19-6-2019

# Ship to Shore Cranes


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# Summary

This report describes the design of an STS-crane that transports containers from the ship to the shore. These STS-cranes often have to lift loads of over 50 tons and should be able to do that to a height of approximately 40 meters.
Using the given requirements, different concepts were first made for the trolley, the boom and the hoist. Drawings of all these concepts were then made to explain and show the mechanism behind the several designs. Per component, the best design was chosen and the three designs were put together to form one final design that is used in the remainder of the project.

After having made the final design choice, initial dimensions were set up. These initial dimensions were used for for instance the material choice. The chosen material for the entire frame of the crane, for example, is a structural steel S420, which has a higher yield strength than conventional steel. This material choice for the S420 steel is then used as input for the FEM-package, in which the deflection in the several components of the crane can be determined. By using solid legs and a hollow frame, it was managed to stay within the deformation limits and also have a high enough safety factor to prevent buckling.

Then, the machine elements were determined. Some components were calculated, such as the gears and the key, and other components, such as the bearings, were determined from a catalogue. From the stress calculations of the gears, it became visible that the gears need extra heat-treatments to be able to withstand the high stresses. When those extra treatments are done, the gears should perform well.

Now that the dimensions of the beams and the forces are known, an in-depth analysis of specific parts of the crane could be done. There was looked at connections that needed to be bolted or welded to determine sizes for the bolts and welds.

Lastly, Mohr's circle was applied to the shaft in the drum, as this shaft is exerted to torque and shear force. With using the calculations for the shear force and the torsion moment, it was determined that the maximum shear stress in the shaft is equal to 28.2 MPa.

The calculations done in FEM combined with the in-depth analyses of the chosen components show that the designed crane will not deform too much or undergo buckling. Therefore, the crane will function as wanted with the determined machine elements.

# Contents

# 1 Introduction

Nowadays a lot of goods are transported over the sea using container ships. When those ships arrive at their destination the containers need to be transported from the ship to the shore. The easiest way to do this is by using a Ship-to-Shore crane (STS crane). In this report, the designing process of such a crane is documented.
Firstly all the requirements for such a crane are determined. Then, by using a morphological diagram multiple concepts will be created for the main parts, namely the boom, the trolley and the hoist. Drawings are created for all of them to make clear what the idea is and from these concepts, a final design will be chosen which will be used for the rest of the project. Initial dimensions are then set up to be able to select materials. Using constraints, literature and the CES Edupack, the materials will be selected for specific components of the crane.
After having specified the materials and the initial dimensions, calculations can be done. With the help of the FEM-package in MatLab, it will be determined whether the crane will stay within the given limits and dimensions and weight will be optimized. Besides doing calculations in FEM, more calculations will be done in order to choose a drive system for the hoist. With these calculations, the shafts, keys, bearings and gears that are going to be used are determined.
Then, there will be zoomed in on specific parts of the crane where parts are welded or bolted and where a highly loaded hinge is used. Widths of welds and bolt- and bearing sizes will be determined using these in-depth analyses. The last part that will be researched in-depth, is a shaft that is in a multi-axial stress state. Using Mohr's circle, the maximum shear stress and/or normal stress will be determined for this specific shaft.

# 2 Problem Definition

The crane consists of a couple of key points that have to be designed in much detail. The boom, trolley and hoist mechanism have to be designed and after that, a frame has to be created. The purpose of this assignment is to design a fully-operational STS crane.

The boom has some requirements that need to be taken into consideration before designing it. The maximum outreach has to be 45 meters. This length is chosen because the width of a vessel is limited to 35 meters, this means that there is a buffer of 10 meters, so the containers can be placed all over the ship. The maximum wind speed, which is 72 km/h, should also be looked at because different concepts of a boom will have a different contact area with the wind. The boom also needs to be able to be lifted up, so a container ship can move easily under the crane. To assure the boom will remain lifted, a locking mechanism has to be proposed. For lifting the boom a hinge has to be designed that can withstand all the forces that will appear. Furthermore, a concept that elaborates on a lifting mechanism has to be made. The maximum weight of a single container is 50 tons, this means that the right materials have to be chosen so the boom will not fail. Choosing right materials is also important when looking at the maximum operating temperatures: -40°C to 50°C. There are also some maximum deflection values in three directions: perpendicular to gantry rails (along the boom): 5 mm, parallel to gantry rails: 60 mm and vertical: 150 mm.

Secondly, a trolley should be sketched. The trolley has one function, which is moving along the boom. The speed range of the trolley is 180-210 m/min. This asks for a drive mechanism that powers the trolley continuously and one which can easily switch between forward movement and backward movement. Besides that, there also has to be thought of something that stops the trolley. Then, how is the trolley attached to the boom, because different boom concepts require different clamping ways. There should also be looked at the temperature range because it is not preferable that the trolley cannot move anymore when it is freezing.

The third part is the hoist mechanism. This part also only has one function, which is lifting the containers up and down. The speed range of the hoist is 90-120 m/min. For lifting the containers, the maximum weight of the containers is an important requirement. The motor(s) should be capable of lifting the containers without any problems. Wind speed is probably the most important point of attention for this mechanism because the containers have to be kept stable when they are lifted in case of heavy winds. To provide very precise placement of the containers, movement perpendicular to the boom on the trolley itself should be realised.

# 3 Boom

## 3.1 Functionality

The main purpose of the boom is to be able to deal with the forces applied on it by the weight of the containers it is lifting and to be able to lift itself under an angle to let ships pass underneath it. To create a concept for the boom a morphological diagram has been set up. Out of this diagram, three different concepts have been created.

## 3.2 Morphological Diagram

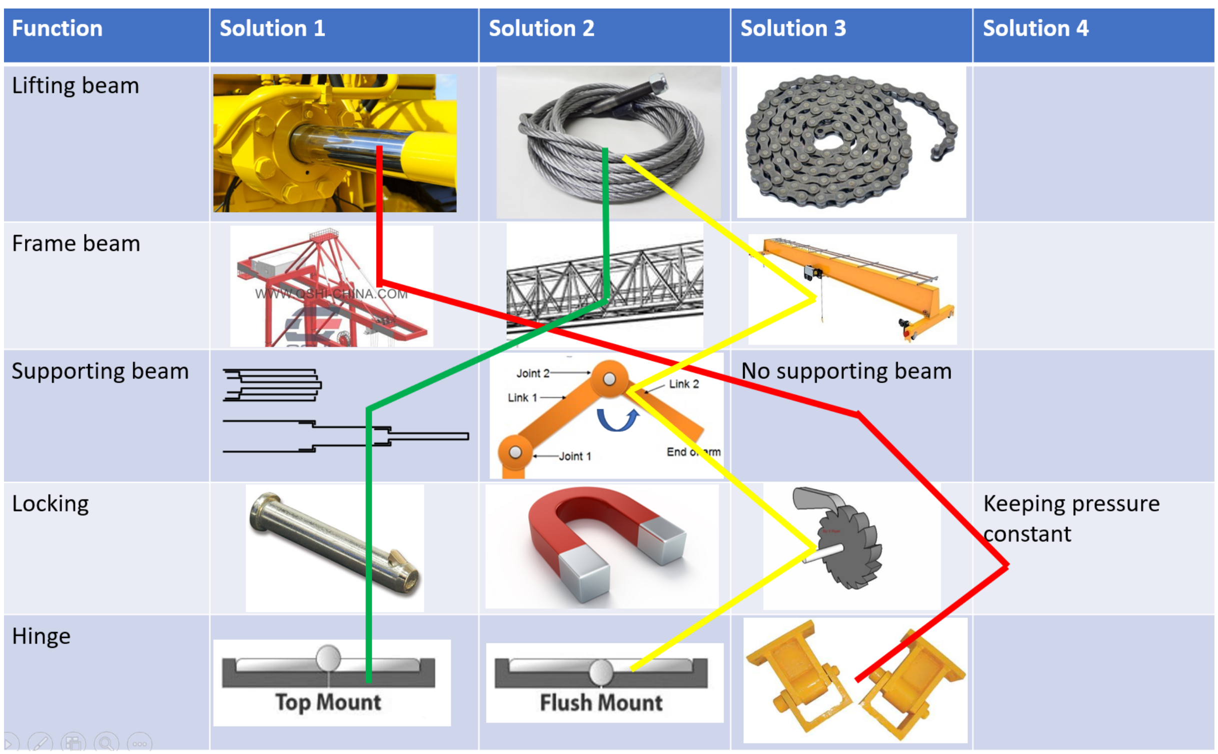


Figure 1: Morphological diagram boom

For the boom, five different parts/mechanisms were distinguished that were found to be important for the functionality of the boom. These are the lifting mechanism, the frame of the beam, the supporting beam, the locking mechanism and the hinge, which are all displayed in Figure 1.

For the lifting mechanism, three solutions were thought of: a hydraulic mechanism, a rope mechanism and a chain mechanism. The rope mechanism works with two ropes running from the front to the back of the crane while being spun over the main pillar of the crane. Using a winch the ropes can be retracted because of which the boom will lift. The chain mechanism works more or less the same but it uses chains instead of ropes.

For the frame, the solutions were: A hollow shape with supports in the middle, a single beam lattice frame and a simple I-beam.

The supporting beam should be able to support the boom but also make room for the boom to lift. Two solutions were created for the supporting beam, namely: a telescoping arm and a folding structure.

The telescoping arm is attached to the main pillar and the boom via two hinges. When the boom is being lifted the different components in the telescope arm fold into each other to make room for the boom to lift. The folding structure consists of two separate beams that are connected by a plate in the middle with pins. When the boom is being lifted the beams will fold inwards so that space is created for the boom. For the hydraulic mechanism, a supporting mechanism is not needed since it acts as a support and a lifting mechanism.

For the boom to stay in its place a locking mechanism was needed. Three mechanisms were thought of, namely: a pin mechanism, a magnet and a ratchet mechanism. The pin and ratchet mechanisms can only be used while using a rope mechanism for lifting the boom. When the boom is fully lifted a pin can slot into a hole in the winch to prevent it from lowering the ropes and thus locking the boom from going down. The ratchet mechanism works with a ratchet wheel and a pawl. The pawl is placed on top of the ratchet wheel because of which the wheel can only turn one way. The ratchet wheel is placed along the same axis as the winch. So as the boom is being lifted it is constantly being locked in place by the pawl preventing movement in the other direction. For the boom to be lowered down again the pawl should be slid of the ratchet wheel. A more general solution is a magnet that keeps the boom in place although this would probably not work that well seeing as the boom is quite heavy and the rest of the structure would also be attracted to the magnet. For the hydraulic mechanism, no locking mechanism is needed as the hydraulics also does this already. The only requirement for the hydraulics is that the pressure should remain constant.

Finally, a hinge should be created. For the hinge three solutions were created: A top mounted hinge, a flush mounted hinge and a 360 hinge. The top mounted and the flush mounted hinge are more or less the same. They both have interlocking teeth that are connected with a pin. the only difference is that the top mounted hinge is placed on top of the boom and the flush mounted hinge is incorporated within the top of the boom.

The lines in the morphological diagram represent the three different concepts that have been chosen. Also, per concept, the general overview is shown in the first picture, and the accompanying sketches of the three concepts are shown in Appendix A.1, Appendix A.2 and Appendix A.3 respectively.

## 3.3 Concept with an I-beam and a Rope Lifting Mechanism

### 3.3.1 Drawings

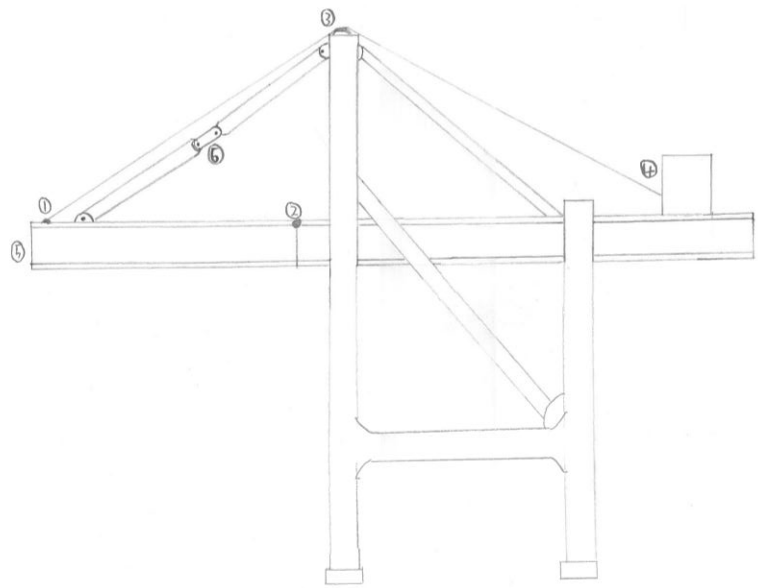

Figure 2: Overview of the I-beam Concept

### 3.3.2 Explanation

This concept works with several ropes that are spun from the boom over the main pillar onto the back of the crane. The boom can be lifted by using a winch to retract the ropes.

The support between the boom and the main pillar is created by two beams and a plate in the middle with pins to connect them. By using pins the beams can move freely so that when the ropes are retracted and the boom rises the support beams can be folded in.

The hinge used is a flush mounted hinge which means it is integrated into the beam itself. The beams used are I beams.

For locking the boom in place a ratchet mechanism is used that is attached to the axis of the winch.

### 3.3.3 Pros and Cons

Pros:

- The locking mechanisms lock the boom in place during every step the boom makes while lifting which prevents it from falling back down during the lifting itself and not just when the boom is already lifted;
- The folding mechanism makes it so that the boom can be lifted the highest of the concepts.

Cons:

- The solid I-beam makes for a heavy boom compared to the other boom frames;
- The rope is only supported from the bottom which means that the rope can be dislocated quite easily when lifting the boom.

## 3.4 Concept with a Hollow Beam and a Hydraulic Lifting Mechanism

This concept uses a hydraulic lifting mechanism to lift up the boom. The frame of this concept is hollow with a couple of cross-sectional support beams. It does not need a locking mechanism or lift support since the hydraulic lifting mechanism is capable of fulfilling those functions. The hinge used in this concept can rotate almost 360 degrees.

### 3.4.1 Drawings

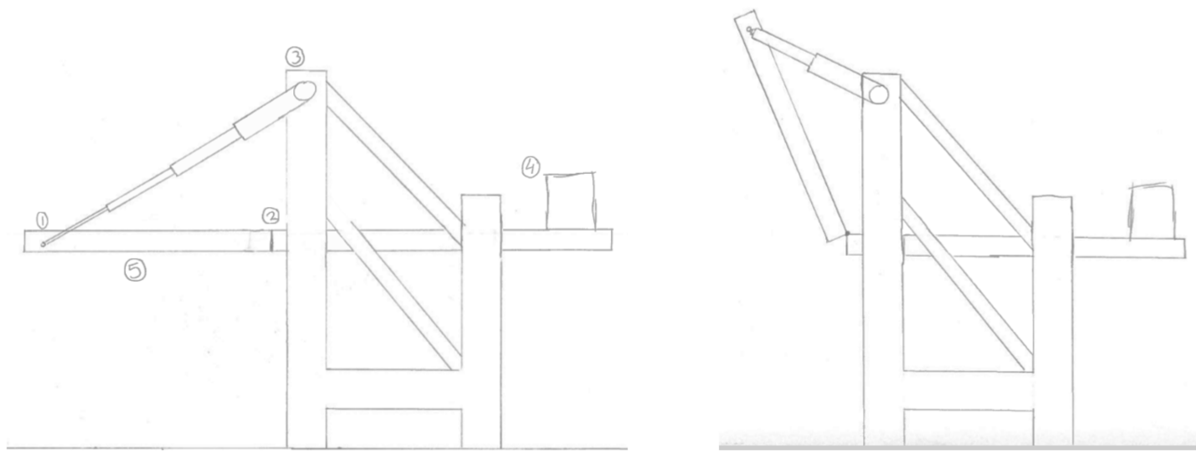


Figure 3: Side views whole crane: left with boom horizontal, right with boom lifted

### 3.4.2 Explanation

From Figure 3 it can be seen what the concept looks like when the boom is horizontal and when the boom is lifted. Numbers are placed at important points of the crane. These points are elaborated in more detail in Figure 68, 69 and 70.

In Figure 68 the hinge is drawn. In each beam, there are two holes and the left and right part of the boom are connected by a pin. In Figure 69 the connection with the hydraulic arm and the boom and frame is sketched. The arm is connected with the boom and frame by a pin. Figure 70 shows the boom. The motor at point 4 has to make sure that the pressure can be changed easily to let the hydraulic arm function.

### 3.4.3 Pros and Cons

Pros:

- No boom support is needed since the hydraulic arm is capable of fulfilling that function;
- The boom weighs a lot compared to other structures.

Cons:

- The hinge can rotate more than 180 degrees, which can cause that the boom will not be perfectly horizontal when lifted down;
- Hydraulic lifting has a much lower speed than lifting with a rope.

## 3.5 Concept with a Lattice Construction Beam and Rope Lifting Mechanism

The lifting mechanism uses a wire rope to lift the boom. The frame of this concept consists of two hollow frames that are connected with beams that are placed in triangles.

### 3.5.1 Drawings

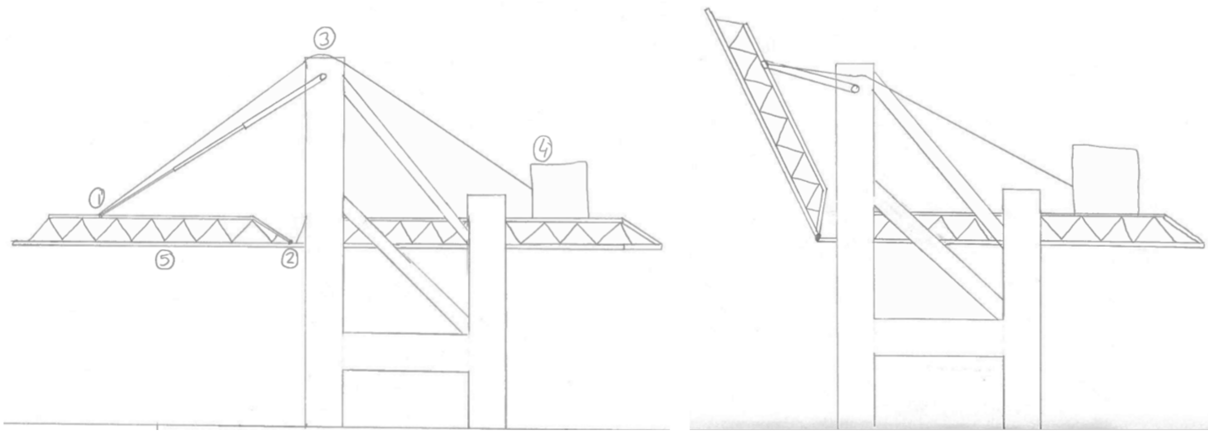


Figure 4: Side views: left with beam horizontal, right with beam lifted

### 3.5.2 Explanation

From Figure 4 it can be seen how the concept looks like when the boom is horizontal and when the boom is lifted. Numbers are placed at important points of the crane. These points are elaborated in more detail in Figure 71, 72, 73, 74 and 75.

This concept uses a single beam lattice construction for the boom, see Figure 75. A top mount hinge is used as can be seen in Figure 72. The rope is connected to the boom like in Figure 71 and the rope is being led by two wheels at the highest point of the crane, as seen in Figure 73. The two wheels make sure that when the boom is almost lifted vertically, the rope is still being led by a wheel (the situation with the red coloured rope). The locking mechanism uses a pin that is pushed through the rope winder like in Figure 74.

### 3.5.3 Pros and Cons

Pros:

- The boom has a reduced wind area and lighter overall construction;
- The hinge cannot rotate more than 180 degrees, so the boom will always be perfectly horizontal positioned.

Cons:

- The hinge does not allow the trolley to move on top of the boom;
- The hole for the pin on the rope winder has to be perfectly aligned with the position of the pin. Otherwise, the pin cannot be pushed through.

## 3.6 Concept Choice

For the final boom concept, a combination between the I-beam and the lattice construction beam concept has been chosen. It will have a lattice construction beam and a two-wheel rope support from the lattice concept and a ratchet locking mechanism, folding boom support and a flush mounted hinge from the I-beam concept. The lifting mechanism is the same for both which is the rope mechanism.

For the frame, a single beam lattice construction frame was chosen because it ensured that the forces put on the boom by the weight of the containers are divided more equally over the complete structure while remaining light compared to the I-beam structure.
The two-wheel rope support was not held into account in the morphological diagram but it is still important when using a rope mechanism for lifting to keep the ropes in place.
The ratchet locking mechanism is the safest option of the ones available because it makes it so that the boom will not be able to move downwards during the lifting process even during loss of power because of the pawl that prevents the wheel from moving other way and thus the winch as well.
For the boom support, the folding beams are chosen because they make for the furthest possible lifting of the boom as less space is needed between the main pillar of the crane and the boom for the folding beams compared to the telescoping arm and the hydraulic mechanism.
What type of hinge is used is not that important as long as it is positioned at the top of the boom so the trolley can move along the bottom of the beam without any obstructions. The flush mounted hinge was chosen.

# 4 Trolley

## 4.1 Functionality

The main purpose of the trolley is to move the hoist over the length of the boom. When moving the trolley along the boom, the hoist should be stable and not swing too much, which could be dangerous for the attached containers. Two concepts for the trolley were made according to the different designs of the boom. There are two possible concepts explained below: one for a solid I-beam and one for two separate I-beams.

## 4.2 Morphological Diagram

Possible mechanisms for the trolley are contained in the morphological diagram, shown in Figure 5. Some of the mechanisms are more reliable than the other. As a result, concepts for a trolley can be derived according to the paths shown in the figure below. An explanation for the two choices that are indicated with the red and green line are given on the next page.

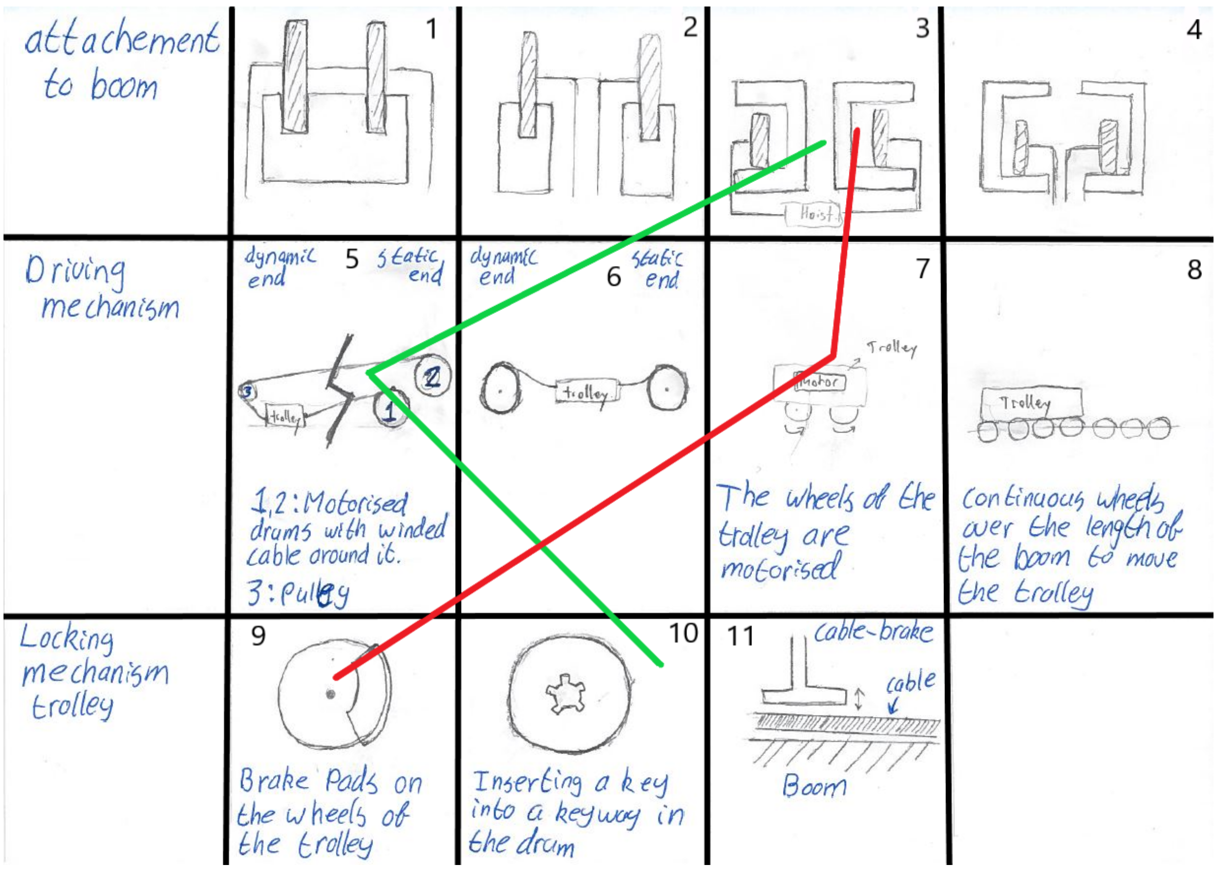


Figure 5: Morphological diagram

In the first row of the morphological scheme, different possibilities of attaching the trolley to the boom are shown. The first option is to put the trolley on top of an I-beam. This can be done in two different ways. The first way would be with one solid I-beam, with two wheels in each side of the gaps of the beam, shown in cell 1 of the morphological scheme. This would, however, cause a problem for the hoist, as it has no place to attach to the trolley. Another option would be with two separate I-beams, where the wheels are attached like shown in cell 2. This would give a possibility for the hoist to be attached between the beams. This seems like a good option, but when constructing the boom with to separate I-beams triangles need to be attached between them for extra strength. This means there is no continuous gap between the beams, preventing the hoist to be able to be attached to the trolley and move along the boom. The second option (cell 3) would be putting the trolley underneath the I-beams, preventing the problem that the first option would have: The triangles will not be in the way of the trolley. Here, the wheels are attached on the outside of the I-beams and the trolley will be attached underneath. Cell 4 shows another option of hanging the trolley underneath the boom, but this time, rather than on the outside of the I-beams, the wheels will be attached on the inside. This would be a possible option, but it will be less strong than the one in cell 3 as it may bend inwards due to forces pushing it down. This concludes that option 3 is probably the best one for the final concept.

For the driving mechanisms, there is the option of attaching cables to the trolley, and moving them using motorized drums. This again gives rise to two different possibilities. The first option, cell 5, would be having two drums with cables on each of the two I-beams on the static part of the boom. On the dynamic part, a pulley will then be attached over which one of the cables moves. This cable is then attached to the front of the trolley. The other cable would be attached to the backside of the trolley. In this way, the trolley can be pulled forward and backwards. Another option would be having one motorized drum on the static part of one beam and another one on the dynamic part. This is then done for both beams again. In this way, less cable is needed and the pulley at the end of the beam can be omitted as well. This is shown in cell 6. It may cause a problem though as the drums need to be synchronized to keep the cables connected to the trolley in tension. When both drums are on opposite sides of the boom, synchronizing them is harder than when they are next to each other. Therefore, the first option might be better. Another option, cell 7, would be without cables but with motorized wheels on the trolley itself. In this case, the trolley is moved immediately by the wheels. The last option, cell 8, is attaching wheels to the boom rather than on the trolley. The trolley will have rails in this case and is pushed by the turning wheels on the boom. This option is relatively difficult and the wheels on the boom all need to be synchronized. Consequently, the most applicable options would be the concepts in cell 5 and 7.

Furthermore, a locking mechanism needs to be designed to stop the trolley from moving further even though no extra power is put out anymore. This can be done by attaching a brake to the wheels of the trolley, shown in cell 9. This is a simple, but probably effective way of stopping the trolley. Another possible solution is making a keyway in the drums and putting in a key to prevent them from moving, cell 10. This is of course, only possible when cables, and therefore drums, are used. The last designing option, cell 11, would be by having a cable break, meaning pushing something against the cable to prevent it from moving. This option, however, is not very suitable as it will damage the cable and the boom quickly. This means option 9 and 10 are the best options.
Combining the best options of all three of the designing parts gives two different possible trolley concepts. These are shown in the morphological scheme as a red and a green line.

## 4.3 Trolley Underneath one I-beam, Driven with Motors on The Wheels

### 4.3.1 Drawings

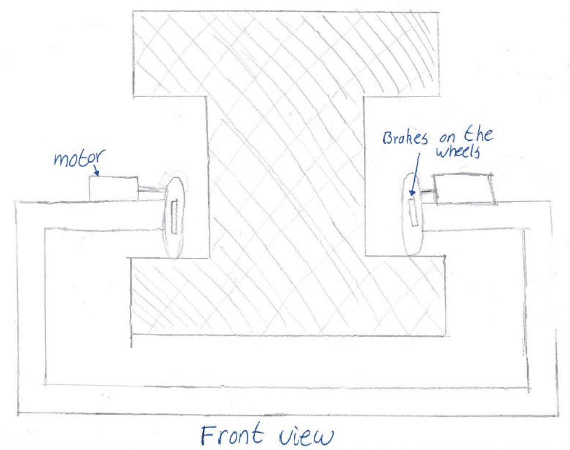


Figure 6: Concept 1, Front view

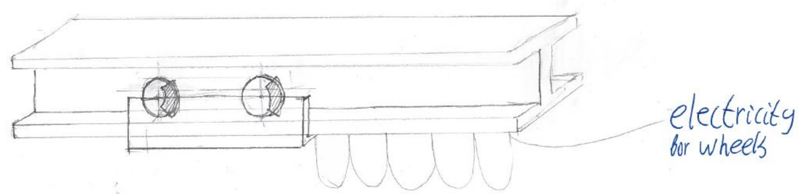


Figure 7: Concept 1, 3D-view for electricity

### 4.3.2 Explanation

As can be seen from Figure 7, there are four vertical wheels that are mounted in between the I-beam. The platform is located under the beam, the hoist will be attached to this platform for lifting purposes. The driving mechanism of the trolley is a motor and it can be attached to either one of the side or both sides of the trolley. In Figure 6, the possibility of attaching a motor to both of the sides is shown.
In Figure 7, brakes can also be seen on the wheels. These can either be disc brakes or brake pads. This will be chosen later in the design stage if this the chosen concept.

### 4.3.3 Pros and Cons

Pros

- The braking blocks are a simple and efficient way to stop the movement of the trolley;
- The trolley can still operate at a reduced speed if one motor happens to fail;
- The movement of the trolley is accurate as all wheels are directly driven with individual braking systems.

Cons

- The electricity cables may be damaged easily as they are subjected to various weather conditions.

## 4.4 Trolley Underneath two Beams, Driven by Cables

### 4.4.1 Drawings

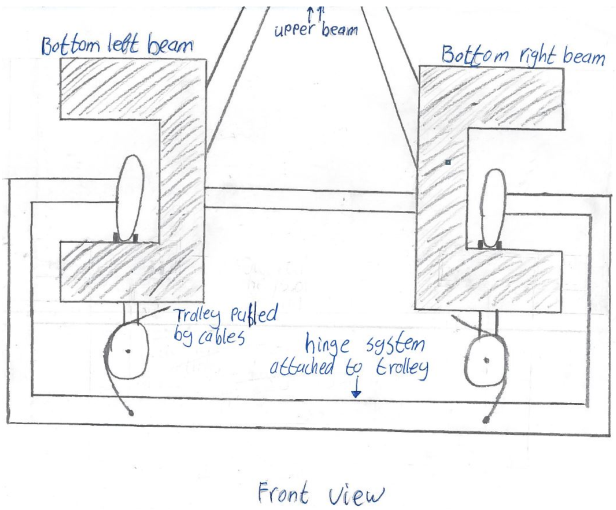


Figure 8: Concept 2, Front view

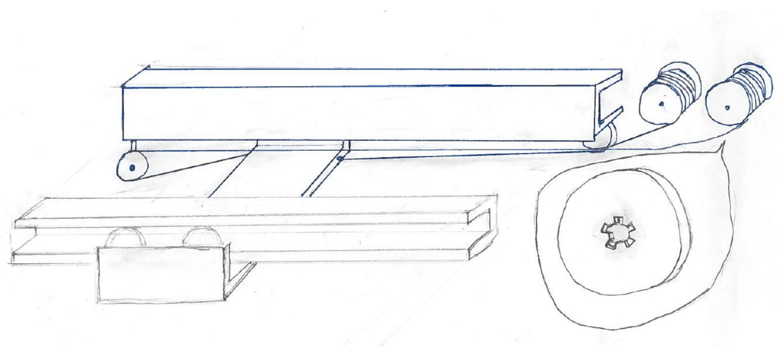

Figure 9: Concept 2, 3D-view for cables

### 4.4.2 Explanation

The trolley is attached to the two lower beams of the boom. This is done by putting rails in both of the sides of the beams, on which the trolley will ride. This can be seen in Figure 8.
As shown in Figure 9 there is lots of room for the hoisting system to be attached to the trolley. As the trolley hangs below the beams, it will also not interfere with the triangles that are incorporated in this boom design.
In Figure 9 the cable management is also shown for this concept. There will be two drums at the end of each of the two beams. The first one of the two cables goes below the beam and is attached to the front of the trolley, as shown in Figure 8. The other cable also goes underneath the beam via the pulley and is then attached to the back of the trolley. This way the trolley can easily go forward and backwards.
The stopping mechanism is shown in the zoom-in in Figure 9. It is a key way in which a key will be inserted to stop the movement of the drum when the trolley has to be stopped.

### 4.4.3 Pros and cons

Pros

- There are no electricity cables on the boom;
- Not much maintenance is needed.

Cons

- The two drums on both of the beams have to be synchronized;
- When the trolley has to be stopped, the motor has to be stopped first, and then turn a bit back or further to fit the key in;
- Maintenance is harder to carry out on the motor as there is a whole drum around it.

## 4.5 Concept Choice

As described in the above sections, the pros and cons were determined for both concepts. The pros and cons of the first concept seem to outweigh the pros and cons of the second concept. The first concept, the one with motorized wheels, can be operated easier than the concept in which the trolley is driven by cables, and is, therefore, more accurate.
Besides that, if a motor were to malfunction, this would not cause as big of a problem as for the cable design. The motorized trolley has at least two motors, meaning that one motor can still do the job at a lower operating speed if the other one falls out. If this would happen to one of the motors of the second concept, this could have bigger implications. As the motors are synchronized, one of them stalling could mean that either the function for driving forward or backward will not work anymore, which is quite a big problem.
The stopping mechanism for both of the concepts is also a point of notion. As all of the wheels on the first concept have separate breaks, the trolley can be stopped very fast and very accurate. The second option, on the other hand, has some tolerance in the locking of the motors, as the keyway in the drum has a certain number of slots. This means that when the trolley is stopped, it could happen that the motor would have to turn a bit backwards or forward to make sure the key fits in the keyway. Turning the motor again after having stopped the trolley, is something that should be avoided.

From all of the above, it is therefore concluded that the motorized trolley concept is chosen. This does not necessarily mean that it has to be installed on one single I-beam, as having two separate I-beams does not change anything in the design of the trolley.

# 5 Hoist

## 5.1 Functionality

The hoist is a mechanical device with the main purpose of lifting and lowering heavy loads vertically. The loads, in this case, are containers with products inside. To power the hoist the use of an electric motor is mostly used. When lowering or lifting the container the movement must be smooth and stable, which means: the forces in the spreader must be equalized, even when a strong wind hits the lateral sides of the container or the loads are not equally distributed inside the container.

