TAMPEREEN TEKNILLINEN YLIOPISTO
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# MECHANICAL DESIGN AND ANALYSIS OF A PORTABLE CRANE UNIT

Master of Science Thesis

Examiner: prof. Kari Koskinen
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# ABSTRACT

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The aim of this project is the design of a mechanical machine. There is the necessity in one of the research teams (UNEXMIN) of the university in the development of auxiliary equipment for their project. During the academic work the design of a workingsite crane has been achieved.

As most of machine designs during the development of the design an iterative process has been followed in order to validate each part of the unit. Research about different solutions has been done so that the decision about which type of crane suits all the requirements imposed by the project. With all that information as starting point the design has been constantly changing due to different proposals for its different mechanical components.

Throughout the whole study a validating process has been used. That is to say, a design in a software CAD has been develop based on pre-dimensioning calculations. Then the final design is subjected to various simulation tests done by finite element analysis (FEM). Once the simulations of each of the components of the crane are considered to be reliable enough from the mechanical and structural point of view then the validating process is ended and the design is finally completed.

# PREFACE

This last project in my studies signifies the end of a long lasting life as student. I would like to thank my supervisor Jussi Aaltonen for giving me the opportunity to work in such a great environment and for all of the effort he put during the guidance of my work.

I cannot go on without thanking the whole UNEXMIN research team of Tampere University of Technology for their help with the development of the thesis and for making me feel as a member of the team from the very first day.

Lastly, I would also like to thank my family. It would not have been possible to get to this point without your support. Thanks for all of your advices and your infinite tolerance towards me.

Barcelona, 22.06.2017

Bastian Bravo Gutiérrez

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# LIST OF SYMBOLS AND ABBREVIATIONS

| | |
|---|---|
| CAD | Computer aided design |
| FEM | Finite Element Analysis |
| FMEA | Failure mode and effects analysis |
| VAC | Voltage alternate current |
| 3D | Three dimensions |

| | |
|---|---|
| $A$ | Area |
| $A_t$ | Tensile-stress area of the bolt |
| b | Wide of the profile |
| $C$ | Stiffness factor |
| $C_a$ | Basic dynamic axial load rating (slewing bearing) |
| $C_{0a}$ | Basic static axial load rating (slewing bearing) |
| $C_r$ | Basic dynamic radial load rating (slewing bearing) |
| $C_{0r}$ | Basic static radial load rating (slewing bearing) |
| $d_m$ | mean diameter of the power screw |
| $d$ | Diameter (inner diameter) |
| $D$ | Diameter (outer diameter) |
| $\epsilon$ | Parameter of eccentricity |
| $E_p$ | Young module |
| $F_{i,j}$ | Forces acting on the element 'i' in the direction of the axis 'j' |
| $g$ | acceleration of gravity |
| $h$ | Height of the profile |
| $I$ | Moment of inertia |
| $k_i$ | Stiffness of the element 'i' |
| $L$ | Length of the power screw |
| $L_i$ | Length of the reference 'i' |
| $L_h$ | Estimated amount of hours of duration (slewing bearing) |
| $L_{thd}$ | Length of the threaded bolt |
| $m_i$ | mass of the load with the reference 'i' |
| $M_i$ | Moment in respect to the point 'i' |
| $n$ | Number of threads of the power screw |
| $N$ | Number of bolts |
| $N_{cr}$ | Euler buckling load |
| $p$ | pitch of the power screw |
| $P$ | Pressure applied to the hydraulic cylinder |
| $P_{eq}$ | Estimated equivalent load on the slewing bearing |
| $R_{i,j}$ | Reaction in the support 'i' in the direction of 'j' axis |
| $SF$ | Safety factor |
| $y_{max}$ | Distance from the axis of the profile to the extreme fiber |
| $\sigma_i$ | Stress on the element 'i' |
| $\sigma_{cr}$ | Critical buckling stress |
| $\tau_i$ | Shear stress on the element 'i' |
| $\alpha$ | Angle of the thread |
| $\delta$ | Inclination angle of the boom |
| $\mu_p$ | Friction coefficient of the power screw |

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# 1. INTRODUCTION

The need of one of the research teams of the university to design support equipment for their own project has led to the start of this academic work. A custom lifting device is needed so that it can be placed in a working place with special requirements.

Particularly, during the thesis a worksite crane is designed to be place inside of mine. Many challenges are introduced into the project due to this condition which will be explained progressively during the following chapters.

First of all, directives and regulations concerning to the design of machines and concretely to the use and design of worksite cranes are commented in Chapter 2.

During Chapter 3, all different feasible options that can be considered to be designed are discussed so that the reader can have an overview of an immense background that can be studied in this field.

All Chapter 3 is dedicated to present all the requirements that must be taken into account all along the project. In addition here it can be found the description of the solution that is considered the best one to be developed for the purpose of this thesis.

The whole review of the most important mechanical components are described in Chapter 4. Mainly, pre-dimensioning theoretical knowledge, simulations done by finite element analysis (FEM) and the final design of each discussed component are introduced with all details.

Concerning the Chapter 5, the analysis of the counterweight is done so that the portable crane unit do not tip over. Different situations are taken into account so that a proper estimation is of the load used as a counterweight is calculated.

During chapter 6, the whole assembly of the worksite crane is discussed with detail. The most important features are described such as the motion of the crane and the procedure that should be followed to assembly the whole unit.

Eventually, in Chapter 7, a failure Mode and Effects Analysis (FMEA) is reported. Therefore, by using this tool an approach to potential failures that may occur can be examined carefully.

## 1.1 Objectives and Scope of the Master thesis

The main goal of the project is to put into practice and all the knowledge acquired during the studies by designing a mechanical machine such as a portable crane. To achieve that goal the following task have to be done during the entire work.

1) Research and gather data about different solutions involving worksite cranes as well as directives and regulations concerning the design of lifting machine devices
2) Pre-dimension every part of the crane and select commercial products from providers if it is possible
3) Design the portable crane unit components assembly them by means of the use of a CAD software
4) Analysis using theoretical background collected throughout the career, mechanical engineering handbooks
5) Simulate using a finite element analysis software and redesign if it is needed
6) Elaborate blueprints of every component with enough detail to be manufactured
7) Extract conclusion about the final design and the overall process of the project.

