Design of a STS-crane

Project Design Report

Group 8

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1 Introduction

Nowadays in an online world in which more and more purchases are made on the internet, more and more goods are transported all over the world by container ships. This results in qualities like reliability, productivity and efficiency are becoming more and more imported in the modern-day harbours that are processing mega ships, which are also becoming larger over time. These modern demands require well-engineered equipment to work with, with the so called ship-to-shore cranes playing a central role as the loading and unloading of the container ships forms the key activity of a harbour. These cranes are prone to technical improvements due to the fact that ships are becoming bigger and the modern demands as stated above are becoming more and more important. Due to these changes in the technical world, it is desired to design new cranes regularly as new technical improvements are invented regularly over time. These modern STS-cranes with these newly invented techniques are already being designed by several different companies. However, while making concepts, new technical improvements or new combinations of these newly invented technical improvements can be made which may fulfill the modern demands better than the regular used STS-cranes. Furthermore, going through the designing process of a STS-crane, engineering experience is being gained by the design team which contributes to their engineering professionality. The purpose of this study is therefore to design a modern STS-crane that fulfils the modern demands in large ports of efficiency, reliability and productivity within the stated technical requirements[1].
This report contains firstly a problem and requirement analysis. After this, the report contains a part containing concepts followed by a part containing the detailed design forming the final product using the Finite Element Method, with elaborations on machine elements and mechanics of materials.

2 Summary

To realise the purpose of this report, namely designing a modern STS-crane that fulfils modern demands, the following procedure was followed by a team of first-year mechanical engineering Bachelor students from the University of Twente:
At first the problem and given requirements were analysed out of which several different design requirements were made to which the final product should be judged.
Secondly, different concepts were introduced for the boom, trolley and hoist with respect to the design requirements as these three components form the main parts of the STS-crane. These concepts form the three different main components of the crane were separately evaluated according to common sense, the design requirements, function fulfillment and from a mechanics and machine elements perspective in order to be able to give a well-reasoned concept choice. Out of this, a well-reasoned concept choice followed with argumentation.
It was chosen to use the cable lifting mechanism with disc brakes on the drum as safety feature; the MOT system for hoisting and the underhanging trolley. The chosen concepts of the three different main parts of the STS-crane were then combined and worked out further. Materials were assigned to the different components, everything was defined in 3D space and the weight of the crane was minimized while still fulfilling all the requirements using the Finite Element Method.
Elaborations on different machine elements in the hoist mechanism, a highly loaded hinge, a highly loaded bolt connection and a highly loaded weld were also done.
Finally Mohr's circle was applied to a point in the boom hinge to assure the maximum stress was under the yield stress of the shaft material.
All in all, a STS-crane was designed which fulfilled all requirements.

3 Problem definition [1]

In order to be able to design a proper product, the following problem definition is stated:

A ship to shore container crane has to be designed that fulfills the following requirements:

- The design must be a fully operational STS crane, this means all the components in the design must be well defined and connected to the frame (e.g. no flying components).
- The crane must meet the dimensional requirements. See subsection 'Dimensional requirements'.
- The crane must meet the operational requirements. See subsection 'Operational requirements'.
- The crane must withstand the loads that is subjected to during operation (container & spreader weight + wind load). Dynamic loads are neglected in this project. See subsection 'Operational requirements' for the load cases.
- The crane design must be operational and suitable for the environment conditions (temperature and wind speed). See subsection 'Operational requirements'.
- The boom, trolley and hoist mechanisms must be fully designed and configured based on the given load cases.

3.1 Dimensional requirements

Maximum vessel length: 300 m
Maximum vessel width: 35 m
Maximum vessel height: 40 m
Measured from the waterline to the vessel's highest point.

Maximum container length: 12 m
Maximum container width: 2.4 m
Maximum container height: 2.6 m

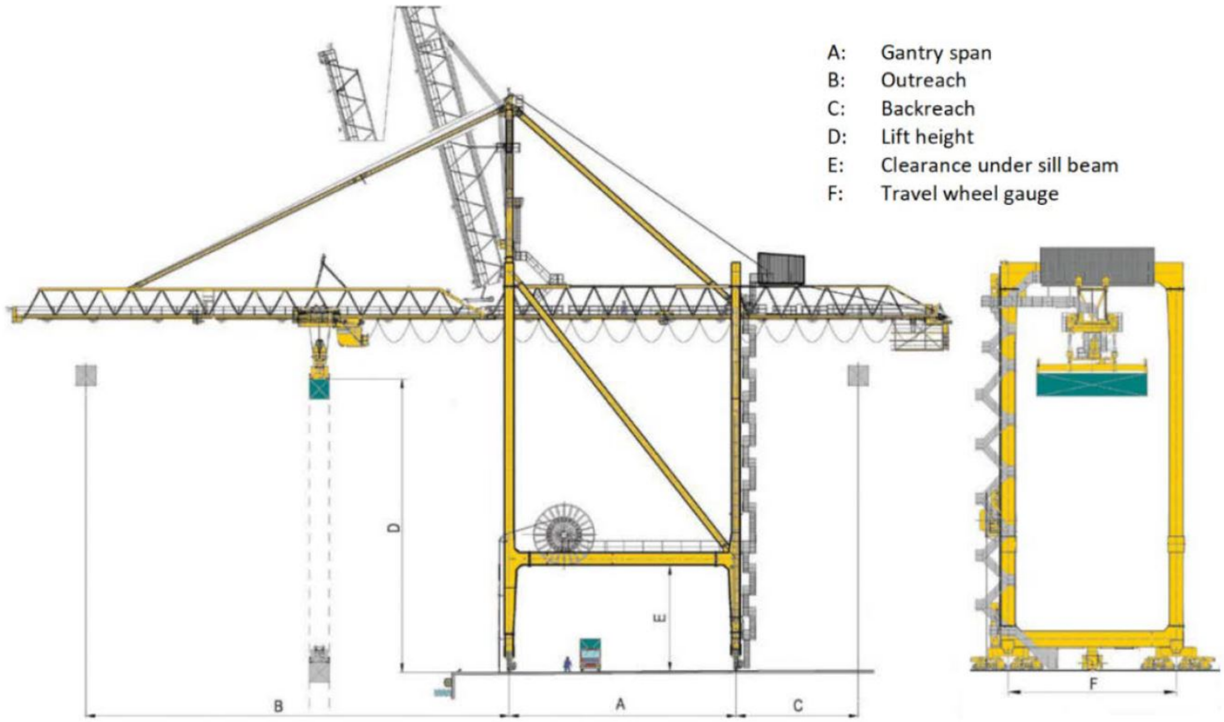

Figure 1: Characteristic dimensions of a STS crane [1]

Maximum outreach(B): 45m
Maximum lift height(D): 40m

3.2 Operational requirements

The crane should be able to operate in the following conditions:

- Maximum weight single container including spreader: 50 tons
- Maximum wind speed: 72 km/h (240 N/m2)
- Ambient temperature: -40 °C to 50°C

Lifting the containers should have the following speed range:

- Hoisting: 90-120 m/min
- Trolley: 180-210 m/min

Maximum deflections at the boom tip:

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

4 Concept Phase

To be able to make a well defined concept choice, multiple concepts were devised for every main component of the functions of the boom, hoist and trolley. These different components were; the hoist mechanism, the lifting mechanism for the boom and the trolley itself. These different components were compared in weighing tables, which gave an indication for which concept scored the best on all different engineering factor combined.
The concept which was considered the best was then chosen to be used in the rest of the project and was worked out further. All detailed and elaborated explanation with weighing tables and argumentation of all different concepts can be found in the appendix section 10.1 till section 10.3.
The following concepts were chosen for the different components:

The hoisting mechanism
For the hoisting mechanism it was chosen to use a Machinery On Trolley (MOT) system. In this system all of the machinery necessary to lift the containers is located on the trolley itself. This reduces the amount of total cable length necessary in the design, which reduced uncertainties regarding controlling the load. A schematic overview of the system can be seen in figure 2.

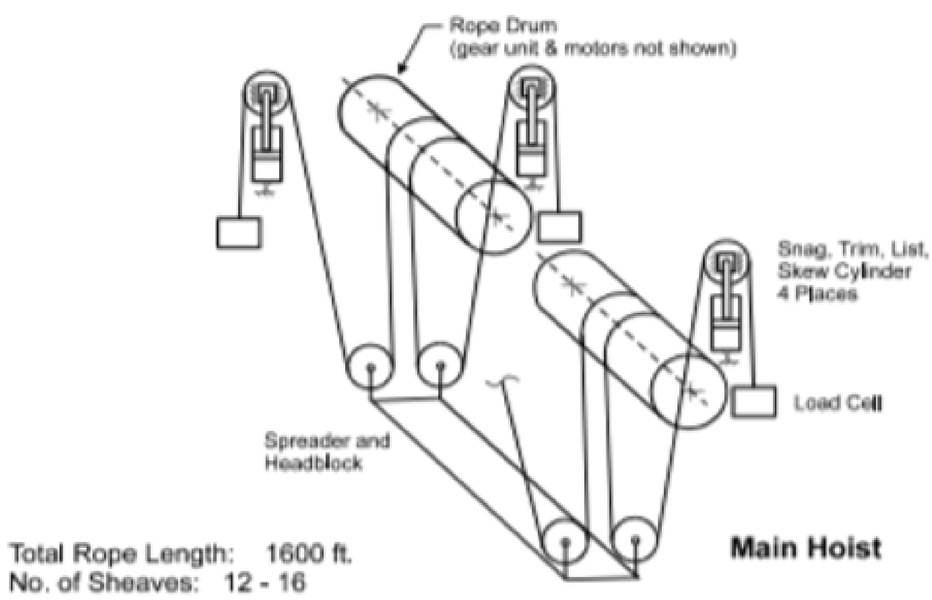

Figure 2: Hoisting concept MOT crane

Lifting mechanism for the boom
For the boom lifting mechanism it was chosen to use a set of cables to lift the boom. This system is the most practical and low-cost solution from a production perspective. This system is also the most save. A schematic drawing of the system can be seen in figure 47. For the locking and braking of the boom, it was chosen to implement a pin-lock system combined with a disc brake system on the drum containing the cables as this combination offers a safe and reliable solution. A schematic overview can be seen in figure 51.

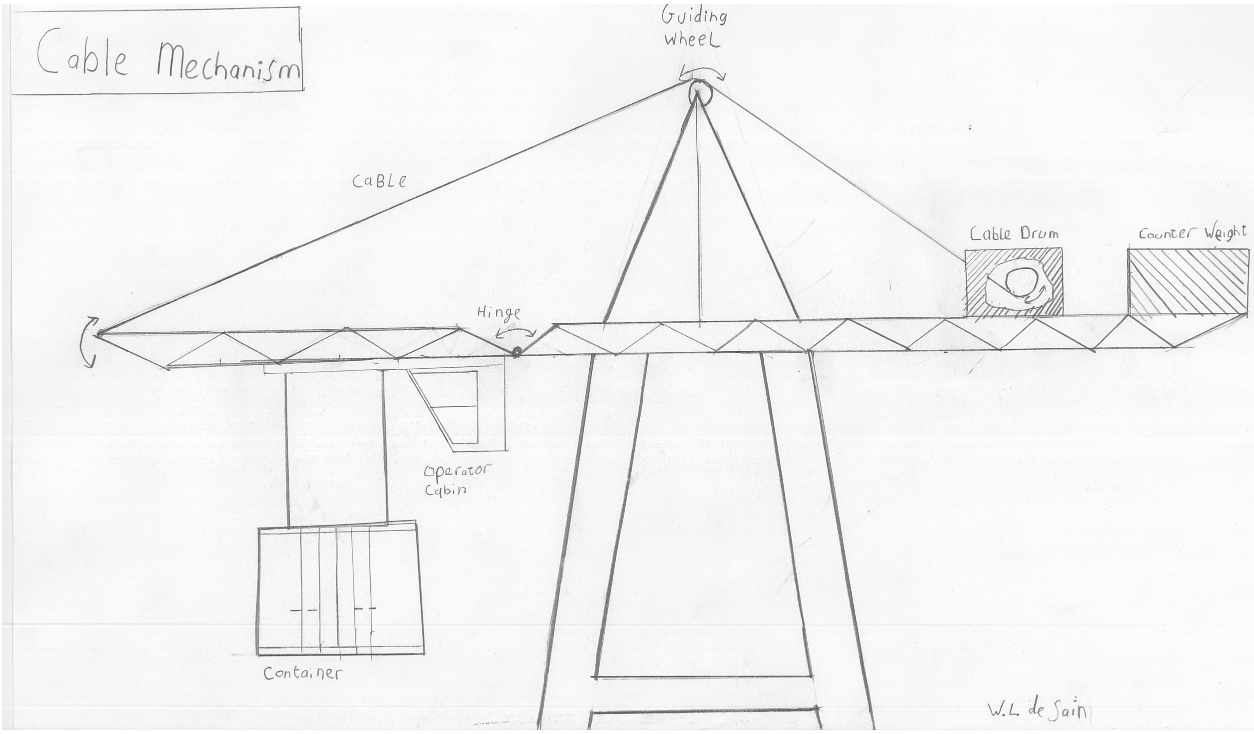

Figure 3: Cable concept

The trolley

For the trolley it was chosen to use a under hanging trolley system. The advantages of an under hanging trolley system is that it is able to lift relatively heavier loads and that it will not interfere with the boom's truss structure as mentioned later in this report. A schematic overview of the system can be seen in figure 4.

Figure 4: Under hanging Trolley

5 Dimensioning

5.1 General assumptions

In order to make further decisions in the design of the different parts of the crane, assumptions have to be made regarding the different rough dimensions of the crane. This was done according to the technical description of a STS-crane by Liebherr [2].
At first, some dimensions of the crane were assumed within the average values of the technical description of Liebherr.

- It was assumed that the space between the two supports of the boom is 50 metres. This in order to have maximum length on which the centre of mass can move without the crane tipping over.
- It was assumed that the back-end of the crane from the support has a length of 25 metres in order to be able to place machinery and to extend the back-reach of the trolley.

Now these lengths are fixed, the place and the weight of the counterweight can be calculated. It is decided that the counterweight will be placed in-between the distance of the two supports of the crane frame. This because the centre of mass of the total crane has to lay always between these two supports at all locations of the trolley with its weight to prevent the crane from tipping over. Because the trolley can be located at the top of the crane as well as at the far back-end of the crane, the counterweight should be placed between the two supports and must have a sufficient weight in order to keep the centre of mass between the two supports. The location and weight of the counterweight is calculated with static equations of the free-body diagram of the total crane where the weight of every slender member is assumed to be zero in order to simplify the calculations. These in-depth calculations can be found in section 10.4.1

5.2 Boom dimensioning

5.2.1 Boom truss system

Truss design

In order to make sure the boom can handle all the forces, a truss design is chosen to create a strong and tough structure. As a basis for the design, the trolley width is chosen. This will be about 6 metres, so the width of the bottom side of the truss system will be about 6 metres.

A simple Warren truss with a triangular cross sectional is chosen with a 45°angle between the bottom triangular truss member and the horizon, as this forms a simple yet strong structure. As for now a cross sectional area with all sides having a length of 6 metres (the trolley width) is chosen as this structure will give the most equal distribution of forces. If another structure turns out to be stronger when the FEM package is used in Matlab, the truss design will be reassessed. The distance M between two truss elements turns out to be 12 metres using geometry.

It is chosen to use slender members with a square cross-section in order to be able to mount the girders of the trolley under the boom.

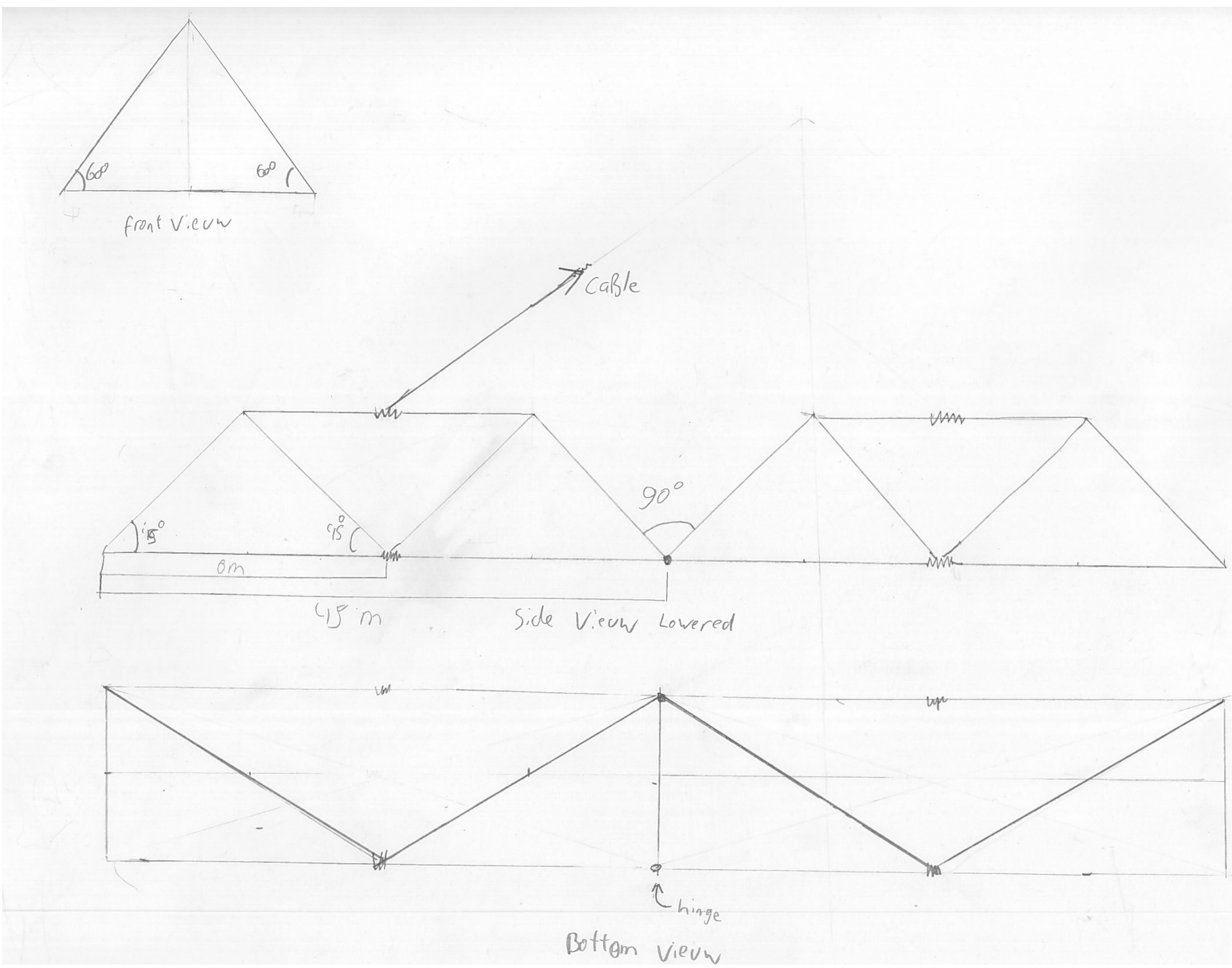

Figure 5: Truss design

Hinge point

It is chosen to place the hinge point on the nearest place where the boom meets the support frame of the crane, as can be seen in the figure below. This, in order to place the weak hinge point as close as possible to the most sturdy point of the crane.

Using the geometry of the truss system, the distance of the hinge point to the frame will be M/2 = 6 metres. Making sure that the boom has a minimum length of 45 metres in combination that an integer is needed for the amount of M's results in a total boom length of 54 metres from the tip of the boom to the crane frame with 4 * M.
The lifting angle is 80°as mentioned later, so the functional angle on which the hinge must be able to operate is also 80°. By leaving all truss-angles intact, an operateable angle of 90°is created. This is enough to let the hinge function properly.
The structure of the hinge point can be seen in the figure below. A shaft is fixed in a hole in the beam in the middle and is put through the socket of the other beam that contains bearings in both holes. The shaft can be fixed with a split pin. The exact machine elements parts will be defined later.

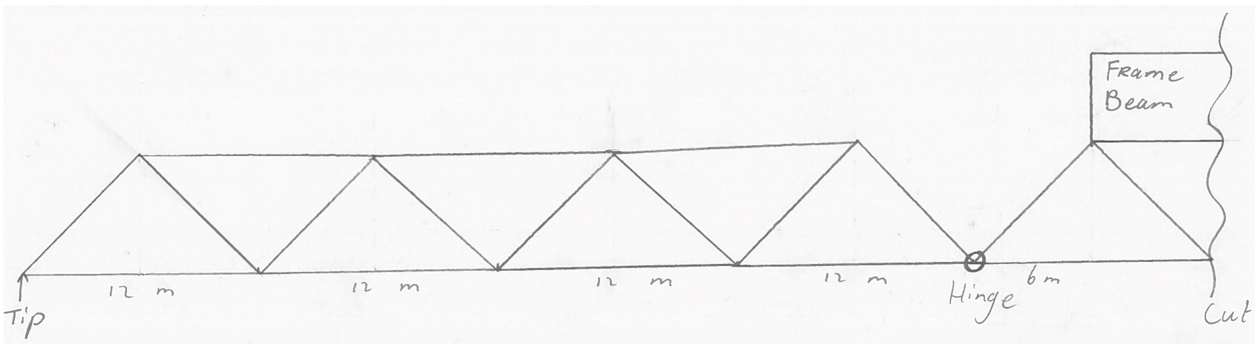

Figure 6: Boom truss design

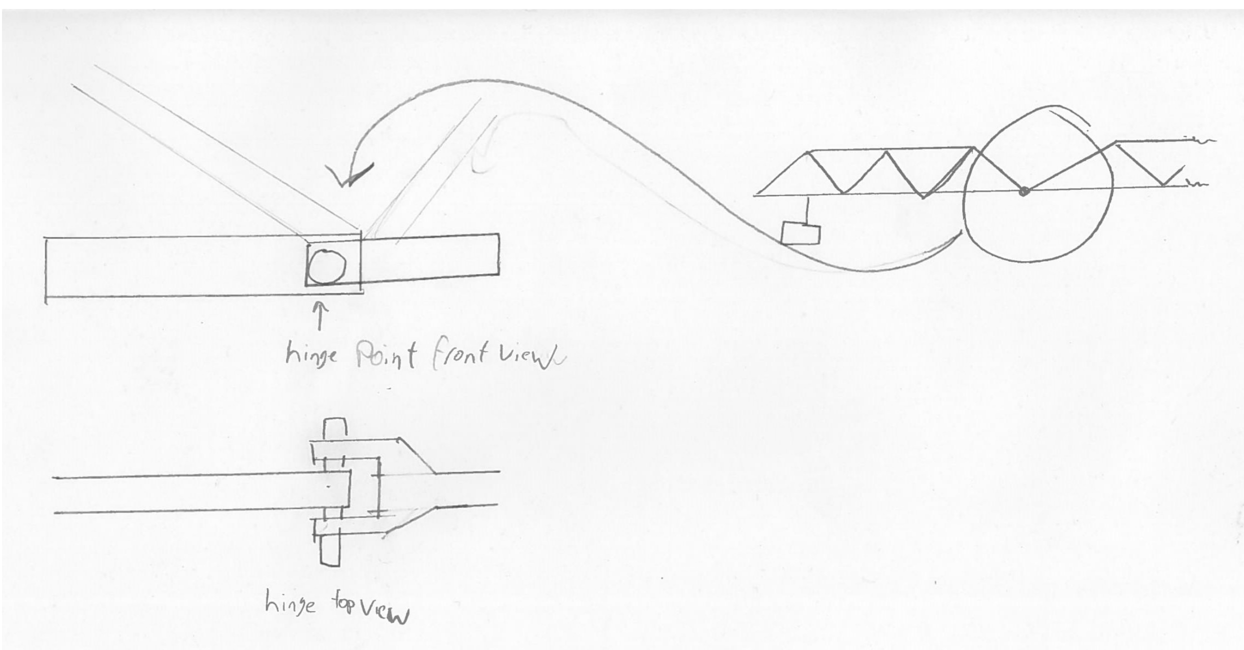

Figure 7: hinge design

Dimensions
With geometry and knowing that every side of the triangular cross-section has a length of 6 metres, it can be calculated with geometry that M = 12 metres and the length from the tip of the boom to the hinge point is 4 * M = 48 metres. The length from the tip of the boom to the crane frame will be Length + M = 54 metres.

Boom material
In order to choose a suitable construction material for the boom, CES Edupack is consulted. By first searching for 'Crane', the most commonly used material type for constructing cranes is chosen. This resulted in the choice for the material class low-alloy steel, as this type of metal is cheap, widely available and easy to machine. For choosing the specific material it is chosen that the density of the material must be as low as possible and the yield strength must be as high as possible in order to get the strongest boom in combination with the lowest lifting force needed as this depends on the weight of the boom. To choose the most suitable material, a simple graph with all low-allow steels is made in CES with the logarithm of the yield strength on the y-axis and the logarithm of the density on the x-axis.
Using the formulas $Mass = \rho * V$ and $\sigma_{yield} = \frac{F_{yield}}{A}$, the performance index $M = \frac{\sigma}{\rho}$ was determined, which has to be maximized for choosing the most suitable material.
Implementing this performance index in CES lets CES rank the best suitable materials. In the low-alloy steel class, AISI 9255 oil quenched & tempered at 205 °C comes out as the most suitable material.
This material is widely available, easy to machine, easy to deform, has a good weldability and is recyclable. The material can be protected by using a coating as for example paint. All of this makes this material a logical choice.

5.2.2 Lifting mechanism

Angle in lifted position
To allow ships as tall as possible to pass underneath the crane the boom needs to be able to lift as high as possible. In an ideal world a lifting angle of 90°with respect to the horizontal position would be favorable. However if the boom would pass the 90°mark the boom would start to fall towards the structure of the crane without a way to stop it from colliding with it. To make sure this will not happen the lifting angle needs to be lower. A safety factor of 10°was chosen so the maximum lift angle would become 80°

Height of extension and cable placement on boom
To determine the height of the extension on top of the crane and the length on which the cables are going to be placed on the boom, the follow points have to be realized:

- In order to keep enough tension on the cables and to be sure that the cables do not come loose from the guiding wheels in the extension, it is chosen that the angle of the cable between the boom and the guiding wheels is parallel with the horizon.
- The more perpendicular the cable is to the boom, the more effective pulling force occurs as the effective pulling force is perpendicular to the boom.
- The height of the extension and the distance between the placement of the cable to the boom and the foot of the extension must be the same.
- The higher the extension is or the closer the cable is placed to the foot of the extension, the more perpendicular the cable will be to the boom, so more effective pulling force will occur.
- With placing the cable more to the tip of the boom, less effective pulling force is needed to lift the boom as the arm to the hinge point becomes longer. However, by doing this, the angle between the boom and the cable will be lower reducing the effective pulling force.
- The maximum vertical deflection which is allowable in the boom tip is 150 mm.

It is chosen that the maximum deflection through the whole boom should be maximum 150mm in order to be able to determine the weight of the extension on the crane and the length on which the cables should be attached to the boom.
This is done by comparing two deflection situations where the load of the trolley is placed at the tip of the boom and in the middle of the length between the cable attachment and the hinge point. The result of this is that the cables of the lifting mechanism should be placed at 29.3 metres from the hinge point of the boom

and that the height of the guiding wheels in the extension on the top of the crane should be placed at a height of 28.9 metres from the boom.
Knowing this, it can be calculated that the length of the cables from the top of the extension to the boom in not-lifted position is about 40 metres and in lifted position about 4 metres.
The in-depth calculations can be found in the appendix.
It is chosen to use two cables, attached at both bottom sides of the booms truss system in order to keep the boom more stable. Furthermore, one cable must be able to hold the crane when the other cable might fail.

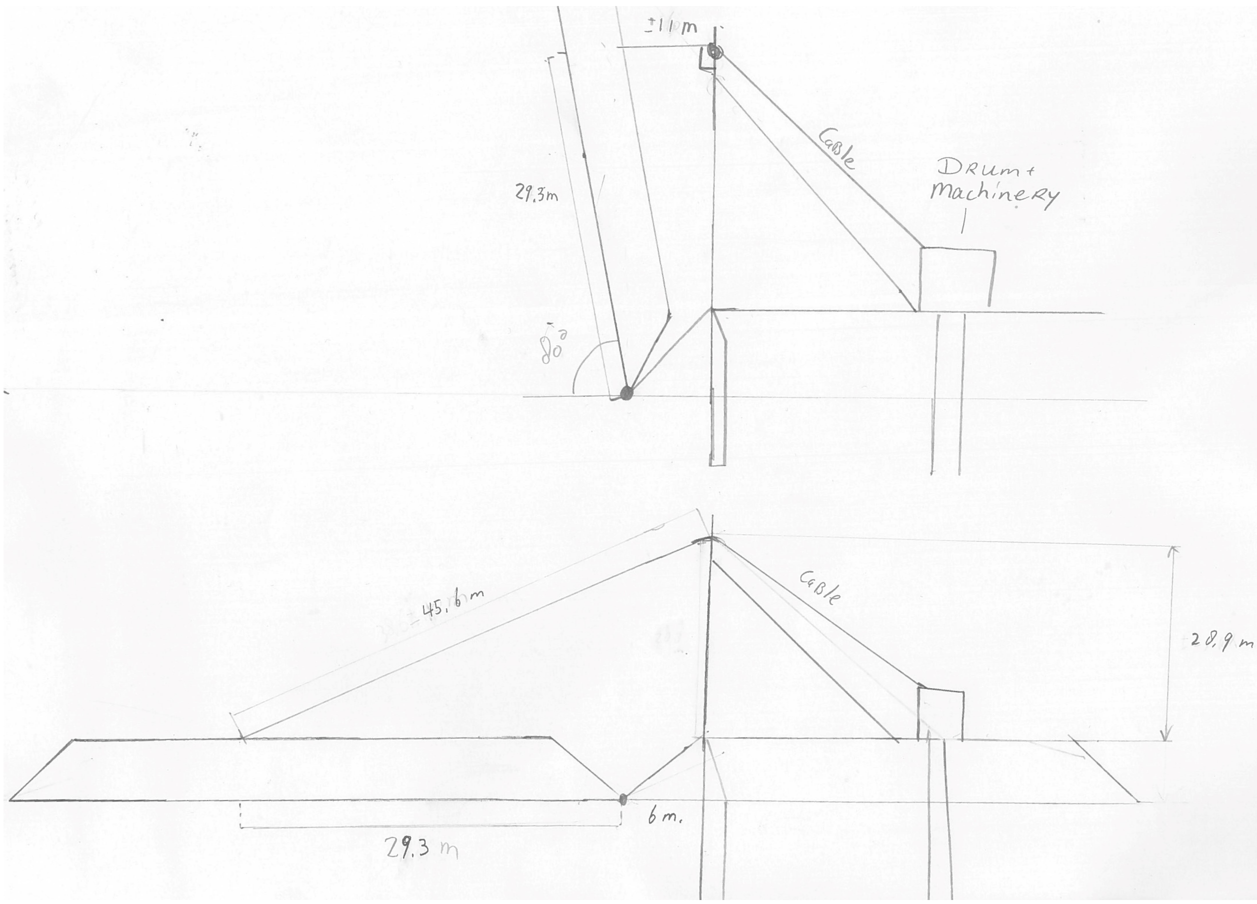

Figure 8: Cable placement on the boom in lowered and lifted state

5.2.3 Cable drum with brake system

Placement
It is chosen to make a cable drum with two larger discs with holes (for the pin lock system and to create a place for the working of the disc brakes) to the sides as shown in the sketch below.
It is chosen to place the whole system at the point where the boom is placed at the support frame of the crane in order to prevent extra deflection and the creation of an extra moment. The shaft of the drum will be placed at the inside of the boom and will be fixed with a fixed bearing and a loose bearing which are placed in two beams that are placed in horizontally beams between two truss members at both diagonal sides of the boom as can be seen in the figure below.
An internal frame in the boom can be made to fix all the components for the disc brakes and to make a sturdy fixed point where the pins of the pin-lock system can be pushed in. In this way, the safety braking systems and the cable drum are combined in one shaft.
The cables can be attached to the drum with a clamping plate to create a relatively flat fixation.
The propulsion and motor can be placed just after the drum in the boom as enough space in the boom is

available.

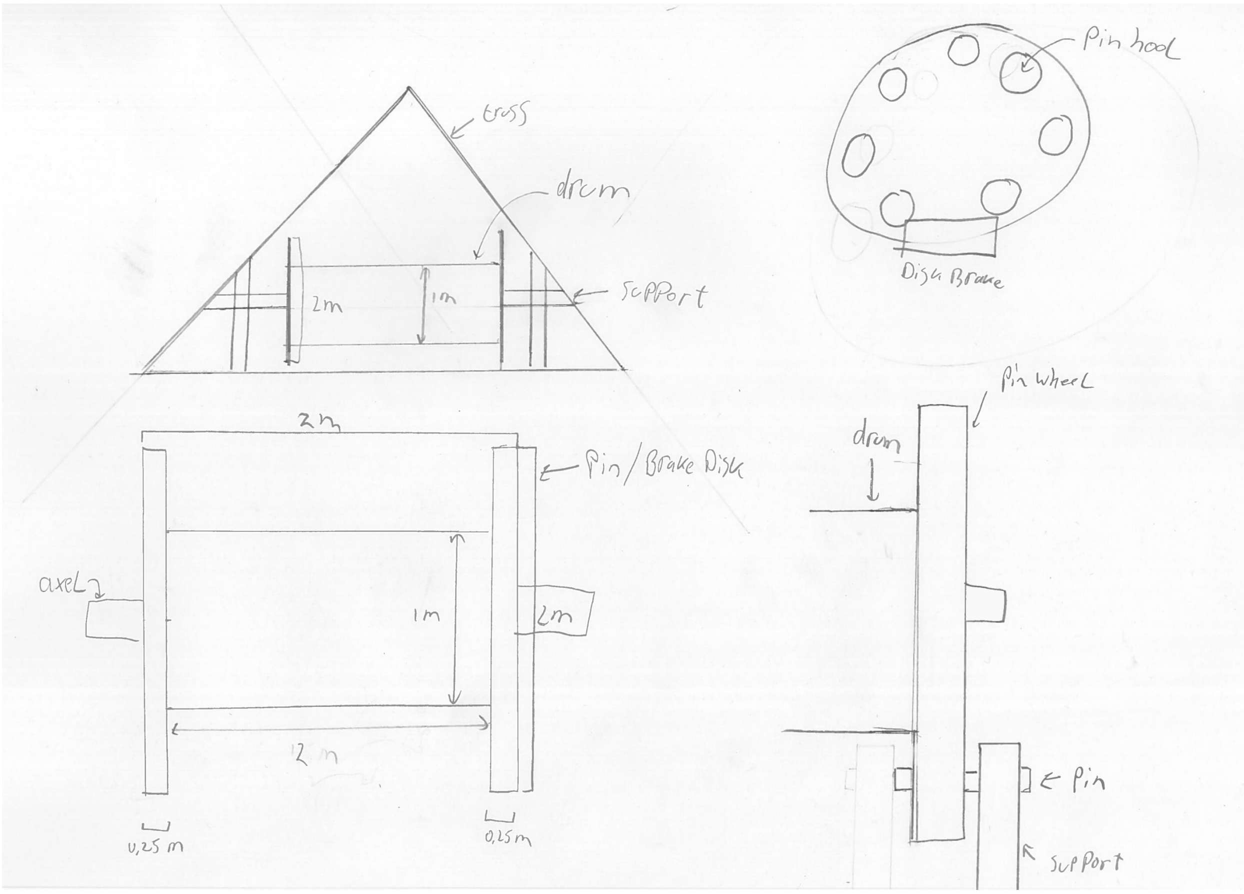

Figure 9: Drum and breaking design sketch

Dimensions
It is chosen to give the cable drum a diameter of 1 metre. Which means that the drum has to rotate about 11 times to lift the boom. In order to place the disc brake system and the locking pins, a disc with a diameter of 2 metres with holes at a radius of 0.75 metres is placed at both sides of the drum. Around the drum, a frame is built to fix the disc brakes too and to create a fixed point to put the locking pins through.
It is assumed that the total length of the drum including the side discs will be about 2.5 metres of which 1 metre is used by each rope to wind up.By placing the drum at a height of 0.5 metres above the bottom of the truss, there will be enough space in between the sides of the truss to place both the drum and its propulsion. Shaft ends can be welded to the sides of the side discs and machined afterwards in order to create a useful shaft to put through the bearings.

Electric cables
All electric cables to deliver power or for communication for the different components can be guided and fixed along the boom to the crane frame and to the ground. A simple schematic is shown below.