## 5.2 Morphological Diagram

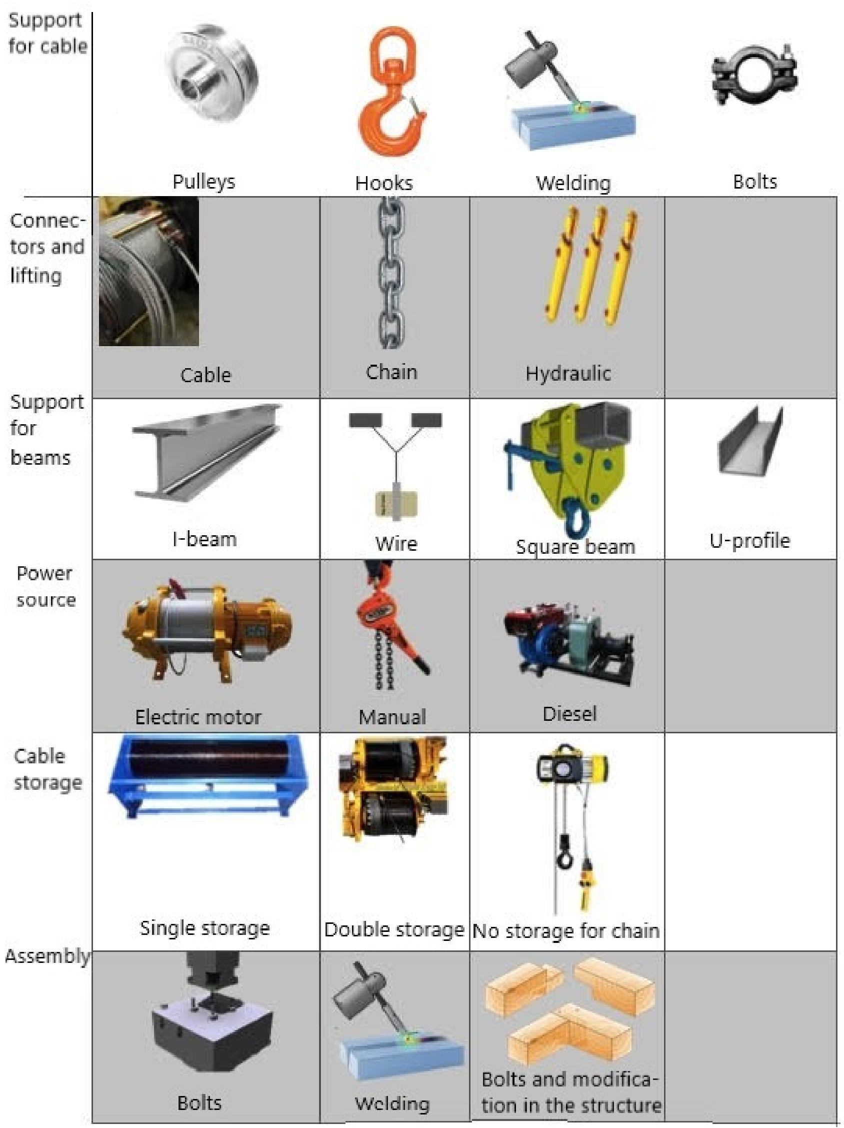


Figure 10: Morphological diagram

## 5.3 Choosing the Right Concept

**Support for cables**

To use pulleys cables are necessary, specifically in U shape of a cable at the end. This means the two endings of a cable are connected to the same section, in this case, a motor or other mechanics to pull the cable and lift the container. The use of a pulley has the benefit that if an undesired movement horizontally occurs, the free movement of the cable does not cause stresses in the endings that connect the cable to the supports. Also, another benefit is that the contact area of the cable with the pulley is bigger if just the endings are connected to the support, as for example for endings welded to the supports, or the use of bolts to attach the cable to the support. If the use of hooks is required, the endings of the cable should be either welded or attached with a clamping device. This means the stress from the load is applied to the weldings or the bolts. This situation must be prevented because welding could cause internal stresses that lead to failure, and the clamping could slip if the cross-sectional area is reduced due to elongation or the shape of the cable could flatten because of the perpendicular forces. This problem is solved if pulleys are used instead. The clamping mechanism could also be attached directly to the beam, but again there would be too much freedom of movement for horizontal forces. In the case of welding the wire directly to the beam, this does not give freedom to the cable to move in case of horizontal movement. These forces add extra stress to one ending of the wire and this could cause the wire to have a shorter working life.

**Connectors and lifting**

For cranes, it is common to use either cables or chains as connector lines between the power source and the load. In this design the use of hydraulic connectors is not beneficial, because the distance they have to move the loading is 40 meter, there is not enough space in the trolley to storage the tubes for the hydraulic connectors. Between choosing either chains or metal wires, the second is more commonly used for this design due to safety reasons. If one of the connections is about to fail there is warning that that is happening. And if it fails the chains break apart completely and this could cause the other support to break as well, because higher stresses concentrated in the remaining wires. On the other hand, with metals cables, there is a warning when failure is about to happen. Metal wires are made of smaller with many thinner metal wires. When failure is starting it is visible that the small wires are breaking apart on the surface. Another reason to choose metal wire instead of chains is the shape of each option and how good it allows its storage while adding as less weight as possible to the crane. Chains are not coiled in a shaft, because the connection in the chain is usually parallel to each other, during coiling, this connection can deform and make cracks in the surface to subsequently fail. With cylindrical metal wires, coiling does not affect the overall shape of the wire.

**Support for beams**

This part of the Hoist is where most of the forces are concentrated, therefore a good material should be selected and some options are analyzed. I Beam is the most used for industrial designs because the shape of this material is excellent at distributing the forces applied to it, but there are more possible options, for instance, wires, in some cases are used as supports but in this case the wires are not a good option because this should support more than two elements and the shape of the wire makes this very complicated, also those cables could bend causing problems for the entire system. The square beam is another good option, pretty similar to I Beam but in this case, there is more surface useful where supports can be attached, there is another advantage with this square beam, it is easy to mount different types of components or extra supports. The last option is U profile which is almost half of the square beam, but due to the inaccessibility to attach this component to the hoist it is not recommended to use in this case. In general, the support should be strong enough and be able to withstand most of the weight.

**Power Source**

To pull the connections some sort of on power source is needed, to choose the correct one the maximum weight and dimensions must be considered. For lifting small objects or even a car, manual cranes can be used because of the relatively short distances. But for STS crane this is not a good option, because firstly it would be dangerous for a worker to operate the cable at the top of the trolley, and more than one handle and operators would be needed to lift the weight at the specified requirements (weight and speed). Now between fuel engines and electric motors, the first ones take more spaced and need a constant supply of fuel (diesel). To transport the fuel pipes must be used, these pipes can have unexpected leaks, that could cause the starting of the fire. On the other hand, electric motors are used in these applications, because the size is smaller than that of a diesel motor and electric cables are safe to operate at high operating devices. Regarding the size of both options, for diesel engines, it is also necessary to have a set of gears to adjust the speed to that of the requirements. These gears not just take more spaces but also add extra weight to the boom. An electric motor can be adjusted to the speed required without any other components like gears or extra shafts.

**Storage for cables or chains**

To lift any load the connectors must be pulled. If the connectors are metal wires then they can be coiled in a shaft connected to the power source (mostly electric motors). In the shaft more than one wire can be attached, then the process of coiling can happen in different sections (one for each attached ending wire). This ensures the wire is equally distributed and the movement up and down stays stable. If there is not a section for each ending of wire the soiling process will be disorganized. A disorganized coiling process may cause excessive friction or unexpected issues. Chains are not usually coiled because of the reason explained in the section of connections lines, see above. For chains usually, the endings that are connected to the power source after it has been pulled, it is left free to hang out. Because the chain needs to be long enough to reach the spreader form the ground, the same length of chain would be hanging out when the load is at the high of the crane. This hanging section could cause different problems, such as: get stuck between somewhere in the spreader or other sections of the crane.

**Assembly**

To assemble the motor, the shafts, the supports for pulleys and any other components, the use of different joining mechanisms can be used. The components can be all welded, this could save the process of making holes and other adaptations for putting bolts. The disadvantage is that for maintenance the components must be disassembled. To do this the welding must be broken and bracers and heat. These processes do not just take longer times to be made, but also put risk in the general reliability of the crane. In case the joining mechanism is chosen to be bolts and easy disassembling components, the detailed design of the parts must be adapted and the number of bolts and their dimensions must be chosen carefully. This could take more time in the design process, however, if future maintenance is required, the component can be removed and relocated relatively easy. In the section of the assembly, the last option refers to adapt the shape in the areas of contact, for the components to be joined, in such a way no bolts, welding or other extra component are needed. This process is mostly used for wood structures because the wood itself is easier to give the required shape. However, for metals, this joining technique is usually not chosen due to the difficulty to shape metal. The hardness of metals would require more tools and time to make the complete structure in this way.

**Final elections**

- Support cables: pulleys;
- Connections and lifting mechanism: metal wires;
- Support for beams: I-beams;
- Power source: Electric motor;
- Cable storage: Section for each cable;
- Assembly: Bolts and easy assembly components.

## 5.4 Concept with one Support and two Motors

As can be seen in Figure 11, the concept works with two electric motors A1 and A2 at each side of the hoist perpendicular to the boom. In the middle of the two motors, there is a support C in which two pulleys are attached. To lift and lower the load, two cables are used. Both endings of the cables are connected to one motor, this cable passes through two pulleys in the spreader and then the other 'U' endings of the cable are placed over the pulley in the support C. Also, notice that the motors rotate in opposite directions to lift or lower the loads.

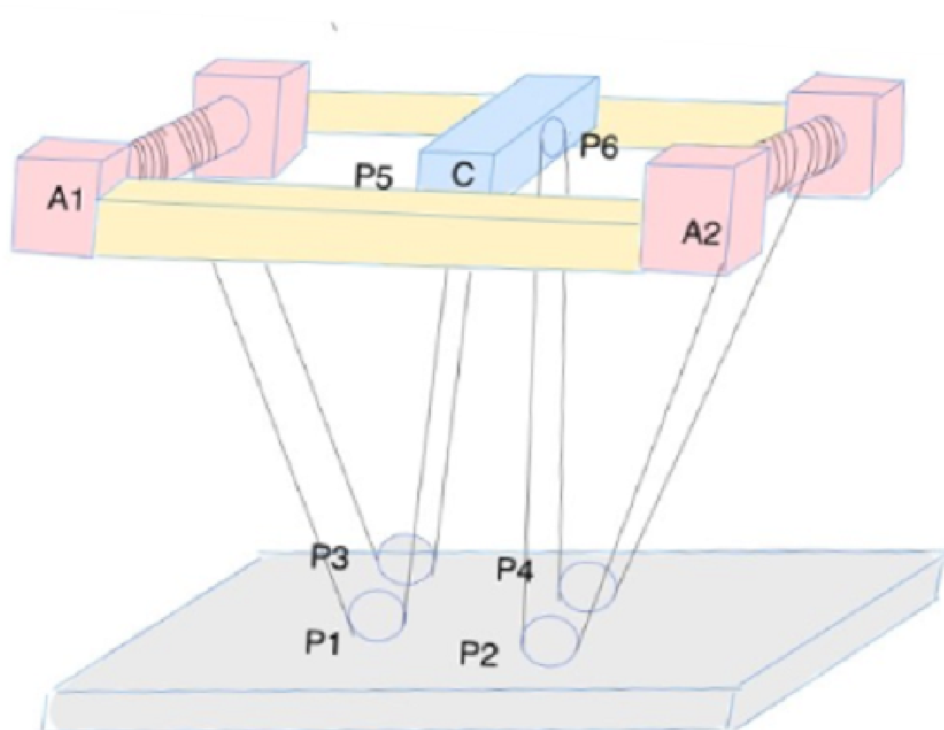


Figure 11: Concept 1, 3D view

### 5.4.1 Advantages

- Because the load of the container is distributed to two motors instead of one, each motor can be smaller and requires less power to operate;
- The load is well-distributed over the eight sections of the two cables;
- When the cable is coiled, it is distributed in the two motors, taking less space than in a one motor crane;
- Wires move in the same direction at the same time, which keeps forces well-distributed, especially in the wheels of the trolley.

### 5.4.2 Disadvantages

- The two motors must always be synchronized to ensure the container is stable during transportation;
- Support C (Figure 11) is loaded with four sections of the cables, which makes the support susceptible to overloading;
- The dimensions of the trolley must be longer than one motor design, which adds stresses to the boom.

## 5.5 Concept with two Supports and one Motor

This concept works with only one motor. The four endings of two cables start from this motor. Two supports are located in each side of the hoist, where the cables are supported on the pulleys. This brings stability to the container that will be lifted and lowered. The load of the container is distributed in eight sections that are well-distributed over the container spreader.

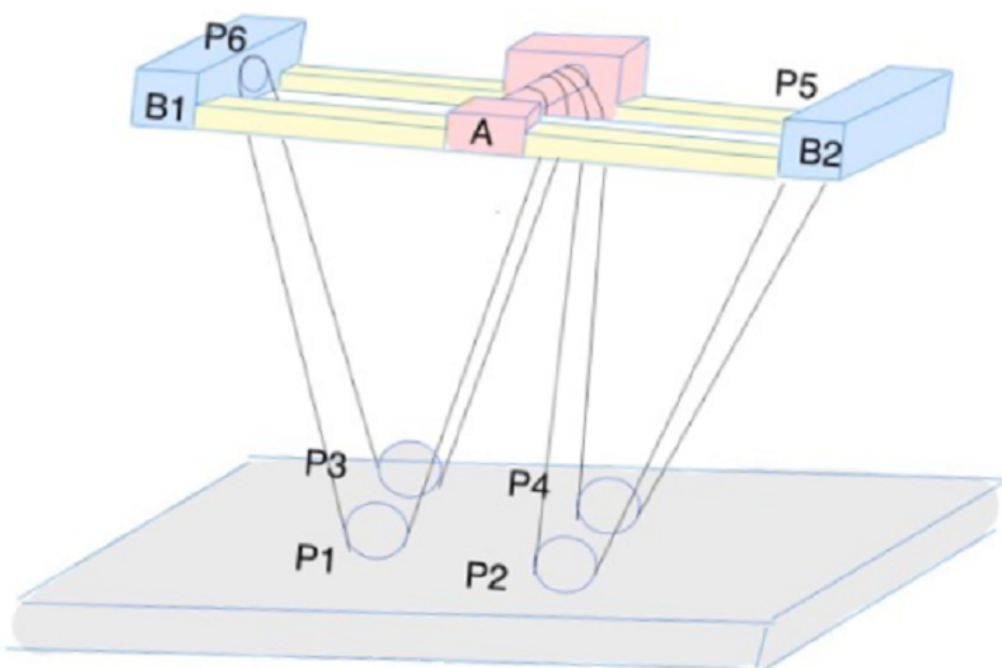


Figure 12: Concept 2, 3D view

### 5.5.1 Advantages

- The eight sections ensure the container to be stable during the operation;
- Only one motor is required and movements are synchronized.

### 5.5.2 Disadvantages

- The motor could be overloaded and get stuck, and this requires more space and power;
- The motor section, where the cables are coiled requires four sections for the ending of the cables which use more spaces than other concepts operation with two motors. This can cause excessive friction or undesired extra stresses because the wires must be coiled tighter;
- Supports may have to be aligned with the coiling section of their cable, instead of being lined with the other support. This can cause not well-distributed forces on the trolley and the components that make movement possible.

## 5.6 Concept Choice

To choose a final concept it must be considered which one gives more reliability and efficiency during operations. Advantages and disadvantages are also taken into account and the Morphological diagram gives more than one solution to different functions of the hoist, with all this information, Concept 1 with two motors and one support will be the most suitable option, most of the disadvantages of this concept could be solved with making the right choice in the final design. For instance, the support C in Figure 12, can be split into two parallel sections, this decreases the stresses in the beam and the load is distributed in two beams, therefore failure is less likely to happen. The final material selection will also play a big role in this decision making, which is all shown in Chapter 9. Synchronization of the motor to always lift and lower at the same time is a matter of programming and electric engineering related, which is not treated in this paper, however, with help of software, it is indeed possible to keep both motors synchronized. The variations on size are not very significant, the concept with two motors will take a bit more space and add a bit of extra weight than the one motor concept. Overall, the benefits of this power source make operations easier and reliable.

## 5.7 Reasons for Dimensions

### 5.7.1 Coiling Shaft

- Length = 2 m
- Diameter = 0.3 m
- Middle section separator diameter = 0.32 m

The length of 2 meters and diameter 30 cm of the shaft is because then the metal wire of diameter 2 cm can be coiled without overlapping over itself.

### 5.7.2 Support for Pulleys

- Length = 2m
- Width = 1 m
- Height = 0.3 m

Because the length of the shaft is two meters, then the support must be the same length as well. The width of 1 m is to pair the dimensions of the beams perpendicular to the boom, where the motor and support rest. Then the pulley can the attachment of the pulleys can be 30 cm, this gives a good gripping area for the pulleys.

#### 5.7.3 Beams Perpendicular to Boom

- Length = 7.5 m
- Width = 1 m
- Height = 0.3 m

These beams form the structure of the trolley in which the hoist components rest; the height of 30 cm is chosen for giving strength to the structure. From there the width of the beam is chosen to be one meter because the motor and the control systems take at least one meter.

# 6 Connections Part by Part

The different parts of the crane should, of course, all be connected to one another. This can be done by using either riveted or welded joints. The wanted joints may vary per part, so every part will be discussed separately.
For determining which part needs which joint, Figure 13 is used. In point A hinged connections are used to the beams that go to part D, as discussed in section 8.1. This also goes for the beam going from B to E. The beams that go from A to H need to be welded on either A or H and need a hinge on the other side. In that way, the beams will not buckle due to thermal expansion as they have some clearance. The same goes for the beams between B and M and between B and G. The other side of the beam will be connected by welding it to the structure, as welding gives a more stable connection than hinges.
The possible danger with welding is the formation of $Cr_{23}C_6$. Chrome is used in the material to keep it from corroding and the carbon is added for extra strength. When welding, however, the parts next to the weld heat up and the carbon and chrome can react together, forming big molecules in the grain boundaries, also known as inter-granular corrosion. This will make the material susceptible to both corrosion and weaker parts. Even though this is a frequent problem when welding metals, due to the low carbon content of the chosen S420 steel, the amount of inter-granular corrosion will be minimal.
As there might still be a small amount of corrosion, the connections of G, M and H are welded, as they have more support from the boom so the small amount of corrosion will affect those parts less. A and B will be hinged to the main structure. At I there will be another hinged connection for the beam from I and J and this beam will be welded at J again.
Furthermore, crevice corrosion can take place between little crevices between the parts of the hinge when it has been raining. By placing the hinges on the vertical beam, less water will stay on the connections and the crevice corrosion will be minimized.

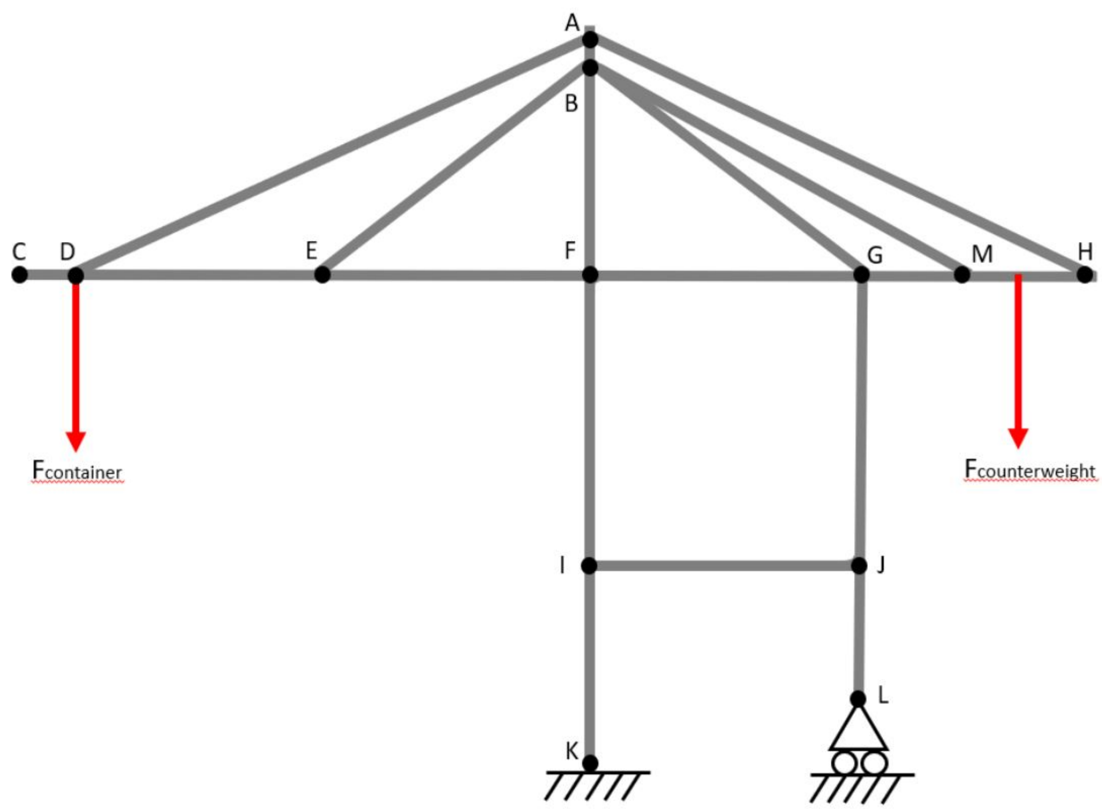


Figure 13: Schematic structure STS crane

# 7 Dimensioning

In this chapter, several initial dimension choices are explained. It is important to understand that most of the dimensions that are chosen for the beams, will be optimized in Chapter 10 using the FEM-package. First, however, after the initial dimensions are selected, the material selection will take place, which is done in Chapter 9.

## 7.1 Boom

### 7.1.1 I-beam

All the dimensions of the I-beams in millimetres can be seen in Figure 14. Also, the dynamic part of the boom is 47 meters long. This part is defined as the two I-beams that are able to be lifted and to which the trolley is connected in operating mode. The initial dimensions are assumed and may be changed in Chapter 10, where the FEM-package is used to optimize the dimensions of all used beams.
There is a distance of 5 meters between both tops of the I-beams. Between both beams triangles are attached, which can be seen in Figure 15 The triangles are 20 cm thick and high and 595 cm long. This is calculated by taking the cosine of the angle between the triangles, which is 45 degrees, and the absolute length between the middle of the I-beams, 550 cm (5 meters + 2·25 cm).

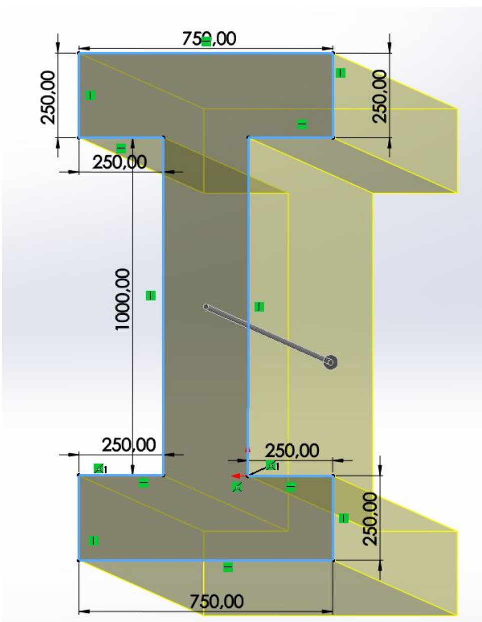


Figure 14: Dimensions I-beam

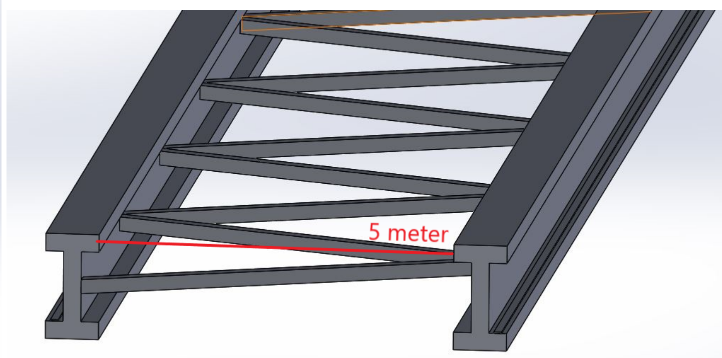


Figure 15: Position of the triangles between the two I-beams

### 7.1.2 Boom Supports

To support the boom when the crane is in operating mode, two times two supports are used. These supports exist out of two beams which are then attached to each other with a hinge, as can be seen in Figure 16. The longest of the two supports is attached at a distance of 24.795 m from the surface of the I-beam. It has this specific measurement as the diameter of the end of the beam is set at 410 mm (so the radius is 205 mm), adding the radius to the attachment height makes it that the highest point of this support is located at 25 m. Besides the height, there is also the distance from the attachment point to the end of the boom where the hinge point is. This distance is equal to 44.795 m. Combining those two distances, the length of the support is determined to be 51.48 m. This length is split up into three separate lengths. Two of those will be for the two separate beams, being 36 m and 14.98 m. The remaining 0.5 meters is taken by the hinge which connects the two beams.

The same procedure is done for the small support, which is located 2 m lower vertically, and horizontally the support is located 25 meters more to the right. This makes the total length of the small support equal to 30.48 m. All these dimensions can be seen in the schematic sketch in Figure 17.

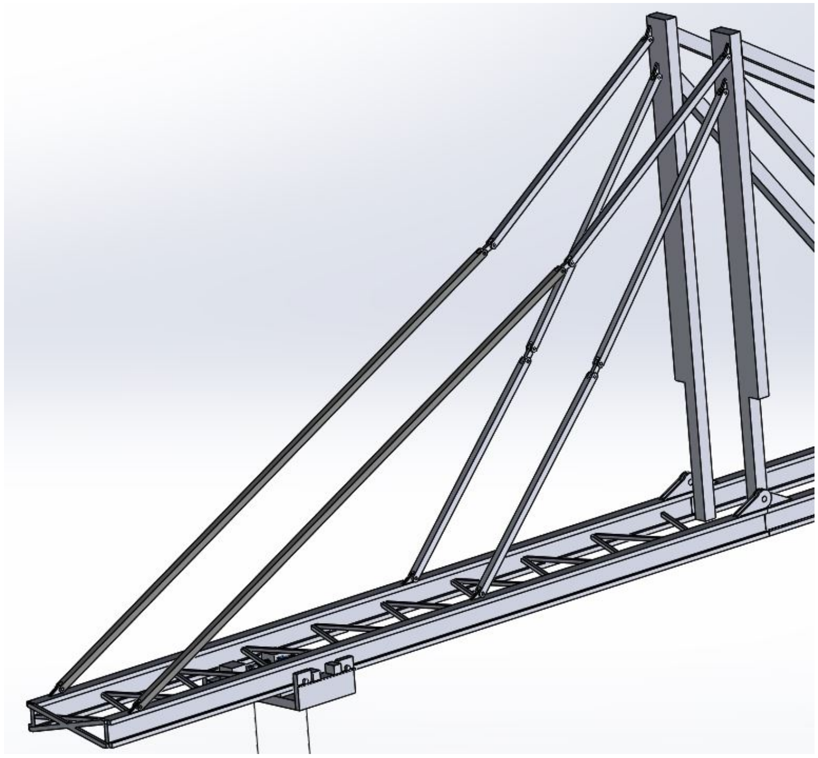

Figure 16: Support of the boom

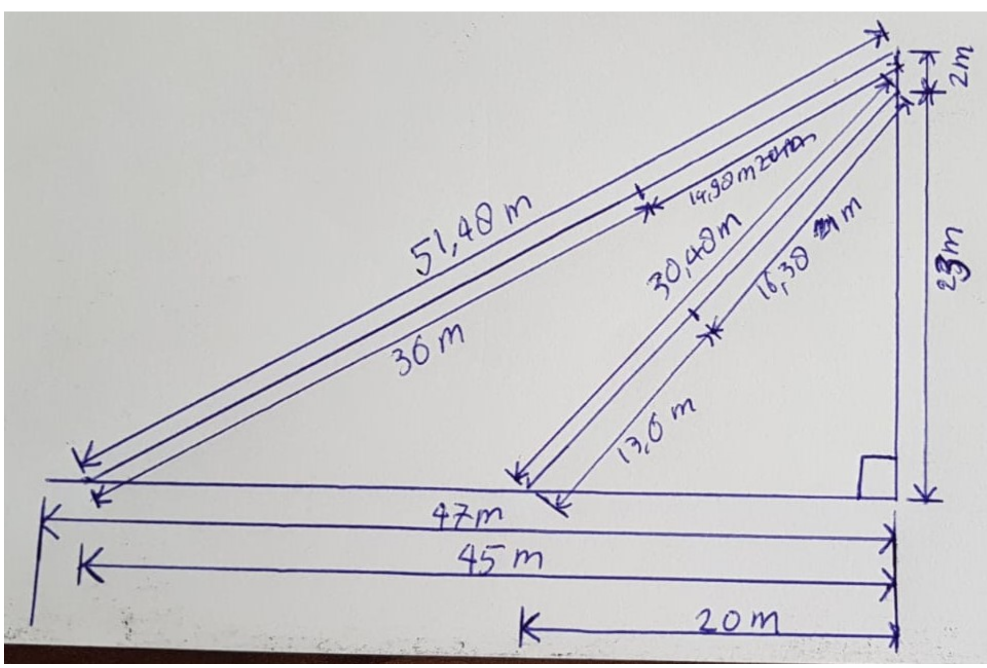


Figure 17: Schematic sketch of the crane with dimensions

Next, there will be zoomed in on the dimensions of one of the supporting beams. In the three figures, it can be seen on which part is zoomed in. In Figure 18, the dimensions of the height and width (400 mm) and the radius of the connection at the end (205 mm) are shown. These dimensions are again assumed and might be altered if the calculations in Chapter 10 show that those dimensions are not big enough. Besides height and width being optimized, materialization will also be done for all the parts of the frame in Chapter 9. At the end of the beam, the hinge is attached, which is shown in Figure 19. The end of the hinge fits exactly in the hole of the supporting beam. The straight part in the middle of the hinge ensures that the supporting beams can fold inward enough, which happens when the boom is lifted. The folding mechanism of the supporting beams is displayed in Figure 20.

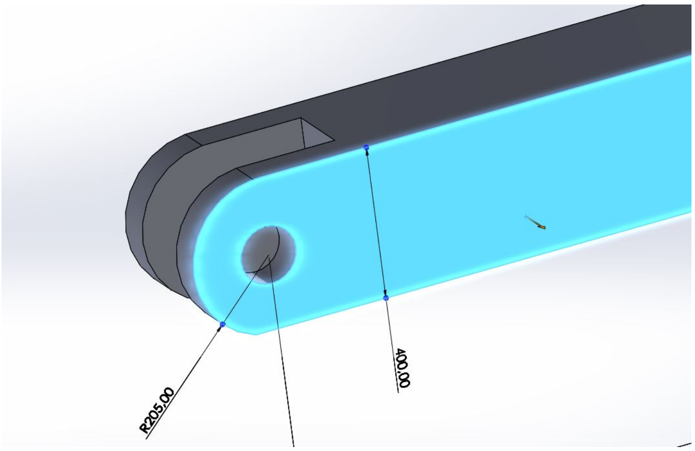


Figure 18: Part of a supporting beam

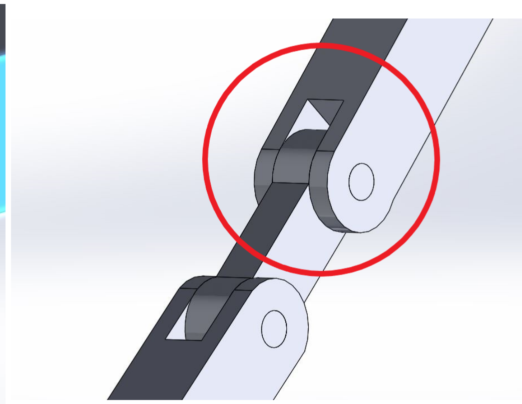

Figure 19: The hinge system in the supporting beams

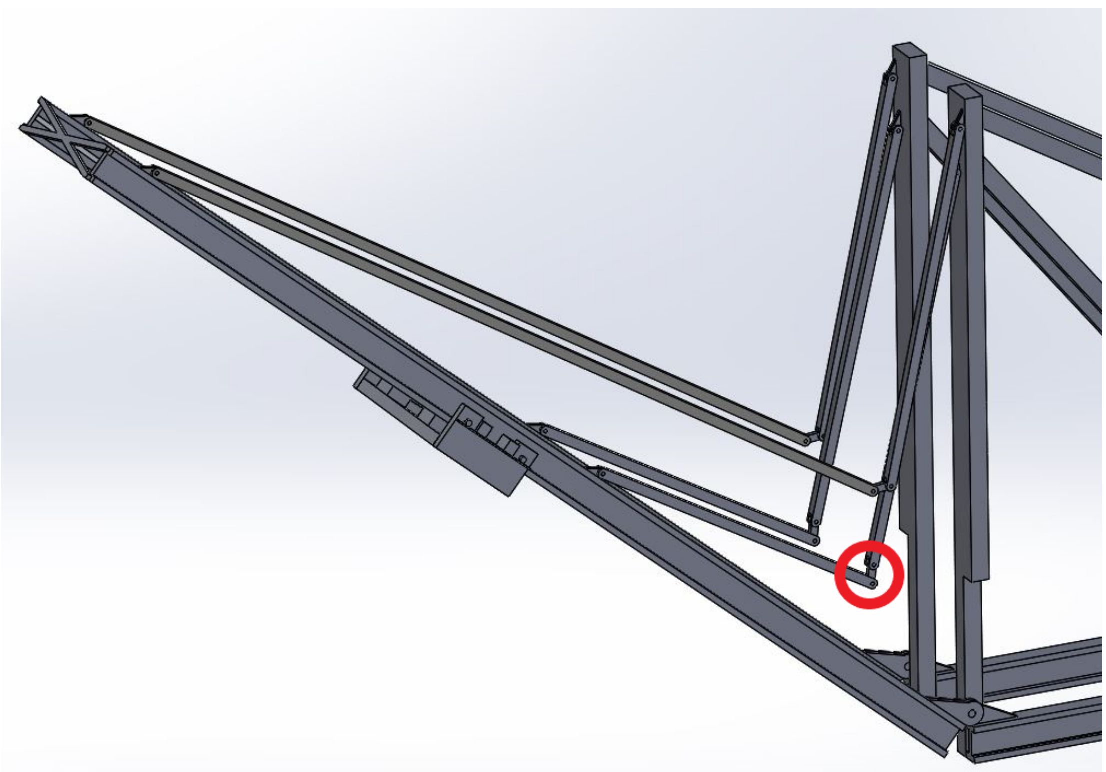

Figure 20: The supporting beams folding inwards when the boom is the lifted

## 7.2 Hoist

### 7.2.1 Drum

The maximum lifting height of the hoist is equal to 40 meters. The chosen concept for the hoist has a cable running from the drum to the container to the support and back. The drum will be split up into two separate parts, meaning that each side has to wind up a cable length of 80 meters. An explanatory picture is shown in Figure 21. Cable parts 1 and 3 will be wound up on one side of the drum, and cable parts 2 and 4 will be wound up on the other side. The total length of the cable is therefore equal to approximately 160 meters (a little bit extra length is needed for connecting the cable to the drum etc).

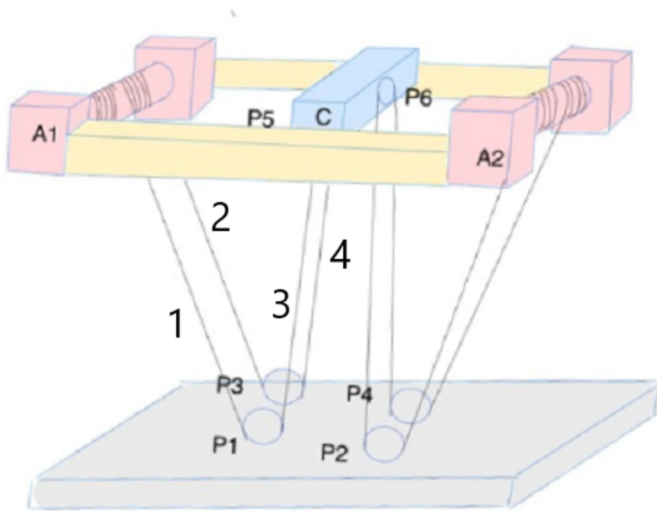


Figure 21: Length of cable on the drum of the hoist

For the calculation for how much space this would take on the drum, a couple of factors are assumed first, which were partly also already explained in the hoist part.

- The drum has a diameter of 30 cm;
- the cable has a diameter of 2 cm;
- and the cable can go over itself once, to decrease the necessary length of the whole drum to store the cable.