# 2. DIRECTIVES AND REGULATIONS

Throughout the development of a mechanical design project various challenges are constantly being introduced to the designer. An important fact that every project has in common is that after the design is complete it will probably be introduced to the market and therefore the corresponding department of the local government will commit the engineer to respond against every possible failure that can happen in the future. That means that the final design has to not only accomplish to all the calculations but also to be in order with the regulations that the government applies in the field that the work is coping with. For that reason in every design project it is a must to do research of all laws and regulations related to the type of work that is planned to be done. By proceeding that way the designer can at least know what cannot be done when the design is about to be started and it will help to avoid further problems from the beginning. To follow the standards is also a good practice to every technical professional given that following that path it will probably complete the final work with more ease.

In this project the first directive that has been taken into account is the '***Machinery Directive 20016/42/EC'***. This directive contains an extensive list of titles and references regarding to every specific topic so that it is easier to do the read and research of the desired information. Basically the directive is divided in two type of standards: '*A-type'* and '*B-type'*. The most important references of this directive are listed at the end of this chapter.

The first of those two standards provides to the designer a full report of the concepts, terminology that should be used in the project and the design principles that are applicable. Anyway by accomplishing this regulations it is not considered enough to ensure the conformity with other relevant essentials such as health and safety requirements. Thus, is does not give full presumption of conformity to the final design.

In order to fulfill the full conformity so that a design can be placed to the market, '*B-type'* standards must be achieved. These standards cope with very specific topics related to mainly to the machinery safety. In these multiple standards very important topics such as mechanical vibrations on machinery, safety of machinery when a human physical performance is needed to operate the machine, acoustics regulations.

A very important regulation that must be highlighted is the "***CEN EN 13115:2003+A2:2009"***. In this document the safety standards of hand powered cranes are explained. It should be a must to read this European regulation before using the designed portable crane.

The electric winch used to lift the load must be in order with the regulation *'EN 14492-2'* which copes with the requirements for hoisting applications.

Below, the most important standards that are considered when designing a worksite crane are listed. If the reader of this thesis wants to have a further approach to the topic, a good start in the field of the industrial machines design is to introduce to yourself the regulations that must be applied.

- CEN EN 349:1993+A1:2008
  Safety of machinery – Minimum gaps to avoid crushing of parts of the human body
- CEN EN 614-1:2006+A1:2009
  Safety of machinery – Ergonomic design principles – Part 1: Terminology and general principles
- CEN EN 614-1:2006+A1:2009
  Safety of machinery – Ergonomic design principles – Part 2: Interactions between the design of machinery and work tasks
- CEN EN 842:1996+A1:2008
  Safety of machinery – visual danger signals – General requirements, design and testing
- CEN EN 1005-2:2003+A1:2008
  Safety of machinery – Human physical performance – Part 2: Manual handling of machinery and component parts of machinery
- CEN EN 1032:2003+A1:2008
  Mechanical vibration – Testing of mobile machinery in order to determine the vibration emission value
- CEN EN ISO 4414:2010
  Pneumatic fluid power – General rules and safety requirements for systems and their components

# 3. WORKSITE CRANES: STATE OF THE ART

Worksite cranes are known to have existed since at least the Ancient Greeks period (Society, Studies, Journal, & Studies, 2012). This invention replaced other lifting systems, such as ramps or even the force done by a group of people, which had been used previously as a tool to build houses, temples and many other purposes.

The advances made in technology since that time has implied the consequent development in the design cranes that can be seen nowadays in operation. Avoiding those cranes related to the construction of large buildings and skyscrapers, which do not have any interest for this thesis, the range of cranes available used as a worksite crane is very wide given that the conditions of the lifting work might be very different. Therefore in every case a specialist should be the one to choose which one fits better for the desired application. By doing research of those worksite cranes and studying their different mechanisms, conclusions can be taken so that a draft of the adopted solution can be started.

One typical kind of crane that can be used as a worksite crane is the one usually called '*Mobile Crane'* that is because in most cases these cranes are installed on a vehicle. They can also be named as '*Telescopic cranes'*.

***Figure 1.*** *Telescopic crane.* (Palfinger Sany, 2017)

Generally, a motor is used to spoil a wire rope that at the end of the telescopic boom, a hook and sheaves which gives to the whole system a considerable mechanical advantage. The inclination of the boom, as well as the extension of its length, is done by means of a hydraulic cylinder. The total crane capacity is controlled principally by the operating radius. This operating radius is measured horizontally from the center of rotation to the hook. Meanwhile the vertical distance is measured as a mathematic function involving the length of the boom and the angle with the horizontal. Normally, the designer of the crane would elaborate a chart so that every user is aware of the limit working conditions at every possible configuration of the crane. The range of load that can lift this kind of crane goes from 3 up to 50 tons and the limit average speed during the operation is lower than 30km/h. (Chudley & Greeno, 2008)

The assembly consist practically only in setting up the telescopic jib. Therefore it is commonly use when a lifting device is needed and the operation must be done in the shortest period of time as possible. Mainly, it is used in construction of industrial buildings, to lift heavy loads when a short period of time of operation is required and also to help in the assembly of others bigger cranes. The main disadvantage is that it requires a considerable working space due to the telescopic boom.

In fact, various designs based on the classic mobile crane ar nowadays available. Each of them, mainly differ on the vehicle which has the crane attached to it. That responds for example to the need of placing cranes in terrains (rough, off road, railways,etc.) of different conditions. (Norman Spencer, 2008)

As a slightly different crane, the '*mini crawler cranes'*, can also be found. The design of these cranes are clearly based on the *mobile cranes* only that its dimensions are much more reduced.

***Figure 2.*** *Mobile crane.* (GGR Group, 2016)


The most simple worksite crane that can be found in the market nowadays is the one referred as '*Floor crane'* although it can be seen named by many other ways. Many different configurations are possible (Bright Hub Engineering, n.d.): wall mounted (fixed or travelling), floor mounted (attached to an enough resistant foundation) and a floor mounted crane which is not attached to any place but just supported with legs or something similar. For the development of these thesis the only one which is compelling enough to be commented is the last one.