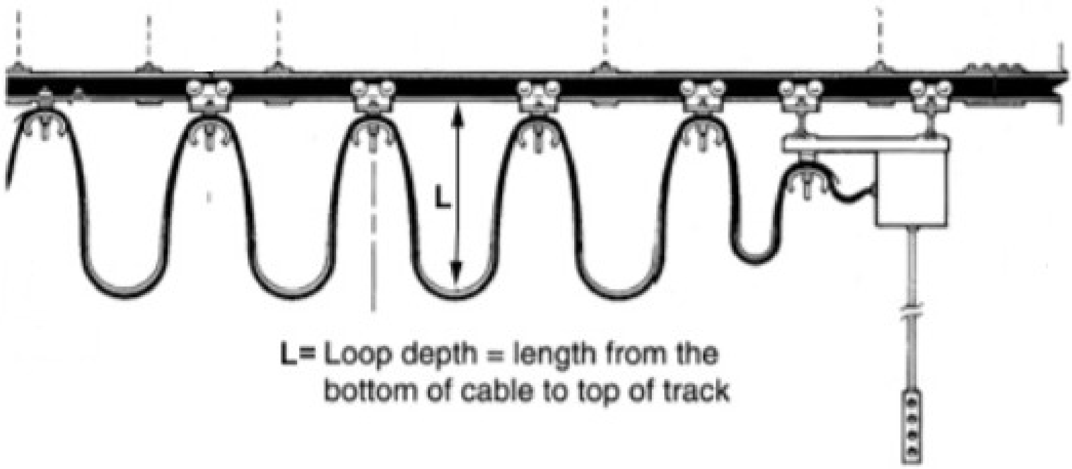

Figure 10: Schematic overview of a festoon system

Cable material and size

In order to be able to choose the right cable material and diameter, the website of Mennens (www.mennens.nl a big Dutch steel cable supplier) was consulted. There a common type of corrosion resistance steel cable was chosen, which is commonly used in gantry cranes and can be delivered in large diameter. From this, the "Steel wire rope - 6x36ws+iwrc" which is galvanized and has a maximum stress of 1960 MPa was chosen.
As the weight of the boom is not yet known, an assumption of the cable diameter has to be made. As circa 1 metre of drum length is available for 1 cable and circa 10 rotations have to be made in order to lift the boom, the maximum cable diameter can be about 100 mm. Knowing this, the 76 mm diameter cable was chosen as this is the largest diameter available, having a minimum breaking force of about 4 Mega Newton. By using two cables, the pulling force of the lifting system will be 8 Mega Newton.
If this is not enough after the FEM package is consulted, the cable design will be reassessed.
According to information received from the technical staff of Mennens, the minimum bending radius has to be between 10 and 15 times the diameter of the cable. By using a drum with a 1 metre diameter and a cable with 76 mm diameter, this requirement is fulfilled.

Drum material

Cable drums are usually made out of grey cast iron class FG 200 [7], this material is also known as EN-GJL-200. After citing CES Edupack it became clear that this material is relative easy to machine and has a relative high yield stress. However, it has poor weld ability. Therefore, a heat treatment is required before an after welding. However, due to the relative small size of the drum this does not pose a big problem.

5.3 Hoist dimensioning

In this section the goal is to fully define the hoist mechanism in 3D space. The first thing is to come up with a list of machine elements and then define each element in 3D space, after which all elements can be put together in the final hoisting design. To define the elements in 3D space, certain design parameters will have to be set up in order to determine suitable dimensions for each element. Next to that, a material will be assigned to each element of the hoisting mechanism.

Machine elements of the hoist mechanism:

- Drums
- Sheaves
- Snag-load system (explained in next section)
- Wire Rope
- Motor
- Gear reduction system
- Drum support bearing/sheave bearing

5.3.1 Snag-load system

Containers sometimes jam against the cell guides (certain guides on the ship that prevent the containers from moving) due to irregularities in the guides. This causes a sudden stress rise in the wire ropes. In order to suppress this impact, a snag-load system is introduced. A snag-load system basically functions as a damper for when the ropes are subjected to high forces. There are 4 dampers, one for each fixed end of the wire rope. Dampers will especially be very useful for the MOT hoisting mechanism since it has a low total rope length. The wire rope can be attached to the snag-load system via a sprocket. It was chosen to not go further in depth about the snag-load system, since it is out of the scope for this project.

5.3.2 Design parameters

- There must be a balance between sheave/drum diameter and wire rope lifetime.
- The hoist should be able to lift at maximum 60 tons with a speed which should be between 90-120 m/min.
- The wire rope must be able to withstand the load of the spreader, container and wind.
- The rope should be able to resist an impact.
- The sheaves and drums should resist the stresses caused by the tension in the ropes.
- The grooves in the drums and sheaves must be designed in such a way that they maximize wire lifetime and minimize cost.
- The bearings should ensure a smooth rotational motion of the drum (and sheaves).

5.3.3 Wire Ropes

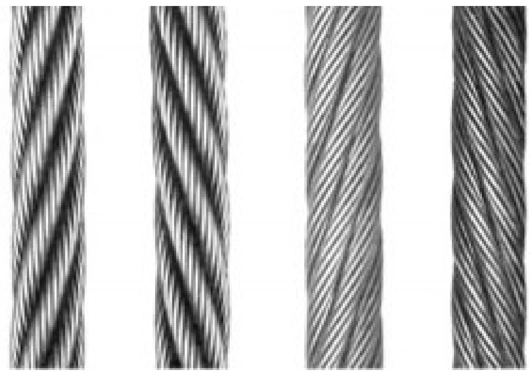

Figure 11: Types of ropes: (a) RHO (b) LHO (c) RHL (d) LHL[3]

As can be seen from the figure above, there are basically 4 types of wire ropes: right-hand ordinary lay (RHO), left-hand ordinary lay (LHO), right-hand Lang's lay (RHL) and left-hand Lang's lay (LHL). To design a suitable wire rope for the hoist mechanism, a few steps have to be followed:

- Determining the safety factor.
- Calculating the design load.
- Select the type and diameter of wire rope.
- Take into account the minimum bend radius.

Firstly, a safety factor will be determined. Hoist wire ropes normally have a safety factor of about 6 [4]. However, due to the use of the snag-load system, the safety factor can be reduced to about 3 [4].

In order to determine the design load, the following formulas are used:

$$Load\ to\ be\ lifted\ per\ rope = Total\ load(Load\ spreader + Container) \div 8 = (60000 \times 9.81) \div 8 = 73575N$$

$$Assumed\ factor\ of\ safety = 3$$

$$Design\ load = 2.5 \times Load\ per\ rope \times assumed\ factorof\ safety = 2.5 \times 73575 \times 3 = 551812.5N$$

It has to be mentioned that the influence of the wind was taken into account at first. However, the effect of the wind on the tension in the ropes is negligible, it was therefore not included in the calculations of the wire rope diameter. The in-depth calculations about the effect of the wind can be found in the appendix.

In order to be able to choose the right cable material and diameter, the website of Mennens (www.mennens.nl a big Dutch steel cable supplier) was consulted. There a common type of corrosion resistance steel cable was chosen which is commonly used in gantry cranes and can be delivered in either RHO or LHO form. From this, the "Steel wire rope - 6x36ws+iwrc" which is galvanized and has a maximum stress of 1960 MPa was chosen. From the table provided by Mennens, a diameter of 30 mm was chosen corresponding to a minimal breakingforce of 628 kN. A diameter of 28 mm was not chosen since this has a minimal breakingforce of 547 kN.

According to information received from the technical staff of Mennens, the minimum bending radius has to be between 10 and 15 times the diameter of the cable. By using a drum with a diameter of 0.9 meter and a cable with a diameter of 30 mm, this requirement is fulfilled.

5.3.4 Sheaves

Basically all installations using wire ropes that run over sheaves require careful sheave design in order to enhance the service life of the wire rope, which is why sheave design is so important. Sheave design both contains dimensioning the sheave as well as what material is used. For example, Metal sheave surfaces require to be harder than the wire ropes. Material selection for the sheaves will be discussed further on.
The sheave design parameters are strongly related to the size of the wire rope that will eventually run over it. The diameter of the wire rope influences for example the groove depth, sheave diameter and the groove radius.
Firstly, standards are introduced regarding the relation between the diameter of the wire rope and the pitch diameter of the sheaves. This ratio according to table 14 is set to 30. Where the pitch diameter of the sheave is 30 times as large as the wire rope diameter. Why this ratio was chosen will be explained within the section about drums.
One of the most important design aspects of the sheaves are the grooves. The grooves in the sheaves should be just a little larger than the diameter of the wire rope in order to prevent pitching and binding of the wire rope strands. Making the grooves in the sheaves to large will result in the ropes to flatten under tension. Making the grooves to small causes rope distortion.
Groove radius should be chosen in accordance with the rope radius in a very specific way. Groove radius = 1.05 x nominal rope radius. Besides that, the groove depth is something to take into account as well. Most often the groove depth is taken as twice the nominal diameter of the wire rope. The bottom of the sheave groove should be a circular arc over an angle of 120 degrees. The groove angle should be between 40 and 45 degrees.
Out of the wire rope selection it follows that the diameter of the wire rope is set to 30mm. From there it is now possible to compute what the pitch diameter of the sheave and the groove radius as well as the groove depth should be in order for the sheave to enhance the service time of the wire ropes.

$$Groove\ radius = 1.05 \times 15 = 15.75mm$$

$$Sheave\ pitch\ diameter = 30 \times 30 = 900mm$$

$$Groove\ depth = 2 \times 30 = 60mm$$

A schematical overview of a sheave containing all the described design parameters of a sheave is given down below in figure 12.
As of now, only a suitable material choice remains in order to have the sheaves designed. The material where sheaves are made out of are often as important as the other design criteria for sheaves. The applicability of three different types of materials will be discussed below. Aluminum is light weight but too soft, which is not preferable considering how often the sheaves should be replaced because of this. Steel is quite similar to aluminum yet heavier and harder. Besides, the availability and price of steel makes it a very suitable material class. Plastics. Although plastics are light weight and resistant to any kind of corrosion, plastics are not hard and strong enough to withstand the load. The materials that are often used in sheaves for this application are either mild steels or cast irons. It was found that ropes running over steel sheaves have a faster wearing rate than ropes running over cast irons. A material with a a sufficient yield strength, hardness and corrosion resistance is wanted. Therefore, sheaves, in our case, are made out of grey cast iron FG 200 [7]. This material is only slightly susceptible to corrosion and has suitable yield strength and hardness,thus CES EduPack 2018. Aside from a clear material choice there are also multiple examples of pulley designs which incorporate multiple materials in order to wed strength with proper resistance characteristics.

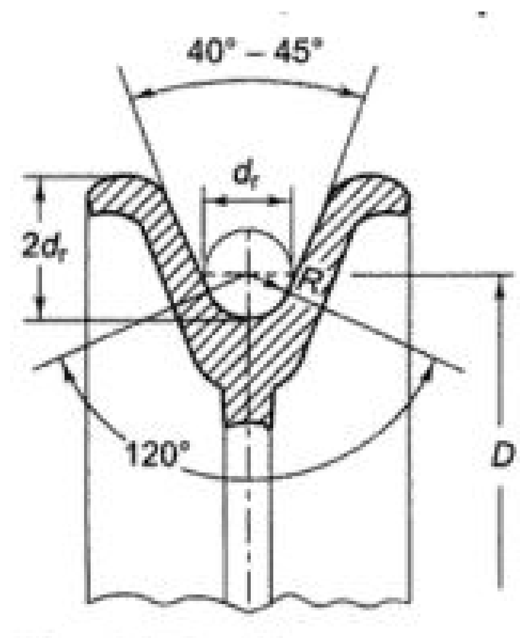

Figure 12: Schematic overview of the design parameters of a sheave

5.3.5 Drums

The drum and sheave diameters(D) are almost always standardized with respect to the wire rope diameter (d).

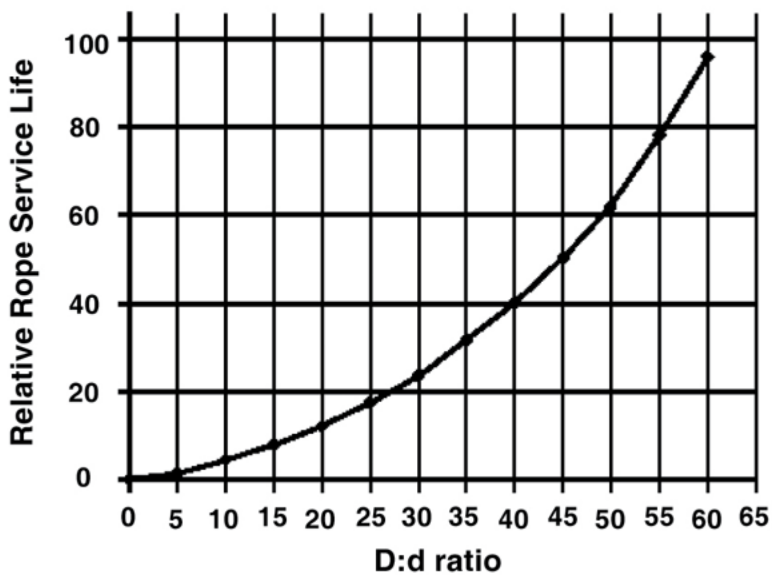

Figure 13: Rope service life plotted against the D/d ratio [5]

As can be seen from figure 13, the lifetime of the wire ropes increases with increasing D/d ratio. However, the costs and centrifugal force increase with increasing diameter. These consequences call for small sheave and drum diameters. On the other hand, there is a minimum ratio between drum/sheave and wire rope diameter. Since the diameter of the wire rope is known, it can be determined what the drum and sheave diameter must be according to table 14. [6] It follows that a minimum ratio of 23 must be used. In order to improve the lifetime of the wire ropes, a final D/d ratio of 30 will be used.

$$D \div d = 30$$

$$D = d \times 30 = 30 \times 30 = 900mm$$

The width, groove radius (r), groove depth (h) and pitch (p) of the drum must also be determined. It is recommended to multiply the wire rope diameter (d) by 0.54 to get the groove radius (r). To get the groove depth (h), the wire rope diameter (d) must be multiplied by 0.374.[6]:

$$r = 0.54 \times d = 0.54 \times 30 = 16.2mm$$

$$h = 0.374 \times d = 0.374 \times 30 = 11.22mm$$

The pitch can be determined in accordance with the groove radius. The minimum pitch can be obtained by multiplying the groove radius by 2.065 [6]. A minimum pitch length was chosen due to the fact that it significantly reduces the width of the drum. Therefore, the drum takes up less space.

$$p = 2.065 \times r = 2.065 \times 16.2 = 33.45mm$$

CONSTRUCTION	Suggested Min. D/d ratio
6x19 S IWRC	34
6x26 WS IWRC	30
6x25 FW IWRC	26
6x36 WS IWRC	23
Python® Multi	20
Python® Super 8	20
Python® Power 9	26
Python® Ultra	26
8x36	18
19x7 / 18x7	34
19x19	20
Python® Compac 18	20
Python® Lift	20
Python® Hoist	20
Python® Compac 35	20

Figure 14: D/d ratios

The maximum lift height of the hoist mechanism is 40 metres. The diameter of the drums is 0.9 m, the circumference of the drums is therefore 2.83 m, which means that every rotation of a drum winds up 2.83 m of wire around the drum. This means that the drums will have to rotate about 14 times to wind up 40 metres of wire rope However, in order to lift the spreader 40 metres, the drums should wind up twice the amount of wire rope due to the use of the sheaves.

$$n = lift\ height \times 2 \div circumference = 28.27$$

In order to calculate the absolute minimum width of the drum, the pitch will have to be multiplied by the amount of rotations and this number will have to be multiplied by two due to the fact that there will be two ropes per drum. However, this width is not realistic due to the fact that there will be excess areas on the outside of the drum. The actual width of one drum will therefore be estimated at 2.2 meter.

$$Minimum\ width\ drum = p \times n \times 2 = 1891.26mm$$

Lastly, a material will have to be assigned to the drums. This will be the same material as used for the drums used in the boom hoisting mechanism: grey cast iron class FG 200.

5.3.6 Bearings and lubrication

The drums are supported by two drum supporting bearings, these ensure a smooth circular motion. There could also be bearings located inside the sheaves. A lubricant will have to be used in order to prevent excessive wear of the wire ropes. As mentioned in the concept phase, excessive lubricant must be avoided since it has a bad influence on the environment. The bearings in the sheaves are often roller bearings. They reduce friction and also provide a smooth rotation.

5.3.7 Total assembly

Since all elements are now fully dimensioned, everything can be put together in one final design. It is important to carefully choose the position of all elements and to take into consideration the amount of space they will take up.

As already mentioned in the section of the trolley, 4 beams will connect the trolley to a platform beneath with all the machinery for the hoist mechanism. The machinery consists out of the drums, motors, gear reduction system and the snag load system. It was chosen to arrange the motors, drums and gear reduction system in such a way that it takes up the least amount of space. In between two drums there is a gear reduction system which is driven by two motors, as can be seen from figure 15. Since the width of the drums is estimated at 2.2 meter, the whole system is estimated to be about 5.2 meters wide. This will therefore easily fit onto the platform beneath the trolley, also leaving space for possible other equipment, but this is beyond the scope of this project. A schematic overview of where the hoist mechanism is located and how it looks like is depicted below.

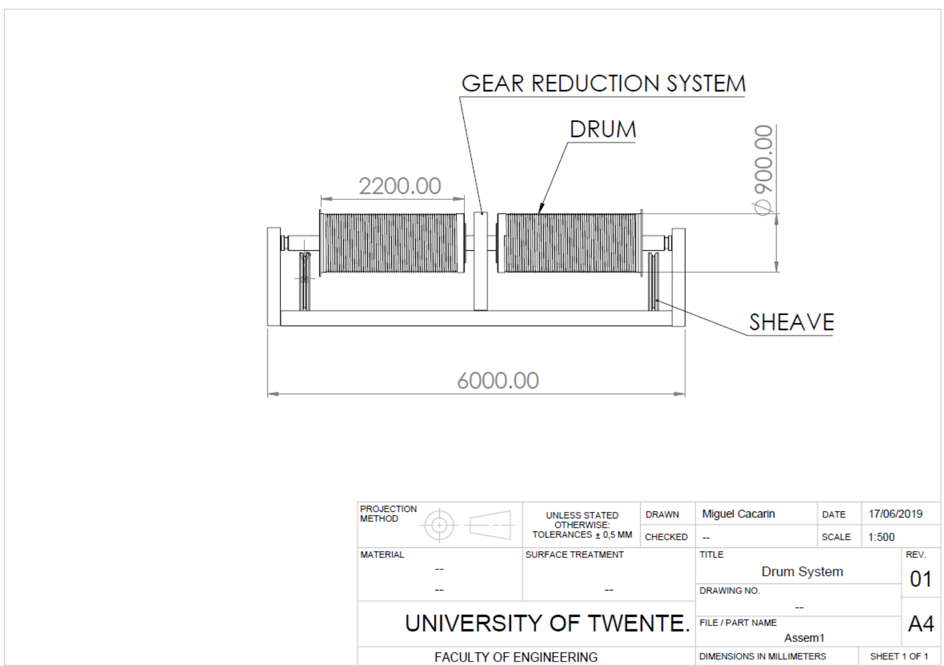

Figure 15: Front view of the hoist mechanism

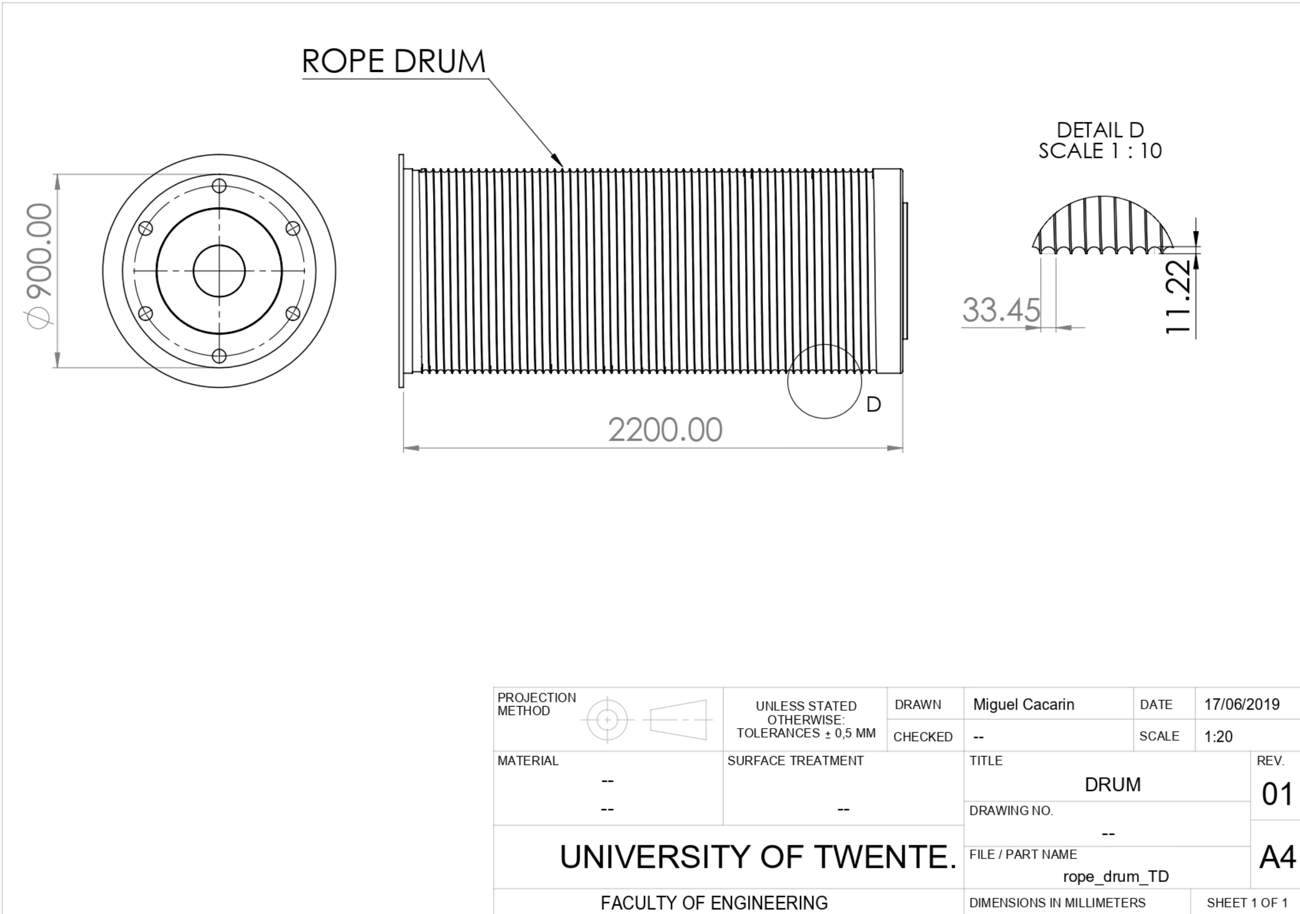

Figure 16: Rope Drum

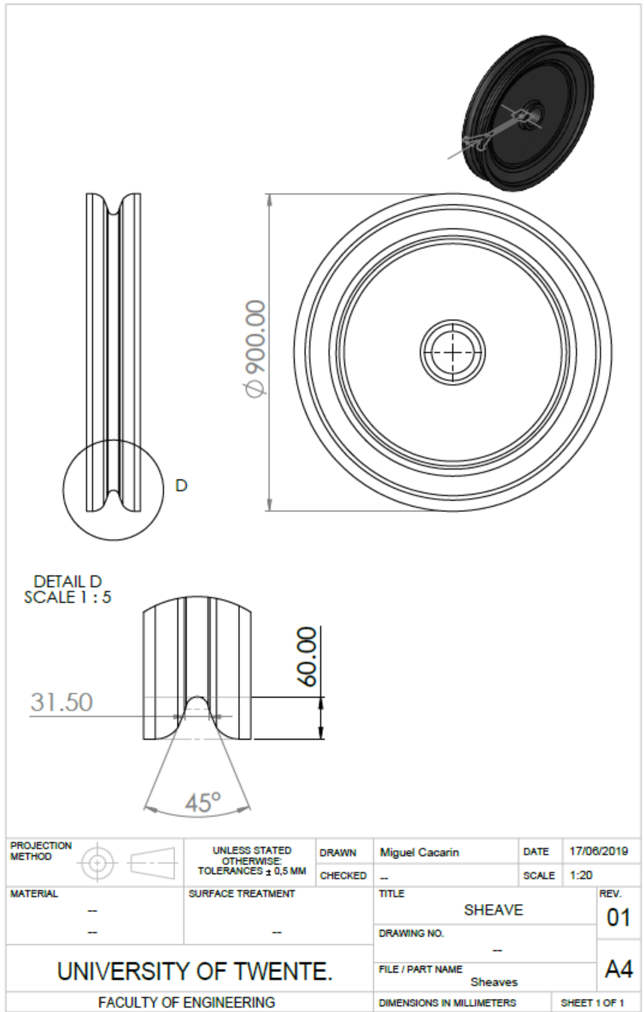

Figure 17: Sheaves

5.4 Trolley Materialization and 3D Model

5.4.1 Trolley Dimensioning

In this section of the report the crane's trolley will be fully defined in 3D space. Firstly, the requirements of the trolley are determined. Secondly, the size of the trolley will be determined. Later, the material choice will be documented. Then the rail system will be discussed. The last part will discuss the way in which the trolley's platform will be connected to the rail system.

Trolley requirements

- Trolley should be able to move at 180 - 210 m/min
- Hoist machinery should be able to be installed on the trolley
- Rail mechanism should be installed at the inside of the girders, the trolley hangs directly under these.
- Trolley should be able to run across the boom, from one end to the other.
- Trolley should be able to lift 50 tonnes of container load, the weight of the hoist mechanism and it's own weight.

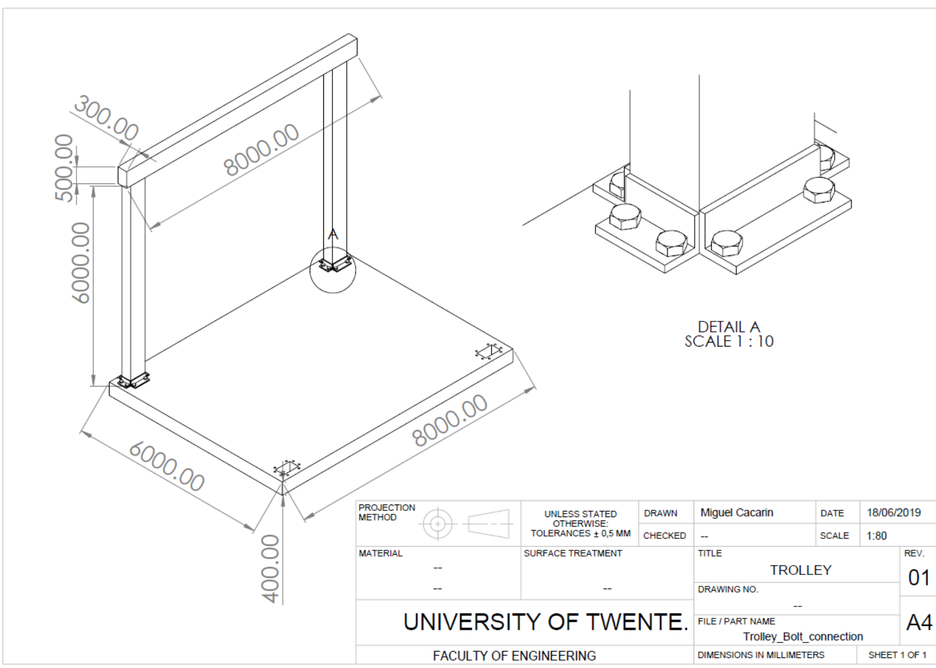

Figure 18: Trolley Dimension Overview Sketch

Trolley size

The width of the trolley is set at 6 metres. This value is chosen, because of the fact that the girders are 6 meters apart from each other. In this way, it is possible to place the rail system at the inside of the girders and the trolley hangs straight in-between and under the girders. Straight under hanging is preferable. However, it has to be taken into account that the width of 6 metres can change, due to the calculations that have to be done in the FEM package. This package calculates from the middle of the beams and the thickness of the beams is not jet known, so we cannot fix set the width of 6 metres.

- If the trolley is wider than the width of the girders, the trolley hangs completely under these. Only the rail system hangs in between the girders, the rest is useless empty space. The material is bend twice, to get to the right dimensions. These bending locations are under constant stresses, which is unwanted.

Another reason for not wanting to have a trolley that is much larger than the width of the girders has to do with the frame of the crane. If the trolley moves across the boom, it might collide with the frame, causing damage.

- If the trolley is considerably smaller than the width of the girders, the trolley can hang in-between the girders and underhangs completely. This is not a straight up issue, but when taking a closer look, again there is a lot of unused space.

Based on these considerations it is prefered to have a trolley width of at maximum not more than the width of the girders.
The length of the trolley is set at 8 metres. With the length at 8 metres, the surface area of the trolley is roughly 48 square metres. Not exactly, because some surface area is lost to the thickness of the material. With logical reasoning and literature research, it is believed that this area is large enough to house the hoist mechanism. The length of the hoist mechanism will be 6 metres if the machinery is mounted on the trolley. Furthermore, the length of 8 metres is also chosen, because of the chosen motor system to move the trolley along the girders. These so called truck systems are 4 metres long and to divide the forces best it was decided to install two of these on each side of the trolley. This system will be explained in more depth later in the report. The 4 bars have a thicker platform at the point where they touch the trolley platform. Here, 8 bolts per bar are mounted to keep the structure as a whole.

5.4.2 Rail system

Since the trolley is guided along the boom over a rail system, it was decided to document the rail system in the dimensioning of the trolley. The rails are exposed to huge loads from the trolley. Especially around the hinge point of the boom, where there is a little gap in the rails so that the boom can be lifted up, these loads are causing damage. It was decided that preferably a standardized rail system was chosen, which is built on these extreme conditions.

The company Gantrex inc., located in Nijvel, Belgium, has specialized itself in STS-crane rails. This company has researched the problems that arise commonly in STS-crane rails. These problems are deformation due to the loads, wear which results in shocks and vibrations, which leads to more damage, cracks that occur due to vibrations and failed clamps that occur due to shocks. With these problems in mind, Gantrex inc. has developed a new rail system in which these problems are minimized. The material used for this rail system is a Chromium-Vanadium-alloy. This alloy is able to withstand huge loads without deforming. Gantrex inc. has used the 110Cr-V alloy, which has a tensile strength of 1080 MPA and a yield strength of 700-800 MPA, according to CES Edupack. This alloy is also very resistant to abrasion, corrosion and oxidation, due to the chromium. Thermal expansion is low, roughly 11 microstrain/Celsius, where steels normally have between 5 and 35 microstrain/Celsius. Furthermore, to decrease shocks and vibrations, Gantrex inc. has developed a special vulcanized rubber layer, which is put in the rail at the hinge point of the boom. Another positive aspect of the Gantrex inc. rail system is that it can be precisely machined into the right dimensions. There is no fixed dimensions that have to be followed per definition.
At last, Gantrex inc. has used this rail system for a couple of years now and has its own engineers to install it.
All of the above considered, it was decided to choose this rail system material. [9] [10] [11] [12]

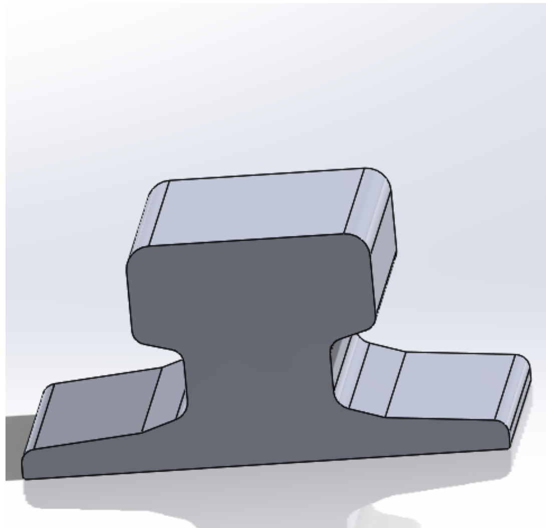

Figure 19: Railway shape sketch

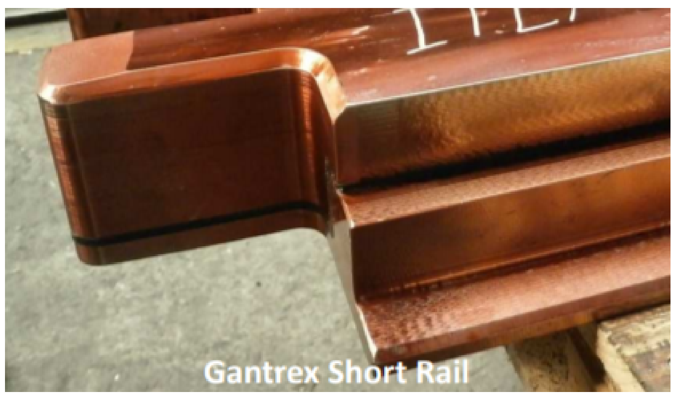

Figure 20: Gantrex inc. rail, close up of the rail how it is designed at the hinge point of the boom, including a black rubber strip near the bottom.

5.4.3 Wheels on rails

As mentioned before, a special type of trucks are being used, which not only effects the length of the trolley, but also the rails. It was decided that, in order to distribute the forces as much as possible, two trucks of 4 metres are installed on both sides of the trolley. The truck system chosen has 2 wheels per truck. In total, there are thus 4 wheels on each side and 8 wheels on the rails guiding the trolley. As can be seen in the sketch below, the wheels enclose the rails at both sides and the inside section of the wheel is 8 centimetres wide. These design aspects do not cause a problem with the chosen rails, because Gantrex inc. can machine the rails for this purpose.

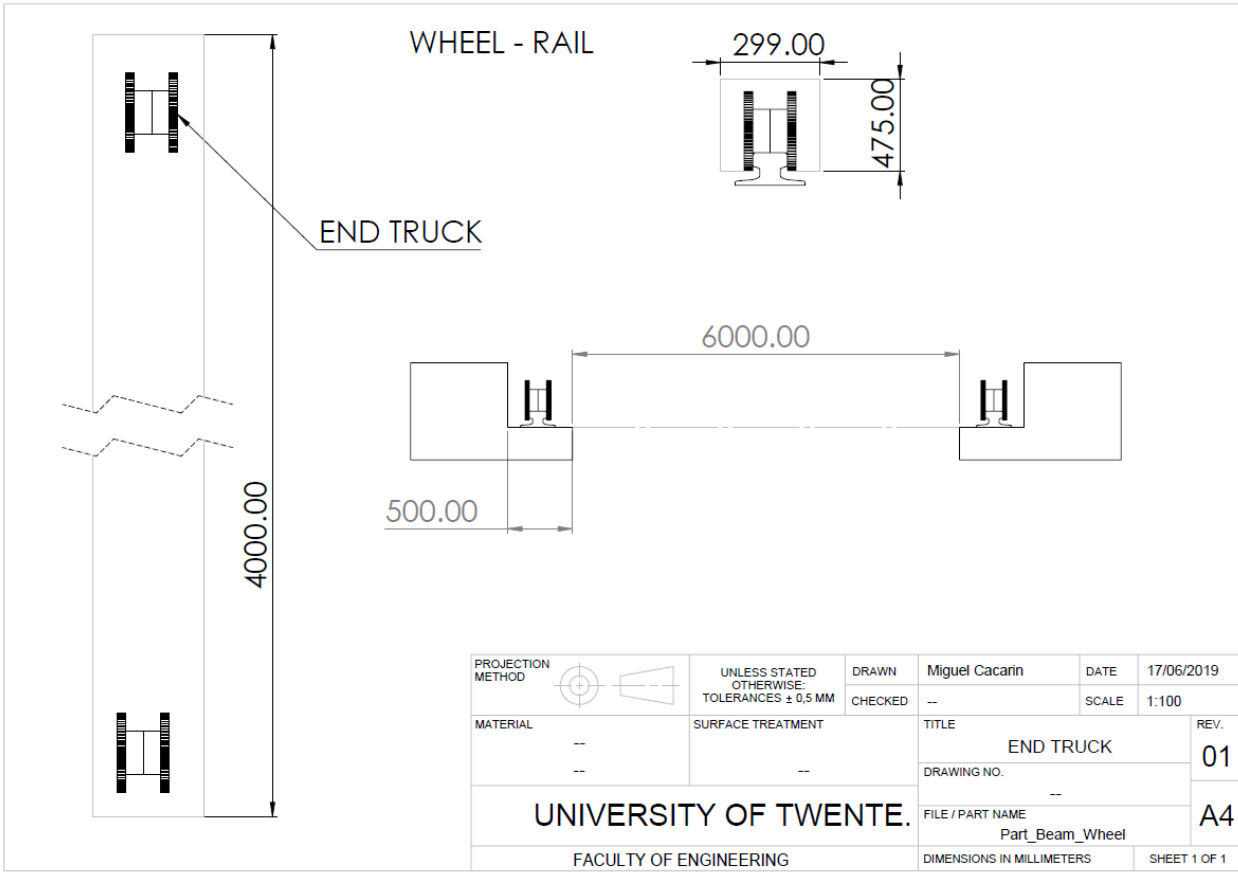

Figure 21: Sketch with the dimensions of one truck, top and front view and sketch with the current dimensions and placement of the trolley in between the girders.

5.5 Mounting the trolley platform to the rail system

As can be seen in the overall design sketch, the trolley's platform is connected to the rail system by means of 4 bars. These bars are 6 metres long, so that the hoist mechanism can fit in between the trolley's platform and the girders. These bars are made from the earlier discussed HSLA 945A steel, with the same reasoning behind it. The bars are bolted to the trolley's platform, because of the good machinability of the steel. On the other side of the bars, where these are connected to the bars that houses the motors. This connection will be welded.

