University of Twente

Group 5

Design and Mechanics

Authors:
Thijs Tijman op Smeijers (s2143658)
Luis Perez Alvarez (s2029413)
Ties Loohuis (s2184486)
Luuk Wijnant (s2178370)
Jasper Wijnen (s2083361)
Ivan Belev (s2118661)
Jeffrey Alvarez Romero (s2032147)
Jesper Kussendrager (s2164949)

Tutor:
Björn Nijhuis

November 4, 2021

Contents

1 Introduction

Ship-to-shore (STS) container gantry cranes are widely used in harbours all around the world. They have major advantages over other types of harbour cranes because they have the option to lift their boom to make large vessels pass and to move themselves across the harbours shoreline. The main task of the crane roughly is as follows: grabbing a container from a vessel, hoisting it vertically, moving it horizontally and placing it onto a truck, after which the next container can be grabbed, et cetera.

Several companies made a substantial amount of progress designing safe and efficient cranes. However, accidents still occur, mostly due to operational faults committed by the cranes operators. There is still progress to be made regarding designing STS cranes in such a manner that these faults are prevented by designing intelligent mechanisms, and in a way that the impact of these faults diminishes if they would ever occur. The purpose of this study is therefore to propose a safe and efficient STS high profile gantry crane.

Firstly, different concepts for five different mechanisms of an STS crane will be created. These mechanisms include the hoisting, trolley, boom structure, boom retraction and boom hinge mechanism. From these concepts, a final design will be compiled and elaborated. Then, suitable materials for different components of the crane will be chosen. After that, an optimal frame and boom will be designed, calculations regarding stresses in some components of the crane will be performed and machine elements for different components will be chosen.

2 Problem Definition

A ship to shore (STS) high profile gantry crane was designed regarding the following 6 requirements [1]:

Requirement 1
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).

Requirement 2
The crane must meet the following dimensional requirements:

- Vessel dimensions: a length of 300m, a width of 35m and a height of 40m
- Container dimensions: a length of 12m, a width of 2.4m and a height of 2.6m
- Extra dimensions: a maximum outreach (B) of 45m and a maximum lifting height (D) of 40m (Figure 1)

Requirement 3
The crane must meet the operational requirements.

- Hoisting speed is 90-120 m/min.
- Trolley speed is 180-210 m/min.
- Boom lifting mechanism should be able to raise 60-90 degrees relative to the horizon.
- Maximum deflection at the boom tip in three directions: 5mm perpendicular to gantry rails, 60 mm parallel to gantry rails and 150mm vertical

Requirement 4
The crane must withstand the loads to which it is subjected during operation. Dynamic loads are neglected.

- Maximum weight of a container including spreader is 50 tons.
- Maximum wind force is 240 N/m^2.

Requirement 5
The crane design must be operational and suitable for the environment conditions (temperature and wind speed).

- Ambient temperature during operation: -40°C to 50°C

Requirement 6
Boom, trolley and hoist mechanisms must be fully designed and configured based on the given load cases.

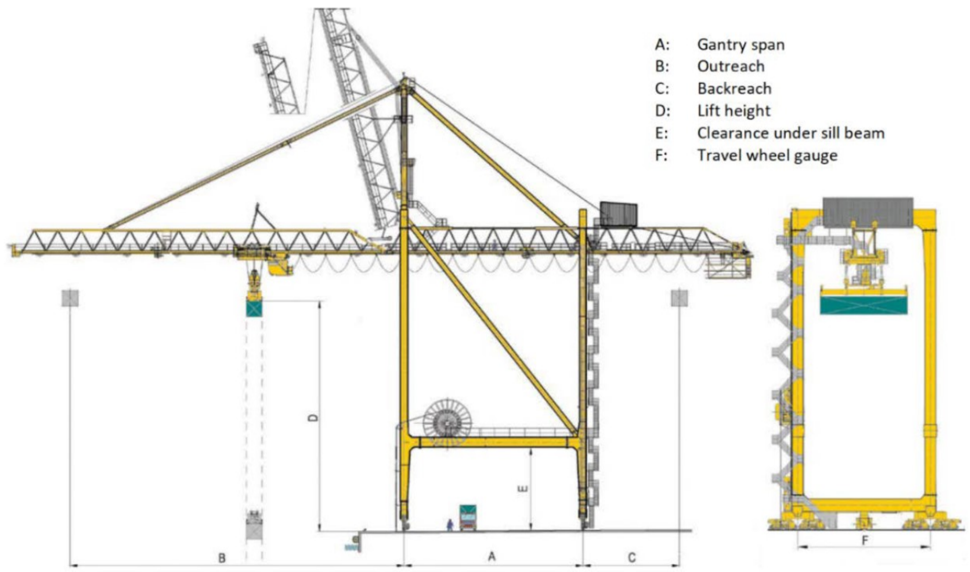
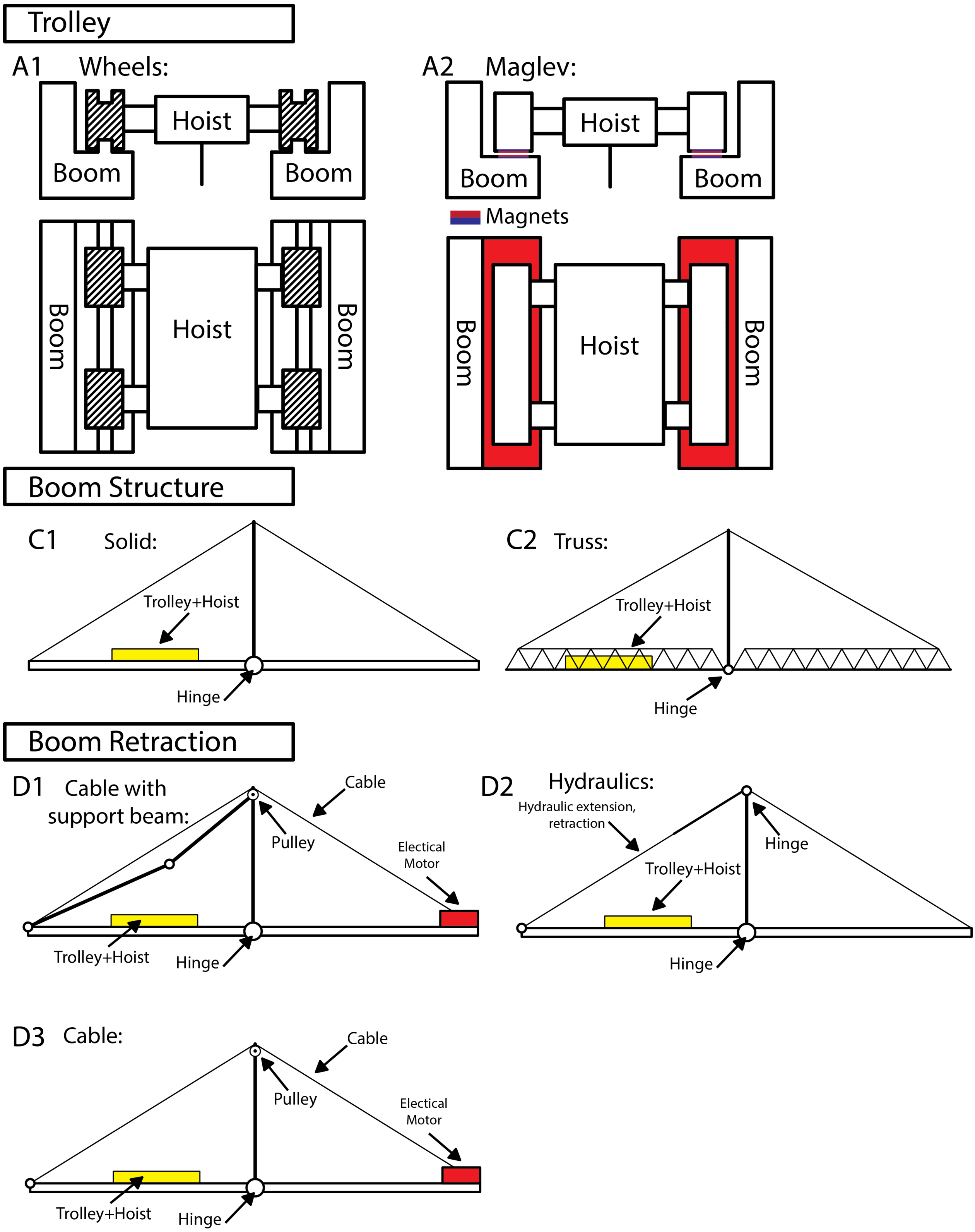


Figure 1: STS crane [1]

3 Conceptual Design

3.1 Conceptual Design Sections

Possible concepts for the boom, trolley and hoist mechanisms were made and accumulated into an overview in Figure 2. Below, all of the conceptual designs are explained into detail, with given pros and cons.



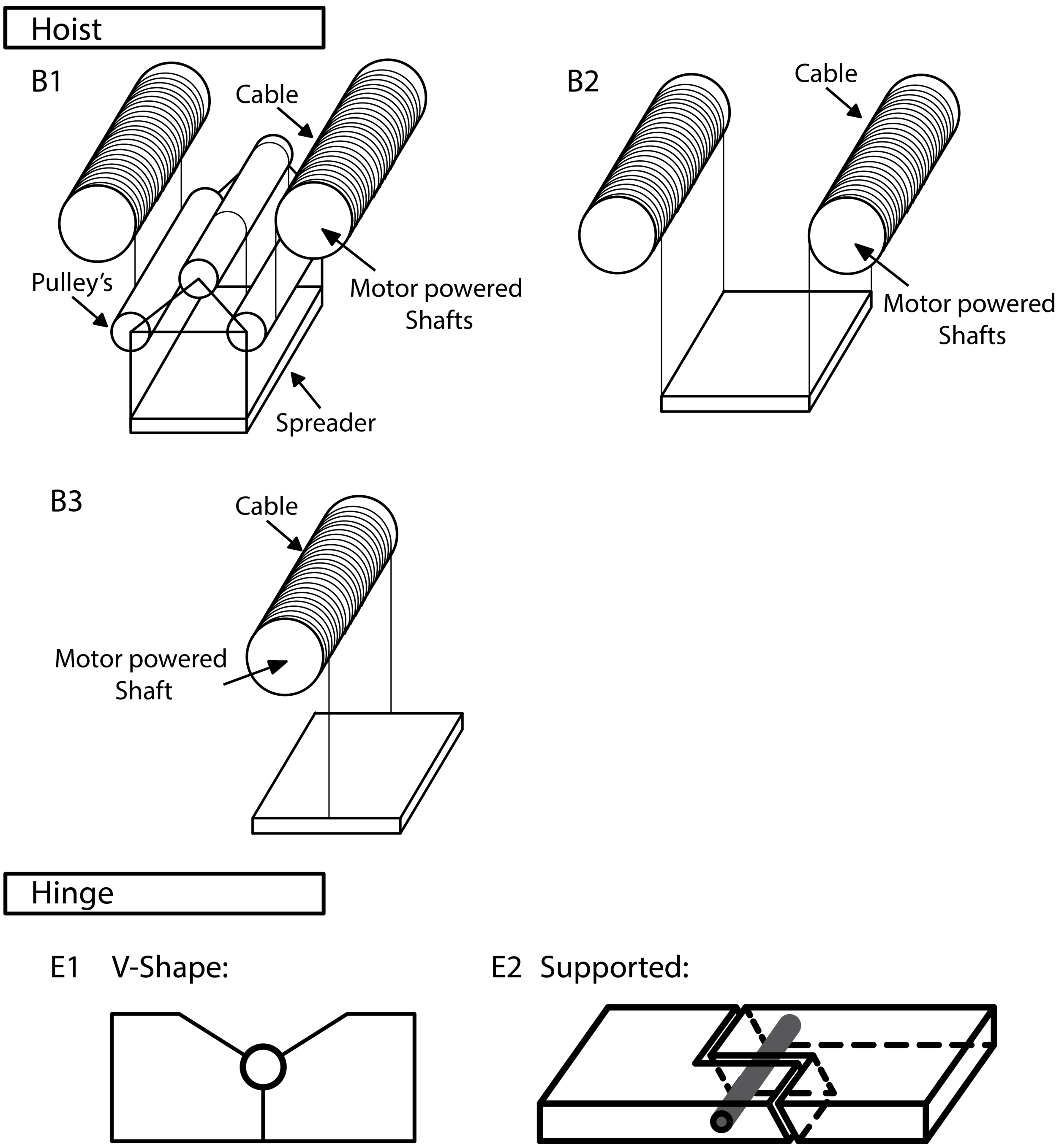


Figure 2: Overview of the concepts

3.1.1 Trolley Mechanism

A trolley is a device which can travel along the length of the boom while carrying the weight of spreader and container. In addition to this, the trolley could also be used to house the hoisting mechanism. There are few alternatives, and each will be explained. For the trolley the following possibilities were considered:

- Wheels
- Maglev

A1. Wheels

The first concept, which can be seen in figure 2 at A1, is for moving the trolley over the boom is with wheels. Wheels which are made out of steel or out of rubber are considered. Calculating the force needed to move the trolley is done using the following equations.

$$F = mg \tag{1}$$

As this force is the same as the normal force, F can be replaced by the normal force, F_n.

$$F_{friction} = \mu_w F_n \tag{2}$$

For μ_w there are two different values, there is a Static friction coefficient and a Dynamic coefficient of friction [2]. As long as the object is not moving, the static coefficient of friction should be used. The moment the object starts to move, the dynamic coefficient of friction should be used. It is decided to use the static friction coefficient as this value is in both cases higher, and therefore this is the maximum force the engine must deliver to be able to move the trolley. Only the maximum weight of the container is taken in the calculations. For the steel wheels the force needed to move the trolley is 362.97 kN, and for the rubber wheels 588.60 kN needed. Lastly the power which is needed can also be calculated.

$$P = Fv \tag{3}$$

In this equation, F is equal to $F_{friction}$. As the maximum velocity is 3.5 m/s, the power needed for the steel wheels is 1,270.40 kW and for the rubber wheels is 2,060.10 kW. These powers are calculated using a static coefficient of friction. This indicates the maximum possible power requirement, which is for startup. As will be seen later during the detailed design of the trolley, this coefficient is used during the shaft analysis, as during the startup the shaft will endure higher forces than when rolling. For rolling movement a rolling coefficient is used.

When looking at wheels there are more options within this. There are different methods to power the trolley. This could be done with a wire/chain which travels around the boom allowing the trolley to be pulled to the back of the crane and return to the front. When using this method, there should also be a load somewhere with a pulley on it, making sure there is no tension in the cable when the boom is lifted up. This weight with the pulley can be seen in figure 3, it is noted at number 1. The other option is the use of an electrical motor on the trolley itself directly driving the wheels.

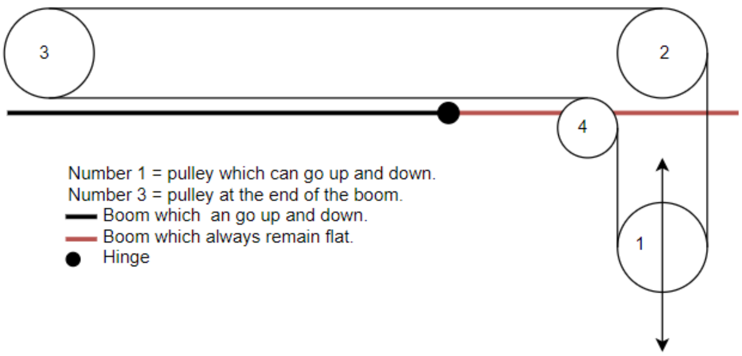


Figure 3: Trolley pulley system used when the boom is lifted

Furthermore, the design of the wheels is considered. A track is required to ensure that the trolley does not sway or slide perpendicular to the boom's length. The two main options for this are either the wheel has an H-shape traveling on a ridge on the tracks, which fits into the groove of the wheels. The alternative is the opposite, where the tracks have a groove and the wheels slot into them. The issue with the latter is having a small groove will increase the stress at that point, due to the smaller area. Having H-shaped wheels will allow for a wider wheel and therefore the weight of the trolley and everything that it's carrying is distributed over a larger area, while still having a relatively small groove.

Also when the boom will be pulled up, the trolley should be at the part of the boom which will not be lifted. This can be done with sensors, so the boom only can go up if the sensors detect that the trolley is not at the liftable part of the boom.

A2. Maglev

The concept of maglev was considered as due to its friction reducing property. It can be seen in figure2 at A2. In short, magnetic levitation is a method by which an object is moving over another object with no support other than magnetic fields [3]. As there is no physical contact between the boom and the trolley, the required force to move the trolley forwards in reduced. However, the friction force acting on the trolley, with wheels for example, is a small additional energy requirement in respect to the energy needed to power a maglev trolley. The following equations are used to calculate the required power input.

$$F = BLq \tag{4}$$

As this is for a constant amount of time, q (charge) can be replaced with I (current). In equation 5, I is made the subject of the formula.

$$I = \frac{F}{BL} \tag{5}$$

$$P = IV \tag{6}$$

Lastly to find the power required the voltage and formula 6 is needed. For this a low voltage value of 120 V is used but even then the power requirement is 29.4 MW. Even this however is a very large power requirement, meaning this method is very wasteful and is not very beneficial for the design.

3.1.2 Hoisting Mechanism

For the hoisting mechanism, different concepts with the use of pulleys and shafts are considered. Also, a comparison between different locations for the motor proving energy for the hoisting mechanism is performed. The following two concepts are considered:

- Two shafts
- Two shafts, three pulleys
- One shaft

B1. Two shafts, three pulleys

The concept using two shafts and three pulleys can be seen in figure2 at B1. It contains two shafts, three pulleys, four support plates and two different cables. At the bottom of the design, the load is distributed over the four support plates. When looking at the pulley system, for each pulley, this load is distributed over a cable going to one of the 2 shafts and a cable going to the middle pulley. This implies that one cable should be able to hold 0.125 times the total load, leading to a load of 6.25 tons on each cable.

The necessary power capacity of the motor can be calculated using equation 3. F can be calculated using equation 1, in which m is equal to 50,000 kg and g is equal to 9.81 m/s^2, leading to a gravitational force of 490,500 N. Now using equation 3 again, with the velocity v equal to a maximum of 120 m/min (= 2 m/s), a necessary power capacity P of 981.00 kW.

A major pro of distributing the load is the fact that it decreases the necessary strength of the cables. This distributes the load even more times than the two shafts concept, and if necessary, more pulleys could be added to further decrease the load on each cable and thereby decreasing the load the motor needs to hoist. A con of this concept over the two shafts concept is the fact that more parts are necessary, increasing the production costs and also the friction (more pulleys). Another con would be that the hoisting motor will be hoisting the load slower because of the 3 pulleys added.

B2. Two shafts

The second concept, which is well visible in figure2 at B2, contains two shafts, both connected to the same motor (which is left out of the figure) by gears and two different cables which are wounded around the shafts. The load is spread over the two cables, which both also split up the load. The spreader is directly connected to each of the four cables in this concept.

A major pro of spreading the load over 4 cables and 2 shafts is the fact that it decreases the necessary strength of the cables. For the ease, it will be assumed there are 4 cables holding the load. Thus, one cable should be able to hold 0.25 times the total load. As stated in problem definition, this total load equal 50 tons, leading to a load of 12.5 tons in each of the four cables. The necessary capacity of the motor in this design is 981.00 kW as well, since the total load is the same.

B3. One shaft

The last hoisting concept, visible in figure2 at B3, contains just one motor powered shaft with a cable wounded around it, splitting up the load of the spreader in two. For ease, it will be assumed there are 2 different cables holding the load, meaning one cable should hold 0.5 times the total load. This leads to a load of 25 tons in each of the cables The necessary capacity of the motor in this design is the same as for the other two concepts, namely 981.00 kW.

Location of the motor

The hoisting mechanism will need a motor to be able to hoist. This motor can be either AC or DC. The location of the motor is variable in each of the concepts. It can be either inside the vertical moving part or in the stationary part of the hoisting mechanism, which is directly on the trolley itself or on the back of the boom.

The main pro of having the motor in the stationary part is the fact that the weight of the motor does not have the to be hoisted. If the motor is placed on the trolley, it only has to be moved in horizontal direction over the trolley mechanism and if it is placed on the back of the boom, it can be stationary. When the motor would be placed on the back of the boom, the cables of the hoisting mechanism should be guided over the boom and should retrace when the trolley is moving back and expand when it is moving forward. This would only be possible for the one shaft concept.

It both cases, the cables of the hoisting mechanism should move as fast as the trolley moves. This can be done by letting the motor of the hoisting mechanism operate at the same speed as the the trolley mechanism. A significant advantage of this design is the fact that the motor can be larger than for the other two designs, meaning e.g. a more powerful motor can be used. A con of this design would be that the motor should be able to operate at different speed, meaning more gears are necessary, increasing the friction and thereby lowering the efficiency. More friction is also generated in the cables because they are stretched out over the boom and thereby having contact with guiders. Another con would be the fact that the motor needs to provide high torque since the arm (the boom) is considerably long.

3.1.3 Boom Structure

In the case of the boom, two different concepts were considered, the first one a single beam structure or more accurately called single girder mechanism and the second one a truss structure. In order to choose the best concept, there are two important constraints to take into account. The outreach of the boom in this case a maximum of 40 meters and the maximum load of a container to be lifted, a maximum of 50 tons. Considering this, the main differences of both concepts will be mentioned below.

