application

Because the slabs of the scissors are rigidly welded to each other, the linkage has no variable length. The pin holes of the scissors and the crane elements have thus one fixed spot of intersection on the pontoon, which means that the crane needs to be precisely positioned at an exact spot at each boarding process. This might consume significant time. Other than that, no excessive time will be lost, considering the simple pin fittings at the ends. An acceptable rating can be assigned.

10.6.9 Number of personnel

The scissor linkage with the dimensions as determined before would have a weight of approximately 210 kg if a steel density of 8000 kg/m^3 is assumed. However, because the structure exceeds the design force by a big margin with these dimensions, the cross-sectional area can be lowered for the purpose of the assessment of this indicator.
Because the relation between the weight of the object and the cross-section is linear, the relation between the force resistance and the weight can be considered linear due to the simplified loading case ($F = \sigma * A$). The force resistance of 518kN exceeds the reference force of 62.5kN by a factor of 8.3. For the force resistance indicator to remain satisfactory, a minimum factor of 3 is taken for this case, so that a minimum force resistance of 187.5kN is required. The current force resistance can thus be lowered by a factor of 518/187.5 = 2.76. The weight can thus potentially be reduced by this factor as well, giving 210/2.76 = 76 kg. This is an acceptable weight for two workers to carry around for an object with good gripping possibilities. An acceptable rating can thus be assigned.

10.6.10 Engineering complexity

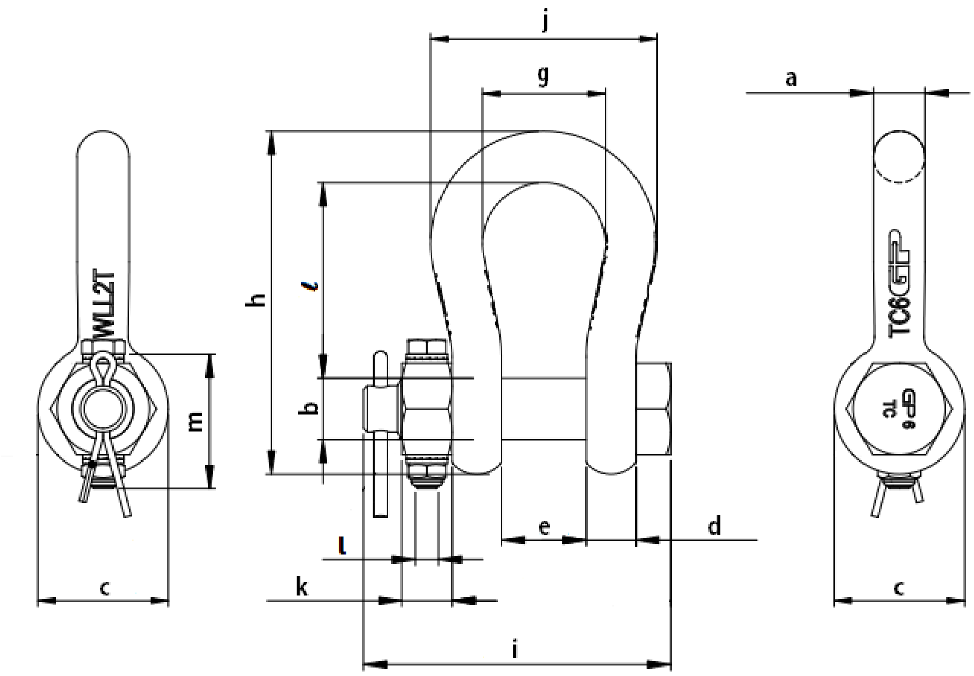
The only indicator that requires improvement is the preload. The problem is related to the clearance at the connection point in between the pin holes and the pins due to the desired clearance fit. Similarly to the analysis made for fishplate pin configuration for this indicator in section 10.1.10, engineering a solution for the preload which conserves the ability of all four linkages to be engaged at the same time requires considerable complexity, like a pin hole with a variable diameter. Doubling the size and weight of the linkages is not an option, because it would drop the ratings for the manageability indicators. A marginal rating is thus assigned here.