With those assumptions, the calculations can be made: One winding around the drum would take up $2 \cdot \pi \cdot 0.15 = 0.94\ m$. That means that the total amount of windings is equal to: $\frac{80}{2} \div 0.94 = 42.4 = 43$ windings. These 43 windings would take up $43 \cdot 0.02 = 0.86\ m$. This is done twice, meaning that the total length taken by the wound up cable is equal to $0.86 \cdot 2 = 1.72\ m$. A gap between the two cables is also incorporated in the design to ensure the cables will not touch each other while the winding process is happening. This gap length is set at 28 cm to create a shaft of a length of exactly 2 meters. All the above mentioned dimensions can be seen in Figures 76 and 77 in Appendix B.1 where the dimensions are also indicated. In Chapter 15.4 there will also be taken a look at the shaft in the drum as it is in a multi-axial stress state. Mohr's circle will then be applied to this element.

Another important dimension is the distance between the middle of both the drums. This distance is set at 4 m to provide stability to the spreader. The length of the container is namely equal to 12 meters. The spreader will cover this whole length. Two times two cables are used from the hoist to attach the spreader to, meaning the spreader will be the most stable if the first cable is attached 4 meters from one end, and the other cable is attached 8 meters from that same end, resulting in a gap between the cable pairs of 4 meters. A Figure showing these dimensions on the crane structure can be seen in Figure 78 and 79 in Appendix B.1. As stated earlier, this distance between the shafts is used.

# 8 Locking mechanism

First, a ratchet mechanism was chosen to lock the drum for lifting the boom in place and thus preventing the boom from lowering. However, the stress in the ratchet would be to high and slotting the ratchet out between the teeth would not be feasible. So for locking the boom in place, a new mechanism was thought of in the form of a thruster drum brake of which a schematic overview is given below.

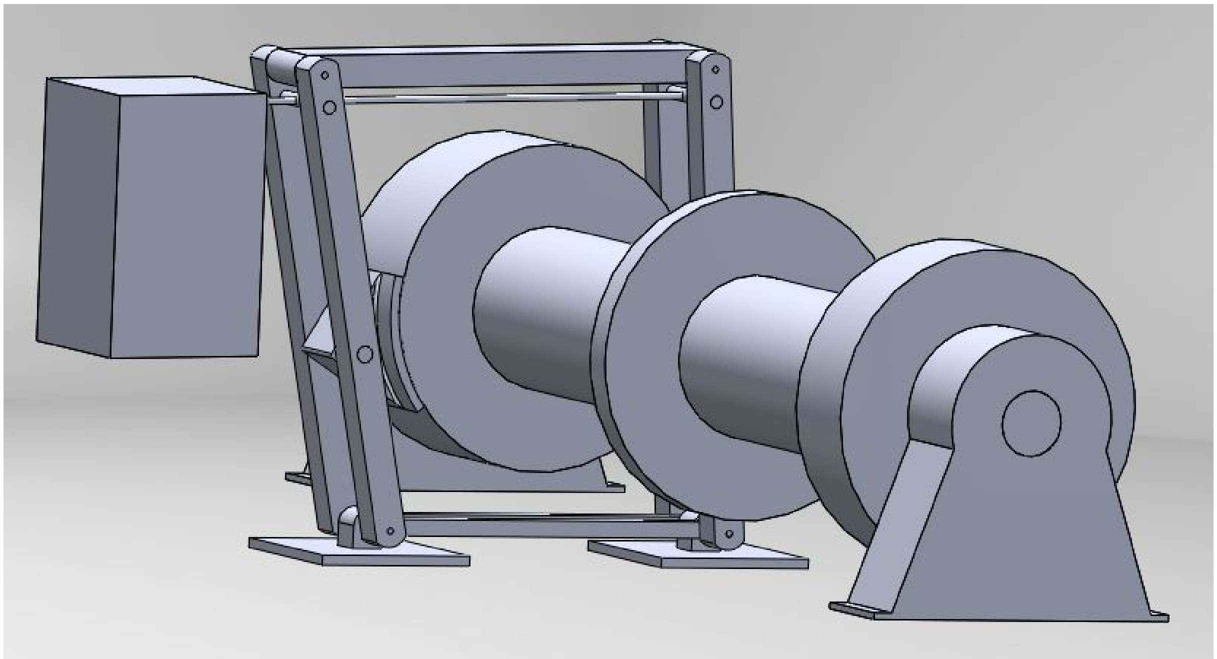

Figure 22: New locking mechanism

In 22 it can be seen that the brake is placed over the drum where two pads are connected to the surface of the drum. On the axis above the drum a spring is attached. While the motor is not being driven the spring will be uncompressed because of which the pads will be pressed against the drum and thus preventing the drum from rotating. By using this spring system the motor does not need to be driven constantly for locking the drum in place. The motor is represented by the grey box on the side of the drum. When the motor is being driven the spring will be compressed. Then the brake pads are dis attached from the drum which enables movement of the drum so the boom can be lifted or lowered.

# 9 Material Selection

In Figure 23 the components are shown for which materials will be chosen. In this chapter, all those components are discussed and a material is chosen according to the condition the component will have to operate under.

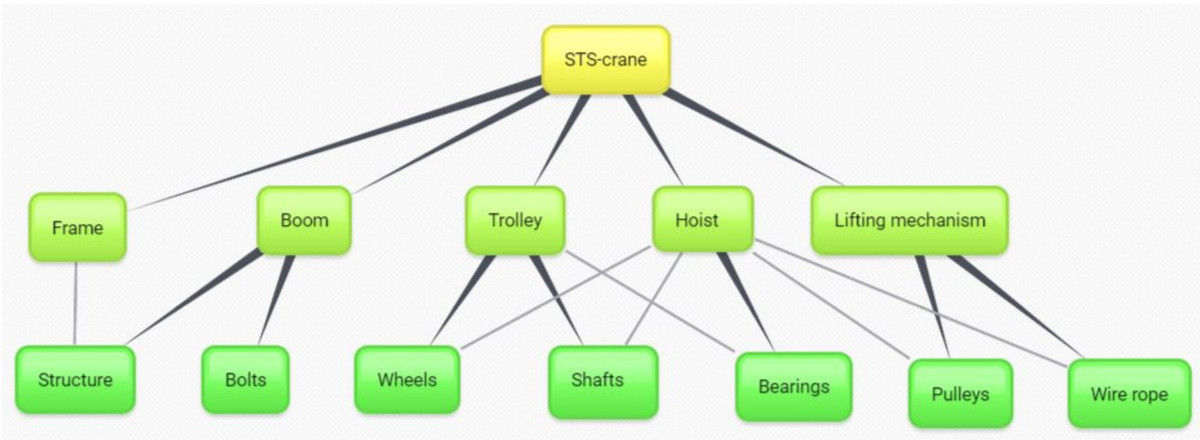


Figure 23: Product structure for the components where materials are chosen for

## 9.1 Boom, Trolley and Frame

The material of the boom needs to be strong in order to prevent deflection and needs to be corrosion resistant to make sure the crane can last for a long time. For the beams of the trolley and the rest of the frame, this is also true, as those are meant for absorbing the forces. This means that both parts will need the same kind of material. Except for strength and corrosion resistance, the material can only contain a low amount of carbon, as the heat of welding will otherwise get make the material weaker due to the diffusion of the carbon. This would cause a higher chance of cracks and should, therefore, be prevented. This means that the material should be low carbon- corrosion resistant- metal. Making use of the CES program, six suitable low-alloy-high-strength metals are selected. These are: S300, S350, S355, S420, S500 and S550. All metals are structural steels, denoted by the S in their name, which means they are suitable for building large constructions and are all corrosion resistant. Even though all of these materials could be suitable for the crane, some have advantages over others. The higher the number used in the name, the higher the minimum yield strength. A high yield strength is optimal as the plastic deformation will only start with higher stresses. On top of that S355 and S420 are used frequently in cranes, so those have been looked into more in detail. The biggest differences between S355 and S420 are the alloys available in the metals, next to carbon. S420 generally contains more silicon than S355 according to the CES program, increasing the strength and hardness of S420, having a Young's modulus between 200 and 221 GPa and a yield strength of 420 MPa (which is why the name is S420). This is favourable as it causes the deflection to decrease. When looking into the costs, the prices of the steel turned out to be almost similar and their Young's Modulus is similar as well [1] [2] [3].
As S420 steel has a higher yield strength and has more alloying elements that have a positive effect on the structure, it was chosen to be the most suitable steel. It is corrosion resistant and can be produced in I-beams, as needed for the boom and frame structure. For the trolley beams, this is the same.

## 9.2 Materials for Bearings and Bolts & Nuts

Nuts and bearings are small parts in which a lot of stresses are concentrated. Translating this to properties, it means that a high yield limit and a Young's modulus around 200 GPa would be enough. But it is also important to have some protection against corrosion, especially oxidation because the working ambient of the crane is close to the sea. This means salt in the air plus humidity and rains can cause oxidation.
Bolts are more likely to fail due to higher tensile stresses than the material can stand. Translating this to properties nuts must have certain properties: high yield strength to withstand tensile forces, good strain value to use it as a warning when bolts start to fail and the price must be reasonable. On the other hand for bearings, they must withstand compression forces and have a high Young's modulus because it is important for bearings to keep their spherical shape to make movement smooth. What also goes for nuts, bearings must be corrosion resistant. For both parts the working temperature must be minimum 40°C, meaning no cracking should happen at this temperature.

**List of materials for bolts and nuts:**
Both materials are stainless steel, with similar composition. The only difference is a small percentage of nickel in AISI 403.

**AISI 410:**
Young's Modulus: 195 GPa-205 GPa
**AISI 403:** Nickel (0-0.6%) decreases the temperature in which perlite starts to form
Young's Modulus: 195 GPa-205 GPa

Table 1: Materials for bolts

| Material | $\sigma$ (MPa) | Heat treatment |
|---|---|---|
| AISI 410 | 276-310 | Annealed |
| AISI 403 | 245-550 | Annealed |
| AISI 403 | 550-620 | Intermediate tempered |
| AISI 403 | 620-700 | Hard tempered |

**Material selected**
From Table 1 above it can be seen than **AISI 403** hard tempered has the highest yield limit. This is why this is the chosen material for the nuts. Also, the price is favourable with $1.27 per kilogram. The strain is 12 to 20 %, which would give a warning when the bolt is about to fail. The working temperature is between -73°C and 800°C, which satisfies the requirement of a minimum working temperature of -40°C.

Table 2: Materials for bearings

| Material | Compressive stress (MPa) | Price $\frac{USD}{kg}$ | Working temp °C | Heat treatment |
|---|---|---|---|---|
| ASTM CF-16F, Stainless steel | 265-285 | 2.9-3.4 | -200 - 743 | Annealed; water quenched |
| High alloy steel 250 | 245-550 | 29.1 | -73 - 490 | Mareged at 430°C |
| Low alloy steel, AISI 9310 | 510-630 | 0.97-1.13 | -43 - 665 | Annealed; water quenched |

Evaluating Table 2 above, it looks like AISI 9310 seems the best option, it has the highest compressive strength and the lowest price. However, the problem with this material is the lack of protection against oxidation. There are ways to cope with this disadvantage, such as a coating material or sacrificial protection. Bearing friction, however, would take out the coating and an extra sacrificial component will increase costs. That is why **ASTM CF-16F**, stainless steel is chosen: price and protection and working are favourable, compression stress will have to be compensated with the size of the bearings.
On the other hand, bearings must have isotropic properties (same properties in any direction). Also, the parts in which balls and rings of the bearing make contact are polished to decrease friction during operation. The material for the rings and balls are the same, however, the crystal structures are different because of the production process.

## 9.3 Wire Cables

The material of the wire cables needs to have a high Young's modulus (190-210 GPa) to prevent deflection. Since the STS-crane is situated in an environment close to the sea, there is a higher risk of corrosion. Therefore the material needs to be very corrosion resistant. Stainless steel is the most suitable option because of its mechanical properties, anti-oxidation properties and lower price compared to other materials. Since cables are often annealed during the production process, a carbon concentration of 0.5 to 0.8 % is necessary. Using these criteria and the CES program, three materials were selected. All the materials found have a Young's modulus of 190-210 GPa and a price of 0.913 to 1.06 EUR/kg. Those materials were then compared on three different properties, namely yield strength, tensile strength and the carbon concentration. The results of this can be found in Table 3.

Table 3: Materials for wire cables

| **Material** | $\sigma_y$ **(MPa)** | $\sigma_t$ **(MPa)** | **Carbon Concentration (%)** |
|---|---|---|---|
| AISI 440A | 1490-1820 | 1610-1970 | 0.6-0.75 |
| AISI 440B | 1670-2050 | 1740-2130 | 0.75-0.95 |
| AISI 440C | 1710-2090 | 1770-2170 | 0.95-1.2 |

As seen in Table 3, the first material - Stainless steel, martensitic, AISI 440A, tempered at 316°C – has the lowest yield strength and tensile strength, but has a carbon concentration within the set range. The second material - Stainless steel, martensitic, AISI 440B, tempered at 316°C – has a higher yield strength and tensile strength. However, the carbon concentration has an upper limit which does not lay within the set range. The last material - Stainless steel, martensitic, AISI 440C, tempered at 316°C - has the highest yield strength and tensile strength but the carbon concentration is too high. Since only **Stainless steel, martensitic, AISI 440A, tempered at 316°C** has the carbon concentration needed for the production process, this is the material that was selected.

## 9.4 Wheels of Trolley and Pulleys

The most common material used for the wheels of trolleys is Molybdenum because it is a strong material with a high Young's modulus making this material very good to supporting stresses because pulleys also support really high stresses this material is perfect to fulfil those qualities that pulleys require.

Table 4: Materials for Wheels of trolley and Pulleys

| **Material** | **Young's Modulus (GPa)** | **Yield Strength (MPa)** | **Thermal Expansion (ustrain/°C)** |
|---|---|---|---|
| Molybdenum, Alloy 362 | 305-325 | 740 - 910 | 5.4 - 6.2 |
| Molybdenum, Alloy 363 | 305-325 | 770 - 950 | 4.7 - 5.5 |
| Molybdenum, 360 grade | 315-335 | 130 - 350 | 4.8 - 5.5 |

Because there are not a lot of wheels and pulleys, the costs do not play that big of a big role. Therefore the best properties are the objective of these simple products that are important parts for the proper functioning of a crane.

Molybdenum, Alloy 362 and Molybdenum, Alloy 363, have very similar properties. They just have a small difference with thermal expansion which should be taken into account for a product like a wheel that increases in temperature for friction forces and high stresses located in the wheels and pulleys.
The third option, Molybdenum, 360 grade, has a high Young's modulus, but the elastic limit of this material is very low compared to the first two options, and these products need a high enough elastic limit otherwise it will deform easily. This all gives that the final material will be **Molybdenum, Alloy 363, TZM**.

# 10 Strength and Stiffness Calculations

## 10.1 Free Body Diagrams

Before starting the strength and stiffness calculations in FEM, two FBD's will be made of two critical positions of the trolley with a container. Those positions are indicated in Figure 24 and Figure 26 which are both made in Matlab. In the first position, the load will be exerted on the tip of the boom. Here, the biggest deformation can take place, which is why this situation will be investigated in-depth. The second position is when the load is exerted right next to the first legs. Here, there will be high compression forces in the legs and it will be researched if due to those forces the deformation limit is not overwritten.

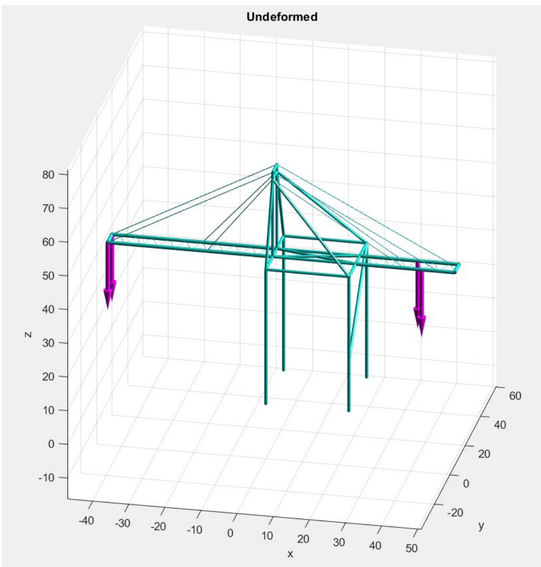


Figure 24: 3D view of the first FBD

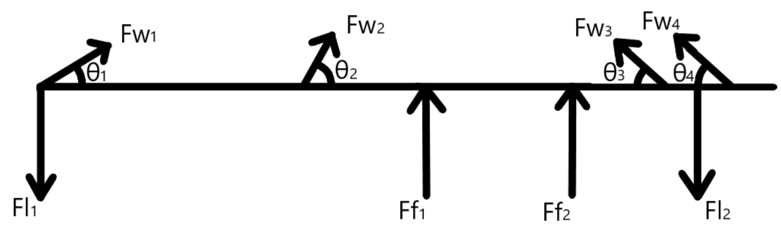


Figure 25: FBD with the load at the end of the boom

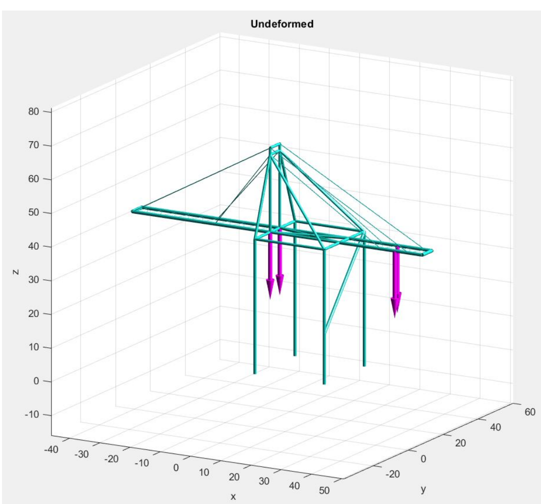


Figure 26: 3D view of the second FBD

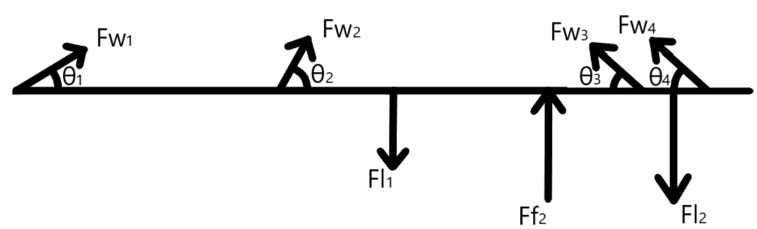


Figure 27: FBD with the load at the first leg of the boom

Next to the figures, the FBD's of the boom for both situations are shown in Figures 25 and 27. All of the angles theta for both FBD's are known (see the list on the next page) and will be the same. The forces indicated in both FBD's, are the forces at the most important positions of the crane. Fw is the force due to the wire-ropes, Fl is the force due to the load and Ff is the force of the frame exerted on the boom. With the help of the FEM-package, it will be investigated if the structure will not surpass the deflection limit at those critical positions and that will be shown in the course of this chapter.

- $\theta_1 = 29.1°$
- $\theta_2 = 49.0°$
- $\theta_3 = 30.7°$
- $\theta_4 = 50.9°$

## 10.2 Introduction

To calculate what the stresses and their effects are on the structure of the STS crane, the package FEM-frame-3D in Matlab is used to compose a model with all the beams and stresses. The results that FEM gives, are the local stresses of each element in X-, Y- and Z-direction and also the principal stresses in each element. To compose the model, each connection is treated as a node and from these nodes, elements are connected. The initial dimensions of the different elements are assumed from different pictures of real STS-cranes. Other inputs that the FEM package requires, are the density and Young's Modulus of the used material. For this model, the properties of steel S420 are used to model the whole structure. S420 steel has a Young's Modulus of 210.5 GPa and a density of 7800 kg/m$^3$.

## 10.3 Process

The design process of the crane in Matlab is done in three main steps. The first step is defining the starting dimensions of the crane, making the model in Matlab and then making sure that less or the same displacement than the maximum displacement specified in the requirements is followed, which can be seen in the list below.

- Perpendicular to gantry rails (along the boom): 5 mm
- Parallel to gantry rails: 60 mm
- Vertical: 150 mm

Not only the boom but also the frame of the crane with initially assumed dimensions is modelled in Matlab to get a full picture, as there are more deformations than only those at the tip of the boom. Other components, such as the legs, will also undergo deformations when the load is applied at the tip of the boom. To keep a realistic view, a limit is also set on the deformation of all the other components. This limit for all the other components is set as double the limit of the deformation of the tip of the boom. That means 10 mm in the x-direction, 120 mm in the y-direction and 300 mm in the z-direction. The second step that will be taken has to do with buckling. As buckling is the first failure mechanism regarding compression that will occur in the crane, it has to be taken into account when designing all the separate dimensions of the crane. The last step will then be to minimize the weight of the crane, with optimizing the dimensions of all the beams. This, of course, has to be done while still staying within the displacement limits and above the critical buckling load, which will be treated in the coming sections.

## 10.4 Kinds of Beams

For the entire crane, different kinds of beams are used as can be seen in Figure 28. The initial dimensions for the beams are given below, which are assumed according to pictures and literature of STS-cranes. Most of these values, however, will change after the weight optimization. The final values for all the dimensions are shown four pages further.

- The beams with number 1 are part of the frame, which are modelled as hollow square beams. The beams have an outer length and width of 70 cm and an inner length and width of 30 cm, meaning that the beams have an all-around wall thickness of 20 cm. The next four kinds of beams have already been shown in Chapter 7, so they will only be recapped shortly;

- Beams number 2 are the boom, modelled as I-beams;
- The beams with number 3 are the supports of the boom;
- Number 4 are the beams that support the supports;
- And number 5 are the beams that connect both of the I-beams of the boom.

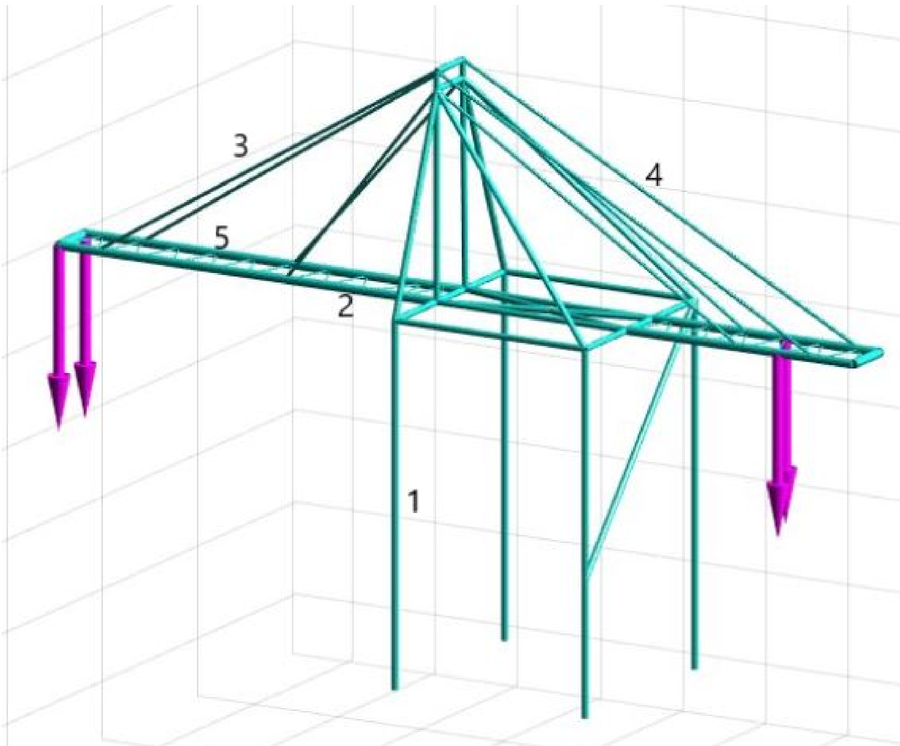


Figure 28: The different kind of beams used in the model

## 10.5 Wind During Operation

To deal with the maximum wind speed of 72 km/h as said in the project description, extra forces are incorporated in the Matlab file. This is done by means of the wind force exerted onto the area of the container and the boom. A sketch of the container with the wind blowing on it is shown in Appendix C.2. In the y-direction of the crane, the area of the container on which the wind blows is equal to 2.4·2.6 = 6.24 m$^2$. Multiplying this with the amount of force at a wind speed of 70 km/h (240 N/m$^2$), the result is 1497.6 N in total. Half of this force is then modelled as an extra force that acts on two points of the boom (both the connection points of the cables with the hoisting mechanism). The same procedure is done for the x-direction, which gives a total force which is equal to 12·2.6·240 = 7488 N. This is again modelled as two times 3744 N on two nodes of the boom.
Besides the wind exerted on the container, the wind is also exerted onto the boom itself, which will cause some displacement. As the surface area in the x-direction is very small, only the y-direction will be considered. The boom is 47 meters long and 1.5 meters high, meaning that the surface area is equal to 1.5·47 = 70.5 m$^2$. This gives a force in the y-direction with a magnitude of 16920 N. This would normally be a distributed force over the entire length of the boom. With the current skill set, this is however not yet possible, which is why this force is modelled at the node in the middle of the boom. Also, the wind does not have a big influence on the structure. The magnitude is in the order of $10^4$ while the force of the trolley and container is in the order of $10^7$.

## 10.6 Weight Optimization

To carry out the weight optimization, there should be looked at the mechanism that can cause the first failure of the crane. This failure mode that will occur the first in the crane structure is buckling. Buckling will happen in the parts that are under compressive forces, so that means this will mostly occur at the bottom of the frame. On those parts, the whole weight of the crane is exerted. To calculate the critical load for buckling, a formula is used that assumes that the beam is clamped on one side (the connection of the bottom of the beam with the land) and on the other side, the compressive force is exerted, see Figure 29. With assuming this system in the bottom beams of the frame, a critical load $P_{crit}$ is given by $P_{crit} = \frac{\pi^2 \cdot E \cdot I}{4 \cdot L^2}$. A list with the meanings and the values of the variables are given in a list below.

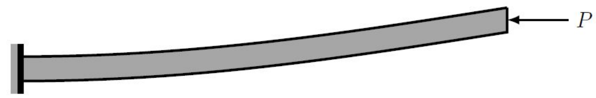


Figure 29: Buckling in a beam

- E is the Young's modulus of the used material for the boom, which is S420 steel. This material has a Young's modulus between 200 GPa and 221 GPa, so an average of 210.5 GPa is taken;
- I is the moment of inertia for the beam, which is calculated with I = $\frac{1}{12} \cdot 0.7^3 \cdot 0.7$ - $\frac{1}{12} \cdot 0.3^3 \cdot 0.3 =$ 0.0193 $m^4$ for the leg;
- L is the length of the beam, which is equal to 40 meters.

All of the above gives a $P_{crit}$ of 6276 kN. With this $P_{crit}$, the critical stress is then calculated: $\sigma = \frac{F}{A}$, with F equal to $P_{crit}$ and A the cross-sectional area of the beam ($0.7 \cdot 0.7 - 0.3 \cdot 0.3 = 0.4\ m^2$). Those values give $\sigma$ to be equal to 15.69 MPa.

The maximum value of the compressive stress in the structure should always be lower than this to not fail due to buckling. When running the Matlab code however with the indicated values, it was observed that the maximum compressive stress in the indicated points (the legs) is equal to 29.22 MPa, a value way higher than the maximum allowable stress. To try to reduce the maximum compressive stress in the structure and increase the value for $P_{crit}$, several steps are taken:

- The dimensions of the triangles are reduced to 0.1 m by 0.1 m, as in all the iterations the triangles did not take that high of a load (0.5-1 MPa). This reducing of the dimensions will reduce the weight of the entire structure, meaning that the force exerted on the bottom beams will be lower, resulting in lower compressive stress;
- The same is done for the supports, the dimensions of those beams are decreased from 0.4 m by 0.4 m to 0.2 m by 0.2 m. This is again done to decrease the mass of the entire structure;
- The last change that is done, is to increase the inner and outer width of the hollow beams of the frame. The outside width went to 0.9 m (from 0.7 m) and the inside width to 0.5 m (from 0.3 m). This change will increase the load-carrying capabilities of the frame as there is more area to carry the whole construction. This change will however also increase the mass of the structure again, but the increase in the area has a bigger positive effect on the load carrying capabilities than the negative effect of the increased mass.

With those new dimensions of most of the beams, the maximum allowable stress as well as the maximum compressive stress in the crane changes. Using the same formula as above, the maximum allowable stress turns out to be 28.67 MPa and the maximum compressive stress equals 23.33 MPa, which can also be seen in Appendix C.2. As can be seen from those values, the maximum compressive stress is lower than the maximum allowable stress, meaning that the structure will not fail due to buckling. There is even a safety factor of $\frac{28.67}{23.33} = 1.23$, meaning that there is some tolerance.

**Safety Factor**

Furthermore, it is important to look if this safety factor is a reasonable safety factor for this application. For buckling, often a safety factor between 1.5 and 3 is used as there is a static load, but the design data (such as dimensions and other loads as the wind etc.) are not very confident [4]. The frame is redesigned to try to reach this safety factor between 1.5 and 3.

- The frame is split up into two parts. The first part consists out of the legs of the crane, which are essentially four long beams and the second part is the frame on top of the crane, which connects the boom and the supports;
- The legs are now modelled as solid square beams with a length and width of 1.1 meters. Making the legs solid, will again increase the load carrying capability of the legs;
- The part of the frame that is not part of the legs, will keep being hollow square beams. The hollow beams are again used to decrease the overall weight. The outside height and width of this part of the frame are 0.7 m and the inside height and width are 0.3 m.

As there are quite some components in the crane now, clearer pictures with the different beams are shown in Figures 83 and 84 in Appendix C.2. Also, from this point on, it was discovered that another beam of the frame was loaded with the maximum compressive stress (25.11 MPa), which is indicated in Figure 30. This beam was however also almost two times shorter with a length of 24.61 meters, which will have a big impact on the critical buckling stress (shorter length means higher buckling stress). That is why for the rest of the calculation, there will be looked at if the front legs (which are the longest parts under high compression) and the indicated beam (called shorter beam in the rest of this section) will have a safety factor which is within the given range.

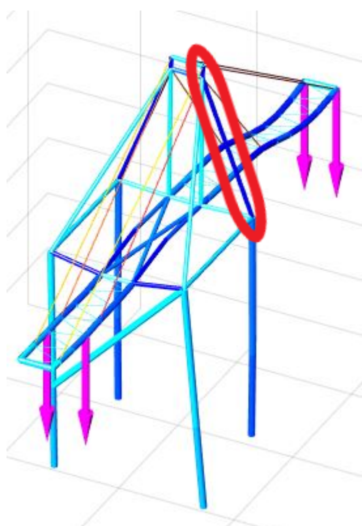

Figure 30: The beam with the highest compressive stresses in the structure

When filling in the formula for the critical buckling stress again with the dimensions of the indicated beam and the front legs, the two critical stresses are 32.73 MPa for the front leg and 41.45 MPa for the shorter beam. Both calculations can be seen in Appendix C.3. Then, the stress in both of the components will be taken from Matlab. The stress in the leg is equal to 9.83 MPa and in the shorter beam, it is equal to 25.11 MPa (the maximum compressive stress in the whole crane). Those values give safety factors for the leg of: $\frac{32.73}{9.83} = 3.33$ and for the shorter beam of: $\frac{41.45}{25.11} = 1.65$. These values indicate that the safety factor for the shorter beam is within the two limits and the safety factor for the leg is even above the upper limit, which is a good sign.

**Final Optimization**

As both safety factors are still above the minimum safety factor, it means that the dimensions of the crane can be reduced one final time in order to reach the lowest possible weight. After continuously filling in new values to stay above the safety factor of 1.5 and also stay within the limits for deformation at the tip of the boom and the rest of the structure, the final dimensions are the following:

- Beams 1: The legs have a height and a width of 0.9 meters;
- Beams 2: The outside width is equal to 0.8 meters and the inside width is equal to 0.55 meters, meaning that there is an all-around wall width of 12.5 centimetres;
- Beams 3: The dimensions of the I-beam remain unchanged;
- Beams 4: The supports have a height and a width of 0.2 meters;
- Beams 5: The supports of the supports have also a height and a width of 0.2 meters;
- Beams 6: The height and width of the triangles between the beams of the boom have a height and a width of 0.1 meters.

With all the indicated dimensions, the safety factors for the two most important beams and the general displacements and maximum stresses are calculated and shown in Figure 31.

As can be seen in the figure, the values of the deformation in the X, Y and Z direction for both the tip of the boom and the remaining structure are still within the specified bounds. The critical buckling stresses are now as followed: for the leg, it is 21.91 MPa and for the short beam it is 67.36 MPa. From Figure 31, it can be seen that the safety factor for buckling for both the legs and the short beam is higher than the minimum safety factor of 1.5.

**Other Lifting Position**

The previous calculation was done with the trolley being at the tip of the boom, lifting containers from the ship. The other position which could possibly cause high stresses in especially the legs is when the trolley with a container is right above the legs of the crane, as is indicated in Figure 32. At this point the compressive forces are equal to 11.45 MPa, meaning that the critical buckling stress of 21.91 MPa (safety factor of 1.91) is still not reached within the safety limit factor. The same goes for the shorter beam, which at this point of the trolley has a compressive stress of 23.61 MPa, which still does not reach the limit of 67.36 MPa (safety factor of 2.85). As the crane functions safely in those two critical positions, it can safely be assumed that in the other possible positions of the trolley the frame will also not undergo buckling.

```
X_displacement =
    1.3147
max_X_displacement =
    9.6280
Y_displacement =
    1.1983
max_Y_displacement =
    4.9254
Z_displacement =
   60.0589
max_Z_displacement =
   92.8793
Max_Compressive =
  -3.0949e+07
Max_Tension =
   6.7363e+07
Safety_factor_leg =
    1.7159
Safety_factor_short_beam =
    2.3190
Total_Mass =
   3.0731e+06
```

Figure 31: The final amount of deflection in the tip of the boom

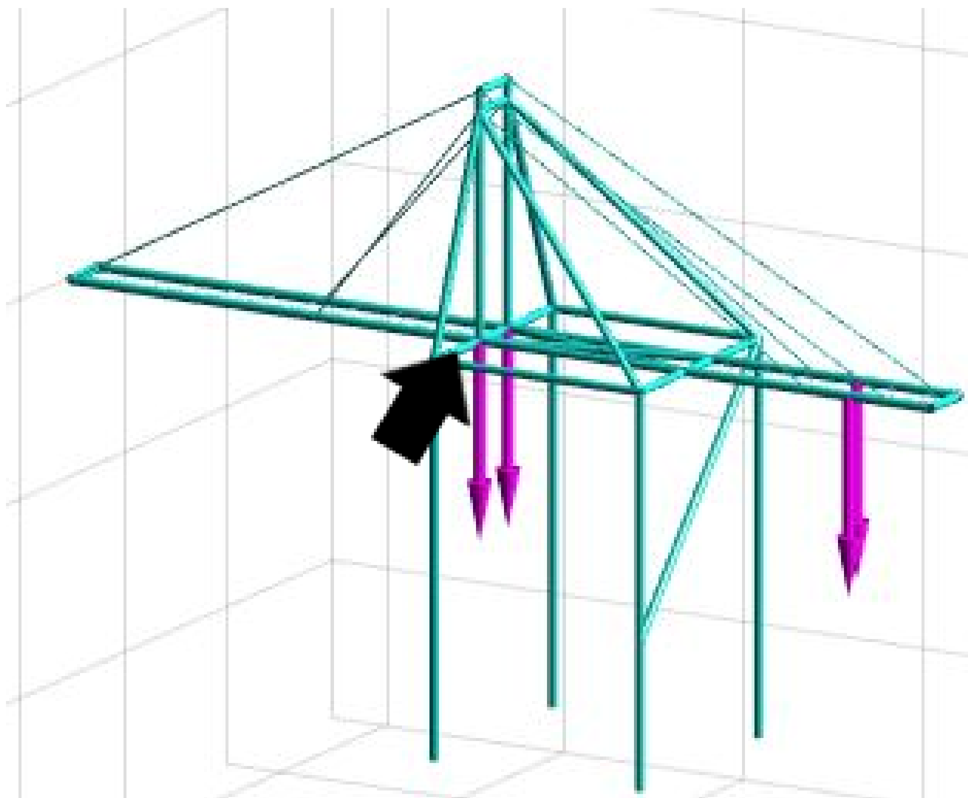

Figure 32: The other position of the trolley where high compressive stresses could occur in the legs

**Tensional Stresses**

There will also be taken a look at the maximum tensile stresses in the crane, as they should be below the yield strength of the S420 material. The maximum tensional stresses occur in the supports of the boom, where this value is equal to 67.36 MPa, which can also be seen in Figure 31. In relation to S420, having a yield strength of 420 MPa, the maximum tensile stresses are a factor of $\frac{420}{67.36} = 6.24$ lower, meaning that there is more than enough tolerance.