Solid beam (C1) vs Truss system (C2)

- The first and main difference between these structures is that beams will support flexural loads, which means shearing and bending forces; while the forces in a truss structure will be supported by members which will experience either tension or compression.
- There are outreach limitations for single girder structures to safely lift heavy loads while; on the other hand, the unique properties of a triangular shaped object allow trusses to cover longer distances more reliably.
- Truss cranes may be designed with single or double bridge beams. Since the trusses carry the load, the bridge beam rail size can be minimized. In many cases the hoist high hook elevation is above the bottom chord of the trusses.
- Truss structures offer enhanced stability and strength and are lightweight compared to the beam one, due to the reduction in the use of material. It is important to point out that the boom will experience not only vertical but also lateral loads due to wind conditions. To withstand the lateral loads, cross beams with diagonal bracing is often employed, ensuring the necessary stability.
- In economic terms, truss structures use less material which mean saving money but it is needed to consider that the production cost would increase due to the manufacturing process like the welding of

all of the joints, although nowadays automated welding processes exist, the possibility of making the process not only more expensive but more time consuming remains for example when applying a heat treatment to correct unexpected deficiencies in the crystal structure of the material. On the other hand, more material is required for the solid structure which also increases the mass of it.

- Single Beam structures in conclusion do have a limit in terms of capacity and span and for this reason is not commonly used in STS cranes for the huge loads to lift. What is more common is the use of twin girder boom with cross bracing to give it lateral stability.

3.1.4 Boom Retraction Mechanism

Three options have been designed for the retraction of the boom:

- Cables with a support beam
- Hydraulic System
- Cables

D1. Cables with support beam

In the first design, which can be seen in figure2 at D1, there is a beam with a hinge in the middle which protracts when the boom is released and holds it's load during work, relieving the rope of the tensile stresses. When the boom is being lifted the ropes and the electric motor still do all the work. An important detail of this design option is that the position of the rope and the top pulley need to be adjusted in a way that the force in the rope is not parallel to the retractable beam and it is possible for it to fold. A loose hinge will be used in the middle of the beam, in order for it to easily bend when the load is taken by the cable. With this design the boom will be held up entirely by the ropes and it will be locked in the upper position by the motor. The shaft of the mo-
tor will be locked using a drum break, because it does not need a continuous supply of energy to hold the shaft.

Pros

- With the addition of the beam there is no constant strain on the rope during work, when the loads are the highest.
- The beam is more sturdy than the rope and it will hold the boom more steadily.
- The beam is quite short when compared to the ropes, so thermal expansion will not have such a critical effect on it.

Cons

- Using the beam makes the system more complicated, with more moving parts, therefore more can go wrong.
- The more complex system will certainly be more expensive to produce.
- Due to the added weight of the beam, when folding the boom there will be a greater load on the ropes and the motor.

D2. Hydraulic System

The second boom retraction concept is the idea of using hydraulics. The hydraulic system is pulls from the left point of the boom with the purpose of lifting the boom and making a rotational movement around the central hinge. The hydraulic system consists in one hollow cylindrical container with an internal cylinder with the form of a piston, this internal cylinder will move outside or inside the hollow cylinder depending in which side there is the higher pressure. Hydraulic systems use an in-compressible liquid to generate the pressure and therefore to move the internal piston, oil is mostly used for this purpose. The hydraulic system is attached in the top of the crane and when it is activated, it will lift the boom. However, in order to have a clear overview of the forces acting on the hydraulic system some assumptions were made, they can be seen in Table 1, finding that the maximum necessary diameter is 0.72m which is inside realistic parameters.

First of all, the gravitational force was calculated using the mass and the gravity, leading to a gravitational force of **1,175 kN**. After a summation of forces and taking into account that the load is equally distributed through the boom, the force acts in the middle. The force of the left point of the boom is equal to a half of the load giving **588 kN**. Then, to calculate the force through the hydraulic extension, it is necessary to calculate the angle using the width and the height. This gives an angle of 23.96°. To calculate this force, the force on A (left point) must be divided by the sine of the angle, which is equal to **1,450 kN**.

Secondly, the external hydraulic diameter was calculated in order to compare the concepts and the validity of this one. Then with the use of equation 7, the bigger diameter can be calculated using the assumptions of table 1. After using the equation $d_2 = \sqrt{\frac{4 \cdot 1,450,000}{7\pi} + 500^2}$, the approximate diameter is **0.72 mm**.

$$F_1 = P_1 \pi \frac{d_2^2 - d_1^2}{4} \tag{7}$$

Table 1: Assumptions for hydraulic system

	Pulling
Weight boom	120 tones
Max. Force	1450kN
Internal pressure 1	7 bars
Internal pressure 2	0 bars
Diameter internal cylinder	0.5m
Diameter hollow cylinder	max. 0.72m
Boom length	45m
Vertical beam	20m
Hydraulic length	50m

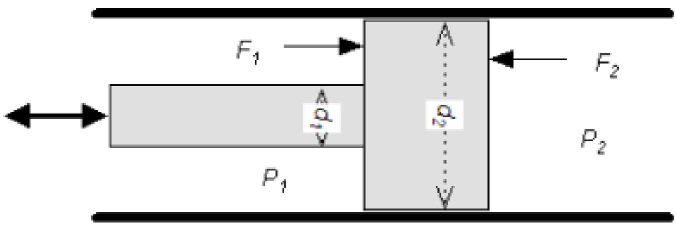


Figure 4: Lifting hydraulic system

<u>Pros</u>

- Has an excellent accuracy because the forces can be controlled very well. Therefore, it is capable of providing a constant force regardless of speed changes
- Brief calculations show that taking into account the forces to carry, the maximum hydraulics's diameter will be 0.80 meters which is not unrealistic.
- Multiplication of force: a fluid power system can multiply forces simply and efficiently from a fraction of a pound, to several hundred tons of output.

<u>Cons</u>

- Oil can leak due to the high pressures, this can affect the safety of the working environment
- The distance of hydraulic cylinder has to be around 50 meters which is not safe if the necessary precision for a hydraulic system is taken into account
- Because of the large length and its weight the hydraulic will tend to create bending forces in it, this could damage the retraction system itself and lead to collapse of the boom.

D3. Cables

The third retraction mechanism seen in figure2 at D3 is just a metal rope attached to the front of the boom, which then goes over a pulley on top of the crane structure and ends up connected with an electrical motor at the rear side of the crane. When the boom is being lifted, it will be held up by locking the cable with the motor. To prevent the shaft of the motor from moving a drum break will be used, because it does not require a constant power input in order to hold the shaft still for prolonged periods of time.

Pros

- Only using a rope for holding the weight of the boom is a very simple mechanism with without many moving parts.
- Metal ropes are already available in a wide range, so fewer custom parts will be needed.
- As a result of the above mentioned characteristics, the use of ropes is a quite price efficient option.

Cons

- If only a rope is used for bearing the weight of the boom there will be constant strain on the rope, which might lead to premature failure
- When the boom is retracted the rope needs to be properly locked, to prevent any movement, which might prove difficult.
- The rope can be subject to thermal expansion, which may lead to the failure of the whole construction.

3.1.5 Boom Hinge Mechanism

The hinge mechanism is a vital addition to the overall design as it ensures the boom is able to be lifted up by the mechanisms mentioned above. For the hinge mechanism, the following two concepts are considered:

- Supported
- V-shaped

E1. Supported

The supported hinge concept can be seen in figure 2 at E1 In this concept the hinge is naturally locked when the boom is horizontal. The contact area is quite large, distributing the force exerted on the beam, which results in less likeliness off failure. However, an additional locking mechanism would be preferable to decrease the load on the boom and cables/beams holding the boom.

E2. V-shaped

As can be seen in figure 2 at E2 the v-shaped mechanism has the hinge placed in the middle of of the boom structure, with a flat bottom part which naturally when the boom is released and a V-shaped gap on the top to allow for retraction. Although the design of the hinge automatically locks the boom in place when released and prevents further downward movement the forces on the hinge are too big and additional support from cables or an extra beam is required to hold the full weight of the structure.

3.2 Choosing a Final Design

Table 2: Names of each concept/subconcept in the weighing table

Trolley		
A1	Wheels	
	A1.1	Motor directly on trolley wheels
	A1.2	Motor at back of the boom
A2	Maglev	
Hoist		
B1	Two shafts, three pulleys	
B2	Two shafts	
B3	One Shaft	
	B3.1	Motor directly on trolley
	B3.2	Motor at back of the boom
Boom Structure		
C1	Solid beam	
C2	Truss system	
Boom Retraction		
D1	Cables with support beam	
D2	Hydraulics	
D3	Cables	
Boom Hinge		
E1	Supported	
E2	V-shaped	

Weighting table

Table 3: Weighing Table

		Trolley			**Hoist**				**Boom Structure**		**Boom Retraction**			**Boom Hinge**	
		A1		**A2**	**B1**	**B2**	**B3**		**C1**	**C2**	**D1**	**D2**	**D3**	**E1**	**E2**
	WF	**A1.1**	**A1.2**				**B3.1**	**B3.2**							
Force Distribution	**2**	2	3	5	4	3	2	3	1	4	4	3	3	4	3
Life Span	**2**	3	2	4	3	3	3	3	4	3	4	3	3	4	3
Manufacturability	**1**	4	3	3	4	4	4	4	4	3	3	2	4	3	3
Assemblability	**1**	4	2	3	3	4	4	4	3	3	3	3	4	3	3
Failure Damage	**3**	5	3	2	3	2	2	1	2	3	4	2	1	3	3
Power Usage	**2**	3	3	0	4	4	4	4	2	3	2	4	3	5	5
Cost	**2**	4	3	1	3	4	3	2	4	4	3	2	4	3	3
TOTAL		47	36	32	44	42	38	36	34	43	44	35	37	47	43
Best		A1.1			B1				C2		D1			E2	

3.2.1 Weights

Force Distribution

The force distribution in each of the designs is important to be considered. More force distribution means less stresses on the materials used, decreasing the chance of failing. It also means there are less stress variations, possibly decreasing the fatigue of the materials used.

<u>Trolley</u>
A1.1 gets 2 points for force distribution and A1.2 gets 3 points. Both designs use the same H-shaped wheels which ride along a track. These wheels distribute the gravitational force quite well. A1.2 has the motor of the trolley on the back of the boom, distributing the force over the crane even more than A1.1. A2 gets 5 points for force distribution, since magnetic levitation distributes all of the gravitational forces of the

trolley over its area on the track.

Hoist
B1 gets 4 points, since it uses pulleys along with two shafts, which distribute the gravitational force from the spreader and its load over the cables. B2 gets 3 points, because it just uses two shafts (drums) which distribute the load as well, but less than B1. B3.1 gets 2 points for force distribution and B3.2 gets 3 points. Both hoist designs are identical, only the place of the motor varies. Having the motor at the back of the boom in B3.2, instead of on the hoisting mechanism itself in B3.1, increases the force distribution over the crane.

Boom Structure
C1 gets 1 point for force distribution. Just a solid beam is used, which hardly distributes the gravitational forces acting on the boom. C2 gets 4 points, because a truss system is used, which distributes these forces acting on the boom very well.

Boom Retraction
4 points are gained by D1, because the cables holding the boom are supported by a beam when the boom is horizontal and operating, distributing the gravitational forces on this mechanism. D2 and D3 both get 3 points, since they both just use multiple cables (for D2 this are cables connected to hydraulics), which distribute the gravitational force over each other.

Boom Hinge
E1 gets 4 points for force distribution, because the contact area of the two parts connected by the hinge is large. E2 gets 3 points, because the contact area of these parts is quite large. However, the hinge is under great tension in this design, since it is not fully implemented in the boom but more as a separate part on top of the boom.

Life Span

The lifespan of the materials needs to be considered in order to avoid possible failures. Materials are dealing with e.g. corrosion, erosion and fatigue, which makes them more likely to fail. If certain parts are still in working services when their lifespan expired, they can cause accidents in the working area.

Trolley
A design with wheels would have a relatively long life span, however, A1.2 which uses a chain, would have a slightly lower life span due to the effect of wear and stretching on the chain compared to A1.1. On the other hand, A2, which uses maglev, would have a longer life span. This is due to the lack of moving parts.

Hoist
All hoisting designs operate in a very similar way and have the same power and weight requirements. The way that each operates is the same and therefore their life span would be the same, or at least very similar.

Boom Structure
The life span of B1 is quite a long one. This is because it is simply a solid beam. This means its less impacted by corrosion (simply made slightly thicker) and is one single part. B2 however is a truss system, which consists of multiple sections welded together, meaning more parts. The slender members are also thinner as each has to carry less weight. Because of this, their life span is similar but slightly longer for the single beam.

Boom Retraction
D2 and D3 have as expected a similar life span as they have a similar underlying structure, with cables. However, D1 has a better life span due to its rigid beam reducing tension in the cable, increasing cable life and therefore the overall life of the part.

Boom Hinge
Lastly the two designs for hinges rely on very similar parts so have a similar life span, however, due to the improved force distribution effect of E1, its life span is expected to be longer due to a smaller force being placed on the hinge pin which holds a lot together.

Manufacturability

The manufacturability can be explained as the ease of making a specific product. This concerns the shape the product will have as well as the material that the product will be made of, these two main aspects are important when determining the best manufacturing process that could be employed on making the product, thus also reducing manufacturing costs.

Trolley
For the design of the car, A1.1 obtained the best score, since the wheels of the car are connected directly to the engine, which would be similar to a small car with an electric motor. For what existing technology could serve as a guide to make this design. A1.2 consists of the same design with the only difference that the engine would not be connected directly to the wheels of the trolley but at the end of the boom which could increase the difficulty in the design of the trolley a little, taking into account that it would have to be moved by cables through the motor. For this reason it ties in the second place along with A2, the maglev mechanism. A2 would use two sets of magnets, one to repel the trolley and the other to move it. Although it consists of less movable parts and therefore fewer parts to manufacture, new alloys and manufacturing techniques in superconductors and cooling systems are necessary for the manufacture of these systems. As the trolley is a relatively small system that does not require high speeds compared to trains that use maglev technology, the difference in manufacturing is not much.

Hoist
For the hoisting system, all three designs have got the same score as the process for the manufacturing of the shafts is the same for all designs as well as the material these will be made of. The difference lies on the number of shafts needed that is not a significant problem for the manufacturing.

Boom Structure
With respect to the structure of the boom structure, the solid design C1 obtained 4 points unlike the design with trusses C2 that obtained 3 points. The manufacture of the beam is relatively easier compared to that of the trusses that need welding and bracing in all the joints, requiring critical attention in the manufacturing process and inspection process.

Boom Retraction
In the case of the retraction mechanisms of the pen; D3 obtained the highest score since it only involves the installation of cables that are already designed nowadays with good load capacities and elastomeric technology. D1 is the following on the list due to the included beams that must be carefully manufactured and connected in order to allow the good movement of these parts. This increases the difficulty and time in the process. The last design is D2, which is the hydraulic system that could mean a slightly more difficult process of design and manufacture, since the tolerances in both cylinders must be carefully calculated, especially because these would have a considerable length, these factors without mentioning the installation process makes it the least favourable mechanism in terms of manufacturing.

Boom Hinge
For the boom hinge mechanism, again E1 supported hinge and E2 v-shaped got the same score as there is not a big difference in the manufacturing of both. For both, the connection of both parts and the contact surfaces are practically similar but should be carefully designed in order to avoid unexpected stresses on neither the connection of both movable parts nor the beams themselves.

Assemblability

The assemblability can be defined as the ease of assembling each of the components of the crane into a whole fully functional crane. This mostly concerns the difficulty of assembling at high altitudes in the assembly of a crane. The more and the heavier the components at high altitude are, the harder it becomes to lift the parts to this high altitude and put all the parts together. This eventually result in a lower weight number.

Trolley
A1.1 gets 4 points and comes out on top of the 3 different designs. This is because the motors and the trolley itself can be put together on the ground and lifted onto the tracks, while A1.2 and A2 are spread out over the boom and need to be attached to the crane. Because of this A1.2 and A2 both have to be assembled at high altitudes. However in the case of A2 the components (mostly the magnetic plates) are lighter and smaller compared to the heavy engine block of A1.2.

Hoist
There is not that much difference in this case, because this part is not that heavy or complicated to assemble. Although A1 is still a bit more complicated than the other designs, which is why it gets a 3 and the others get a 4. Furthermore, we are not taking the assembly of the motor into account in A3.1 and A3.2, because the motor was already graded in the trolley section.

Boom Structure
If it is considered that both the structures are around the same weight.(In reality they are not, but both structures are extremely long and heavy, so although the truss will be lighter the difference will be minimal relative to the total weight of the boom.) Then it will be as complicated for both structures to lift the boom to such heights, so they will both get the same number.

Boom Retraction
D3 with cables is in this case the best option, because of the ease of assembly of cables. A support beam will make the assembly harder, because there are more and heavier components, so D1 gets a lower grade. For D2 there needs to be a relatively large hydraulic system for the design to work, so the same as D1, more and heavier components result in a 3 for D2 compared to a 4 for D3.

Boom Hinge
The boom hinge is considered to be small and light relative to the whole crane, so this should not be a huge problem for the assemblability. The difficulty of inserting the shaft and combining the components goes for both designs, so E1 and E2 both get a 3.

Failure Damage

The failure damage is a factor that needs to be considered due to safety implications. Safety is difficult to measure, however the failure damage is much more simple, this would be the damage caused in cost and amount of lives at risk. The higher each design weight is, the better it is. The scores are given in a worst case scenario.

Trolley
In a worse case scenario for A1.1 and A1.2, the motors could fail, however for this design the trolley would simply stay in its place. Despite being difficult to retrieve, this means the design doesn't contribute to any more damage. On the other hand, if a chain is used, there is a lower failure damage score due to the damage a snapping high tension chain can cause. A2 relies on electromagnets keeping the trolley off the surface of the boom. If this system fails, similar to an electric motor failing, the whole weight (up to 50 tons) would fall onto the boom. This would create damage to the boom and in a worse case scenario could cause the boom to fail.

Hoist
The first two designs for the hoist are quite similar in the method at which they hoist, but B1 has extra support from its 3 pulleys, this reduces tension in the cables, allowing more strain. This means the failure damage score is higher than that for the design without the pulleys, B2. B3.1 and B3.2 are simply a single shaft winding up and down. This has less support and the second version of the design has it located at the rear of the crane. Here if there is failure there is the added impact of the crate lowering if the trolley would continue to move to the centre of the boom. This decreases their score.

Boom Structure
Of course both designs of the boom can cause catastrophic damage if they were to fail. However, C1 is a solid beam, so has more concentrated weight and therefore the ability to create more damage than C2.

Boom Retraction
Boom retraction is an important feature of the crane and failure during this event would cause a lot of damage due to fall the boom would take before reaching the locking mechanism. The D1 is one with a support beam. This takes the load when the boom is in a horizontal position but if the cable were to snap then it would take a large section of the impact, reducing damage. D2 is the design with hydraulics. This takes the force off the cable making the likely hood of damage lower but if failure were to occur it would have no back up system and would create a large quantity of damage. But this design is not as bad as D3, which has only a cable which, if it would fail, would cause catastrophic failure.

Boom Hinge
The boom hinges are both very similar in the ability to prevent a large amount of damage if they were to fail. If complete failure of the hinge, the amount of failure would be dependant on the boom retraction mechanism, as the whole boom weight would be on that.

Power Usage

The power usage of each concept needs to be considered because the crane which is part of will be functional for the better part of the day and in some cases even over night. The power usage of the different part can significantly influence the cost over time and the ability of the crane to function overall.

Trolley
The trolley designs using wheels, A1.1 and A1.2, need quite little energy for their motorised wheels because the wheels support the weight by themselves without requiring any power input and they need power only to move forward and back, therefore they get 3 points. The other design, A2, requires constant power while working not only to move but to support the weight. The power it needs is more than 30 MW, which is why it earns 0 points.

Hoist
In all 4 hoist designs, the weight lifted by the motor(s) is the same and therefore they get the same amount of points. The power necessary for the hoisting is not that much, only about 500 kW, which is why it scores 4 points.

Boom Structure
The solid beam concept C1 scores 2 points and the truss system concept C2 scores 3 points for power usage. The solid beam will be quite heavy and therefore it will require more power to lift it. However, the solid beam can be made out of a certain shape, for example an I-shape, which does weigh as much as a complete solid boom, for example. The truss system will most likely be less heavy than a solid beam and therefore got 1 point more than the solid beam concept.

Boom Retraction
Both D1 and D3 get 3 points for power usage and D2 gets 4 points. D1 and D3 both use cables to lift the boom, but in D1, the cables need to lift an additional support beam (or more). This increases the power usage for D1 compared to D3. D2 uses hydraulics, which are able to lift high loads because the force in the hydraulics is spread evenly, decreasing the total amount of power usage.

Boom Hinge
Both hinge designs, E1 and E2, do not use additional power. This means that they both get the same amount of points, which is 5.

Cost

The costs of the different components is important to consider, especially for the customer. The higher the cost are for the crane the less appealing it is to buy such a crane. How the components are manufactured materials and where they are made of will be the biggest cost items.

Trolley
A1.1 gets 4 points for the cost and A1.2 gets 3 points. For A1.1 the motor is directly on the trolley so no cables are needed to pull the trolley over the beam, which reduces the cost. Assuming that the motors which are used are the same in both cases, A1.1 will be a bit cheaper. Therefore A1.1 has one point more than A1.2. A2 gets 1 point for the cost, since maglevs are really expensive in purchase cost and even more expensive in use phase because the power needed for operating is really high.

Hoist
B1 gets 3 points because there are besides two shafts also three pulleys needed, which will rise the costs. B2 gets 4 points, since only two shafts are needed. So it will be cheaper than B1 and therefore B2 has one point more. B2.1 gets 3 points for the cost and B2.2 gets 2 points. For B2.1 the motor is directly on the trolley so no cables are needed to pull the trolley over the beam, which reduces the cost. Assuming that the motors which are used are the same in both cases, B2.1 will be a bit cheaper. Therefore B2.1 has one point more than B2.2.

Boom Structure
Both C1 and C2 get 4 points for the cost. Since for both options, probably the same amount of material will be used.