10.6.11 Fabrication complexity

The only potential option of modification found in the engineering complexity section above is the pin hole with variable diameter. Because the structure of this potential adaptation is relatively unknown in advance, a specific assessment cannot be carried out for this indicator. It can be assumed however that it would exist out of small components that need to be constructed with high precision. It can be expected that it would require significant effort at the fabrication process. A marginal rating is thus assigned here as well.

11 Appendix C (shackle dimension sets)

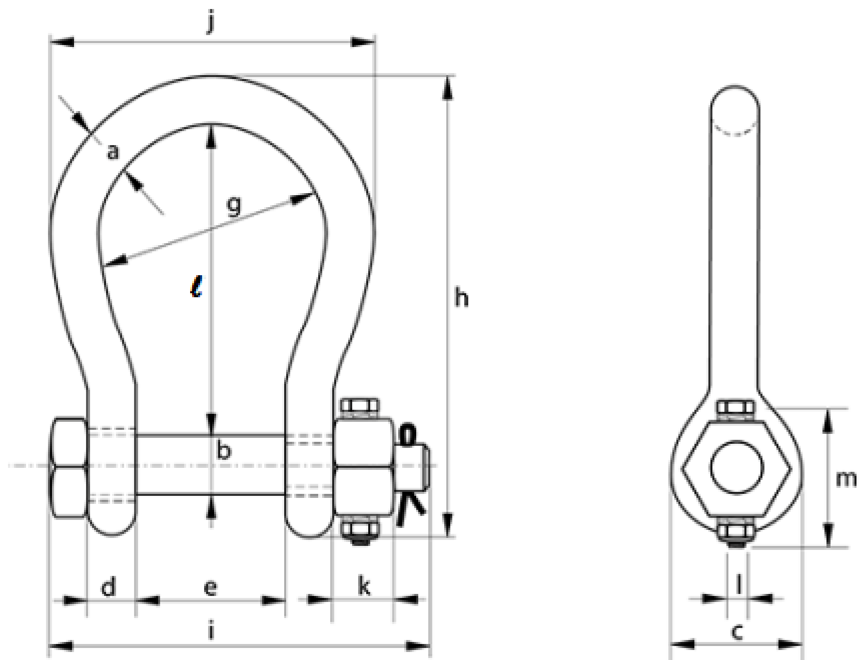
11.1 Standard bow shackle dimension sets



working load limit (ton)	Net weight (kg)	diameter bow (mm) a	diameter pin (mm) b	diameter eye (mm) c	width eye (mm) d	width inside (mm) e	length inside (mm) ℓ	width bow (mm) g	length (mm) h	length bolt (mm) i	width (mm) j	thickness nut (mm) k	Securing bolt thread (mm) l	securing bolt length (mm) m
2	0.42	13,5	16	34	13	22	51	32	90	80	59	13	M6	35
3,25	0.74	16	19	40	16	27	64	43	110	98	75	17	M6	40
4,75	1.18	19	22	46	19	31	76	51	129	115	89	19	M6	45
6,5	1.77	22	25	52	22	36	83	58	144	130	102	22	M8	50
8,5	2.58	25	28	59	25	43	95	68	164	150	118	25	M8	55
9,5	3.66	28	32	67	28	47	108	75	186	166	131	27	M10	60
12	4.80	32	35	73	32	51	115	83	201	184	147	30	M10	65
13,5	6.54	35	38	79	35	57	133	92	227	197	162	33	M10	70
17	8.19	38	42	88	38	60	146	99	249	202	175	19	M8	75
25	14.0	45	50	104	45	74	178	126	300	243	216	23	M8	90
35	19.9	50	57	112	50	83	197	138	332	269	238	26	M10	100
42,5	28.3	57	65	132	57	95	222	160	378	301	274	29	M12	110
55	39.6	65	70	145	65	105	260	180	433	329	310	32	M12	120
85	62.0	75	83	167	75	127	330	190	530	381	340	39	M12	140