**Conclusion**

As was shown in the previous chapters, the deformations are all within the bounds, the maximum compressive stress is below the buckling stress and the tensional stresses are well below the yield strength. This means that the forces shown in the FBD's at the beginning of the chapter remained within the specified limits. It also shows that the assumption of using S420 steel for the entire framework of the crane was a good decision the final mass can be calculated. With all the final values, the total mass of the crane can be determined. This value is again calculated with Matlab with the values from the areas and the density of the used material. As shown in Figure 31, the final mass is equal to 3073 tonnes.

## 10.7 Thermal Expansion

The usage of the FEM-package is now finished, but there is still one more thing to look at and that is thermal expansion. Due to the possible temperature changes from -50°C to 40°C, the material will expand. The part of the structure where this thermal expansion will have big impacts, is the hinge point of the boom, see Figure 33.

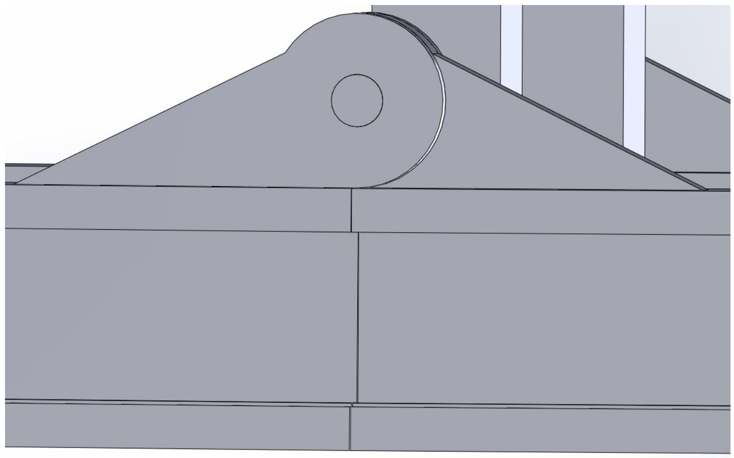

Figure 33: Position where thermal expansion has a big impact

The boom is a very long structure, so thermal expansion will play a big role here. With the following formula, $u_T(L) = \alpha \cdot \Delta T \cdot L$, the thermal expansion can be calculated if one of the ends is free and the other one is fixed. This is the case for the boom, so: $u_T(L) = 12 \cdot 10^{-6} \cdot 90 \cdot 47 = 0.0508m = 50.8\ mm$. This is quite much above the deflection limit in the x-direction, which is why there is looked at ways to decrease this thermal expansion.

The thermal expansion can be reduced in several ways. This is done by looking at which variables are involved in the formula for the thermal expansion. A is the cross-section and $\alpha$ and E are material properties, so that means choosing another material can drastically change the thermal expansion, as there are two variables involved. Materials with a very low thermal expansion coefficient, such as for instance silicon nitride ($2.6 \cdot 10^{-6}$), do not always have a high yield strength (60 MPa), which is necessary for an STS-crane [5] [6]. To find the right balance between the yield strength and the thermal expansion coefficient, another research to find the right material should be done. The resulting material of this research should have a low enough thermal expansion coefficient to not elongate too much and a high enough yield strength so that the material will have a high enough safety factor. There has been looked further into other materials that have the same yield strength and a comparable price, as the current yield strength has already been used in all kind of calculations and the material should not get too expensive. In the CES program, no materials were found that have comparable properties and a thermal coefficient with a factor of 10 times lower. As the material choice in this project was done before doing the calculations and doing more research now into new materials takes too much work for the given time-frame, this should be done in further research.

Besides the material properties, the cross-sectional area also has an effect on the thermal expansion coefficient. Reducing the cross-section will reduce thermal expansion. When reducing the cross-section, however, all of the FEM calculations will have to be done again as decreasing dimensions, with keeping the same load, will increase the stresses. It would have to be checked again if the critical beams mentioned in this chapter would still be within the given safety factors for buckling. As this is not feasible anymore in the given time-frame, this investigation could be done in further research where a right balance between material properties, cross-sectional area and stresses should be selected, but this will not be discussed further in this project.

# 11 Drive System

The motor is a vital component in many industrial mechanisms. However, choosing a motor depends on a number of factors. For the STS crane, the requirements and environmental conditions are according to the lists below. Appropriate machine elements that are used in the hoist such as bearings, shafts and keys will be determined accordingly.

## 11.1 Hoist

There are various possibilities for the driving mechanism of the hoist in an STS crane. One main factor is to be able to handle a heavy load of up to 50 tonnes. According to this requirement, 4 motors of a size larger than 0.75 kW must be chosen.
The hoist consists out of 4 motors, therefore the load can be divided by four. This means that every motor has to carry a load of 12 500 kg.

The following things are given:

- Speed: 90-120 m/min
- Loading capacity: 12.5 tons

To operate fast, which is important nowadays, the maximum hoisting speed of 120 m/min is chosen to make the calculations with. This linear speed of 120 m/min can be converted in rotational speed given that the radius of the drum is 0.15 m. The rotational speed of the hoist is then 127.32 rpm or 13.33 rad/sec.

To make a choice on the motor type, the power input, $P_{in}$ has to match the power output $P_{out}$. The power output is yet unknown but can be calculated with the following formula.

$$P = T \cdot \omega \tag{1}$$

Where Power is in Watt, Torque is in Nm and $\omega$ in rad/sec. The torque of the load can be calculated with a formula for hoisting applications: $T = m \cdot g \cdot \frac{D}{2}$, which gives $T = 12\,500 \cdot 9.81 \cdot \frac{0.3}{2} = 18\,393.75$ Nm. Now the power can be computed, $P_{out} = 18\,393.75 \cdot 13.33 = 245$ kW.

To calculate $P_{in}$ a safety factor of 1.2 is applied. Then the power input is $245 \cdot 1.2 = 294$ kW. So a motor with a power of around 300 kW is needed. From a catalogue of TECO an electrical motor has been selected with the following properties [7]:

| FULL LOAD rpm | FRAME NO. | EFFICIENCY | | | POWER FACTOR | | | CURRENT | | TORQUE | | | | ROTOR GD² kg-m² | NOISE SOUND POWER NO-LOAD dB(A) | APP. WEIGHT kg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | FULL LOAD (%) | 3/4 LOAD (%) | 1/2 LOAD (%) | FULL LOAD (%) | 3/4 LOAD (%) | 1/2 LOAD (%) | FULL LOAD (A) | LOCKED ROTOR %FLT | FULL LOAD kg-m | LOCKED ROTOR %FLT | PULL UP %FLT | BREAK DOWN %FLT | | | |
| 1485 | 315D | 95.1 | 95.1 | 94.3 | 89.0 | 88.0 | 84.5 | 566.0 | 675 | 206.70 | 140 | 115 | 230 | 31.200 | 108 | 1860 |

Figure 34: Properties electric motor

This motor has a full load torque of 206.70 kgm, which is 2027.729 Nm. The full load rpm = 1485, which is 155.51 rad/sec. The power output of this engine is 315 kW, which means that $P_{in}$ = 315 kW.

In Figure 35 a simplified drive system is sketched. This can be used to determine the ratio i of the system.

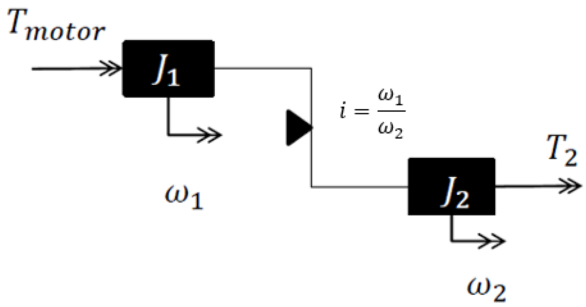


Figure 35: Simplified drive system

Looking at this simplified drive system, the following can be filled in:

- $T_{motor}$ = 2027.729 Nm
- $\omega_1$ = 155.51 rad/sec
- $T_2$ = 18 393.75 Nm
- $\omega_2$ = 13.33 rad/sec

Given this, the ratio i can be calculated. i= $\omega_1/\omega_2$ = 155.51/13.33 = 11.66. This is the TV (train value) for the gear system, in other words the total reduction ratio of the rotational speed is 11.66.

Given the TV of 11.66, a two stage transmission is needed where VR1 (velocity ratio 1)·VR2 (velocity ratio 2) = 11.66. From the graph (slide 32 gear drives machine elements) VR1 is set to be 3.6. VR2 $= \frac{TV}{VR1}$ = 3.24. Now the drive system can be sketched as follows:

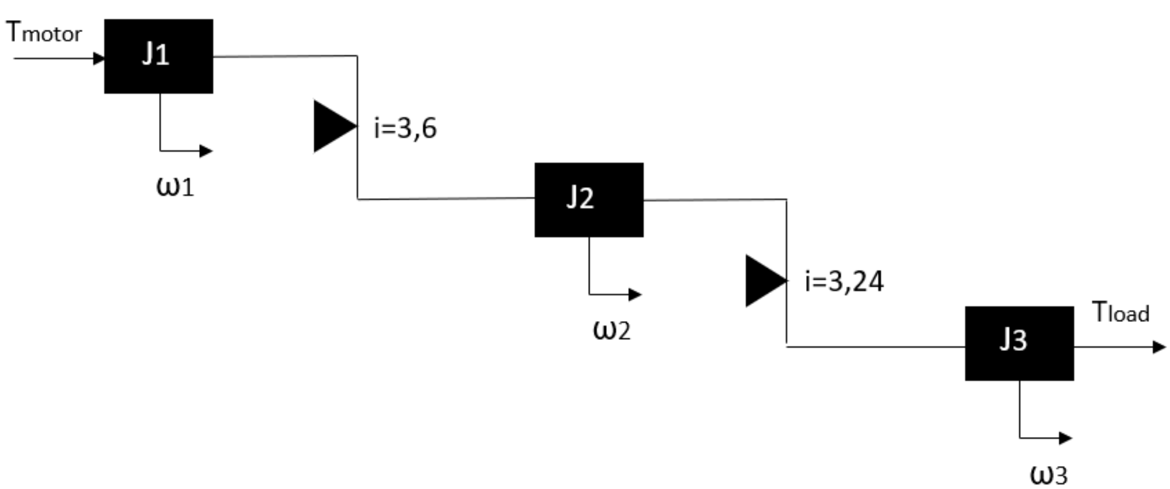


Figure 36: Initial drive system

For Figure 36 $\omega_1$ = 155.51 rad/sec and so $\omega_2 = \frac{155.51}{3.6}$ = 43.197 rad/sec. $\omega_3$ is then $\frac{43.197}{3.24}$ = 13.33 rad/sec.

## 11.2 Gear Design

In Figure 37 a schematic gear system can be seen. Z1 till Z4 are the 4 gears that are connected to each other. The shaft between Z2 and Z3 is in reality not that long that it can be seen, but for clarity, it has been sketched exaggerated. Z2 and Z3 are actually right next to each other. First, the number of teeth

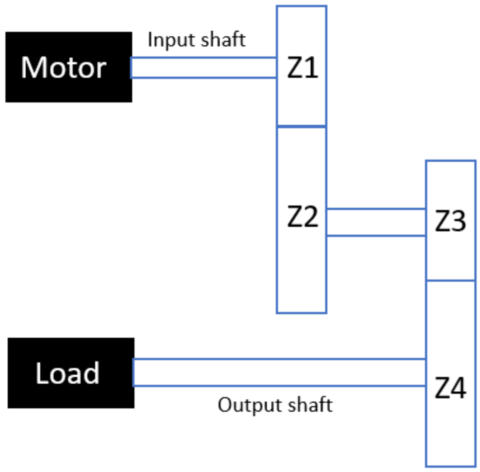


Figure 37: Schematic gear system

for the pinion (Z1) is chosen. Typically for hoisting applications, the number of teeth for the pinion lies between 14 and 20. To avoid undercutting this number should also be larger than 17. The number of teeth for Z1 = 18.
The number of teeth for the gear (Z2) will then be 18·3.6 = 64.8. To avoid a common divisor (and thus avoid a hunting tooth) 67 is chosen since 67 is a prime number. Now the actual ratio is 67/18 = 3.72
Z2 and Z3 are connected by a shaft, that is in reality not visible because there will not be that much space between the two gears.
Pinion Z3 has the same number of teeth as Z1: 18 and also for the same reasons. Then the number of teeth for the gear (Z4) is 18·3.24 = 58. To avoid a common divisor 59 is chosen since 59 is a prime number. Now the actual ratio is 58/18 = 3.22. The actual train value of the gears is $3.72 \cdot 3.22 = 11.99$.

Now a module has to be selected. The module is chosen to be 6 since strong teeth are needed that can withstand high forces and transmit high power. With the module being set, the pitch diameters of the gears can be calculated.

- $D_{z1}$: 6·18 = 108 mm
- $D_{z2}$: 6·67 = 402 mm
- $D_{z3}$: 6·18 = 108 mm
- $D_{z4}$: 6·59 = 354 mm

The width of the gear also has to be specified. A value of 1 is chosen for the width-diameter ratio, which means that the face width is 108 mm for the gears Z1 and Z3. To let the gears Z2 and Z4 match properly with Z1 and Z3, the face width of Z2 and Z4 is also 108 mm.

All the different rotational speeds per gear can also be calculated now:

- Z1: 1485 rpm
- Z2: $\frac{1485}{3.72} = 399.2$ rpm
- Z3: 399.2 rpm
- Z4: $\frac{399.2}{3.22} = 124.0$ rpm

Also the torque transmitted per gear is known and for every gear taken, a loss of 3 percent is assumed:

- Z1: 2027.729 Nm
- Z2: 2027.729·3.72·0.97 = 7316.9 Nm
- Z3: 7316.9 Nm
- Z4: 7316.9·3.22·0.97 = 22 853.5 Nm

The power is calculated now:

- Z1: P = 2027.7·155.5 rad/sec = 315 kW
- Z2: P = 7316.9·41.8 rad/sec = 305 kW
- Z3: P = 305 kW
- Z4: P = 22 853.5·13.0 rad/sec = 297 kW

For a clear overview, all the values have been put in a table

Table 5: Overview Gears 1 till 4

| Gear | Speed (rpm) | Speed (rad/sec) | Torque (Nm) | Power (kW) |
|---|---|---|---|---|
| **Z1** | 1485 | 155.5 | 2027.7 | 315 |
| **Z2** | 399.2 | 41.8 | 7316.9 | 305 |
| **Z3** | 399.2 | 41.8 | 7316.9 | 305 |
| **Z4** | 124.0 | 13.0 | 22 853.5 | 296 |

The updated drive system is shown in Figure 85 in Appendix D.

## 11.3 Bending Stress

For the gears, martensitic stainless steel is chosen as it is castable and has the following properties:

- Young's modulus = 195-205 GPa (200 GPa is chosen)
- Yield strength = 755-835 Mpa (800 MPa is chosen)
- This material has excellent durability against water (fresh and salt), which is important in for a container crane.

Then the bending stresses of every gear were computed to determine if every gear will not crack under pressure. The formula for the bending stress is:

$$\sigma_b = \frac{F_t}{b \cdot m} \cdot Y_{Fa} \tag{2}$$

Where $F_t$ is the tangential force on the tooth in N, b the face width in mm, m the module in and $Y_{Fa}$ the form factor. The form factor depends on the number of teeth of the gear and the profile shift x which is in this case 0. The value of the form factor can be looked up in table DIN 3990.

$\sigma_b$ Z1: $F_t = \frac{T}{0.5 \cdot D_p} = \frac{2027.7}{0.5 \cdot 0.108} = 37\ 537$ N. The form factor for Z1 is from DIN 3990 and it is 3.0 for Z1. The actual bending stress is then: $\frac{37\ 537N}{108 \cdot 6} \cdot 3.0 = 173.8$ N/mm$^2$.

$\sigma_b$ Z2: $F_t = \frac{7316.9}{0.5 \cdot 0.402} = 36\ 402.5$ N. The form factor for Z2 is from DIN 3990 and it is 2.4 for Z2. The actual bending stress is then: $\frac{36\ 402.5N}{108 \cdot 6} \cdot 2.4 = 134.8$ N/mm$^2$.

$\sigma_b$ Z3: $F_t = \frac{7316.9}{0.5 \cdot 0.108} = 135\ 498.1$ N. The form factor for Z2 is from DIN 3990 and it is 3.0 for Z3. The actual bending stress is then: $\frac{135\ 498.1N}{108 \cdot 6} \cdot 3.0 = 627.3$ N/mm$^2$.

$\sigma_b$ Z4: $F_t = \frac{22\ 853.5}{0.5 \cdot 0.354} = 129\ 115.8$ N. The form factor for Z4 is from DIN 3990 and it is 2.3. The actual bending stress is then: $\frac{129\ 115.8N}{108 \cdot 6} \cdot 2.3 = 458.3$ N/mm$^2$.

### 11.4 Contact Stress

The contact stress for every gear has to be determined as well. The formula for the contact stress is:

$$\sigma_h = \sqrt{\frac{F_t}{b \cdot D_1} \cdot \frac{u+1}{u}} \cdot Z_h \cdot Z_e \tag{3}$$

Where $\sigma_h$ is the contact stress in N/mm$^2$, $F_t$ is the tangential force in N, b is the gear width in mm, $D_1$ is the pitch diameter in mm, u is the teeth ratio, $Z_h$ is the curvature factor which is 2.5 for standard spur gears, $Z_e$ is the elasticity factor in N/mm$^2$ and E is the E modulus in N/mm$^2$. $Z_e$ is also given as $\sqrt{0.175 \cdot E}$. For the gears the E modulus is set to be 200 GPa which is 200 000 N/$mm^2$, because this the value for carbon steel that is typically used for gears.

$\sigma_h$ Z1:

$$\sigma_h = \sqrt{\frac{37\ 537}{108 \cdot 108} \cdot \frac{3.72+1}{3.72}} \cdot 2.5 \cdot \sqrt{35\ 000} = 948\ Mpa \tag{4}$$

$\sigma_h$ Z2:

$$\sigma_h = \sqrt{\frac{36\ 402.5}{108 \cdot 408} \cdot \frac{3.72+1}{3.72}} \cdot 2.5 \cdot \sqrt{35\ 000} = 478\ Mpa \tag{5}$$

$\sigma_h$ Z3:

$$\sigma_h = \sqrt{\frac{135\ 498.1}{108 \cdot 108} \cdot \frac{3.22+1}{3.22}} \cdot 2.5 \cdot \sqrt{35\ 000} = 1824\ Mpa \tag{6}$$

$\sigma_h$ Z4:

$$\sigma_h = \sqrt{\frac{129\ 115.8}{108 \cdot 354} \cdot \frac{3.22+1}{3.22}} \cdot 2.5 \cdot \sqrt{35\ 000} = 983\ Mpa \tag{7}$$

### 11.5 Conclusion Gears

From this table, it can be concluded that the bending and contact stresses in gear Z3 are relatively high. From table 9-3 (slide 29 of machine elements Gear Drives) that an allowable bending stress of 611.2 MPa is allowed (MPa is equal to N/mm$^2$) if the steel has a grade of 3. This means that the steel has to be case-hardened and carburized. Thus, for the production of gear Z3, heat treatments have to be conducted. Then the bending stress is still higher than the allowable bending stress, but this is negligible, otherwise a larger module for the gears Z3 and Z4 has to be chosen. For gears Z1, Z2 and Z4 the bending stress is far below the allowable bending stress for grade 1 which is 310 MPa. Grade 1 is only hardened, so that is the only thing that needs to be done for these gears.

For the contact stresses in the gears, it is the same story as for the bending stresses. The maximum allowable contact stress for a gear that is made out of steel with grade 3 is 1896 MPa. So again, it is clear that Z3 needs additional heat treatments to perform without any problems. For gears Z1, Z2 and Z4 again no additional heat treatment is needed because the values of the contact stresses are below the maximum allowable stresses for grade 1 which is 1172 MPa. Grade 1 means that the steel is only hardened.

Since gears Z1 and Z2 are far below the allowable stresses, a smaller module could maybe be chosen. This means that the strength of the tooth will decrease and so the bending and contact stresses will increase. A smaller module means that the size of the gears can be reduced, which can make the gears cheaper. Another 'solution' could be a reduction in the face width of the gears Z1 and Z2. Again this will increase the bending stresses and contact stresses, but it will reduce the weight and probably the price.
On the other hand, for the gears Z3 and Z4, a larger module would be better so that the actual bending and contact stresses of gear Z3 are far below the allowable stresses.

Choose module = 8 for gears Z3 and Z4:

- $D_{z3}$: $8 \cdot 18 = 144$ mm
- $D_{z4}$: $8 \cdot 59 = 472$ mm

Then the new bending and contact stresses are:
$\sigma_b$ Z3: $F_t = \frac{7316.9}{0.5 \cdot 0.144} = 101\ 623.6$ N. The form factor for Z2 is from DIN 3990 and it is 3.0 for Z3. The actual bending stress is then: $\frac{101\ 623.6N}{108 \cdot 8} \cdot 3.0 = 352.9$ N/mm$^2$.

$\sigma_b$ Z4: $F_t = \frac{22\ 853.5}{0.5 \cdot 0.472} = 96\ 836.9$ N. The form factor for Z4 is from DIN 3990 and it is 2.3. The actual bending stress is then: $\frac{96\ 836.9N}{108 \cdot 8} \cdot 2.3 = 257.8$ N/mm$^2$.

$\sigma_h$ Z3:

$$\sigma_h = \sqrt{\frac{101\ 623.6}{108 \cdot 144} \cdot \frac{3.22+1}{3.22}} \cdot 2.5 \cdot \sqrt{35\ 000} = 1369\ Mpa \tag{8}$$

$\sigma_h$ Z4:

$$\sigma_h = \sqrt{\frac{96\ 836.9}{108 \cdot 472} \cdot \frac{3.22+1}{3.22}} \cdot 2.5 \cdot \sqrt{35\ 000} = 738\ Mpa \tag{9}$$

It can be seen that the bending stresses as well as the contact stresses have reduced significantly. Increasing the module from 6 to 8 was needed to get this result.

For a clear overview a table is created which is shown below.

| Gear | Module | Pitch Diameter (mm) | Bending stress (N/mm$^2$) | Contact stress (N/mm$^2$) |
|---|---|---|---|---|
| **Z1** | 6 | 108 | 173.8 | 948 |
| **Z2** | 6 | 402 | 134.8 | 478 |
| **Z3** | 8 | 144 | 627.3 | 1369 |
| **Z4** | 8 | 472 | 458.3 | 738 |

## 11.6 Moments of Inertia

For the inertia of the mass, a mass of 12 500 is used because 4 motors are used, so 50 tons divided by 4 is 12.5 tons. The radius of the drum is 0.15 m.
So that gives the inertia of mass = mass$\cdot$ $r^2 = 12\ 500 \cdot 0.15^2 = 281.25\ kg \cdot m^2$
For the inertia of the rope drum, the formula for a solid cylinder is used to calculate the moment of inertia.
Inertia of rope drum = 0.5$\cdot$mass$\cdot r^2 = 0.5 \cdot (\pi \cdot\ 0.15^2 \cdot 2 \cdot 8000) \cdot 0.15^2 = 12.72\ kg \cdot m^2$
The total inertia load is the inertia of the mass plus the inertia of the rope drum.
Inertia load = $281.25 + 12.72 = 293.97\ kg \cdot m^2$

Inertia load reflected on motor = $\frac{J_L}{efficiency \cdot i^2} = \frac{293.97}{0.97 \cdot 0.97 \cdot 11.99^2} = 2.2\ kg \cdot m^2$

Inertia ratio = $\frac{reflected\ inertia}{inertia\ of\ the\ motor} = \frac{2.2}{32.2} = 0.07$

An inertia ratio of 0.07 is relatively low. This means that the motor is a bit oversized. But for fast moves and frequent starts and stops an inertia ratio less than 2 is preferred, so it meets that criteria. For improvement a smaller motor can be chosen to reduce weight and costs, but for now it will remain the same engine.

## 11.7 Shafts

The standard shaft dimension according to the selected motor is given in the catalogue, as a result, suitable material must be chosen for the required torque [8]. First, it is assumed the material that is going to be used for the shaft which is mild steel with a shear stress of approximately 220 Mpa. It is also assumed that the weight of both shafts that would contribute to the inertia is included in the efficiency of the transmission.

From the catalogue, the shaft diameter specified for the selected motor is 95 mm[8]. The shaft diameter corresponds to the total length of the motor due to the fact that the size of every component is standardized.

- Torque of input shaft : 2027.7 Nm
- Torque of output shaft : 22 853.5 Nm

$$Shear\ stress = \frac{Torque \times Radius}{Polar\ moment\ of\ Inertia} \tag{10}$$

From the equation given, values of shear stress for the output/ input shaft are determined from the according the diameter:

- output = 132.2 MPa
- input = 12.04 MPa

This means that for the shaft diameter of 95 mm from the given motor frame (315D) is sufficient to withstand the input torque of 2027.7 Nm. However, this diameter is very large compare to the required input torque but it it the only size available for the given motor.
Moreover, the diameter of the output shaft is yet to be determined, and since this shaft size is not standardized a formula to find the minimum diameter is given below.

$$\frac{Torque}{Polar\,moment\,of inertia} = \frac{Shear\,modulus}{Radius} = 0.10\ m \tag{11}$$

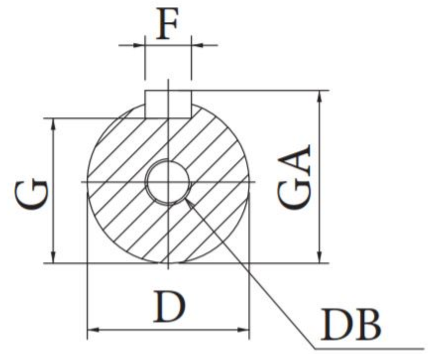


Figure 38: Technical Drawing of the shaft

| HA | HE | K | L | LL | O | UB | SHAFT EXTENSION | | | | | | | | | BEARING | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | D | E | EB | EF | EG | F | G | GA | DB | DRIVE END | OPPCSITE DRIVE END |
| 45 | 485 | 28 | 1704 | 372 | 200 | M72x2 | 95 | 170 | 160 | 5 | 48 | 25 | 86 | 100 | M24 | 6322C3 | 6322C3 |

Figure 39: Shaft size and the bearings for frame 315D

## 11.8 Bearings

According to the shaft size, both of the bearings inside the motor are also given in the catalogue. This can be seen from Figure 39, where the bearing on the drive end is a 6322C3 bearing and on the opposite drive end too. Normally, one of the bearings is fixed on one end and free on the other to allow the tolerance of the shaft. Calculations for a bearing are done in Chapter 12, where there will be taken a loot at the bearing in the hinge point of the boom.

## 11.9 Key/Spline

For the input shaft, the dimensions of a key can be computed with the help of source [9]. Given are: maximum torque is 2027.7 Nm, rotational speed n= 1485 revs/min. and $d_{shaft} = 95$ mm. The material of the shaft is hardened steel and the key is made out of C45, because keys are usually made of this material.
First, the dimensions of the key are determined with source [9], the width is then 25 mm and the height is then 14 mm. This can also be seen in figure 38 and figure 39, the width (F) is 25mm and the height of the key (GA-G) 14 mm.
Next, is determining the material of the key. This is usually C45, so that material is chosen.
The yield strengths of the materials are:

- Key (C45) = 340 N/mm$^2$
- Shaft (hardened steel) = 345 N/mm$^2$
- Gear (martensitic stainless steel) = 800 N/mm$^2$

Then the allowable shearing stress for the key and the allowable compressive stress for the weakest material is determined. In typical industrial applications a safety factor N = 3 is chosen. The weakest material in this case is the shaft.
Allowable shear stress of the key: $\tau = \frac{0.5 \cdot \sigma_{yield}}{N} = \frac{0.5 \cdot 340}{3} = 56$ N/mm$^2$.

Allowable compressive stress of the shaft: $\sigma = \frac{\sigma_{yield}}{N} = \frac{345}{3} = 115$ N/mm$^2$.
Now, the minimum length of the key based on the shearing stress of the key can be computed:

$$\tau = \frac{F}{A_s} = \frac{2 \cdot T}{d \cdot b \cdot l} \tag{12}$$

Where $\tau$ is the allowable shear stress of the key, T is the torque, d is the diameter of the shaft, b is the width of the key and l is the length of the key.

$$l = \frac{2 \cdot T}{\tau \cdot d \cdot b} = \frac{2 \cdot 2\ 027\ 000}{56 \cdot 95 \cdot 25} = 30.48\ mm \tag{13}$$

The minimum length of the key based on the compressive stress of the weakest material can also be computed:

$$\sigma_d = \frac{F_t}{A_c} = \frac{4 \cdot T}{d \cdot h \cdot l} \tag{14}$$

Where $\sigma_d$ is the allowable compressive stress of the shaft, h the height of the key and the rest is the same as in the previous equation.

$$l = \frac{4 \cdot T}{\sigma_d \cdot d \cdot h} = \frac{4 \cdot 2\ 027\ 000}{115 \cdot 95 \cdot 14} = 53.01\ mm \tag{15}$$

So the minimum key length should be based on the compressive stress, then the key length is load carrying length + b = 53.01 mm + 25 mm = 78.01 mm. This is not a problem since the face width of the gear is only 108 mm. That means that more than half of the key is in contact with the gear. Then all the dimensions for the key are: (width x height x length) 25 mm x 14 mm x 78.01 mm.

For the output shaft, the same materials are used, only the diameter is different, namely 101 mm. Furthermore, the torque is much higher now: 22 853.5 Nm. The dimensions of the key are again determined with help of source [9], the width is then 28 mm and the height is 16 mm.

Now, the minimum length of the key based on the shearing stress of the key can be computed:

$$l = \frac{2 \cdot T}{\tau \cdot d \cdot b} = \frac{2 \cdot 22\ 853\ 500}{56 \cdot 101 \cdot 28} = 288.61\ mm \tag{16}$$

The minimum length of the key based on the compressive stress of the weakest material can also be computed:

$$l = \frac{4 \cdot T}{\sigma_d \cdot d \cdot h} = \frac{4 \cdot 22\ 853\ 500}{115 \cdot 95 \cdot 16} = 523.0\ mm \tag{17}$$

So the minimum key length should be based on the compressive stress, then the key length is 523.0 mm + 28 mm = 551.0 mm. This is a problem since the face width of the gear is only 108 mm.

A solution could be to choose another material for the key and the shaft, but it would probably not make such a big difference since the torque remains very high. Another solution would be to just change the face width of the gears, but this would also mean that the contact and bending stresses of the gears will change.

A better solution would be to make use of involute splines instead of a single key. Involute splines have a greater contact area and thus the stresses are equally distributed over outside of the shaft and the inside of the gear. Furthermore, these are made for the highest torque applications. There are standard involute splines available with an outer diameter of 98 mm, these have 10 tooth [10]. This can fit on the output shaft which has a diameter of 101 mm. Then only some machining on the output shaft has to be applied to get the right profile in it.

### 11.10 Conclusion

All the parts of the hoist have now been elaborated. A short overview will be given. For the hoist, 4 motors will be used, which all have a power output of 315 kW. A two-stage gearbox is used to reduce the rotational speed and increase the torque with a factor of 11.99. This leads to an inertia ratio of 0.07, which is relatively low for a hoisting system. Probably the motor could have been smaller, but because a safety factor of 1.2 was applied, a power of 294 kW was required. So the engine could have been smaller if no safety factor was applied. It was calculated that the two most important shafts, the input and the output shaft, have a diameter of 95 mm and 101 mm respectively. This led to the use of involute splines instead of a simple key for the output shaft to connect the shaft to the gear. The input shaft, as well as the bearings, are standardized from the catalogue.

# 12 Highly Loaded Hinge

For the highly loaded hinge in the design, the boom hinge is chosen to be designed. Some dimensions for this hinge already have limits, such as the width. The hinge should be able to fit on the boom and as the boom has a width of 750 mm, the hinge can have a maximum width of 750 mm too. This hinge consists of three main parts, as can be seen in Figure 40. The two smaller parts of the hinge that point forward are approximately half of the width of the bigger hinge that points backwards. This will also be done for designing the final dimensions for the hinge. For the first assumption, it is modelled that entire will fit exactly on the boom, so that will be the maximum width of 750 mm. The big middle part of the hinge will have a width of half of this width, so 375 mm. The two smaller part on the side will then also both be approximately $\frac{375}{2} = 187.5\ mm$. This will be a bit less because there will be a bearing in between the parts to reduce friction which also takes up some space.

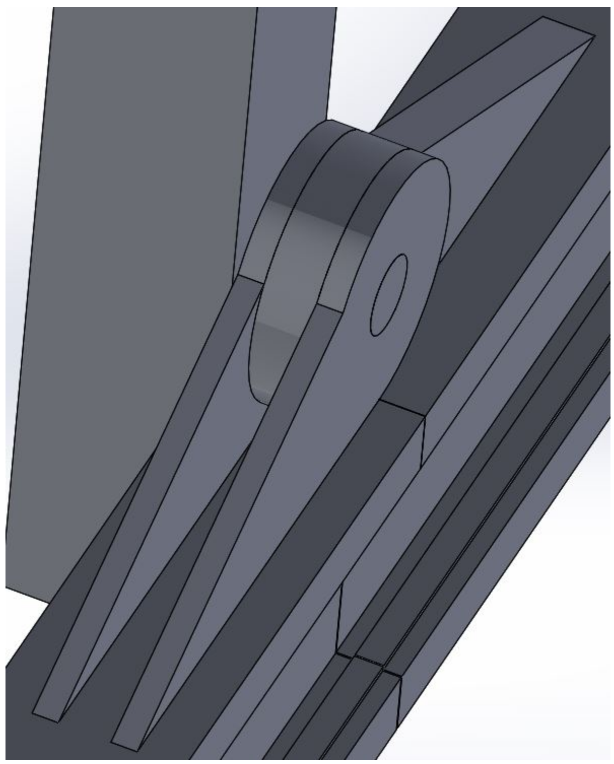

Figure 40: The chosen hinge point

It is also good to know that the hinge will carry none to almost no load when the crane is in normal operating mode. The weight of the dynamic boom is then exerted on the static boom and is also held by the supports. The hinge will, however, take a high loading when the boom has to be lifted. The total mass of the boom is then equal to 472 390 kg (obtained from the Matlab script) which is exerted on the two hinges. That means that one hinge will have to carry 236 195 kg at maximum.
In the hinge, there will be a plain bearing to make sure that the hinge can rotate. This bearing will have the same length as the width of the biggest hinge of the three, which will be equal to half of the width (375 mm again). With this dimension, the diameter of the bearing can also be determined. For most industrial applications, a length to diameter ratio (L/D) of 0.60 is recommended [4]. For this reason, the diameter of the bearing will be equal to $\frac{375}{0.60} = 625\ mm$.

## 12.1 Bearing Pressure

Then, the following formula will be used to calculate the bearing pressure: $p_L = \frac{F}{b \cdot d_L} \leq p_{L\ lim}$. The variables mean the following:

- $p_L$ is the bearing pressure $[N/mm^2]$
- $F$ is the bearing load $[N]$
- $b$ is the bearing length $[mm]$
- $d_L$ is the bearing diameter $[mm]$
- and $p_{L\ lim}$ is the allowable pressure $[N/mm^2]$

Plugging in the above-mentioned values and F = 236 195 · 9.81 = 2 317 072.95 N, it gives that $p_L$ is equal to $\frac{2\ 317\ 072.95}{375 \cdot 625} = 9.89$. This $p_L$ value will have to be compared with $p_{L\ lim}$ factors of certain materials to find out if it is within the limit. In Table 6, it can be seen that the obtained value of 9.89 is within the maximum $p_{L\ lim}$ for all the materials, which is a good sign. As can be seen, when Cu-Sn alloys are chosen, the bearings will have a big enough safety factor regarding the maximum $p_{L\ lim}$. This value would be equal to $\frac{25}{9.89} = 2.53$ which should be big enough to cover any fluctuations, which is why Cu-Sn alloys are chosen as the material for this bearing.