Boom Retraction
D1 gets 3 points, D2 gets 2 points and D3 gets 4 points. D1 uses cables and a support beam, the cost will not be that high because the components are very standard. D3 has one point more because only cables are used, so the cost will decrease because there is less material needed. Hydraulics (D2) get the least amount of points, since the purchase cost of hydraulic systems is in general high.

Boom Hinge
For the boom hinge there are two options which are quite similar to each other. They both use the same amount of material, therefore E1 and E2 both get 3 points.

3.3 Conclusion

As can be seen in Table 3, for driving the trolley over the boom, the design with having the motor directly connected (by a transmission) to the wheels from the trolley (A1.1) performs the best. The wheels should be made out of steel instead of rubber, because when using wheels out of steel, the least amount of power required for riding the trolley over the boom. For the hoist there are three options for lifting the containers. The best scoring concept is the design containing two shafts and three pulleys (B1). However, the option with two shafts (B2) has only two points less, so this could also be an option. For the boom structure, there is an obvious difference in points given, in which the truss system (C2) scores the highest. Also for the boom retraction, there is an obvious difference in points given in which the cables with a support beam (D1) score the highest. Locking the boom vertically will be done using a slotted shaft in between two booms or just above. For the boom hinge, it is chosen to use the supported hinge design (E1), because this design scores the highest. An overview of the final concept can be seen in table 4.

Table 4: Final concept

Trolley	Motor directly on trolley wheels (A1.1)
Hoist	Two shafts, three pulleys (B1)
Boom Structure	Truss system (C2)
Boom Retraction	Cables with support beam (D1)
Boom Hinge	Supported (E1)

4 Assigning Materials

Table 5: Fixed-free table to determine the performance index of Equation 12

Fixed			**Free**		
L	Length	m	σ_y	Yield strength	MPa
F	Force	N	ρ	Density	kg/m^3
			A	Area	m^2

Mass for a certain component:

$$m = AL\rho \tag{8}$$

Formula for yield strength:

$$\sigma_y = \frac{F}{A} \tag{9}$$

By combining Equation 8 and Equation 9, a new equation will be formed:

$$m = \frac{F}{\sigma_y} L\rho \tag{10}$$

Simplifying the equation:

$$m = \frac{\rho}{\sigma_y} LF \tag{11}$$

Therefore, the performance index for maximal yield strength and minimal weight is:

$$M = \frac{\sigma_y}{\rho} \tag{12}$$

By using this performance index in the program CES for some components and by using information about materials/parts currently used in this industry, materials for different components of the final design were chosen. These different components only include parts with large stresses/loads on them.

The following properties and limits are important for all of the components for which materials were chosen:

- A given minimum service temperature, which equals -40°C [1].
- A given maximum service temperature, which equals 50°C.
- Low price, since it is wanted to keep the price per kg material as low as possible.
- Good corrosion resistance, since the crane operates directly near salt water from the sea, which can result in corrosion. Also general oxidation could occur.

4.1 Frame

For the frame of the crane, the following properties and limits are important to consider:

- High yield strength, since plastic deformation alters the structure of the frame. This can ultimately lead to stress concentrations and thus failure of the frame.
- Low density, since a low weight of the frame is favourable when it is transported from the production factory to the harbor and when it gets assembled. However, it is not paramount, since most likely a certain type of steel will be used and all steels have a quite similar density.
- High elongation at fracture, since it is desired to prevent fracture. A "warning" in the form of plastic deformation would be favourable. A minimum elongation of 20% strain is taken.
- Fracture toughness, because the arising/existence of cracks should not immediately lead to fracture. A minimum is set to be 50 MPa$\sqrt{m}$.

It is chosen to use a type of steel, because it is cheap and the most common used material for frames in the STS crane industry. A cheap material is very important for the frame, because it is the component of the crane with the largest volume, so the most material is needed for this. When putting the performance index of Equation 12 into CES and setting limits according to the list above, the following three main materials are suggested: carbon steel, low alloy steel, stainless steel.

Although corrosion resistance is very important for the frame, the material itself does not need to be corrosion resistant. A layer of paint can be applied onto the frame, which will prevent corrosion from occurring. This layer will need to be applied every few years of maybe even more frequently, since it will wear away when time passes. This means that stainless steel is not an ideal option. It also has the highest price per kg out of the three suggested materials by CES.

Now, low alloy steel and carbon steel are left to be chosen. Most of the suggested materials are quenched and/or annealed. Since the frame consists of huge beams, these treatments are not practical. Therefore, these options are not valid.

Now, the two best options are either high strength low alloy steel (cold rolled) or carbon steel (as rolled). The main difference between these comes with the yield strength and the elongation at fracture. Choosing an optimal yield strength, the low alloy steel would be preferable. However, the yield strength just needs to be sufficient to withstand the maximal load in the frame and both yield strengths will most likely be. This will later on be checked when the FEM analysis is performed. Furthermore, the carbon steel has a higher elongation at fracture than the low alloy steel, namely 26% vs 17% strain, which is why the first is chosen. It concerns the following material: **Carbon steel, AISI 1117, as rolled**.

4.2 Hoist shafts (drums)

For the two shafts (drums) of the hoist, the following properties and limits are important to consider:

- High yield strength, since plastic deformation is not wanted. The forces on the shafts are very high, namely 0.5 times the total load (50 tons).
- Low density, since a low weight of the shafts is favourable, because the trolley, in which these shafts are located, is accelerating and decelerating a lot. Less weight means less necessary power to move the trolley, since there is less friction.
- High elongation at fracture, since it is desired to prevent fracture. A "warning" in the form of plastic deformation would be favourable. A minimum elongation of 20% strain is taken.
- High fracture toughness, because the arising/existence of cracks should not immediately lead to fracture. A minimum is chosen to be 50 MPa$\sqrt{m}$.

When putting the performance index of Equation 12 into CES and setting limits according to the list above, different types of stainless steels, zinc-alloys and polymers are suggested.

A polymer would not be a good option, because of its low yield strength. Thus, stainless steels and zinc-alloys are left. Stainless steels have considerably higher yield strengths than the zinc-alloys, namely about 400 MPa. Additionally, they are excellently resistant against salt water flow, whereas the zinc-alloys are only acceptably resistant against it. However, zinc-alloys are cheaper than stainless steels, approximately €3 per kg. Considering the shafts are vital components for the crane, price is less important than yield strength and corrosion resistance. Therefore, a stainless steel will be chosen. Choosing an optimal yield strength, while also striving for the lowest price per kg, the following material is chosen: **Stainless steel, duplex, ASTM CD-4MCu, cast, water quenched**.

4.3 Boom hinge pin

For the pin of the boom hinge, the properties and limits similar to the hoist shafts are important to consider. However, density (weight) is not as important, since the pin is not moving when the crane is operating. Also, corrosion resistance for the material itself is not as important, since the pin with its bearings can be sealed of from the outside by a seal.

Although there are some differences between the boom hinge and the hoist shafts, the materials suggested by CES are the same. Thus, the material chosen is: **Stainless steel, duplex, ASTM CD-4MCu, cast, water quenched**.

4.4 Hoist cables

The hoist cables will be ordered from a company as standard part. Important properties and limits for them are listed below:

- Dimensions, a (calculated) length of 40m and a diameter of 30 mm.
- High yield strength, since plastic deformation is not wanted. The weights on the cables are quite high, namely 0.25 times the total load (50 tons). This results in force of 122,625 N per cable. Using Equation 37, a diameter of 30 mm and a safety factor of 2, the required yield strength should at least be equal to 346.96 MPa.
- Low density, since a low weight of the cables is favourable, because they are being hoisted a lot. Less weight means less power needed for the motor.
- High ductility, since the cables need to be able to be wounded up around drums.
- High elongation at fracture, since it is desired to prevent fracture. A "warning" in the form of plastic deformation can prevent catastrophic damage from occurring.
- High fracture toughness, because the arising/existence of cracks should not immediately lead to fracture. A prevention to this is already implemented a bit in the cable design itself, since the steel wire cables consist of multiple small cables. If cracks grow in one of these small cables, the rest of cables are not affected by this.

- High fatigue strength, since the stresses in the cables are varying, since containers are being picked up and released constantly.

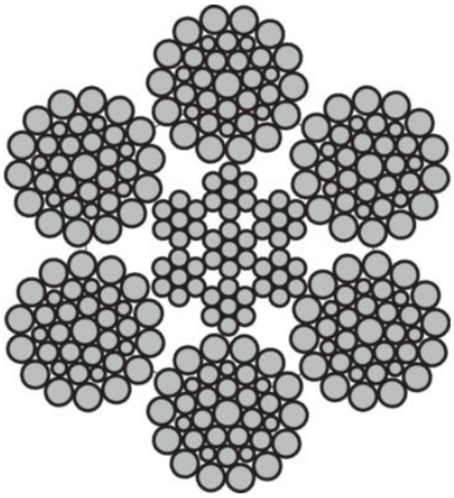


Figure 5: Steel wire cable design [4]

Using Equation 1, the maximum force in the cables is 0.25 times the total load (50 tons) times the gravitational acceleration, which equals 122,625 N. Now the stress in the cable will be calculated by using Equation 37. Firstly, taking a diameter of 30 mm^2 for the cables (which was calculated previously), the area A becomes 706.86 mm^2. Therefore, the stress in each of the four cables is equal to 173.47 MPa. Taking a safety factor of 3.5, this means that the yield strength of the cables should be at least 3.5 times 173.47, which equals 607.15 MPa.

	Part Code	Diameter mm	Min. breakingforce kN 1960 N/mm²	Weight kg/100m
▸	01.G10---280G	28	547	312
▸	01.G10---300G	30	628	358
▸	01.G10---320G	32	715	408
▸	01.G10---340G	34	807	460
▸	01.G10---360G	36	904	516

Figure 6: Table with different dimensions for the steel wire cables [4]

After some thorough research, the following steel wire ropes, which meet the requirements above, were chosen: **Steel wire rope - 6x36ws+iwrc**. The design can be seen in Figure 5 and some of their specifications can be seen in Figure 6. The ropes are manufactured by the company Mennens [4].

4.5 Trolley wheels

The trolley wheels will be ordered from a company as standard part. The important properties and limits for the wheels are quite similar to those of the most loaded gear, which can be seen in the section above. However, some of these properties, limit and their reasons vary, so they will still be examined separately below:

- High yield strength, since plastic deformation is not wanted and the stresses in the wheels are quite high. The 4 wheels need to be able to at least support the load of the spreader and container combined, which equal 50 tons, which subsequently equals 490,500 N, making it 122,625 N per wheel. The total weight will actually be higher, because the trolley will contain e.g. shafts and motors as well. For now, a safety factor of 2 will be taken, resulting in a minimum force of 245,250 N per wheel.
- High elongation at fracture, since it is desired to prevent fracture. A "warning" in the form of plastic deformation can prevent catastrophic damage from occurring.

- High fracture toughness, because the arising/existence of cracks should not immediately lead to fracture.
- High fatigue strength, since the stresses on the trolley wheels are varying constantly when it is operating.
- Low thermal expansion, because this could result in wheels which do not move easily over the rails anymore.
- Low friction coefficient, since the wheels are constantly moving in contact with the rails. However, the wheels should not slip on the rails.
- The wheels should be slotted/have a key way, because they are powered by a motor (and not e.g. pulled by a rope).

Figure 7: Trolley wheel design [5]

As elaborated in the section *Conceptual Design*, it was chosen to use H-shaped wheels for the trolley. After some thorough research, the following trolley wheels, which meet the requirements above, were chosen: **Crane wheel B 200 55 45 H7 KG 013**. The wheels are manufactured by the company Karl Georg [5]. The design can be seen in Figure 7. The material of which these wheels are made is C45 and no further specifications were given by the firm. A great option this company offers is the heat treatment "slip free hardening of the running surface", which makes the wheels obtain a better grip on the rails.

4.6 Most loaded bearing

The bearings were ordered from a company as standard part. Only the most loaded bearing in the final design will be chosen, since this bearing will be able to handle the smaller loads as well. However, the type of bearing may vary (e.g. ball bearing, roller bearing). The most loaded bearings in the final design are the two bearings connected to the shafts (drums) in the hoist. Important properties and limits for the material choice of the bearings are listed below:

- High yield strength, since plastic deformation is not wanted and the stresses in the bearing are very high.
- High elongation at fracture, since it is desired to prevent fracture. A "warning" in the form of plastic deformation can prevent catastrophic damage from occurring.
- High fracture toughness, because the arising/existence of cracks should not immediately lead to fracture.
- High fatigue strength, since the stresses within a bearing are varying quite a lot (with a certain constant frequency) when it is rotating.
- Low thermal expansion, because this could result in bearings which do not rotate easily anymore.

- Low friction coefficient, since the balls/cylinders in the bearing are constantly moving in contact with the outer and inner shell of the bearing.

Corrosion resistance for the bearing material itself is not that important, because the bearing can be sealed of from the outside with a seal. The final material choice (type of bearing) for the highest loaded bearing will be determined in the section *Machine Elements*.

4.7 Most loaded gear

The gears were ordered from a company as standard parts. Only the most loaded bearing in the final design will be chosen, since this bearing will be able to handle the smaller loads as well. However, the type of gear may vary (e.g. spur, helical). The most loaded gears are the gears connecting the motors and the shafts of the hoist. Important properties and limits for the material choice of gears are listed below.

- High yield strength, since plastic deformation is not wanted and the stresses in the gear are very high.
- High elongation at fracture, since it is desired to prevent fracture. A "warning" in the form of plastic deformation can prevent catastrophic damage from occurring. A minimum elongation of 20% strain is taken.
- High fracture toughness, because the arising/existence of cracks should not immediately lead to fracture.
- High fatigue strength, since the stresses in a bearing are varying quite a lot (with a certain constant frequency).
- Low thermal expansion, because this could result in gears which do not rotate easily against each other anymore.
- Low friction coefficient, since gears are constantly moving in contact with other gears.

When putting the performance index of Equation 12 into the program CES, four different types of stainless steels are suggested. The yield strength of the gears should just meet the stress in the gears times a certain safety factor. This will be checked in the section *Machine Elements* when the maximum stresses in the maximum loaded gear is known. For the gears, it is desired to have a high fracture toughness, so they can take high loads and because of that plastically deform a lot before they actually fracture. Therefore, the final material choice for the highest loaded gear is: **Stainless steel, duplex, Ilium P, cast, water quenched**.

5 Analysis Final Design

5.1 Frame

5.1.1 FBD of the Frame

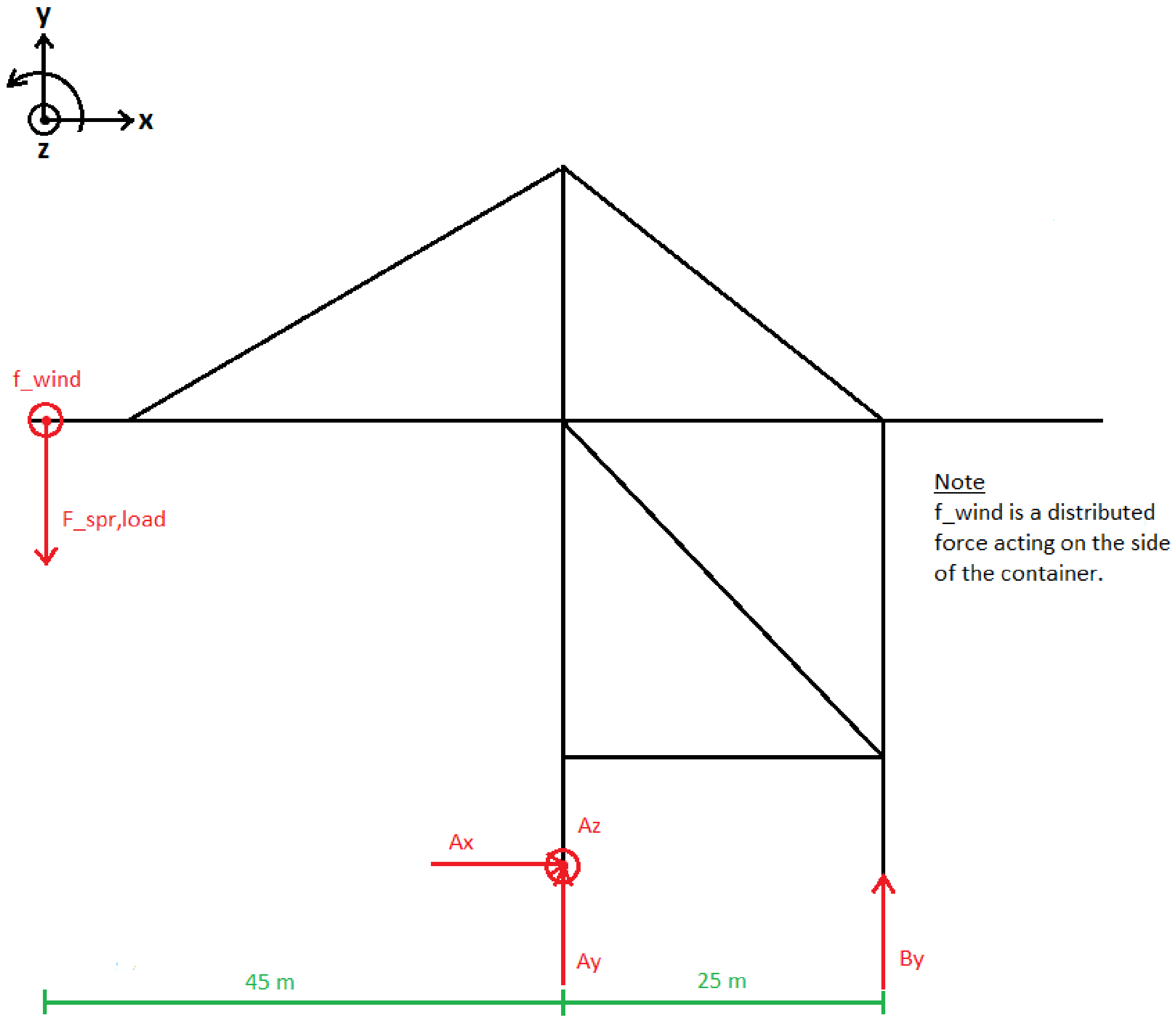


Figure 8: FBD of the frame in 2D

A FBD of the frame in 2D can be seen in Figure 8. The reaction forces of the supports were calculation by using this FBD and the given external forces $F_{spr,load}$ and f_{wind}. The distributed force f_{wind} is equal to 240 N/m^2. In order to calculate F_{wind}, this value should be multiplied with the area of the side of the container. This area was set to be equal to the largest side of the container, yielding a length of 12 m and a height of 2.6 m. The direction of the wind could act in all directions, but what is shown above is a critical condition. There are no other forces in the z direction and therefore the crane will have less rigidity is this direction. Because of this, the load case being calculated is one which would create the most disruption in the structure. Doing this ensures that the frame is strong enough for all scenarios. $F_{spr,load}$ was calculated by using Equation 1, with m equal to 50,000 kg (50 tons) and g equal to 9.81 m/s^2. This results in $F_{spr,load}$ = 490,500 N. The load case being calculated has the force placed right at the end of the boom. This is so that a worst case scenario is being calculated. At this point the moment of the boom is at a maximum and would only be smaller in actual applications. This ensures that the crane is strong enough for all other load cases. The equilibrium equations of the frame are elaborated below:

$$\sum F_x = 0 = A_x \tag{13}$$

$$\sum F_z = 0 = f_{wind} \cdot 2.6 \cdot 12 + A_z \tag{14}$$

$$\sum F_y = 0 = -F_{spr,load} + A_y + B_y \tag{15}$$

$$\sum M_{@A} = 0 = F_{spr,load} \cdot 45 + B_y \cdot 25 \tag{16}$$

First of all, A_x can be calculation using Equation 13. This results in $A_x = 0$. Equation 14 can be used to calculate A_z. It results in A_z = -7,488 N. Next, equation 16 can be used to calculate B_y, which results in B_y = -882,900 N. Finally, Equation 15 can be used to calculate A_y. Filling in the equation results in A_y = 1,373,400 N.

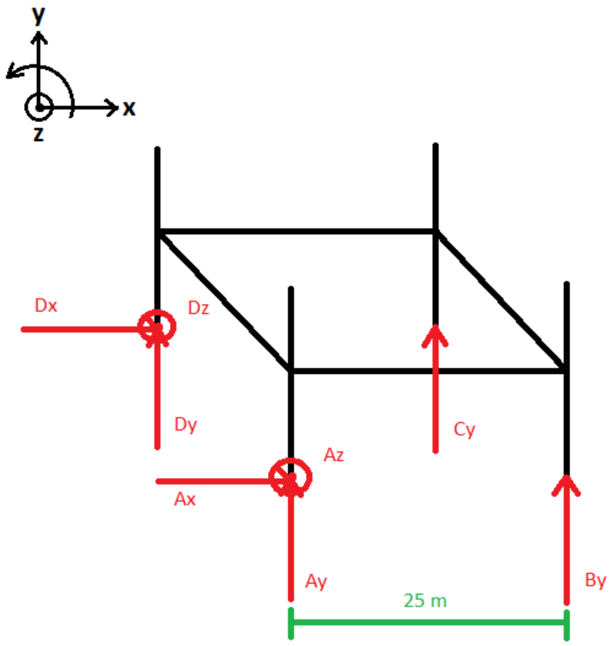


Figure 9: Part of FBD of the frame in 3D

In reality, the frame is of course built in 3D and consists of 4 supports instead of just 2. This results in more support forces in total, which means that the forces calculated above change. It is chosen to use 2 hinge supports and 2 roller supports. The forces acting on the frame due to these supports can be seen in Figure 9, in which the the lower part of the frame is drawn in 3D. Because the frame of the crane is almost entirely symmetrical in 3D about the X-axis, the calculated forces at the support in 2D can be used to calculate those in 3D.