Basically it is a simplified '*mobile* crane' but in this case it is fixed to the floor (as the static cranes that can be seen during the construction of large buildings). The structure is composed by different profiles, normally made of steel, which support all the stresses provoked by the load. Many different configurations are possible with this small crane. However, these cranes usually have a small telescopic jib which can vary its inclination by means of a small hydraulic cylinder actioned by a hand pump. The main advantage is that they can be assembled and disassembled with ease even by people who does not have any advanced knowledge. In addition, that means that these type of unit are considered as portable units. That saves time during the project and it can have an effect onto the final budget by reducing the hours needed from the operator. The load capacity depends on the model and it often goes from 400kgs up to 3 tons depending on many things like the radius of operation, the counterweight that must be added, etc. As the *mobile crane*, the floor crane lifts the load by means of a hook attached to a wired rope (it can also be fiber depending on the working conditions) which is spoiled into a drum. These drums are directly connected through a designed gearbox to motor that can overcome all the required torque to lift the load. Nevertheless, some of these cranes can be actioned without the help of any motor but the force applied by a person in the end of the cable.

Another type of crane is the '*Spider Crane'* which is similar to a floor crane. Its main difference resides in the many legs that it uses a support of the crane which reminds to a spider. This solution provides the ability to perform more skilled lifting operations and it is versatile in terms of that it can be used in many different terrains. A part from having a telescopic boom, many of these cranes have also a mechanism that allows a rotation motion.

# 4. REQUIREMENTS OF THE PORTABLE CRANE UNIT

In every type of design the requirements that have to be taken into consideration shall be defined so that it saves time to both designer and customer. In many projects this is hard part to develop at the beginning and it usually occurs that the designer starts working in its first proposal without knowing exactly which one is the optimal direction that he should be focusing his ideas. Therefore, time is lost and nowadays it signifies that the project is not as cost-effective as it must be.

During this chapter all the requirements which has been specified at the very first moment of the thesis are described as well as the description of the adopted solution.

## 4.1 Requirements and specifications of the crane

As it is mentioned in the introduction chapter the worksite crane is going to be placed inside of a mine. That mine has small dimensions and also the tunnels to access to the launch point where the unit is going to be exactly placed do not allow to enter the mine with objects of large dimensions. For that reason, the crane has to be portable and its components must be designed to be assembled inside of the mine. Also, knowing that the entrance to the operating place is small, it is assumed that all the components are going to be carried by persons and not by machines. That leads to the requirement that the weight of the components should be reduced as much as possible.

The load to be lifted is estimated to be proximately to 110kg. All components must withstand the stresses produced by that force. The lifting height is unknown but it has been specified as a minimum of 60 meters.

Concerning the dimensions of the mine it has been specified that the height of the crane should be around 1,5 meters. In addition giving that the load has a considerable weight to be lifted by a person, the inclination of the boom must be variable so that for instance, a person can attached the load to the crane without lifting it.

The length of the lifting device is defined by the fact that it is unknown where the load to be lifted should be placed. Thus, the second requirement is that the length must be adjustable so that the lifted item can be placed without the need of moving the whole assembly which is considered to be impossible due to the mine dimensions.

To avoid moving the unit more than it is needed a mechanism to rotate the crane must be designed. That, combined with the extensible boom, should be enough to operate with the crane fixed on a specific place.

Reliability must be achieved given that a failure during the operation of the crane may cause a great loss of money, time and in the worst scenario it might harm a person.

To sum up, the most important features to be achieved are the following ones:

1) The mass of the crane and its components must be as low as possible.
2) The unit must be portable.
3) The machine must be designed to lift at least 110kg at its maximum length.
4) The minimum lifting height is specified as 60 meters.
5) The height of the portable crane should around 1,5 meters.
6) The length of the boom must be adjustable.
7) The crane must have a mechanism to rotate the boom.
8) The final design must be as reliable and robust.

## 4.2 Description of the adopted solution principle for the crane unit

Having clarify and justify all the requirements and specifications a decision is made so that the designing process can be carried on. In this study, the adopted solution is a portable crane with a telescopic boom in order to have a variable length and a rotating column. It is a very common type of crane that can usually be found as a commercial product in many places. Nonetheless, given the situation and the requirements that this design has to fit, a customized unit has to be designed. During this subchapter all the decisions about the most important components of the crane are discussed.

First, the material used in most parts of the crane is an alloy of aluminum. This material provides enough robustness. The yield strength and the ultimate strength of the aluminum alloys are hard enough taken into consideration the mass of the load that is going to be lifted. Besides the aluminum alloy can machined and also welded if it is needed. Other options that could have been considered are materials such as iron, steel and its different alloys. Those materials have been discarded because of its weight in comparison the aluminum. There are providers located in Finland that can provide profiles to this project which helps with the time of delivery and it reduces the cost at the end of the project.

The assembly of the unit must be done in the working place. That is to say, all different parts are designed to be conveniently assembled by using pins, bolts and any other kind of removable unions. Welding between two different parts of the crane must be avoided, otherwise it will not be possible to disassembly the machine once the operation is done.

The required lifting height is big enough to discard any type of lifting device that requires the direct interaction with a worker. That would mean designing a system of pulleys to multiply the force of the person and more systems to that component itself such as a brake to avoid the load to drop down in case of accident. In this case, an electric winch is considered to be the best option the lift the load. To fit the requirement of 60 meters, it is only needed a drum big enough to have the correct length of steel rope spoiled in it. Designing this component is complicate even for an experienced engineer and it increases the cost of the whole project. For that reason, a commercial electric winch which suits all the desired requirements is selected.

The main boom has to have a variable length. That is called in this field a telescopic boom. Generally, it consist in two profiles one within the other. The one with bigger dimensions is the fix one and the smaller one slips inside of the fix profile by having the help of a mechanism.

There are many ways to design this mechanism. Two of them have been considered in this dissertation. A hydraulic cylinder actioned by a hand pump can be installed inside the fixed boom with the piston rod attached to the beginning of the small boom so that by adding or subtracting a fluid to the cylinder a linear motion can be produced with ease. On the other hand, a power screw actioned by a handle and installed the same way as the cylinder mentioned above can be designed. This power screw can be manufactured in aluminum so that the weight of the crane do not increase more than expected because of desired of having a telescopic boom. Both options can be purchased from providers located in Finland. Hence, the hydraulic cylinder is discarded because the extra weight of the cylinder and also because the power screw is purely mechanical and it implies a more robust and reliable design.