Design 3D

To design this piece in 3D, It was taken as reference a real overhead crane design by Kito Crane, in order to complete our requirements. The trolley that has been designed is made up of two standardized I beams forming the body where the hoist mechanism will be mounted, and this will be below the boom, the next part are the end trucks in which we can find the wheels that allow the movement 2 wheels per end truck to be exact and also the driving system (electric motor) which transmits its power through gears that are connected to the wheels, the end trucks will be running on a rail that will help with its displacement fig.32 , The overlapping is smooth, avoiding vibrations and desalination due to the mechanism provided by the rail. [1] An important benefit of this type of design is that we can count on much more space and does not hinder any other part

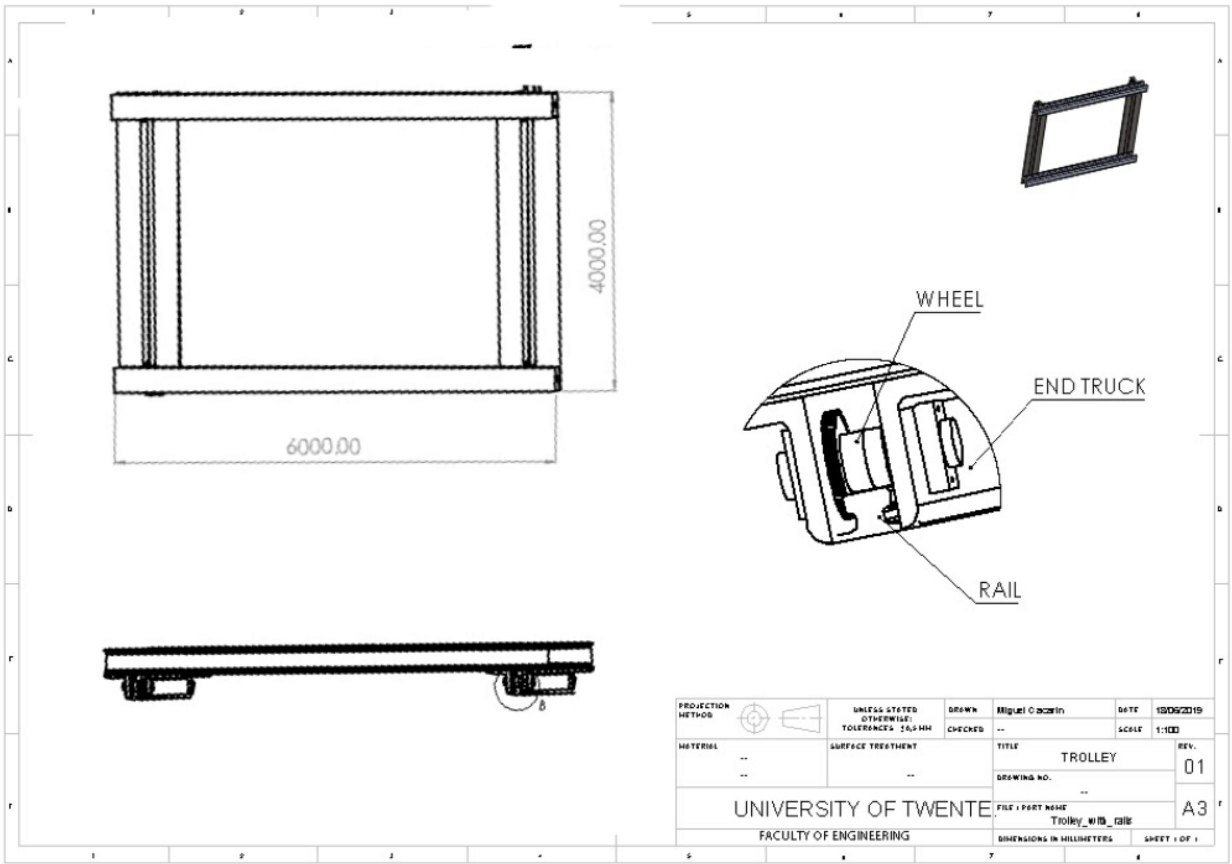

Figure 22: Trolley

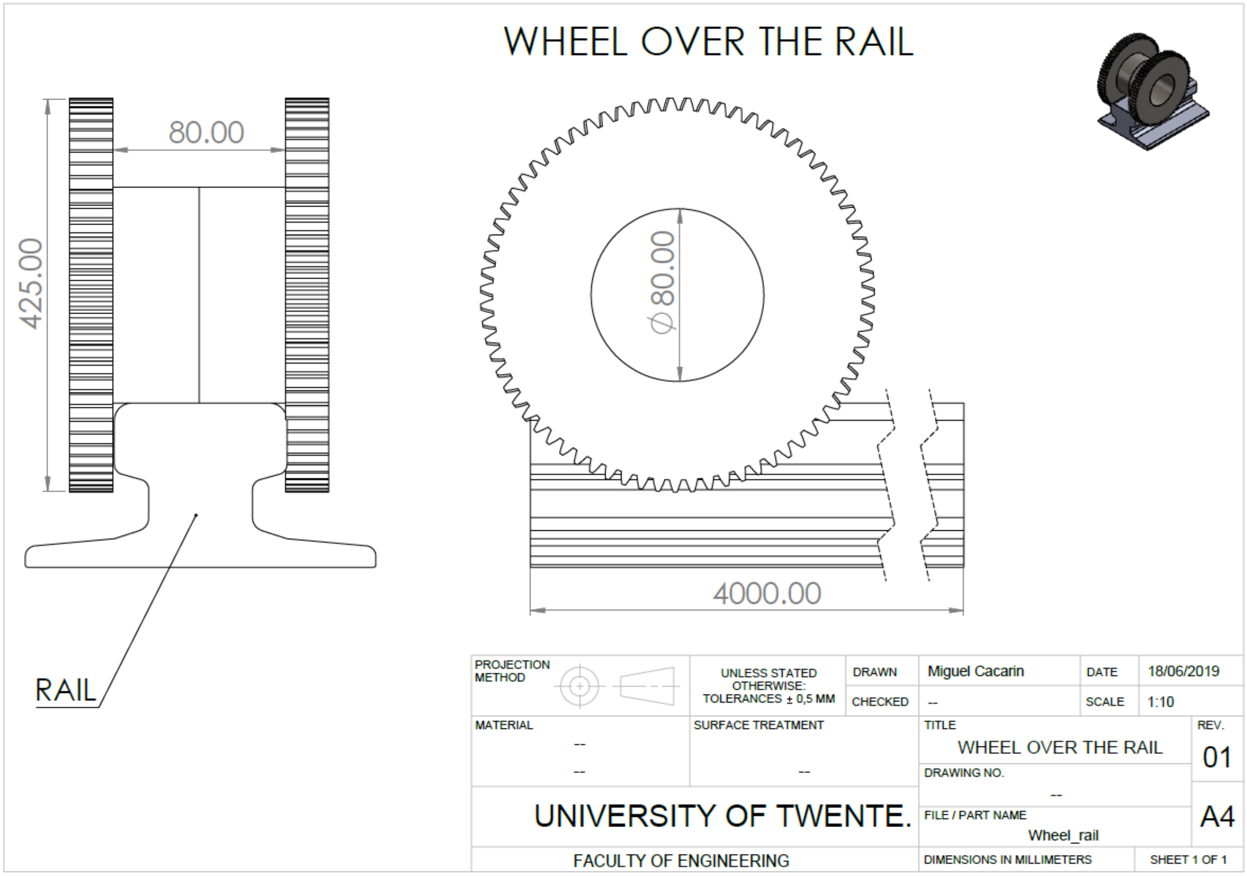

Figure 23: Wheel over the rail system

5.6 Static Calculation of Trolley

The Trolley is composed of four trucks in total with 2 trucks on each side, and each truck on its own contains two wheels. The length of each truck is 4m, 0.3m in width, and 0.5m in height (Appendix 9.5).

For simplification; in this static calculation the two trucks on each side are visualized as one single truck with four wheels. The the trolley must resist a load of 50 tons which is equivalent to $490,000 Newtons = F$, the load is split equally amongst the two trucks of the entire trolley. Hence each truck bares a load of $245,000 Newtons = P$.

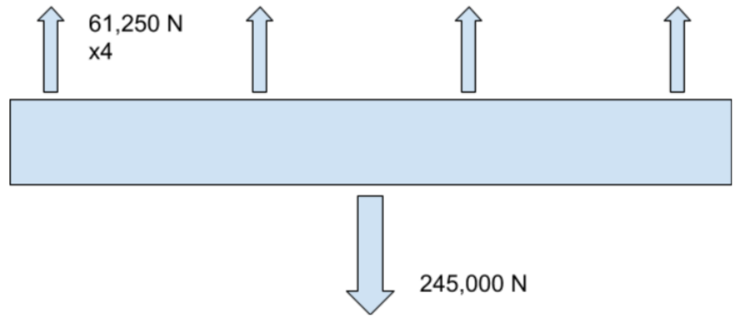

Figure 24: Single Truck

To determine the forces of the Trolley, a free-body diagram of one of the single trucks is created as seen above. With four reaction forces representing the wheels, and the load baring on the single truck.

The system is then further decomposed into the main contact area for the trolley system. Since there are four wheels with a load baring of 245,000 Newtons equally displaced upon all of them, each wheel thus retains $61,250 Newtons$

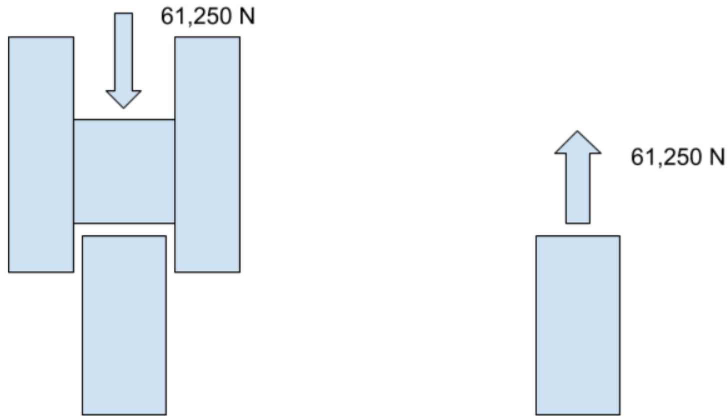

Figure 25: Static representation of Contact area between wheel and rail

The free-body diagram above shows the contact area between the rail and the wheel of the trucks.

Using the equation from Material Science, Stress was calculated

$\sigma = \frac{F}{A}$

$F = 490,000N$

$P = \frac{490,000N}{2} = 245,000N$

The Trolley was split up into it's truck components. Each truck component bares a load of $245,000N = P$. The truck is then split up into its four wheels.

The FBD of the Wheel and contact area has a reaction force of $\frac{245,000}{4} = 61,250N$

For reference to the dimensions of the single truck see (Appendix 11.5)

The Area of the Contact area is $A = b * h = 0.4 * 0.08 = 0.032m^2$

$\sigma = \frac{61{,}250N}{0.032m^2}$

$\sigma = 1.9MPa$

5.6.1 Material

The material choice for the trolley is important as it it is the main movement and load carry system. Since the trolley acts a STS trolley, it thusly operates in a humid salt environment and exposed to corrosion. Therefore, not only should the material choice for the trolley have a high strength, but also be corrosion resistant.

Material choice

To determine the material for the Trolley research was done on common materials used in crane construction and manufacturing of STS trolleys. First a more general conclusion of material class was found. HSLA (High Strength Low Alloy) was found to be used in many construction cranes and STS trolleys [15]. The prosram CES EduPack was used to determine if the material class would have a high enough yield strength when loaded. The range of yield strength for the HSLA class ranged from 500 and 590 MPa and the maximum stress on the wheels is $1.9MPa$. Therefore the material is a suitable choice as the yield strength is far greater than the stress of the load applied. The materials class was then specified to the material chosen for the STS crane trolley which is HSLA 945A Steel [13].

The reason the STS trolley is made of HSLA 945A is that this material has a high strength and high resistance to corrosion. Cranes are also made from Carbon Steels. These steels have a carbon percentage of 0.015 - 0.5. These carbon steels are not as strong as HSLA steels. Furthermore, since HSLA steels contain about 0.05 carbon and include other such elements such as, chromium, molybdenum, vanadium, titanium, and niobium. They are better for welding in comparison to carbon steels.

Both Carbon steels and HSLA steels are used in construction of Cranes. However HSLA steels are better suited for greater loads, welding, and corrosion resistance.

HSLA 945A was chosen as it is the highest ranked HSLA steel amongst others based on its varying properties.

Rank	Weldability	Formability	Toughness
Worst	980X	980X	980X
	970X	970X	970X
	965X	965X	965X
	960X	960X	960X
	955X, 950C, 942X	955X	955X
	945C	950C	945C, 950C, 942X
	950B, 950X	950D	945X, 950X
	945X	950B, 950X, 942X	950D
	950D	945C, 945X	950B
	950A	950A	950A
Best	945A	945A	945A

Figure 26: HSLA Grade Comparison [13]

The Society of Automotive Engineers (SAE) maintain standards for different grades of HSLA steels. The table above ranks the different HSLA grades from worst to best based on their properties.

As can be seen in the table above the HSLA 945A ranks the best in weldability, formability, and toughness.

Bearings for Trolley
Research was done on the type of bearings used in STS trolleys. Roller bearings were found to be the bearing used in STS trolley components, and thus is chosen to be the type of bearing in this trolley design. Roller Bearings are designed to carry heavy loads, the bearings are designed with a cylinder instead of a sphere. This decision increases the surface area for the distributed load. This however means that the roller design can only resist radial loads, and not thrust load.

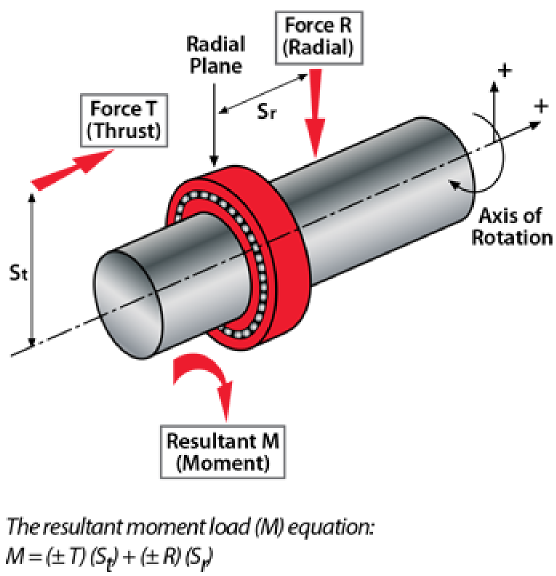

Figure 27: Static Drawing of Bearing [8]

The material choice
For roller bearing should have low friction, high corrosion resistance, withstand a high load, so that the trolley can carry 50 tons. The standard material choice for the bearings which withstand high loads is Standard 52100 Chrome bearing steel. This Standard is graded from SAE. The bearings are also coated with a highly corrosion resistant phosphate coating. The bearings are pre lubricated to reduce friction.
52100 chrome steel is composed of 1.5 percent chromium and the hardness for this material ranges from 60-64 on the Rockwell hardness scale (Rc).
Maintenance
To change out the roller bearings it can take up to 10-11 days, this includes grinding down the rails and fixing the trolley wheels. This is caused by bearing's developing a groove along the contact region of the rails which causes damage to the bearings over time. The rails need to grinded down due to the load deformation on the rails. The bearings are also designed to have access from the operator for inner and outer re lubrication. Furthermore, the bearings and sealed to further hinder corrosion and wear.
To combat the thrust loads on the bearings for the trolley, a low speed gearing is spur, thusly to eliminate these thrust loads.

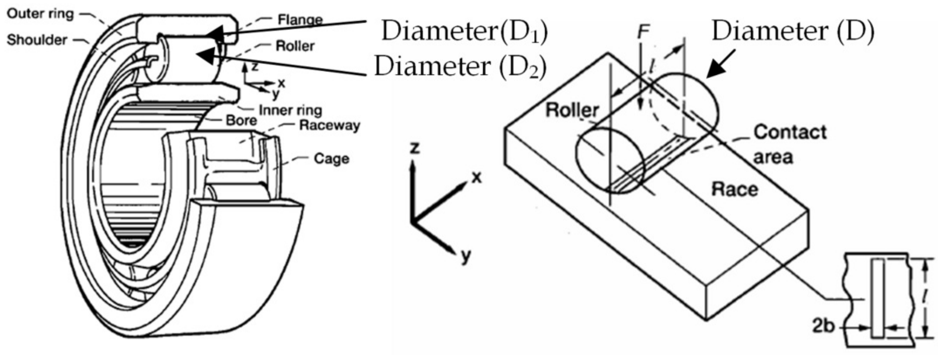

Figure 28: Schematic of Bearing [14]

5.7 Linear Guideways

Linear guideways are the rail tracks over which the crane moves along the harbour. Just like a regular train rail, the linear guideways consists of 2 parallel rail tracks. The crane has 4 contact points with the rails, 2 bars at the front and 2 bars at the end rail. The rails are positioned 42 metres apart. As can be seen in Figure 65, this distance is how far the bars are apart from each other. The total weight of the crane is 2650.6 tonnes. This weight has been determined with the FEM package, which will be explained later in the report. Since the complete weight of the crane rests on the linear guideways, these rails must have a high strength. It was decided to use the aforementioned Gantrex Inc. 110Cr-V alloy as the material for these rails. The rails is exaclty the same as the rails along which to trolley moves, see figures 29 and 30. As can be seen from the calculations in appendix 13.4.3, each bar that has a contact point with the rails, needs to have at least a contact area of 0.0464 square metres. This can be achieved by installing multiple large rail wheels, so that the weight is neatly distributed over, sometimes up to 8 wheels per rail, and that the contact area is always large enough. The empty space in between the bars and the guideways can be used to lay a road. This allows for trucks to move under the crane and pick a container up or give it to the crane. [25]

6 Finite Element Method and Final Structure Dimensions

In this subsection. The Finite Element Method in Matlab is used to compute the minimum weight of the beams in the crane while fulfilling the stated requirements in this section. Furthermore, all of the dimensions of the crane are fully defined according to the results of the FEM analysis. It is chosen to choose a worst-case scenario to optimize the strength and stiffness of the crane, namely the situation were the trolley force is acting as a tip force on the tip of the boom.

6.1 FEM initialization

At first, all the nodes and truss/frame elements were defined in the FEM package with the properties of the boom material giving figure 29. It was chosen to place the boom and the extension on top of the crane on a simple, yet strong crane frame based on the Liebherr crane frames in the Liebherr catalog [2] as these frames turn out to work good in practice and that not much material is needed for a strong structure. In the FEM model defintion, only the four nodes at the bottom of the crane frame were restricted in X, Y and Z linear and rotary movement.

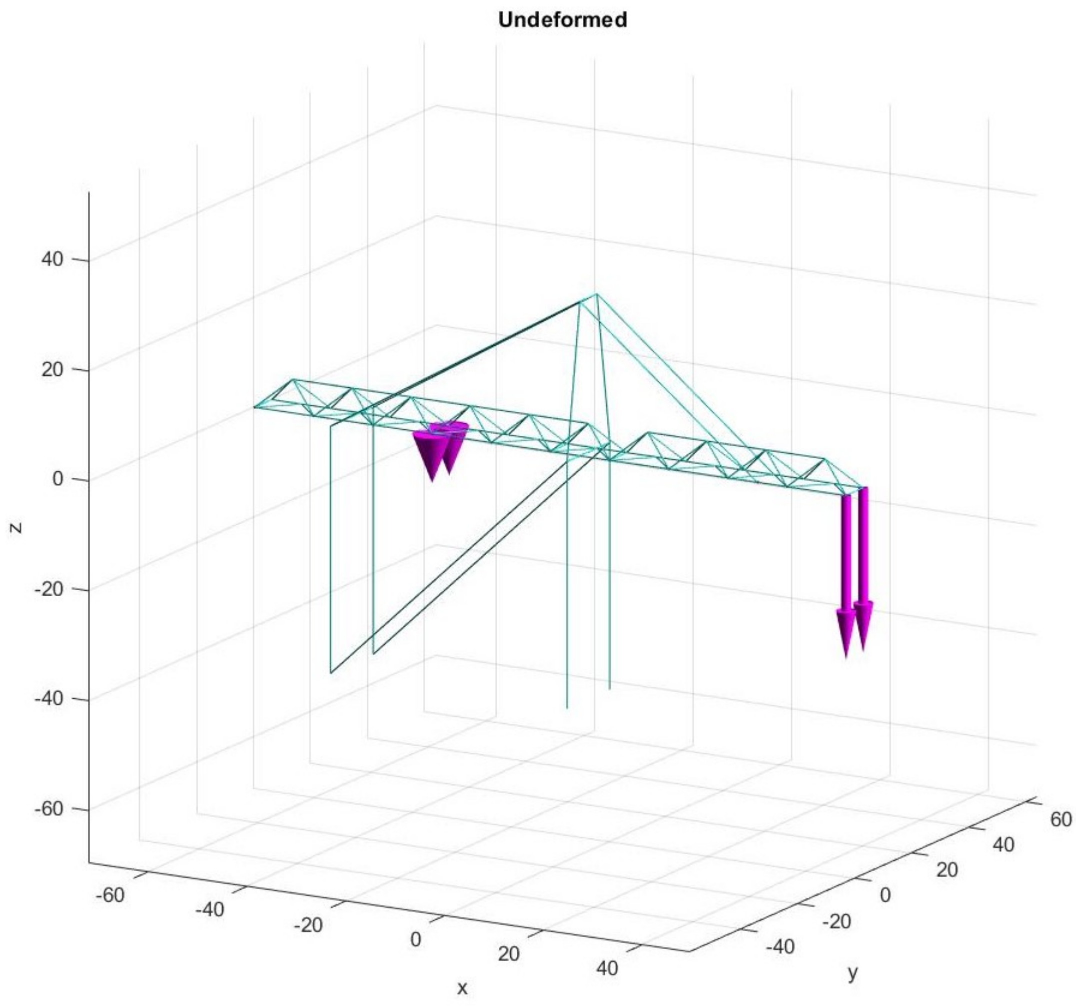

Figure 29: FEM model definition in undeformed shape

As can be seen, all dimensions of the frame are adapted to the truss sizes of the boom. It is chosen to keep as close to the original conceptual dimensions as possible. The final dimensions of the crane structure are shown in the technical drawings further in this report.

Three external loads were implemented, namely the trolley load + trolley weight, the load of the counter-weight as calculated and the maximum wind force as given in the project description. [1] All loads are multiplied with a chosen safety factor of 2. All forces were applied from the FBD's in figure 30 and figure 31 .

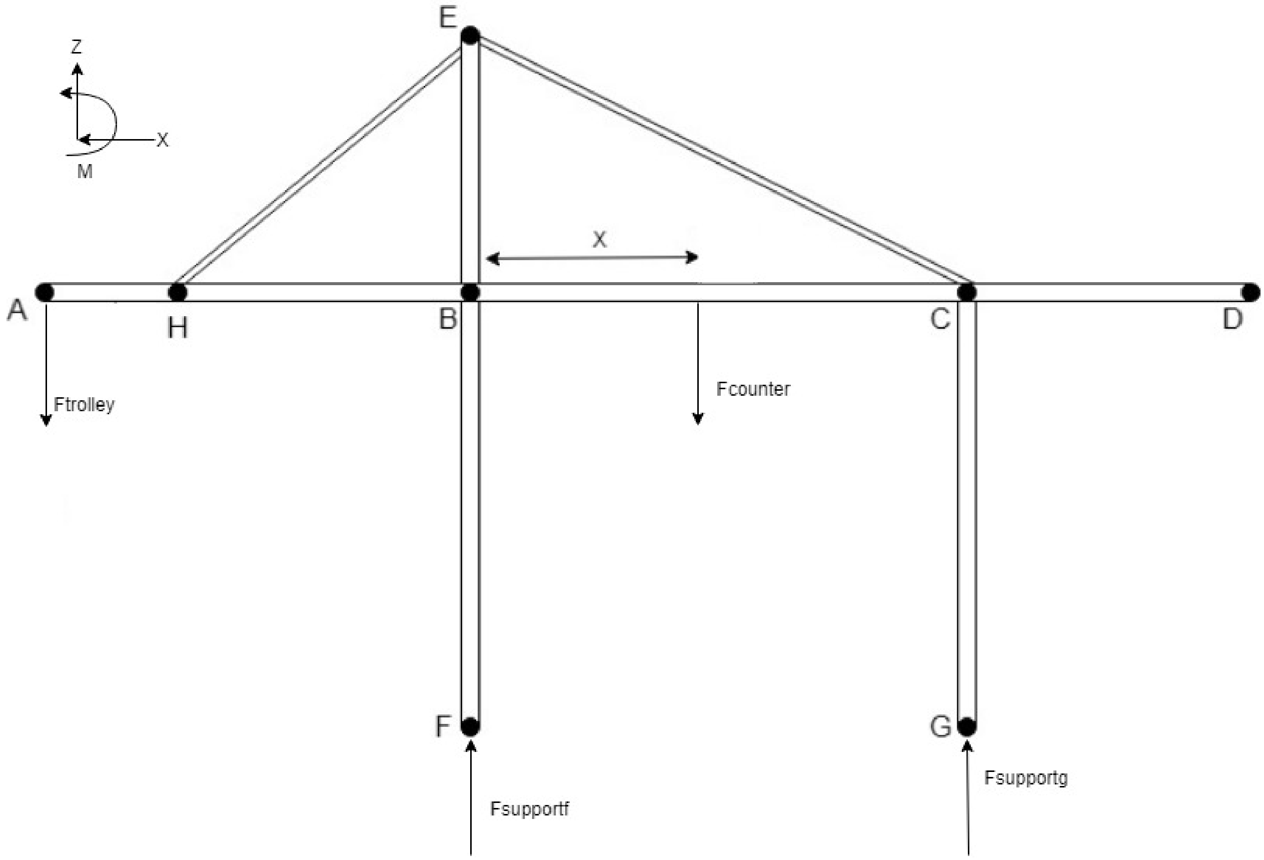

Figure 30: FBD of the side of the crane

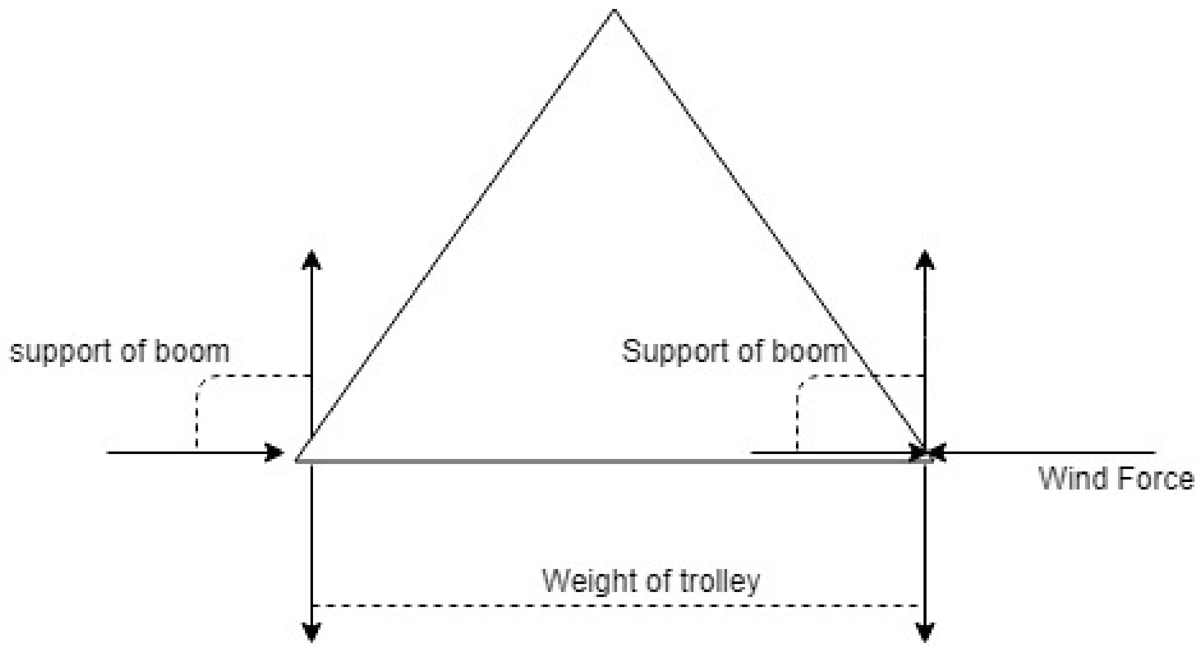

Figure 31: FBD of the front of the crane

The wind force is assumed as the force acting on the area given by (total boom length * the boom height) * safety factor. It was chosen to model with 5 different massive beams with a square cross-section, namely with a height of 0.25, 0.5, 1, 1.5 and 2 metres. At first, all beams were assigned with beams with cross-sectional length of 0.25.
It was chosen to model the steel cables as beams as cables can not be implemented in FEM.
The initial deformation can be seen in figure 32.

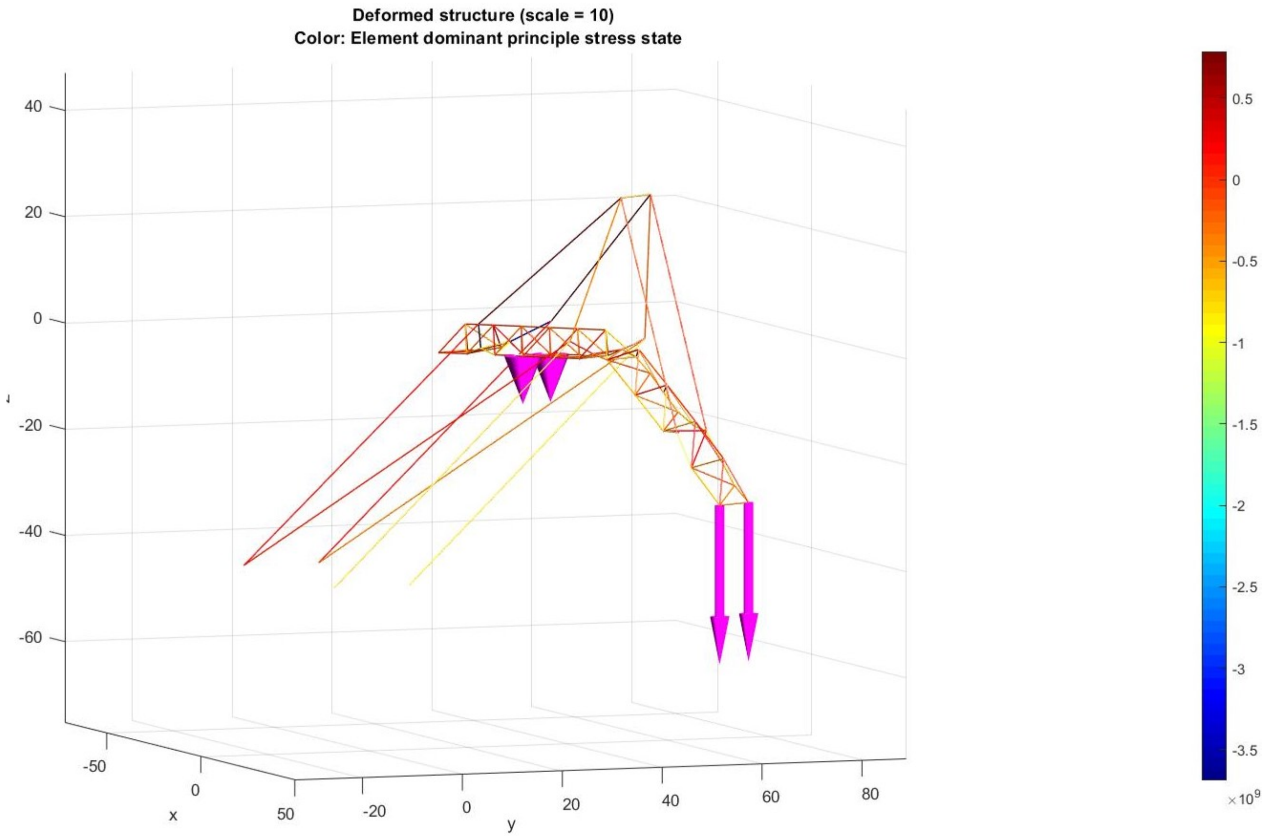

Figure 32: Initial deformation with stress scale

As can be seen, the structure clearly fails. This is solved with the following three iterations.

6.2 FEM Optimization

6.2.1 Iteration 1

The goal of the first optimization was to fulfill the displacement requirements at the tip of the boom, namely 5mm in y-direction, 60 mm in x-direction and 150 mm in z-direction.
Thicker beams were assigned to critical elements like the crane frame according to the deformation matrix in FEM. Also, a few extra elements were placed to reduce a large amount of shear stress. These were placed at the side of the boom and at the bottom of the boom at the locations were the cables are attached to the boom and were the boom is attached to the crane frame at the back of the crane. A close up of the location were the cables are attached to the boom with the extra elements can be seen in figure 33
The deformation figure after the first iteration can be seen in figure 34.
The deformations on the tip of the boom were:

Table 1: Deformations after iteration 1

	U_X (m)	U_Y (m)	U_Z (m)
Node 6	-1.5856	5.0992	-145.02
Node 15	-2.0161	4.9061	-144.39

These deformation were considered to be decent enough to continue to the next iteration. Also in this iteration, a situation were the trolley force was placed at the place were the force of the counterweight is acting was also made in order to check whether or not the structure would fail when the trolley is placed within the supports of the crane frame. In this situation no failure or non-acceptable displacements occurred.

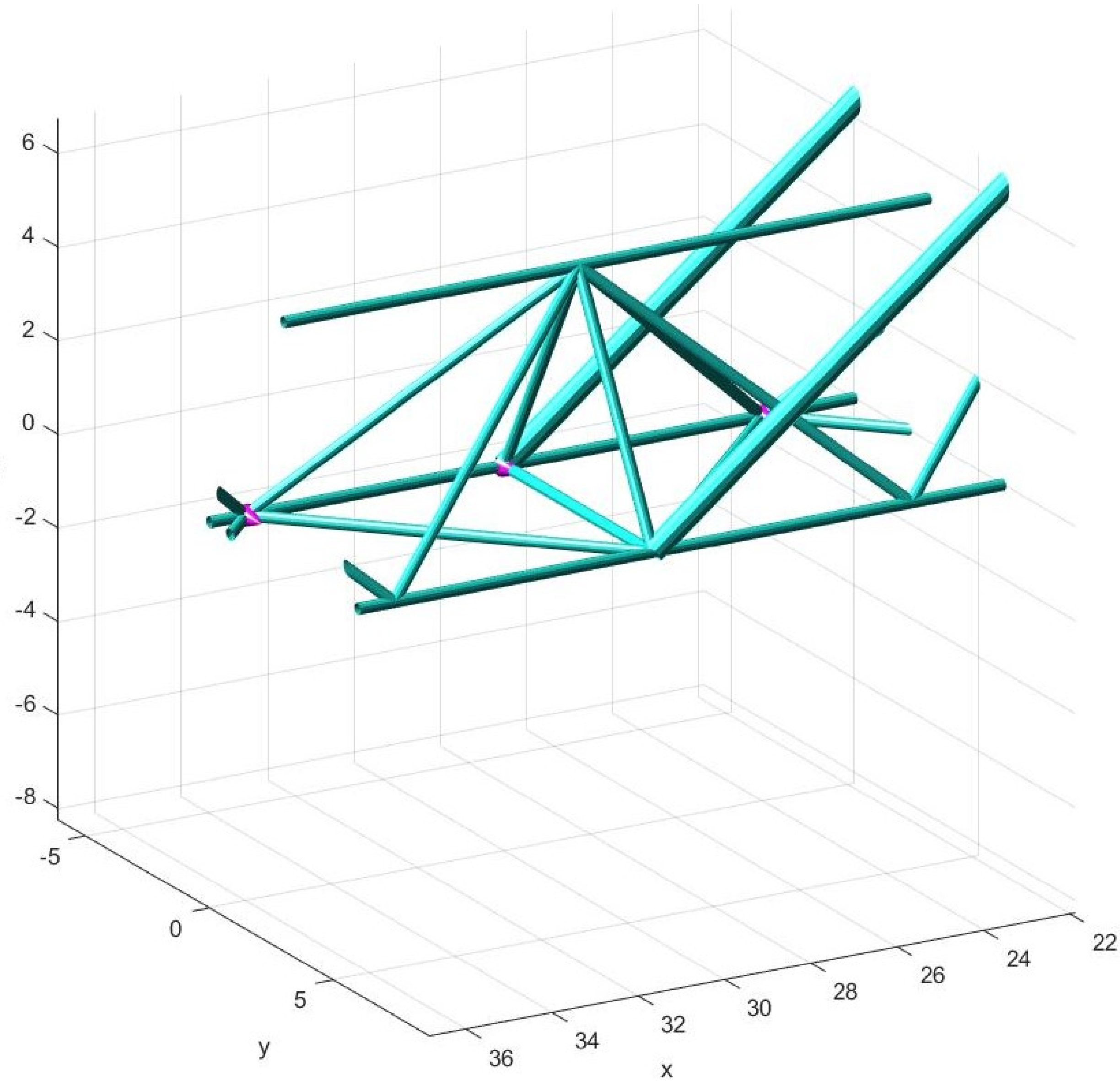

Figure 33: Close-up of the cable attachment points with extra elements

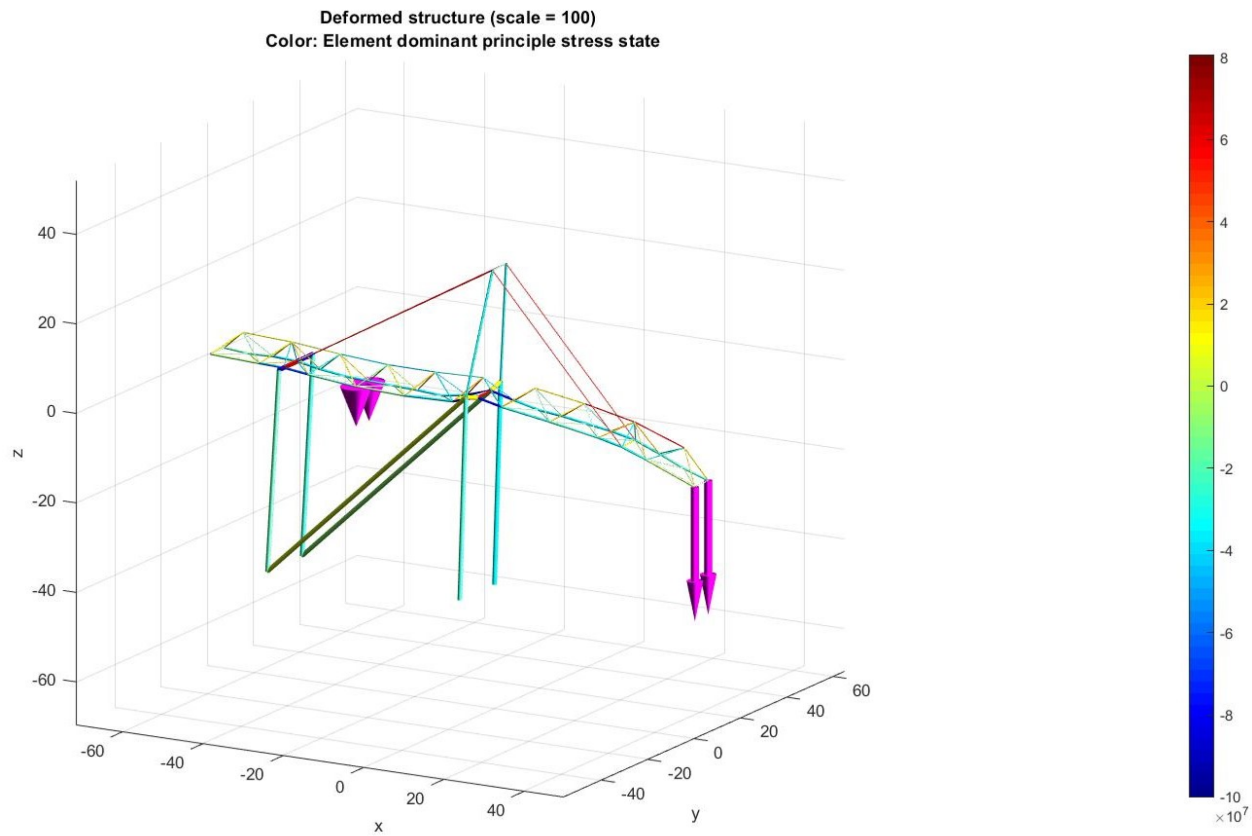

Figure 34: Deformation figure after first iteration

6.2.2 Iteration 2

The goal of the second iteration is to make sure all beams do not buckle and all stresses in the beams remain below the yield stress in order to prevent plastic deformation. This was done by letting FEM calculate all the internal reaction forces and moments with its stresses.
$\sigma_y = 2.00$ GPa was used in normal direction of the cross-section of the beam as worst-case material property. The maximum compressive stress was derived using a mechanical buckling analysis of the situation in figure 35 to simulate one frame element.
The analysis gives a buckling mode of $v_{critical}(x) = -(\delta + \frac{M_B}{F})cos(\frac{\pi}{2}x) + \delta + \frac{M_B}{F}$ were v is the vertical displacement, δ the total displacement at x = total length, M_B the internal moment at the point where F is acting. This gives a critical load of $F_{critical} = \frac{\pi^2 EI}{L^2}$ where E is the materials Youngs modulus, I is the moment of inertia of the cross-section of the beam and L the beams total length. This was implemented in the FEM package to find the elements with excessive internal stresses.