Firstly, the total force in X-direction will be calculated. Using the A_x calculated for the frame in 2D, the new A_x and D_x are equal to this value (which is 0) divided by 2. This results in $A_x = D_x = \frac{0}{2} = 0$. The same procedure will be used to calculate the remaining forces. This results in $A_y = Dy = \frac{1{,}373{,}400}{2} =$ 686,700 N, $B_y = C_y = \frac{-882{,}900}{2} =$ -441,450 N and $A_z = D_z = \frac{-7{,}488}{2} =$ -3,744 N.

5.1.2 Frame Design FEM

The frame analysis was done by using a FEM analysis. This was done in a few iterations to optimise the design. Firstly, some forces had to be calculated so that they could be applied in the FEM. There were two forces to be calculated, the first of which is the weight of the container and spreader acting downwards on the boom. This force is equivalent to the total weight (50 tons) multiplied the gravitational acceleration of 9.81m/s^2. Secondly, the maximum force of the wind needs to be calculated. This is done by using equation 17 below, where area A is equal to the product of the two longest sides of the container, 2.6m and 12m. The constraints that were used at this point are the first three legs of the frame are fully constrained and the other constrained only in the y direction, beyond this there is nothing constrained. This will be adjusted later.

$$F = F_{w,max}A \tag{17}$$

The latter force acts in the Z-direction. The first iteration was done with a relatively simple structure with very little z plane stiffness. The rectangular beams had dimensions of 10cm by 15cm. This results in max displacement of 4mm in the x direction, 2mm in y direction and 22m in z. A maximum stress was also found to be 1GPa. These values are of course not acceptable and a second iteration was done. A table of all values can be found in appendix A.1

The second iteration included a cross on the back side of the frame.There was little effect on the displacements in x and y direction however the maximum displacement in z was drastically decreased to 5 m. This is in two locations which are the two front legs of the crane, so some additional stiffness is required there. In addition to this a rotation of the front leg is also around 40cm in the x direction. No changes were made to the cross sectional area of the beams. A max stress was found to be 600MPa in the front leg. More stress values are shown in appendix A.1

The third iteration included the needed stiffness for the front side of the crane, this reduced the deformation to a couple of millimeters. There was no change to area. The maximum displacement is the rear leg of the crane which is deforming by 2m so some additional stiffness was added to the rear two legs, just as done by the front. As before the maximum stress is on the front leg, which is 260MPa. The rest of the stresses are shown in appendix A.1

For the fourth iteration the rear leg displacement was corrected as said above. This final adjustment reduces the displacement; in x to a maximum of 3mm, in y to a maximum of 1.1mm, and in z to a maximum of 4.4mm. The maximum stress is 17.5MPa in the recently added cross beam. The dimensions are still kept at 10cm by 15cm. The full list of stresses can be seen in appendix A.1

The constraints were adjusted so that the front two legs are constrained but the rear two are only constrained in the y axis, but free to move in both z and x. This has little effect on the stresses and displacements in the design. In addition to this the location of the supporting beam on the boom was adjusted during boom analysis, resulting in the final design shown in figure 10, using scale 5 to view it in realistic terms. In addition to this final stress values are shown in appendix A.1. The area of each of the elements are kept at the same area, 10cm x 15cm. There is no need for larger beams as maximum stress does not exceed 20 MPa. There is a possibility to make certain elements in low stress areas with a lower area. However, this would increase manufacturing costs, which is not wanted. Despite this however, as will be seen, different sized elements are used in the boom calculations.

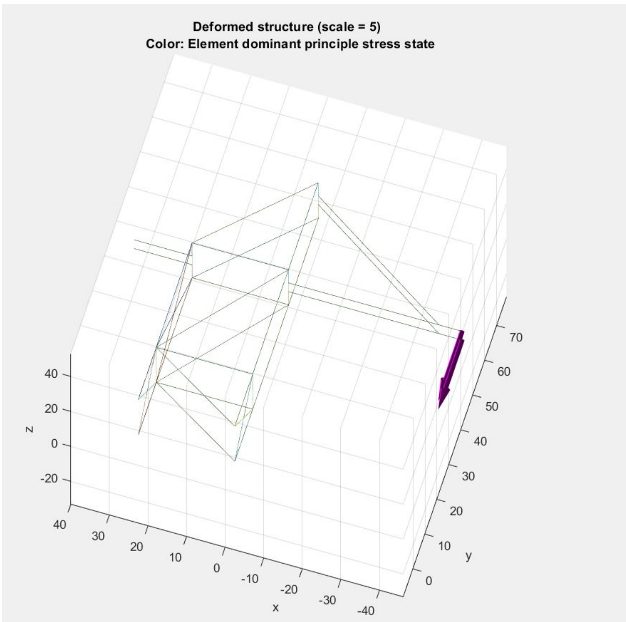


Figure 10: Frame Deformation Scale 5

5.2 Boom

For the boom analysis, FEM was also used. For this the design was first made such that it is suitable and then in a few iterations it will be made to decrease the weight of the boom. The boom is constrained fully in two locations, at the hinge and at the supporting beam.

For the first iteration a design was made that full fills the requirements. The element diameter was chosen to be 0.05m and a wall thickness of 1e-2. This resulted in a max displacement of 8.5mm in the x direction, 9.7mm in the y direction and 4mm in the z direction. The maximum stress is found to be 279 MPa in an element just over halfway through the beam. This is adjusted later by changing the cross sectional area in this element. The stresses are shown in figure A.2

For the second iteration, some supporting elements on the bottom of the boom were removed. As the boom requires less support in the z axis, these supports are not required. This removes 1/3 of all the elements on the bottom of the boom, saving a lot of weight. This increases the displacement in the z direction to 4.5mm. The maximum stress is unchanged.

For the final iteration the thickness of the high stress beams was increased so that the stress in those elements could be decreased. Some elements are completely solid to increase their area, whilst others only a thicker wall. In addition to this some supporting beams around where the load of the crate is hung were supported to decrease their displacement and stress. This beam is also the most stressed beam at 136MPa. Most of the beams sit below 100MPa with a few at approximately 100MPa.

The full list of stresses compared to iteration 1 are shown in Appendix A.2 The max displacements are similar to that above, a max of 4.6mm in x, 14.6mm in y and 5mm in z. However, the maximum displacements of the boom are important. As seen in figure 11, which uses a displacement factor of 1000, there is a total displacement of 5mm in the z direction. On top of this it is seen that the displacement along the boom, in the x direction is also around 5mm. Lastly, as seen in figure 12 the total displacement in the y direction is 18mm. An image with scale 1 is shown in figure 13

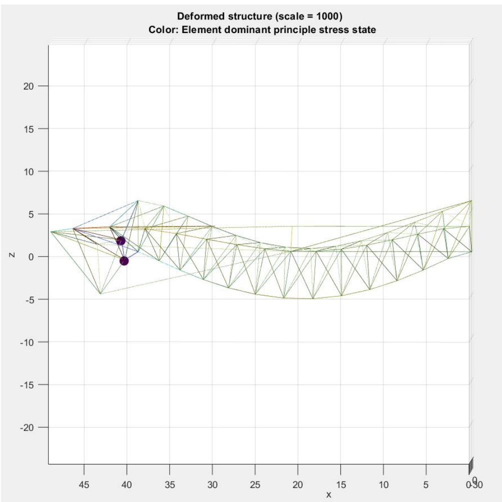


Figure 11: Max Displacement Boom in Z and X directions

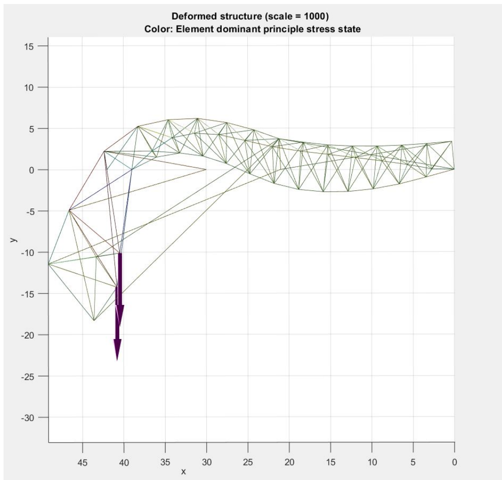


Figure 12: Max Displacement Boom in Y direction

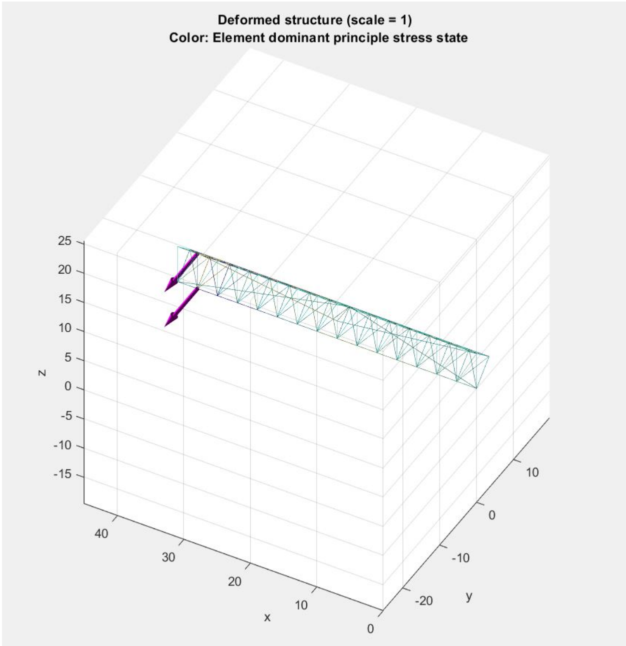


Figure 13: Boom Deformation Scale 1

5.2.1 Boom Cables

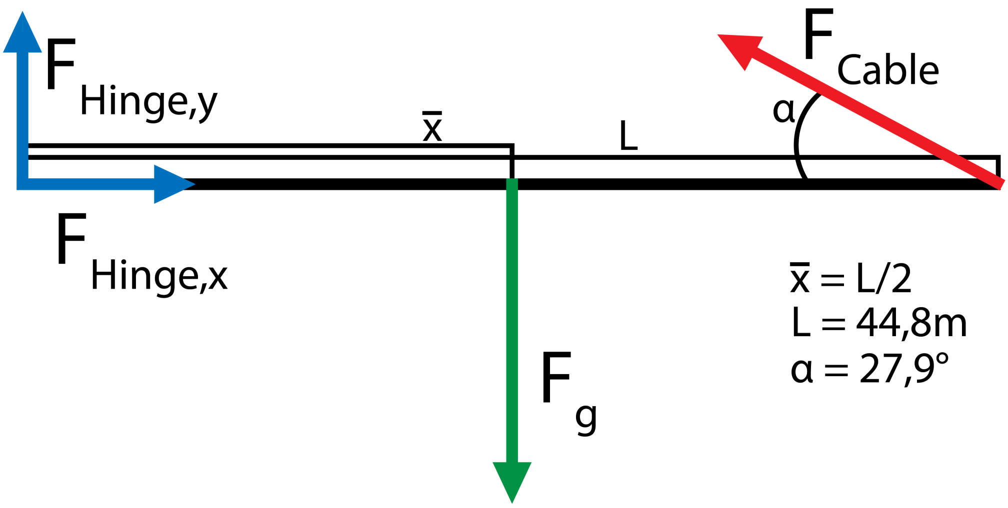


Figure 14: FBD of the cable situation

The boom can be retracted using two cables connected to a motor located at the back of the crane. The weight of the boom was estimated to be around 20 tons. The calculated cable force is then 105 kN per cable. Assuming the centre of mass is located at the middle of the boom. To determine the thickness of the cables. The *steel quality* is S355, which means that the yield strength is 355 N/mm^2, the minimum $safety\ factor$ is 3.5, as is usual for gantry cranes.
The thickness of the cables can be calculated using equation 18:

$$D = \sqrt{\frac{safety\ factor \cdot F}{steel\ quality \cdot 0.25\pi}} \tag{18}$$

Using this equation, the calculated minimum diameter of the cable is 36.83mm.

5.3 Extra Analysis

5.3.1 Bolt Calculation

The hoist is mounted to the trolley with some pre-loaded bolts, the amount requaired and at which diameter is calculated. The total tension force over all bolts is the weight of 50 tons, which equals 490500 N. There is additional force of 7488 N caused by the wind. However this is compensated with friction due to pre-loading. The static steel on steel coefficient of friction is 0.7, this means that the total pre-load force is found using equation 20

$$F_{pre} = F_{clam} + F_t \tag{19}$$

$$F_{pre} = F_t + F_w/\mu \tag{20}$$

The total force on the bolt is therefore 490500 + 7488/0.7 = 501,197.14 N. To finish of the calculation a safety factor of 2 is used. Creating a total force of 1002394.286 N. At least 4 bolts will be used to ensure the weight is equally distributed, this would result in a force of 250,598.57 N per bolt. If a grade 10.9 bolt is used, it will have a yield strength of 100*100*0.9 = 900 MPa. Using equation 21 the tensile stress area can be calculated and from that the diameter can be found.

$$A_t = F/\sigma_t \tag{21}$$

If the values of max force and yield strength are applied to equation 21, the value of A_t is found to be 278.44 mm^2 for each of the 4 bolts. It is seen from bolt tables that the minimum diameter is 24 mm. Meaning that 4, M24 bolts are used.
A lower grade could also be used. If the grade is 5.6, the yield strength would equal 500*0.6 = 300 MPa. Therefore, using equation 21 a tensile area is found to be 835.32 mm^2, requiring a diameter of 42mm. This can be reduced further by increasing the number of bolts. If 8 bolts are used, one at the edge of each corner, the force will be equally divided by those 8. This leaves max force per bolt at 125,299.29 N and a tensile area of 417 mm^2. This leaves us with a diameter of 30mm.
In conclusion, 8 M30 grade 5.6 bolts are used to connect the hoist to the trolley.

5.3.2 Weld Calculation

The supporting beam is attached to the boom through two hinges, these hinges are welded to the boom. The total force for lifting the boom up will fully be on these two welds. The total tension force on the welt is half of the weight of the boom. Since the total weight of the boom was estimated at around 20 tons, the tension force will be 20 tons. This equals 98,100 N. Each of the two welds will have a force of 49,050 N on it. The welds will be subjected to tensile stress and twisting moment. The tensile force is found using equations 22 and 23, where b and d are the length and the with of the hoist.

$$A_w = 2 * b + 2 * d \tag{22}$$

$$f_{tension} = \frac{P}{A_w} \tag{23}$$

If the values of b, d and P are applied to equations 22 and 23, the value of $f_{tension}$ is 47.16 N/mm. Next the twisting stress need to be calculated. Since the retraction cable is placed under an angle of 27 degrees, there will be a twisting moment in x and y direction. First the point of the weld that experiences the greatest force needs to be calculated using equations 24 and 25, where a is the center point of the hoist. In this case $a = 85mm$.

$$T_y = F_y * a \tag{24}$$

$$T_x = F_x * a \tag{25}$$

Now the total twisting moment can be calculated using equations 26 and 27.

$$J_w = \frac{(b+d)^3}{6} \tag{26}$$

$$f_{T(x,y)} = \frac{T_x + t_y) * \frac{1}{2a}}{J_w} \tag{27}$$

Finally the vectorial combination of the forces can be calculated using equation

$$F_r = \sqrt{(f_{T(x,y)})^2 + (f_{tension})^2} \tag{28}$$

To determine the weld thickness the the maximum force must be divided by the allowable stress.

TABLE 20–2 Allowable shear stresses on fillet welds

A. Steel

Electrode type	Typical metals joined (ASTM grade)	Allowable shear stress
E60	A36, A500	18 ksi (124 MPa)
E70	A242, A441	21 ksi (145 MPa)
E80	A572, Grade 65	24 ksi (165 MPa)
E90		27 ksi (186 MPa)
E100		30 ksi (207 MPa)
E110		33 ksi (228 MPa)

Figure 15: Allowable shear stresses on fillet welds

Using figure 15 the allowable shear stress is chosen as $\sigma = 145MPa$. Now all the values are known for calculating the weld thickness, using equation 29.

$$A = \frac{F_r}{\sigma} \tag{29}$$

Filling in all the values in the equations 22 to 29 gives an filled weld throat of $A = 0.34mm$ (A = t)

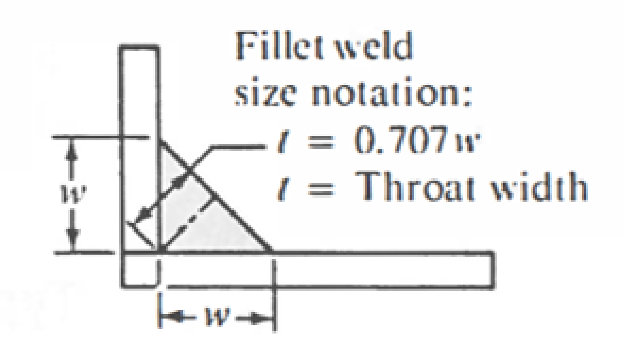


Figure 16: A filled weld[6]

In figure 16 the filled weld size notation can be seen, using this notation the minimum weld size can be calculated using equation 30, where $t_0 = 0.707$

$$w = \frac{t}{t_0} \tag{30}$$

Filling t and t_0 in equation 30 gives $w = 0.48mm$. Since the plate thickness of the hoist (that will be welded on to the boom) will be 25 mm, from figure 17 can be concluded that the minimum leg size for filled weld is 7.94 mm.

TABLE 20–4 Minimum weld sizes for thick plates

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

Figure 17: Minimum weld size for thick plates[6]

5.3.3 Mohr's Circle

To help find absolute maximum values of stress, and also the combination of stresses within a certain part, a Mohr's circle is created for this part. The shaft of the hoist is susceptible to a large load and a Mohr's circle will be made for this part. The calculation will be done for the critical condition, where the whole force acting on the shaft is acting through the center. Firstly the forces acting at that point must be found. This is done by the method of sections and results in the following M(x) and S(x) graphs. As seen by figure 19 and figure 18, the critical condition is then in the center and this will then also be the point where the Mohr's circle will be calculated.

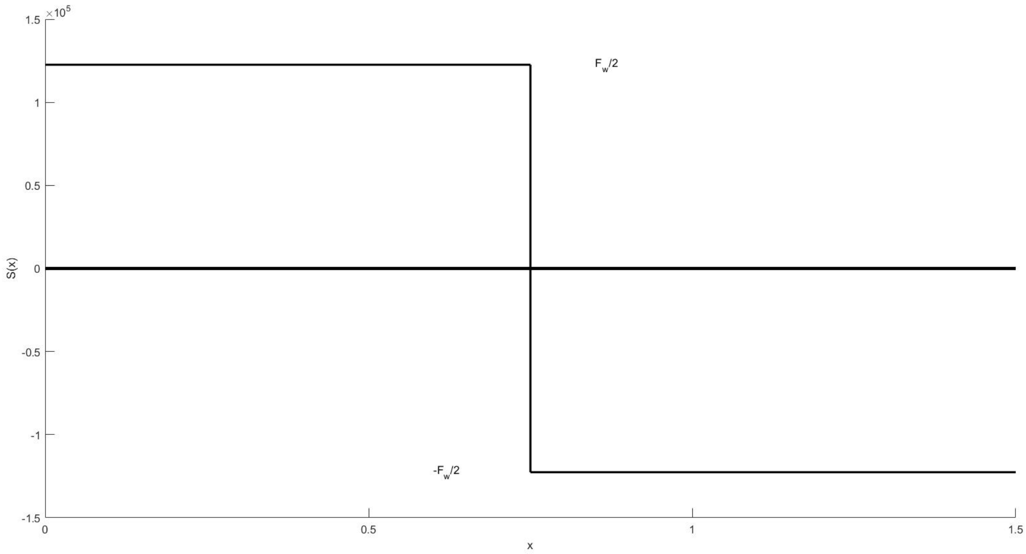


Figure 18: Shear Force Diagram

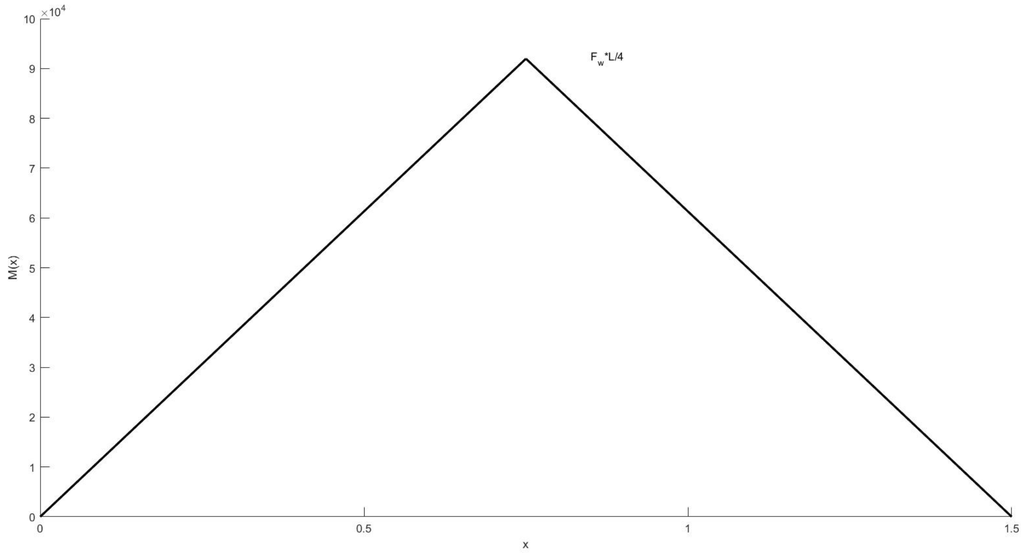


Figure 19: Moment Diagram

The forces calculated first need to be translated to their respected stresses. The moment creates a bending stress according to equation 31. The shear force creates a stress as seen in equation 32.