Figure 99: Standard bow shackle dimension sets [35]

11.2 BigMouth bow shackle dimension sets



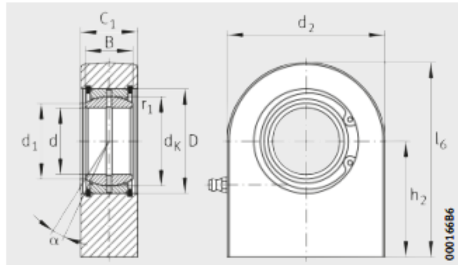
working load limit (ton)	Net weight (kg)	diameter bow (mm) a	diameter pin (mm) b	diameter eye (mm) c	width eye (mm) d	width inside (mm) e	length inside (mm) ℓ	width bow (mm) g	length (mm) h	length bolt (mm) i	width (mm) j	thickness nut (mm) k	Securing bolt thread (mm) l	securing bolt length (mm) m
4,75	2.08	22	25	52	22	63	112	88	173	157	132	22	M8	50
6,5	3.14	25	28	59	25	75	135	105	204	183	155	25	M8	55
8,5	4.36	28	32	66	28	82	148	115	225	205	171	27	M10	60
9,5	5.95	32	35	72	32	90	162	126	248	224	190	30	M10	65
12	7.87	35	38	79	35	100	180	140	274	245	210	33	M10	70
16	10.2	38	42	88	38	106	216	159	319	248	235	19	M8	75
25	16.7	45	50	103	45	127	248	175	370	296	265	23	M8	90
30	25.0	50	57	118	50	146	273	207	411	332	307	26	M10	100
55	45.0	65	70	145	65	165	314	213	487	389	343	32	M12	120
75	70.0	83	83	164	83	184	330	254	537	455	420	39	M12	140

Figure 100: BigMouth bow shackle dimension sets [34]

12 APPENDIX D (ROD END DIMENSION SETS)

Hydraulic rod ends

With rectangular welding face
Requiring maintenance
Open design



GF..-DO
Steel/steel

Dimension table · Dimensions in mm

Designation[1]	Mass	Dimensions						
	m ≈kg	d	D	B	d_K	d_1	d_2	h_2
GF20-DO	0,35	**$20_{-0,01}$**	35	$16_{-0,12}$	29	24,2	50	38
GF25-DO	0,53	**$25_{-0,01}$**	42	$20_{-0,12}$	35,5	29,3	55	45
GF30-DO	0,87	**$30_{-0,01}$**	47	$22_{-0,12}$	40,7	34,2	65	51
GF35-DO	1,5	**$35_{-0,012}$**	55	$25_{-0,12}$	47	39,8	83	61
GF40-DO	2,4	**$40_{-0,012}$**	62	$28_{-0,12}$	53	45	100	69
GF45-DO	3,4	**$45_{-0,012}$**	68	$32_{-0,12}$	60	50,8	110	77
GF50-DO	4,4	**$50_{-0,012}$**	75	$35_{-0,12}$	66	56	123	88
GF60-DO	7,1	**$60_{-0,015}$**	90	$44_{-0,15}$	80	66,8	140	100
GF70-DO	10,5	**$70_{-0,015}$**	105	$49_{-0,15}$	92	77,9	164	115
GF80-DO	15	**$80_{-0,015}$**	120	$55_{-0,15}$	105	89,4	180	141
GF90-DO[4]	23,5	**$90_{-0,02}$**	130	$60_{-0,2}$	115	98,1	226	150
GF100-DO[4]	31,5	**$100_{-0,02}$**	150	$70_{-0,2}$	130	109,5	250	170
GF110-DO[4]	48	**$110_{-0,02}$**	160	$70_{-0,2}$	140	121,2	295	185
GF120-DO[4]	79	**$120_{-0,02}$**	180	$85_{-0,2}$	160	135,6	360	210