To rotate the crane another mechanism must be designed. The first idea to achieve this goal was to design a spur gear box actioned by a handle so that a person could manually rotate the crane. This option was discarded because of it large weight and its complexity. Nowadays, most cranes rotate commonly by the use of a slewing bearing which consist in two races with different inner diameters assembled along with a support which at the same time holds the balls of the bearing. This type of bearing resist axial and radial forces and a momentum too. In this concrete design, it can be installed at the lowest part of the column attached with bolts to both, column and the base support. Its simplicity and the fact that it is a commercial product that can be found easily is the reason why it has been finally chosen as the mechanism to rotate the crane.

# 5. MAIN COMPONENTS OF THE PORTABLE CRANE UNIT

In this chapter the main components of the portable crane unit are described. In every subchapter the pre-dimensioning calculations are explained as well as the reasons of the decisions that had been taken. The motion of the mechanical parts is also briefly explained. In many cases, it occurs that the calculations are not trivial. For that reason in this chapter many times a simulation done by FEM analysis is commented and evaluated.

## 5.1 Telescopic boom

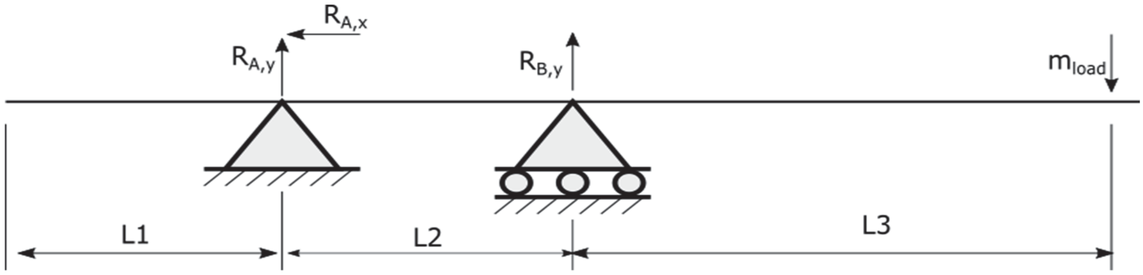
The telescopic boom as a very important component of the crane. The design is meant to be operating inside of a mine and that does not give to the users many choices about how and where to place the portable crane even considering that its dimensions are not big. For that reason, an extensible arm to control the position of the lifted load is needed.

First of all, a calculation to check the safety, from the structural point of view, of the profiles that are going to be used is needed. The telescopic boom consist mainly in two square tub profiles selected from the catalogue of a Finnish aluminum manufacturer named PURSO (Purso, 2016). The dimensions of the smaller profile are 80x80x4mm and the dimensions of the bigger one are 100x100x4mm. The material used is the aluminum 6063-T4.

The biggest one is used a structural support of the boom and also as a container of the whole mechanism that provokes the other profile to have a relative motion to this fixed profile. As it has been previously discussed this can be done by many ways but in this project it has been concluded that a power screw is the most reliable option to be used for this purpose.

To define the dimensions of the boom a calculation considering all loads is needed. In this case the following free-body diagram is considered:

***Figure 3.*** *Free-body diagram of the telescopic boom*



The physical equations describing the problem using the Newton's laws are the following ones:

$$\sum F_x = R_{Ax} \tag{1}$$

$$\sum F_y = R_{Ay} + R_{By} - m_{load}g \tag{2}$$

$$\sum M_A = m_{load}g \cdot (L_2 + L_3) - R_{By} \cdot L_2 \tag{3}$$

Where:

- $R_{Ax}$: Reaction in support A in the direction of x axis.
- $R_{Ay}$: Reaction in support A in the direction of y axis.
- $R_{By}$: Reaction in support A in the direction of y axis.
- $m_{load}$: Mass of the load
- $g$: Velocity of gravity

The distances between the points considered in Figure 3 are shown in the following table.

***Table 1.*** *Distances of between supports in the telescopic boom*

| Ref. | Distance (mm) |
|---|---|
| $L_1$ | 565 |
| $L_2$ | 360 |
| $L_3$ | 1.240 |

Considering equilibrium then equations (1), (2) and (3) are equal to zero. Is trivial to see that the value of the reaction in support A in the direction of the horizontal axis is also equal to zero. From the equation (3) the reaction in support B can be found directly by operating mathematically.

$$R_{By} = \frac{m_{load}g \cdot (L_2 + L_3)}{L_2} = 4.796\ N \tag{4}$$

Knowing the value of the reaction in support B then equation (1) can be used to find the value of the reaction in A.

$$R_{Ay} = R_{By} - m_{load}g = 3.716{,}9\ N\ (downwards) \tag{5}$$

The results of the static analysis are resumed in Table 2.

***Table 2.*** *Reactions in the supports of the telescopic boom*

| Ref. | Force (N) | Direction |
|---|---|---|
| $R_{A,x}$ | 0 | - |
| $R_{A,y}$ | 3.716,9 | Downwards |
| $R_{B,y}$ | 4.796,0 | Upwards |

Having calculated all the values of the reactions it is possible to analyze the stress diagrams in order to calculate the safety factor of the telescopic boom.

First of all the equations describing the shear stress can be easily found.

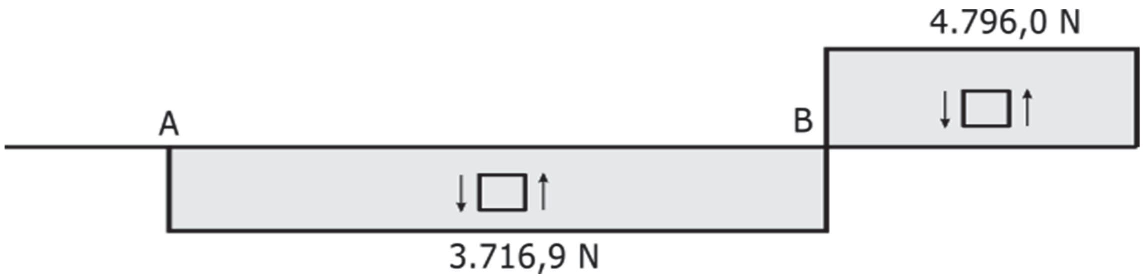
$$From\ x = 0\ to\ x = A: v = 0 \tag{6}$$

$$From\ x = A\ to\ x = B: v = R_{Ay} \tag{7}$$

$$From\ x = B\ to\ x = 2{,}165: v = R_{By} - R_{Ay} \tag{8}$$

As a result the shear diagram can be drawn as shown in Figure 4.