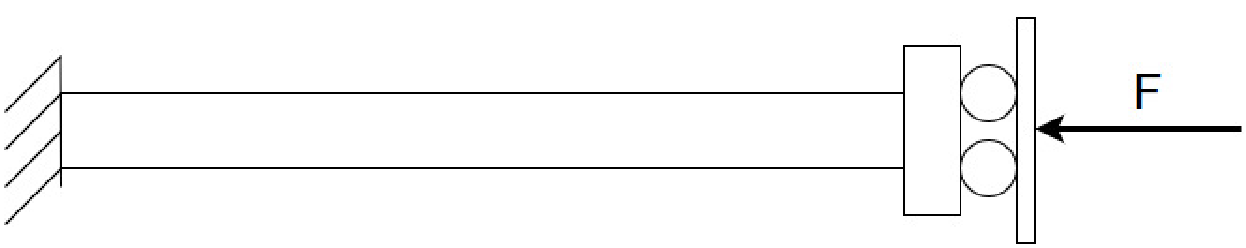

Figure 35: Buckling model

To achieve the goal, the FEM package was extended to find stresses which were outside our constraints. It appeared that with iteration 1, all buckling was corrected and no stress was higher than the worst-case yield stress of the material. This resulted that no changes had to be made in iteration 2.

6.2.3 Iteration 3

The goal of this iteration is to reduce the weight of the construction while the requirements are still fulfilled. This was done by adjusting the cross-sectional area and thicknesses of the beams, giving the following beam specifications:

Table 2: Final Beam Dimensions

	Cross-sectional height (m)	Wall thickness (m)
Beam 1	0.25	0.1
Beam 2	0.4	0.08
Beam 3	1	0.25

In general, beam 3 was assigned to the elements of the crane frame as these elements have to withstand large weights of the whole crane. Furthermore, a mixture of beams 1 and 2 were assigned to the truss elements in the boom depending on whether or not the elements have to withstand larger stresses like in places as the hinge point, the place where the cable is attached to the boom and the places where the boom is attached to the crane frame.

Also, the structure still fulfilled the requirements regarding deformation in the tip of the boom, buckling and yield stress.

Furthermore, the force in the cable element was determined in order to check the initial stated cable diameter. The total principal force acting on the cable is 3286.2 kN. Although the calculated force is lower than the initial stated 400 tons per cable and the cable diameter can thus be down scaled, it was chosen to keep with the initial stated cable diameter as to include an extra small safety factor.

The maximum stress found in the construction was 6.7850e7 Pa in the element placed in X-direction on top of the boom directly after were the cables are attached to the boom. This stress is still way below any maximum stress.

The weight of the total structure was successfully reduced from 16827 tons of steel in iteration 2 to 2650.6 tons of steel after iteration 3. Giving a reduction of circa 14200 tons. This means that the goal of this iteration was achieved. The deformed structure can be seen in figure 78.

The final Matlab code for the model definition can be found in the appendix.

6.3 Final Dimensions

Ones the FEM simulation was completed technical drawings could be made of the crane. These drawings can be found in section 10.9. The three different types of beams are described in section 6.2.3.

7 Machine Elements

7.1 Calculating forces and torque in hoisting drums

The total force pulling down is,

$$60000 \times 9.81 = 588600N$$

Due to the sheaves attached to the spreader, only half of this force is experienced by the drums, which is equal to 294300N. Therefore, the load acting on one drum is 147150N. Besides that, an efficiency of 75 percent is assumed. The total force acting on the drum then becomes,

$$147150/0.75 = 196200N$$

The radius of the drum was previously set to 0.9m. To calculate the torque on the drum. The total force acting on the drum should be multiplied with the radius of the drum.

$$T_{drum} = 196200 \times 0.45 = 88290Nm$$

In order to calculate the power that is required for the motor, the torque on the drum should be multiplied with the angular speed of the drum. To calculate the angular speed of the drum the circumference and the linear hoisting speed are needed. Circumference of the drum is equal to 2.83 m. The wanted hoisting speed is 2 m/s. In order to achieve this lifting speed, the drum should rotate 2/2.8 rev/s , which equals 4.44 rad/s.

$$Power = Angular\ speed \times Torque = 4.44 \times 88290Nm = 392400W$$

The power that has just been calculated sets a requirement for the motor to be chosen. That is, the power of the motor should roughly equal 400kW.

7.1.1 Motor Choice

A motor can be selected according to the power required. With the help of MATLAB, a suitable motor could be selected. For each motor it was determined whether or not it can produce a sufficient amount of torque and if it has a suitable inertia ratio. Some assumptions were made:

- The maximum torque a motor can produce is the continuous (rated) torque divided by 0.6.
- The hoisting speed should be able to accelerate to 2m/s within 5 seconds.
- The wall thickness of the drum is 3 centimeter (although it does not really influence the outcome).

A 400 kW motor was chosen according to Catalog D 81.1 of Siemens [16]: Simotics SD self-ventilated or forced-air cooled motors - cast-iron series 1LE5604 Performance Line: 400 kw, 4-pole, 1492 rpm at 50Hz.

$$Inertia\ ratio = 1.4721$$

$$Max\ Torque\ Motor = 4266.7Nm$$

$$Torque\ Drum\ Continuous = 88290Nm$$

$$Torque\ Drum\ Acceleration = 130540Nm$$

$$Torque\ Gear = 149990Nm$$

T_{Gear} should be larger than $T_{DrumAcceleration}$ and $T_{DrumContinuous}$ since this implies that the motor will be able to provide enough torque to accelerate a container to the required speed within 5 seconds. The inertia ratio is quite low so this implies that the motor is pretty big and might therefore be too expensive. However, this is not necessarily a bad choice since it results in a high performance.

Table 3: Stress and geometry values found with the MATLAB script

	Diameter/width (mm)	**Allowable Bending Stress (MPa)**	**Bending Stress (MPa)**	**Allowable Contact Stress (MPa)**	**Contact Stress (MPa)**
Pinion 1	285/175	448.1	35.33	1551	416.88
Pinion 2	285/175	448.1	135.75	1551	833.57
Pinion 3	285/175	448.1	435.84	1551	1516.4
Gear 1	1095/175	448.1	26.95	1551	212.68
Gear 2	915/175	448.1	105.84	1551	465.22
Gear 3	810/175	448.1	344.24	1551	899.49

7.1.2 Determining gear ratios, size and stresses

A lot of information needs to be gathered in order to design suitable gears. Most of the information comes from the slides of the University of Twente[22]. Since designing gears is an iterative process, a MATLAB script was made to do the calculations. The script can be found in the appendix. In the upcoming paragraphs, a short explanation of the script will be given as well as the found values.
It was chosen to use a three-stage-transmission since this fits best with the total gear ratio of 35.19. The velocity ratio (gear ratio) of the first stage was chosen to be between 3.4 and 4.1 [22]. The number of teeth of the pinions were set to 19 since it is a prime number and any number higher than 17 avoids undercutting. Next to that, the amount of teeth for the pinion gears must ideally be between 14 and 20 for hoisting [22]. The number of teeth of all gears in the system are prime numbers. After following an iterative process to determine the other velocity ratios, the following values were found:

$VR_1\ (Velocity\ Ratio\ 1) = 3.84$
$VR_2 = 3.21$
$VR_3 = 2.84$
$N_p\ (Number\ of\ teeth\ pinion) = 19$
$N_{g1}\ (Number\ of\ teeth\ gear\ 1) = 73$
$N_{g2} = 61$
$N_{g3} = 54$
$Train\ Value = 35.058$

Since we know the amount of teeth on the gears, a trial value for the module can be picked with which we can calculate the diameter of the gears. With the width-diameter-ratio found in a table in the slides [22], the width (bp) can be determined. Since we now know the exact geometry of all the gears, the bending and contact stress can be determined. The formulas were picked from the slides whereas the book of machine elements was used to determine the constants in the formulas and the allowable bending and contact stresses.[23] After an optimization process, the values in table 3 were found. The optimization process basically consists of adjusting the module, width of the gears and the material until all stresses are lower than the allowable stresses. As can be seen, all stress values are lower than the allowable stresses found in the machine elements book. It could be more cost efficient to select cheaper materials for the gears that experience lower stresses.

Material: AISI 8620 Carburized and Case-Hardened Grade 2 ($E = 205GPA, \rho = 7850kg/m^3$)
$Module = 15$

7.1.3 Overview of all inertia's and transmissions

In this section, a schematic overview of the drive system and a table with all inertia's and transmissions are given. Next to that, the final inertia ratio (so including all gears) is calculated.

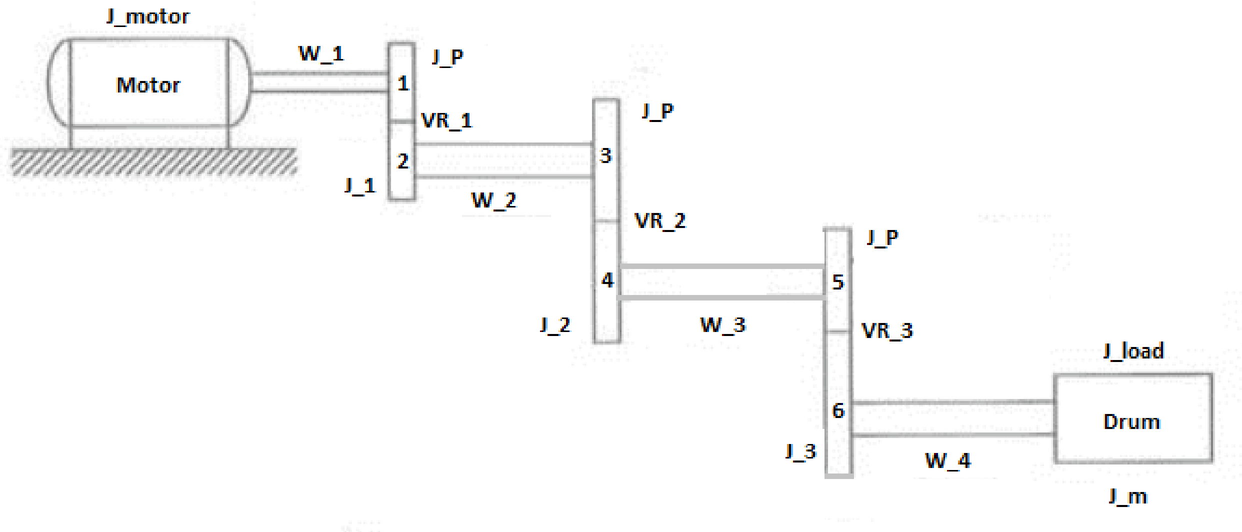

Figure 36: Schematic overview of the drive system

Table 4: Overview of all inertia's and transmissions

	Moment of Inertia (kgm2)			Angular Speed (rad/s)			Velocity Ratio
Motor	7.16		**Shaft 1**	156.24		**VR_1**	3.8421
Pinion Gears	0.88979		**Shaft 2**	40.67		**VR_2**	3.2105
Gear 2 (J_1)	193.89		**Shaft 3**	12.67		**VR_3**	2.8421
Gear 4 (J_2)	94.535		**Shaft 4**	4.4567			
Gear 6 (J_3)	58.056						
Mass (Container + Spreader)	4050						
Load (Drum)	127.59						

The reflected total inertia at the motor side was calculated:
$J_{Reflected} = J_{Motor} + J_p + J_1/(VR_1^2) + J_p/(VR_1^2) + J_2/((VR_1^2)*(VR_2^2)) + J_p/((VR_1^2)*(VR_2^2)) + J_3/((VR_1^2)*(VR_2^2)*(VR_3^2)) + J_Load/((VR_1.^2)*(VR_2^2)*(VR_3^2)) + J_m/((VR_1^2)*(VR_2^2)*(VR_3^2)*(i_D^2)) = 22.69Nm^2$

With this we could calculate the final inertia ratio:
$Final\ inertia\ ratio = J_{Reflected}./J_{Motor} = 3.17$

Due to taking all the gears into account, the inertia ratio got bigger. An inertia ratio of 3.17 may still be a bit low but as said earlier, it results in a high performance. If the inertia of the shafts would have been taken into account as well, the inertia ratio would even be a bit higher.

7.1.4 Shaft calculations

After the angular speeds of all the shafts are obtained as well as the gear ratios, the diameters of the shafts can be calculated. This first step is to calculate the reaction forces and internal bending moments at the positions of the gears. Besides that, the internal shear forces at the positions of the bearings have to determined. From there the shaft diameters can be determined. For shaft 2 and 3, torque is only transmitted between the two gears. Therefore, the diameters at the positions can be determined using both the bending moment and torque at these positions. The shaft diameters at the ends of the shaft can be determined using the vertical shear forces. For shaft 1, torque is being transmitted from the motor to the gear, and thus the bending moment and the torque at the position of the gear can be used to calculate the minimum shaft diameter. For shaft 4, torque is being transmitted from the gear to the drum. The diameter at the position of the gear can thus be calculated using both the bending moment and the torque. A schematic of how the 4 shafts are positioned is given below in figure 37.

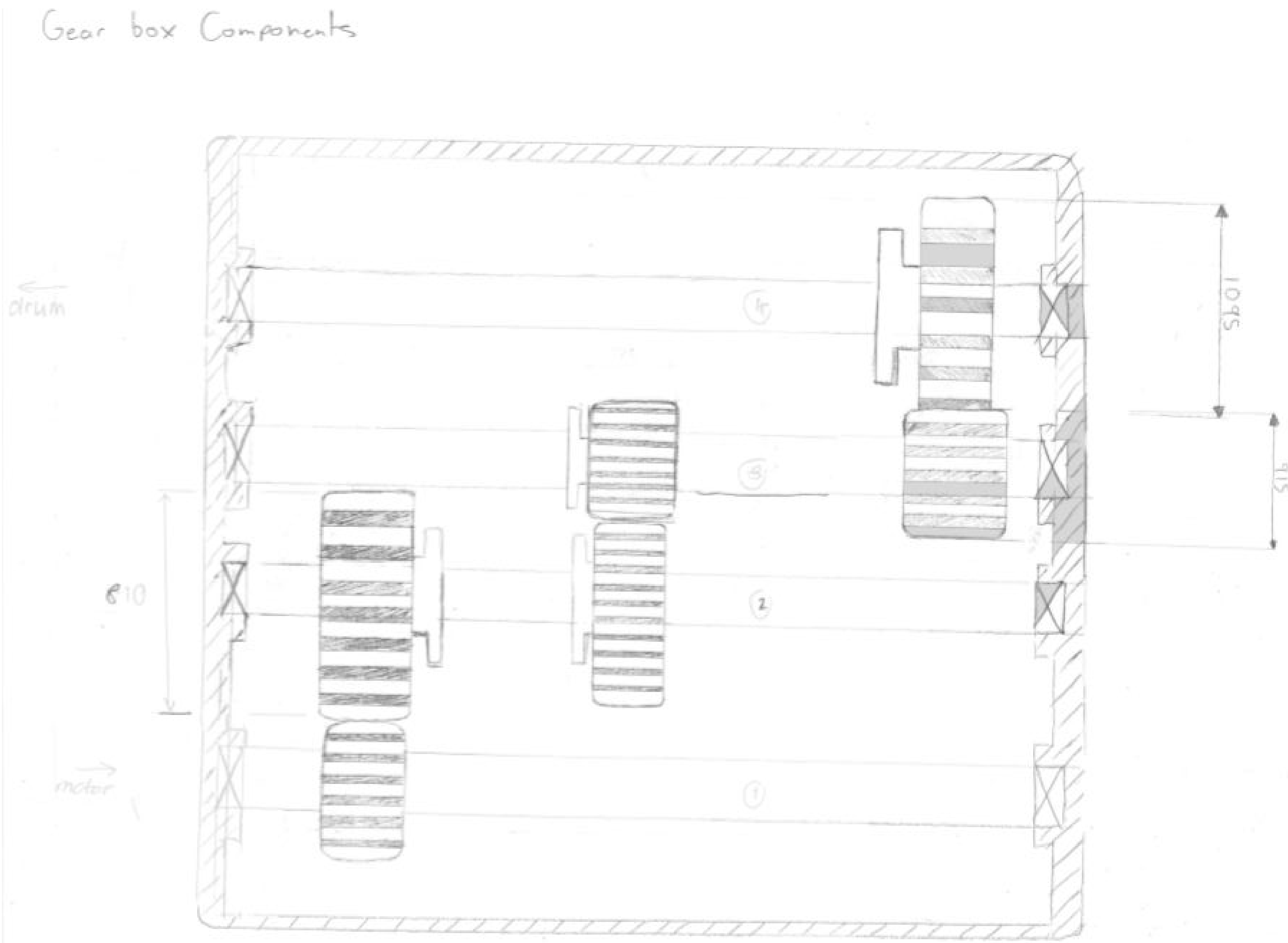

Figure 37: Schematic overview of the gearbox

A table containing all the minimum shaft diameters is for all shafts is given below. All the diameters are given in mm. The Matlab file showing how these values were computed is shown in the appendix.

Position	**1**	**2**	**3**	**4**	**5**	**6**	**7**
Shaft 1	42.9	88.73	9.99				
Shaft 2	46.00	95.83	128.60	204.06	185.51	138.83	40.15
Shaft 3	50.99	162.93	218.05	235.5	214.12	159.95	99.21
Shaft 4	140.39	209.17	92.85				

A very basic schematic of how a shaft looks with 7 diameters is given in the appendix. Where positions 1 and 7 represent the locations of the bearings and 3 and 5 the location of the gears.

7.1.5 Key calculations

In order to determine if keys are a suitable coupling choice, the minimum key length must be determined. This length should be shorter than the gear hub length. Keys have two types of failure modes, by shear and compressive stress, so the minimum key length will be calculated using two formulas: one for the minimum key length based on compressive stress (L_σ) and one for the minimum key length based on shear stress (L_τ). Table 11-1 in the machine elements book [23] was used to determine the standard width (W) and height (H) of the key spline with respect to the shaft diameter. To determine the final key length, the datasheet "Nosta Standard: DIN 6885" [26] was used. The reason the spline was chosen for shaft 4 is because the calculated key length was greater than the allowable key length.

$L_\sigma = \frac{4 \cdot T}{\sigma_d \cdot D \cdot H}$
$L_\tau = \frac{2 \cdot T}{\tau_d \cdot D \cdot H}$
$\tau_d = 0.5 * \sigma\_y/N$
$\sigma_d = \sigma\_y/N$

Table 5: Overview of key calculations

	Allowable Key Length (mm)	**L_\sigma (mm)**	**L_\tau (mm)**	**Key Length (mm)**	**Coupling Used**
Shaft 1	175	45.82	64.15	80	Key
Shaft 2	175	85.01	121.45	140	Key
Shaft 3	175	100.59	134.13	160	Key
Shaft 4	175	298.10	397.50	-	Spline

7.1.6 Hoist Bearing Calculations

As a result of the radial and the axial forces being determined in the section of the shaft design, the bearing for the 4 varying shafts can be determined. Spherical roller bearing were chosen as these types of bearings can withstand axial and radial load. Moreover, these bearings have low friction and are commonly used in rotary shafts and gearboxes. The lifespan of the bearing is determined so that the maintenance of bearing over time can be considered, as well as to test whether the bearing has a similar lifetime to the crane's lifetime. The Formulas used are derived from the catalog of Spherical Roller Bearings [27].

An example method of one of the bearing calculations is shown below
This is an example of the Dynamic loading lifetime calculation, the data used is from [27]

Equations and Constants for Determination of lifetime

The formulas used to calculate the Dynamic loads P_1, P_2 and the static load P_0 were found from the catalog [27] '
$P_1 = F_r + Y_1 F_a$ for $\frac{F_a}{F_r} \leq e$

$P_2 = 0.67 \cdot F_r + Y_2 \cdot F_a$ for $\frac{F_a}{F_r} > e$

$P_0 = 0.6 \cdot F_r + 0.5 \cdot F_a$

The Life Rating L_{10m} ($10^6 revolution$) and 90 percent reliability, and the Life Rating L_{10h} (hrs) and 90 percent reliability were taken from the machine elements book [26], while the Static safety factor S_0 was taken from the Machine Elements Presentation 'Roller Contact Bearings'.
$L_{10m} = (\frac{C}{P})^k$

$S_0 = \frac{C_0}{P_0}$

$L_{10h} = L_{10m} \cdot (\frac{10^6}{60 \cdot n})$

$C = 310kN$ for 60 (mm) diameter bearing
$C = 510kN$ for 150 (mm) diameter bearing
$C_0 = 335kN$ for 60 (mm) diameter bearing
$C_0 = 750kN$ for 150 (mm) diameter bearing

Y_1 and Y_2 are calculation factors which pertain to a specific diameter of a bearing d it's calculated e value as seen below. These factors are then used to determine the Dynamic loads of the bearings.

$Y_1 = 1.8$
$Y_2 = 2.9$

F_a and F_r are the external forces on the shaft determined in the calculation of the shaft diameters 7.1.4

$F_a = 10.082kN$
$F_r = 27.701kN$

k is the exponent which pertains to the type of bearing used, in this case it is $k = 10/3$ as it is a spherical roller bearing used.
$k = \frac{10}{3}$

For shaft 1 and a bearing diameter of 60mm
$n = 1492RPM$ The rotation speed of the shaft

To calculate which Dynamic load formula should be used the following value is calculated
$\frac{F_a}{F_r} = 0.37$

Since the value is 0.37 and there is no e value which corresponds to the calculated value above instead e is chosen to be 0.35 from the catalog [27]

chosen e is 0.35[27]

Due to the e value being smaller than the calculated value 0.37 formula P_2 is used

$P_2 = 0.67 \cdot F_r + Y_2 \cdot F_a$ for $\frac{F_a}{F_r} > e$

Substituting values for P_2 in

$47.8kN = 0.67 \cdot 27.701 + 2.9 \cdot 10.082$

Now L_{10m} can be determined, by substituting the P_2, and the corresponding C value for a 60 (mm) diameter bearing. $C = 310kN$

$508.7 \cdot 10^6$ rotations $= (\frac{310}{47.8})^{\frac{10}{3}}$

Following this L_{10h} is found

5683 hrs $= 508.7 \cdot (\frac{10^6}{60 \cdot 1492})$

The rest of the calculated life ratings hrs can be found below in Table 6, each value was calculated using the same method as shown above.

Table 6: Overview Bearing Lifetime Calculations

Shaft and Bearing Diameter	**Rotational Speed of Bearing (RPM)**	**Lifetime (hrs)**
Shaft 1, D = 60 (mm)	1492	5683
Shaft 2, D = 60 (mm)	388.96	21,555
Shaft 3, D = 60 (mm)	120.98	69,301
Shaft 3, D = 150 (mm)	120.98	423,286
Shaft 4, D = 60 (mm)	42.558	197,004
Shaft 4, D = 150 (mm)	42.558	1,107,750

Explanation of Table

Since the Gearbox was designed with 4 shafts with 2 bearings on each shaft with varying shaft diameters. Shaft 3 and 4 have two differing diameters in the bearings. Therefore as seen in the table above shaft 3 and 4 have two different lifetime calculation while shaft 1 and 2 have the same for each bearing located on shaft 1 and 2.

Safety Factor The Safety Factor was calculated in a similar method as the Life Rating were calculated, however instead the S_0, P_0 formula, and corresponding C_0 constant were used to to find the safety factor for both diameters.

For 60 (mm) diameter bearing, $S_0 = 15.465$

for 150 (mm) diameter bearing, $S_0 = 34.624$

For a spherical roller bearing is it advised to use a bearing with a safety factor above $4.0 \leq S_0$. Since the Safety Factor calculated above for both diameters 60 and 150 (mm), are both well above this value it can be concluded that the bearing are safe for use.

7.2 Highly loaded hinge

A highly loaded hinge in the crane that needs more detailing is the boom hinge.
At first, the forces on the hinge point are determined after which a bearing type is chosen and the hinge point design is worked out further.

7.2.1 Forces on the hinge point

The bearings in the hinge point will be subjected to radial and axial forces. To determine the axial forces, the wind force as assumed in the FEM section is chosen as this way of calculating the wind force already includes a safety factor. A boom area from the tip of the boom to the hinge point of approximately $48 * 5.2 = 250m^2$ gives a wind force and thus an axial force on the bearings of $250 * 240 = 60kN$.
The maximum radial force is determined by making a static analysis of the boom in lifted position as in lifted position the support of the cables in vertical direction is missing, resulting in a higher radial force on the bearings in the hinge point. The Free Body Diagram with only the vertical forces caused by the force F_B can be seen in figure 38 were F_B is the force caused by the weight of the boom, F_H the counter force in the hinge point and F_{s1} and F_{s2} the counter forces in the connections between the boom and the crane frame.

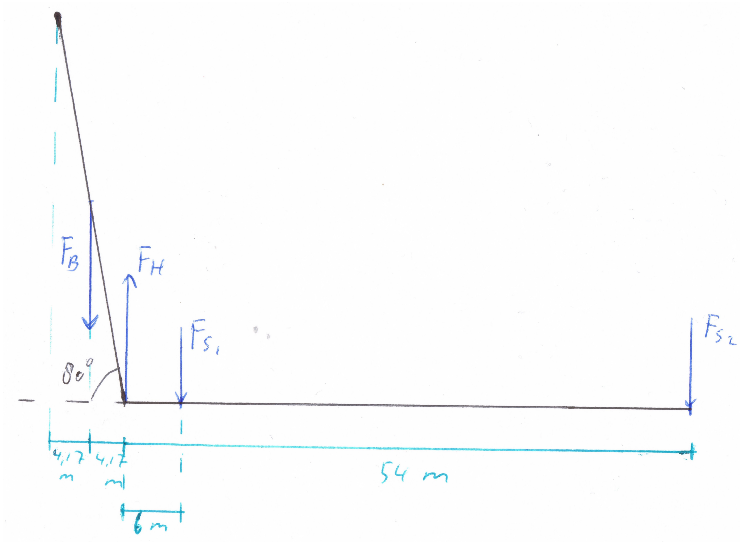

Figure 38: FBD of the crane construction in lifted situation

In this situation, three unknowns are present and two equations can be made, namely the equations of equilibrium for the moments in the hinge point and for the vertical forces. This makes the situation an underdetermined problem. To overcome this, it is assumed that F_B directly acts above F_H as the distance between F_B and F_H is relatively small. This results in $F_H = F_B$. F_B can be obtained from the FEM model resulting in $F_B = 2.05 * 10^5 kg = 2.01$ MNewton.
Taking a total of 4 bearings in mind as decided in section 5, the forces per bearing will be 15 KNewton in axial direction and 502.5 KNewton in radial direction.

7.2.2 Bearing selection

Now that the forces on the bearing are known, the specific bearing type can be chosen.
The specific considerations that have to be taken into account in this situation are that the bearings must withstand large radial as well as axial loads and have to withstand environmental effects. Furthermore, when the bearings are installed, they must be reliable and must have a lifetime as long as the lifetime of the crane as replacing the bearings is a highly impractical job. Also, the friction in the bearings must be as low as possible, so the motor power of the lifting cable drum can be as low as possible.
Taking this into consideration, a rolling contact bearing is chosen as bearing type as they have lower friction, can handle axial forces and last longer than plain bearings.
To chose the specific bearing type, first the minimum shaft diameter at the bearing locations was calculated. It was chosen to use the same material for the shaft as used for the boom as the same material properties are required. Determining the minimum shaft diameter was done with the formula for shafts with only significant vertical shearing force [17] as this situation can be concluded from figure 39.

$$D = \sqrt{\frac{2.94 * K_t * V * N}{s'_n}} \tag{1}$$

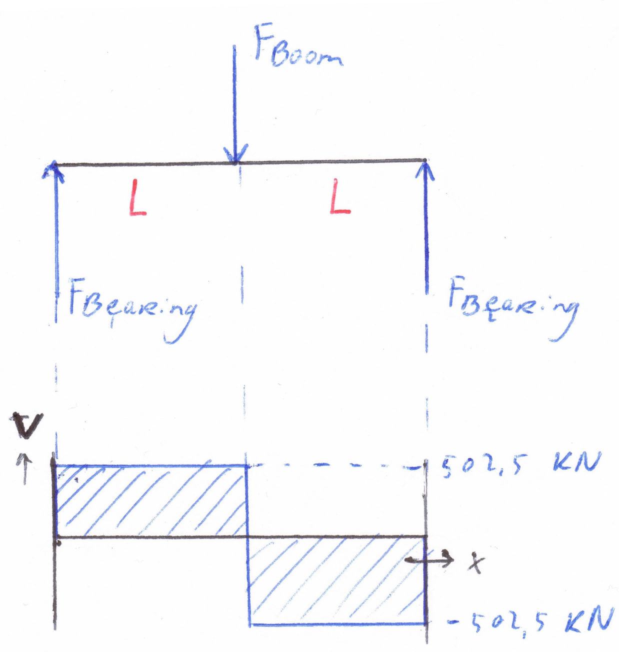

Figure 39: FBD with internal shear diagram of the hinge shaft one side of the hinge point

With K_t = 2,5 for a design with sharp diameter transitions, V = 502,5 KN as obtained from figure 39, N = 2 as a general safety factor [17] and $s'_n = s_n * C_m * C_{st} * C_R * C_s = 475 * 1.00 * 1.0 * 0.7 * 0.81 = 269.325$ MPa as the modified endurance strength as determined using [18].
This gives a minimal shaft diameter of 166 mm, which is rounded to 170 mm to implement another small safety factor and to be able to choose a standard bearing size.
To chose the right bearing type and size, the following has to be calculated with data gained from SKF (a large bearing deliverer):
Static load:

$$P_0 = 0.6 * F_{radial} + 0.5 * F_{axial} = 309.0 \quad KN \tag{2}$$

$P_0 < F_{radial}$, this means that $P_0 = F_{radial} = 502.5$ KN
Minimum static rating load:

$$C_0 = s_0 * P_0 = 1 * 502.5 = 502.5 \quad KN \tag{3}$$

As both the catalogs of SKF [21] and NTN [20] (another large bearing producer) do not provide a single-row, deep-groove ball bearing with the right C_0, it was chosen to look for another suitable bearing in the catalog of NTN. It was chosen to use a spherical roller bearing as these bearings have an excellent radial load capacity and a fair/good axial load capacity [19]. Bearing number 23934 with a bore of 170 mm, an outside diameter of 230 mm, a width of 45 mm and a C_0 of 650 KN fulfills the requirements and is therefore used.

7.2.3 Detailed design

Now the bearing size and the basic design from figure 7 were known, detailed drawings of one side of the hinge point were made. One side, because the hinge point consists of two identical hinge points located at each side of the boom.
It is chosen to construct the hinge as out of b2 from FEM surrounded by a socket with bearings, which is connected to the second b2 beam. The technical drawings of the hinge can be found in the appendix.
The following points are important to take in mind:

- At the outside of the socket over bearing 1, a plate can be bolted with a seal or welded so that the bearing is protected from environmental effects.

- The shaft is placed in the middle beam with a transition fit.
- All bearings are placed with an interference fit in the socket.
- The shaft is placed with an interference fit in Bearing 2 and with a loose transition fit in Bearing 1 in order to allow axial movement with for example thermal expansion.
- A clearance of 10 mm at both sides between the middle beam and the socket is chosen to allow small positioning errors and thermal expansion of all parts.

7.3 Highly loaded bolt

A highly loaded bolt in the crane that needs more detailing are the bolts that connect the trolley's platform to the 4 supporting beams as can be seen in figure 18.
It was chosen to bolt the four pillars with flanges on each side of the pillar to the trolley platform. Each flange is fixed with two bolts, resulting in 32 bolts in total for four pillars.
It was chosen to include a safety factor of 5 due to possible dynamic behaviour of the boom due to possible shocks of the load. As the total mass of the trolley and load was assumed to be 60 tons, the load on the 32 bolts including safety factor will be rounded up to 3.0 MNewton.
This gives a force per bolt of 93.75 KN.
A bolt grade of 12.9 was chosen to be sure the bolts are strong enough to hold the load. These bolts have a proof strength of 970 MPa [23].
As only significant tensile forces occur in the bolts, a minimum bolt area of $A = \frac{F}{\sigma} = \frac{93.75*10^3}{970*10^6} = 96.65 \quad mm^2$.
This results in a bolt size of ISO-metric M16.

Pretensioning
Due to possible dynamic behaviour of the boom, shear stresses in the bolt connection might occur. To prevent that these shear stresses are projected to the bolts themselves, pretensioning of the bolts should be applied. It was chosen to apply the force per bolt = 93.75 KN as pretensioning, because the safety factor of 5 should be enough to compensate the possible stresses.

Bolt connection design
The design of the loaded bolt connection with flanges can be seen in figure 18.

7.4 Highly loaded weld

For the calculations of the highly loaded weld, a weld of a point to which the boom is attached to the crane frame is analyzed. It was decided to choose the connection between element 58 and the continuous beam in element 54 and 55 at node 26 from the FEM model as can be seen in figure 40.

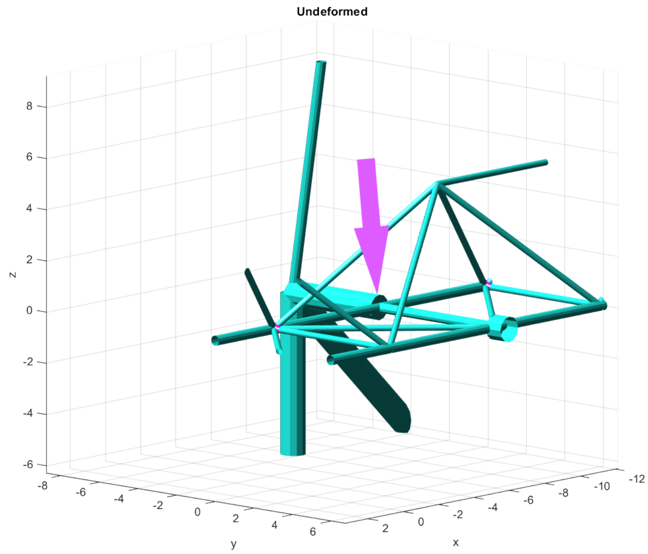

Figure 40: Elements of higly loaded weld

It is chosen to make as much weld length as possible. This results in a side few of the weld as can be seen in figure 41.