Table 6: $p_{L\ lim}$ values for several alloys

| **Plain bearing material** | $p_{L\ lim}$ **design (max.)** / $N/mm^2$ |
|---|---|
| Sn/Pb alloys | 5 (15) |
| Cu-Pb alloys | 7 (20) |
| Cu-Sn alloys | 7 (25) |
| Al-Sn alloys | 7 (18) |
| Al-Zn alloys | 7 (20) |

## 12.2 Lifetime

Next, the lifetime of the bearing can be determined with the following formula: $L_h = \frac{K_a}{p \cdot V} \cdot f_p \cdot f_c \cdot f_d \cdot f_m$. The variables have the following meaning:

- $L_h$ is the estimated life [h]
- $pV$ is the pV factor [MPa.m/s]
- $K_a$ is the application type constant
- $f_p$ is the load correction factor
- $f_c$ is the application characteristic factor
- $f_d$ is the bearing size factor
- $f_m$ is the shaft material factor

The last five factors can be determined using a file in which all sorts of situations are shown [11]. Also, the pV factor is determined with: $p \cdot V = p_L \cdot V$, where $p_L$ is the value used from the previous part and V is the linear velocity [m/s]. The linear velocity is determined by: V = $\frac{\pi \cdot d_L \cdot n}{60\ 000}$ in which $d_L$is the bearing diameter [mm] and n is the rotational speed [rpm].
For the rotational speed, another assumption will be done. It is assumed that the crane will be lifted to an angle of 63.89° in 2 minutes time, which was the maximum lifting angle according to SolidWorks, see Figure 86 in Appendix E. That means that the amount of rpm is equal to $\frac{63.89}{360} \div 2 = 0.0887$. Also, with the use of the file in which all factors are stated, the specific values are determined, which are shown below:

- $K_a$ = 800 (rotating load)
- $f_p$ = 1 (because p $\leq$ 10)
- $f_c$ = 3.0 (the bearing is constant immersed in a lubricant and the average temperature will be around 20°C.
- $f_m$ = 1.5 (the shaft material is first assumed to be made of hardened steel, as this has a higher yield strength than stainless steel)
- $f_d$ = 0.4 (the shaft diameter will be bigger than 150 mm)

With all those values, the formulas will be filled in:

- V = $\frac{\pi \cdot 625 \cdot 0.0887}{60\ 000} = 2.90 \cdot 10^{-3}$ m/s
- pV = $9.89 \cdot 2.95 \cdot 10^{-}3 = 2.87 \cdot 10^{-2}$ MPa.m/s
- $L_h = \frac{800}{2.92 \cdot 10^{-3}} \cdot 1 \cdot 3.0 \cdot 1.5 \cdot 0.4 = 50\ 140$ hours = 5.72 years.

So the estimated lifetime of the bearing is equal to 5.72 years, which seems like a reasonable number for such a big bearing.

The technical drawings for all the components of the hinge, are shown in Appendix E.

# 13 Highly Loaded Weld and Bolt Connection

## 13.1 Introduction

To manufacture the crane, the type of connection must be chosen. They may be welding, soldering or nuts among others. In this chapter, the analysis of two types of connections for two specific locations is carried out using the FEM-package in Matlab. This model was already explained before in Chapter 10. The result will be applied to choose the dimension of nuts or the thickness of welding in two specific connections points. The connection points chosen were specially selected because they have the highest stresses of the entire crane.
From Figure 41, it can be seen that the colours of beam A and B have the darkest red colour, which means those are the beams in which the highest tensile stress is applied. Explanation more in detail about, shapes, calculations, etc, are presented in the following sections.

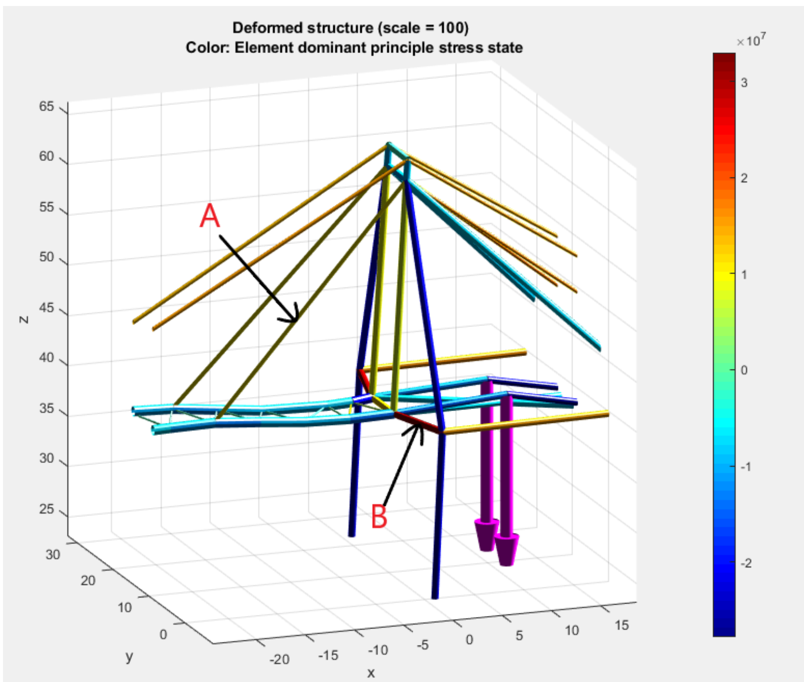


Figure 41: Beams with highest stresses

## 13.2 Bolts

There are different ways of connections and bolts is one of the most used in the biggest pieces. Therefore the purpose of this section is to prove that bolts used in the crane are strong enough for these stresses. In order to prove this, one of the sections with the biggest stresses is analyzed. In the starting point of beam 'A' shown in Figure 41, there is a hinge attached to the boom with bolts. The selected material for bolts, as explained in Chapter 9, is AISI 403: Nickel (0-0.6%) with a yield strength of 640 MPa.

**Process**
The first step is to determine the grade of steel for the bolts using the yield strength: 640 MPa of the selected material. With this information, a grade of 8.8 can be used as shown in the following calculation.

$$Grade\ 8.8 =$$
$$R_m = 8 \cdot 100 = 800$$
$$R_{el} = R_m \cdot 0.8 = 640\ N/mm^2$$

Now with the given grade and a force of 5 498 000 N, it can easily be estimated how many bolts are needed to get a reasonable area for those bolts. When 5 bolts are used:

$$Area = \frac{Force}{5 \cdot Grade}$$

$$Area = \frac{5\ 498\ 000 \cdot 2}{(5 \cdot 640)}$$

$$Area = 143.18\ mm^2$$

At the end safety factor of 2 has been taken into account, giving a final result of five bolts of M16.

## 13.3 Welding

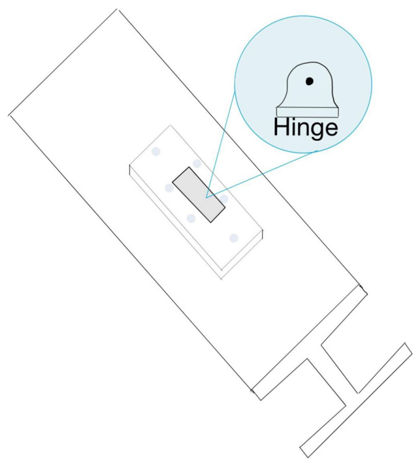


Figure 42: Bolts

The structure of the crane is shown in Figure 42 in which the hinge is supported and is assembled mostly with welds in the connection points. In order to choose the right type and right dimensions of the welding, it is important to consider in which moment the beams are subjected to the highest stresses. For example for beam A in Figure 41, the highest stresses are subjected when the container is in the furthest position from the structure. However, for the structure itself where the joints are connected with welds, the highest stresses are subjected when the container is in the middle of the beam, this was shown in Figure 30 in Chapter 10.

**Process**
There are four steps to find the correct thickness of the welding.

1. Select the type of welding that fits the most. For this purpose a fillet around the the B beam is selected, as shown in Figures 43 and 44 because it gives the longest contact with beam C.

2. Identify the type of stress to which the joint is subjected: As can be seen in Figure 41, the B-beam is attached to the boom. The main function for this beam is to connect the structure to the boom and carry the weight of boom on that section. The types of stresses present in the joint are tension, shear and bending.

3. Analyze the forces vectorially at the points of the weld, due to each type of stress: The values of the following forces and stresses are calculated with the FEM-package of Matlab. The script can be seen in Appendix C.2. In the next part, the three kinds of stresses will be analyzed.

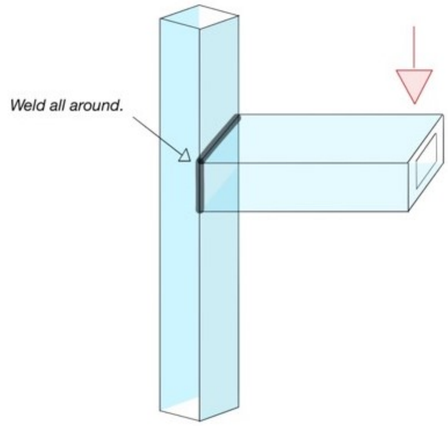


Figure 43: An all around weld

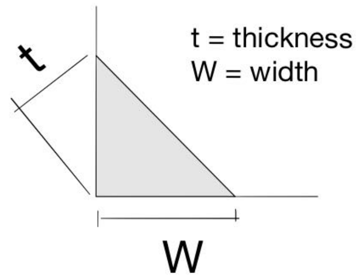


Figure 44: Fillet weld size notation, where t = sin(45)·w

**Normal**
The tension is taken from the highest principal stress ($\sigma_P$) of the beam B, this value is $3.18 \cdot 10^7$ Pascals. To get the normal force that generates tension, the cross-sectional area (A) must be considered. The dimension of the hollow beam B is a width (w) of 0.55 m, a height (h) of 0.8 m and a thickness (t) of 0.125 m. Those values give the normal force to be equal to:

$$F_N = \frac{\sigma_P}{A} = 2.67 \cdot 10^7\ N \tag{18}$$

From this, the normal stress ($\sigma_W$) felt on the welding is given by:

$$\sigma_W = \frac{F_N}{A_w} = 9.34 \cdot 10^6\ Pa \tag{19}$$

Where $A_w$ is the geometry factor given by $A_w = 2h+2w$. This refers to a square beam with welding all around the contact surface.

**Shear**
The force that causes shear to beam B are the forces acting on the z-axis, in the other end as seen in Figure 43. This force is given by the weight of the beam and the container. From the FEM model the shear force ($F_s$) is $9.22 \cdot 10^5$ N.
The shear stress ($\sigma_S$) if given by;

$$\sigma_S = \frac{F_s}{A_w} = 3.22 \cdot 10^5 Pa \tag{20}$$

**Bending**
The forces that causes bending is the same shear force multiplied by the length from the point in the structure to the connection point on the boom. This length is 8.75 m. So the bending force ($F_b$) is:

$$F_b = F_s \cdot L = 8.06 \cdot\ 10^6\ N \cdot m \tag{21}$$

And the bending stress caused by this force is:

$$\sigma_B = \frac{F_b}{S_w} = 1.09 \cdot\ 10^7\ Pa \tag{22}$$

Where $S_w$ is the geometry factor, given by $S_w = h \cdot w + \frac{b^2}{3}$

All the stresses have now been calculated.

4. Combine the forces vectorially at the point of the weld where the force appears to be maximum. The shear force and the bending force are combined because they are acting in the axis parallel to the structure, while the normal force is acting perpendicular to the structure. The total stresses are given by the following formula:

$$\sigma_T = \sqrt{\sigma_W^2 + (\sigma_S + \sigma_B)^2} = 1.56 \cdot 10^6 \ Pa \tag{23}$$

5. Determine the weld thickness by dividing the maximum force by the allowable stress. The allowable stress of an electrode type E110 is 228MPa. The throat width (t) of the welding is:

$$t = \frac{1000 \cdot \sigma_T}{\sigma_{allow}} = 6.85 \ mm \tag{24}$$

Now, applying a safety factor (N) of 3:

$$t = t \cdot N = 20.56 \ mm \tag{25}$$

The width of the fillet welding is then: $\frac{t}{sin(45)} = 24.16 \ mm$

To select the final weld size, table 20.4 from the book Machine Elements [4] is used.

**TABLE 20–4 Minimum weld sizes for thick plates**

| Plate thickness (mm) | Minimum leg size for fillet weld (mm) |
|---|---|
| ≤12.7 | 4.76 |
| >12.7 − 19.05 | 6.35 |
| >19.05 − 38.1 | 7.94 |
| >38.1 − 57.15 | 9.525 |
| >57.15 − 152.4 | 12.7 |
| >152.4 | 15.875 |

Figure 45: Table for welding size

From the third option on the table, the minimum size of the welding would be 7.94 mm.

**Conclusion**
The type of connection is chosen according to the condition that specific joint is undergoing during operation, in this chapter only two joints were analyzed. For beam A in the connection to the boom, 5 bolts M16 were chosen and for the beam B connected to the structure welding with leg size of minimum 7.96 mm is required.
The bolts and nuts chosen in these sections may be the same for parts of the crane that face the same conditions as the ones analyzed here. For instance, the process to find the specific type of connection for other different parts, follow the same structure taking into consideration their own working conditions.

# 14 Linear Guideways

The forces on the linear guideways need to be calculated to get an idea of which stresses they need to handle. The rails do not handle the forces alone but do that together with the I-beam. The calculation has been done using the FBD in Figure 46. In this figure it can be seen that there is a force, F, going downwards. This force is the weight of the hoist and the container hanging onto it times the gravitational acceleration divided by two. It is divided by two as there is only looked at one side of the hoist and the weight of the hoist and container is distributed over the two beams. This force is the only force working on the rails, so for the trolley to be in equilibrium this means that the force that the guideway needs to handle is the same force F. This is:

$$F_g = m \cdot a = \frac{109.52 \cdot 10^3}{2} \cdot 9.81 = 537\ kN$$

So the force acting on one rail is 537 kN. This force is exerted through two of the wheels of the trolley to the rail. That means that at the position of the wheels the rail would deform a lot if it was not situated on the I-beam. But because the rails are situated on the I-beam, the deformation of the rail is equal to the deformation of the I-beam. The deformation of the I-beam with the trolley being at a specific position has already been calculated and is shown in Chapter 10.

As the force is not working directly underneath the wheel, the force will also create a moment M on the wheels. This moment needs to be counteracted again by the rails. The moment is the force times the distance between the force and the wheels. As there are two wheels on each side of the boom, the moment that is acted out by one wheel is half of the size of the total moment.

$$M = \frac{F \cdot L}{2} = \frac{537 \cdot 1.4}{2} = 376\ kN \cdot m$$

The rails are bolted to the I-beam. This is done as welds can cause an uneven surface, that might prevent the wheel from being able to ride over the rails. The SolidWorks model is shown in Figures 47 and 48. The dark red part is the rails on which the wheel moves. The rails are 2 cm high and 11 cm wide. As the position where the rail is located is 25 cm wide and there should still be enough tolerance on both sides of the rail to fit the wheel, the 11 cm was chosen.

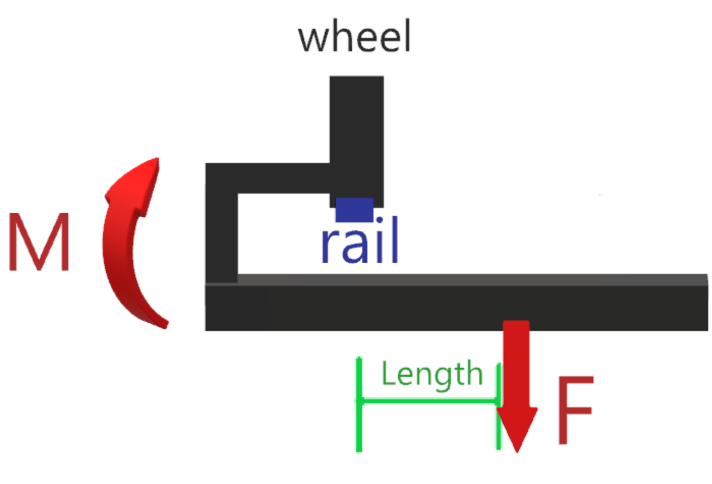


Figure 46: FBD of forces on the rails

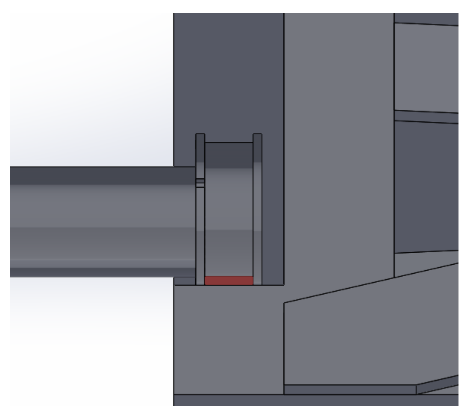

Figure 47: SolidWorks model of rails

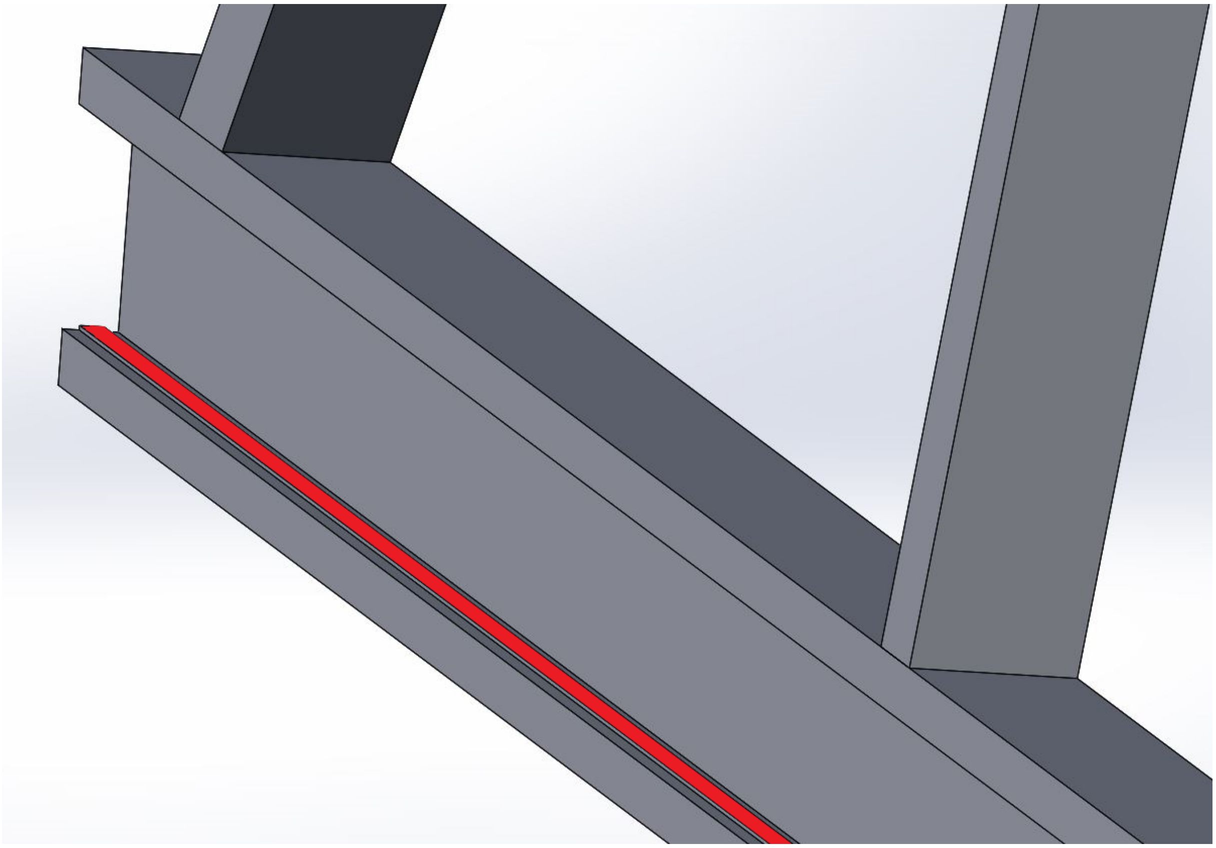

Figure 48: Rails situated on the I-beam

# 15 Driveshaft Design

First, the driveshaft will be designed. This is mainly focused on determining all the different diameters of the shaft. As shown in Figure 37, the driveshaft in the gearbox is connected to gear Z4, meaning that the calculated values of this gear will be used for designing the driveshaft. There are several formulas for calculating the diameter of a shaft. When the section of the shaft only has significant shear stress, the formula is given by:

$$D = \sqrt{\frac{2.94 \cdot K_t \cdot V \cdot N}{S_n'}} \tag{26}$$

In this formula, $K_T$ is the stress concentration factor, which is different for specific cases, such as 3.0 for a retaining ring groove (located at a gear for instance to prevent horizontal movement) and 2.5 for a sharp fillet (which is used for the other parts of the shaft). V is the vertical shear force in N. N is the design factor, which is generally taken as 2 and $S_n'$ is the modified endurance strength, which is determined by taking the endurance strength $S_n$ and multiplying its value with certain factors. The factors will be explained a bit further down.
The formula, however, for calculating the diameter of the shaft when there is bending stress and/or torsional shearing stress, is given by:

$$D = [(\frac{32 \cdot N}{\pi})\sqrt{(\frac{K_T \cdot M}{S_n'})^2 + \frac{3}{4} \cdot (\frac{T}{S_y})^2}]^{\frac{1}{3}} \tag{27}$$

For using the above formula, first a material will be chosen. This material is chosen to be cold drawn steel AISI 1050. For this material, the yield strength ($S_y$) and ultimate tensile strength ($S_u$) are determined, which are equal to 580 MPa and 690 MPa respectively. N is the design factor, $K_T$ is the stress concentration factor and $S_n'$ is the modified endurance strength.
The endurance strength is given by source [4], which is determined according to the ultimate tensile strength. This endurance strength is equal to approximately 265 MPa. The above described factors are $C_m$ (material factor), $C_{st}$ (type of stress factor), $C_R$ (reliability factor) and $C_s$ (size factor). With the help of [4], the values are respectively determined to be 0.8, 1, 0.75 (taking a reliability of 99.9% as the crane should be very safe) and $C_s$ turns out to be 0.7753. The last value is obtained by taking a trial diameter of 100 mm, which gives $C_s$: 0.859 - 0.000837·100 = 0.7753. All of the above gives the modified endurance strength $S_n'$ to be equal to: $265 \cdot C_m \cdot C_{st} \cdot C_R \cdot C_s = 123.27$ MPa.
The torque (T) in the shaft is equal to the torque transmitted by the last gear. This is gear number 4, which has a torque of 22 853 Nm which is equal to 22 853 000 Nmm. Then, the last value that has to be determined in order to calculate the diameter of the shaft, is the bending moment M. This bending moment will be determined by using a static analysis, which is shown in the next section.

## 15.1 Static Analysis

The goal of the static analysis is to calculate the internal forces in the shaft. In this way, the maximum stresses can be calculated and put into Mohr's circle. First, the forces that are acting on the shaft are determined and after that, the equilibrium equations for the moment and for the forces in the y-direction are used. The shaft has two bearings with both a vertical force and a gear with a vertical force and a force in the z-direction. The bearings do not take axial force as there is no force acting in the horizontal direction in the shaft. If this were to be the case, bearings that are able to take axial loads (such as thruster bearings) would have to be incorporated. Furthermore, the cable with container acts on the shaft as well in a vertical direction. Lastly, there is a torsion force in horizontal direction applied by the gear and an equal reaction torque at the place of the load.

As can be seen in Figure 49 on the next page, there are several forces acting on the shaft. As described above, starting from the left: The vertical force of the bearing, the vertical force and force in z-direction of the gear, the vertical force of the load and the vertical force of the bearing.

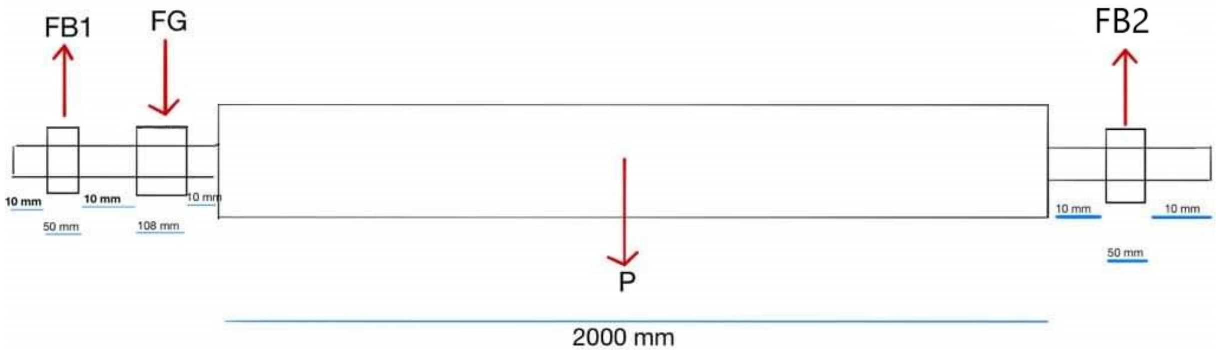


Figure 49: FBD of the shaft with the forces and dimensions indicated (not on scale)

The forces generated by the gear are calculated by the following formulas, given by the powerpoints of Machine Elements:

- $W_t = \frac{T}{(D/2)}$
- $F_{Gx} = W_t$
- $F_{G9y} = \tan\theta \cdot W_t$, in which $\theta$ is the pressure angle, which is generally 20 degrees

D is equal to the pitch diameter of the $4^{th}$ gear, which is 472 mm. This value of D, gives $W_t$ equal to $\frac{22\ 853\ 000}{472/2} = 96\ 834.75$ N. That means that $F_{Gx}$ is equal to 96 834.75 N and $F_{Gy}$ is equal to $\tan(20) \cdot W_t = 35\ 244.97$ N. Then, P is equal to the force generated by the load, which is 537 195 N as shown in previous Chapters. There are now only two unknowns left, which are both the radial forces that are generated by the two bearings. These unknowns can be determined by using the equilibrium equation for the moment at the first bearing and the equilibrium equation for the forces in the y-direction.

$$\sum M@_{B1} + \circlearrowleft = -F_{Gy} \cdot 89 - P \cdot 1153 + F_{B2} \cdot 2188 = 0$$

Which gives $F_{B2}$ is equal to $\frac{F_{Gy} \cdot 89 + P \cdot 1153}{2188} = 284\ 516.74$ N

Now, the forces in the other bearing can be calculated using the equilibrium of forces in y-direction:

$$\sum F_y \uparrow + \ = F_{B1} - F_g - P + F_{B2} = 0$$

So $F_{B1} = F_g + P - F_{B2} = 287\ 923.25\ N$

With the above-shown values, the shear diagram (V(x)), the torque diagram (T(x)) and the bending moment diagram (M(x)) can be made, as shown in Figures 50 till 52. With those values and some assumed dimensions, the bending moments can be determined. As can be seen from the shear diagram, there are three main sections in the shaft. The bending moment will be calculated at the point where the line changes its path, because a force is applied at those places. The first part of the shear diagram goes from 10+25 = 35 mm to 10+50+10+54 = 124 mm. This is the length from the middle of the first bearing to the middle of the gear, which can be seen in Figure 49. For this, some dimensions were assumed. Those assumed values are the widths of the bearings (50 mm) and the distances between the different components, as seen in Figure 49. The face width of the gear was already calculated in Chapter (108 mm).
Now, using the magnitude of the force in this part of the shaft (287 923.25 N), the bending moment is calculated to be: $287\ 923.25 \cdot (124 - 35) = 25\ 625\ 169.72$ Nmm. This moment acts on the middle of the

gear. Then, a force facing downwards is exerted by the gear, meaning that the shear force at the point of the gear will decrease with the magnitude of the shear force created by the gear. The new value of the shear force is then equal to 287 923.25 - 35 245 = 252 678.25 N. This shear force is exerted onto the shaft until the middle of the drum is reached, which is where the load is applied. The distance from the middle of the gear to the middle of the drum is equal to (10+50+10+108+10+1000) - 124 = 1188 - 124 = 1064 mm. With those two values, the bending moment at the load turns out to be 25 625 169.72 + 252 678.25 · 1064 = 294 474 833.4 Nmm. This value can be checked by calculating the bending moment at the position of the load by coming from the other (right) side of the shaft. The value is then equal to (2223 - 1188) · 284 516.75 = 294 474 830.8 Nmm, which only differs 2.6 Nmm from the other calculated value, which means that the value is correct.

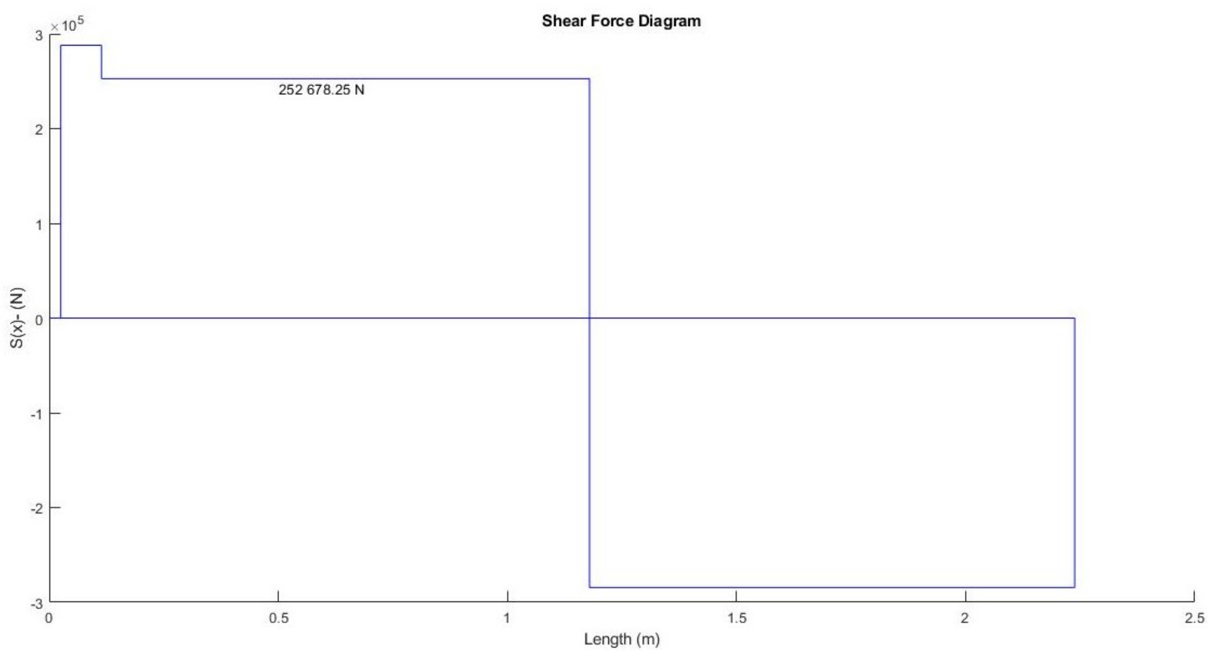


Figure 50: Shear diagram

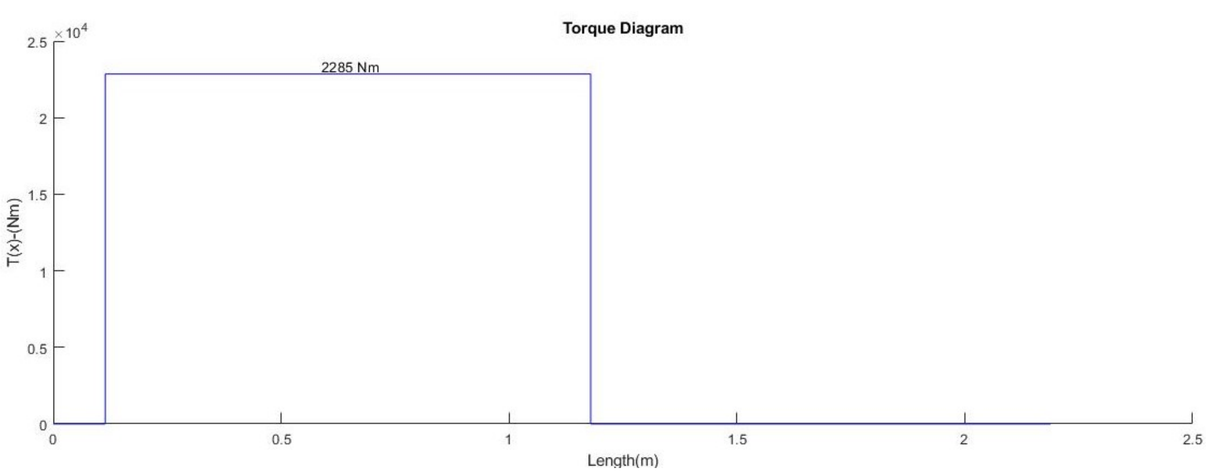


Figure 51: Torsion diagram

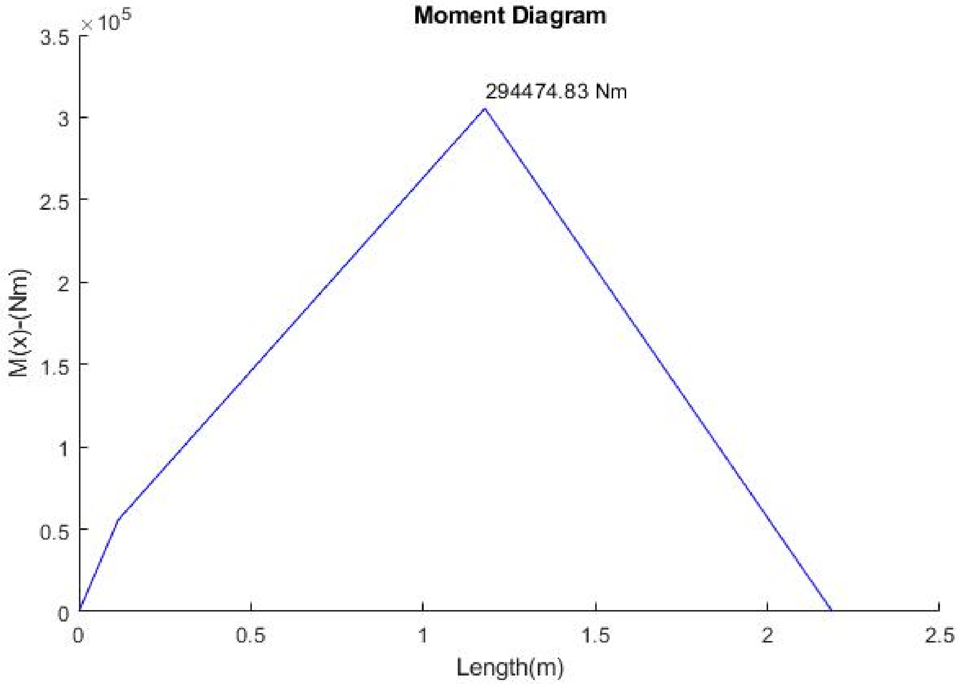


Figure 52: Moment diagram

### 15.1.1 Continuation Design Driveshaft

With the two calculated bending moments and the bending moment at the bearings being equal to 0, the various diameters of the shaft can be calculated by using Equation 26 for the first and fourth section and Equation 27 for the two remaining sections.

- For the first section: D = $\sqrt{\frac{2.94 \cdot 2.5 \cdot 287\ 923.25 \cdot 2}{123.27}} = 185.30$ mm
- For the second section: D = $[(\frac{32 \cdot 2}{\pi})\sqrt{(\frac{3.0 \cdot 25\ 625\ 169.72}{123.27})^2 + \frac{3}{4} \cdot (\frac{22\ 853\ 000}{580})^2}]^{\frac{1}{3}} = 233.46$ mm
- For the third section: D = $[(\frac{32 \cdot 2}{\pi})\sqrt{(\frac{2.5 \cdot 294\ 474\ 833.4}{123.27})^2 + \frac{3}{4} \cdot (\frac{22\ 853\ 000}{580})^2}]^{\frac{1}{3}} = 495.51$ mm
- For the fourth section: D = $\sqrt{\frac{2.94 \cdot 2.5 \cdot 284\ 516.74 \cdot 2}{123.27}} = 184.20$ mm

It can now be seen that the diameter of the thickest part of the shaft is equal to 495.51 mm. This will be the diameter of the drum, which is higher than the first assumed 300 mm. A drum with a diameter of almost half a meter, still seems reasonable for such a large construction.