$$\sigma_x = yM(x)/I \tag{31}$$

Where y is the position in the y axis, M(x) is the moments and I is the second moment of area.

$$\sigma_y = -S(x)Q(y)/(It(y)) \tag{32}$$

Where S(x) is the shear force, Q(y) is the first moment of area at that point, I is the second moment of area and t(y) is the thickness. The first moment of area is found from the following equation.

$$Q(y) = \int ydA \tag{33}$$

This is done by replacing the area and limits as ones from a circle, and is found to be $2/3R^3$[7]. The second moment of area is $\pi/4R^4$.
Lastly the torque of the shaft itself is translated to a shear stress by equation 37

$$\tau = rT(x)/J \tag{34}$$

Where r is the radius, T(x) is the torsion and J is the torsion constant.
The center of Mohr's circle is found using σ_{avg}. This is equal to $(\sigma_x + \sigma_y)/2$. The centre lies at the coordinate $(\sigma_{avg}, 0)$. The circle is made in MatLab and the equation 35 is used to plot this and is seen in figure 20. The point A as seen is the value of stresses at $\theta = 0$

$$R = \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy^2}} \tag{35}$$

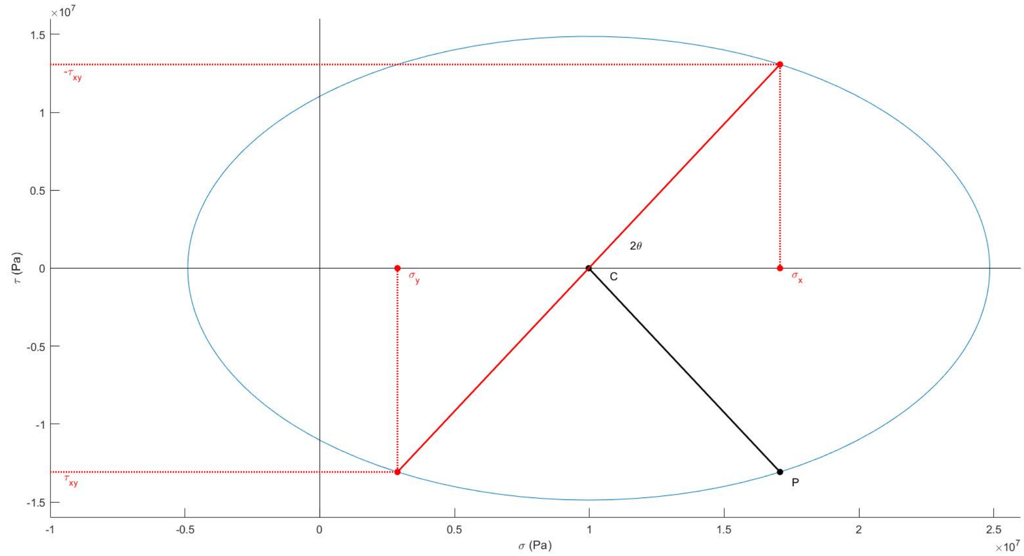


Figure 20: Mohr's Circle

Lastly the angle theta is calculated. This is found using equation 36

$$tan(2\theta_p) = \frac{2\tau_{xy}}{\sigma_x - \sigma_y} \tag{36}$$

θ at P is found to be 30.75° meaning P is 61.5° below the x axis. This means he maximum normal stress is found at 61.5° and is equal to σ_{avg} + R = 24.85 MPa. The maximum shear stress is equal to R and is found at 151.5° and is equal to 14.87 MPa.

6 Machine Elements

6.1 Machine Elements Hoist

6.1.1 Cables and outer shafts dimensions

The spreader (where the container is connected to) is connected with 4 cables on the hoist. The weight one cable will 'carry' is thus equal to $F = 125kN$. To determine the thickness of the cable and the thickness of the two main shafts in the hoist, there are made a few assumptions. The *steel quality* is S355, which means that the yield strength is 355 N/mm², the $safety\,factor$ is 2 and the length of the shaft is taken at

l = 1500 mm. Equation 37 is used as the starting point for determining the thickness of the cables.

$$\sigma = \frac{F}{A} \tag{37}$$

$$\frac{steel\ quality}{safety\ factor} = \frac{F}{0.25\pi D^2} \tag{38}$$

Rewriting equation 38 gives equation 39, with witch the thickness of the cable can be calculated.

$$D = \sqrt{\frac{safety\ factor \cdot F}{steel\ quality \cdot 0.25\pi}} \tag{39}$$

The thickness of the cables should be D = 30 mm. With help of the thickness of the cables, the thickness of the shaft can be calculated, figure 10 is used to make it more clear.

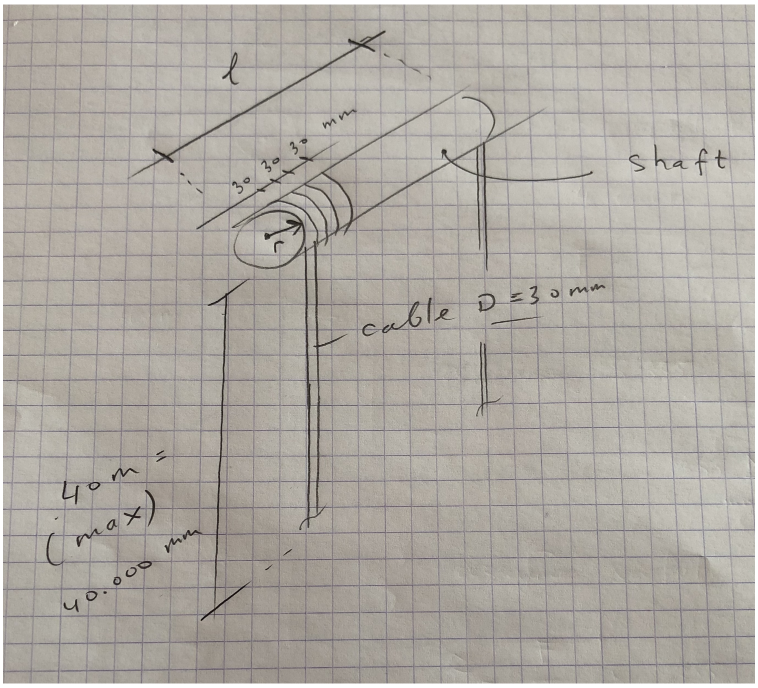


Figure 21: Cable on the shaft

In the figure above the width of the shaft, the distance from ground to shaft and the cable can be seen. To determine the thickness of the shaft, the width of the cables is used. Since the maximum load on the shaft already is considered in equation 10, it is not necessary to take it again into account. The maximum length of the cable from ground to hoist is 40 m, L_0 = 40,000 mm and the width of the shaft l = 1500 mm.

$$L_0 = 2\pi r n \tag{40}$$

$$n = \frac{0.5l}{D} \tag{41}$$

Substituting equation 41 into 40 gives equation 42.

$$L_0 = 2\pi r \frac{0.5l}{D} \tag{42}$$

To determine the ratio of the shaft, equation 42 should be rewritten.

$$r = \frac{L_0 D}{2\pi 0.5l} \tag{43}$$

Since L_0, D and l are known values, these can easily be filled in into equation 43, which gives a radius of r = 260 mm. The diameter of the shaft is twice the radius thus D = 520 mm.

6.1.2 Inner shaft of the hoist drum

Now the final material for the shafts on the hoist is chosen, the final diameter of the inner shaft can be determined. **Stainless steel, duplex, ASTM CD-4MCu, cast, water quenched** is chosen as material for the shafts. This material has the following characteristics.

Table 6: Characteristics of ASTM CD-4MCu Steel

S_y	=	505	MPa
S'_n	=	335	MPa

The moment M in point A is calculated with help of figure 22.

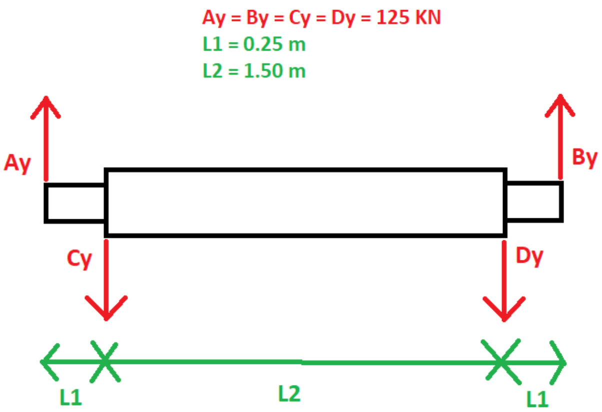


Figure 22: FBD of the shaft

Using the initial shaft diameter (520 mm), the torque on the shaft is determined. Furthermore the design factor, stress concentration factor and the bending moment also are known and can be seen in the following table.

Table 7: Values for calculating the thickness of the shaft

T	=	$140{,}820 \cdot 10^3$	Nmm
N	=	2	
K_t	=	2.5	
M	=	$31{,}250 \cdot 10^3$	Nmm

Since all the necessary values are known to calculate the thickness of the shaft, they can be inserted in the following equation.

$$D = \left(\frac{32N}{\pi} \sqrt{\left(\frac{K_t M}{S'_n} \right)^2 + \frac{3}{4} \left(\frac{T}{S_y} \right)^2} \right)^{\frac{1}{3}} \tag{44}$$

Filling the values into the equation gives a diameter of 189.82 mm for the inner shaft.

6.1.3 Shaft bearings

The shafts are mounted on the hoist using bearings. The hoist will be used to handle large quantities of containers each day. This task consists of may start-stop cycles because of which roller bearings will be used. The main stresses acting on the shafts will be radial stresses, therefore a cylindrical roller bearing will be used on one side and a spherical roller bearing will be used on the other side to hold the shaft in place. Also since the crane will operate in close proximity to the sea, which is a quite corrosive environment, all the bearings need to have a rubber seal to prevent them from being damaged by the environment.

In order to calculate the specific properties of the bearings first L_{10h} must be determined. It is assumed that the bearings of the the hoist should not be replaced more than once a year, which gives a L_{10h} equal to 8,766 hours.

$$L_{10m} = \frac{L_{10h} 60n}{10^6} \tag{45}$$

Afterwords L_{10m} needs to be calculated using Equation 45. Since L_{10h} is known only the revolutions per minute n need to be calculated. The shaft rotates with 7.69 rad/s , which gives us an n equal to 73.4 rpm. Plugging all the values into the equation gives a L_{10m} equal to 38.6·10^6 revolutions.

$$L_{10m} = \left(\frac{C}{P}\right)^k \tag{46}$$

Now that L_{10m} has bee found, the basic dynamic load rating C can be determined using Equation 46, in which P is the equivalent dynamic load and k is a constant based on the type of bearing used. With the radial force F_r being equal to one forth of load on the hoist, so 12.5 tons or 123 kN and the axial force F_a being equivalent only to the force of the wind blowing on the container, which is 6.8 kN, it was concluded that $F_r >> F_a$, so the axial force could be neglected. This gives an equivalent force P equal to F_r, so 123 kN. After plugging these values in the equation, it was concluded that the bearing needs to have a dynamic load rating C of at least **370 kN**.

Considering the coefficient C and the dimensions of the shafts, the bearings were chosen using the bearing selector from the company SKF [8]. The two bearings chosen were the **cylindrical roller bearing NCF 2938 CV** and the **spherical roller bearing 23930 CC/W33**. The cylindrical bearing has a dynamic load rating of 440 kN with an inner diameter of 190 mm and a width of 42 mm, while the spherical one has a C equal to 499 kN, inner diameter of 190 mm and a width of 52 mm, which makes them fit almost perfectly onto the shaft. Despite axial forces being neglected in the calculation due to their small size in comparison with the radial forces, the axial forces do still play a part at 7.5 kN, this along with the spherical bearing holding the shaft in place is the reason why the spherical roller bearing is chosen instead of two cylindrical bearings.

6.1.4 Guide ways

In order to guide the cables, so they roll up in an orderly manner in the shafts, the shafts should be manufactured with a spiral surface shape where the thickness of the rope exactly fit. Therefore the distance between these spirals should be 30 mm. Besides these guide ways, a mechanism should be incorporated to the shafts in order to move the cables while rolling up and unrolling.

6.1.5 Electric motors

In order to obtain the power needed for the motor to lift the load, the following equation is used.

$$P = Tw \tag{47}$$

Being T the total torque in the system, which means the torque on the drum, plus the torque of the dead lift and the torque to accelerate the lift.

$$T_{total} = T_{drum} + T_{dead-lift} + T_{accelerate-lift} \tag{48}$$

This formula give just an overview of what is needed to get the total torque. To calculate each of the previous mentioned torques, 3 formulas were used and those are listed below:

$$T_{drum} = I\alpha \tag{49}$$

$$T_{dead-lift} = Wr \tag{50}$$

$$T_{accelerate-lift} = mar \tag{51}$$

Now the process to get each of the equations will be explained.

6.1.6 Hoist Torque

Drum Torque

The drum torque is the torque required to accelerate the drum. For the it the moment of inertia is given by the shape of the drum and the dimensions in the following graph and formula.

$$I = \frac{M(R_1^2 + R_2^2)}{2} \tag{52}$$

As the graph shows, the drum radius is 0.26m and has a thickness of 0.06m.For the mass of the drum we refer to part 4.2 Selection of material for the drum, getting a specific mass of 7800kg/m^3 for the material and the volume is calculated using $\pi(R_1^2 - R_2^2)L$ that follows $\pi(0.26^2 - 0.20^2)1.5$ which gave a volume of 0.13m^3. The mass can now be calculated from $(7800kg/m^3) \cdot 0.13m^3$, giving a total mass of 1014kg. After filling equation 52 it follows $\frac{1014(0.26^2 + 0.20^2)}{2}$ with the calculated values, which in the end give a Moment of Inertia on the drum of 54.55 kgm^2.

After this a linear acceleration of $1m/s^2$ is assumed and the angular acceleration is calculated to be $\alpha = a/r = 1/0.26 = 3.85$rad/s^2. Once the moment of inertia and the angular acceleration are found, they can be plugged in equation 49, obtaining a torque of **420Nm** for both drums.

Dead Lift Torque

This is the torque required to hold the dead weight of the lift.
In order to get this torque, the weight is calculated by taking the mass of the load and multiplying it by the gravity and the radius of the drum, which gives **127400Nm** of torque.

First Acceleration Torque

This is the torque required to accelerate the lift. For the first acceleration of the load the mass is multiplied with the acceleration and the radius of the drum, which give a torque of **13000Nm**.

Total Torque

At the end the total torque is calculated by filling in equation 48 with the values written in bold giving a $T_{total} = \mathbf{140820Nm}$

Now that the torque is known, the power needed to lift the load can be calculated with equation 47. By using the formula for the angular velocity v/r it is calculated to be w=**7.69m/s**, which in turn results in a power P of **1.08MW**

As can be seen the power required to lift the load is too high for just one motor so it was decided to use two motors, each one of these running one drum and for that reason the total power should be divided by two, which means a power of **541kW** for each motor.

After an analysis focused on advantages and disadvantages of two different types of motors being AC and DC it was decided to use a DC motor based on the following reasons:

- Adjustable speed
- Reversible diction of rotation
- Acceleration and deceleration can be controlled
- Dynamic Breaking
- Quick response of acceleration

The maximum power to be used was 600 kW. Searching on an electric motors catalog of the Nidec company [9], a motor with a maximum power of 581 kW was found and based on other parameters such as torque, speed and moment of inertia, this motor was chosen for the crane hoist. When taking an efficiency of 0.94 into account, the power will actually be 546kW. This in turn shows that this motor nearly matches the required power for the system. The characteristics of the motor are listed in Table 8 and Table 9.

Table 8: Motor Specifications

Max. Power	581 kW
Moment of Inertia	23.9 kgm^2
Efficiency	0.94
Voltage	460 V
Speed	1030 RPM
Torque	5385 Nm

Table 9: Motor Dimensions

Length	1.83 m
Height	0.7 m
Width	0.7 m
Shaft diameter	132 mm

6.1.7 Gears

Gear ratio

In order to get the gear ratio the following formula was used:

$$i = \frac{w_{motor}}{w_{drum}} \tag{53}$$

The rotational speed of the drum was previously calculated being $w_{motor} = 7.69$ rad/s. The speed of the motor can be found in the table above, but first it needs to be converted into rad/s. The formula is $(rpm * 2\pi)/60$, giving a speed of 108 rad/s. Finally, after putting this values into formula 53, a gear ratio i of **14** was found.

Gear dimensions

The diameter of the rotor shaft was taken as reference, for the dimension of the gear connected to it, which will be the smaller gear. The rotor shaft diameter as specified by the seller is 132mm. For the gears there are some dimensions in common for the four gears in the gearbox according to figure 31. Those are listed in table 10

$$m = \frac{D}{N} \tag{54}$$

After calculating the module with an assumption of diameter of 200mm the following result $m= 200/25 = 8$ was obtained. Then, the following equations were used to calculate the values in figure 23. Their values can be found in table 10.

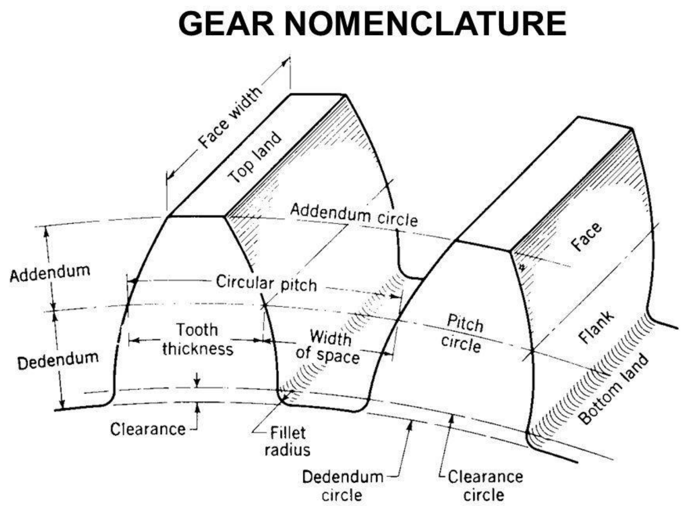


Figure 23: Gear Nomenclature

$$A = 1 \cdot m = 1 \cdot 8 = 8 \quad (55)$$

$$D = 1.25 \cdot m = 1.25 \cdot 8 = 10 \quad (56)$$

$$C = 0.25 \cdot m = 0.25 \cdot 8 = 2 \quad (57)$$

$$h_{tooth} = a + b = 8 + 10 = 18 \quad (58)$$

$$PTH = 2 \cdot a = 2 \cdot 8 = 16 \quad (59)$$

$$t = \pi \cdot \frac{m}{2} = \pi \cdot \frac{8}{2} = 4\pi \quad (60)$$

Table 10: Shared gear dimensions

Module	8
Addendum (A)	8 mm
Dedendum (D)	10 mm
Clearance (C)	2 mm
Height of tooth (h_{tooth})	18 mm
Projected tooth height (PTH)	16 mm
Tooth thickness (t)	4π mm

There are some specific dimensions used for each gear. As the total train value resulted in 14, the safest gear ratios for spur gears are between 1 to 8 , so it was decided to divide the train value between 1 to 2 and 1 to 7, thus making the power transmission safer. Therefore, if you see figure 31 the smallest gears have similar dimensions and will be named as gear 1 and gear 3. The other gears, both differ from these ones. Gear 2 will be the medium sized gear and the biggest gear will be named as gear 4. The dimensions listed below present the individual dimensions of the gears.

The first teeth number was assumed to be 17, as the common teeth number for hoisting systems for the first gear was between 14 and 20 and 17 is a prime number.

$$D_{pitch} = N \cdot m \quad (61)$$

$$D_{outside} = D + 2 \cdot a = D + 2 \cdot 8 = D + 16 \quad (62)$$

$$D_{root} = D - 2 \cdot b = D - 2 \cdot 10 = D - 20 \quad (63)$$

Table 11: Dimensions gear 1 and 3

Number of teeth	25
Pitch diameter	200 mm
Outside diameter	216 mm
Root diameter	180 mm
Inner shaft diameter	132 mm

Table 12: Dimensions gear 2

Number of teeth	51
Pitch diameter	408 mm
Outside diameter	424 mm
Root diameter	388 mm
Inner shaft diameter	132 mm

Table 13: Dimensions gear 4

Number of teeth	176
Pitch diameter	1408 mm
Outside diameter	1424 mm
Root diameter	1388 mm
Inner shaft diameter	190 mm

Gear forces

In order to calculate the gear forces, the equation below was used for each of the gears. Taking into account the power **581 kW**, angular velocity and diameter specified in table 14 .

$$F_t = \frac{2P}{w \cdot d} \tag{64}$$

With the tangential force, the radial and normal force can be calculated for each of the gears. To do this calculation, the pressure angle $\theta =$ **20 degrees** is needed in the following formulas:

$$F_r = F_t \cdot tan(\theta) \tag{65}$$

$$F_n = \frac{F_t}{cos(\theta)} \tag{66}$$

The result of the forces calculations will be showed in table 14.

Gear Bending stresses

To calculate the stress that tend to bend the gear teeth, the following calculation was applied for each gear:

$$\sigma_b = \frac{F_t * Y_{fa}}{b * m} \tag{67}$$

where, F_t is the tangential force, Y_{fa} is the form factor for standard metric module based gears, b is the width and m is the module. The width which is **110 mm** and module which is 8, are the same for the calculation of the bending stress of all the gears.

The result of the bending stress calculations will be showed in table 14

Gear contact stresses

The maximum allowable tangential force at the pitch circle, without incurring in surface failure. and is described by the following equation,

$$\sigma_H = \sqrt{\frac{F_t \cdot (u+1)}{b \cdot D_1 \cdot u}} \cdot Z_H \cdot Z_E \tag{68}$$

being F_t the tangential force, u is the teeth ratio between the gears, b the width which is the same, Z_H is the curvature factor and is an standard value of 2,5 for spur gears, Z_E is the elasticity factor that depends on the Young's modulus $E = 205$GPa, which is $Z_E = 189.406969$ and D_1 is the diameter.