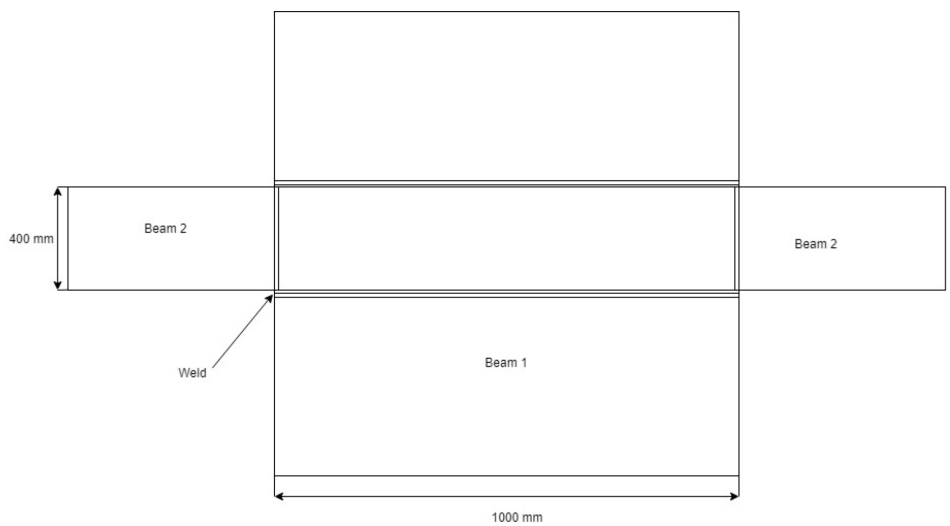

Figure 41: Highly loaded weld side view

7.4.1 Calculation of Weld Thickness

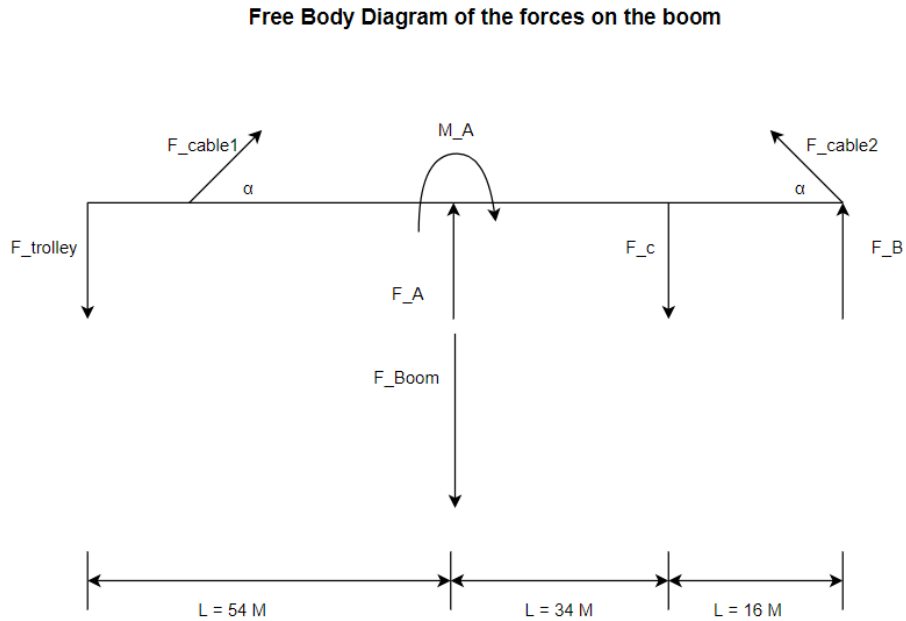

Figure 42: Free Body Diagram for the weld calculations

In the above free body diagram, all forces that are exerting a force on the boom are determined. Point A is where the weld is located. For the weld calculations, it was decided to look for the worst case scenario. In this case, the trolley is at the very end of the boom, causing a massive moment around point A. It was decided to set the reaction force Fb to 0 N. In this way it was ensured that the forces in point A were maximal. Using the results from the FEM analysis, it was possible to determine FBoom, FC and the Fcables. In appendix ... can be found how all reaction forces and moments were determined. With these forces known, it was possible to determine what kind of forces act on the weld, and use the right formulas to determine the weld thickness.

The Safety factor for the weld calculation was decided to be 3 as the point at which the weld takes place consists of shear forces, a bending moment and due to the change in the weight of the crane (from the hoist lifting) there is some possible dynamic deflection which happens at this point. Hence the Shear Force and the Twisting Moment were multiplied by 3 so that the thickness of the weld would be safe enough for when the crane is in use.

$V = 8.139 \cdot 10^6 N$
$T = 2.466 \cdot 10^5 N/m$
$C_v = 200(mm)$
$C_h = 584(mm)$
$x = 416(mm)$

b and d are the length and height of the weld as seen in Figure 41
$b = 1000(mm)$
$d = 400(mm)$

The allowable shear stress was found in the Machine elements book [26] this allowable shear stress was chosen as it is the highest stress. $\sigma = 228MPa$

The A_w and J_w formulas were derived from and schematic view of the weld in [26]
$A_w = 2 \cdot b + d$

$J_w = \frac{(2 \cdot b+d)^3}{12} - \frac{b^2 \cdot (b+d)^2}{(2 \cdot b+d)^2}$

Substituting the values of b and d in we get
$A_w = 2400(mm)$
$J_w = 0.335(m^3)$

The formulas for the forces f_s, f_{th}, and f_{tv} were taken from the machine elements book [26] These formulas are used to calculate the forces which in turn are used to find the resultant force f_r.
Substituting A_w, J_w, C_v, C_h, V, and T into the following equations:

$f_s = \frac{V}{A_w}$

$f_{th} = \frac{T \cdot C_v}{J_w}$

$f_{tv} = \frac{T \cdot C_h}{J_w}$

$f_s = 3391.25N/mm$
$f_{th} = 147.22N/mm$
$f_{tv} = 426.41N/mm$

Following this the resultant force f_r can be determined, by substituting in the calculated forces

$f_r = \sqrt{f_{th}^2 + (f_{tv} + f_s)^2}$

$f_r = 3820N/(mm)$

Now that the resultant force f_r was found the thickness of the weld can be determined, by using the resultant force f_r and the allowable shear stress determined from the book [26] $t = \frac{f_r}{\sigma}$

$t = 16.75(mm)$

Conclusion of Weld Calculations

As a result of the calculations the thickness of the weld was determined $t = 16.75(mm)$. This is a large enough weld for the crane as from the external force and twisting moment derived V and T multiplied by safety factor of 3 gives a large enough safety zone for the weld and thus the crane will be able to operate correctly.

8 Mohr's Circle

When designing a highly loaded part, it can be interesting to apply Mohr's circle to determine the maximum stresses in the material. These determined stresses can be used to check whether or not the designed part fulfills the stated requirements
In this section, Mohr's circle is applied to a point P in the highly loaded hinge shaft from subsection 7.2.

8.1 Selecting worst case scenario

The worst case scenario in the shaft which gives the highest stresses is the case were the middle beam is completely shifted against the inside of the socket while the boom is in lifted position. This gives a space of 20 mm between the middle beam and the other inside of the socket and the highest possible shear forces.

8.2 Determining Forces and Location of P

As can be seen in the FBD in figure 43, only external vertical forces occur. The horizontal wind forces on the boom do not exert a significant axial force on the shaft, because these forces are very small relative to the vertical forces created by the weight of the boom and because these wind forces are compensated by the socket and the middle beam themselves and are thus not transferred to the shaft.
From the FBD in figure 43, it can be concluded that a point at cross-section A is interesting to apply Mohr's circle too as this gives the longest arm for an internal moment and the highest shear force.

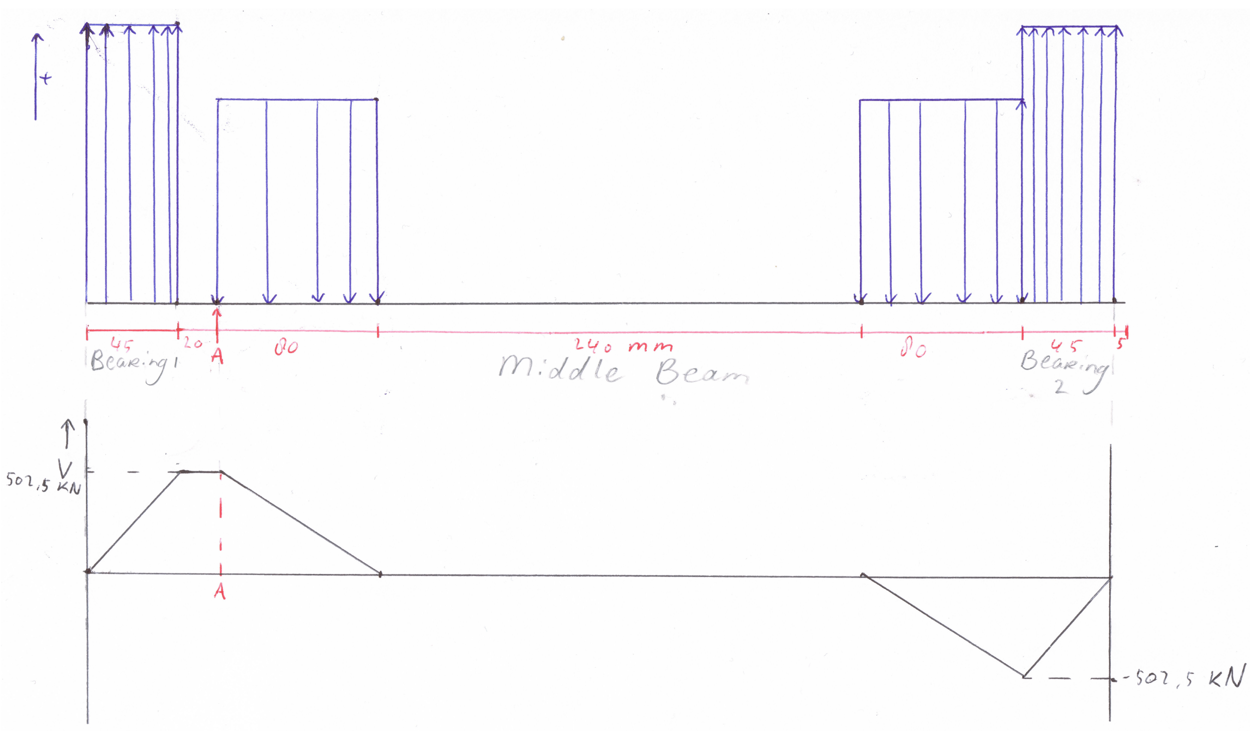

Figure 43: FBD of the hinge shaft with shear diagram

As can be seen, the shear force in A equals 502.5 KN. The internal moment can be calculated with the method of sections, assuming that the arm is 20mm as bearing 1 clamps the beam and that the distributed load can be replaced by a point force of 502.5 KN. This gives an internal moment of 20.10 kNm in A. When a cut is made at A, the stress states at A will be as in figure 44.

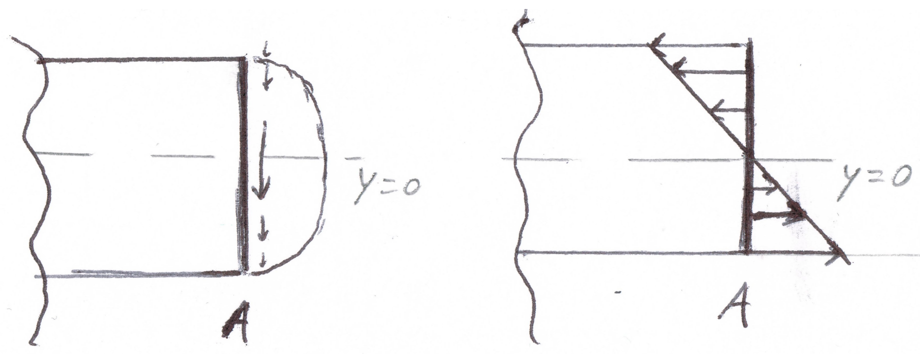

Figure 44: Stress states in A. Left due to shear. Right due to internal moment.

An interesting point on the cross-section of A to take for Mohr's cycle is a point with bending stress as well as shear stress. As the shear stress is maximal at y = 0 mm and the bending moment M_z is maximal at y = 85 mm or y = -85 mm, it is chosen to take point P for the Mohr's circle halfway on y = 42.5 mm.
The magnitudes of these stresses were calculated with the following formulas:

$$\sigma_{bending} = -\frac{y * M_z(x)}{I} \tag{4}$$

$$\tau_{xy} = -\frac{S_y(x) * Q_z(y)}{I * t(y)} \tag{5}$$

With the moment of inertia of a circle:

$$I = \frac{1}{4}\pi r^4 = \frac{1}{4}\pi(85 * 10^{-3})^4 = 4.0998 * 10^{-5} \quad m^4 \tag{6}$$

With the first moment of area of a circle of:

$$\int_{-\sqrt{(85*10^{-3})^2+y^2}}^{\sqrt{(85*10^{-3})^2+y^2}} \int_{y}^{85*10^{-3}} y \quad dydx = (0.007225 - y^2)\sqrt{(85 * 10^{-3})^2 - y^2} \quad m^3 \tag{7}$$

With $S_y(x) = 502.5$ kN and $M_z(x) = 20.10$ kNm results in $\sigma_{bending} = \sigma_x = -20.836$ MPa and $\tau_{xy} = -33.208$ MPa.
Now Mohr's circle can be constructed with the following formulas:

$$Radius = \sqrt{(\frac{\sigma_x - \sigma_y}{2})^2 + \tau_{xy}^2} = \sqrt{(\frac{-20.836 * 10^6 - 0}{2})^2 + (-33.208 * 10^6)^2} = 34.804 * 10^6 \tag{8}$$

$$\sigma_{average} = \frac{\sigma_x + \sigma_y}{2} = \frac{-20.836 * 10^6 + 0}{2} = -10.418 \quad MPa \tag{9}$$

This results in the following circle:

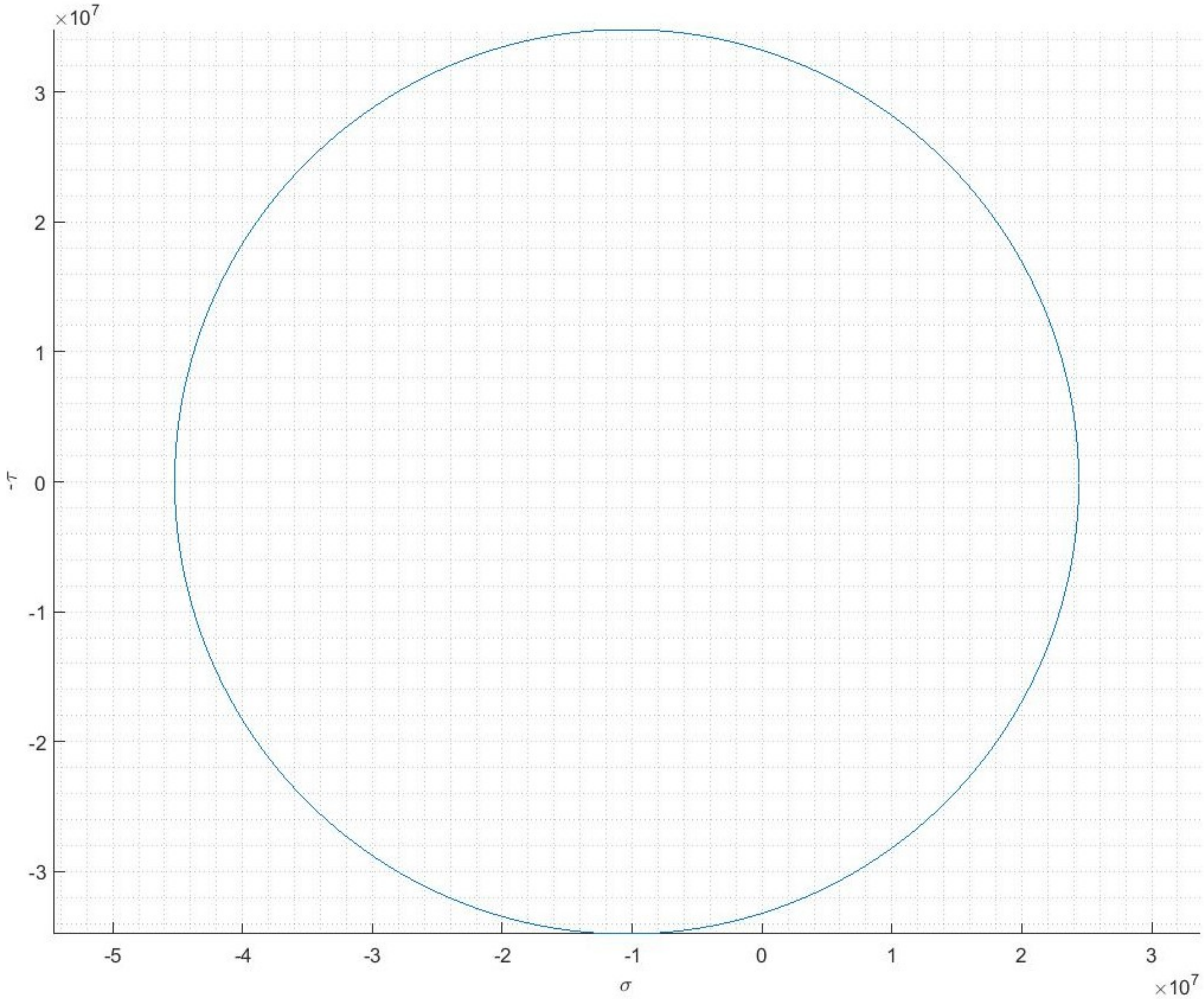

Figure 45: Mohr's Circle of Hinge Shaft Boom

8.3 Conclusion of Mohr's Circle analysis

Out of Mohr's circle, it can be concluded that $\sigma_{max} = \sigma_{average} - R = -10.418 * 10^6 - 34.804 * 10^6 = -45.222 * 10^6 \quad Pa = -45.222$ MPa and that $\tau_{max} = R = 34.804$ MPa.
This is of way below the shafts material (AISI 9255 oil quenched & tempered at 205 °C) yield stress of 1890 MPa and maximum shear stress of 538.65 MPa.[24]
It can be argued that the shaft diameter can be lowered based on this conclusion. However, this would not be a wise decision regarding safety factors.

9 Conclusion

After the design procedure, the following can be concluded:

- A cable based system was chosen for the lifting mechanism with a disc brake system on the drum as safety feature. For hoisting a Machinery On Trolley was chosen. For the trolley an underhanging trolley system was chosen.
- Different materials were assigned to the main components of the boom, hoist and trolley.
- A truss design and crane frame with different square beams was designed in 3D according to a Finite Element Method analysis. This resulted in a total crane construction weight of 2650.6 tons of AISI 9255 oil quenched steel.
- Elaborations were made on the machine elements in the hoist system. A drive system was designed with a motor. The chosen motor is: Simotics SD self-ventilated or forced-air cooled motors - cast-iron series 1LE5604 Performance Line: 400 kw, 4-pole, 1492 rpm at 50Hz.
- Linear guideways were designed according to the Gantrex rail system. Also, the highly loaded weld in the crane frame, a bolt in the trolley system and the hinge connection in the boom were designed and calculations were made.
- Mohr's circle was applied to the highly loaded shaft in the boom hinge. Out of this it became clear that the shaft is strong enough to withstand the loads.

At the end, a working STS-crane is designed that fulfills all given requirements.

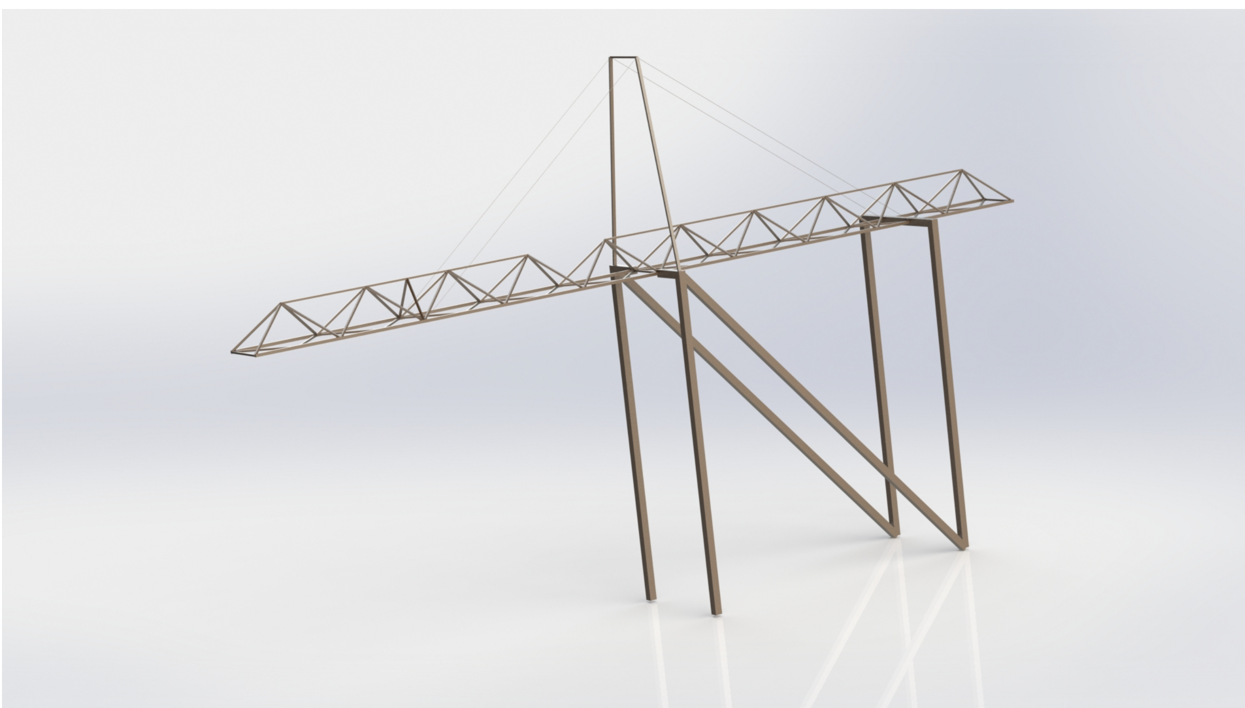

Figure 46: Final Crane construction

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10 Appendix

10.1 Boom concepts

The boom of the crane is the arm on which the trolley can move forwards and backwards. It should be able to be lifted up and down, so it must have a hinge in it. Furthermore it should fulfill the requirements stated in the problem definition. In order to design the different mechanisms of the boom, concepts were made of the lifting mechanism and the locking/braking mechanism. For the boom itself, a truss-based design is chosen as this is the strongest way and most weight-saving way to build a long arm on a crane.

Lifting Mechanism The function of the lifting mechanism is to lift the boom up when it does not have to pick up containers and lower the boom when it has to do its lifting task. Before lifting, the trolley is moved behind the hinge point to the back of the crane, so it is not located on the boom while the boom is lifted or lowered. The lifting mechanism has to fulfill the requirements stated in the problem definition.

Cable mechanism
This widely used system consists of strong cables mounted to the boom, guided over a extension on the top of the crane and then mounted on a drum which is connected to a powered shaft. With this the boom can be lifted by rolling up the drum as can be seen in the figure below.

Advantages:

- It is a simple to make and cheap system so there are few parameters that can cause failure.
- The drive train is placed on the back of the cranes, so all power train components are centralized.

Disadvantages:

- Cables must be checked or replaced regularly in order to prevent failure of the cables.
- The cable mechanism is easy to produce.

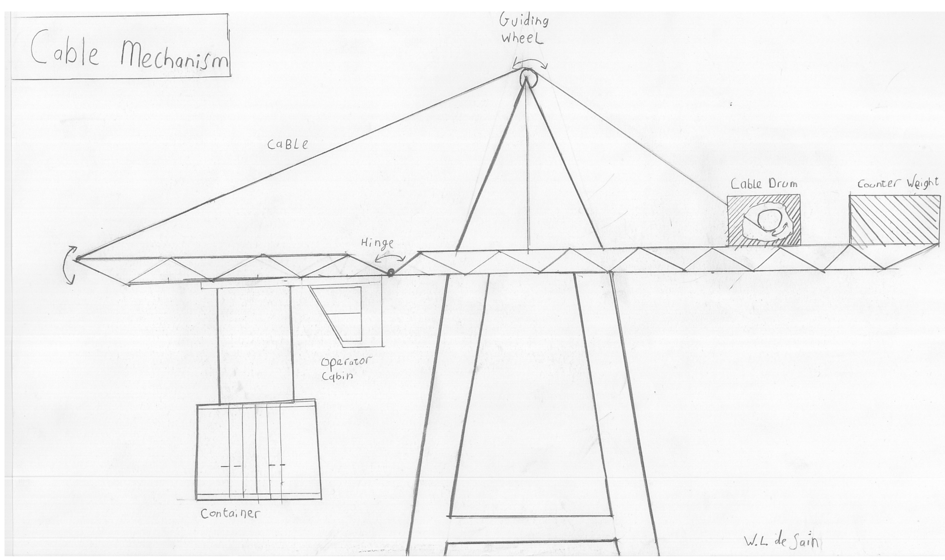


Figure 47: Cable concept

Gear rack mechanism
This system consists of a gear rack or set of gear racks mounted between the boom and the extension on top of the crane. A gear or set of gears is placed in the extension on the crane which can lift the boom by bringing in the gear rack by turning the gear. A disadvantage of such a mechanism is that the powertrain

has to be placed in the extension on the crane in order to drive the gear which drives the gear rack. Also, producing such long gear racks is very expensive and very difficult to produce. For this principle, two options are possible:

Using a curved gear rack:
By using a curved gear rack the gear rack can be mounted fixed at the boom. When lifting, the curve in the gear rack is needed to follow the track of the point onto which the gear rack is fixed to the boom. The disadvantages of using this type of gear rack is that a moment can be created around the point on which the gear rack is fixed to the boom, this can be prevented by mounting the gear rack with a hinge to the boom. Also the curved gear rack might interfere with the back end of the crane if the boom is lifted up, so that has to be taken into account when designing the crane in more detail. The advantage of using this type of gear rack is that the actual pulling force is normal to the boom, which results in lower power needed on the powertrain.

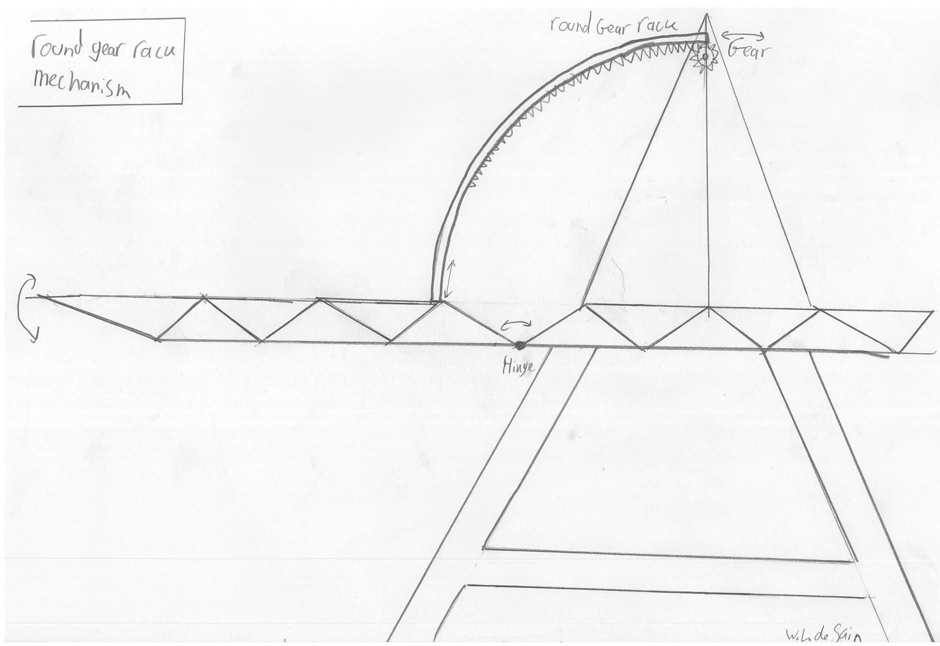


Figure 48: Curved gear rack concept

Using a straight gear rack:
By using a straight gear rack the gear rack is mounted with a hinge to the boom. The advantage of a straight gear rack is that the creation of a moment around on the point to which the gear rack is attached to the boom is prevented, so no problems with that can occur. Also this system can not interfere with the back of the crane. The disadvantage of a straight gear rack is that the actual pulling force in the connection of the gear rack to the boom is not normal, but in an angle. This means that not all force created by the power train can be used to pull up the boom. This means that a more powerful power train is needed to power a straight gear rack mechanism than a curved gear rack mechanism. However, the force exerted by the power train becomes more and more normal to the boom during lifting, so this disadvantage will become less and less important while lifting.

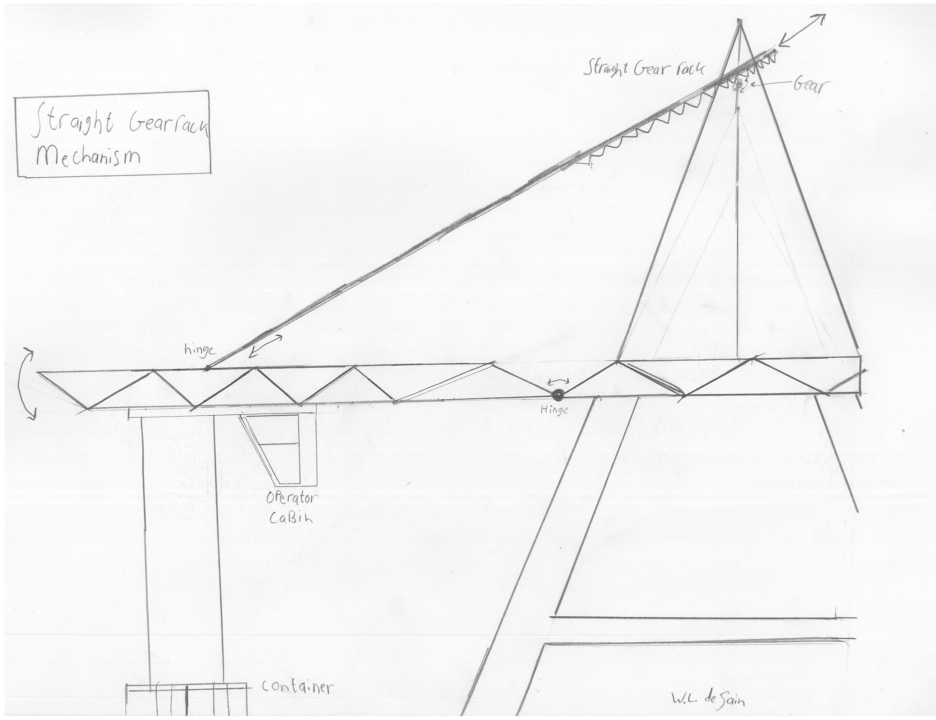


Figure 49: Straight gear rack concept

Pneumatic/hydraulic mechanism

With a pneumatic or hydraulic lifting system, a piston or a set of pistons is mounted to the back of the crane and to a extension on top of the crane before the hinge as can be seen in the figure below. By compressing the piston, the boom will be lifted as a moment is created around the hinge. The disadvantage of this system is that the boom can not be lifted completely as the pistons can not be compressed completely, but this depends on the detailed design of the crane. Also the piston will be heavy and large pistons with high power delivery will be needed to lift the boom as a large weight with a large moment will have to be lifted. Furthermore, these kind of systems are prone to leaks. This might especially be a problem in this case were a very high power output will have to be delivered.

The advantage of this system is that no mechanical power train is needed, but only a compressor and a set of pipes and valves. So less mechanical wear will occur. Also the boom can be placed in every possible position.

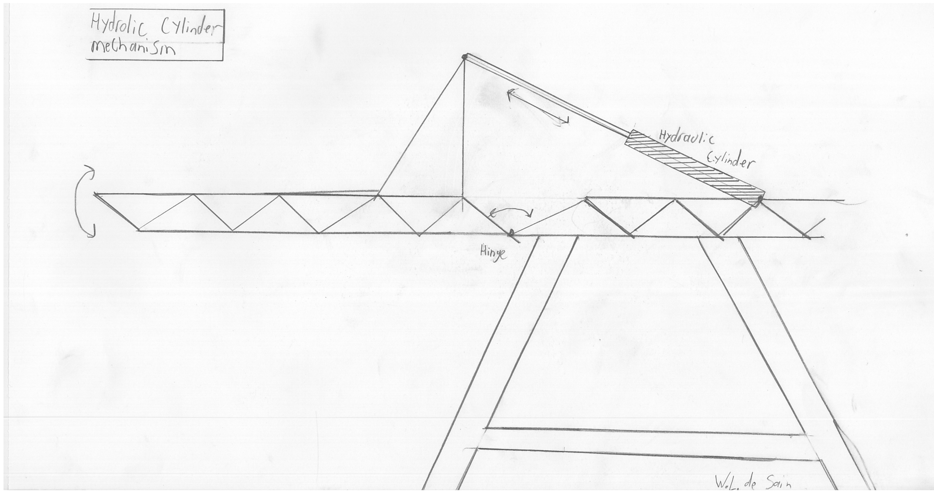


Figure 50: Hydraulic concept

Concept choice

For evaluating the different concepts and in order to make a well-reasoned concept choice, the following weighing table was made:

Table 7: Weighing table for concepts of lifting mechanism

	Weighing factor	Cable concept	Straight gear rack concept	Curved gear rack concept	Hydraulic cylinder concept
Amount of parts	3	4	4	4	5
Safety	5	3	4	4	2
Manufacturing cost	4	5	2	1	3
Weight	3	4	2	2	4
Maintenance cost	2	2	4	4	3
total		62	54	50	55

To be able to asses the different concepts and to make a choice a concept choice a weighing table was made. Five different weighing factors were devised. These weighing factors are: amount of parts, safety, manufacturing cost, weight and maintenance cost. These weighing factors all have a weighing value between 1 and 5 as can be seen in the table. As engineering safety is the number one priority, safety got a multiplier of 5 out of 5. The manufacturing cost weighing factor consists of different components. This includes the difficulty of manufacturing and material types . These components add up to a single weighing factor. A multiplier of 4 out of 5 was chosen for this weighing factor. If a product consists of more parts, it can become more complex. This can reduce the cranes durability. The amount of components is less important than safety and the manufacturing costs so it received a multiplier value of 3 out of 5. The weight of the total construction is also important in the decision process. If the weight of the product is reduced, the production process will be cheaper as well. Weight got a multiplier of 3 out of 5 because of this. The last weighing factor is maintenance costs of the product. During the life of the product it will need maintenance. The amount of maintenance is dependent of the materials and the amount of components. In deciding the concepts the maintenance is less important than the other factors, so it received a multiplier of 2 out of 5.

The cable concept

The cable concept does not have a lot of parts. However, it has more than other concepts in the table therefore it gets 4 out of 5 points. For safety it does not score high, If for example the cables snap the boom will fall. However, this can be partially prevented by using safety features. Therefore the concept gets a 3 out of 5 for safety. It is relatively cheap to manufacture the cable system therefore it gets 5 out of 5 points. The mechanism is also light as it only has a few parts. Therefore it gets 4 out of 5 points. The cables will require more checks then the other concepts this results is more downtime and more maintenance cost. Therefore it only got 2 points for maintenance.

Gear rack concepts

For the gear rack systems most of the points are the same. The gear rack system does not have a lot of parts. However, it has more than other concepts in the table therefore both concepts gets 4 out of 5 points. The gear racks are very strong so it is less likely that they will snap. However, it is relatively difficult to lock the gear rack. Therefore they both score 4 out of 5 points. Gear racks are difficult to manufacture and therefore it will cost more to manufacture then the other concepts. The curved gear rack is even more difficult to manufacture and therefore it gets a lower score then the straight gear rack. The straight gear rack gets 2 points and the curved one only 1. Due to the fact that a gear rack is heavier than a cable due to its size, both concepts score lower on weight. The gear racks are less likely to fail due to fracture because they do not need to be flexible. Therefore less safety checks are needed and so the maintenance costs will be lower than with the cables. So both concepts get 4 out of 5 points.

Hydraulic cylinder concept

The hydraulic cylinder concept uses only a hand full of parts, mainly the cylinder, compressor, and a small boom. It used by far the least amount of parts of all the concepts so it gets 5 out of 5 points of the amount of parts. It is difficult to lock a hydraulic cylinder, therefore if it fails the boom will fall. If it fails it will also spray hydraulic oil at high pressure form the crane. This can damage people and equipment. This oil will also contaminate the surrounding land and water. Therefore it gets 2 out of 5 points for safety .To be able to lift the boom a very large cylinder needs to be made. This is rather difficult to manufacture and therefore the cost of the manufacturing will rise. This results is a score of 3 out of 5 for the manufacturing cost. Due to the small amount of parts in the concept it is not very heavy. Therefore it gets 4 out of 5 points for weight. Hydraulic cylinders are prone to leakage and therefore need to be checked periodically. Therefore it gets 3 out of 5 points for manufacturing cost.
Out of the table, the cable mechanism comes out to be the best choice. This is also the most logical choice as the cable system is the most simple and cheap concept with a few amount of parts and very low weight. Therefore, the cable mechanism is chosen as the final concepts.

10.1.1 Locking/braking mechanism

The function of the locking or braking mechanism is to lock the boom when it is in its most upward or its most downward position and in between these two position in order to relieve the forces on the engine and the power train. Also, when the lifting mechanism of the boom might fail, the locking/braking mechanism must be able to slow down and stop the boom from falling down and destroying the structure of the crane.