Designation[1]					Chamfer dimension	Basic load ratings		Radial internal clearance
	C_1 nom.	C_1 max.	α[2] °	l_6	r_1 min.	dyn. C_r N	stat. C_{0r}[3] N	
GF20-DO	19	20	9	63	0,3	29 600	65 600	0,030 – 0,082
GF25-DO	23	24	7	72,5	0,6	48 300	68 800	0,037 – 0,1
GF30-DO	28	29	6	83,5	0,6	62 300	116 000	0,037 – 0,1
GF35-DO	30	31	6	102,5	0,6	79 900	193 000	0,037 – 0,1
GF40-DO	35	36,5	7	119	0,6	99 100	306 000	0,043 – 0,12
GF45-DO	40	41,5	7	132	0,6	128 000	386 000	0,043 – 0,12
GF50-DO	40	41,5	6	149,5	0,6	157 000	442 000	0,043 – 0,12
GF60-DO	50	52,5	6	170	1	245 000	558 000	0,043 – 0,12
GF70-DO	55	58	6	197	1	313 000	725 000	0,055 – 0,142
GF80-DO	60	63	6	231	1	402 000	804 000	0,055 – 0,142
GF90-DO[4]	65	69	5	263	1	489 000	1 350 000	0,055 – 0,142
GF100-DO[4]	70	74	7	295	1	608 000	1 520 000	0,065 – 0,165
GF110-DO[4]	80	85	6	332,5	1	655 000	2 340 000	0,065 – 0,165
GF120-DO[4]	90	95	6	390	1	952 000	3 400 000	0,065 – 0,165

Figure 101: Rod end dimension sets (Schaeffler) [37]

13 APPENDIX E (TURNBUCKLE DIMENSION SETS)

Green Pin® JJ Turnbuckle CP

Turnbuckle with jaw-jaw end-fitting and cotter pins, generally to ASTM F1145-92

Article code	working load limit	take up (inch)	A diameter thread (inch)	B length closed position	C length open position	D length closed position	E opening jaw	F length inside	G diameter pin	H thickness jaw eye	I diameter jaw eye	Net weight (kg)
SSGPGG1906	2,36	6	3/4	369	487	439	24	38	16	16	41	2.59
SSGPGG1909	2,36	9	3/4	444	640	514	24	38	16	16	41	3.13
SSGPGG1912	2,36	12	3/4	520	792	590	24	38	16	16	41	3.42
SSGPGG1918	2,36	18	3/4	670	1096	740	24	38	16	16	41	4.51
SSGPGG2212	3,27	12	7/8	561	826	638	27	42	19	19	48	4.93
SSGPGG2218	3,27	18	7/8	713	1132	790	27	42	19	19	48	6.41
SSGPGG2506	4,54	6	1	447	554	532	31	50	22	20	54	5.18
SSGPGG2512	4,54	12	1	598	859	683	31	50	22	20	54	6.43
SSGPGG2518	4,54	18	1	750	1168	835	31	50	22	20	54	8.08
SSGPGG2524	4,5	24	1	903	1470	988	31	50	22	20	54	8.56
SSGPGG3212	6,9	12	1 1/4	641	914	748	44	71	29	26	68	11.2
SSGPGG3218	6,9	18	1 1/4	803	1228	910	44	71	29	26	68	13.6
SSGPGG3224	6,9	24	1 1/4	962	1539	1069	44	71	29	26	68	15.0
SSGPGG3812	9,71	12	1 1/2	675	942	806	52	71	35	28	80	17.1
SSGPGG3818	9,71	18	1 1/2	825	1244	956	52	71	35	28	80	19.3
SSGPGG3824	9,71	24	1 1/2	980	1551	1111	52	71	35	28	80	21.8
SSGPGG4518	12,7	18	1 3/4	938	1316	1092	60	86	41	33	90	27.3
SSGPGG4524	12,7	24	1 3/4	1089	1621	1243	60	86	41	33	90	31.7
SSGPGG5024	16,8	24	2	1151	1671	1338	63	93	51	40	107	49
SSGPGG6424	27,2	24	2 1/2	1255	1831	1480	75	114	57	41	143	88
SSGPGG6924	34	24	2 3/4	1348	1882	1604	90	110	70	41	158	103