***Figure 4.*** *Shear diagram of the telescopic boom*




As the telescopic boom is subjected to bending, it is very important to estimate the moment diagram. The equations to find the moment in every part of the boom are:

$$From\ x = 0\ to\ x = A: M = 0 \tag{9}$$

$$From\ x = A\ to\ x = B: M = R_{Ay} \cdot x \tag{10}$$

$$From\ x = B\ to\ x = 2{,}165: M = R_{Ay} \cdot x - R_{By} \cdot (x - 1{,}24) \tag{11}$$

Therefore the moment diagram is represented in Figure 5 by giving values to the equations (9), (10) and (11).

***Figure 5.*** *Moment diagram of the telescopic boom*


Checking the safety factor for the smaller profile should be enough to prove that the boom will withstand the efforts that is going to be subjected when lifting the load. The profile is subjected to bending. The theory used to express the maximum stress on the beam mathematically is estimated using the equation *(12)* (Norton, 2006).

$$\sigma_{b,max} = \frac{M}{I} \cdot y_{max} = \frac{M}{W} \tag{12}$$

Where:

- $\sigma_{b,max}$: Maximum stress due to bending.
- $M$: Maximum moment calculated in the moment diagram.
- $I$: Inertia of the profile.
- $y_{max}$: Distance from the axis to the extreme fiber

The inertia of the square tub can be found using the equation (13).

$$I = \frac{D^4 - d^4}{12} \tag{13}$$

Where:

- $D$: Outer side length of the square tub.
- $d$: Inner side length of the square tub.

The value of the distance from the axis to the extreme fiber is easy to find. The square profile is subjected to bending so the mentioned distance cannot be anything else that the half of the length of the outer side of the tube.

$$y_{max} = \frac{D}{2} \tag{14}$$

It can be observed in Figure 5 that the maximum moment is estimated in 471,16 Nm. Thus, the maximum stress can be calculated using equation (12):

$$\sigma_{b,max} = \frac{M}{I} \cdot y_{max} = 38{,}97\ Mpa \qquad (15)$$

The tensile strength of the material can be found in the engineering data of the manufacturer. Its value is considered to be 120Mpa for the purpose of this project (Aluminium City, 1999). Knowing the maximum stress that the profile is going to be subjected and the tensile strength of the material the safety factor can be easily estimated.

$$SF = \frac{\sigma_{tboom}}{\sigma_{b,max}} \cong 3{,}1 \qquad (16)$$

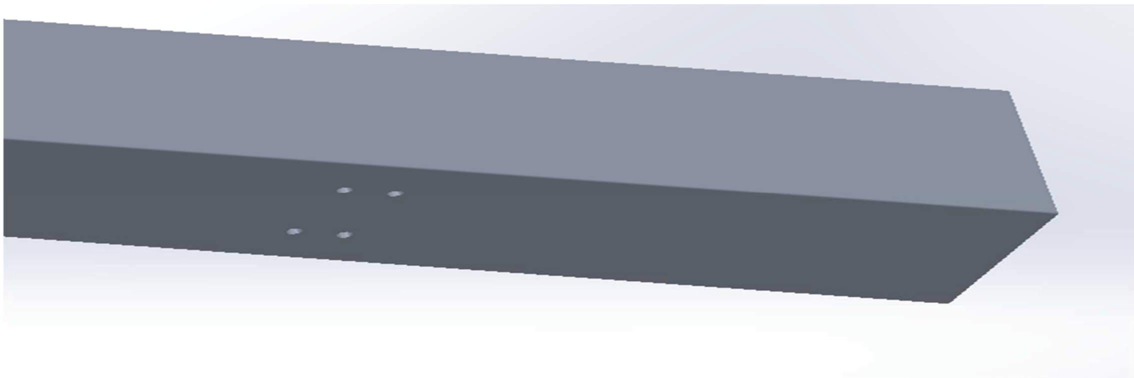
The value of a safety factor of 3,1 is considered to be feasible for the purpose and safety of the project.

In a professional environment hand calculations are usually only as a pre-dimensioning tools. The final verification of the design before it is going to be manufactured and properly tested is done by using FEA software. This type of software allows the team of engineers to simulate the working conditions and therefore to solve the equations given by the theory of the continuum mechanics which are a really complex and laborious task that does not fit the time delivery requirements of the industry work nowadays.

For the sake of this design of the portable crane, a FEM simulation in the software SOLIDWORKS is run in order to verify that form the structural point of view the telescopic boom is correctly designed.

In the simulation the force of provoked by the weight of the load is applied where the pulley, that holds the steel wire at the end of the boom, is attached with 4xM5 bolts.

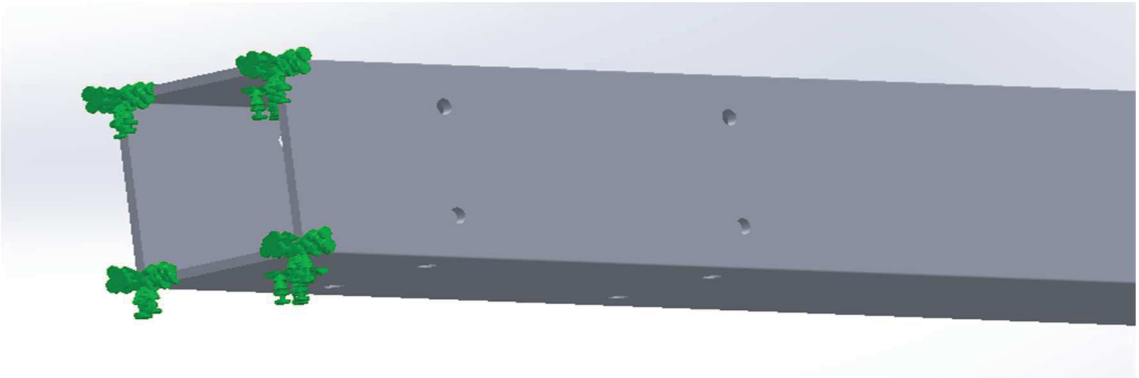
***Figure 6.*** *Drilled M5 holes on the boom for the attachment of the pulley*



Gravity of 9,81m/s$^2$ is applied in the software so that the calculations take into consideration the value of the weight of the material. It will provoke an extra downwards force at the center of mass of the boom.