Besides the calculated bending moment for the gear in the y-direction, there is also a bending moment in z-direction as the gear also exerts a force in that direction on the shaft. The length of the arm on which this force $F_{Gx}$ acts is equal to the radius of the shaft. This is $\frac{233.46}{2}$ which is 116.73 mm. This gives that the bending in the z-direction is equal to 96 834.75 · 116.73 = 11 303 520.37 N. This is a factor $\frac{25\ 625\ 169.72}{11\ 303\ 520.37} = 2.27$ smaller than the bending moment in the y-direction, meaning that the shaft should mainly be designed to withstand the bending moment in the y-direction.

With all the calculated diameters, a model is made in SolidWorks and a technical drawing will also be made, as shown in the next section. The obtained values for the bending moment and shear force will among others be used to construct Mohr's circle.

## 15.2 Shaft model and technical drawing

According to the diameters calculated in the previous sections, the model of the shaft made in SolidWorks and the technical drawing including dimensions is shown below. The spline can be seen on the shaft in Figure 53. A bigger version is shown on page 69.

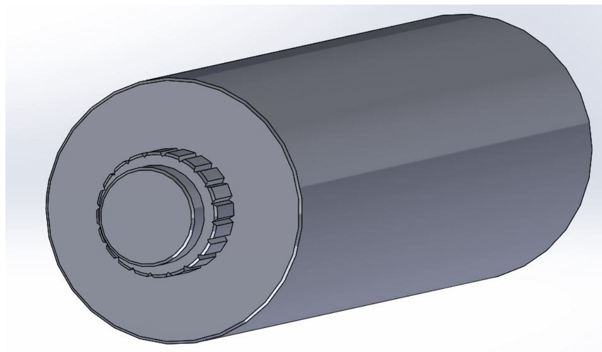

Figure 53: Model of the shaft made in SolidWorks

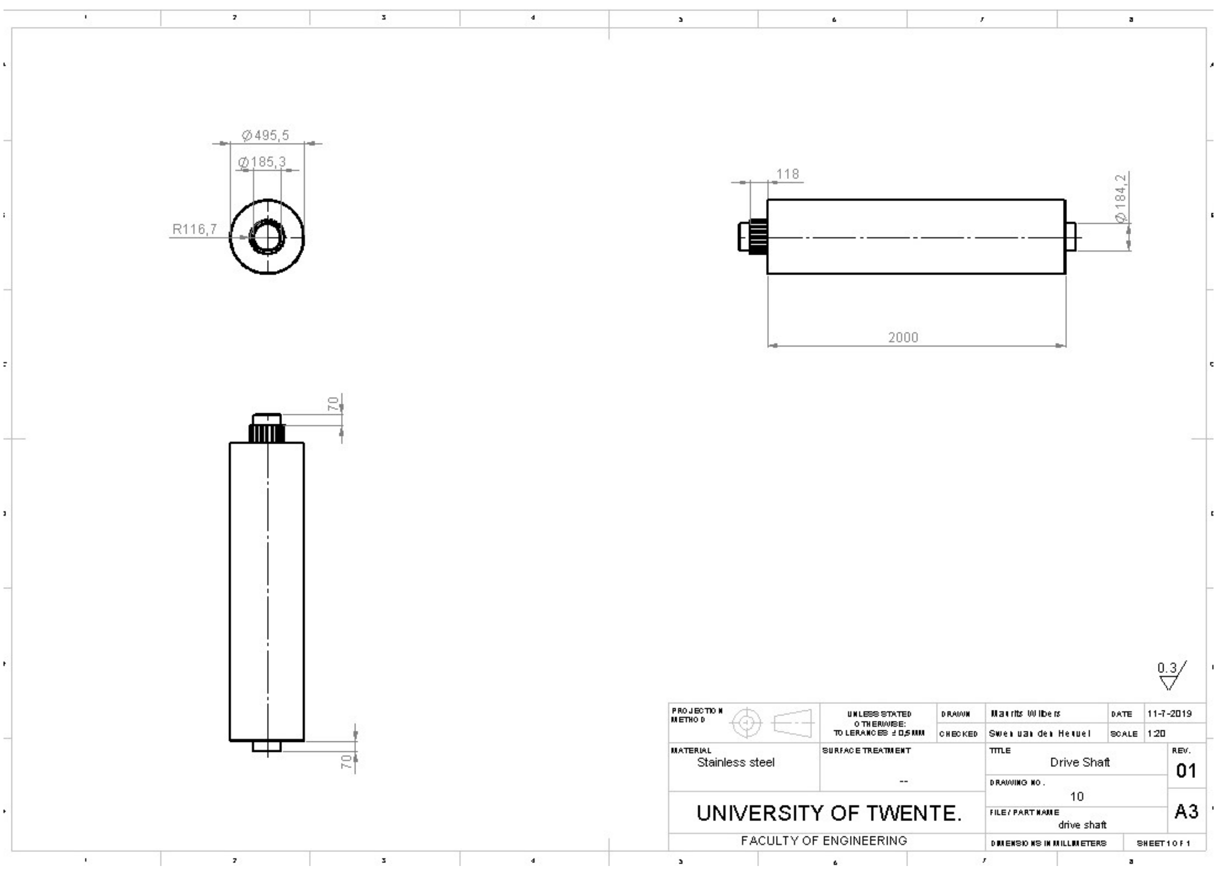


Figure 54: Technical drawing of the shaft

## 15.3 Stress Distribution

At the critical section of the shaft, the stress distributions are drawn. This critical section is at the place of the load, and then at the top of the shaft, displayed with a red dot. At this section, the force of the load is applied, which is the highest force in the shaft. Also, at the top of the shaft, the torsion and the bending moment are maximum. There is no shear force, but since the bending moment has more impact than the shear force, the critical section will be at the top instead of the middle of the shaft, where the shear force is maximum. This can be seen by using Figures 55 till 59. Figure 59 shows a part of the combined distribution of shear and torsion, where the shear increases the downwards arrows in the torsion distribution.

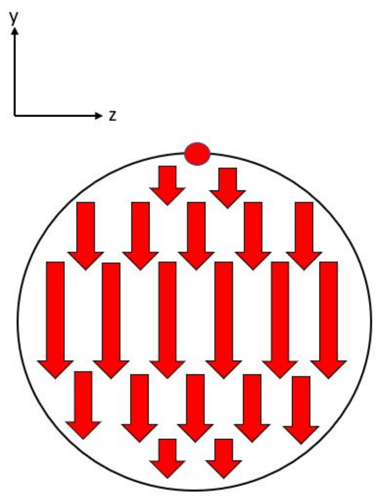


Figure 55: Shear distribution

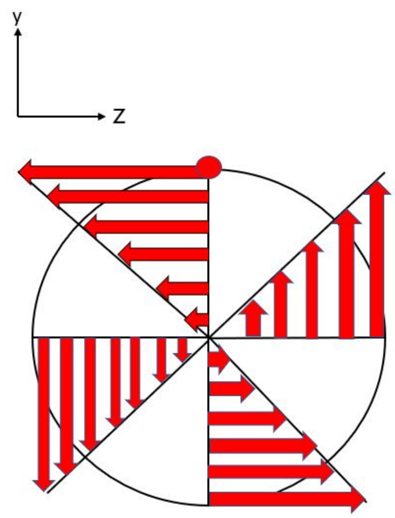


Figure 56: Torque distribution

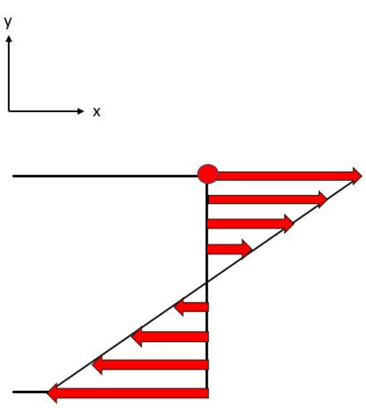


Figure 57: Bending moment distribution in the x-direction

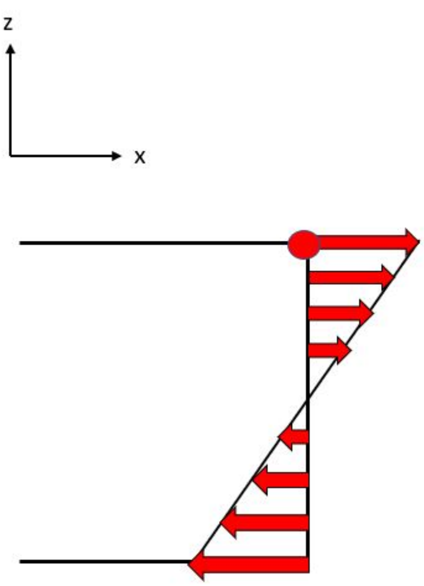


Figure 58: Bending moment distribution in the z-direction, which has a lower impact than the bending moment in the x-direction

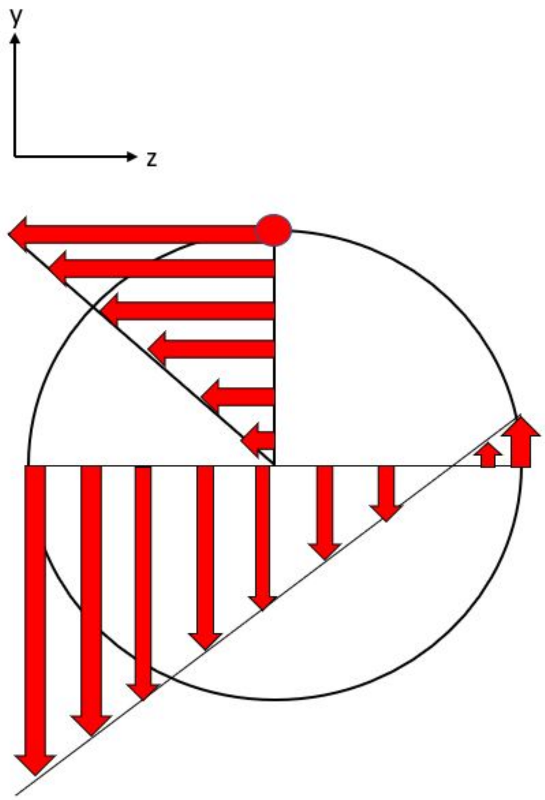


Figure 59: Combined shear and torsion distribution

### 15.4 Mohr's Circle

The shear stress due to the torsion moment is calculated using the following formula:

$$\tau(x,y) = r \cdot \frac{T(x)}{J}$$

In which r is the radius, T(x) is the torsion moment and J is the polar moment of area. Filling in the values that are determined gives:

$$\tau(x,y) = \frac{495.51}{2} \cdot \frac{22\ 853\ 000}{\frac{\pi}{2} \cdot \frac{495.51}{2}^4} = 0.96\ MPa$$

The shear stress due to the shear force is calculated using the following formula:

$$\tau(x,y) = \frac{S(x) \cdot Q(y)}{I \cdot t(y)}$$

In which S(x) is the shear force, Q(y) is the second moment of area, I is the moment of inertia and t(y) is the thickness of the circle, taken at the middle of the circle (diameter). Q(y) for a circle is equal to $\frac{2}{3} \cdot r^3$ and I = $\frac{\pi}{4} \cdot r^4$. As the shear force at the top of the shaft is 0 however, there will not be any shear stress at this point due to the shear force.

The normal stress due to the bending moment is calculated using the following formula:

$$\sigma(x,y) = -y \cdot \frac{M(x)}{I}$$

In which y is the distance from the centroid to the outer diameter (radius), M(x) is the bending moment and I is again the moment of inertia. Filling in all the values obtained from the previous sections, with the bending moment being negative, gives:

$$\sigma(x,y) = -\frac{495.51}{2} \cdot \frac{-294\,474\,833.4}{\frac{\pi}{4} \cdot \frac{495.51}{2}^4} = 24.65\ MPa$$

With the calculated values, Mohr's circle can be made, which is displayed in Figure 60.

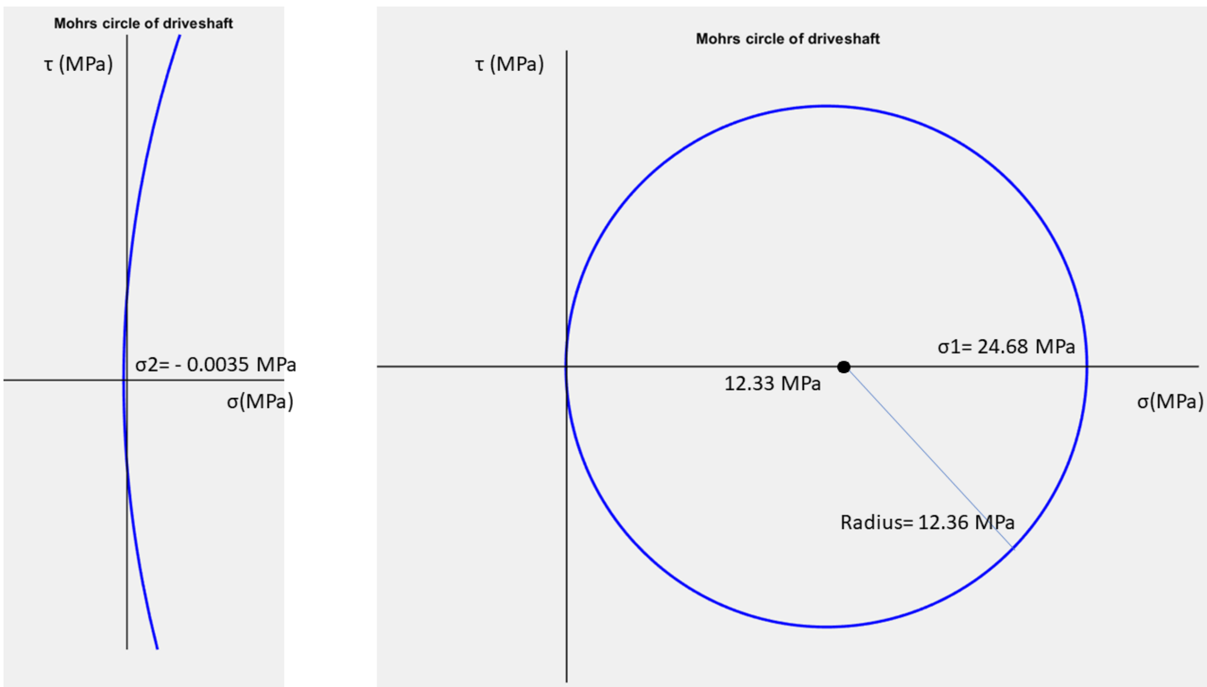


Figure 60: Mohr's circle

The center of the circle $\sigma_{avg}$ is at $\frac{\sigma_x+\sigma_y}{2} = \frac{24.65}{2} = 12.33\ MPa$. The radius R is then equal to $\sqrt{(\frac{\sigma_x+\sigma_y}{2})^2 + \tau_{xy}^2}$, which is $\sqrt{12.33^2 + 0.96^2} = 12.36\ MPa$. And lastly, the principal stresses $\sigma_{1,2}$ are equal to $\sigma_{avg} \pm R$, so $\sigma_1 = \sigma_{avg} + R = 24.68\ MPa$ and $\sigma_2 = \sigma_{avg} - R = -0.0035\ MPa$.

It can be seen that the normal stress has a big impact, which is expected. The bending moment due to the load (which creates the normal stress) is bigger than the applied torsion by the motor (which creates the shear stress).

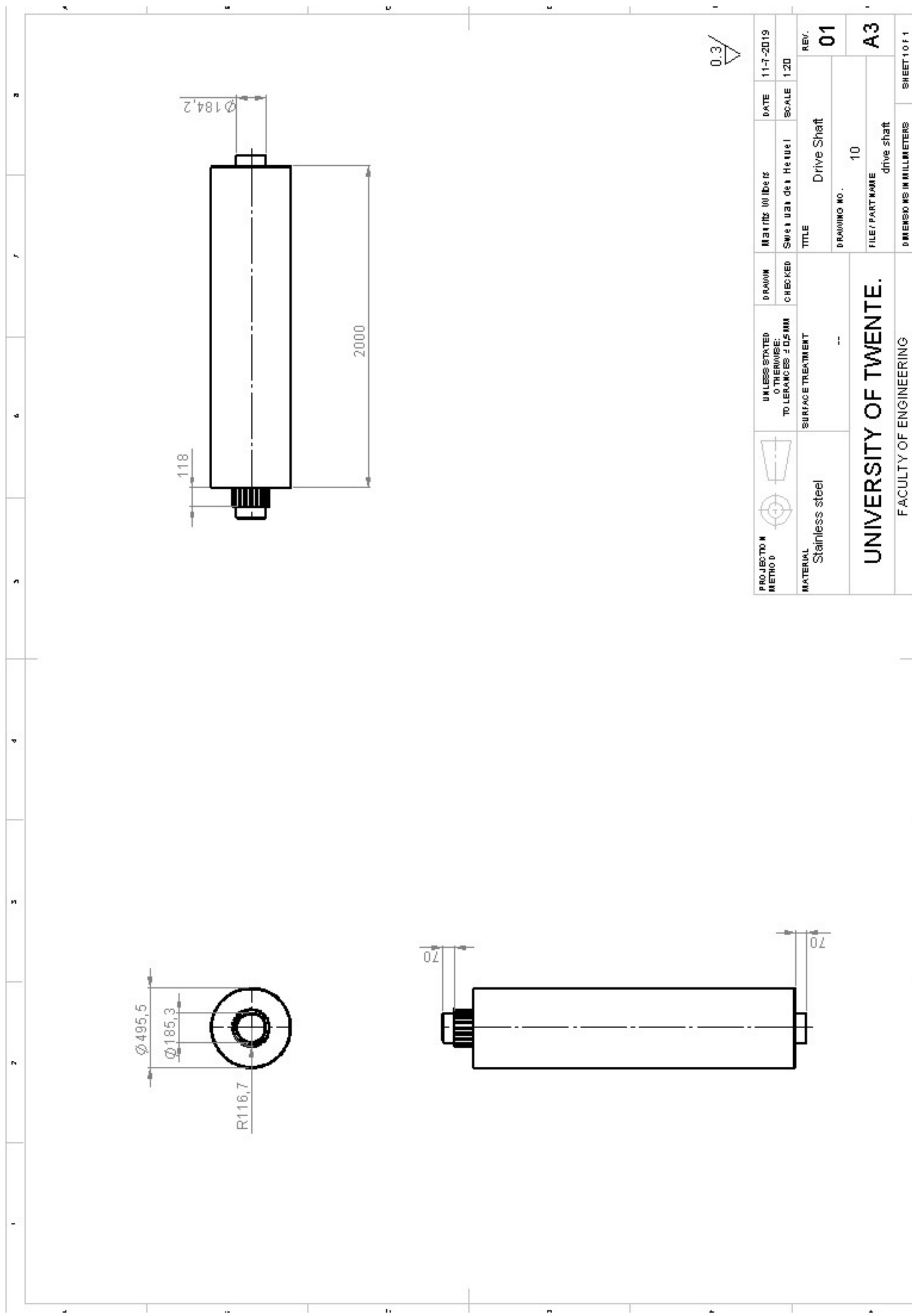


Figure 61: Technical drawing of the shaft

# 16 Conclusion

The goal of this project was to design a high profile STS-crane. By first setting the requirements for the crane, creating concepts and determining some initial dimensions, a start in the design was made. Then using all information determined before, materials were chosen and it was decided on how to connect all parts. On top of that, a suitable motor was chosen for the hoist. With that, the final dimensions were determined and the design was optimized. For the final details, the machine elements were determined using calculations and the catalogue and specific important parts were analyzed in more detail to show the crane will not fail. In conclusion, it has been found that the crane deflection stays within the requested boundaries and contains the wanted machine elements. Furthermore, materials have been selected that will allow the crane to function as desired. Finally, the dimensions of the crane are as required, leading to a final, detailed STS-crane design.

# References

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# Appendices

## A Boom

### A.1 First Concept

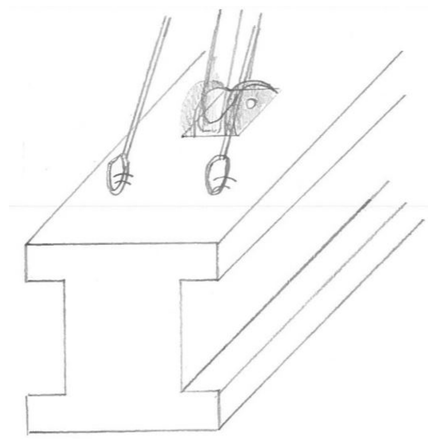


Figure 62: Zoom in of point 1, the front of the boom

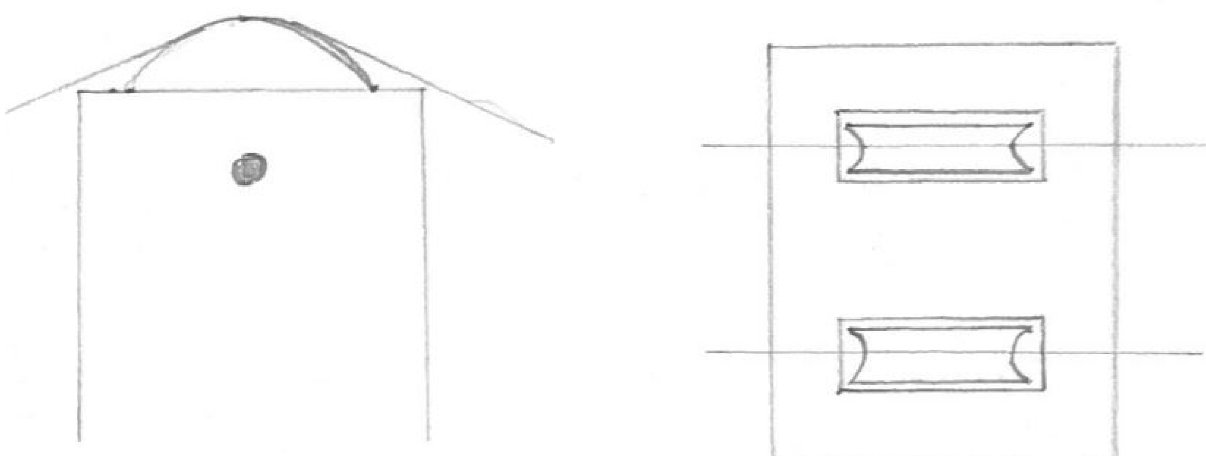


Figure 63: Zoom in of point 2, the top of the crane

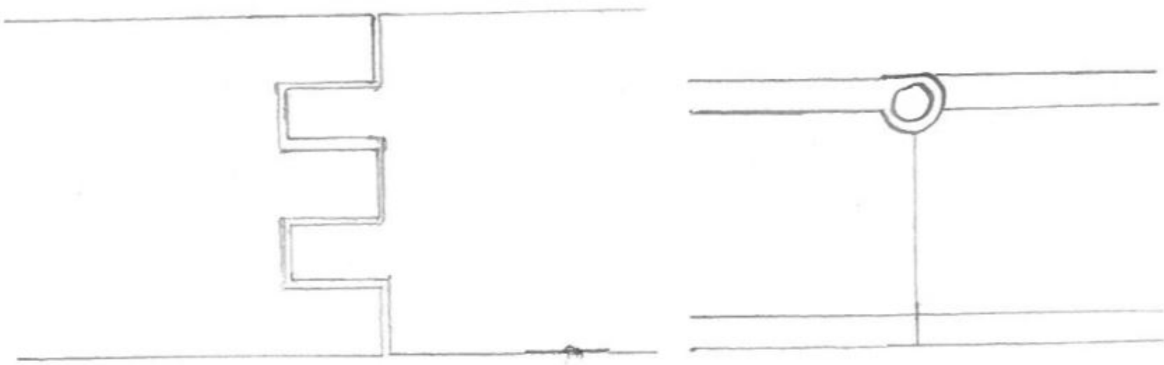


Figure 64: Zoom in of point 3, the hinge

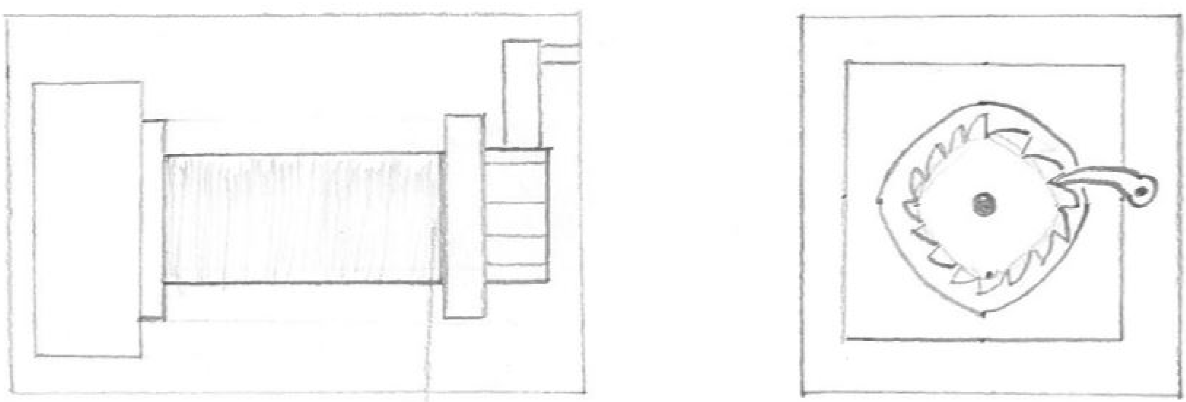


Figure 65: Zoom in of point 4, the machinery housing

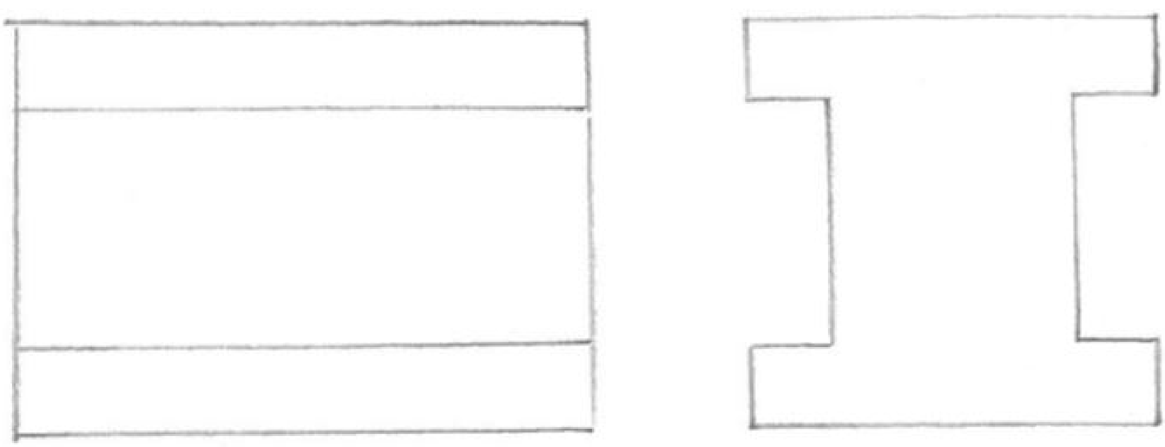


Figure 66: Zoom in of point 5, the frame of the beam

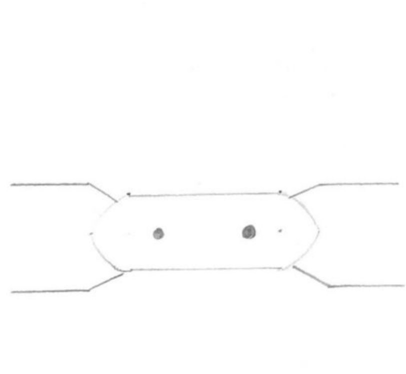


Figure 67: Zoom in on point 6, the connection point of the support

## A.2 Second Concept

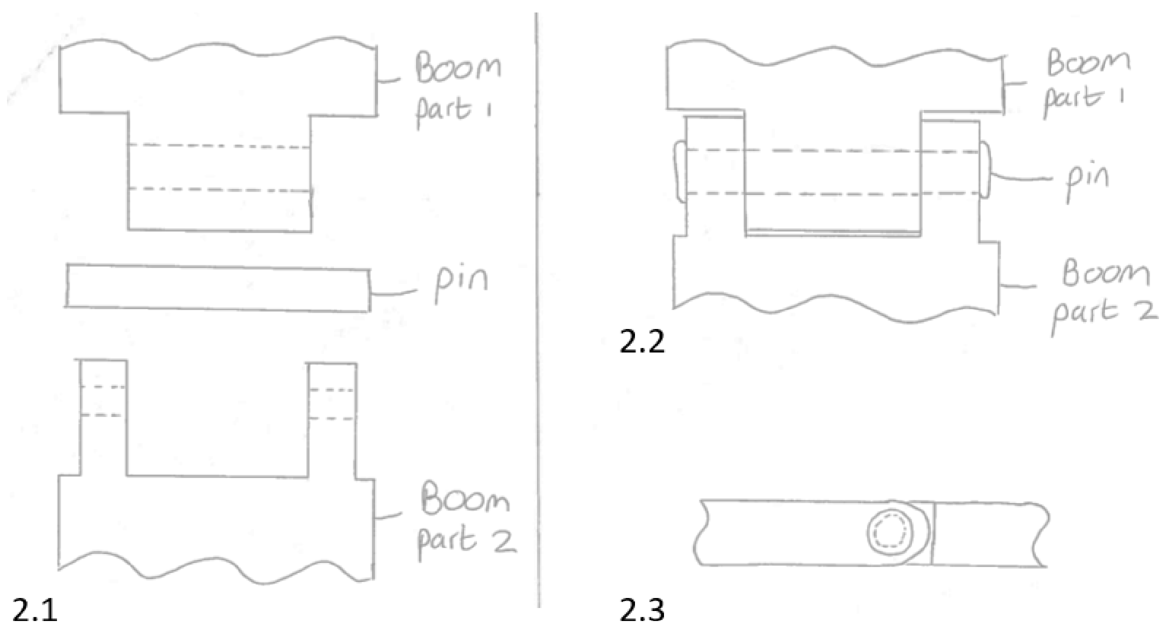



Figure 68: Top view disassembled hinge (2.1), top view assembled hinge (2.2), side view hinge (2.3)

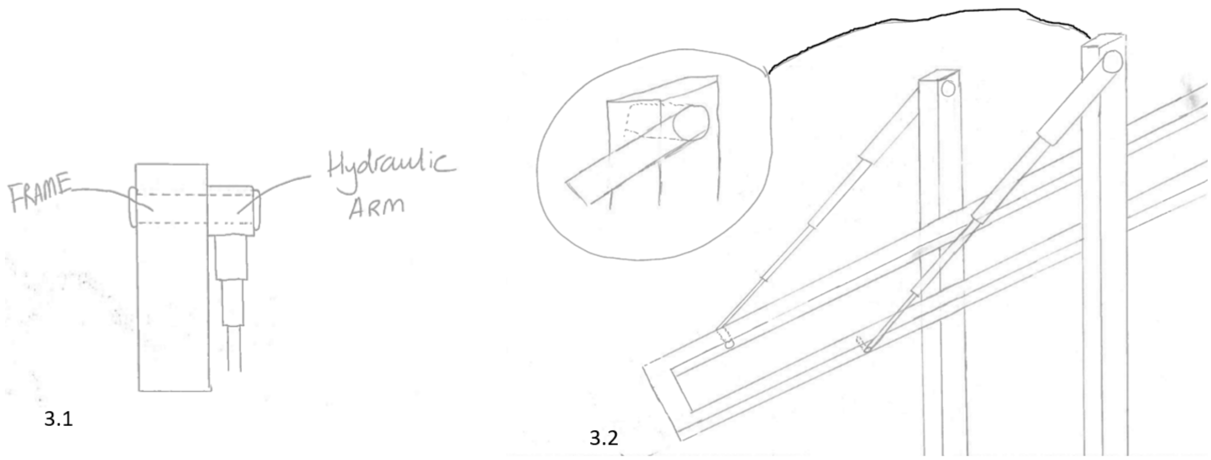



Figure 69: Front view boom support (3.1) and 3d view boom support (3.2)

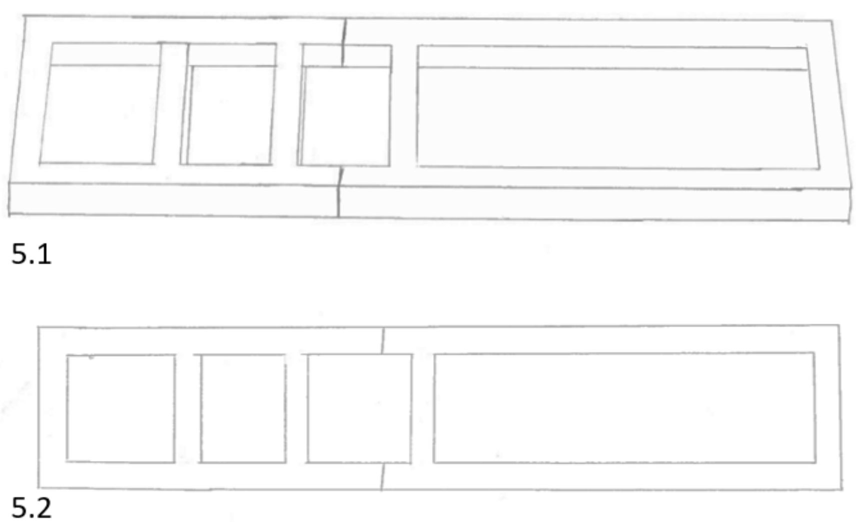
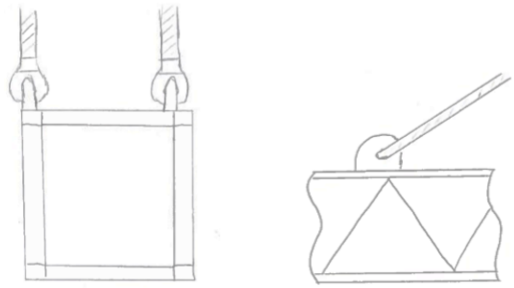


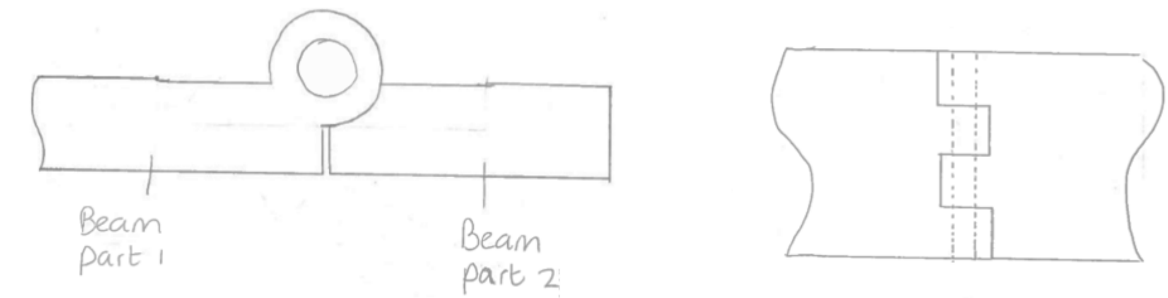

Figure 70: 3d view boom (5.1) and top view boom (5.2)

## A.3 Third Concept



1.1 1.2

Figure 71: Front view boom support (1.1) and side view boom support (1.2)




2.1 2.2

Figure 72: Side view hinge (2.1) and top view hinge (2.2)

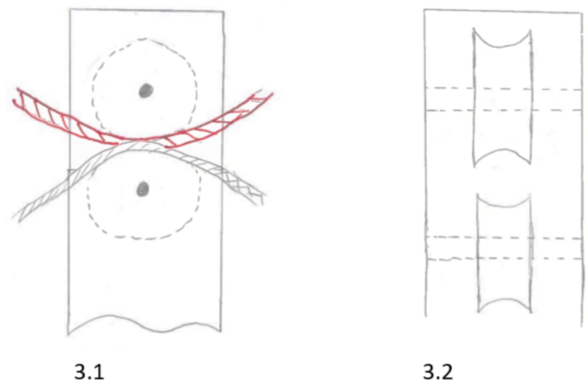



Figure 73: Side view boom support (3.1) and front view boom support (3.2)