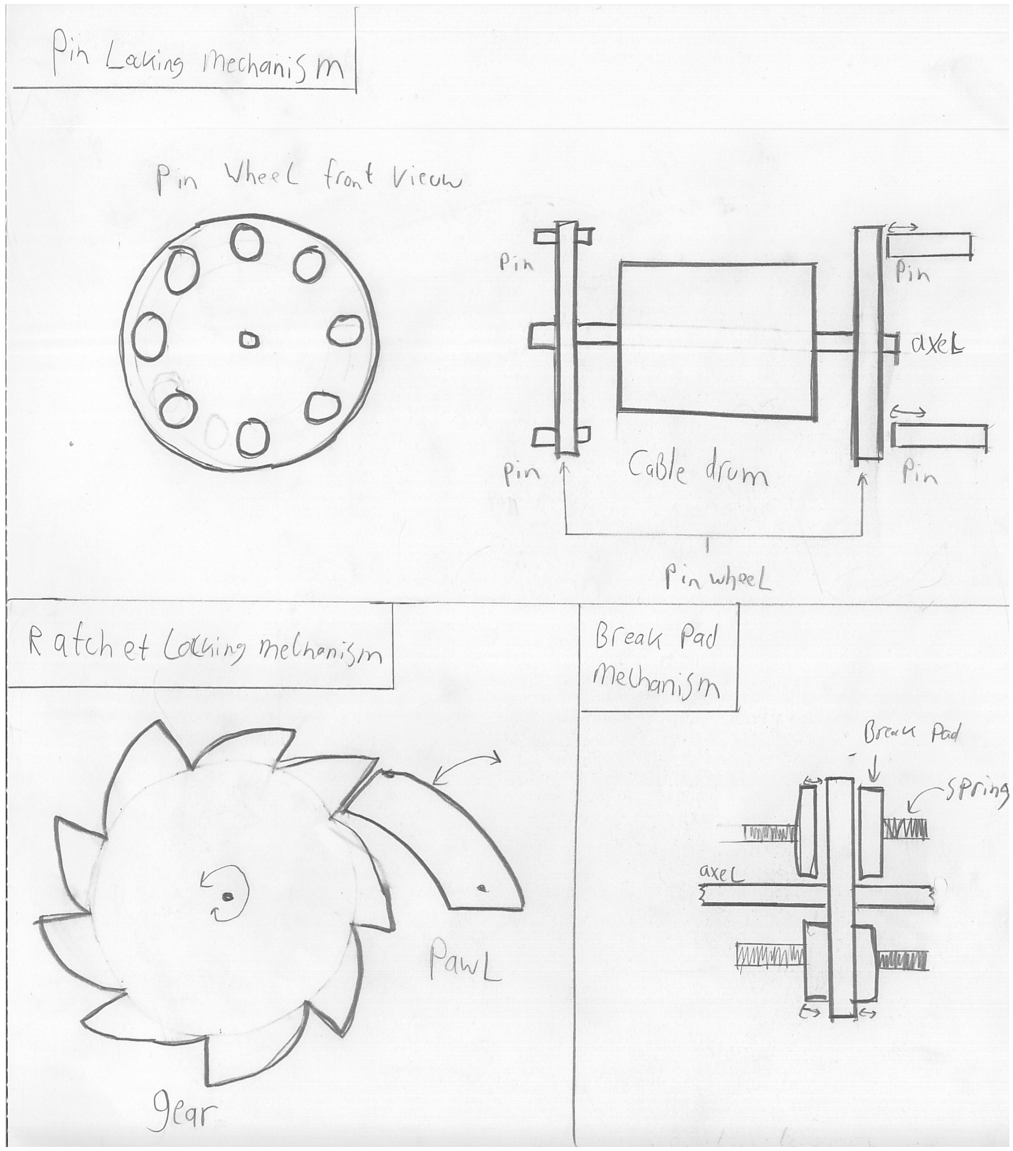


Figure 51: Locking mechanism concepts

Pin mechanism

With a pin mechanism a wheel is connected on the drive shaft with holes as can be seen in figure 51. When the crane is lifted up or down completely or it is wanted that the crane is fixed somewhere halfway the lifting, pins can be put through the holes into holes in the frame of the crane. This has the result that the force exerted by the lifting mechanism on the power train is relieved and transferred to the frame so that a fixed whole is created. The pins can be connected to for example electromotors in order to push them in or take them out the holes.

The advantage of this system is that it is relatively simple and the boom can be locked at different lifting positions.

The disadvantage of this system is that if failure occurs and the boom might fall down, this system is not able to stop the boom as the wheel with holes will spin to fast for the pins to go in. This means that with using this system, an extra safety mechanism must be used.

Disc brake mechanism

The principle of this system is similar to a regular car disc brake or brakes used in trains. For this system a wheel has to be mounted to the driving shaft of the power train in order to be make disc brakes work.

The advantage of this system is that multiple disc brake units can be placed on one wheel (for example 4) to be able to create a large braking force. Furthermore, a safety feature can be implemented in such a system where the brakes are fully braking when for example a power failure occurs in the crane. This can be done by for example springs behind the brake blocks that are pushed in when the brake is off and are released when failure occurs. Also the crane can be locked in every position wanted.
The disadvantage of this system is that the brake blocks are prone to wear and must be checked/replaced regularly to prevent failure. Also when a lubricating substance (for example oil) comes between the wheel and the brake pads, the braking force is drastically reduced. This has to be taken into account when designing the crane in detail. Also, very strong brakes are needed to cancel out the moment on the drive shaft created by the weight of the boom in lowered position.

Ratchet mechanism
This mechanism consists of a gear connected to the drive shaft with a pall that that does not influence the movement of the gear but falls between the teeth of the gear when the gear turns the other way round. This system is widely used in ratchet wrenches. When it is desired to let the drive shaft turn the other way round, the pall has to be lifted up so it can not block the backwards movement of the gear.
An advantage is that this kind of system is relatively simple and cheap to make.
The big disadvantage is that a large force will be exerted on the pall as it has to hold back the weight of the whole boom. This results in a very high force on the gear attached on the drive shaft which can deform the gear, which is highly undesirable.

Safety beam
An extra safety feature can be build in the crane by placing two beams that are connected to each other by a hinge on the front of the boom and a extension on top of the crane as shown in the figure below. Thus an elbow like construction. When the crane is lifted, these set of beams are also folded. When the boom of the crane is fully lowered, these two beams function as a lock as the extension on the top of the crane can exert a pulling force on the boom, keeping the boom in place. This will relieve stresses from the drive train when the boom is in its lowered position.
This feature can not lock the boom in lifted position or half-lifted position. However, it can be added with for example the pin-lock mechanism to prevent the boom falling further than the fully lowered position when failure of the lifting system occurs.

Concept choice
For the locking/braking mechanism the pin-lock mechanism, the disc brake mechanism and the safety beam were chosen to be used in the final concept. All of these different mechanisms contribute in their own unique way to the safety of the crane. With combining these three mechanisms, all disadvantages of the different mechanisms are covered by the advantages of the other mechanisms.

- The safety beam makes sure the cables are relieved from tension when the boom is lowered down completely and stops the crane in its lowered down position when failure of the cables occur.
- The pin-lock mechanism makes sure that the boom can be locked in every position between the lowered and lifted position, so that tension on the cables is relieved and wear of the cables is reduced.
- The disc brake mechanism makes sure that when the cables fail and the boom falls down, the boom is stopped or slowed down. This prevents destruction of the whole construction. This mechanism can also be used to lock or slow down the boom for a short time when that is demanded.

With this, the safety of the design is optimized, which is number one priority in engineering.

10.2 Hoist Concepts

10.2.1 Hoisting

A hoist is a device used within a STS (ship to shore) crane to lift containers up by means of a drum. The drum or either multiple drums are powered using motors. Due to the turning of the drum, the wire rope, which is connected to the containers, wraps around the drum causing the container to rise. There are different ways in which the hoisting mechanism is integrated within the crane system. The hoist can either be fixed and powered on the trolley (machinery trolley) or from another location (rope towed trolley system), the different systems are described below.

10.2.2 Rope towed trolley system (RTT)

On an RTT crane, the machinery of the hoisting mechanism is placed inside a machinery house which is usually located somewhere at the backreach of an STS crane. Ropes will go from the trolley all the way to the machinery house and the boom tip.

Long ropes and a lot of sheaves Since the machinery of the hoist is not mounted on the trolley, the trolley will be lightweight. This reduces the load on the wheels of the trolley which then ensures less wear. On the other hand, a lot of rope is needed in order for the ropes to reach the machinery house and the boom tip. There will be more sagging and stretching when the rope lengths are high. This reduces the accuracy and responsiveness of the hoisting mechanism. This is especially a problem with big STS cranes since rope lengths will be even longer.

Concepts Two concepts for a hoisting mechanism of a RTT crane are shown below. The fixed ends in the left figure represent the boom tip, the drums are located in the machinery house and the sheaves above the container are mounted on the trolley. The other figure basically shows the same mechanism but now the drums serve as fixed points.

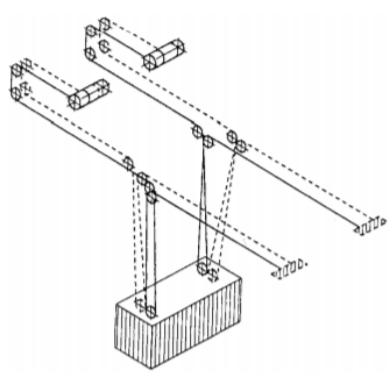


Figure 52: First hoisting concept RTT crane

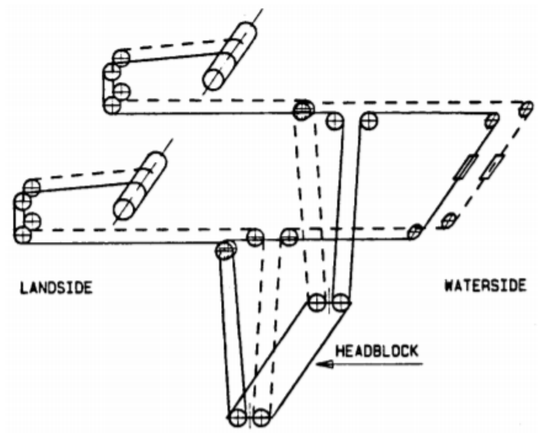


Figure 53: Second hoisting concept RTT crane

10.2.3 Machinery on trolley crane (MOT)

The MOT crane has the complete hoisting and trolley mechanism mounted on the trolley. This results in some disadvantages and advantages with respect to the RTT system which will be discussed in the following paragraphs.

Environmental issues related to the different crane systems One of the environment related problems that is occurring comes from the lubricated wire ropes. Within a RTT system, these lubricated wires run over the entire length of the crane causing dripping of lubricant (grease,oil) either on the landside or worse, in the water. Eliminating the lubricant results in larger amounts of wear which eventually results in having to replace the wire ropes more often. This compared to a MOT crane, where due to the absence of the ropes running over the length of the crane, the total amount of lubricated wire rope is reduced. Consequently, less lubricant is being dropped resulting in fewer environmental problems.

Comparison in controlling of the load Within a MOT the wire ropes running to the trolley across the length of the crane are eliminated. Therefore, both stretching and sagging of the wire ropes are creating less of a problem due to the reduction in the amount of wire rope. Due to less stretching and sagging of the ropes the MOT has improved load control compared to the RTT system.

Increased wheel loads Accounting for both the increased weight of a MOT system trolley and the larger weight of the crane itself in order to reach improved strength, MOT systems experience 15 percent higher wheel loads than RTT systems. Due to these increased wheel loads, MOT cranes undergo more wear resulting in increased replacement costs, which is of course not favorable.

Skidding Due to the increased weight of the trolley, it has been assumed that the MOT is greatly subjected to skidding during heavy rain or icy conditions. After all, these problems turned out not to be of great significance.

Ease of maintenance Due to the close packing of the machinery of both the trolley and the hoist mechanism on the trolley, due to limited space availability, maintenance of the machinery can be rather difficult compared to the less strict packing of the machinery on a RTT crane system.

Festoon system Due to the machinery located on the trolley in a MOT system, the power supply towards the trolley is significantly increased, resulting in a heavier festoon system. Consequently, the shock loads can cause excessive wear. Sometimes extra measures have to be taken in order to support the festoon system, resulting in higher (maintenance) costs and a more complex system.

Concept As can be seen from figure 3, the drums are now located above the spreader. The hoisting mechanism is mounted on the trolley which results in a heavier trolley but less rope use. The load cells act as fixed points. There is a reason for the use of sheaves on the spreader: less force is exerted on the drums and the ropes since the force get divided over the ropes leaving the sheaves. The drawback of this is that the drum has to cover twice the distance for the same amount of vertical movement of the spreader compared to a system without sheaves.

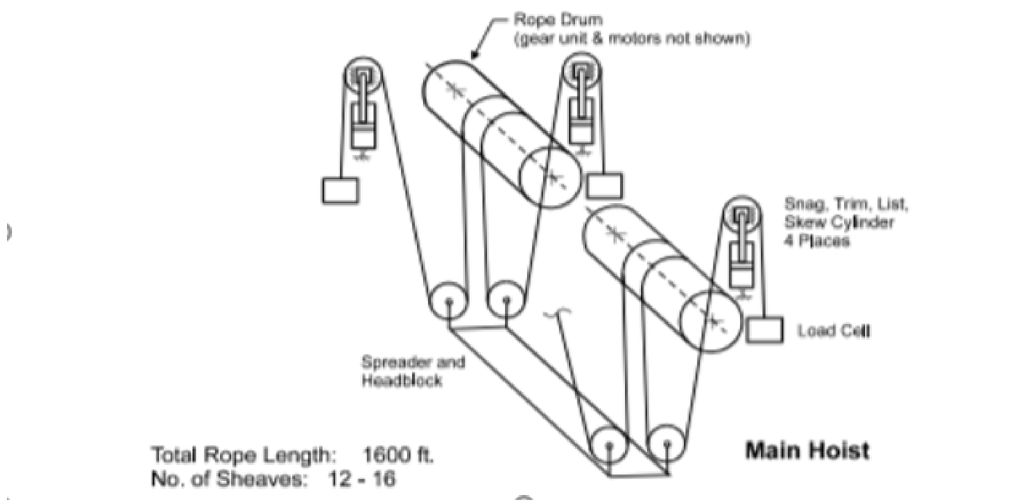


Figure 54: Hoisting concept MOT crane

10.2.4 Hoist concept choice

By making use of a weighing table, a choice can be made between the different concepts. The table shows crucial properties which have to be considered in order to make a suitable choice. The weighing factors show the relative importance of each property. Each property of both concepts is given a score between zero and five, zero being the lowest and five being the highest. These scores will be multiplied with the weighing factors corresponding to the properties. All the scores are then added to form the total score. The concept with the highest score is assumed to be the best according to this rating system.

Table 8: Concept choice weighing table hoist

	Weighing factor	**RTT**	**MOT**
Amount of parts	4	2	4
Controlling of the load	4	3	5
Safety	5	4	4
Environmental impact	2	2	4
Weight	4	4	2
Maintenance	3	3	2
Total		**72**	**78**

Evaluation of the table In the next couple paragraphs it will be explained why these properties and weighing factors were chosen. The scores given to the two concepts will be addressed as well.

Amount of parts The amount of parts are important for many reasons: it for example influences the amount of maintenance needed, the complexity of the system, the accuracy and the responsiveness of the system. Since the amount of parts influence the functioning of the hoist a lot, the weighing factor is somewhat higher: 4. For the hoisting system, we are especially focusing on the sheaves and ropes. The sheaves and ropes are the critical parts since they need to be inspected regularly on excessive wear. The RTT system has way more sheaves and it uses a lot more rope than the MOT sytem. Consequently, the total amount of wear will be bigger for the RTT system which means that the RTT system needs more maintenance. The RTT system also uses more rope which reduces accuracy and responsiveness due to sagging and stretching of the ropes. In conclusion, more parts negatively effects the crane which is why the RTT system gets a lower score.

Controlling of the load The controlling of the load has to do with the accuracy of the mechanism as well as the ease of operating the system. Due to the long cables used in a RTT system, taking into account the sagging and stretching of these cables, the RTT might be more inaccurate as well as harder to operate. Therefore, the MOT scores higher on this property. From a customers perspective, the ease of operating is considered very important. Besides that, an easy to operate system could possibly save time, which is why the weighing factor of this property is set to 4.

Safety Safety is of course the most important property: something that is not safe, will not be used. The weighing factor is therefore 5. There are no significant differences in safety between the concepts so the concepts are considered to be equally safe. For this reason, they get the same score.

Environmental impact The only, yet subtle, contribution to any kind of harmful effect to the environment comes from the lubricated cables. These cables tend to drip their lubricant on either land side or in the water. MOT therefore scores better since cable length is reduced significantly. Since this effect just plays a minor role the weighing factor is only 2.

Weight Due to the replaced machinery on the MOT system, the weight of the trolley is considerably greater than the RTT trolley. Since in a MOT system the machinery is moved from the back-reach of the crane towards the front side of the crane, a considerable amount of weight is removed from the back end. This might result in having to increase the counterweight on the crane. The increased weight also results in higher wheel loads on the trolley. This increases wear which then increases the costs. Besides that, a heavier trolley requires both a stiffer and stronger boom structure. Therefore MOT scores worse on this property compared to a RTT system. As discussed earlier, the weight of the trolley influences the crane a lot and therefore it gets the weighing factor 4.

Maintenance Lastly, maintenance should be considered as well. More maintenance means more costs and less operation time, which is not favorable. It is of course less important than for example safety but it still plays an important role which is why the weighing factor is 3. Due to close packing of machinery on the trolley in a MOT system, maintenance is regarded hard with respect to the RTT system because the parts are hard to reach. On the other hand, the RTT system needs more maintenance since the amount of parts is greater, possibly resulting in more parts to be replaced. Maintenance of a RTT system is still considered to be harder. Therefore, the score of the RTT with respect to this property is greater than the score for the MOT.

10.2.5 Conclusion

As can be seen from the table, the MOT system has a higher total score. Out of this it can be concluded that the MOT system is the most suitable concept in this case. This seems logical due to the fact that literature nowadays favors the MOT system over the RTT system since RTT systems are not able to keep up with the increasing development in the shipping industry.

10.3 Trolley Concepts

The trolley is the component of the crane that moves the hoist from side to side, over a rail, along a (double) girder. In essence, there exist two types of trolleys, manual and gear operated trolleys. Manual operated trolleys are suitable for components that weigh up to around 2000 kilograms. These are mainly in use to help operators in placing/lifting heavy components and their precision is not that high. Gear operated trolleys can be made to lift and move components which weigh 120 tonnes. These trolleys are also much higher degree of accuracy when positioning their lift. Gear operated trolleys can e.g. be found in STS-cranes. Therefore, the trolley concepts designed are gear operated trolleys. Manual operated trolleys are no longer mentioned. Gear operated trolleys are powered by an electrical engine, our concepts are designed on that basis.

10.3.1 Main Components of a Trolley

Hoist Mechanism The hoist mechanism is used for lifting or lowering a load. It can either be found directly on the trolley itself, or at the back of the STS-crane. In case the hoist mechanism is installed on the trolley itself, the trolley should be designed differently than if that is not the case.

Figure 55: Hoist Mechanism

Girder The girder is the component of the crane over which the trolley moves along the boom. Girders are part of the boom, but have to be considered when designing the trolley as well. There are two types of girder-trolley systems, single or double girder. Both girder-trolley systems have advantages and disadvantages, which will be outlined below.

Single Girder On a single girder system, the trolley moves along one girder beam. The trolley hangs directly under the girder. The advantages of single girder cranes is that they are cheaper to make than double girder systems, because one beam requires less material and installation effort than two beams. The disadvantage of single girder cranes is that their maximum load capacity tends to end at twenty tonnes, which makes them unsuited for large STS-cranes.

Figure 56: Single Girder

Double Girder Double girder-trolley systems consists of two beams, along which the trolley can run. The trolley is situated in the middle of the girders, either above or below the beams. Double girder systems are more expensive to make, because of the second beam and since there is a second beam, the installation effort will be a bit higher, because the girders have to be perfectly adjusted to each other. If they are not, the trolley cannot move along the rails on the girders. However, the great advantage of double girder systems is that these can support up to 120 tonnes of load, over a much longer span than single girder cranes, which makes them very well suited for use in STS-cranes. Double girder systems can also be used to install maintenance platforms, combines magnetic reels and lights.

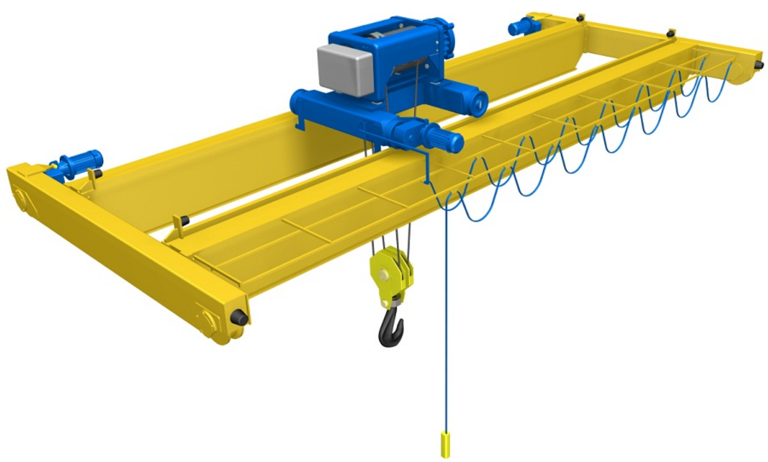


Figure 57: Double Girder

Wheels and rails to move the trolley Since STS-cranes tend to be double girder-trolley systems, only the rails for those cranes will be discussed. In double girder STS-cranes, the railway tends to be on the inside or on top of the girder. STS-crane rails and wheel tend to be the widely used asce-rail and wheels.

Bumpers Bumpers are located on both ends of the girders. They make sure that the trolley cannot move any further and shall therefore not fall of the rails.

10.3.2 Three Different Concepts

10.3.3 Concept 1

Rail system below the crane with dual movement x and y direction.

Figure 58: Double Girder x and y direction

In this trolley concept, the trolley can move back and forwarth across the boom as well as moving sidewards in between the two girders. In order to allow for side movements, either the trolley should be small enough or the girders should be far enough apart of each other to allow for this kind of movement. This is problematic, as trolleys on STS-cranes have the width from girder to girder to be able to hold the load. Making the girders stand further apart and keeping the same large trolley is also problematic, because that would make the crane even larger, for a minimal gain in accuracy. The advantage of a dual movement trolley is that containers can be placed with more accuracy, because of micro adjustments with the side movement. Another advantage of a dual movement trolley system is that it is no longer necessary to move the whole crane for, say 0.5 meters, to place the container up the ship or to pick it up. The disadvantage of a dual movement system is that an additional girder system is needed. Two girders are needed to allow movement along the boom, while two other girders are needed for the displacement in perpendicular direction. The actual trolley shall be fixed to the last mentioned girders. This is to make sure that nothing gets tangled or that components accidentally collide with each other. Another disadvantage will likely be the cost of this system for a marginal gain in accuracy. Apart from 2 additional girders, a second rail system is needed, more electric engines are required and it is a lot harder to make sure that the trolley does not fall of the crane, when there are two more girders that also cannot fall off in any circumstance.
Cabling in the perpendicular movement will be the third issue to solve. If the cables come from the direction of the crane itself, these might get tangled with the second girder system. If the cables come from the perpendicular direction, as can be seen in the picture below, there is no space on a STS-crane for the cables in that direction, so there is a difficult and probably expensive system required to solve this issue.

Concept 2 Rail system lying on top of the crane with single movement in y direction.

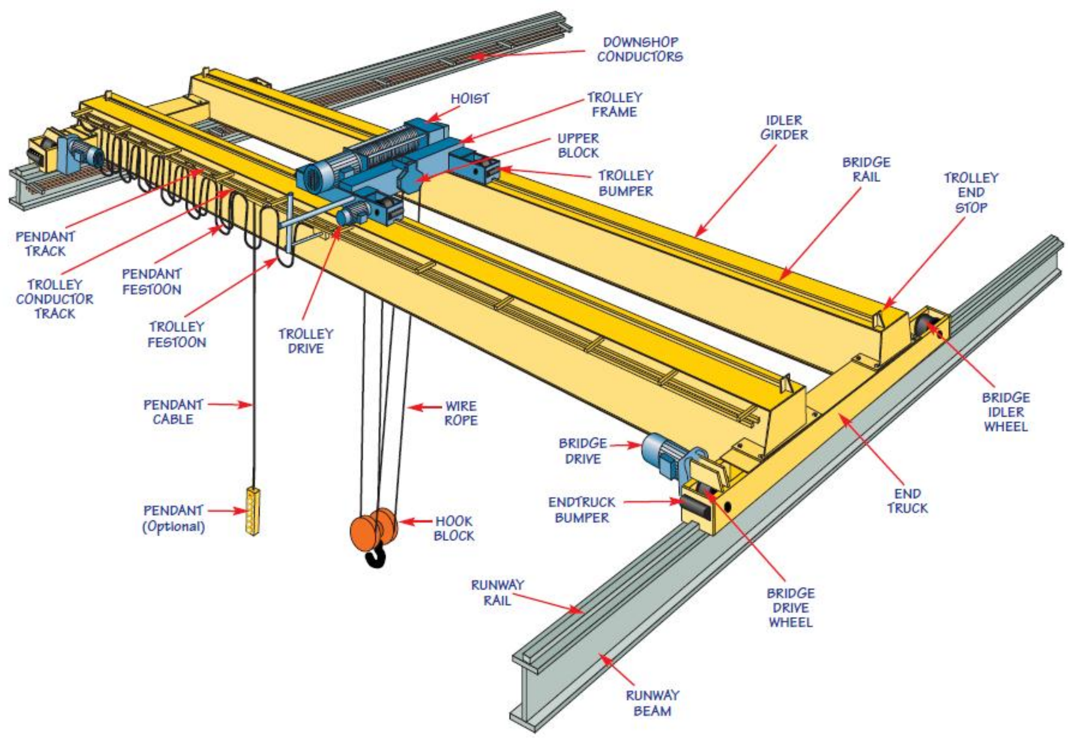


Figure 59: Overhead Trolley x direction

This trolley system only allows for movement in the direction along the boom. The trolley itself is situated above the girders and runs along rails. The advantage of such a system is that all cables can simply be installed next to a girder, without having to take components in the cross directional area into account. Another advantage is that this system is much cheaper than the first concept, as less material is needed and because this system is much simpler.

A tiny disadvantage is that for a very small correction movement, the whole crane needs to be moved, but as discussed above, that sacrifice is justified.

There are three significant aspects to take into account with a trolley system in a STS-crane like this. The first one being the truss structure of the crane. In case this structure is very low above the girders and in triangle shape, the trolley might not be able to move along the boom. It needs to be taken into account how the truss structure is designed. The second aspect has to do with the housing of the machinery and again the trusses of the boom. In most STS-cranes, the housing of the machinery is at the back of the STS-crane and below the girders. If the trolley is above the girders, it might be needed that the machinery housing and the cabling is above the girders as well. This might interfere with the trusses of the crane. The last aspect has to do with the spreader. If this component is pulled up too far, it might hit the girders, causing significant damage to the girder, trolley system and spreader itself.

Concept 3 Trolley system in between the two girders

Figure 60: Under hanging Trolley

This trolley concept also solely allows movement along the boom longitudinal direction. The difference with the previous concept is that the trolley itself does not run on top of the girders, but hangs in between and a bit under the girders. This has the advantage that the trolley cannot interfere with the crane's trusses. Furthermore, the cabling can simply be installed along one of the girders and cannot get tangled with other components. Another advantage is that, since the trolley hangs lower, the spreader cannot collide with the girders, which might be a problem with concept 2.

10.3.4 Comparison Between Crane Trolley Concepts:

- Top running cranes tend to carry greater loads than under running cranes
- Girder system can carry large loads up to 120 tonnes, while single girder cranes carry between 10-20 tonnes. Therefore, a double girder system will be used
- Maintenance of concepts 2 and 3 will be significantly easier than concept one, because the trolley system is much simpler.
- Therefore, concepts 2 and 3 will be cheaper than concept 1.
- The 2-direction movement system of concept one allows that the crane can be stationary while picking up or dropping a container from / on the ship, which is safer.
- Concept 2 might give problems with the spreader when pulling it up, concept 3 does not have that problem at all.

10.3.5 Advantages of the Machinery on the trolley:

- All the wheels have individual motors on the trolley hence if a motor, brakes down the trolley can still move at reduced speed
- Each wheel has its own braking system, therefore the entire system does not fail if one of the brakes stops working
- All components are easily accessible for maintenance
- Extended trolley wheel lifetime, due to non-skewing trolley movement
- Accurate alignment of the wheels

10.3.6 Advantages of Trolley sitting in between the two girders:

- Advantages of Trolley sitting in between the two girders:
- Compressive forces instead of tensile
- Reduced obstruction of other components ie trusses
- Lifting capacity is greater than under hanging trolleys

10.3.7 Table with weighing factors

In this table the three concepts are analyzed on five aspects, where every aspect has a certain weighing factor. After analyses, the concepts will have a total point score, where the highest scoring concept probably works best.

Table 9: Weighing Factor

Components	Weighing Factor	Dual Movement Trolley	Single Movement Top running Trolley	Single Movement Under Hanging Trolley
Amount of parts	3	2	4	4
Manufacturing cost	3	2	3	3
Weight	1	3	4	4
Maintenance cost	4	2	4	4
Position on the boom	4	4	2	4
Total	0	39	49	57

The weighing factors are implemented to rank the different aspects according to their importance. It was chosen to give the aspects ''Amount of parts" and ''Manufacturing cost" three points. The reasoning behind this is that these aspects are important, but not of key importance. Less parts is desirable, but not critical. Lower manufacturing cost is desirable, but again not critical. It was chosen to give the aspect ''Weight" factor one. If the trolley weigh less, that is desirable, but in the whole system, the trolley does not weigh much. ''Maintenance cost" and ''Position on the boom" are ranked as the most important factors with weighing factor four. Maintenance will always happen during the crane's lifecycle, so the cost should be low. If the trolley is located on an unfortunate spot, that will severely affect the crane's ability to work and the crane's stability.
As can be seen from this table, the concept where the trolley has a single movement and underhangs the beams has the highest score. The second concept and first concept are in second and third place respectively. The results of this weighing table give a good overview of what concept probably works best. These results do not tell what concept will be chosen, but give a good indication, since these points come from a hand full of aspects.

10.3.8 Reasoning for dual Movement Trolley Weighing Factors

For the amount of parts the Dual movement Trolley received a 2, this is because it is a more complex structure, which included x and y lateral movement. For Manufacturing cost it also received a 2 since it is more complex, and includes more parts to be manufactured. The weight of the system is given a 3 as that it has more parts causing it to be heavier in comparison to the other 2 concepts. This system scores a 4 on maintenance cost. This is because it can be expected that a complex system like this is prone to a lot of (preventive) maintenance to ensure that no component/hinge fails. Position on the boom was rated a 4 this is due to the multiple movements of the trolley on the boom, which leads to more accurate adjustments, and also no need for minor adjustments of the entire crane. Furthermore, this trolley system is under hangs the girders, which is more desirable.

10.3.9 Reasoning for Single movement Top running Trolley and Under hanging Trolley

The scores received for the two single movement concepts are both the same with the exception of the boom placement. Both single Trolley system received a 4 for amount of parts as both do not have dual lateral movement in comparison to the first concept. Manufacturing cost was given a 3 since there are less parts needed and the complexity of the structure is simpler. Both were given a 3 for weight as they both do not include dual movement and so less parts are needed. With regard to the maintenance cost, both trolley systems score the same, a 4. It may be assumed that the maintenance cost of these systems will be lower than the first system, because these two systems are much simpler than the first concept. There are much less critical points in these two systems, which leads to shorter (preventive) maintenance time. Both concepts score the same number, because the concepts are decently similar, where the largest difference is in the position on the boom. The position on the boom is the only differing result for these two concepts. Since the Top running Trolley runs on top of the girders, this could lead to obstruction of the Truss system which supports the girders. In comparison the Under hanging Trolley system does not obstruct the Trusses and so it is given a rank of 4 while the Top Running Trolley a 2.

10.3.10 Chosen Concept

The chosen concept for the Trolley is Concept 3 "Under hanging Trolley", because it scored the highest score when comparing the 3 concepts to each other with usage of the weighing factors, which makes it the most logical choice. Another argument is that it is the most used trolley design for STS-Cranes. Moreover it can be concluded from literature analysis that this type of trolley can not obstruct the truss system of the crane and this system can be designed to carry the weight of the loads as required.

10.4 Boom calculations

10.4.1 Determining weight and location of counter weight

To make sure that the crane will not collapse if its used, especially if the trolley is at full load at one of the ends of the boom.Therefor a counterweight is needed. To be able to calculate the location and the wight of the counter weight. There are two extreme situations: the trolley is at the far right of the trolley is at the far left. If we look at the drawing in figure 61, it comes clear that the beam is supported at two points. If the trolley is located at the far left of the beam, the reaction force in point C should be zero as if the force would be tensile the crane would fall over, and if the force would be compressive the counter weight is heavy-er then it should be.

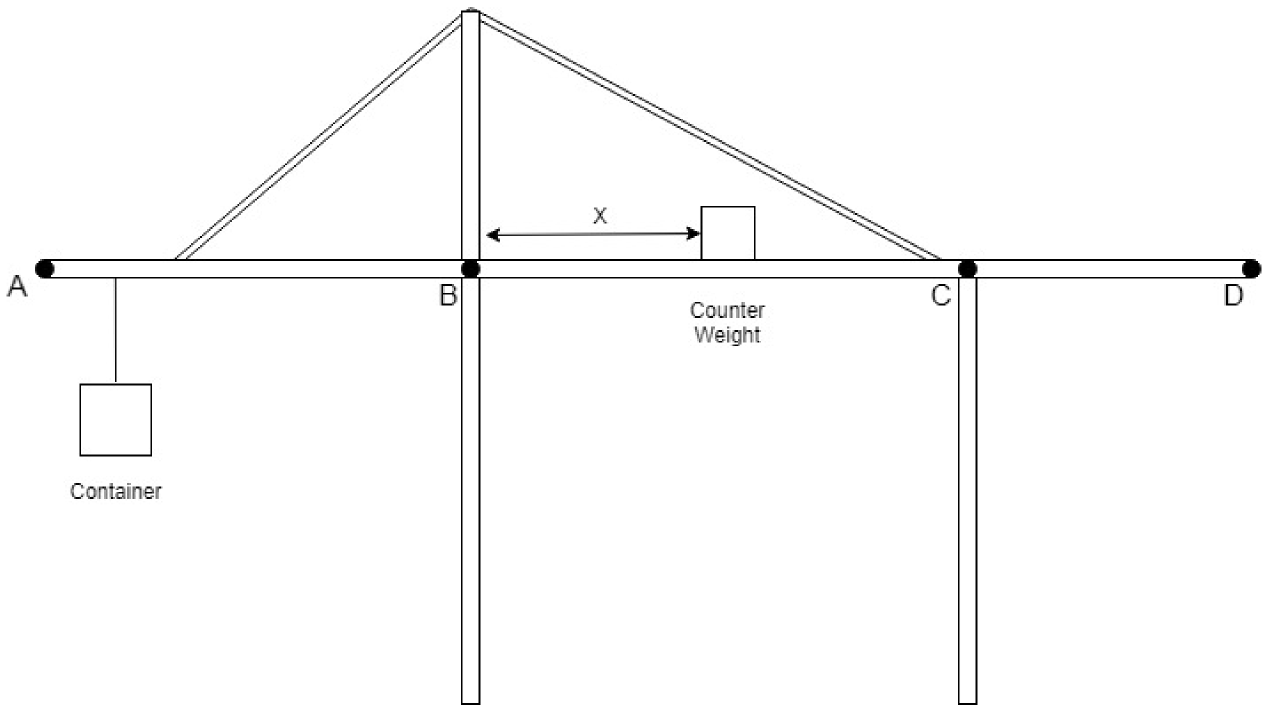


Figure 61: Situation sketch of the STS crane

The free body diagrams show the forces acting in both situations.

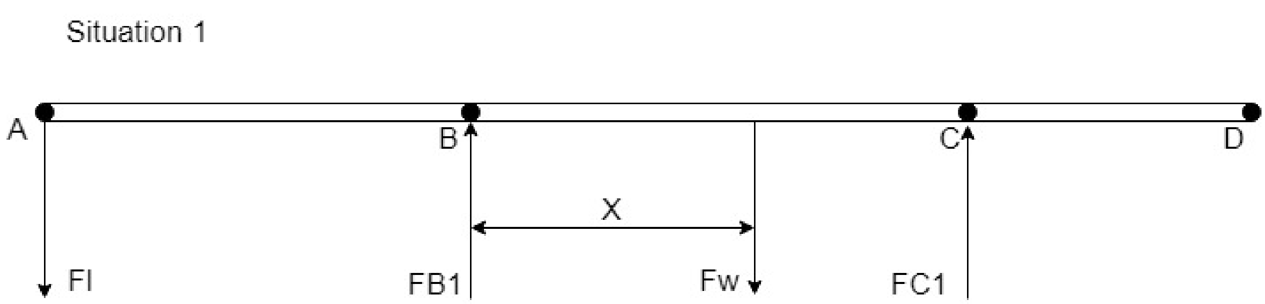


Figure 62: FBD with trolley located in the left position

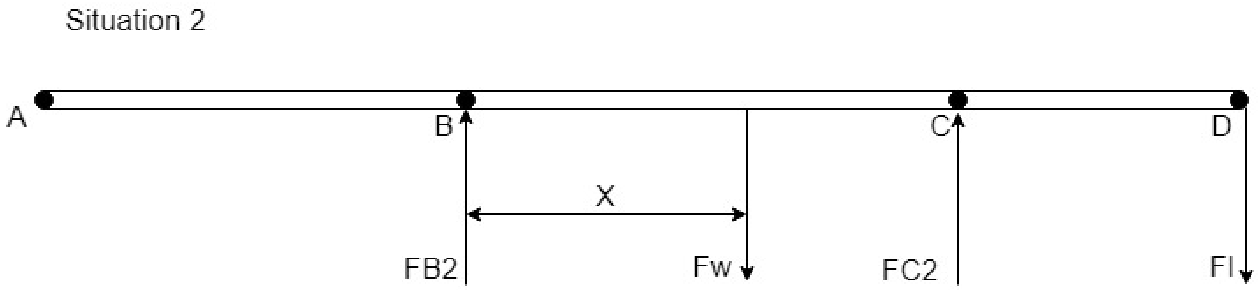


Figure 63: FBD with trolley located in the right position

After the Free Body Diagrams (FBDs) were drawn the equilibrium equations could be formulated for both situations

situation 1

$$\Sigma M \uparrow + : 0 = -F_L + F_{B1} + F_{C1} - F_W \tag{10}$$

$$\Sigma M_{@b} \circlearrowleft + : 0 = AB * F_L - F_W * X + F_{C1} * BC \tag{11}$$

Situation 2

$$\Sigma M \uparrow + : 0 = -F_L + F_{B1} + F_{C1} - F_W \tag{12}$$

$$\Sigma M_{@b} \circlearrowleft + : 0 = (BC + CD) * F_L - F_W * X + F_{C2} * BC \tag{13}$$

This gives the following list of variables :

- F_L Force of the trolley load.
- F_{B1} Force on support b.
- F_{B2} Force on support b.
- F_{C1} Force on support C.
- F_{C2} Force on support c.
- F_W Force of weight
- X Place of the weight
- AB Length AB
- BC Length BC
- CD Length CD

The following variables are known

- $F_L = 60,000kg\ or\ 588600N$
- $F_{B2} = 0$
- $F_{C1} = 0$
- $AB = 54m$
- $BC = 50m$
- $CD = 25m$

This leaves us with 4 unknowns and 4 equations. This mean that it is possible to solve this system of equations.

from the equilibrium equations the following equations can be derived for the 4 unknowns

$$F_{C2} = \frac{F_L * (AB + CD)}{BC} + F_{C1} + F_L \tag{14}$$

$$F_w = \frac{F_L * (AB + CD)}{BC} + F_{C1} - F_{B2} \tag{15}$$

$$F_{B1} = \frac{F_L * (AB + CD)}{BC} - F_{B2} \tag{16}$$

$$X = \frac{F_L * AB * BC + F_{C1} * BC^2}{F_L * (AB + CD) + F_{C1} * BC - F_{B2} * BC} \tag{17}$$

If the known variables are used to calculate the unknowns, the following numbers are obtained:

- $F_{B1} = 9.3 * 10^5 N$
- $F_{C2} = 1.5 * 10^6 N$
- $F_W = 9.3 * 10^5 N$
- $X = 34m$

10.4.2 Placing the cables/determining height of guiding wheels

In order to be able to calculate the ratio between the height of the extension on the crane and the length on which the cable is attached on the boom from the hinge point, calculations are made using statics.
This is done by saying that the maximum deflection in the whole boom must be the same, as the maximum deflection occurs at the points where the trolley is located at the tip of the boom and in the middle between the cable attachment and the hinge point.
Knowing this, the length between the hinge point and the cable attachment (L) as shown in figure 64 and 67 can be calculated by stating that the maximum deflection with the trolley on the tip of the boom = maximum deflection with the trolley at 1/2 L.
It is assumed that all members are slender members, which are prismatic and homogeneous.
For this statement, the following two calculations were made:

Situation 1
Figure 64 gives an overview of the situation where the trolley is located in the middle of length L.