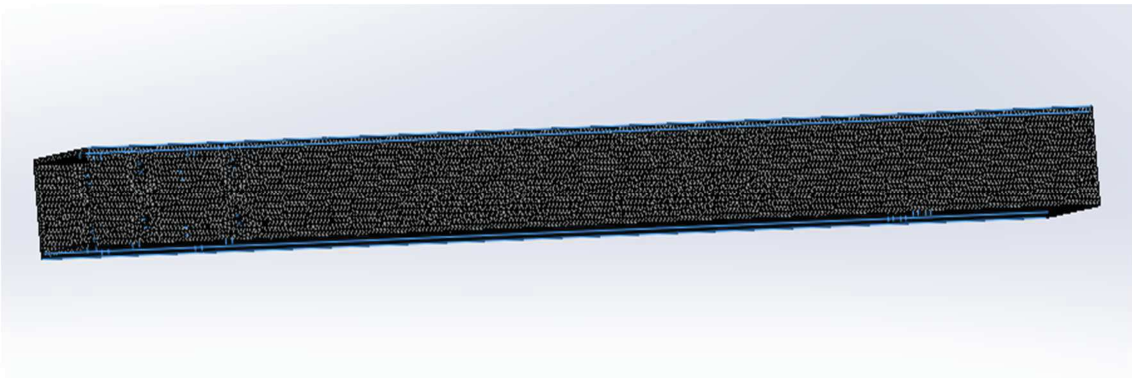
The support is considered to be the inner side of the boom given that it is attached to the bigger profile of the boom as well as two drilled holes for 2xM10 bolts which are also being used as a fix attachment between the two parts.

***Figure 7.*** *Simulation of the support on the telescopic boom*



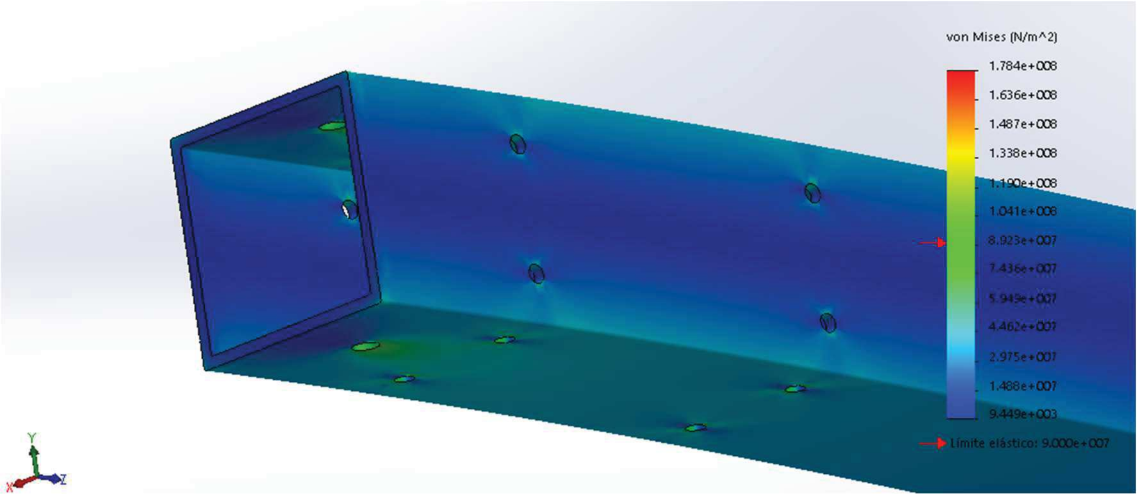
One very important feature to control in FEA analysis is the size of the mesh and the type of geometry of the mesh. The explanation of the whole topic is so extensive that it is considered to be out of the scope of the project. However, as a minor explanation, the smaller the geometry of the finite elements are used the more nodes the whole software will generate and it can lead to better get better results in the end. The generated mesh is shown on Figure 8.

***Figure 8.*** *Mesh of the telescopic boom*



Having done all pre-process tasks it is now available to run the calculations and obtain the results of the stresses and deformations in the boom. First, the maximum value of the equivalent von Misses tension on the boom is found on the drilled holes which is considered to correct given that holes are a point where tensions tend to be concentrated. That maximum value is close to the 89,2 Mpa.

***Figure 9.*** *Von Misses stress on the telescopic boom*

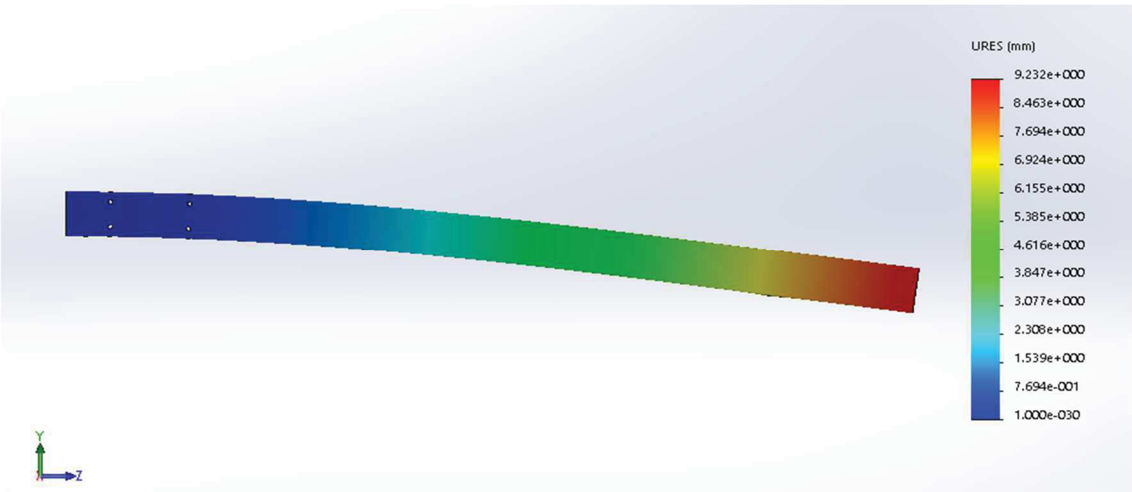



Taking into account the value obtained in the simulation the safety factor is still considered to be feasible.

$$SF = \frac{\sigma_{tboom}}{\sigma_{b,max}} \cong 1{,}3 \tag{17}$$

Secondly, even if the stresses are considered to be under control is important to verify that there is not any local deformation that exceeds a value that is considered to be the limit.