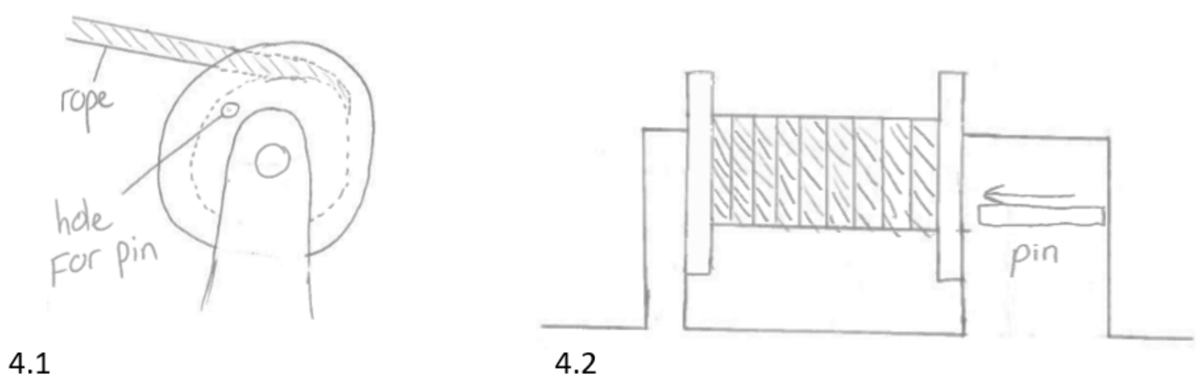



Figure 74: Side view locking mechanism (4.1) and front view locking mechanism (4.2)

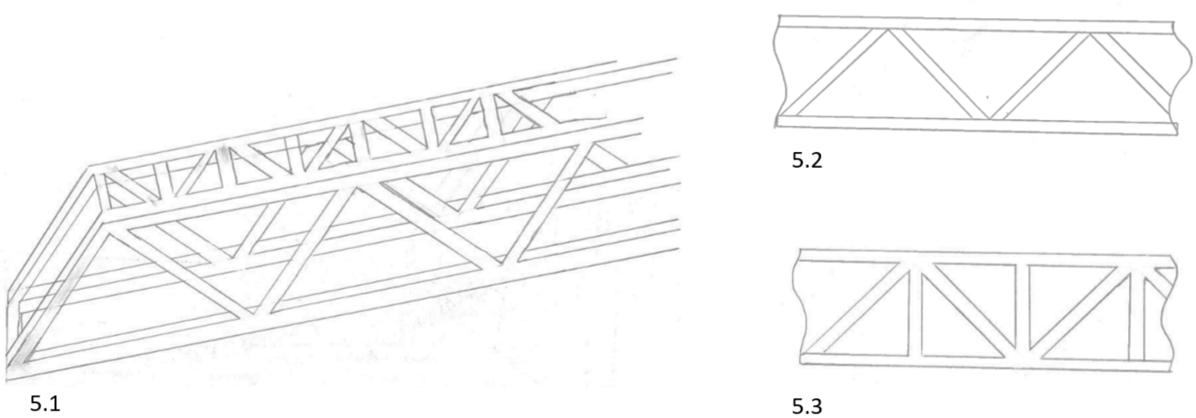



Figure 75: 3d view boom (5.1), side view boom (5.2) and top view boom (5.3)

# B Dimensioning

## B.1 Figures

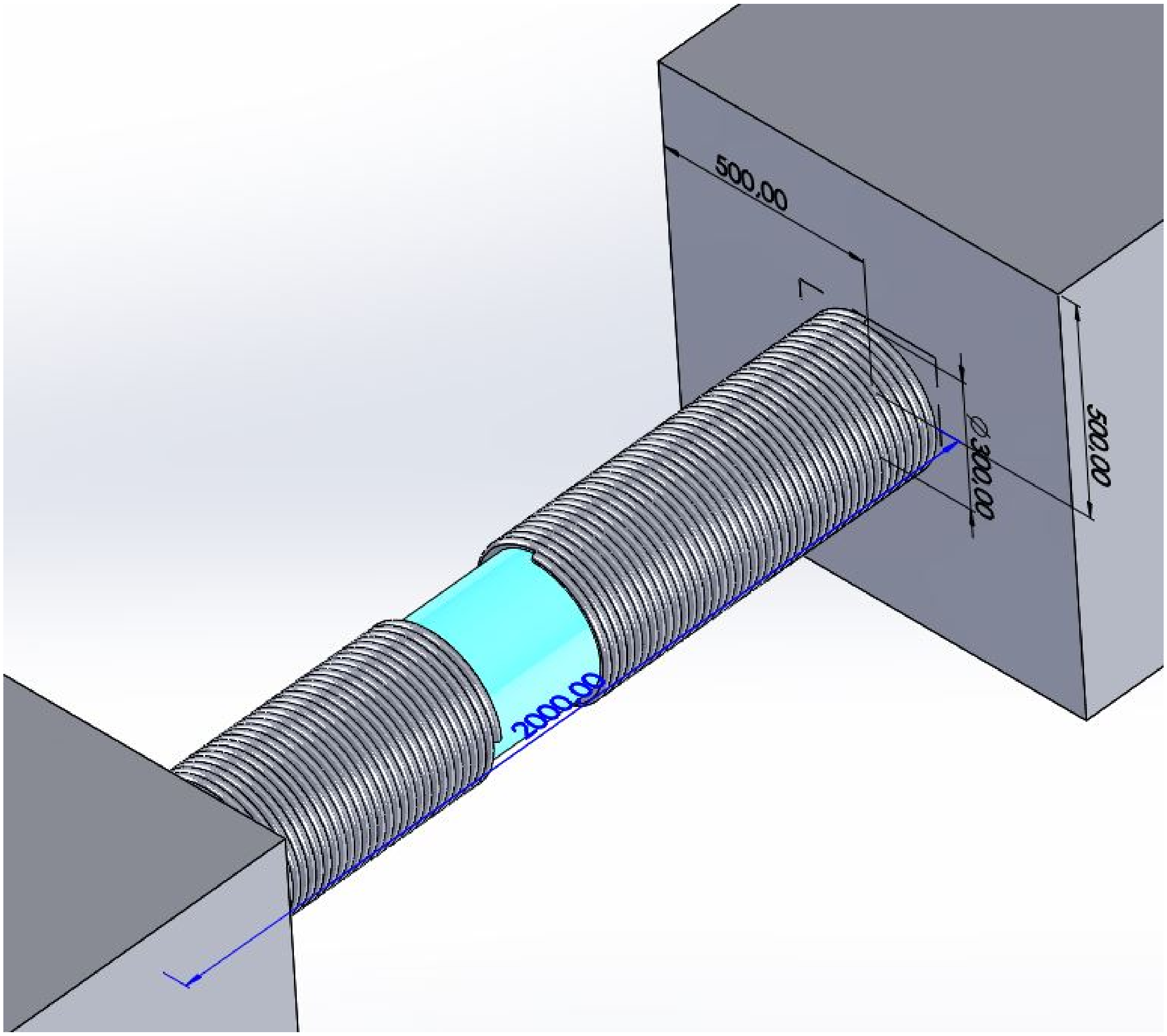



Figure 76: The length and diamter of the drum

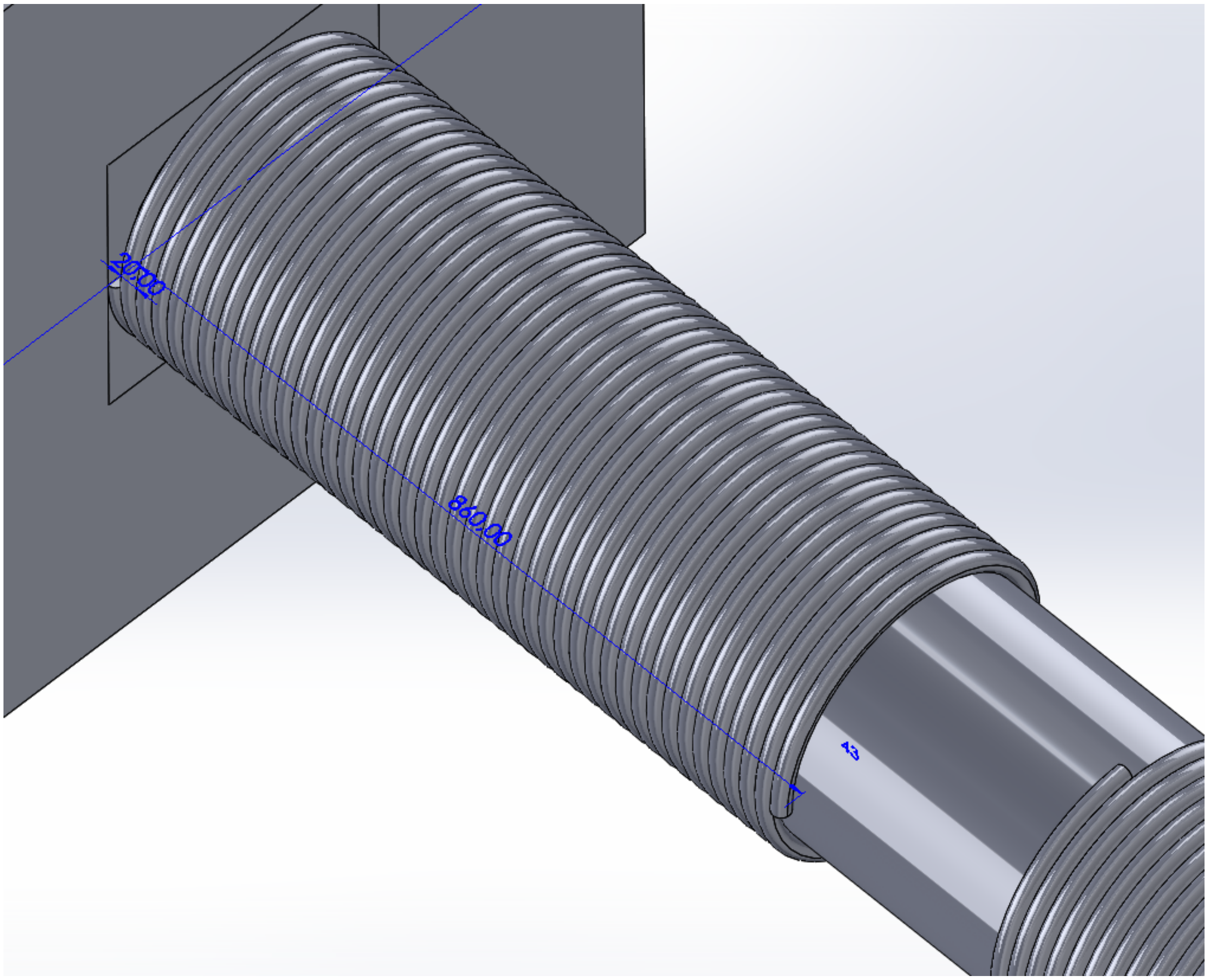



Figure 77: The drum of the hoist, where it can be seen that there are 43 windings (at the end, the cable will go over itself and go back to the front which can not be seen in the picture)

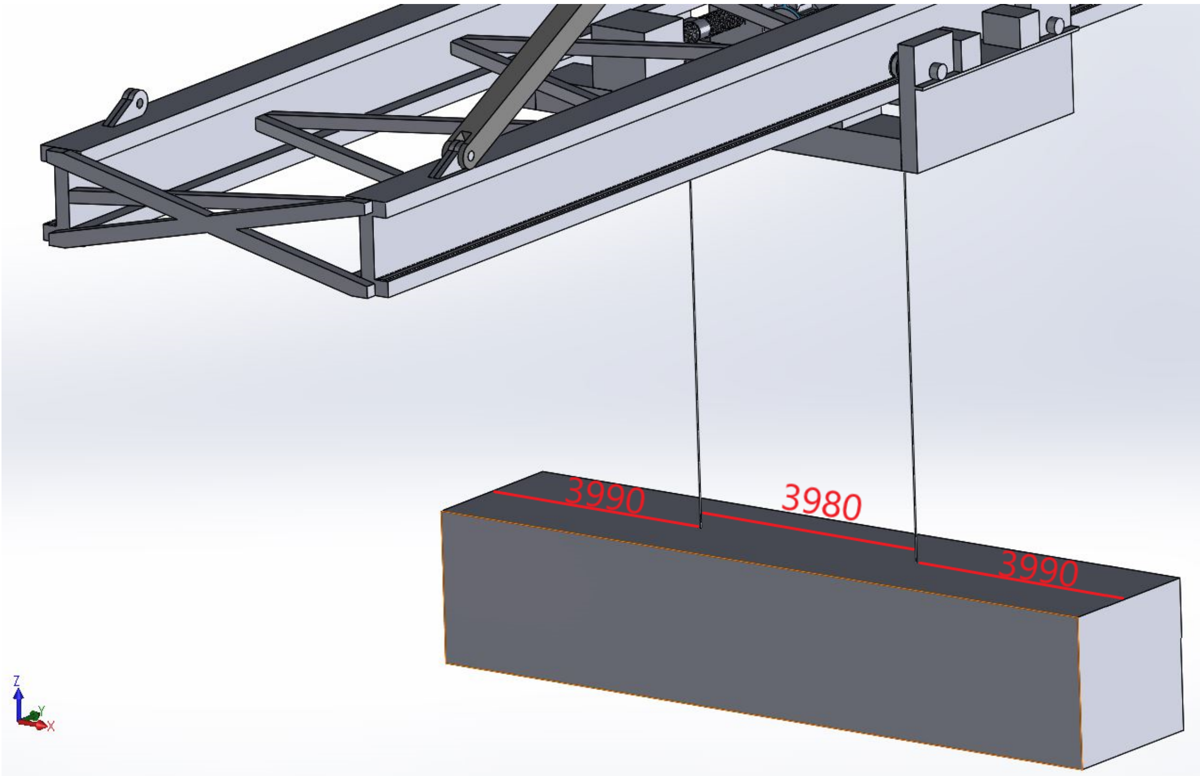



Figure 78: The dimensions of the cables coming from the hoist that are attached on the (invisible) spreader on the container. As the distance is approximately equal to three times 4 meters, the load is distributed equally

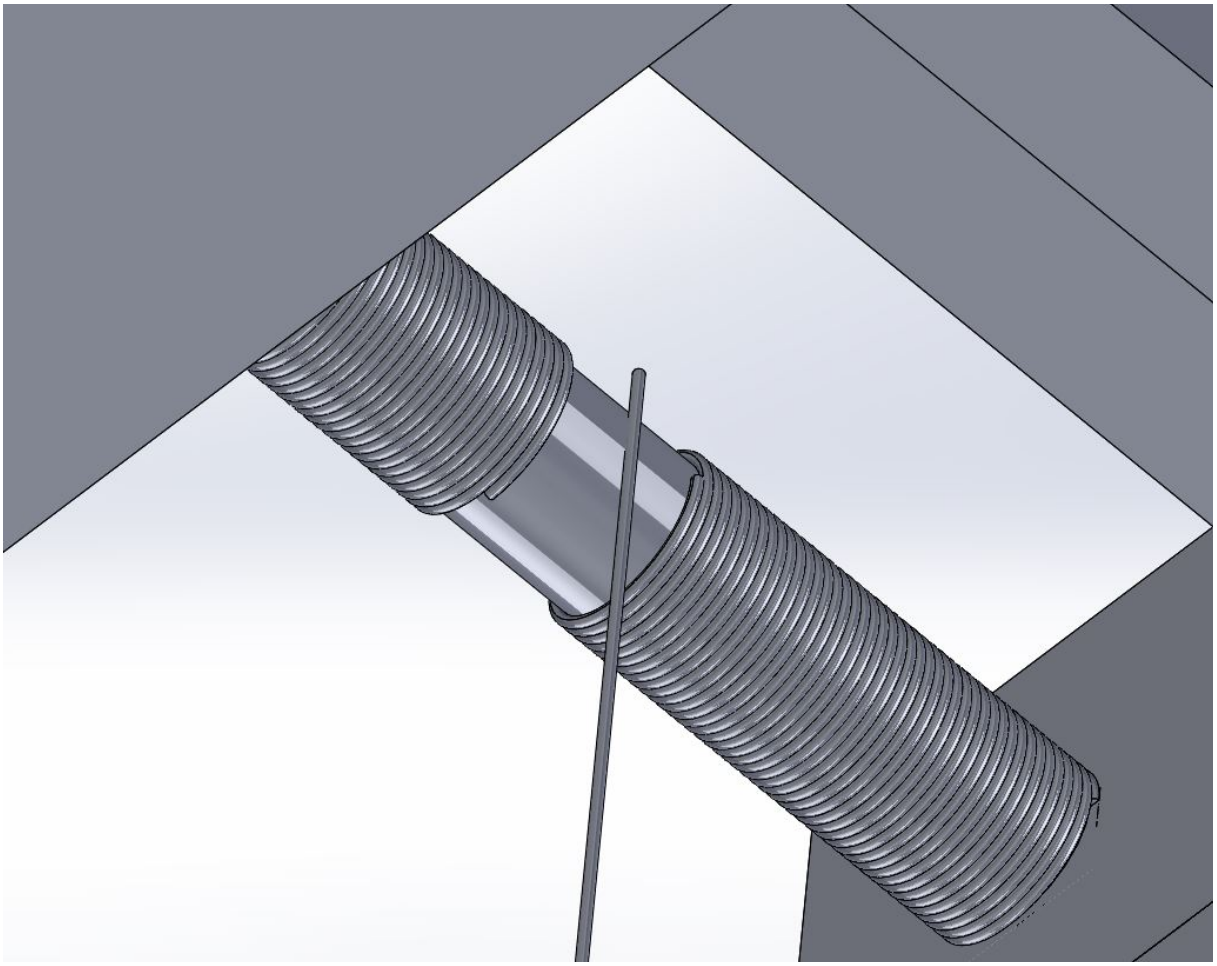


Figure 79: The cable coming from the container will fit perfectly on the drum from the hoist with the mentioned dimensions

# C Strength and Stiffness Calculations

## C.1 FBD's and Matlab code

First, the code is shown where the force is exerted at the end of the boom:

```
%% Definition of material properties
% define Elastic modulus, take for all elements the same E. E is here
% defined for steel. nu is the poisson factor, necessary for calculating
%the shear modulus.
addpath support_functions_Frame_3D
E = 210.5e9;                                                    %S420 steel
nu = 0.3;
G = E/(2*(1+nu));
rho = 7800;

%%% define coordinates of the nodes [meter]
xyz_nodes = [0,0,0
             23,0,0
             0,23,0
             23,23,0
             0,0,15
             23,0,15
             0,23,15
             23,23,15
             0,0,40
             23,0,40              %10
             0,23,40
             23,23,40
             0,8.75,63
             0,14.25,63
             0,8.75,65
             0,14.25,65
             0,8.75,40            %node 17 of new boom
              4.57,8.75,40
              9.14,8.75,40
              13.71,8.75,40       %20
              18.28,8.75,40
              22.85,8.75,40
              27.42,8.75,40
              31.99,8.75,40
              36.56,8.75,40
              41.13,8.75,40
              45.7,8.75,40
             0,14.25,40           %node 28
              2.29,14.25,40
              6.86,14.25,40       %30
              11.43,14.25,40
              16,14.25,40
              20.57,14.25,40
              25.114,14.25,40
              29.71,14.25,40
              34.28,14.25,40
```

```
            38.85,14.25,40
            43.42,14.25,40
            45.7,14.25,40       %node 39
           23,8.75,40           %nodes of the back, 40
           27.57,8.75,40
           32.14,8.75,40
           36.71,8.75,40
           41.28,8.75,40
           45.85,8.75,40
           50.42,8.75,40        %final of back 8.75 in y
           23,14.25,40          %nodes of the back
           25.3,14.25,40
           29.87,14.25,40
           34.42,14.25,40       %50
           39.01,14.25,40
           43.58,14.25,40
           48.15,14.25,40
           50.42,14.25,40
           11.5,8.75,40         %Nodes above land
           11.5,14.25,40];      %Idem

       %Connection 15 25
%% define which nodes are connected by elements and
elements = [1, 5
           2, 6
           3, 7
           4, 8
           5, 9
           6, 10
           7, 11
           8,12
           6, 12
           %5,6
           %6,7          %10
           %6,8
           %8,7
           %5,7
           %9,6
           %9,7
           %11,8
           9,13
           9,10
           11,12        %20
           9,17
           10,40
           11,14
           10,13
           12,14
           13,14
           14,16
           13,15
           16,15
           13,21        %30
```

```
14,33
15,27
16,39
13,43
14,50
15,45
16,53
17,28
28,11
40,47           %40
47,12
13,17
14,28
17,55
55,40
56,47
28,56
17,47
28,40
17,18
18,19
19,20           %50
20,21
21,22
22,23
23,24
24,25
25,26
26,27
28,29
29,30
30,31           %60
31,32
32,33
33,34
34,35
35,36
36,37
37,38
38,39
27,39
17,29           %70
29,18
18,30
30,19
19,31
31,20
20,32
32,21
21,33
33,22
22,34           %80
34,23
```

```
            23,35
            35,24
            24,36
            36,25
            25,37
            37,26
            26,38
            38,27       %end boom with triangles
            40,41       %90
            41,42
            42,43
            43,44
            44,45
            45,46
            47,48
            48,49
            49,50
            50,51
            51,52       %100
            52,53
            53,54
            54,46       %end of back boom
            40,48
            48,41
            41,49
            49,42
            42,50
            50,43
            43,51       %110
            51,44
            44,52
            52,45
            45,53
            53,46];

%% define outer diameter and wall thickness of each element [meter]
% C = [outer diameter, wall thickness]
h1 = 0.25;  %Height base I beam
h2 = 1;     %Height middle part I beam
b1 = 0.75;  %Width base
b2 = 0.25;  %Width middle part I beam
h3 = 0.1;   %Height triangles
b3 = 0.1;   %Width triangles
h4 = 0.2;   %Height support
b4 = 0.2;   %Width support
h5 = 0;     %Filling for the squared beams
b5 = 0;
h6 = 0.2;   %Height support support, changed to 0.4 from 0.75
b6 = 0.2;   %Width support support, changed to 0.4 from 0.75
h7 = 0.9;   %Outer heigth and width
b7 = 0.9;
h8 = 0.55;  %Inner height and width (So wall thickness all around of 20 cm)
b8 = 0.55;
```

```
b9 = 0.8;
h9 = 0.8;

C =[h5 , b5 , h7 , b7
    h5 , b5 , h7 , b7
    h5 , b5 , h7 , b7
    h5 , b5 , h7 , b7
    h5 , b5 , h7 , b7
    h5 , b5 , h7 , b7
    h5 , b5 , h7 , b7
    h5 , b5 , h7 , b7
    %h8 , b8 , h7 , b7   %Beams without any load were removed and beams
    %that were in the way of transporting the container to land
    %h5 , b5 , h7 , b7        %10
    %h8 , b8 , h7 , b7
    %h8 , b8 , h7 , b7
    h5 , b5 , h7 , b7
    %h8 , b8 , h7 , b7
    %h8 , b8 , h7 , b7
    %h5 , b5 , h7 , b7
    %h8 , b8 , h7 , b7
    h8 , b8 , h9 , b9
    h8 , b8 , h9 , b9
    h8 , b8 , h9 , b9        %20
    h8 , b8 , h9 , b9
    h8 , b8 , h9 , b9
    h8 , b8 , h9 , b9
    h8 , b8 , h9 , b9
    h8 , b8 , h9 , b9
    h8 , b8 , h9 , b9
    h8 , b8 , h9 , b9
    h8 , b8 , h9 , b9
    h8 , b8 , h9 , b9
    h5 , b5 , h4 , b4        %30
    h5 , b5 , h4 , b4
    h5 , b5 , h4 , b4
    h5 , b5 , h4 , b4
    h5 , b5 , h6 , b6
    h5 , b5 , h6 , b6
    h5 , b5 , h6 , b6
    h5 , b5 , h6 , b6
    h8 , b8 , h9 , b9
    h8 , b8 , h9 , b9
    h8 , b8 , h9 , b9      %40
    h8 , b8 , h9 , b9
    h8 , b8 , h9 , b9
    h8 , b8 , h9 , b9
    h8 , b8 , h9 , b9
    h8 , b8 , h9 , b9
    h8 , b8 , h9 , b9
    h8 , b8 , h9 , b9
    h8 , b8 , h9 , b9
    h8 , b8 , h9 , b9
```

```
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2          %50
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2          %60
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2
h5 , b5 , h3 , b3          %70
h5 , b5 , h3 , b3
h5 , b5 , h3 , b3
h5 , b5 , h3 , b3
h5 , b5 , h3 , b3
h5 , b5 , h3 , b3
h5 , b5 , h3 , b3
h5 , b5 , h3 , b3
h5 , b5 , h3 , b3
h5 , b5 , h3 , b3
h5 , b5 , h3 , b3          %80
h5 , b5 , h3 , b3
h5 , b5 , h3 , b3
h5 , b5 , h3 , b3
h5 , b5 , h3 , b3
h5 , b5 , h3 , b3
h5 , b5 , h3 , b3
h5 , b5 , h3 , b3
h5 , b5 , h3 , b3
h5 , b5 , h3 , b3
h1 , b1 , h2 , b2          %90
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2          %100
```

```
    h1 , b1 , h2 , b2
    h1 , b1 , h2 , b2
    h1 , b1 , h2 , b2
    h5 , b5 , h3 , b3
    h5 , b5 , h3 , b3
    h5 , b5 , h3 , b3
    h5 , b5 , h3 , b3
    h5 , b5 , h3 , b3
    h5 , b5 , h3 , b3
    h5 , b5 , h3 , b3        %110
    h5 , b5 , h3 , b3
    h5 , b5 , h3 , b3
    h5 , b5 , h3 , b3
    h5 , b5 , h3 , b3
    h5 , b5 , h3 , b3];      %115

%% split elements into smaller sections

element_split = [];

%% define constraints for each node
% x,y,z, rotx, roty, rotz; 0 = no constraints, 1 = constraint

%Look more into the constraints
xyz_constr = [1, 1, 1, 0, 0, 0
              1, 1, 1, 0, 0, 0
              1, 1, 1, 0, 0, 0
              1, 1, 1, 0, 0, 0
              0, 0, 0, 0, 0, 0
              0, 0, 0, 0, 0, 0
              0, 0, 0, 0, 0, 0
              0, 0, 0, 0, 0, 0
              0, 0, 0, 0, 0, 0
              0, 0, 0, 0, 0, 0        %10
              0, 0, 0, 0, 0, 0
              0, 0, 0, 0, 0, 0
              0, 0, 0, 0, 0, 0
              0, 0, 0, 0, 0, 0
              0, 0, 0, 0, 0, 0
              0, 0, 0, 0, 0, 0
              0, 0, 0, 0, 0, 0
              0, 0, 0, 0, 0, 0
              0, 0, 0, 0, 0, 0
              0, 0, 0, 0, 0, 0        %20
              0, 0, 0, 0, 0, 0
              0, 0, 0, 0, 0, 0
              0, 0, 0, 0, 0, 0
              0, 0, 0, 0, 0, 0
              0, 0, 0, 0, 0, 0
              0, 0, 0, 0, 0, 0
              0, 0, 0, 0, 0, 0
              0, 0, 0, 0, 0, 0
              0, 0, 0, 0, 0, 0
```

```
                0, 0, 0, 0, 0, 0        %30
                0, 0, 0, 0, 0, 0
                0, 0, 0, 0, 0, 0
                0, 0, 0, 0, 0, 0
                0, 0, 0, 0, 0, 0
                0, 0, 0, 0, 0, 0
                0, 0, 0, 0, 0, 0
                0, 0, 0, 0, 0, 0
                0, 0, 0, 0, 0, 0
                0, 0, 0, 0, 0, 0
                0, 0, 0, 0, 0, 0        %40
                0, 0, 0, 0, 0, 0
                0, 0, 0, 0, 0, 0
                0, 0, 0, 0, 0, 0
                0, 0, 0, 0, 0, 0
                0, 0, 0, 0, 0, 0
                0, 0, 0, 0, 0, 0
                0, 0, 0, 0, 0, 0
                0, 0, 0, 0, 0, 0
                0, 0, 0, 0, 0, 0
                0, 0, 0, 0, 0, 0        %50
                0, 0, 0, 0, 0, 0
                0, 0, 0, 0, 0, 0
                0, 0, 0, 0, 0, 0
                0, 0, 0, 0, 0, 0];

%% define external forces/moments
% [node number, direction(1=x,2=y,3=z 4=moment x, 5= moment y,
% 6= moment z), magnitude]

ext_forces = [27,3,537195          %Force is equal to 109.5 tonnes of load
              39,3,537195
              26,1,3744                     %Wind, 1:1 to boom, worst case
              26,2,748.8
              38,1,3744
              38,2,748.8
              22,2,16920
              51,3,537195                              %Also 109.5 tonnes
              44,3,537195];
 %Nodes 53 and 45 gave smaller deformations though
 %Nodes at the end of boom where load is applied: 27 and 39
 %Nodes above first stands 17 and 28
 %Nodes above land: 55 and 56

%% define plot settings
% Plot 1
plot_undeformed        = true;
plot_undeformed_labels = true;
plot_undeformed_forces = true;
plot_deformed          = false;
plot_deformed_labels   = false;
plot_deformed_forces   = false;
```

```
% Plot 2
plot_undeformed_tubes  = true;

% Plot 3
plot_deformed_tubes    = true;
color_deformations     = false;
color_stress           = true;

% Plot deformation scaling: visual displacements will be multiplied
% with this factor
f = 50;

%% Calculation of model parameters and element splitting (DON'T EDIT!)
% Calculate area of each element
A1 = 2.*C(46 4:67 4,1).*C(46 4:67 4,2)+C(46 4:67 4,3).*C(46 4:67 4,4);
%Boom

A2 = 2.*C(88 4:101 4,1).*C(88 4:101 4,2)+C(88 4:101 4,3).*C(88 4:101 4,4);
%Boom

A3 = C(22:29,3).*C(22:29,4);
%Support and support support

A4 = C(68 4:87 4,3).*C(68 4:87 4,4);
%Triangles

A5 = C(102 4:113 4,3).*C(102 4:113 4,4);
%Triangles

A6 = (C(1:9,4).*C(1:9,3)) (C(1:9,1).*C(1:9,2));
%Main frame

A7 = (C(34 4:45 4,4).*C(34 4:45 4,3)) (C(34 4:45 4,1).*C(34 4:45 4,2));
%Small frame

A8 = (C(10:21,4).*C(10:21,3)) (C(10:21,1).*C(10:21,2));
%Small frame

A = [A6;A8;A3;A7;A1;A4;A2;A5];

% Calculate area moments of inertia for I beams and square beams

%I = [(1/64)*pi*( C(:,1).^4    (C(:,1) 2.*C(:,2)).^4)  ,...
%(1/64)*pi*( C(:,1).^4    (C(:,1) 2.*C(:,2)).^4)];

I1 = [2.*((1/12).*(C(46 4:67 4,2).*(C(46 4:67 4,1).^3)+(((C(46 4:67 4,3)...
     ./2)+(C(46 4:67 4,1)./2)).^2).*C(46 4:67 4,1).*C(46 4:67 4,2)) +...
     ((C(46 4:67 4,4).*C(46 4:67 4,3)).^3)./12),...
     2.*(1/12).*(C(46 4:67 4,2).^3).*C(46 4:67 4,1)+(1/12).*...
     (C(46 4:67 4,4).^3).*C(46 4:67 4,3)];

I2 = [(2/12).*(C(88 4:101 4,2).*(C(88 4:101 4,1).^3)+(((C(88 4:101 4,3)...
     ./2)+(C(88 4:101 4,1)./2)).^2).*C(88 4:101 4,1).*C(88 4:101 4,2))...
```

```
    + ((C(88 4:101 4,4).*C(88 4:101 4,3)).^3)./12,...
    (2/12).*(C(88 4:101 4,2).^3).*C(88 4:101 4,1)+(1/12).*...
    (C(88 4:101 4,4).^3).*C(88 4:101 4,3)];

I3 = [1/12.*(C(22:29,3).*C(22:29,4).^3+0.*C(22:29,1).*C(22:29,2)),...
     1/12.*(C(22:29,3).*C(22:29,4).^3+0.*C(22:29,1).*C(22:29,2))];

I4 = [1/12.*(C(68 4:87 4,3).*C(68 4:87 4,4).^3+0.*C(68 4:87 4,1).*...
    C(68 4:87 4,2)),1/12.*(C(68 4:87 4,3).*C(68 4:87 4,4).^3+0.*...
    C(68 4:87 4,1).*C(68 4:87 4,2))];

I5 = [1/12.*(C(102 4:113 4,3).*C(102 4:113 4,4).^3+0.*C(102 4:113 4,1).*...
    C(102 4:113 4,2)),1/12.*(C(102 4:113 4,3).*C(102 4:113 4,4).^3+0.*...
    C(102 4:113 4,1).*C(102 4:113 4,2))];

I6 = [1/12.*(C(1:9,3).*(C(1:9,4).^3)) 1/12.*(C(1:9,1).*(C(1:9,2).^3)),...
     1/12.*(C(1:9,3).*(C(1:9,4).^3)) 1/12.*(C(1:9,1).*(C(1:9,2).^3))];

I7 = [1/12.*(C(30:45 4,3).*(C(30:45 4,4).^3)) 1/12.*(C(30:45 4,1).*...
    (C(30:45 4,2).^3)),1/12.*(C(30:45 4,3).*(C(30:45 4,4).^3)) 1/12.*...
    (C(30:45 4,1).*(C(30:45 4,2).^3))];

I8 = [1/12.*(C(10:21,3).*(C(10:21,4).^3)) 1/12.*(C(10:21,1).*...
    (C(10:21,2).^3)),1/12.*(C(10:21,3).*(C(10:21,4).^3)) 1/12.*...
    (C(10:21,1).*(C(10:21,2).^3))];

I = [I6;I8;I3;I7;I1;I4;I2;I5];

%New formula of moment of inertia for squares
J = sum(I,2);

% Automatic element split
[xyz_nodes,elements,element_split,A,I,J,C,xyz_constr] = ...
     split_elements(xyz_nodes,elements,element_split,A,I,J,C,xyz_constr);
```