The result of the contact stress calculations will be showed in table 14. Using a safety factor of 2, each gear should have a minimum yield strength of 2 times its contact stress. The maximum loaded gear is gear 3, with a contact stress of 35.42 MPa. This means that the material chosen should have a yield strength of at least 70.84 MPa. The material chosen for the most loaded gear in the section *Assigning Materials*, (stainless steel, duplex, Ilium P, cast, water quenched), has a yield strength of 365 MPa, which means that it meets this requirement.

Table 14: Gear Forces and Stresses

	Gear 1	**Gear 2**	**Gear 3**	**Gear 4**
Angular speed	107.86 rad/s	53.93 rad/s	53.93 rad/s	7.7 rad/s
Torque	5385 Nm	10773.2 Nm	10773.2 Nm	75411.6 Nm
Tangential Force	53.87 N	52.81 N	107.73 N	107.18 N
Radial Force	19.61 N	19.22 N	39.21 N	39 N
Normal Force	57.33 N	56.20 N	114.64 N	114.06 N
Y_{Fa}	2.725	2.35	2.725	2.1
Bending Stress	0.166813 N/mm^2	0.141027 N/mm^2	0.3336 N/mm^2	0.2558 N/mm^2
Contact Stress	28.6975 N/mm^2	19.8936 N/mm^2	35.4233 N/mm^2	13.31165 N/mm^2

6.1.8 Keyway of most critical situation

At gear 4 the amount of torque is the highest, as can be seen in table 14. For a shaft with a diameter of 190mm, the height of the key should be 25mm and the width should be 45mm. Substituting equation 70 in equation 69 and rewriting for l yields equation 71.

$$\tau = \frac{2 \times T}{d \times b \times l} \tag{69}$$

$$\tau_d = \frac{0.5 \times S_y}{N} \tag{70}$$

$$l = \frac{4 \times N \times T}{d \times b \times S_y} \tag{71}$$

Filling in the values obtained from table 14, and assuming a safety factor N of 3, the minimum length of the key way was found to be 311.3mm.

6.1.9 Scheme of drive system

In figures 25 and 26, a scheme of the drive system is presented. It includes four parts: the motor, the biggest gear, the transmission and the load. Other gears and shafts were neglected because their inertia's are relatively small. The first three boxes indicate the angular speed of respectively the motor, gear and drum. The last box indicates the linear speed of the load.

The torque at the beginning of the graph is the torque given by the motor. As there are two motors, the added torque of both were put in the figure as well as the added moment of inertia of the motor. For the produced torque, the torque calculated with equation 52 was used. As well as for the motors, the moment of inertia was calculated for the two drums. The mass of the load is located in the last box.

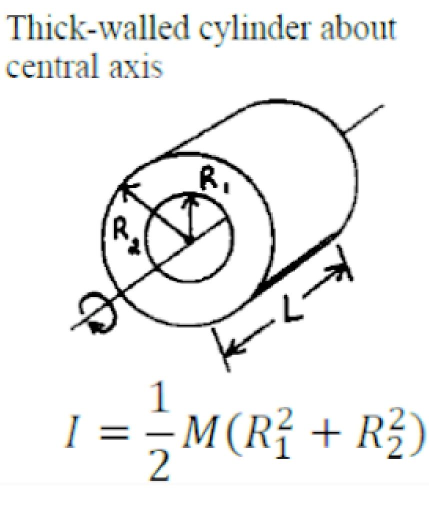


Figure 24: Moment of Inertia Formula [6]

The graph of figure 25 represents the individual moment of inertia of the main components in the drive system and figure 26 represent the reflected moment of inertia which is the inertia from the load that is translated through the drive components back to the drive input of the axis of motion.

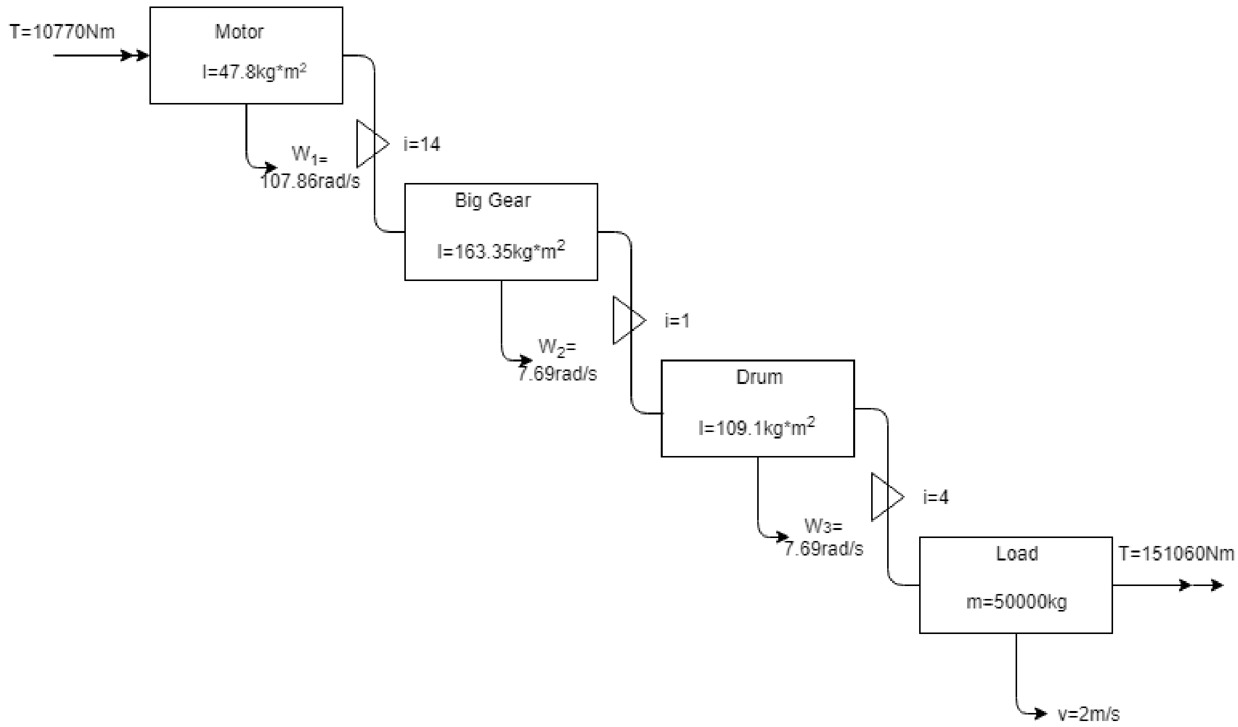


Figure 25: Scheme of Drive System

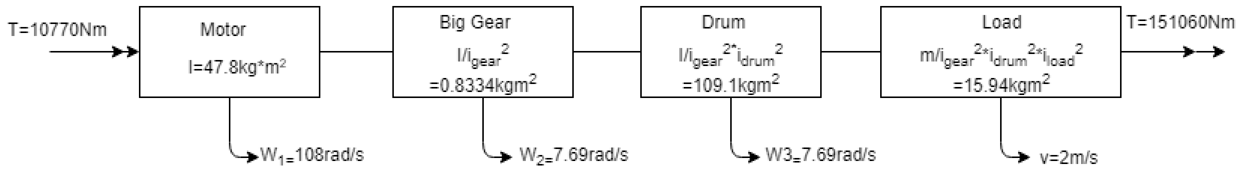


Figure 26: Scheme of Drive System

Lastly the inertia ratio can be found. The inertia ratio is the sum of the reflected inertia's divided by the motor inertia. In this case this is $125.87/47.8 = 2.6$. This is a reasonable value for a motor that is always active, stopping and starting, with relatively precise movements.

6.2 Machine Elements for Trolley

6.2.1 Shafts dimensions

The wheels on the trolley are directly driven by motors, which means they will be slotted. To determine the diameter of these shafts, Equation 72 was used.

$$D = \left(\frac{32N}{\pi}\sqrt{\left(\frac{K_t M(x)}{S'_n}\right)^2 + \frac{3}{4}\left(\frac{T}{S_y}\right)^2}\right)^{\frac{1}{3}} \tag{72}$$

Firstly, some quantities in this equation can be derived from the material used for the shafts, which is **Stainless steel, duplex, ASTM**. Their values can be seen in Table 15.

Table 15: Characteristics of ASTM CD-4MCu Steel

S_y	=	505	MPa
S'_n	=	335	MPa

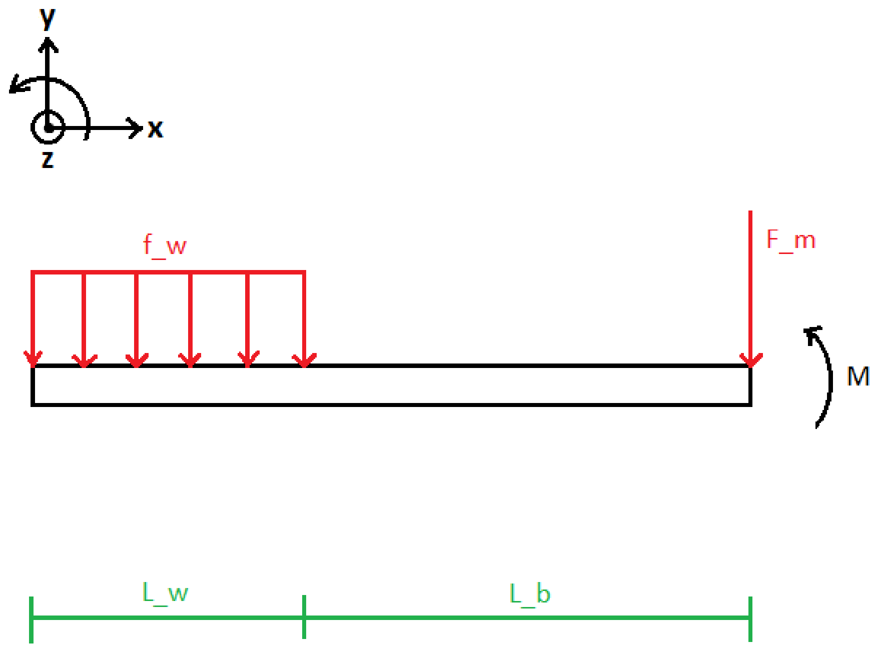


Figure 27: FBD of the trolley wheel shaft

The moment $M(x)$ was calculated using the FBD in Figure 27 along with the equilibrium equations following from this FBD. In this FBD, f_w is the total load on the wheel as distributed load, F_m is support reaction force of the motor and M is the support reaction moment of the motor.

f_w was already known, since it was the total load (50 tons) divided by 4 (4 wheels) and divided by the width of the wheels L_w, which is chosen to be 6 cm. This gives an f_w of 204,375.0 N/m. The length L_b was chosen to be 8 cm. Depending on the width of the bearings used on these shafts, this length might change. The shaft was "cut" two times over the lengths x_1 and x_2, which gave two different equations for $M(x)$:

$$M_1(x) = -f_r L_w x_1 + f_r L_w(\frac{1}{2}L_w + L_b) \tag{73}$$

$$M_2(x) = 0.5 f_r(x_2 - L_b)^2 - f_r L_w x_2 + f_r L_w(\frac{1}{2}L_w + L_b) \tag{74}$$

From the equilibrium equations of the FBD in Figure 27, F_m and M were determined and expressed in f_r, L_b and L_w. They are already filled in in the equations of $M_1(x)$ and $M_2(x)$.

For the calculations of the diameter of the shaft, the maximum moment will be used, since the shaft should be able withstand this moment. $M_1(x)$ is a linear function with a negative slope, so its value will be maximal for $x = 0$. Filling this in gave a maximum $M_1(x)$ of 13,488.75 Nm. $M_2(x)$ is not a linear function, so its derivative was calculated and set equal to 0 to find for which x the function has an extreme value. The x following from doing this was $x = L_w + L_b$. However, this gave a minimum value of $M_2(x)$ instead of a maximum. The value for which $M_2(x)$ was maximal turned out to be at $x = 0$, which gave a value of 20,028.75 Nm. Therefore, the value for $M(x)$ in Equation 72 is 20,028,750 Nmm.

$$\omega = \frac{v_{max}}{r} \tag{75}$$

The torque T is calculated using several equations. To ensure the shaft can withstand its working conditions, the maximum torque will be used. Firstly, the angular velocity was calculated using Equation 75, in which v is the velocity and r is the radius of the wheels. To eventually obtain the highest torque, ω should be as small as possible and therefore v should be as small as possible. The minimum velocity was given, being equal to 180 m/min (= 3 m/s) [1]. The minimum value is used because this value will eventually give the highest torque which the shaft should be able to withstand. An assumption was made regarding the radius, being 15 cm. When filling these values in into Equation 75, the angular velocity ω becomes 20 rad/s.

$$T = \frac{P}{\omega} \tag{76}$$

Now that ω is known, it can be used to calculate the maximum torque T by using Equation 76. The necessary power P was already calculated in the section 3, *Conceptual Design*, and was equal to 1,270.40 kW for steel wheels. Since it is chosen to use 4 wheels, P will be equal to 1,270.40 / 4 = 317.60 kW per wheel. Filling the two known variables in into Equation 76 gives T = 15,880.00 Nm.

In order to calculate D in Equation 72, only N and K_t need to be determined. N is the design, which average value is equal to 2. The quantity K_t is the stress concentration factor, which is taken to be equal to 1.6, because this value is used for shafts which are slotted. The trolley wheel shaft will be slotted on both ends, because the shaft needs to be directly connected to the motor on one side and a wheel needs to be fitted on the other side. An overview on the non-material related quantities and their values can be seen in Table 16.

Table 16: Values for calculating the thickness of the shaft

T	=	15,880.00·10^3	Nmm
N	=	2	
K_t	=	2.5	
$M(x)$	=	20,028.75·10^3	Nmm

Now that all necessary values are known to calculate the diameter of the shaft, these can be inserted into Equation 72. This leads to a diameter D equal to **12.65 cm**.

6.2.2 Bearings

The shafts going through the wheels of the trolley are supported by a bearing or possibly multiple bearings, which are connected to the frame of the trolley. This was done because the load of the spreader and container would otherwise be directly on the motors of the wheels.

To be able to make a proper choice, some calculations were performed, starting with calculating the life rating in hours L_{10h}. It is determined that the bearing should be able to at least last a year and should therefore be replaced every year. This leads to an L_{10h} equal to 8,766 hours.

$$L_{10m} = \frac{L_{10h} 60n}{10^6} \tag{77}$$

Now that L_{10h} is known, the life rating in 10^6 revolutions L_{10m} can be calculated by using Equation 77, in which n is the rotational speed in RPM. Firstly, the rotational speed n will be computed by converting the previous calculated ω from rad/s to RPM. The ω used in this bearing calculation is different than the ω used in the shaft diameter calculation. To find to largest dynamic load rating C on the bearing, ω should be as high as possible instead of as small as possible. This holds for the largest v, which equals 210 m/min (= 3.5 m/s). Using Equation 75, ω becomes 23.33 rad/s, which leads to a rotational speed of 222.82 RPM. Filling in Equation 77 gives an L_{10m} equal to 117.19 ·10^6 revolutions.

$$L_{10m} = \left(\frac{C}{P}\right)^k \tag{78}$$

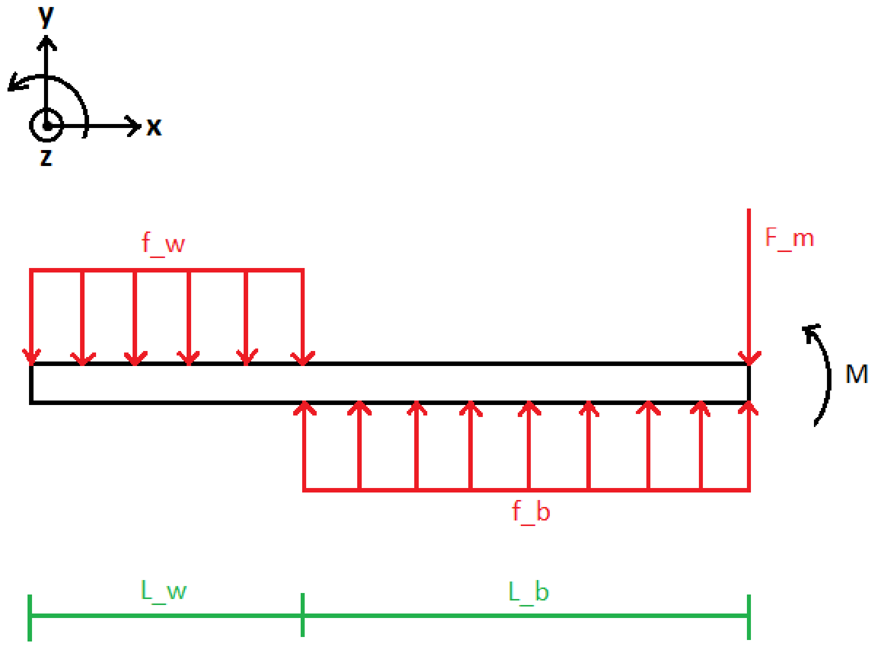


Figure 28: FBD of the trolley wheel shaft with a bearing

Since L_{10m} is known, the basic dynamic load rating C can now be determined by using Equation 78, in which P is the equivalent dynamic load in N and k is a constant. An upgrade of the FBD in Figure 27 can be seen in figure 28, in which the supporting force of the bearing as distributed load, f_b, is added. The distance between the wheel and the bearing and the motor and the bearing is 0 in this FBD, but in reality there will be some space between them to allow thermal expansion to happen. Furthermore, F_m is set equal to 0, so all of the force of the total load f_r will be on the bearing.

P is equal to f_b times L_b, which is the total force acting on a wheel divided by 4, resulting in 122,625 N. The constant k is equal to $\frac{10}{3}$ for cylindrical roller bearings, which is the type of bearing which will be used, since the bearing only has to deal with radial loads. Filling in Equation 78 leads to C = 511,975.63 N, which is approximately **512 kN**.

Since the bearing only has to deal with radial loads, a cylindrical roller bearing will be chosen. As can be seen in Figure 28, there is a moment on the right. Since all of the load in on the bearing, they should be able to withstand this moment. Bearings tend to turn less smooth when they have a moment on them. To prevent this, two bearings next to each other will be used instead of just one. This means that the dynamic load per bearing splits up, meaning that C becomes 256 kN. It would be even better to place an additional bearing on the other side of the wheel, but this gave result in too many unknowns in the equilibrium equations of the new FBD.

Considering this dynamic load on the bearing and the diameter of the shaft, a bearing was chosen using the bearing selector from the company SKF [8]. After some research, it was determined to use the following cylindrical roller bearing: **CRL 40 AMB**. Some information about the bearing can be seen in Table 17 and a picture with indications for the dimensions can be seen in Figure 29. The bearing does not perfectly fit on the shaft, but the shaft could be made somewhat larger, so that it would fit better.

Table 17: Information about bearing used for shaft of trolley wheels [8]

Type	**d**	**D**	**B**	**C**
	mm			kN
CRL 40 AMB	127	228.6	34.925	297

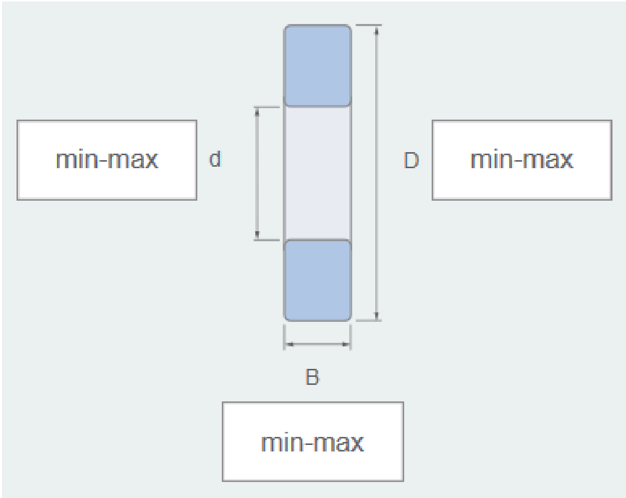


Figure 29: Bearing dimensions [8]

6.2.3 Power requirement trolley motor

The max power is found during the acceleration phase, equation 79 below is used to calculate the total force.

$$F = Ma + F_r \tag{79}$$

$$F_r = \mu R \tag{80}$$

Where R is the normal force, and μ the coefficient of friction.

Applying a rolling coefficient of 0.0025 for steel on steel and an acceleration of 0.5 ms^{-2} a total force of 26kN is found, when applying the angular velocity of the 30cm diameter wheel a total power of 91 kW is found, translating to 22.9 kW per wheel.
A low speed 20kW motor was found fitting the conditional requirements, can turn at 220rpm, which is nearly the exact requirement. The torque is also 800Nm, only slightly below the calculated requirement. This motor requires an AC connection

6.2.4 Inertia of the Trolley

The inertia reflected on the motor of the trolley is dependant on the inertia of the trolley wheels and on the shaft connecting the two.

$$I_w = \frac{1}{2}M(R_1^2 + R_2^2) \tag{81}$$

$$I_s = \frac{1}{2}MR^2 \tag{82}$$

Applying the dimensions of the wheel, 30cm outer and 13cm inner diameter to equation 81 finds an inertia of 0.3636 kg/m^2. Doing the same for the shaft with equation 82 finds an inertia of 0.0310 kg/m^2. As the ratio is one to one the reflected inertia is the sum of the two inertia's.
I_{ref} = 0.3946 kg/m^2
Lastly the inertia of the motor itself must be used to calculate an inertia ratio. This was not given by the manufacturer, but by looking at pictures to estimate size, and using equation 81 an approximate inertia is found to be 1 kgm^2. This can then be used to find an approximate inertia ratio of 0.4. This would realistically be closer to 1 as this approximation places all the mass at the edge of the motor instead of some mass closer to the center.