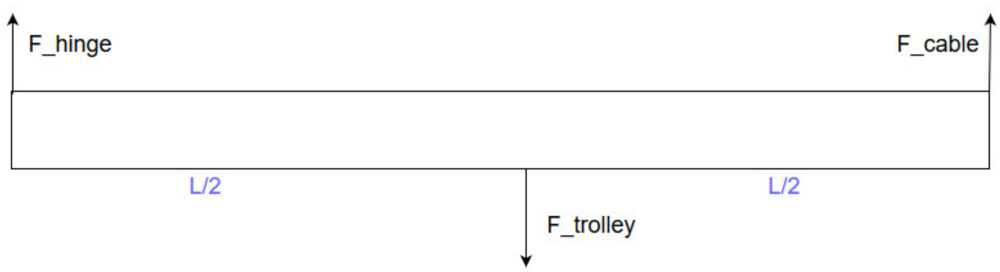


Figure 64: FBD of trolley on half the length between cables and hinge

The equations of equilibrium of F_y and the moment around the point where F_{hinge} is acting gives the forces F_{hinge} and F_{cable} expressed in $F_{trolley}$.
Hereafter, $F_{trolley}$ is called P.

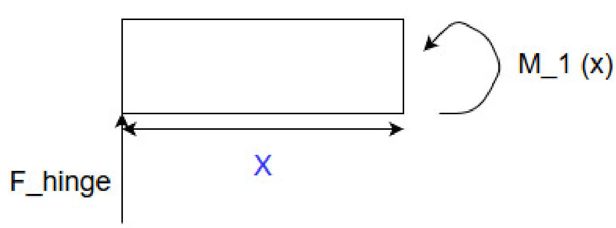


Figure 65: Method of sections first cut

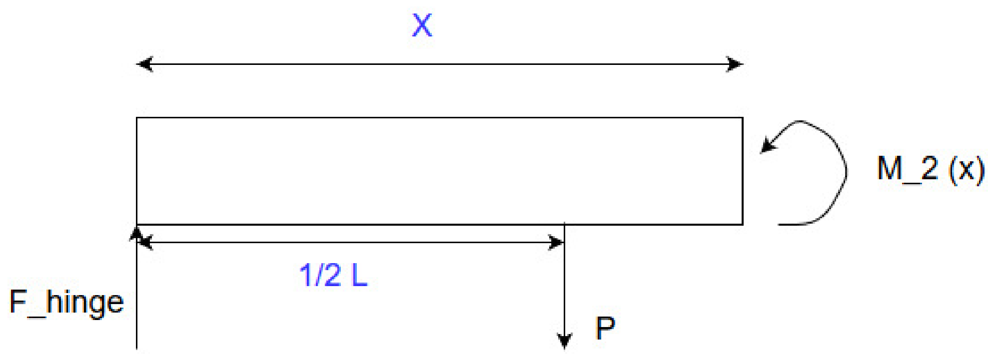


Figure 66: Method of sections second cut

Using the above sections and with determining the equations of equilibrium again, the reaction moments $M_1(x)$ and $M_2(x)$ can be calculated, which are:

$$M_1(x) = PX/2 \qquad for \quad 0 < x < L/2 \tag{18}$$

$$M_2(x) = -PX/2 + FL/2 \qquad for \quad L/2 < x < L \tag{19}$$

Knowing this, the two equations for deflection can be made:

$$EIV_1(x) = PX^3/12 + C_1X + C_3 \qquad for \quad 0 < x < L/2 \tag{20}$$

$$EIV_2(x) = -PX^3/12 + PLX^2/4 + C_2X + C_4 \qquad for \quad L/2 < x < L \tag{21}$$

The following boundary conditions can be stated:
$V_1(0) = 0$
$V_2(L) = 0$
$V_1(L/2) = V_2(L/2)$
$\frac{dV_1(L/2)}{dx} = \frac{dV_2(L/2)}{dx}$

Using these boundary conditions, the equations for deflection can be worked out, giving:

$$V_1(x) = \frac{PX^3}{12EI} - \frac{PL^2X}{16EI} \qquad for \quad 0 < x < L/2 \tag{22}$$

$$V_2(x) = -\frac{PX^3}{12EI} + \frac{PLX^2}{4EI} - \frac{3PL^2X}{16EI} + \frac{PL^3}{48EI} \qquad for \quad L/2 < x < L \tag{23}$$

This gives a maximum deflection on X = L/2 of $V_{max} = -\frac{PL^3}{48EI}$

Situation 2
Now the same calculations are made for the situation where the trolley is placed at the end of the boom. The objective is to calculate the maximum deflection on point X = L + A.
F_{hinge} is hereafter called F_A and F_{cable} F_B.

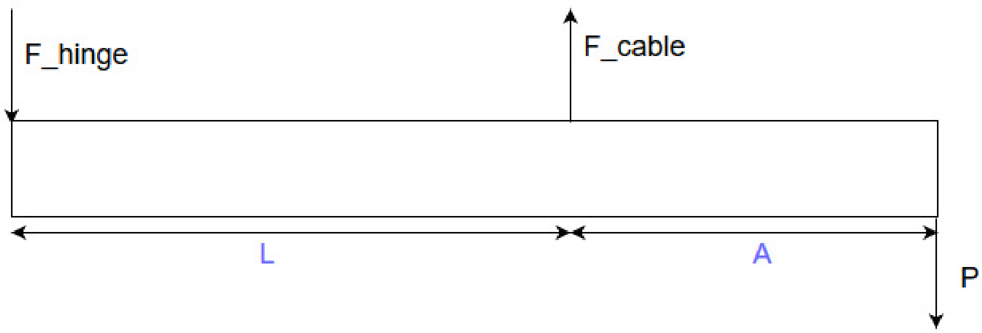


Figure 67: FBD of trolley on the end of the boom

Here the FBD of the whole situation is given.
The equilibrium equations of F_y and the moment around the point where F_{hinge} is acting were determined, giving:

$$\sum F_y \uparrow + = -F_A + F_B - P = 0 \tag{24}$$

$$\sum M_{@F_A} \circlearrowleft + = F_B L - P(L + A) = 0 \tag{25}$$

This gives that $F_A = \frac{PA}{L}$ and $F_B = P + \frac{PA}{L}$
Now two cuts have to be made to determine the equations of deflection.

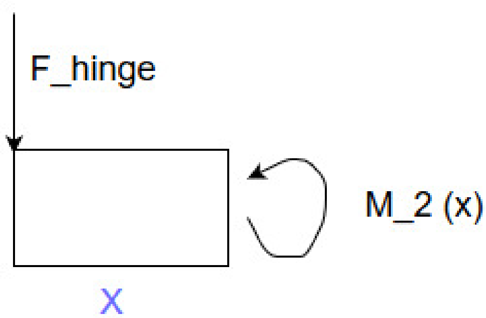


Figure 68: Method of sections first cut

The first cut gives the following reaction moment:

$$M_1(x) = -F_A X \tag{26}$$

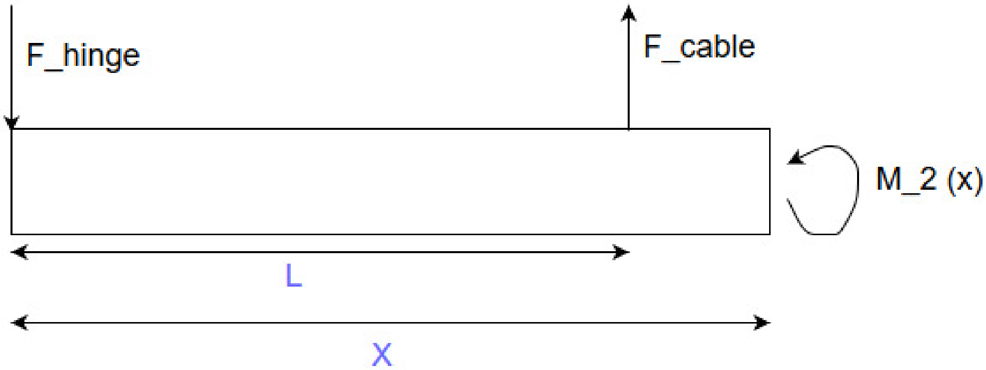


Figure 69: Method of sections second cut

The second cut gives the following reaction moment:

$$M_2(x) = F_B(X - L) - F_A X \tag{27}$$

This gives the following equations for deflection:

$$V_1(x) = -\frac{\frac{1}{6}F_A X^3}{EI} + C_1 X + C_2 \qquad for \quad 0 < X < L \tag{28}$$

$$V_2(x) = \frac{F_B(\frac{1}{6}X^3 - \frac{1}{2}LX^2) - \frac{1}{6}F_A X^3}{EI} + C_3 X + C_4 \qquad for \quad L < X < A + L \tag{29}$$

By using the following boundary conditions, the complete deflection equations can be worked out:
$V_1(0) = 0$
$V_1(L) = 0$
$\frac{dV_1(L)}{dx} = \frac{dV_2(L)}{dx}$
$V_1(L) = V_2(L)$

Out of these boundary conditions, the following complete deflection formulas are determined:

$$V_1(x) = -\frac{\frac{1}{6}F_A X^3}{EI} + \frac{\frac{1}{6}F_A L^2}{EI} X \qquad for \quad 0 < X < L \tag{30}$$

$$V_2(x) = \frac{F_B(\frac{1}{6}X^3 - \frac{1}{2}LX^2) - \frac{1}{6}F_A X^3}{EI} + \frac{\frac{1}{6}F_A L^2 + \frac{1}{2}F_B L^2}{EI} X \qquad for \quad L < X < A + L \tag{31}$$

By now calculating $V_2(L + A) = V_{max}$, given that A + L = 48 metres, L can be calculated. This gives an L of 29.3 metres.
Now given that the maximum angle of the boom with the horizon is 80°and the lifting cable is horizontal when the boom is in lifted position, the height of the extension on the top of the crane can be calculated using geometry. This gives a height of 28.9 metres.

10.5 The effect of wind on the tension in wire ropes

According to the project description the maximum wind speed of 72 km/h corresponds to 240 N/m^2.[1] The maximum wind force (Fw) is obtained by multiplying 240 by the area of the side with the largest area. The gravitational force (Fg) is obtained by multiplying the gravitational acceleration with the maximum weight of spreader and container. From the calculations it follows that the total force in the cables increases by only 47,63 N.

Fg = m x g = 60000 x 9.81 = 588600 N
Fw = 240 x A = 240 x 12 x 2,6 = 7488 N

Fsy = Fg , Fsx = Fw

α = tan(Fsy / Fsx) = 89.27 degrees
Fs = Fsx / cos(α) = 588647,63 N

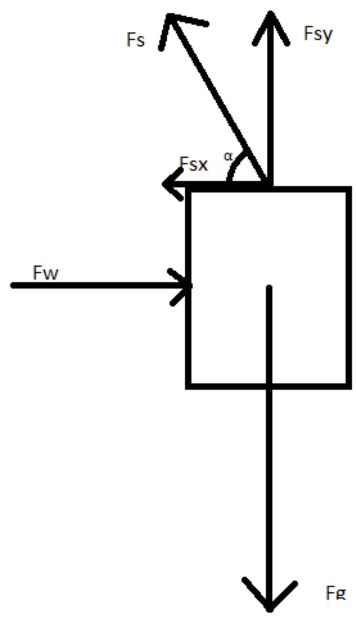


Figure 70: Free body diagram of container subjected to wind force

10.6 Overview of the dimensions of the STS crane

10.6.1 Boom Dimension

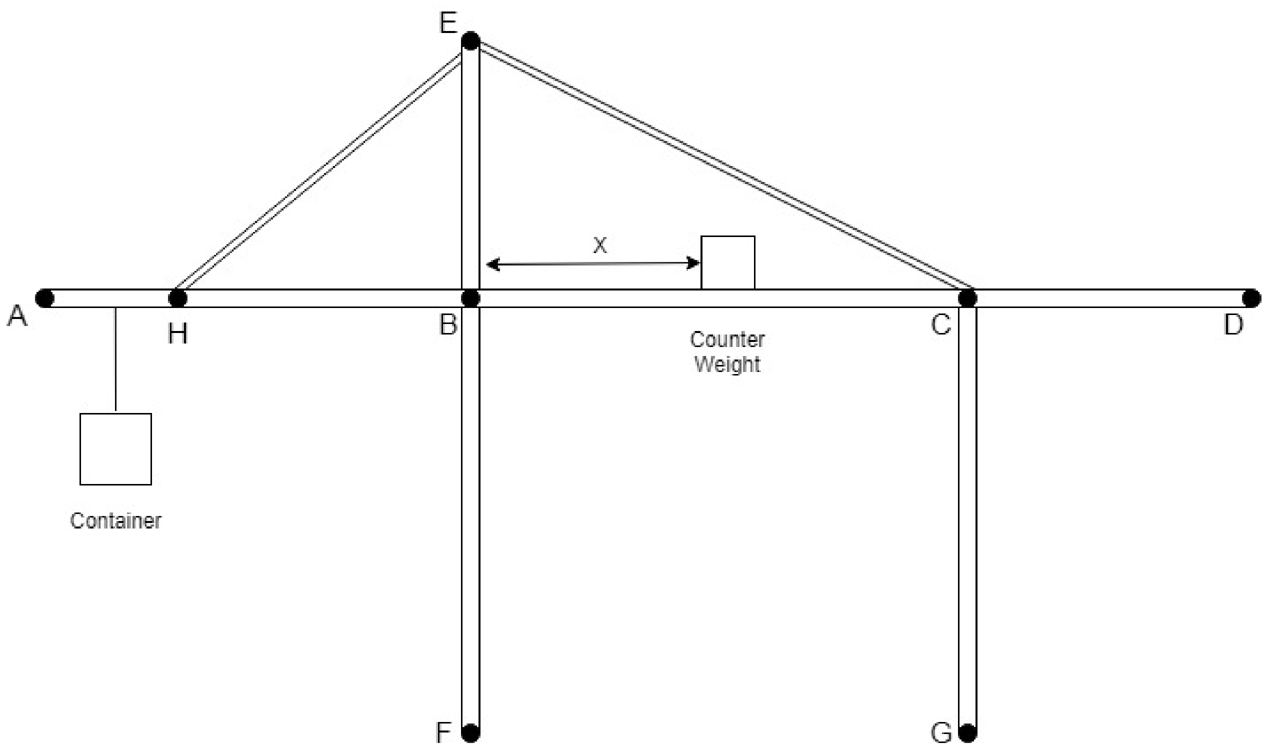


Figure 71: Dimensions of the boom

The dimensions of the crane are given as:

- $AB = 54m$
- $BC = 50m$
- $CD = 25m$
- $BH = 33.5m$
- $BE = 27.1m$
- $X = 34m$

10.7 FBD Single Truck

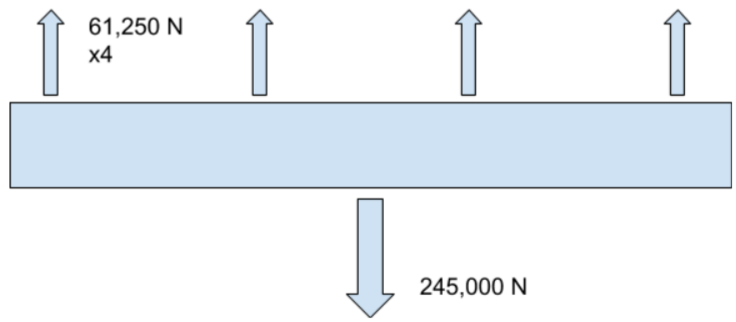


Figure 72: Single Truck FBD

10.8 Technical Drawing

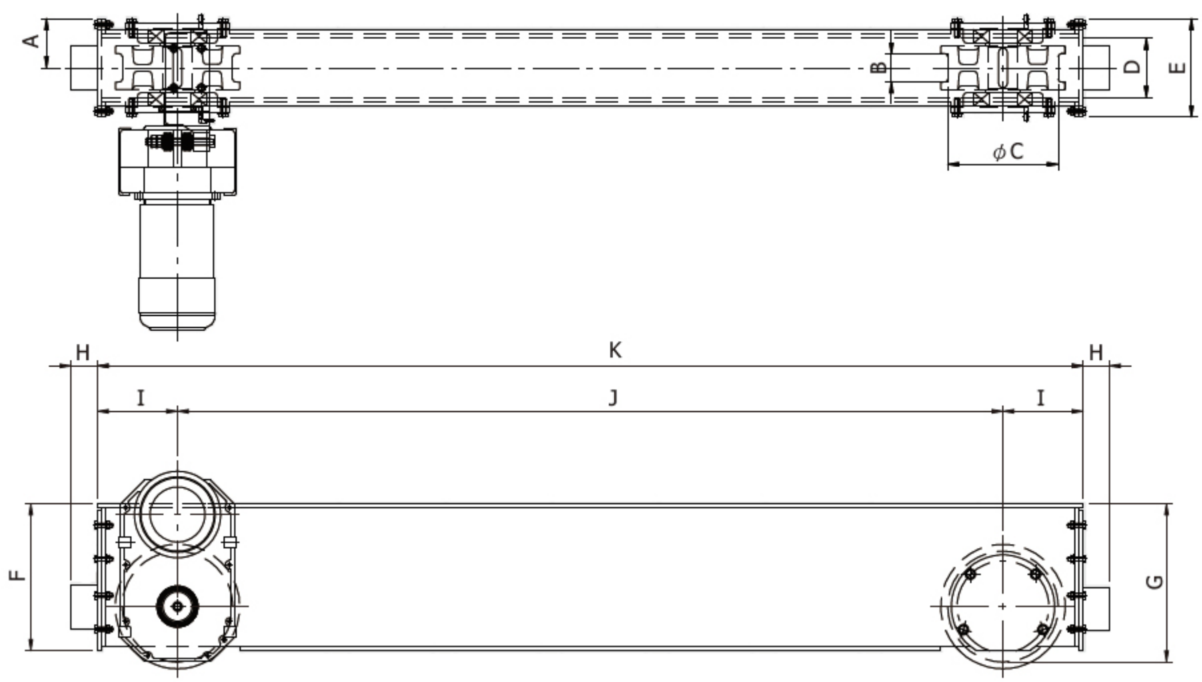


Figure 73: Technical Drawing Single Truck[15]

$A = 15cm$, $D = 19.4cm$, $B = 8cm$, $E = 30cm$, $F = 50cm$, $K = 400cm$, $C = 40cm$,

10.8.1 FEM Model definition

Below, the final model definition of the FEM calculations can be found to see precisely which beam types were assigned to which element.

```
%% 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.

E = 206e9;
nu = 0.295;
G = E/(2*(1+nu));
rho = 7900;

%%% define coordinates of the nodes [meter]
xyz_nodes = [0,0,0
12,0,0
24,0,0
30,0,0
36,0,0
48,0,0
0,6,0
6,6,0
12,6,0
18,6,0
24,6,0
30,6,0
36,6,0
42,6,0
48,6,0
6,3,5.2
18,3,5.2
30,3,5.2
42,3,5.2
-6,0,28.1
-6,6,28.1
-12,0,0
-12,6,0
-6,3,5.2
-6,6,0
-6,0,0
-6,-4.5,0
-6,10.5,0
-6,-4.5,-45
-6,10.5,-45
-24,0,0
-36,0,0
-48,0,0
-24,6,0
-36,6,0
-48,6,0
-18,6,0
-30,6,0
-42,6,0
-60,0,0
-60,6,0
-54,6,0
-54,0,0
-54,-4.5,-45
-54,10.5,-45
-54,-4.5,0
```

```
-54,10.5,0
-72,0,0
-72,6,0
-66,6,0
-18,3,5.2
-30,3,5.2
-42,3,5.2
-54,3,5.2
-66,3,5.2];

%% define which nodes are connected by elements and
elements = [1,2
    2,3
    3,4
    4,5
    5,6
    7,8
    8,9
    9,10
    10,11
    11,12
    12,13
    13,14
    14,15
    1,7
    6,15
    1,8
    2,8
    2,10
    3,10
    3,12
    5,12
    5,14
    6,14
    16,17
    17,18
    18,19
    1,16
    2,16
    2,17
    3,17
    3,18
    5,18
    5,19
    6,19
    7,16
    9,16
    9,17
    11,17
    11,18
    13,18
    13,19
    15,19
    12,21
    4,20
    22,23
    22,24
```

```
1,24
7,24
23,24
7,25
23,25
1,25
22,25
1,26
22,26
25,26
25,28
26,27
20,21
20,27
21,28
28,30
27,29
23,37
37,34
34,38
38,35
35,39
39,36
33,32
32,31
31,22
22,37
37,31
31,38
38,32
32,39
39,33
36,42
42,41
33,42
42,40
33,43
43,40
47,42
42,43
43,46
33,43
43,40
46,44
45,47
28,45
27,44
21,47
20,46
41,50
49,50
40,48
40,50
50,48
48,49
24,51
51,52
```

```
     52,53
     53,54
     54,55
     23,51
     34,51
     22,51
     31,51
     34,52
     35,52
     31,52
     32,52
     32,53
     33,53
     36,53
     35,53
     36,54
     41,54
     40,54
     33,54
     41,55
     49,55
     48,55
     40,55
     4,18
     12,18
     12,4
     43,54
     42,54];

%% define outer diameter and wall thickness of each element [meter]
% C = [outer diameter, wall thickness]

b1=[0.25,0.1];
b2=[0.4,0.08];
b3=[1,0.25];

C=[b2
   b2
   b2
   b2
   b2%5
   b2
   b2
   b2
   b2
   b2%10
   b2
   b2
   b2
   b1
   b1%15
   b1
   b1
   b1
   b1
   b1%20
```

```
b1
b1
b1
b1
b1%25
b1
b1
b1
b1
b1%30
b1
b1
b1
b1
b1%35
b1
b1
b1
b1
b1%40
b1
b1
b1
b1
b1%45
b1
b1
b1
b1
b2%50
b2
b1
b1
b2
b2%55
b2
b3
b3
b1
b2%60
b2
b3
b3
b2
b2%65
b2
b2
b2
b2
b2%70
b2
b2
b1
b1
b1%75
b1
b1
```

```
    b1
    b2
    b2%80
    b1
    b1
    b1
    b1
    b3%85
    b3
    b3
    b2
    b2
    b3%90
    b3
    b3
    b3
    b1
    b1%95
    b2
    b2
    b2
    b1
    b1%100
    b1
    b1
    b1
    b1
    b1%105
    b1
    b1
    b1
    b1
    b1%110
    b1
    b1
    b1
    b1
    b1%115
    b1
    b1
    b1
    b1
    b1%120
    b1
    b1
    b1
    b1
    b1%125
    b1
    b1
    b1
    b1
    b1
    b1];

%% split elements into smaller sections
```

```
element_split = zeros(1,131);

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

xyz_constr = [0,0,0,0,0,0
    0,0,0,0,0,0
    0,0,0,0,0,0
    0,0,0,0,0,0
    0,0,0,0,0,0
    0,0,0,0,0,0
    0,0,0,0,0,0
    0,0,0,0,0,0
    0,0,0,0,0,0
    0,0,0,0,0,0
    0,0,0,0,0,0
    0,0,0,0,0,0
    0,0,0,0,0,0
    0,0,0,0,0,0
    0,0,0,0,0,0
    0,0,0,0,0,0
    0,0,0,0,0,0
    0,0,0,0,0,0
    0,0,0,0,0,0
    0,0,0,0,0,0
    0,0,0,0,0,0
    0,0,0,0,0,0
    0,0,0,0,0,0
    0,0,0,0,0,0
    0,0,0,0,0,0
    0,0,0,0,0,0
    0,0,0,0,0,0
    0,0,0,0,0,0
    1,1,1,1,1,1 %29,ground
    1,1,1,1,1,1 %30,ground
    0,0,0,0,0,0
    0,0,0,0,0,0
    0,0,0,0,0,0
    0,0,0,0,0,0
    0,0,0,0,0,0
    0,0,0,0,0,0
    0,0,0,0,0,0
    0,0,0,0,0,0
    0,0,0,0,0,0
    0,0,0,0,0,0
    0,0,0,0,0,0
    0,0,0,0,0,0
    0,0,0,0,0,0
    1,1,1,1,1,1 %44, ground
    1,1,1,1,1,1 %45, ground
    0,0,0,0,0,0
    0,0,0,0,0,0
    0,0,0,0,0,0
    0,0,0,0,0,0
    0,0,0,0,0,0
    0,0,0,0,0,0
    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 = [6, 3, -600000 %safety factor = 2 for all forces
             15, 3, -600000
             32, 3, -18e4
             35, 3, -18e4
             1,2,12500
             2,2,12500
             3,2,12500
             4,2,12500
             5,2,12500
             6,2,12500
             22,2,12500
             31,2,12500
             32,2,12500
             33,2,12500
             40,2,12500
             48,2,12500];


%% define plot settings
% Plot 1
plot_undeformed        = true;
plot_undeformed_labels = true;
plot_undeformed_forces = false;
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 = 100;

%% Calculation of model parameters and element splitting (DON'T EDIT!)
% Calculate area of each element
A =  C(:,1).^2 - (C(:,1)-2.*C(:,2)).^2;

% Calculate area moments of inertia
I = [(1/12.*C(:,1).^4)-(1/12.*C(:,2).^4),(1/12.*C(:,1).^4)-(1/12.*C(:,2).^4)];
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)
```

10.8.2 MATLAB script for the gear calculations

```
clc , clear , format shortG , format compact
%%
TV = 35.19;
VR_11 = 3.4; %From Graph
VR_12 = 4.1; %From Graph

VR_1_Trial = 4;
VR_2_Trial = 3.3;

N_p = 19;

N_g1_trial = VR_1_Trial.*N_p;
N_g1 = 73 ;   %PRIME NUMBER
VR_1 = N_g1./N_p; %Between VR_11 and VR_12, good

N_g2_Trial = VR_2_Trial.*N_p;
N_g2 = 61; %PRIME NUMBER
VR_2 = N_g2./N_p;

VR_3_Trial = TV./VR_1./VR_2;
N_g3_Trial = VR_3_Trial.*N_p;
N_g3 = 54;
VR_3 = N_g3./N_p;

TV_Actual = VR_1.*VR_2.*VR_3;

m = 15;

Dp = m.*N_p;
D1 = m.*N_g1;
D2 = m.*N_g2;
D3 = m.*N_g3;

W_D_Ratio = 0.6; %value from table
bp_trial = W_D_Ratio.*Dp;
bp = 175;

%% Stresses Pinion 1 (p1)
P = 400000;
K_0 = 1.75;
P_d = P.*K_0;
w_p1 = 156.24;
T_p1 = P_d./w_p1;
F_t_p1 = T_p1./(0.5.*Dp./1000);
```

```
clc, clear, format compact


%% Project Shaft Design
power = 400000; % W
D1 = 1095;      % Diameter of gears 1,2,3 respectively, all
in mm
D2 = 915;
D3 = 810;
Dp = 285;       % Diameter of pinion gear, mm
theta = 20;     % Pressure angle, degrees
Kt1 = 1.5;      % For a well-rounded fillet
Kt2 = 2.5;      % For a sharp fillet
Kt3 = 3.0;      % For a ring-groove
RGF = 1.06;     % Ring groove factor
sy = 572.28;    % MPa, use figure A4-2
su = 813.61;    % MPa, use figure A4-2
sn = 289.59;    % MPa, use figure 5-8
Cs = 0.75;      % Size factor, use figure 5-9
Cr = 0.81;      % Reliability factor
sn_prime = sn.*Cs.*Cr;
N = 2;          % Design factor

%% Shaft 1
% Directly connected to motor
as_shaft1 = 156.24;
T_shaft1 = (power./as_shaft1);

 W_tB = T_shaft1./(1/2.*Dp./1000);
 W_rB = W_tB .* tand(theta);

 RAy = 15720.25;
 RCy = 2245.75;
 RAx = 5721.71;
 RCx = 817.4;

 MBy = -1572.025;
 MBx = -572.1725;
 M_B_Bending = sqrt(MBx.^2 + MBy.^2);

 VAx = -5721.725;
 VAy = -15720.25;
 VCy = -2245.75;
 VCx = -817.4;
```

```
 V_A_Shearing = sqrt(VAx.^2 + VAy.^2);
 V_C_Shearing = sqrt(VCx.^2 + VCy.^2);

  D_shaft1_1 =
10.*((32.*N./pi).*sqrt((3./4).*(T_shaft1./sy).^2)).^(1./3)
  D_shaft1_2 =
RGF.*10.*((32.*N./pi).*sqrt(((Kt3.*M_B_Bending)./sn_prime).^
2 + (3./4).*(T_shaft1./sy).^2)).^(1./3)
  D_shaft1_3 = sqrt((2.94.*Kt2.*V_C_Shearing)./sn_prime)


%% Shaft 2
as_shaft2 = 40.67;      % angular speed shaft 2, rad/s

T_shaft2 = (power./as_shaft2);          % Nm

W_tB = T_shaft2 ./ (1/2.*D1./1000);     % N
W_rB = W_tB .* tand(theta);             % N
W_tC = T_shaft2 ./ (1/2.*Dp./1000);     % N
W_rC = W_tC .* tand(theta);             % N

RAx = 6600.1;    % N, reaction force in bearing A in x-
direction
RDx = 11737.625;    % N, reaction force in bearing D in x-
direction
RAy = 36755;  % N, reaction force in bearing A in y-
direction
RDy = 50228;  % N, reaction force in bearing D in y-
direction

MBx = -660.001;     % Nm, reaction moment in gear B in x-
direction
MCx = -4601.53;      % Nm, reaction moment in gear C in x-
direction
MBy = 5022.8;     % Nm, reaction moment in gear B in y-
direction
MCy = 14702;       % Nm, reaction moment in gear C in y-
direction

M_B_Bending = sqrt(MBx.^2 + MBy.^2);
M_C_Bending = sqrt(MCx.^2 + MCy.^2);

VAx = -6600.1;          % N, Shear force in x-direction
VAy = 50228;            % N, Shear force in y-direction
```

```
VDx = -11737.625;       % N, Shear force in x-direction
VDy = 36755;            % N, Shear force in y-direction

V_A_Shearing = sqrt(VAx.^2 + VAy.^2);
V_D_Shearing = sqrt(VDx.^2 + VDy.^2);

% Calculating the minimum shaft diameter
 D_shaft2_1 =  sqrt((2.94.*Kt2.*V_A_Shearing)./sn_prime)
 D_shaft2_2 =
10.*((32.*N./pi).*sqrt(((Kt1.*M_B_Bending)./sn_prime).^2)).^
(1./3)
 D_shaft2_3 =
RGF.*10.*((32.*N./pi).*sqrt(((Kt3.*M_B_Bending)./sn_prime).^
2 + (3./4).*(T_shaft2./sy).^2)).^(1./3)
 D_shaft2_5 =
RGF.*10.*((32.*N./pi).*sqrt(((Kt3.*M_C_Bending)./sn_prime).^
2+(3./4).*(T_shaft2./sy).^2)).^(1./3)
 D_shaft2_6 =
10.*((32.*N./pi).*sqrt(((Kt1.*M_C_Bending)./sn_prime).^2)).^
(1./3)
 D_shaft2_7 = sqrt((2.94.*Kt2.*V_D_Shearing)./sn_prime)
 D_shaft2_4 = 1.1 .*D_shaft2_5

%% Shaft 3
 as_shaft3 = 12.67;      % angular speed shaft 3, rad/s

        % Design factor
 T_shaft3 = (power./as_shaft3);          % Nm

 W_tB = T_shaft3 ./ (1/2.*D2./1000);     % N
 W_rB = W_tB .* tand(theta);              % N
 W_tC = T_shaft3 ./ (1/2.*Dp./1000);     % N
 W_rC = W_tC .* tand(theta);             % N

 RAx = 2478.375;    % N, reaction force in bearing A in x-
direction
 RDx = 2478.375;    % N, reaction force in bearing D in x-
direction
 RAy = 62197.25;    % N, reaction force in bearing A in y-
direction
 RDy = 228359.75;   % N, reaction force in bearing D in y-
direction
```

```
 MBx = 991.35;        % Nm, reaction moment in gear B in x-
direction
 MCx = -5799.94;      % Nm, reaction moment in gear C in x-
direction
 MBy = 24878.9;       % Nm, reaction moment in gear B in y-
direction
 MCy = 22835.8;       % Nm, reaction moment in gear C in y-
direction

 M_B_Bending = sqrt(MBx.^2 + MBy.^2);
 M_C_Bending = sqrt(MCx.^2 + MCy.^2);

 VAx = 2478.375;            % N, Shear force in x-direction
 VAy = 62197.25;            % N, Shear force in y-direction
 VDx = -57999.375;          % N, Shear force in x-direction
 VDy = 228359.75;           % N, Shear force in y-direction

 V_A_Shearing = sqrt(VAx.^2 + VAy.^2);
 V_D_Shearing = sqrt(VDx.^2 + VDy.^2);

 % Calculating the minimum shaft diameter
  D_shaft3_1 = sqrt((2.94.*Kt2.*V_A_Shearing)./sn_prime)
  D_shaft3_2 =
10.*((32.*N./pi).*sqrt(((Kt1.*M_B_Bending)./sn_prime).^2)).^
(1./3)
  D_shaft3_3 =
RGF.*10.*((32.*N./pi).*sqrt(((Kt3.*M_B_Bending)./sn_prime).^
2 + (3./4).*(T_shaft3./sy).^2)).^(1./3)
  D_shaft3_5 =
RGF.*10.*((32.*N./pi).*sqrt(((Kt3.*M_C_Bending)./sn_prime).^
2+(3./4).*(T_shaft3./sy).^2)).^(1./3)
  D_shaft3_6 =
10.*((32.*N./pi).*sqrt(((Kt1.*M_C_Bending)./sn_prime).^2)).^
(1./3)
  D_shaft3_7 = sqrt((2.94.*Kt2.*V_D_Shearing)./sn_prime)
  D_shaft3_4 = 1.1 .*D_shaft3_5

%% Shaft 4
% Shaft that is connected to the drum / load
 as_shaft4 = 4.4567;
 T_shaft4 = (power./as_shaft4);

 W_tB = T_shaft4./(1/2.*D3./1000);
 W_rB = W_tB .* tand(theta);
```

```
 RAy = 27701.25;
 RCy = 193908.75;
 RAx = 10082.5;
 RCx = 70577.5;

 MBy = 19390.875;
 MBx = 7057.75;
 M_B_Bending = sqrt(MBx.^2 + MBy.^2);

 VAx = 10082.5;
 VAy = 27701.25;
 VCy = 193908.75;
 VCx = 70577.5;
 V_A_Shearing = sqrt(VAx.^2 + VAy.^2);
 V_C_Shearing = sqrt(VCx.^2 + VCy.^2);

 D_shaft4_1 =
10.*((32.*N./pi).*sqrt((3./4).*(T_shaft4./sy).^2)).^(1./3)
 D_shaft4_2 =
RGF.*10.*((32.*N./pi).*sqrt(((Kt3.*M_B_Bending)./sn_prime).^
2 + (3./4).*(T_shaft4./sy).^2)).^(1./3)
 D_shaft4_3 = sqrt((2.94.*Kt2.*V_C_Shearing)./sn_prime)
```