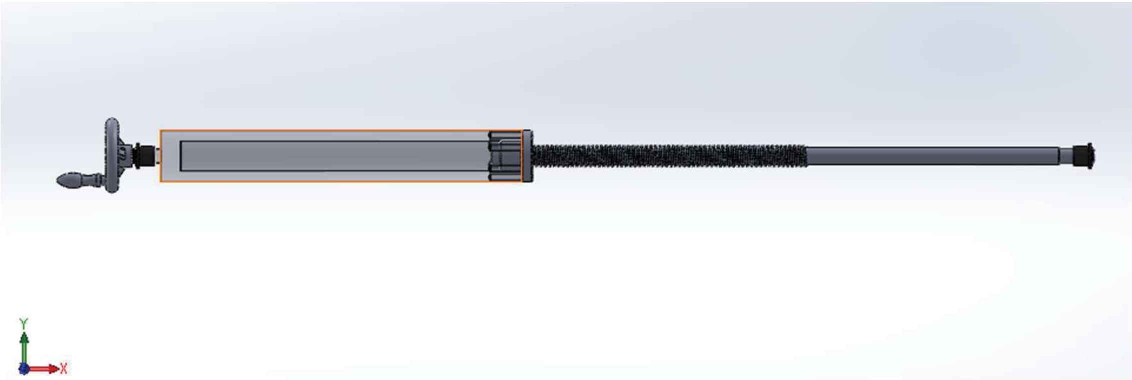
***Figure 10.*** *Deformation of the telescopic boom*




As shown in Figure 10 the maximum deformation can be found at the end of the profile. Nonetheless, it is not considered to be a deformation that can cause any harm given its value is under 1mm.

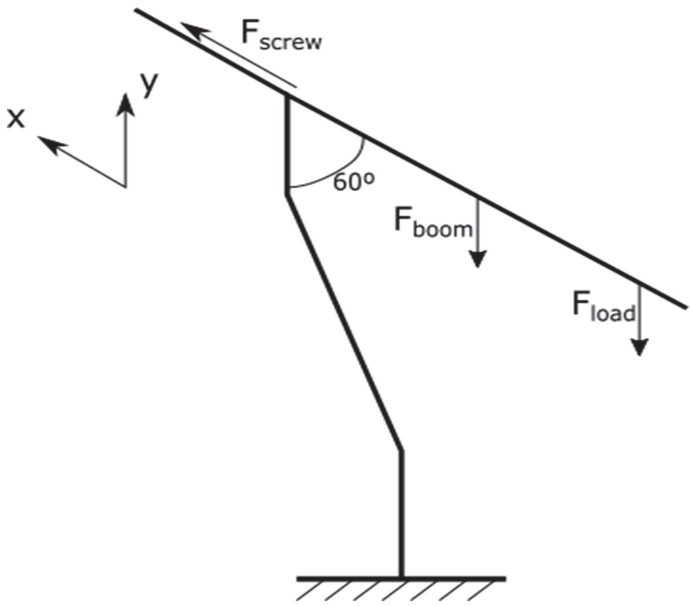
The mechanism that provokes the motion between the two profiles is a power threaded screw (Figure 11). Basically it consists in a threaded profile which is supported by both sides and attached to the inner profile of the telescopic boom. Two self-lubricating bushings are installed so that they serve as a structural support and also it works as a control of the wear avoiding contacts between two metallic materials. The power screw is designed to be actioned by a person by means of rotating a wheel handle.

***Figure 11.*** *Power threaded screw mechanism*




The calculation procedure is retrieved from a well-known engineering design handbook (G.Bydunas & Keith Nisbett, 2011). First of all, a free body diagram scheme is needed to analyze the forces and to estimate the value of the force that the power screw must be able to do in order to lift the entire load.

***Figure 12.*** *Free-body diagram of the external forces on the power screw*




Normally larger calculations are needed in order to explore all the possible scenarios using a mathematically software, such as Matlab, and programming a code which considers all possible angles and forces. In this case, given that is being used a well-trusted procedure and to simplify the calculation of pre-dimensioning of the power screw, the worst case is considered to be considered when the boom is inclined downwards in angle 60 degrees in respect to the main column and the design of the power screw is done according to the values obtained in this estimation.

In this case, the external forces are caused by the action of gravity on the different masses. Both forces have the same direction so as a result to solve this problem it is easier to consider just on force which is the sum of both weights multiply by the acceleration of gravity.

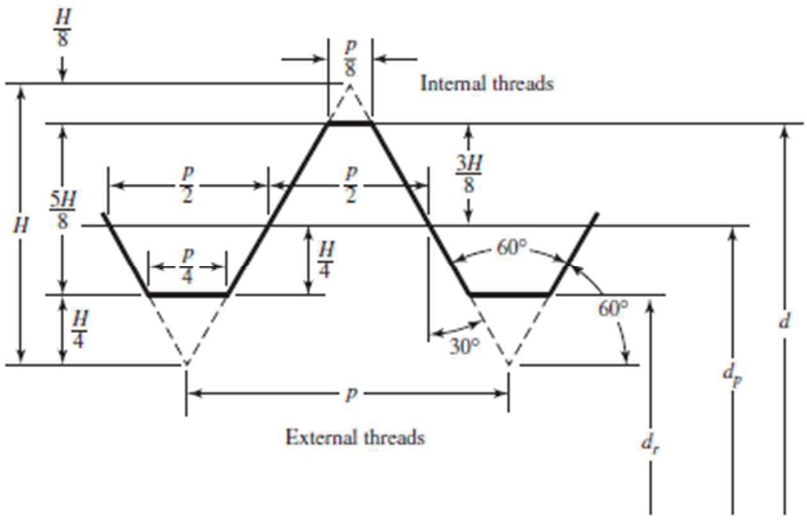
$$F_{lif.load} = (m_{load} + m_{boom})g = 1.124{,}23\ N \tag{18}$$

Knowing that the inclination angle of the boom is 60 degrees the force in the axial direction is calculated by using simple trigonometry.

$$F_{lif.load,x} = F_{lif.load} \cdot \cos(\delta) = 562{,}1\ N \tag{19}$$

The basic dimensions of the power screw are shown in (12) .