6.3 Machine Elements for Boom and Supporting Beam

As the boom is only lifted and lowered a few times a day, it is not necessary to have a form of roller bearing. Instead a plain bearing is used. This will be lubricated periodically to reduce the coefficient of friction between shaft and bearing.

7 3D Models

7.1 Full assembly

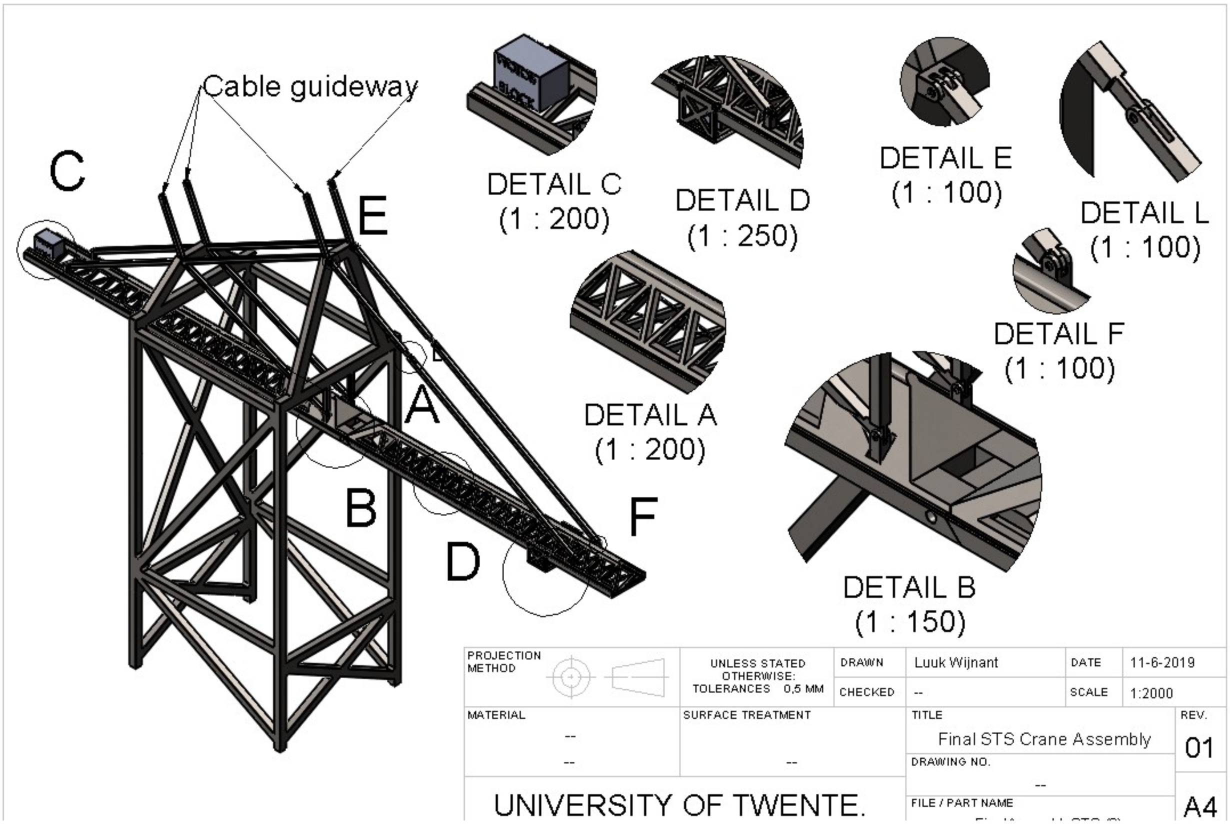


Figure 30: Full assembly of the STS crane

In Figure 30, the full assembly is shown and at the right a few detailed views. (A) The boom structure with the rails for the trolley, a truss system is used to decrease weight while maintaining strength.(B) The main hinge and a hinge connection between the back boom and the frame, to add rotation as a degree of freedom to decrease stresses.(C) The motor block attached to the cables to pull the boom up.(D) The trolley laying on the rails of the boom.The detailed views of the hinges of the retraction system can be seen on (L)(E)(F). At last there is a cable guideway for the cables from the motor block at the back boom to a connection point at the end of the boom.

7.2 Hoist

In Figure 31, the design of the hoist can be seen, at the left a detailed view of the gear box and at the right an isometric view of the entire hoist structure, which is later attached to the trolley. The electrical motors have been modelled as a black box using the dimensions from 9. The drum is driven by the motor using a gear ratio of 2 and after that 7 to accommodate for the large ratio, as using a ratio of 14 at once would result in gear too large of size. One of the smallest options to be used with the 135mm thick motor shaft was found to be around a gear with 25 teeth and a module of 8. The biggest gear dimension is then 1400mm, the 175 teeth multiplied by the module of 8.

As determined in the section *Machine Elements*, it was chosen to use a spherical roller bearing and a cylindrical roller bearing on each of the two shafts (drums). The other bearings in the gearbox will also be a combination of spherical roller bearings (because of the axial wind force and to hold the shaft in place) and cylindrical roller bearings (because of the radial load). However, for the model in SolidWorks, just general ball bearings were used just to show where the actual bearings are located. Furthermore, what cannot be seen in the figure is that both of the motors are connected to the frame with bolts.
Calculating the inertia of the two main shafts, the drum and the motor shaft, is done by using the following equations.

$$m = \rho V \tag{83}$$

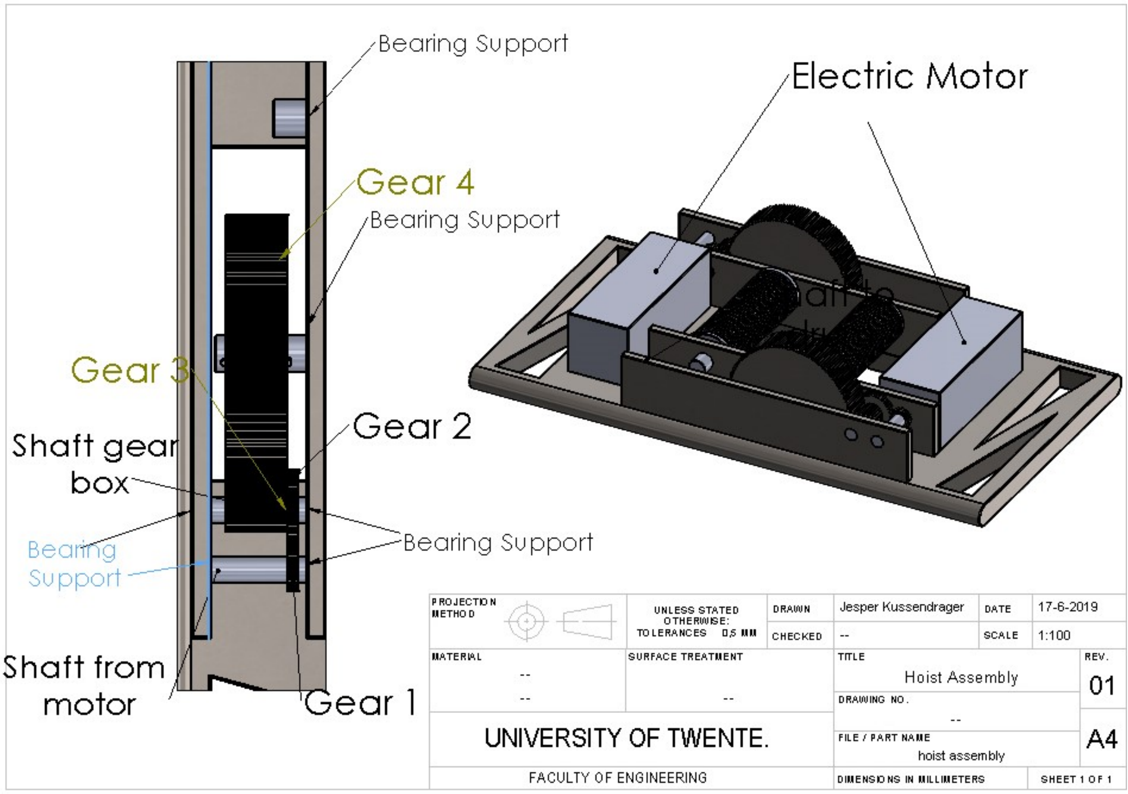


Figure 31: Hoist assembly

$$I = \frac{1}{2}mr^2 \tag{84}$$

Calculating gives that both of the two shafts have a inertia of 130.11 kgm^2 each.
The polar moment of the inertia of the shaft can be calculated using the following equations.

$$J = \frac{\pi}{2}r_0^4 \tag{85}$$

$$\tau = \frac{Tr}{J} \tag{86}$$

The torque on the shafts was already calculated, $T = 140.820KN/m$. Filling the values in the above given equations gives that $\tau = 1.41MPa$.
Second moment of Area can be calculated using the following equation.

$$I_z = \frac{\pi D^4}{64} \tag{87}$$

7.2.1 Trolley

In Figure 32, the design of the trolley is shown, as well as its connection to the hoist. A truss like system was chosen to minimize the amount of mass used for the side frame, which lowers the total stress on the boom. Because of the relatively small needed force to roll along the boom, the electrical motors can be installed within the frame of the trolley.

The power delivery of the trolley and hoist is done using a cable connected to the back of the boom. This cable is held to tension using a design as seen in Figure 33. Because the cables are fixed on their supports and the supports have freedom to move along the axis of the boom, the trolley is still also free to move while the lowest point of the cables is always at least close to the trolley. This way they can't hinder the movement of trolley or come in contact with vessels.

The hoist is connected to the frames of the trolley using a bolted connection. This connection will be

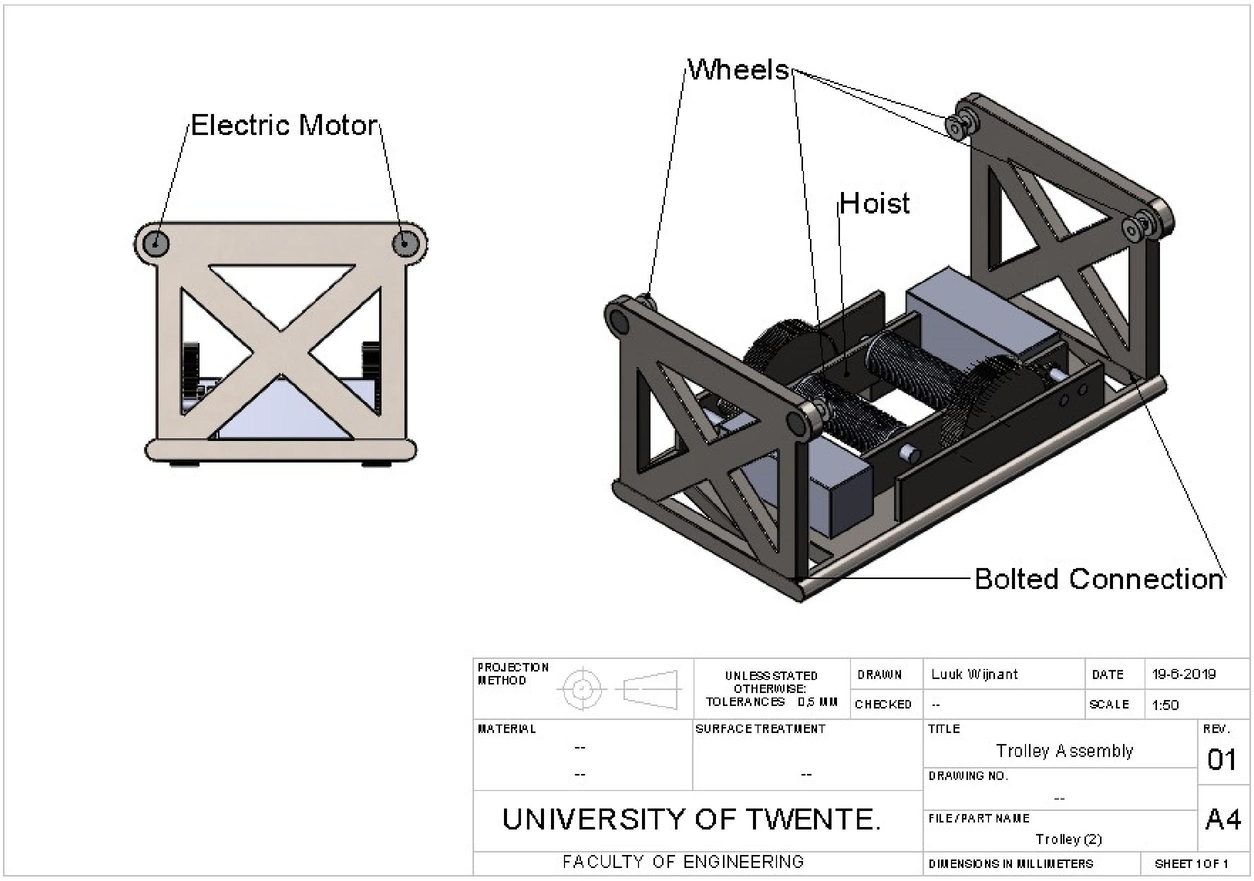


Figure 32: Trolley design

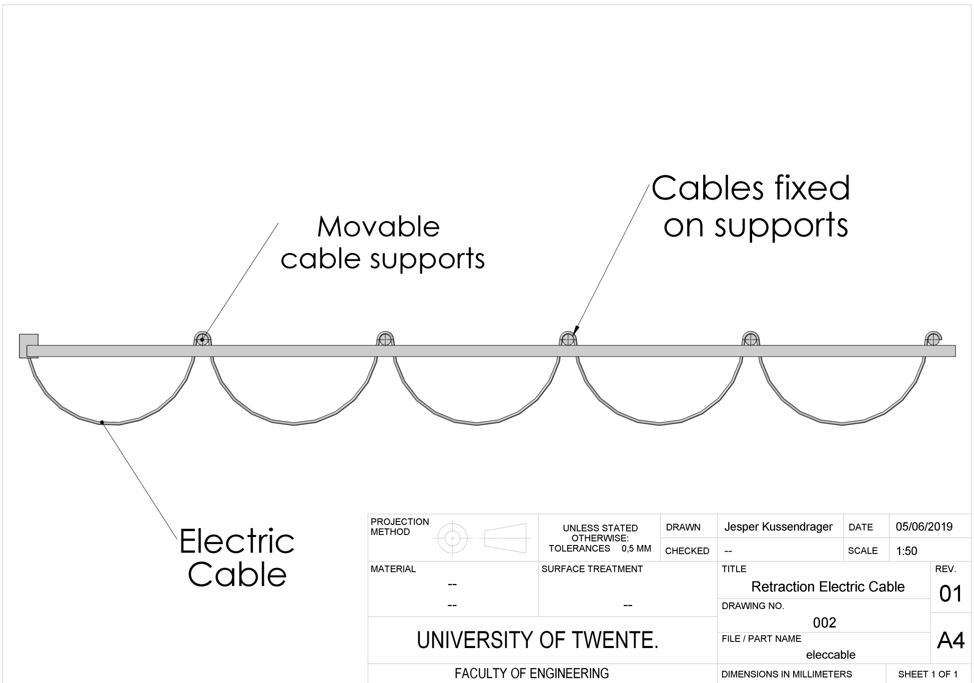


Figure 33: Retraction mechanism electric cable

exposed to a lot of shear stress due to the weight of the container. The connection wheels and motor will also be exposed to this large shear stress, bearings have to be chosen accordingly to ensure safe use of the electric motor.

7.3 Boom

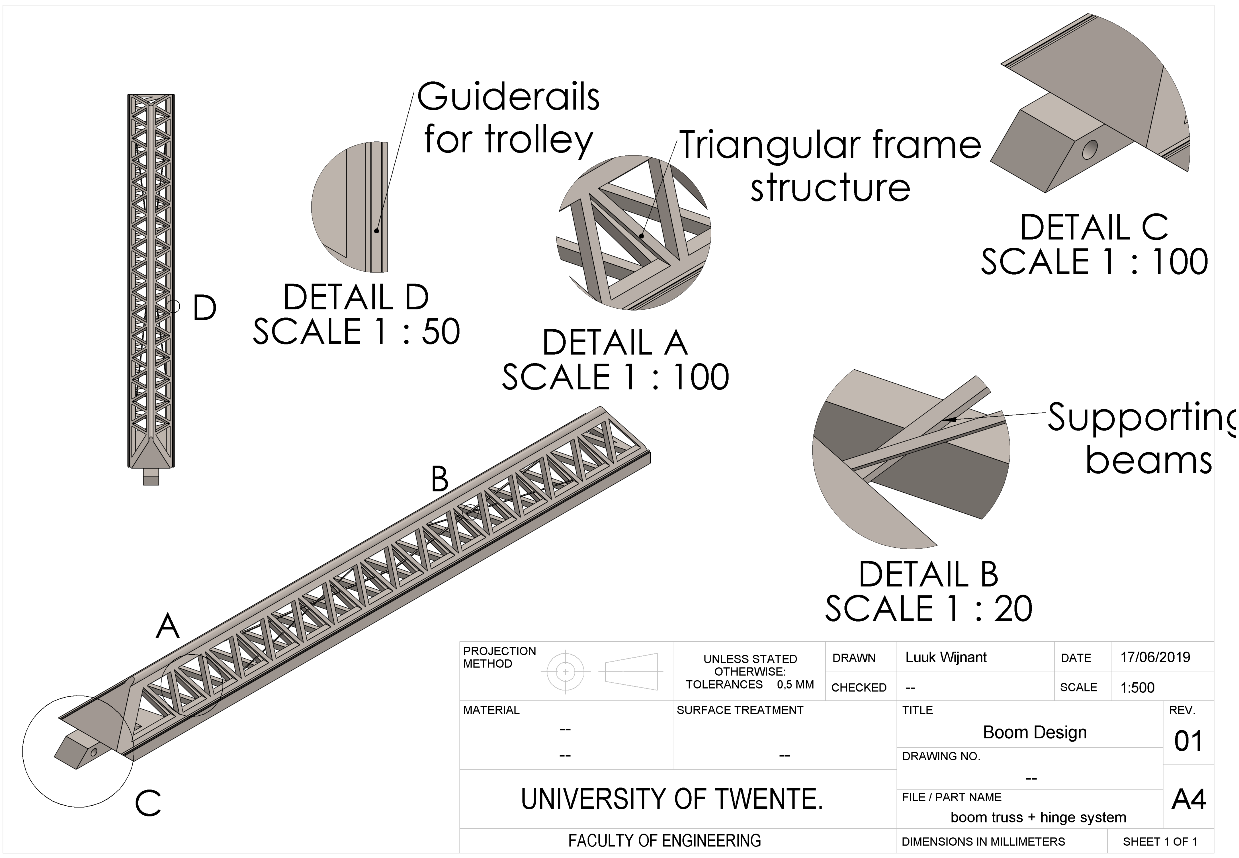


Figure 34: Front part of boom design

For the boom, a triangular frame was chosen as determined from table 3, although slightly altered for ease of modelling. On the side their are guide rails to ensure the trolley follows the right track. The front part of the boom is connected to the back part using a hinge, to maintain a locked position the sloped edge is used to rest against an opposite sloped angle in the back part of the boom. In the middle of the boom, tiny support beams are placed to greatly help stability and strength of the structure as was concluded from the FEM analysis.

7.4 Detailed Technical drawing

A detailed technical drawing of the shaft in the hoist carrying the load of the container is shown below. There are a few important conditions; Firstly the surface roughness of the drum is 0,5 micrometers. This is chosen, because of the interference between the bearing and the shaft. Furthermore, an interference fit is chosen for the shaft and the connection with the gear. This is done to ensure that the gear does not slip in the direction of the shaft. A key-way is also include to make sure that the gear does not slip on the shaft, avoiding jumps in motion or complete failure.