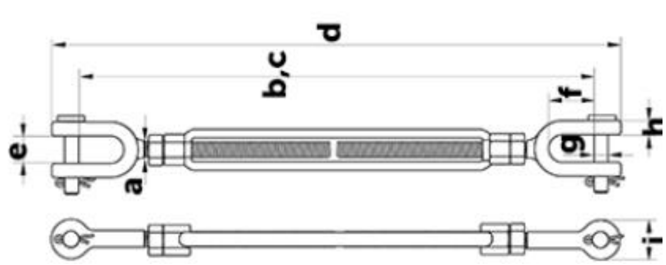


Figure 102: Turnbuckle dimensions (GreenPin)

14 APPENDIX F (EUROCODE WELDING REQUIREMENTS)

BS EN 1993-1-8:2005
EN 1993-1-8:2005 (E)

4.5.3.2 Directional method

(1) In this method, the forces transmitted by a unit length of weld are resolved into components parallel and transverse to the longitudinal axis of the weld and normal and transverse to the plane of its throat.

(2) The design throat area A_w should be taken as $A_w = \sum a\, \ell_{eff}$.

(3) The location of the design throat area should be assumed to be concentrated in the root.

(4) A uniform distribution of stress is assumed on the throat section of the weld, leading to the normal stresses and shear stresses shown in Figure 4.5, as follows:

- $\sigma_\perp$ is the normal stress perpendicular to the throat
- $\sigma_\parallel$ is the normal stress parallel to the axis of the weld
- $\tau_\perp$ is the shear stress (in the plane of the throat) perpendicular to the axis of the weld
- $\tau_\parallel$ is the shear stress (in the plane of the throat) parallel to the axis of the weld.

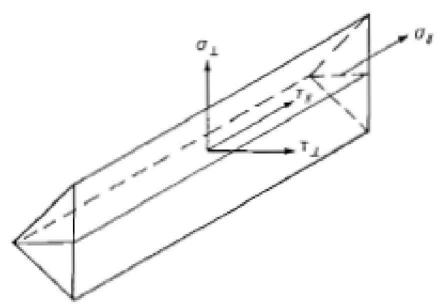


Figure 4.5: Stresses on the throat section of a fillet weld

(5) The normal stress $\sigma_\parallel$ parallel to the axis is not considered when verifying the design resistance of the weld.

(6) The design resistance of the fillet weld will be sufficient if the following are both satisfied:

$$[\sigma_\perp^2 + 3(\tau_\perp^2 + \tau_\parallel^2)]^{0,5} \leq f_u / (\beta_w \gamma_{M2}) \quad \text{and} \quad \sigma_\perp \leq 0.9 f_u / \gamma_{M2} \qquad \text{... (4.1)}$$

where:

f_u is the nominal ultimate tensile strength of the weaker part joined;

β_w is the appropriate correlation factor taken from Table 4.1.

(7) Welds between parts with different material strength grades should be designed using the properties of the material with the lower strength grade.

Table 4.1: Correlation factor β_w for fillet welds

Standard and steel grade			Correlation factor β_w
EN 10025	**EN 10210**	**EN 10219**	
S 235 S 235 W	S 235 H	S 235 H	0,8
S 275 S 275 N/NL S 275 M/ML	S 275 H S 275 NH/NLH	S 275 H S 275 NH/NLH S 275 MH/MLH	0,85
S 355 S 355 N/NL S 355 M/ML S 355 W	S 355 H S 355 NH/NLH	S 355 H S 355 NH/NLH S 355 MH/MLH	0,9
S 420 N/NL S 420 M/ML		S 420 MH/MLH	1,0
S 460 N/NL S 460 M/ML S 460 Q/QL/QL1	S 460 NH/NLH	S 460 NH/NLH S 460 MH/MLH	1,0

Figure 103: Eurocode welding requirements [41]