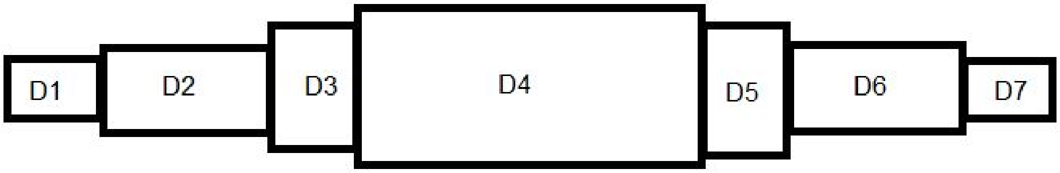


Figure 74: Lay out of the different diameters

10.8.3 Matlab script about the calculations on the linear guideways

```
% Weight of the crane = 2650.6 tonnes
Total_weight = 2650.6 .*10^3     % in kg

% Force imposed by the crane:
Force_crane = Total_weight .* 9.81  % in Newton

% Yield stress Gantrex Inc. 110Cr-V alloy = 700 MPa
Yield_stress = 700.*10^6       % in Pascal

% Implement a safety factor for the stress
Safety_factor = 5

% The maximum allowable stress on the rail is therefore:
Maximum_allowable_stress = Yield_stress ./ Safety_factor  % 140.*10^6
Pa

%{ Using the formula for stress:  Stress = force/area, the required
contact
 ... area, to achieve the maximum allowable stress can be determined.
%}

Contact_area = Force_crane ./ Maximum_allowable_stress    % 0.1857 m^2

%{ There are 4 contact points with the rails, so each contact point
should
... have: %}
Contact_area_per_beam = Contact_area ./ 4                 % 0,0464 m^2
```

10.8.4 Matlab script for the motor calculations

```
clc , clear , format shortG , format compact

%% Motor Characteristics
RPM_Motor = 1492;
J_Motor = 7.16;
T_Motor = 2560;

%% Drum Characteristics
RPM_Drum = 2./(0.9.*pi).*60;
r = 0.9/2;
r_2 = r;
r_1 = r_2 - 0.015;
L = 2.2;
rho = 7.1e3;
m_drum = rho.*(pi.*(r_2.^2 - r_1.^2)).*L;
J_L = 0.5.*m_drum.*(r_2.^2 + r_1.^2);

%% Characteristics of the mass (Container + Spreader)
m = 196200./9.81;
J_m = m.*(r.^2);

%% Angular speed alpha
dw = (RPM_Motor./60).*(2.*pi);
dt = 5;
alpha = dw./dt;

%% Calculating Inertia Ratio
i_G = RPM_Motor./RPM_Drum;
i_D = 1./r;
J_total = J_Motor + (J_L./i_G.^2) + m./((i_D.^2).*(i_G).^2);
Inertia_ratio = J_total./J_Motor

%% Can the motor produce enough torque for acceleration?
J_Lm = J_L + J_m;
T_Drum_Continuous = 196200.*r
T_Drum_Acceleration = J_Lm.*alpha
Max_T_Motor = T_Motor./0.6;
T_Gear = Max_T_Motor.*i_G

%% Motor Choice:
%Simotics SD self-ventilated or forced-air cooled motors - cast-iron series
%1LE5604 Performance Line: 400 kw, 4-pole, 1492 rpm at 50Hz.
```

Figure 75: MATLAB code for motor calculations

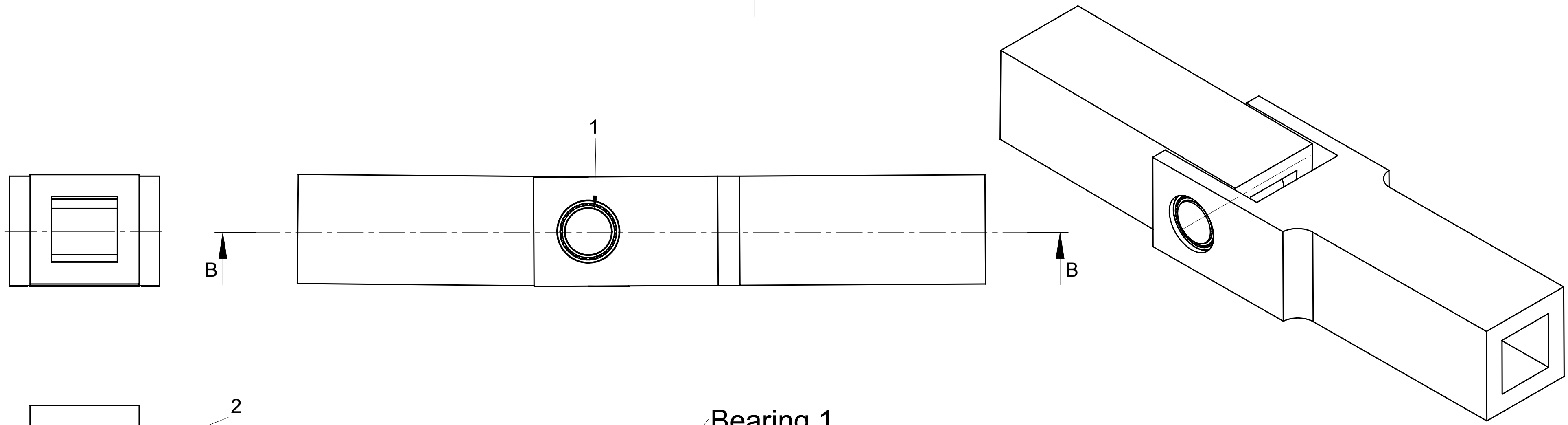
10.8.5 Matlab script for the weld calculations

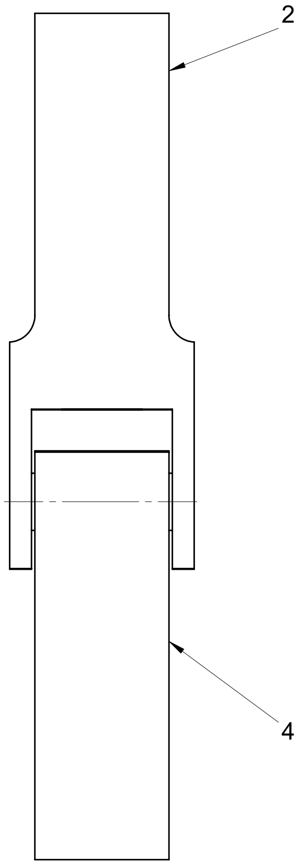
```
% Determine the forces in downward direction:
FB = 0                                                  % in Newton
Fboom = 5.5271 .*10^5                                   % in kg
ForceFboom = Fboom .* 9.81                              % in Newton
FC = 9.3 .* 10^5                                        % in Newton
Ftrolley = 588600                                       % in Newton

Total_force_down = (ForceFboom + FC + Ftrolley)         % in Newton

% Determine the force in the beams:
Local_stress_Node_44  =   5.4774e+07                    % in Pa
Length_of_the_beam = 45.6                               % in meter
Length_of_the_beam_side = 0.25 - 0.1                    % in meter
Cross_sectional_area_beam = Length_of_the_beam_side .^ 2   % in meter^2
Force_in_the_beam = Local_stress_Node_44 .* Cross_sectional_area_beam
                                                        % in Newton

% 2 beams, for simplicity compromised in 1 calculation
Beam_forces_comprised = Force_in_the_beam .* 2          % in Newton

% Correct the forces in the beam to upward y-direction
Angle_of_beams = 37.9                                   % in degrees
Comprised_beam_force_positive_y_direction = Beam_forces_comprised .* ...
  sind(Angle_of_beams)                                  % in Newton

% Determine the total force on the boom
total_force_boom = Comprised_beam_force_positive_y_direction -...
  Total_force_down                                      % - because negative direction



% Now the moments in the material have to be determined.
... Using the sum of the moments = 0 equation, all moments around point
  ... point a are determined

% First, determine the lengths of the segments.

L_trolley_endpoint_A = 54                               % in meter
Length_C_A = 34                                         % in meter

% Since FB = 0 N, there is no moment around A
% Since FA and FBoom cross point A, they don't exert a moment around A.

% Clockwise moments around A get a - sign, counterclockwise moments are
% positive
```

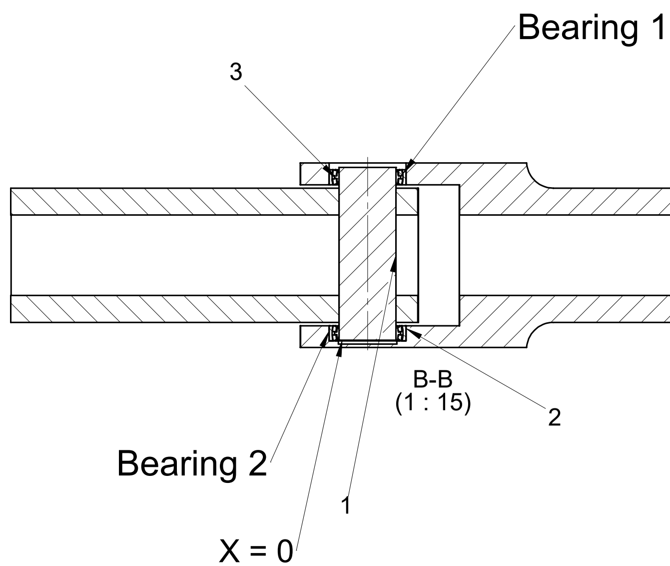
```
% Moment exerted by trolley in end position
Moment_trolley = L_trolley_endpoint_A .* Ftrolley                         % in Nm
Moment_exerted_by_point_C = Length_C_A .* FC .* -1                        % in Nm
Total_moment_around_A = Moment_trolley + Moment_exerted_by_point_C % in Nm

% Determine Fa and Ma
Fa = -1.*total_force_boom                                  % in N, - because of other direction
Ma = -1.*Total_moment_around_A                             % in Nm, - because of clockwise moment
```

10.8.6 Technical Drawings Boom Hinge

On the following pages, the technical drawings of a side of the boom hinge point can be found. The design choices can be found in subsection 7.2.

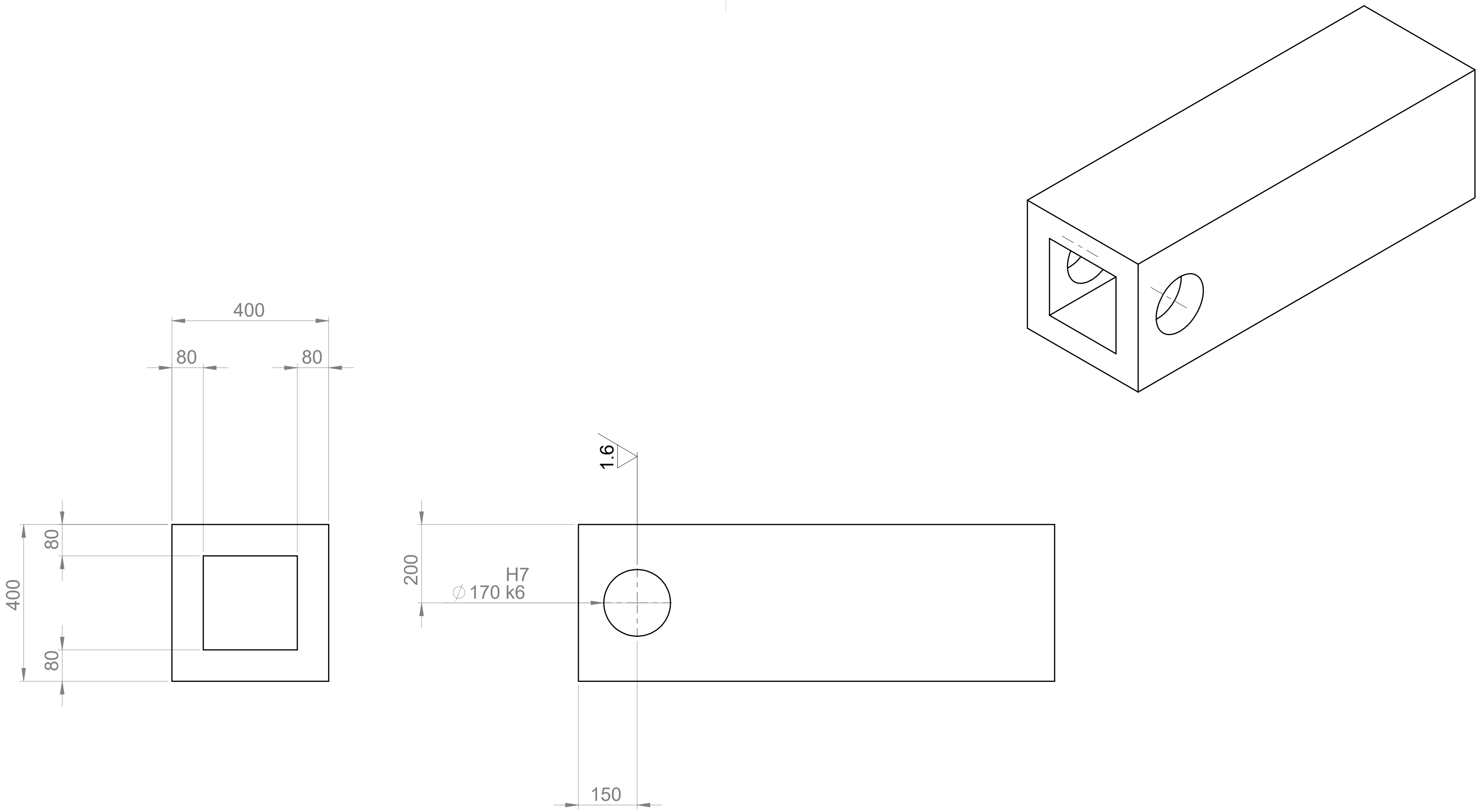




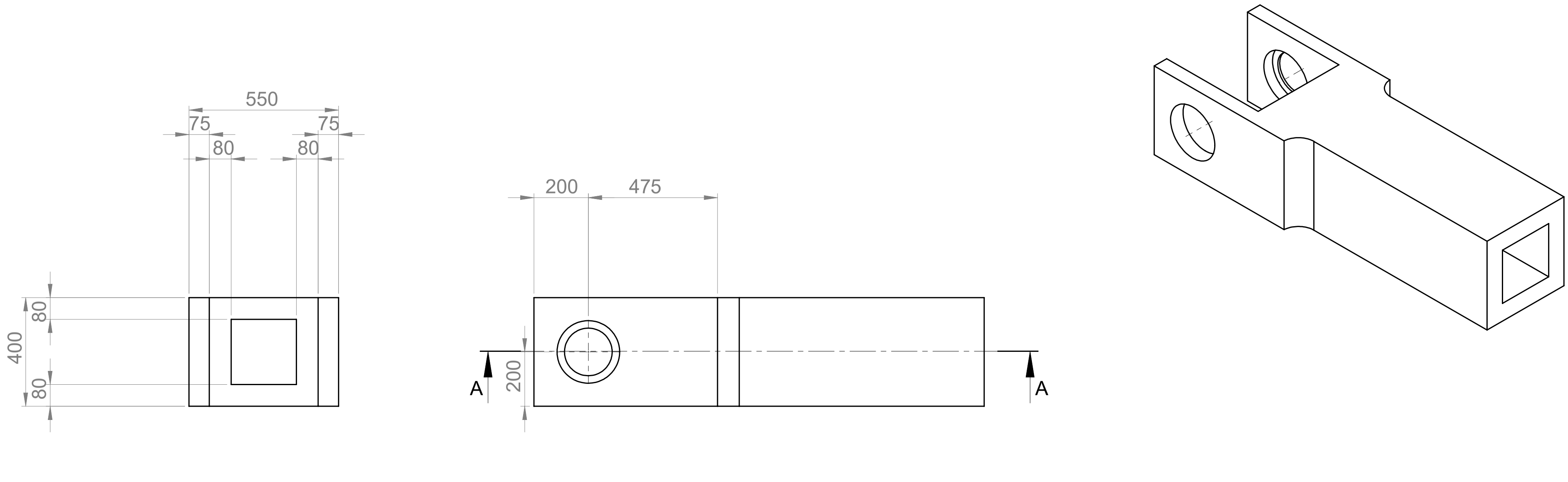


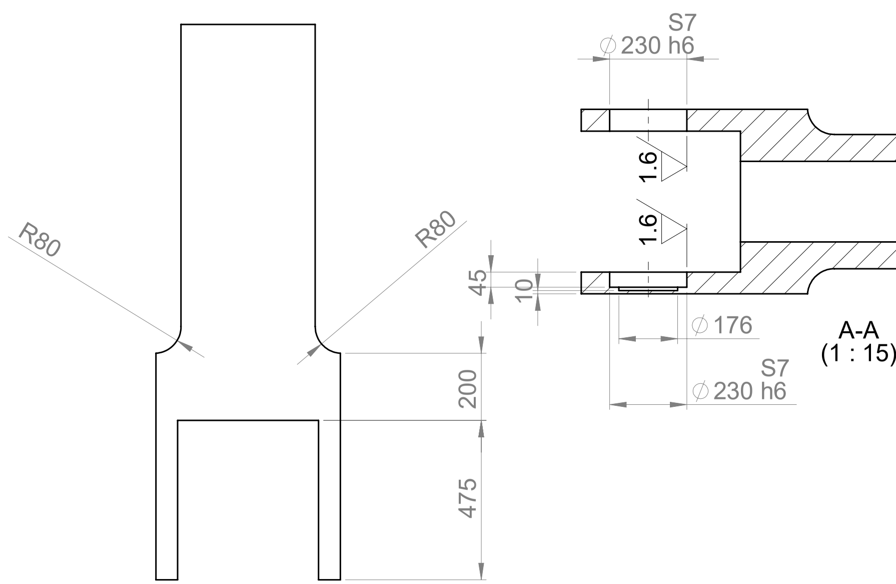
POS.	QTY.	NAME	MATERIAL	NORM / DIMENSION	REMARK
4	1	Beam middle			
3	2	23934			
2	1	Beam side with socket			
1	1	Shaft			

PROJECTION METHOD	UNLESS STATED OTHERWISE: TOLERANCES ± 0,5 MM	DRAWN: T.L. van Rooijen	DATE: 12-6-2019
		CHECKED: W.L. de Sain	SCALE: 1:20
MATERIAL: -- --	SURFACE TREATMENT: --	TITLE: Assembly Boom Hinge	REV.: 01
UNIVERSITY OF TWENTE.		DRAWING NO.: 1	A3
FACULTY OF ENGINEERING		FILE / PART NAME: Assembly Hinge	
		DIMENSIONS IN MILLIMETERS	SHEET 1 OF 1

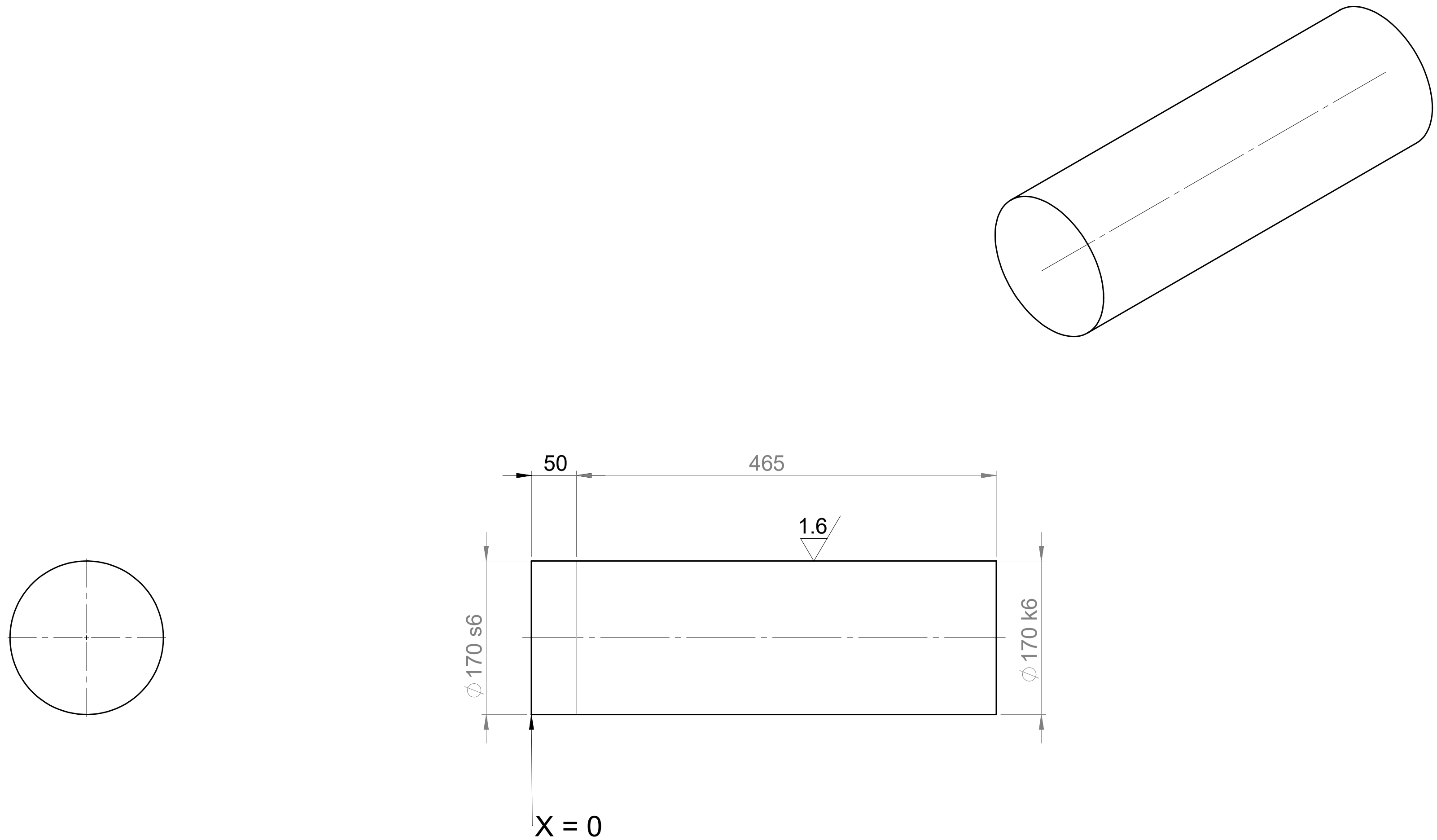


PROJECTION METHOD	UNLESS STATED OTHERWISE: TOLERANCES ± 0,5 MM	DRAWN	T.L. van Rooijen	DATE	12-6-2019
		CHECKED	W.L. de Sain	SCALE	1:10
MATERIAL AISI 9255 --	SURFACE TREATMENT Machined	TITLE Middle Beam Boom Hinge		REV. 01	
		DRAWING NO. 1.1			
UNIVERSITY OF TWENTE.		FILE / PART NAME Beam middle		A3	
FACULTY OF ENGINEERING		DIMENSIONS IN MILLIMETERS		SHEET 1 OF 1	

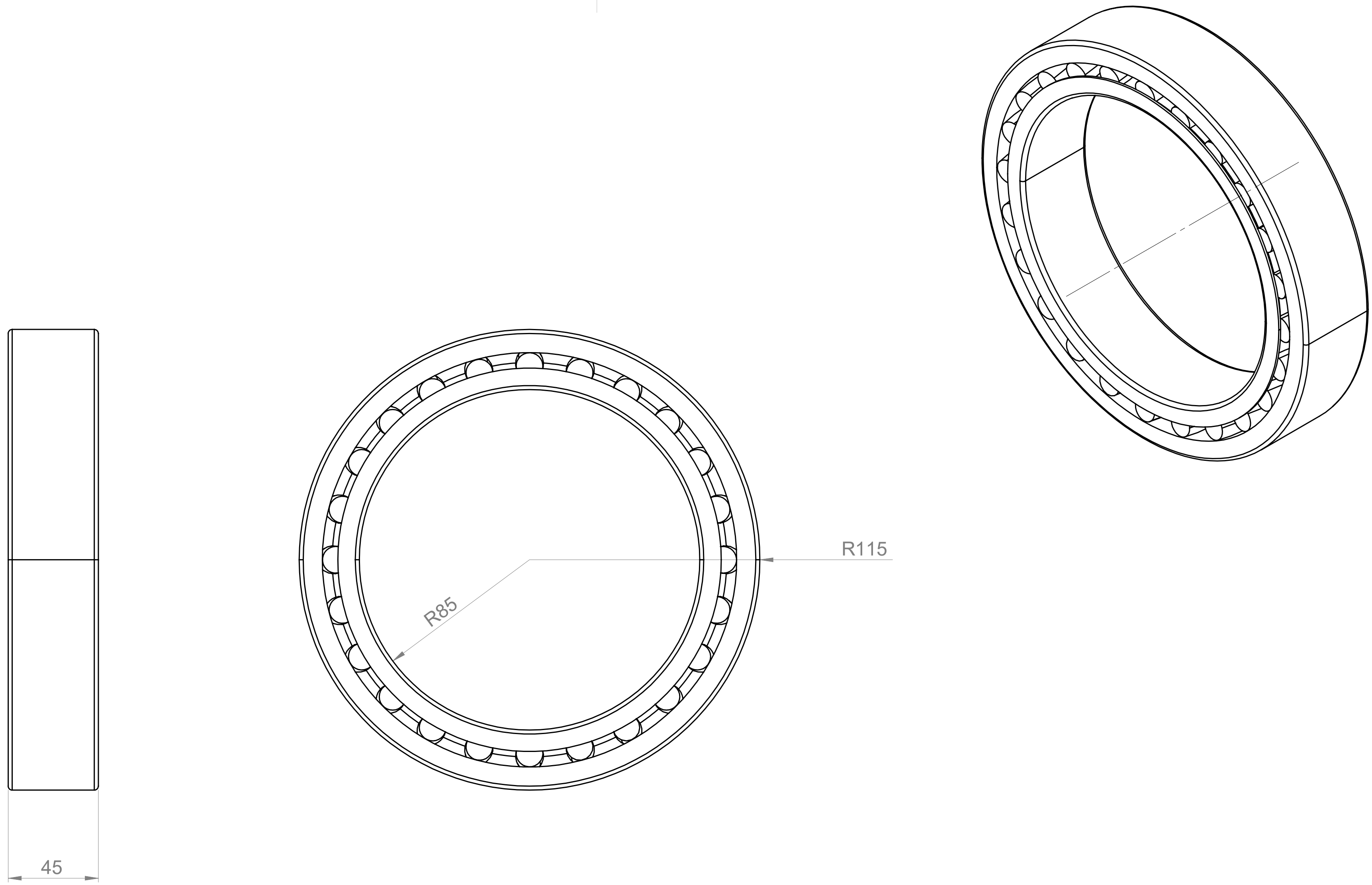




PROJECTION METHOD	UNLESS STATED OTHERWISE: TOLERANCES ± 0,5 MM	DRAWN	T.L. van Rooijen	DATE	12-6-2019	
		CHECKED	W.L. de Sain	SCALE	1:20	
MATERIAL AISI 9255 --	SURFACE TREATMENT --	TITLE Beam with Socket Boom Hinge				REV. 01
		DRAWING NO. 1.2				
UNIVERSITY OF TWENTE.		FILE / PART NAME Beam side with socket				A3
FACULTY OF ENGINEERING		DIMENSIONS IN MILLIMETERS		SHEET 1 OF 1		



PROJECTION METHOD	UNLESS STATED OTHERWISE: TOLERANCES ± 0,5 MM	DRAWN	T.L. van Rooijen	DATE	12-6-2019
		CHECKED	W.L. de Sain	SCALE	1:5
MATERIAL AISI 9255 --	SURFACE TREATMENT Machined	TITLE Hinge Shaft		REV. 01	
UNIVERSITY OF TWENTE.		DRAWING NO. 1.3			
		FILE / PART NAME Shaft		A3	
FACULTY OF ENGINEERING		DIMENSIONS IN MILLIMETERS		SHEET 1 OF 1	



Standard Part from NTN

PROJECTION METHOD	UNLESS STATED OTHERWISE: TOLERANCES ± 0,5 MM	DRAWN	T.L. van Rooijen	DATE	12-6-2019
		CHECKED	W.L. de Sain	SCALE	1:5
MATERIAL -- --	SURFACE TREATMENT --	TITLE NTN 23934 Spherical Bearing			REV. 01
		DRAWING NO. 1.4			
UNIVERSITY OF TWENTE.		FILE / PART NAME Bearing NTN			A3
FACULTY OF ENGINEERING		DIMENSIONS IN MILLIMETERS		SHEET 1 OF 1	

10.9 Final Frame Drawings

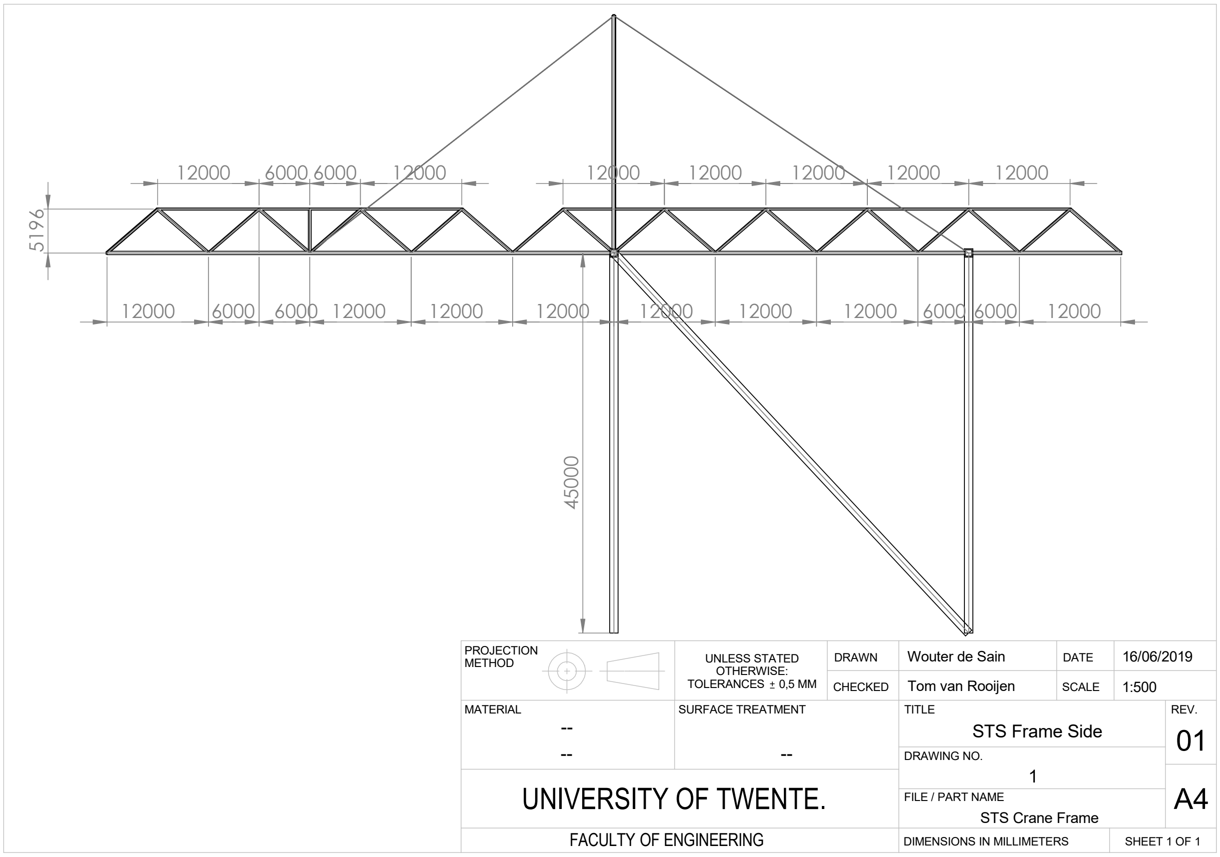
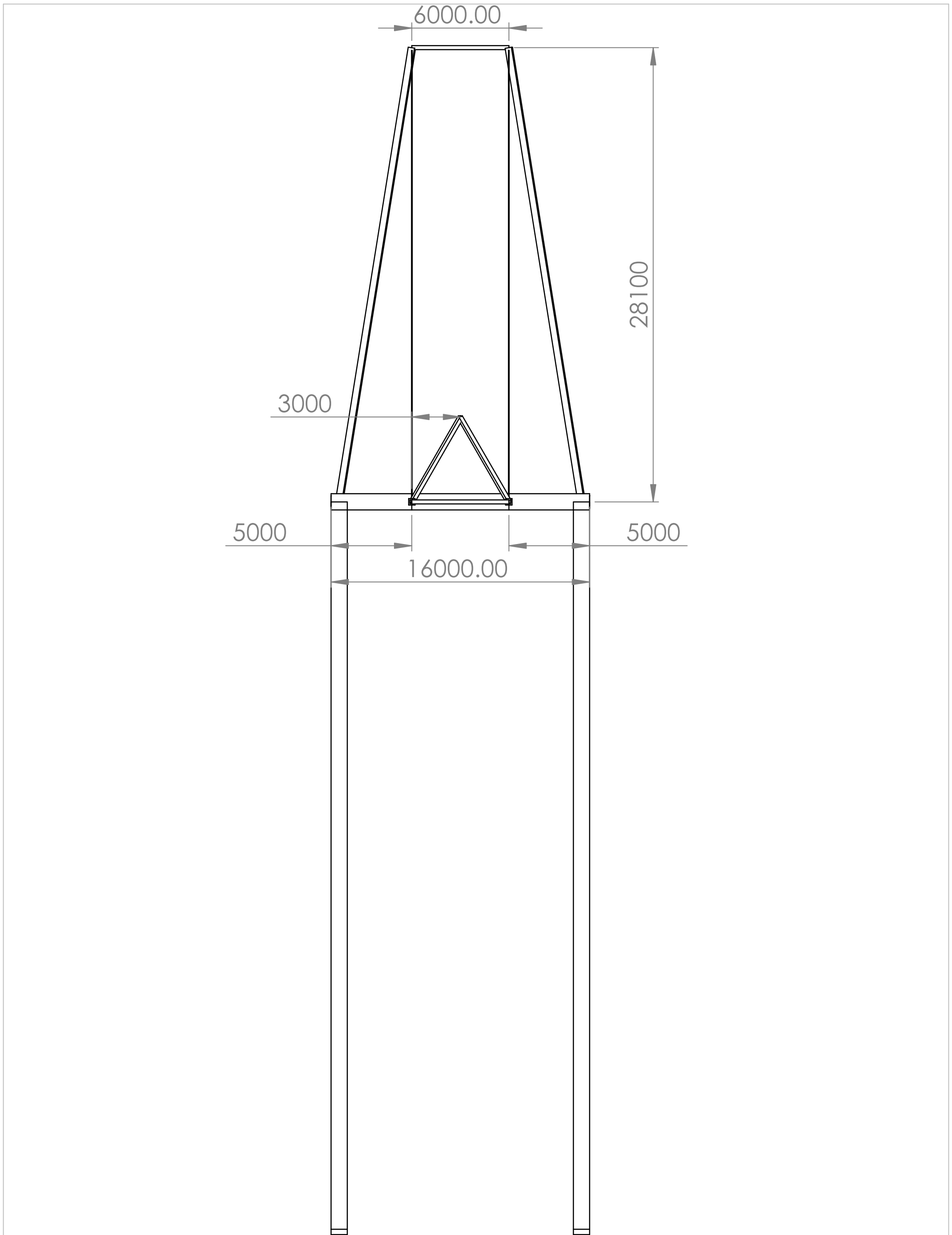
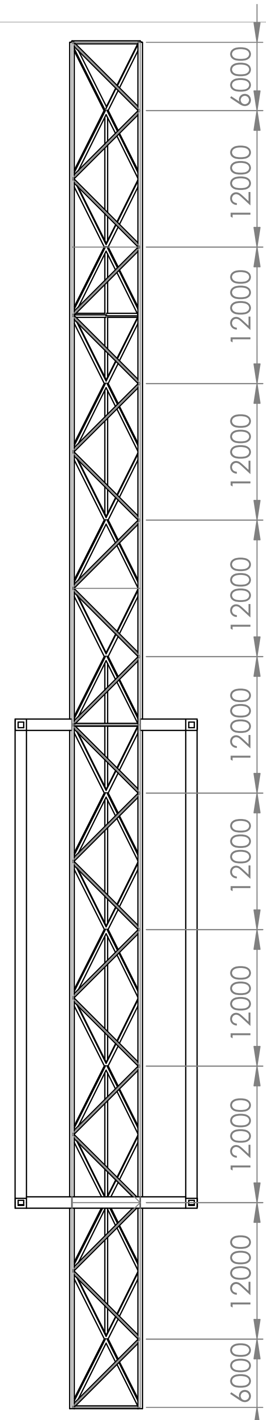


Figure 76: Side view of the STS Crane



PROJECTION METHOD	UNLESS STATED OTHERWISE: TOLERANCES ± 0,5 MM	DRAWN	Wouter de Sain	DATE	16/06/2019
		CHECKED	Tom van Rooijen	SCALE	1:325
MATERIAL -- --	SURFACE TREATMENT --	TITLE STS Crane Frame Front			REV. 01
		DRAWING NO. 2			
UNIVERSITY OF TWENTE.		FILE / PART NAME STS Crane Frame Front			A4
FACULTY OF ENGINEERING		DIMENSIONS IN MILLIMETERS			SHEET 1 OF 1

Figure 77: Front view of the STS Crane



PROJECTION METHOD	UNLESS STATED OTHERWISE: TOLERANCES ± 0,5 MM	DRAWN	Wouter de Sain	DATE	16/06/2019
		CHECKED	Tom van Rooijen	SCALE	1:525
MATERIAL -- --	SURFACE TREATMENT --	TITLE STS Crane Frame Bottom		REV. 01	
		DRAWING NO. 4			
UNIVERSITY OF TWENTE.		FILE / PART NAME STS Crane Frame		A4	
FACULTY OF ENGINEERING		DIMENSIONS IN MILLIMETERS		SHEET 1 OF 1	

Figure 78: Bottom view of the STS Crane