Then, the code is shown where the force is exerted at the position of the first leg:

```
%% Definition of material properties
% define Elastic modulus, take for all elements the same E. E is here
% defined for steel. nu is the poisson factor, necessary for calculating
%the shear modulus.
addpath support_functions_Frame_3D
E = 210.5e9;                                                         %S420 steel
nu = 0.3;
G = E/(2*(1+nu));
rho = 7800;

%%% define coordinates of the nodes [meter]
xyz_nodes = [0,0,0
             23,0,0
             0,23,0
             23,23,0
             0,0,15
             23,0,15
             0,23,15
             23,23,15
             0,0,40
             23,0,40           %10
             0,23,40
             23,23,40
             0,8.75,63
             0,14.25,63
             0,8.75,65
             0,14.25,65
             0,8.75,40         %node 17 of new boom
              4.57,8.75,40
              9.14,8.75,40
              13.71,8.75,40    %20
              18.28,8.75,40
              22.85,8.75,40
              27.42,8.75,40
              31.99,8.75,40
              36.56,8.75,40
              41.13,8.75,40
              45.7,8.75,40
             0,14.25,40        %node 28
              2.29,14.25,40
              6.86,14.25,40    %30
              11.43,14.25,40
              16,14.25,40
              20.57,14.25,40
              25.114,14.25,40
              29.71,14.25,40
              34.28,14.25,40
              38.85,14.25,40
              43.42,14.25,40
              45.7,14.25,40    %node 39
             23,8.75,40        %nodes of the back, 40
```

```
            27.57,8.75,40
            32.14,8.75,40
            36.71,8.75,40
            41.28,8.75,40
            45.85,8.75,40
            50.42,8.75,40      %final of back 8.75 in y
            23,14.25,40        %nodes of the back
            25.3,14.25,40
            29.87,14.25,40
            34.42,14.25,40      %50
            39.01,14.25,40
            43.58,14.25,40
            48.15,14.25,40
            50.42,14.25,40
            11.5,8.75,40       %Nodes above land
            11.5,14.25,40];    %Idem

        %Connection 15 25
%% define which nodes are connected by elements and
elements = [1, 5
            2, 6
            3, 7
            4, 8
            5, 9
            6, 10
            7, 11
            8,12
            6, 12
            %5,6
            %6,7         %10
            %6,8
            %8,7
            %5,7
            %9,6
            %9,7
            %11,8
            9,13
            9,10
            11,12       %20
            9,17
            10,40
            11,14
            10,13
            12,14
            13,14
            14,16
            13,15
            16,15
            13,21       %30
            14,33
            15,27
            16,39
            13,43
```

```
14,50
15,45
16,53
17,28
28,11
40,47        %40
47,12
13,17
14,28
17,55
55,40
56,47
28,56
17,47
28,40
17,18
18,19
19,20        %50
20,21
21,22
22,23
23,24
24,25
25,26
26,27
28,29
29,30
30,31        %60
31,32
32,33
33,34
34,35
35,36
36,37
37,38
38,39
27,39
17,29        %70
29,18
18,30
30,19
19,31
31,20
20,32
32,21
21,33
33,22
22,34        %80
34,23
23,35
35,24
24,36
36,25
```

```
            25,37
            37,26
            26,38
            38,27       %end boom with triangles
            40,41       %90
            41,42
            42,43
            43,44
            44,45
            45,46
            47,48
            48,49
            49,50
            50,51
            51,52       %100
            52,53
            53,54
            54,46       %end of back boom
            40,48
            48,41
            41,49
            49,42
            42,50
            50,43
            43,51       %110
            51,44
            44,52
            52,45
            45,53
            53,46];

%% define outer diameter and wall thickness of each element [meter]
% C = [outer diameter, wall thickness]
h1 = 0.25;  %Height base I beam
h2 = 1;     %Height middle part I beam
b1 = 0.75;  %Width base
b2 = 0.25;  %Width middle part I beam
h3 = 0.1;   %Height triangles
b3 = 0.1;   %Width triangles
h4 = 0.2;   %Height support
b4 = 0.2;   %Width support
h5 = 0;     %Filling for the squared beams
b5 = 0;
h6 = 0.2;   %Height support support, changed to 0.4 from 0.75
b6 = 0.2;   %Width support support, changed to 0.4 from 0.75
h7 = 0.9;   %Outer heigth and width
b7 = 0.9;
h8 = 0.55;  %Inner height and width (So wall thickness all around of 20 cm)
b8 = 0.55;
b9 = 0.8;
h9 = 0.8;

C =[h5 , b5 , h7 , b7
```

```
h5 , b5 , h7 , b7
h5 , b5 , h7 , b7
h5 , b5 , h7 , b7
h5 , b5 , h7 , b7
h5 , b5 , h7 , b7
h5 , b5 , h7 , b7
h5 , b5 , h7 , b7
%h8 , b8 , h7 , b7   %Beams without any load were removed and beams
%that were in the way of transporting the container to land
%h5 , b5 , h7 , b7        %10
%h8 , b8 , h7 , b7
%h8 , b8 , h7 , b7
h5 , b5 , h7 , b7
%h8 , b8 , h7 , b7
%h8 , b8 , h7 , b7
%h5 , b5 , h7 , b7
%h8 , b8 , h7 , b7
h8 , b8 , h9 , b9
h8 , b8 , h9 , b9
h8 , b8 , h9 , b9        %20
h8 , b8 , h9 , b9
h8 , b8 , h9 , b9
h8 , b8 , h9 , b9
h8 , b8 , h9 , b9
h8 , b8 , h9 , b9
h8 , b8 , h9 , b9
h8 , b8 , h9 , b9
h8 , b8 , h9 , b9
h8 , b8 , h9 , b9
h5 , b5 , h4 , b4        %30
h5 , b5 , h4 , b4
h5 , b5 , h4 , b4
h5 , b5 , h4 , b4
h5 , b5 , h6 , b6
h5 , b5 , h6 , b6
h5 , b5 , h6 , b6
h5 , b5 , h6 , b6
h8 , b8 , h9 , b9
h8 , b8 , h9 , b9
h8 , b8 , h9 , b9      %40
h8 , b8 , h9 , b9
h8 , b8 , h9 , b9
h8 , b8 , h9 , b9
h8 , b8 , h9 , b9
h8 , b8 , h9 , b9
h8 , b8 , h9 , b9
h8 , b8 , h9 , b9
h8 , b8 , h9 , b9
h8 , b8 , h9 , b9
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2        %50
h1 , b1 , h2 , b2
```

```
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2          %60
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2
h5 , b5 , h3 , b3          %70
h5 , b5 , h3 , b3
h5 , b5 , h3 , b3
h5 , b5 , h3 , b3
h5 , b5 , h3 , b3
h5 , b5 , h3 , b3
h5 , b5 , h3 , b3
h5 , b5 , h3 , b3
h5 , b5 , h3 , b3
h5 , b5 , h3 , b3
h5 , b5 , h3 , b3          %80
h5 , b5 , h3 , b3
h5 , b5 , h3 , b3
h5 , b5 , h3 , b3
h5 , b5 , h3 , b3
h5 , b5 , h3 , b3
h5 , b5 , h3 , b3
h5 , b5 , h3 , b3
h5 , b5 , h3 , b3
h5 , b5 , h3 , b3
h1 , b1 , h2 , b2          %90
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2          %100
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2
h1 , b1 , h2 , b2
h5 , b5 , h3 , b3
```

```
    h5 , b5 , h3 , b3
    h5 , b5 , h3 , b3
    h5 , b5 , h3 , b3
    h5 , b5 , h3 , b3
    h5 , b5 , h3 , b3
    h5 , b5 , h3 , b3          %110
    h5 , b5 , h3 , b3
    h5 , b5 , h3 , b3
    h5 , b5 , h3 , b3
    h5 , b5 , h3 , b3
    h5 , b5 , h3 , b3];        %115

%% split elements into smaller sections

element_split = [];

%% define constraints for each node
% x,y,z, rotx, roty, rotz; 0 = no constraints, 1 = constraint

%Look more into the constraints
xyz_constr = [1, 1, 1, 0, 0, 0
              1, 1, 1, 0, 0, 0
              1, 1, 1, 0, 0, 0
              1, 1, 1, 0, 0, 0
              0, 0, 0, 0, 0, 0
              0, 0, 0, 0, 0, 0
              0, 0, 0, 0, 0, 0
              0, 0, 0, 0, 0, 0
              0, 0, 0, 0, 0, 0
              0, 0, 0, 0, 0, 0          %10
              0, 0, 0, 0, 0, 0
              0, 0, 0, 0, 0, 0
              0, 0, 0, 0, 0, 0
              0, 0, 0, 0, 0, 0
              0, 0, 0, 0, 0, 0
              0, 0, 0, 0, 0, 0
              0, 0, 0, 0, 0, 0
              0, 0, 0, 0, 0, 0
              0, 0, 0, 0, 0, 0
              0, 0, 0, 0, 0, 0          %20
              0, 0, 0, 0, 0, 0
              0, 0, 0, 0, 0, 0
              0, 0, 0, 0, 0, 0
              0, 0, 0, 0, 0, 0
              0, 0, 0, 0, 0, 0
              0, 0, 0, 0, 0, 0
              0, 0, 0, 0, 0, 0
              0, 0, 0, 0, 0, 0
              0, 0, 0, 0, 0, 0
              0, 0, 0, 0, 0, 0          %30
              0, 0, 0, 0, 0, 0
              0, 0, 0, 0, 0, 0
              0, 0, 0, 0, 0, 0
```

```
            0, 0, 0, 0, 0, 0
            0, 0, 0, 0, 0, 0
            0, 0, 0, 0, 0, 0
            0, 0, 0, 0, 0, 0
            0, 0, 0, 0, 0, 0
            0, 0, 0, 0, 0, 0
            0, 0, 0, 0, 0, 0      %40
            0, 0, 0, 0, 0, 0
            0, 0, 0, 0, 0, 0
            0, 0, 0, 0, 0, 0
            0, 0, 0, 0, 0, 0
            0, 0, 0, 0, 0, 0
            0, 0, 0, 0, 0, 0
            0, 0, 0, 0, 0, 0
            0, 0, 0, 0, 0, 0
            0, 0, 0, 0, 0, 0
            0, 0, 0, 0, 0, 0      %50
            0, 0, 0, 0, 0, 0
            0, 0, 0, 0, 0, 0
            0, 0, 0, 0, 0, 0
            0, 0, 0, 0, 0, 0];

%% define external forces/moments
% [node number, direction(1=x,2=y,3=z 4=moment x, 5= moment y,
% 6= moment z), magnitude]

ext_forces = [17,3,537195         %Force is equal to 109.5 tonnes of load
            28,3,537195
            26,1,3744                     %Wind, 1:1 to boom, worst case
            26,2,748.8
            38,1,3744
            38,2,748.8
            22,2,16920
            51,3,537195                               %Also 109.5 tonnes
            44,3,537195];
 %Nodes 53 and 45 gave smaller deformations though
 %Nodes at the end of boom where load is applied: 27 and 39
 %Nodes above first stands 17 and 28
 %Nodes above land: 55 and 56

%% define plot settings
% Plot 1
plot_undeformed         = true;
plot_undeformed_labels  = true;
plot_undeformed_forces  = true;
plot_deformed           = false;
plot_deformed_labels    = false;
plot_deformed_forces    = false;

% Plot 2
plot_undeformed_tubes   = true;

% Plot 3
```

```
plot_deformed_tubes     = true;
color_deformations      = false;
color_stress            = true;

% Plot deformation scaling: visual displacements will be multiplied
% with this factor
f = 50;

%% Calculation of model parameters and element splitting (DON'T EDIT!)
% Calculate area of each element
A1 = 2.*C(46 4:67 4,1).*C(46 4:67 4,2)+C(46 4:67 4,3).*C(46 4:67 4,4);
%Boom

A2 = 2.*C(88 4:101 4,1).*C(88 4:101 4,2)+C(88 4:101 4,3).*C(88 4:101 4,4);
%Boom

A3 = C(22:29,3).*C(22:29,4);
%Support and support support

A4 = C(68 4:87 4,3).*C(68 4:87 4,4);
%Triangles

A5 = C(102 4:113 4,3).*C(102 4:113 4,4);
%Triangles

A6 = (C(1:9,4).*C(1:9,3)) (C(1:9,1).*C(1:9,2));
%Main frame

A7 = (C(34 4:45 4,4).*C(34 4:45 4,3)) (C(34 4:45 4,1).*C(34 4:45 4,2));
%Small frame

A8 = (C(10:21,4).*C(10:21,3)) (C(10:21,1).*C(10:21,2));
%Small frame

A = [A6;A8;A3;A7;A1;A4;A2;A5];

% Calculate area moments of inertia for I beams and square beams

%I = [(1/64)*pi*( C(:,1).^4    (C(:,1) 2.*C(:,2)).^4)  ,...
 %(1/64)*pi*( C(:,1).^4    (C(:,1) 2.*C(:,2)).^4)];

I1 = [2.*((1/12).*(C(46 4:67 4,2).*(C(46 4:67 4,1).^3)+(((C(46 4:67 4,3)...
     ./2)+(C(46 4:67 4,1)./2)).^2).*C(46 4:67 4,1).*C(46 4:67 4,2)) +...
     ((C(46 4:67 4,4).*C(46 4:67 4,3)).^3)./12),...
     2.*(1/12).*(C(46 4:67 4,2).^3).*C(46 4:67 4,1)+(1/12).*...
     (C(46 4:67 4,4).^3).*C(46 4:67 4,3)];

I2 = [(2/12).*(C(88 4:101 4,2).*(C(88 4:101 4,1).^3)+(((C(88 4:101 4,3)...
     ./2)+(C(88 4:101 4,1)./2)).^2).*C(88 4:101 4,1).*C(88 4:101 4,2))...
    + ((C(88 4:101 4,4).*C(88 4:101 4,3)).^3)./12,...
     (2/12).*(C(88 4:101 4,2).^3).*C(88 4:101 4,1)+(1/12).*...
     (C(88 4:101 4,4).^3).*C(88 4:101 4,3)];
```

```
I3 = [1/12.*(C(22:29,3).*C(22:29,4).^3+0.*C(22:29,1).*C(22:29,2)),...
    1/12.*(C(22:29,3).*C(22:29,4).^3+0.*C(22:29,1).*C(22:29,2))];

I4 = [1/12.*(C(68 4:87 4,3).*C(68 4:87 4,4).^3+0.*C(68 4:87 4,1).*...
    C(68 4:87 4,2)),1/12.*(C(68 4:87 4,3).*C(68 4:87 4,4).^3+0.*...
    C(68 4:87 4,1).*C(68 4:87 4,2))];

I5 = [1/12.*(C(102 4:113 4,3).*C(102 4:113 4,4).^3+0.*C(102 4:113 4,1).*...
    C(102 4:113 4,2)),1/12.*(C(102 4:113 4,3).*C(102 4:113 4,4).^3+0.*...
    C(102 4:113 4,1).*C(102 4:113 4,2))];

I6 = [1/12.*(C(1:9,3).*(C(1:9,4).^3)) 1/12.*(C(1:9,1).*(C(1:9,2).^3)),...
    1/12.*(C(1:9,3).*(C(1:9,4).^3)) 1/12.*(C(1:9,1).*(C(1:9,2).^3))];

I7 = [1/12.*(C(30:45 4,3).*(C(30:45 4,4).^3)) 1/12.*(C(30:45 4,1).*...
    (C(30:45 4,2).^3)),1/12.*(C(30:45 4,3).*(C(30:45 4,4).^3)) 1/12.*...
    (C(30:45 4,1).*(C(30:45 4,2).^3))];

I8 = [1/12.*(C(10:21,3).*(C(10:21,4).^3)) 1/12.*(C(10:21,1).*...
    (C(10:21,2).^3)),1/12.*(C(10:21,3).*(C(10:21,4).^3)) 1/12.*...
    (C(10:21,1).*(C(10:21,2).^3))];

I = [I6;I8;I3;I7;I1;I4;I2;I5];

%New formula of moment of inertia for squares
J = sum(I,2);

% Automatic element split
[xyz_nodes,elements,element_split,A,I,J,C,xyz_constr] = ...
    split_elements(xyz_nodes,elements,element_split,A,I,J,C,xyz_constr);
```

## C.2 Figures

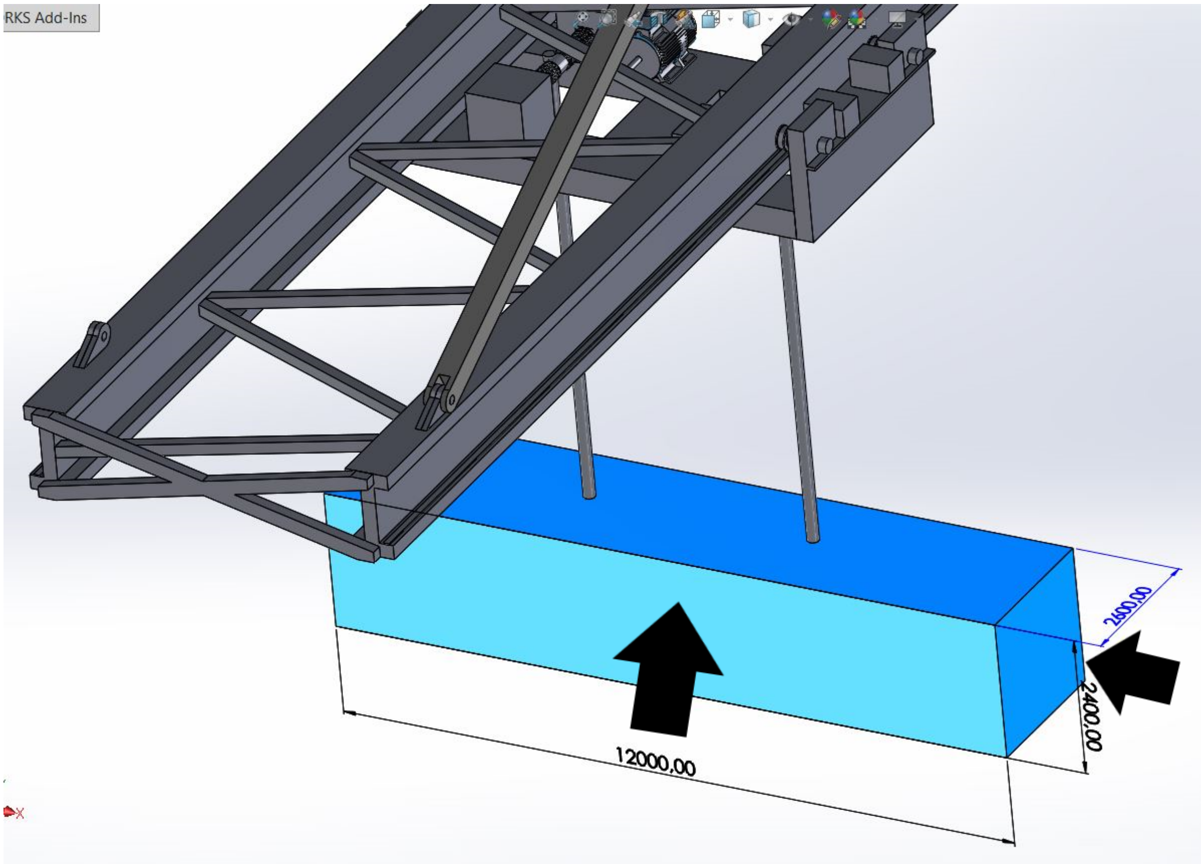



Figure 80: Sketch of the container with the windforces on it in x-, and y-direction. The cables are scaled up ten times to make clear where they are located

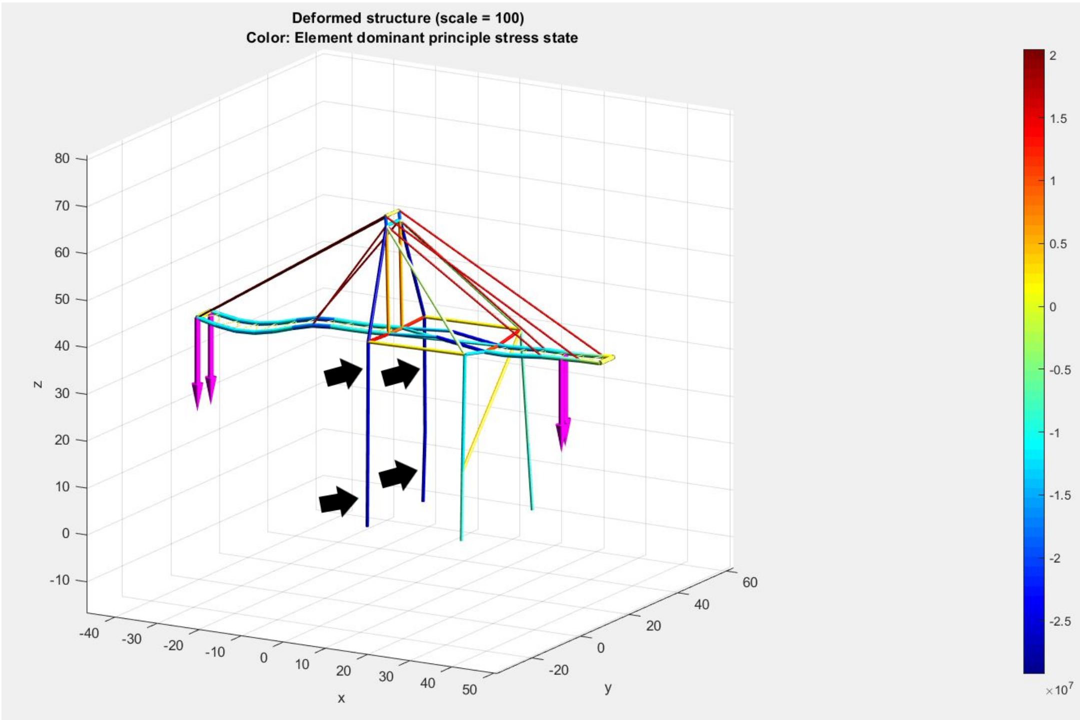



Figure 81: First iteration

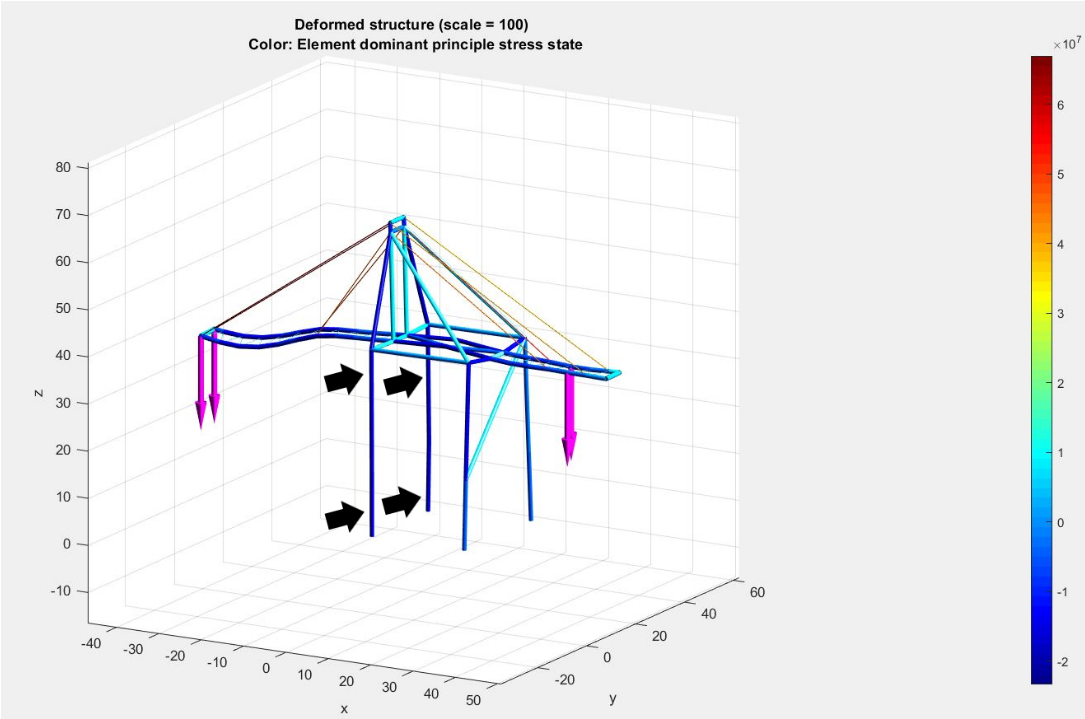



Figure 82: Second iteration

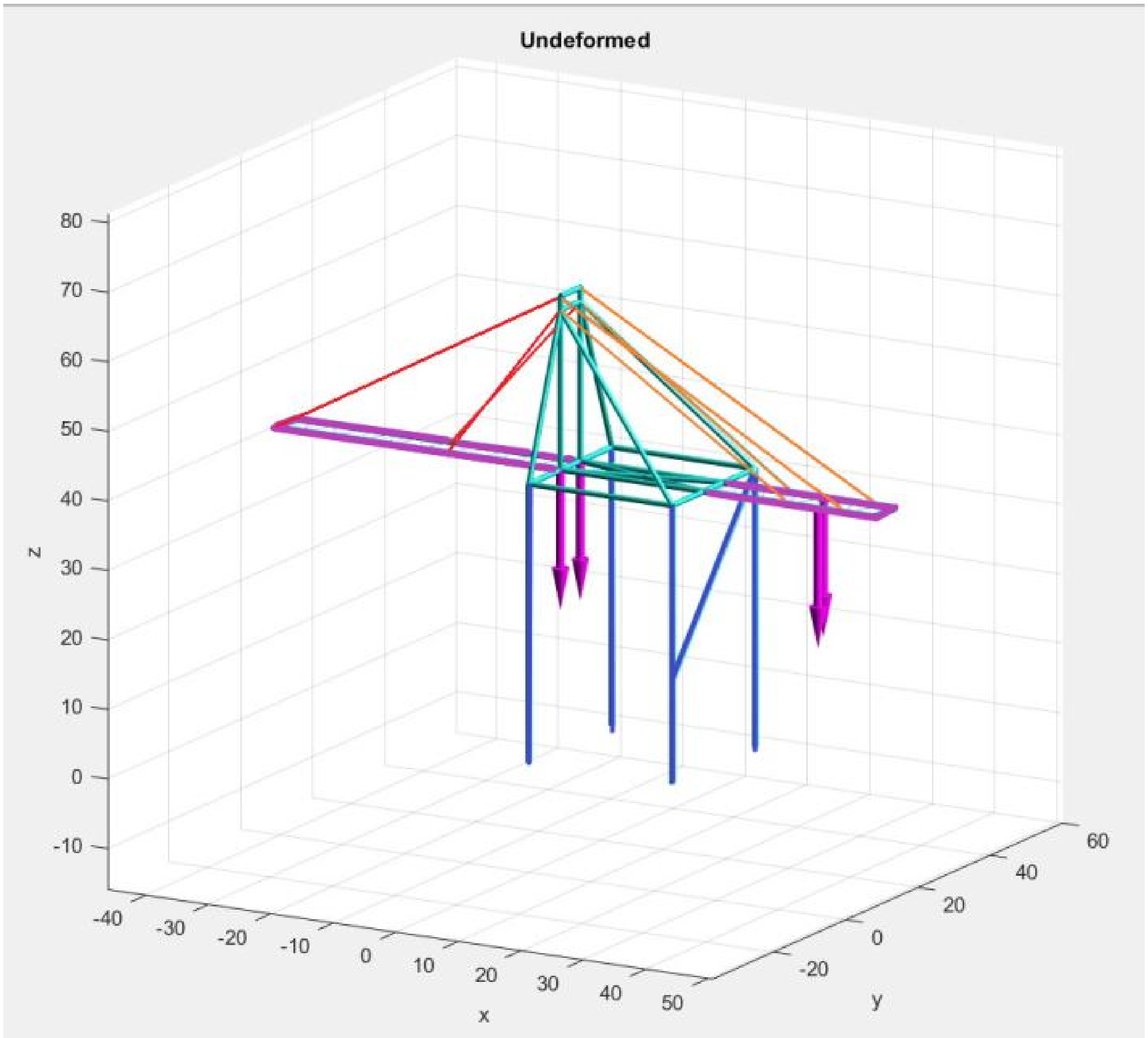



Figure 83: All the different beams used in the model with different colors: The blue beams are the legs, the green beams are the frame, the red beams are the supports, the orange beams are the supports that support the supports, the purple beams are the I-beams of the boom and between the I-beams, the triangles are located. These triangles can not be seen on this picture, but are shown in the next figure in yellow.

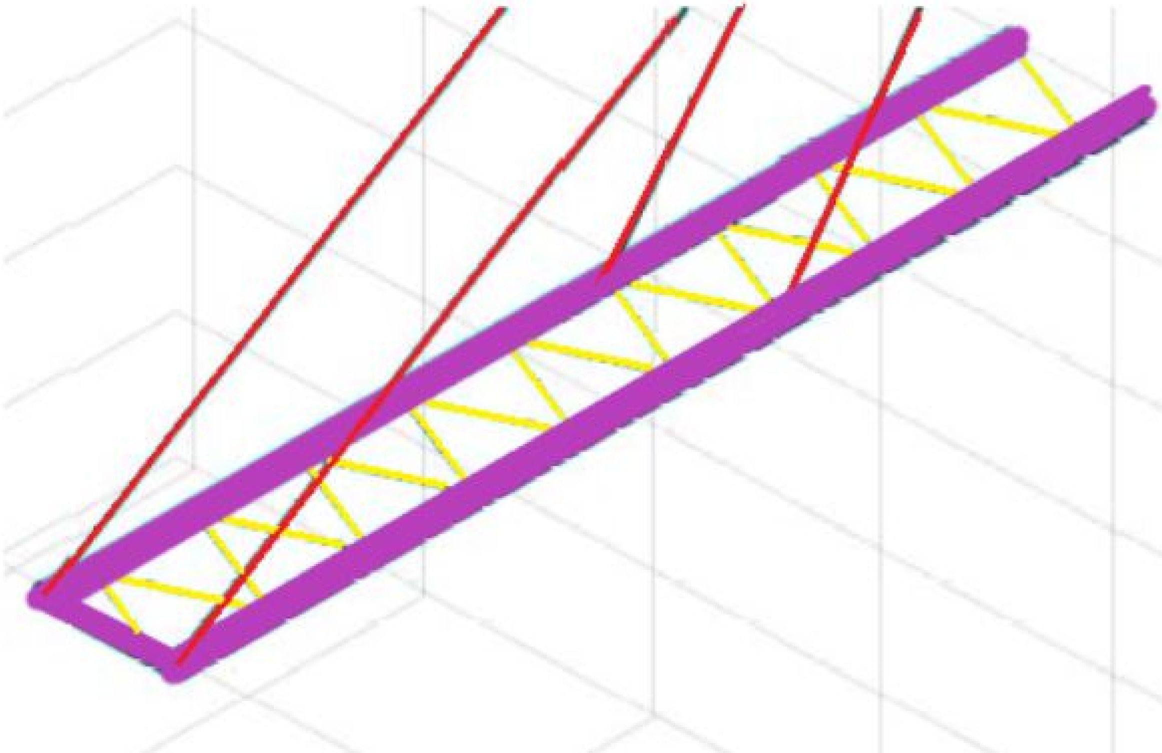


Figure 84: Triangles between the boom indicated in yellow

## C.3 Calculations

**Front leg:**

- E is the Young's modulus of the used material for the boom, which is S420 steel. This material has a Young's modulus between 200 GPa and 221 GPa, so an average of 210.5 GPa is taken.
- I is the moment of inertia for the beam, which is calculated with I = $\frac{1}{12} \cdot 1.1^3 \cdot 1.1 = 0.122\text{m}^4$
- L is the length of the beam, which is equal to 40 meters

All of the above gives a $P_{crit}$ of 39 606 kN. With this $P_{crit}$, the critical stress is then calculated: $\sigma = \frac{F}{A}$, with F equal to $P_{crit}$ and A the cross-sectional area of the beam ($1.1 \cdot 1.1 = 1.21$). Those values give $\sigma$ to be equal to 32.73 MPa.

**Shorter beam:**

- E is the Young's modulus of the used material for the boom, which is S420 steel. This material has a Young's modulus between 200 GPa and 221 GPa, so an average of 210.5 GPa is taken.
- I is the moment of inertia for the beam, which is calculated with I = $\frac{1}{12} \cdot 0.7^3 \cdot 0.7$ - $\frac{1}{12} \cdot 0.3^3 \cdot 0.3 = 0.0193\text{m}^4$
- L is the length of the beam, which is equal to 23 meters

All of the above gives a $P_{crit}$ of 6276 kN. With this $P_{crit}$, the critical stress is then calculated: $\sigma = \frac{F}{A}$, with F equal to $P_{crit}$ and A the cross-sectional area of the beam ($0.7 \cdot 0.7 - 0.3 \cdot 0.3 = 0.4$). Those values give $\sigma$ to be equal to 47.46 MPa.

# D Drive System

## D.1 Gear Design

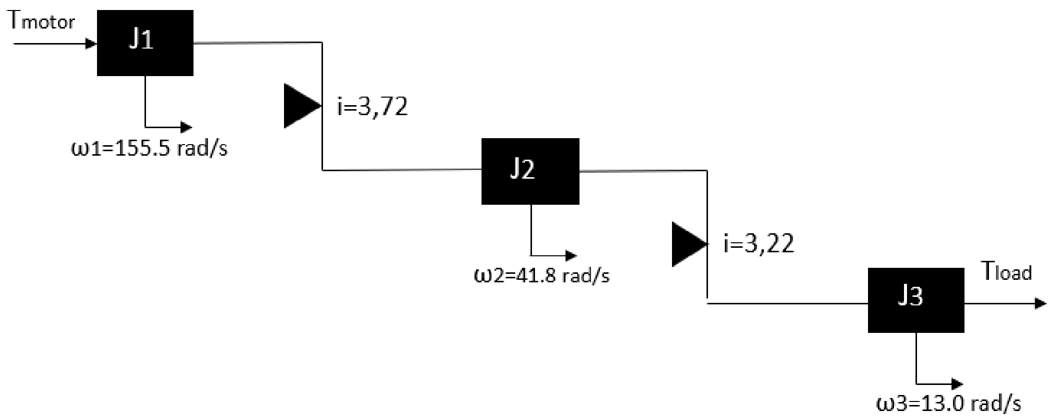



Figure 85: Drive system

# E Highly Loaded Hinge

## E.1 Lifetime

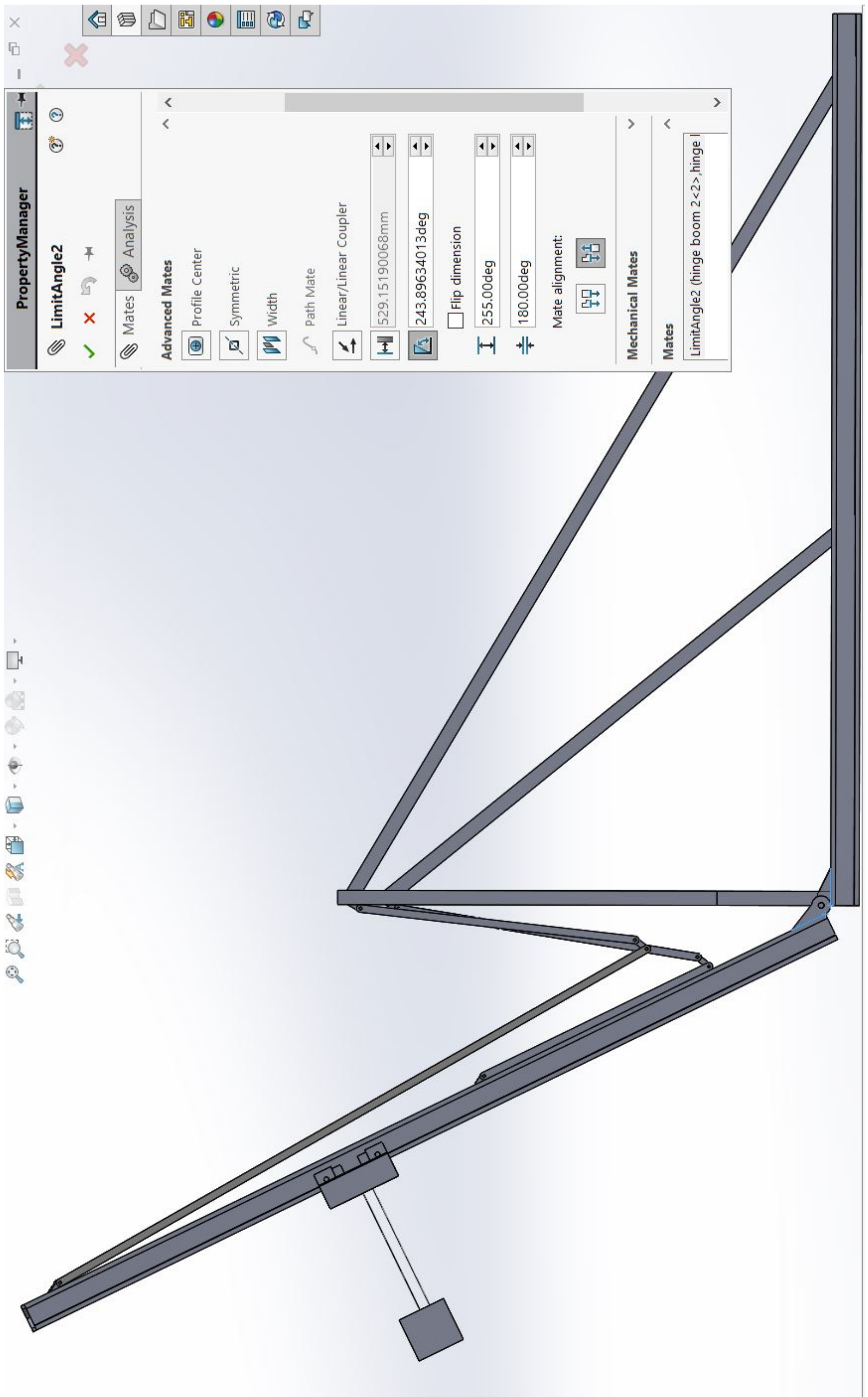



Figure 86: The maximum angle of the crane (63.89°)

## E.2 Technical Drawings