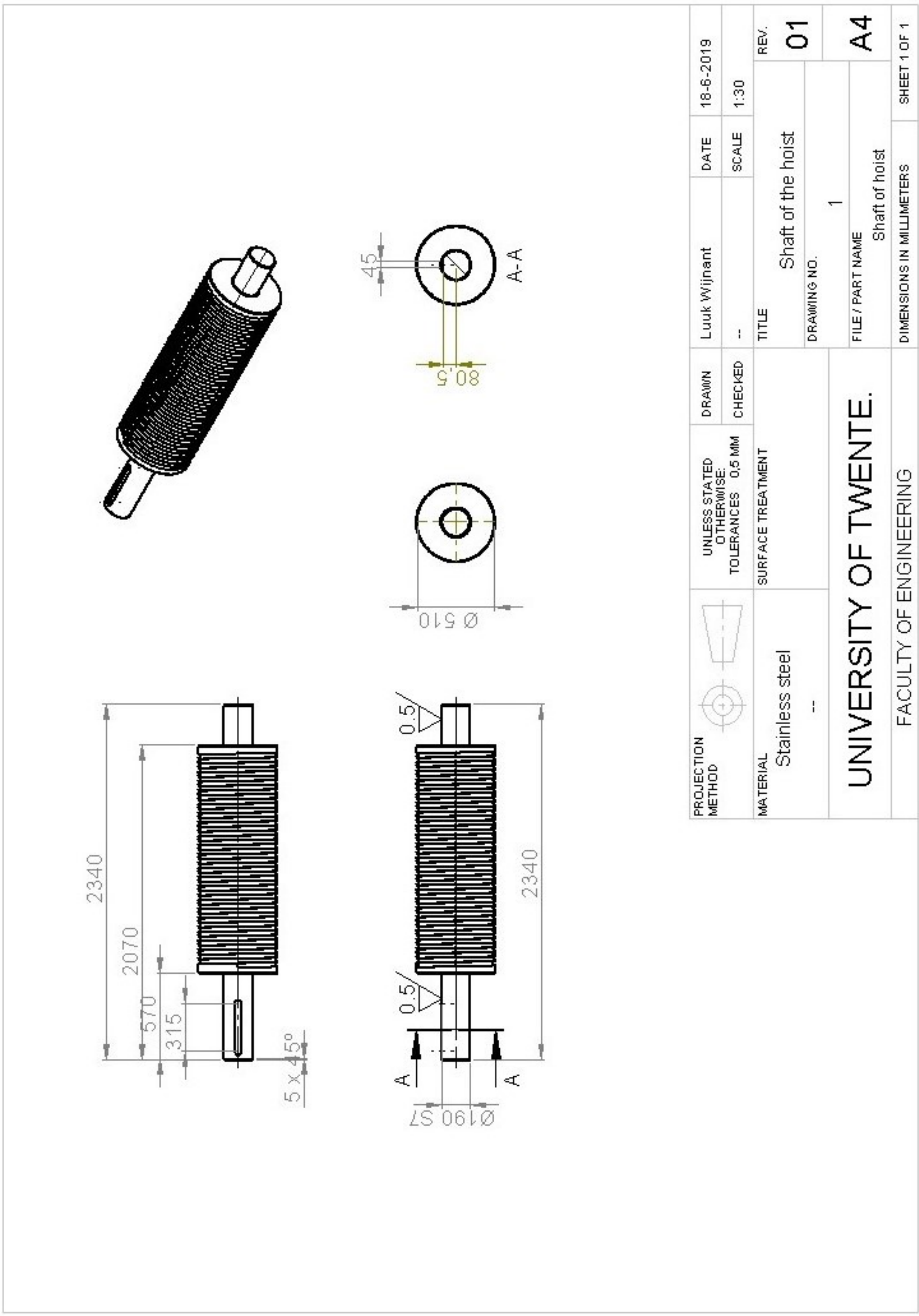


Figure 35: Detailed Drawing

8 Conclusion

In conclusion, it is seen that the goal to design a crane within the constraints of the problem definition has been fulfilled. This can be broken down into all the aspects that were looked at. Firstly, from the FEM analysis it is seen that the maximum displacements are well within their requirements, and that the stresses in elements are low enough to ensure the safety of the crane and those around it. This is also seen in the machine elements analysis, where critical parts are analyzed with multiple safety factors to ensure their life span would fulfill that of the crane itself and also be safe, such that if failure does occur, it is in a safe manner. The latter of course primarily relying on the materials that were chosen. Parts that could cause catastrophic failure if brittle fracture were to occur, use materials with a higher fracture toughness. This is seen in the gears of the hoist for example, where jamming of the gears is a much more safe failure than if brittle fracture were to occur and the hoist would all come falling down. The analysis as a whole allows us to ensure critical parts meet the required function. Lastly, Solidworks drawings and models were made to help visualize the models, allowing some dimensions to be realized and set accordingly. In some cases it even gives extra information about the material size, shape, surface and interaction with other parts. From all of this, it is seen that the designed crane meets the requirements of a working high profile ship to shore crane, whilst carrying out its function safely.

References

[1] dr. J. Hazrati, "Project description - design and mechanics."

[2] dr. R. Bosch, "Calculating friction force."

[3] "Magnetic levitation," 2019. [Online]. Available: https://en.wikipedia.org/wiki/Magnetic_levitation

[4] "Steel wire ropes," 2019. [Online]. Available: https://www.mennens.nl/en/products/steel-wire-rope-6x36ws-iwrc-p58907

[5] "Crane wheels," 2019. [Online]. Available: https://www.karl-georg.de/en/activities/crane-wheels

[6] R. L. Mott, "Machine elements in mechanical design (fourth edition in si units)."

[7] "Moments of areas." [Online]. Available: https://slideplayer.com/slide/2825728/

[8] "Cylindrical roller bearing." [Online]. Available: https://www.skf.com/sg/products/bearings-units-housings/roller-bearings/cylindrical-roller-bearings/high-capacity-cylindrical-roller-bearings/high-capacity/index.html?designation=NCF%202228%20ECJB

[9] Nidec, "Nidec motor catalog." [Online]. Available: http://acim.nidec.com/motors/usmotors/products-and-services/catalogs-and-literature

Appendices

A FEM Data

In the first part of the appendix the data from the different iterations of the FEM models are shown.

A.1 Frame Iterations

Below the 4 different iterations of the frame structure can be seen with the changing stresses in each part.

ElementNR	S1	S2	S3
1	9.8353e+08	-3.9339e-06	-1.842e+06
2	3.2029e+08	1.1474e-06	-7.4769e+05
3	1.7077e+09	-3.6716e-05	-10522
4	1.266e+09	-3.4094e-05	-7.0034e+05
5	5.9755e+08	-1.0133e-06	-2.0345e+06
6	6.7372e+08	4.7058e-05	-1.0791e+05
7	8.4907e+07	4.4703e-08	-4.327e+05
8	4.6797e+05	-1.8626e-06	-1.903e+08
9	5.2716e+06	-2.563e-06	-1.0816e+09
10	1.2414e+07	-1.4901e-07	-3.0031e+08
11	1415.9	0.0005538	-2.4468e+08
12	1.1204e+06	-3.6955e-06	-7.164e+08
13	11197	-0.00010417	-3.3322e+08
14	3.124e+05	5.9009e-06	-6.4501e+08
15	26719	5.6997e-07	-5.2963e+07
16	1.6744e+08	-4.8429e-07	-2.0649e+05
17	1.651e+06	-4.4703e-08	-1.2071e+08
18	8.7617e+07	-1.1176e-08	-1.1591e+06
19	1.2704e+07	5.9605e-08	-2.835e+08
20	5.8406e+08	3.8743e-07	-1.9804e+07
21	5.1491e+08	-0.00011095	-7206.2
22	8323.1	-0.00020733	-7.6668e+08
23	7.438e+08	1.2696e-05	-40906

Figure 36: Frame Iteration 1

ElementNR	S1	S2	S3
1	5.974e+08	-3.2187e-06	-2.6502e+05
2	2.0685e+05	-8.3596e-06	-2.8064e+08
3	5.5748e+07	4.2439e-05	-106.67
4	3.6787e+07	5.5879e-09	-2.4912e+06
5	66224	4.9174e-07	-4.7602e+07
6	3.9006e+08	8.8364e-06	-3.2656e+05
7	1.2946e+07	-1.8626e-09	-3.2904e+06
8	6155.4	1.1716e-05	-6.742e+07
9	1.6406e+05	1.359e-05	-6.0494e+08
10	2.8174e+08	-4.3213e-07	-6.7398e+05
11	1331.4	9.121e-05	-2.6021e+08
12	1.8496e+06	-5.0291e-08	-5.0728e+07
13	1.7503e+07	-3.7253e-09	-1.1294e+06
14	3.0214e+05	-1.1489e-05	-3.8767e+08
15	3.0512e+06	-7.9162e-09	-1.361e+07
16	6.573e+07	2.6077e-06	-12238
17	96378	-1.1642e-09	-5.5421e+06
18	5.7673e+06	2.5611e-09	-73765
19	1.1255e+08	-1.8626e-08	-2.1636e+07
20	2.4777e+07	-6.333e-08	-1.1415e+08
21	3.3229e+08	-1.2368e-06	-4745.7
22	14565	3.9801e-05	-2.3454e+08
23	2.428e+08	3.0845e-05	-13011

Figure 37: Frame Iteration 2

ElementNR	S1	S2	S3
1	1639.8	-7.3807e-08	-5.1833e+06
2	1.9408e+06	-9.3132e-10	-2.4983e+05
3	3.2137e+07	-0.00066747	-12.922
4	2.0511e+07	0.0001183	-22.194
5	21651	-4.4238e-09	-3.9896e+06
6	1.0113e+06	-9.8953e-10	-18721
7	1.5735e+06	-2.3283e-10	-49272
8	1.9359e+05	-1.0186e-10	-53571
9	8.8278e+06	-4.6566e-09	-2.9437e+05
10	5.7192e+06	-5.1223e-09	-6.5786e+06
11	1336.2	-0.00028744	-2.5927e+08
12	158.32	0.0001878	-6.4799e+07
13	3.2026e+07	1.3039e-08	-2.057e+05
14	49632	-3.4925e-10	-1.6568e+06
15	23923	-1.8626e-09	-1.5281e+06
16	78885	-2.3283e-10	-1.9095e+06
17	5598.9	-9.0222e-10	-5.7167e+05
18	9.1306e+05	-7.567e-10	-21646
19	2.8591e+06	-8.1491e-10	-67992
20	6.7025e+05	-1.1176e-08	-2.6407e+07
21	3.3485e+06	-1.8394e-08	-4043.7
22	542.29	-9.3389e-05	-7.8037e+07
23	6.8583e+06	-3.7719e-08	-1236

Figure 38: Frame Iteration 3

ElementNR	S1	S2	S3
1	2.0077	-1.4928e-05	-3.4522e+06
2	1.1435e+05	9.3132e-10	-61.01
3	9.0656e+06	-5.7602e-05	-15.649
4	3.3254e+06	-6.2268e-06	-39.296
5	728.26	-2.6921e-10	-1.0237e+05
6	1.4664e+05	-5.8208e-11	-2736
7	4.6913e+05	-1.8335e-09	-1211.2
8	1.5139e+05	-3.638e-11	-10994
9	3.4997e+06	-2.3751e-06	-16.736
10	1291.1	-5.8208e-11	-1.0209e+05
11	3.0168e+06	1.7884e-06	-25.766
12	11.968	2.3805e-06	-4.3371e+06
13	77339	3.6453e-09	-5.4518
14	103.35	2.1755e-09	-1.0032e+05
15	4577.4	-1.6371e-11	-45139
16	2614.9	-5.0932e-11	-85631
17	3776.7	3.4925e-10	-4.5332e+05
18	81323	-6.5484e-11	-4022.1
19	1.1276e+05	1.2078e-09	-186.3
20	1.2945e+05	-2.9104e-11	-1522.9
21	8.3777	-5.2714e-09	-95297
22	7.4707e+06	-7.9973e-06	-0.99015
23	5.9674e+06	5.1223e-08	-104.37

Figure 39: Frame Iteration 4

A.2 Boom Iterations

In this section the stresses of the different iterations of the FEM model of the boom are displayed. The following stresses correlate to the stresses in each element of the boom. When looking at a side view of the crane:

1 - 15 = Front lower length 16 - 30 = Back lower length 31 - 45 = Top length 46 - 61 = Bottom straight 62 - 77 = Bottom diagonal 78 - 93 = Front vertical 94 - 109 = Back vertical 110 - 114 = Front diagonal 115 -

129 = Front diagonal back 130 - 144 = Back diagonal back 145 - 159 = Back diagonal 160 - end = Extra
The extra elements are extra elements placed within boom or as extra elements on the bottom of the boom, the latter of which was taken out after the first iteration.

ElementNR	Stress
1	1.7667e+06
2	1.5856e+07
3	2.7854e+07
4	3.759e+07
5	4.5144e+07
6	5.0107e+07
7	5.5212e+07
8	4.2528e+07
9	1.0342e+07
10	-2.1529e+07
11	-5.5459e+07
12	-9.5204e+07
13	-1.1421e+08
14	-8.3522e+07
15	-1.4959e+07
16	5.1666e+07
17	4.2357e+07
18	3.3155e+07
19	2.6571e+07
20	2.2196e+07
21	1.9746e+07
22	2.173e+07
23	1.1376e+07
24	-1.3032e+07
25	-3.2817e+07
26	-5.0238e+07
27	-6.8564e+07
28	-6.4147e+07
29	-8.1605e+07
30	-1.7789e+07
31	6.7611e+06
32	4.2467e+06
33	1.1903e+06
34	-1.8149e+06
35	-4.6907e+06
36	-8.4064e+06
37	-6.988e+06
38	1.8819e+06
39	4.7286e+07
40	9.8256e+07
41	1.4935e+08
42	1.9405e+08
43	2.7916e+08
44	1.1409e+08
45	-8.4162e+07
46	0
47	-4.941e+06
48	-4.2188e+06
49	-4.635e+06
50	-4.8486e+06
51	-4.9909e+06
52	-5.518e+06
53	-5.2484e+06
54	-2.0269e+06
55	2.164e+06
56	5.9095e+06
57	9.5104e+06
58	1.5854e+07
59	0
60	-9.61e+07
61	2.748e+07
62	1.4091e+07
63	1.1091e+07
64	9.703e+06
65	7.976e+06
66	6.3615e+06
67	4.5548e+06
68	3.3706e+06
69	6.7317e+05
70	-3.8114e+06
71	-7.5926e+06
72	-1.1579e+07
73	-1.5199e+07
74	-2.2252e+07
75	-5.4296e+07
76	-2.8526e+07
77	-7.3569e+06
78	-6.7763e+06
79	-4.5841e+06
80	-2.7307e+06
81	-7.6499e+05
82	1.0086e+06
83	3.4046e+06
84	4.2874e+06
85	3.3831e+06
86	3.1821e+06
87	2.7758e+06
88	2.7365e+06
89	-7.3612e+05
90	-5.5609e+07
91	-3.292e+07
92	-4.2915e+06
93	-1.2545e+06
94	-3.3229e+06
95	-4.0925e+06
96	-4.7115e+06
97	-5.5774e+06
98	-2.8392e+06
99	-2.0318e+07
100	-3.2235e+06
101	-8.8026e+06
102	-9.6684e+06
103	-1.5295e+07
104	6.5402e+06
105	-1.3812e+08
106	1.168e+08
107	-2.0573e+07
108	-3.01e+05
109	-7.2817e+06
110	-4.2448e+06
111	-3.2472e+06
112	-2.1277e+06
113	-1.8335e+06
114	1.3361e+06
115	-1.5029e+07
116	3.9996e+05
117	-6.1423e+06
118	-8.8148e+06
119	-1.6187e+07
120	3.1329e+05
121	-1.3321e+08
122	1.1648e+08
123	-1.9595e+07
124	5.0583e+06
125	6.508e+06
126	6.2336e+06
127	6.1252e+06
128	5.6702e+06
129	5.8579e+06
130	1.0792e+06
131	-2.61e+07
132	-2.0735e+07
133	-2.0713e+07
134	-2.1003e+07
135	-1.5268e+07
136	-4.8504e+07
137	1.5698e+08
138	2.104e+07
139	-6.3091e+06
140	-3.5532e+06
141	-2.5216e+06
142	-1.3239e+06
143	-4.7287e+05
144	1.0209e+06
145	-2.4517e+06
146	2.3296e+07
147	2.9967e+07
148	3.1296e+07
149	3.2312e+07
150	3.9352e+07
151	7.4222e+06
152	-1.6247e+08
153	1.5567e+07
154	9.1538e+06
155	5.3264e+06
156	4.2044e+06
157	2.8235e+06
158	1.5552e+06
159	1.1646e+06
160	-3.5533e+06
161	1.9491e+07
162	2.5129e+07
163	2.6034e+07
164	2.7029e+07
165	3.3572e+07
166	5.9282e+06
167	-1.5689e+08
168	1.7743e+07
169	3.4094e+06
170	8.8806e+05
171	1.0721e+06
172	9.973e+05
173	1.0351e+06
174	1.9505e+06
175	-1.4614e+06
176	-2.5609e+07
177	-1.8665e+07
178	-1.6454e+07
179	-1.4153e+07
180	-6.3041e+06
181	-3.2642e+07
182	1.5518e+08
183	1.8459e+07
184	1.6377e+07
185	2.2707e+07
186	6.8529e+06
187	1.5681e+07
188	6.6413e+07
189	-1.1587e+07
190	5.3514e+07

ElementNR	Stress
1	-1.674e+07
2	-5.3496e+06
3	5.0872e+06
4	1.3352e+07
5	1.9695e+07
6	2.3891e+07
7	2.7071e+07
8	2.2376e+07
9	9.6684e+06
10	-4.22e+06
11	-1.9078e+07
12	-3.8183e+07
13	-3.7569e+07
14	-1.0137e+08
15	-1.056e+07
16	2.6785e+07
17	1.7745e+07
18	9.5679e+06
19	3.5696e+06
20	-4.6224e+05
21	-2.6322e+06
22	-2.0318e+06
23	-4.3685e+06
24	-9.7818e+06
25	-1.2582e+07
26	-1.2386e+07
27	-1.2103e+07
28	-1.3208e+06
29	-7.8792e+07
30	-1.4177e+07
31	1.6255e+06
32	7.4494e+05
33	-2.2755e+05
34	-1.1924e+06
35	-2.1136e+06
36	-3.3189e+06
37	-2.7703e+06
38	3.6707e+06
39	1.8818e+07
40	3.5398e+07
41	3.6675e+07
42	4.3808e+07
43	7.3405e+07
44	8.3243e+07
45	-3.2262e+07
46	0
47	-1.46e+07
48	-1.1924e+07
49	-9.2487e+06
50	-6.5728e+06
51	-3.897e+06
52	-1.2212e+06
53	1.4546e+06
54	4.1304e+06
55	6.8063e+06
56	9.4821e+06
57	1.2158e+07
58	1.4871e+07
59	0
60	-1.1889e+08
61	-1.7022e+06
62	1.7819e+07
63	1.4828e+07
64	1.1836e+07
65	8.8445e+06
66	5.8528e+06
67	2.8612e+06
68	-1.3048e+05
69	-3.1221e+06
70	-6.1138e+06
71	-9.1055e+06
72	-1.2097e+07
73	-1.5089e+07
74	-1.8122e+07
75	1.0693e+06
76	4.1006e+06
77	-2.6876e+06
78	1.8224e+06
79	-4.6833e+05
80	-1.2602e+06
81	-2.0151e+06
82	-2.6943e+06
83	-1.9306e+06
84	-8.3675e+06
87	-2.9255e+06
88	-3.256e+06
89	8.4428e+06
90	-4.2045e+07
91	7.7486e+07
92	-1.4257e+07
93	1.6544e+06
94	-3.5219e+06
95	-1.1698e+06
96	-4.4379e+05
97	3.2274e+05
98	6.5079e+05
99	1.7535e+06
100	-3.61e+06
101	6.8835e+05
102	-1.9267e+06
103	-2.2931e+06
104	-4.3074e+06
105	3.2253e+06
106	-3.4647e+07
107	7.4704e+07
108	-1.2734e+07
109	2.0035e+06
110	3.7252e+06
111	3.4455e+06
112	3.3691e+06
113	3.0451e+06
114	2.8779e+06
115	8.2288e+05
116	-8.7746e+06
117	-7.5959e+06
118	-8.0073e+06
119	-1.0442e+07
120	-9.0271e+06
121	-1.1308e+07
122	1.2026e+08
123	1.7292e+07
124	-7.2175e+06
125	-4.1897e+06
126	-3.1634e+06
127	-1.9337e+06
128	-9.5165e+05
129	1.8716e+05
130	-5.6179e+05
131	9.2155e+06
132	1.17e+07
133	1.2595e+07
134	1.1517e+07
135	1.4348e+07
136	-1.1016e+06
137	-1.0117e+08
138	1.3571e+07
139	7.4401e+06
140	4.2922e+06
141	3.3122e+06
142	2.1151e+06
143	1.0866e+06
144	4.1552e+05
145	-1.5424e+06
146	5.544e+06
147	6.9612e+06
148	7.3752e+06
149	6.0947e+06
150	8.4353e+06
151	-4.3585e+06
152	-9.3014e+07
153	1.5092e+07
154	-66988
155	-1.9088e+06
156	-1.5827e+06
157	-1.4737e+06
158	-1.1963e+06
159	-5.6124e+05
160	-1.2131e+06
161	-7.787e+06
162	-5.0638e+06
163	-3.3437e+06
164	-3.2678e+06
165	4.8913e+05
166	4.5263e+06
167	1.222e+08
168	1.3793e+07
169	-8.4128e+05
170	6.6727e+06
171	-1.7938e+06
172	6.0745e+06
173	1.659e+07
174	-2.7301e+06
175	1.9135e+07
176	3.4931e+07
177	5.6886e+07
178	-1.3672e+08
179	-3.0639e+07
180	-4.8805e+06

40	5.3194e+06	-1.7427e-05	-15.823
41	18.363	-7.2783e-07	-3.9088e+06
42	6.0488	-8.6888e-05	-5.7291e+06
43	5.7426e+06	0.00020784	-1.7414
44	0.0079255	0.0079255	-1.745e+07
45	8.7745	-4.0195e-06	-2.0626e+06

Figure 40: Frame Iteration 4 (extended)

ElementNR	S1	S2	S3
1	192.28	8.0862e-07	-3.3851e+06
2	7.4784e+05	4.0454e-09	-1435.2
3	3.1692e+07	-5.0686e-05	-110.78
4	1.558e+07	0.00021698	-8.2127
5	5596.2	-2.1828e-11	-1.8042e+05
6	2.3874e+05	-5.8208e-11	-8049.6
7	1.472e+06	5.8208e-10	-6600.7
8	3.5081e+05	-2.3283e-10	-55268
9	3.3851e+06	1.6973e-07	-192.28
10	1435.2	-1.135e-09	-7.4784e+05
11	110.78	5.0681e-05	-3.1692e+07
12	8.2123	0.00026918	-1.558e+07
13	1.8042e+05	-7.276e-12	-5596.2
14	8049.6	-1.1642e-10	-2.3874e+05
15	6600.7	1.397e-09	-1.472e+06
16	55268	-4.3656e-11	-3.5081e+05
17	13413	1.1642e-10	-1.6205e+06
18	1.6205e+06	-3.4925e-10	-13413
19	6.9593e+05	-7.5088e-09	-1809.6
20	1809.6	-1.9791e-09	-6.9593e+05
21	5.2764e+05	-1.947e-08	-101.48
22	7.4888e+05	8.4983e-09	-359.17
23	7.9811e+06	-1.0887e-06	-906.95

Figure 41: Frame Iteration 5