***Figure 13.*** *Power threaded screw* (G.Bydunas & Keith Nisbett, 2011)




The maximum force that a person can apply to the wheel handle is considered in this project on 400N. The validating process consists in selecting a commercial power screw and verifying that the force a person must apply to move the desired load is under the specified maximum force. The equation that is used to calculate that force is:

$$\frac{F \cdot d_m}{2} = \left[\frac{p + \pi \cdot d_m \cdot \sec(\alpha)}{\pi \cdot d_m - \mu_p \cdot \sec(\alpha)}\right] \tag{20}$$

Where:

- $F$: Force applied to the power screw
- $d_m$: Mean diameter of the power screw
- $p$: Pitch of the thread.
- $\mu_p$: Friction coefficient between the thread of the power screw and its housing
- $\alpha$: Angle of the thread (Also named as lead angle)

The selected power screw is from the manufacturer Norelem, whose specification is Tr24x5. The basic dimensions are explained in Figure 14 and in Table 3.

***Figure 14.*** *Scheme of the Norelem power threaded screw* (Norelem, 2017)

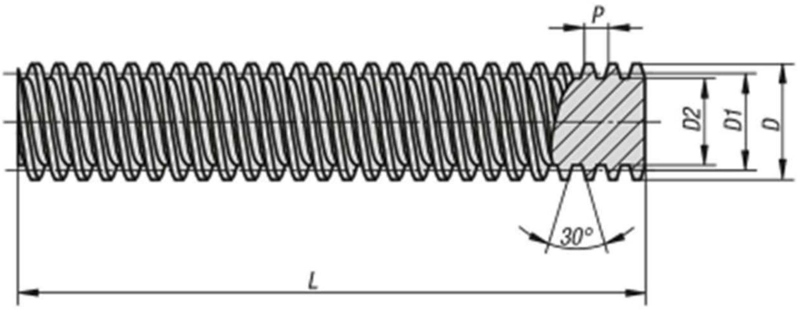



***Table 3.*** *General data of the power screw* (Norelem, 2017)

| Ref. | Dimension (mm) |
|---|---|
| D | 24 |
| D1 (min) | 21,094 |
| D1 (max) | 21,394 |
| D2 | 17,5 |
| L | 1000 |
| P | 5 |

Solving equation (20)(***25***) the value obtained of the force that a person should apply to lift the load is 360,26 N which is considered acceptable for this project.

The desired length of the power screw is set on 340mm. The only limiting factor of the length is that it must be assured that it does not allow the inner profile to go outside of the outer profile. That has been verified using the 3D CAD model. The number of threads needed of the selected power screw is calculated using equation (21).

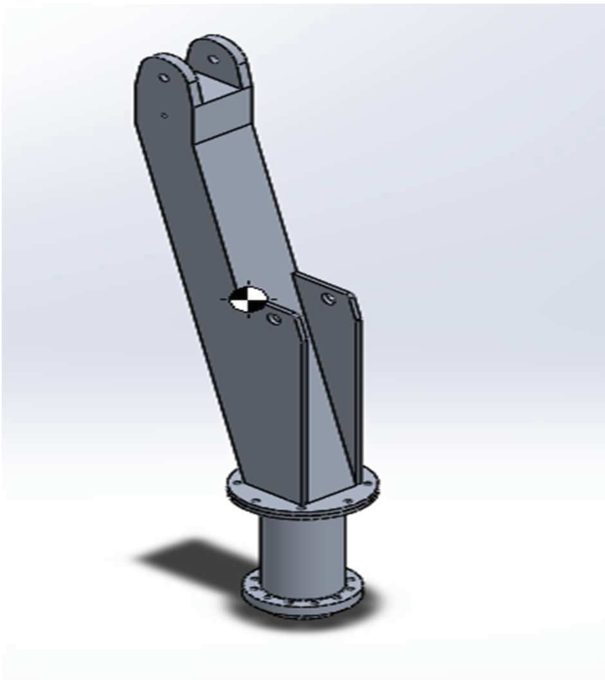
$$n = \frac{L}{p} = 68\ threads \tag{21}$$

## 5.2 Main column

One of the most important parts of the crane regarding to the structural point of view is the main column. The height of the crane is mainly defined by the length of the main column which is set in 1.500mm as specified in the requirements of the crane. The main column has to withstand a great amount of stresses due to the load and the weight of the telescopic beam.

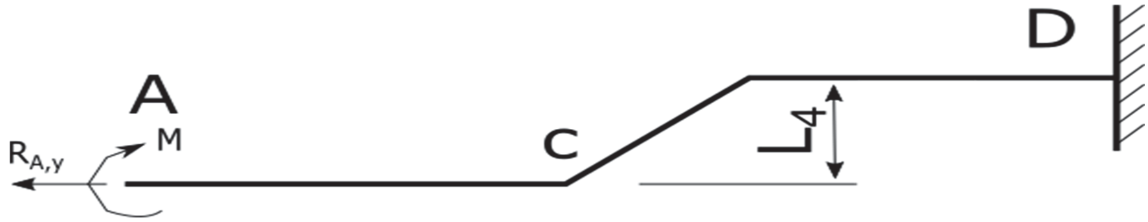
As it can be seen in Figure 15 the main column is formed by two profiles. The profile connected to the base support is a rounded tube of a 125mm diameter. Given that the crane has to have a rotational motion in respect to the floor the rounded tube allows to hold a mechanism in its base in order to fulfill that requirement. For structural reasons the profile is changed to a square tube of 120x120mm to connect to main column with the telescopic jig.

***Figure 15.*** *Main column model*



The free-body diagram is shown in Figure 16. All loads and weight resulting from the analysis of the telescopic boom are considered in order to check the safety of the final design.

***Figure 16.*** *Free-body diagram of the main column*




The reactions are considered to be at the top of the main column where it is connected to the telescopic boom by a pin and at the end of the square profile which is attached to the rounded tube by multiple bolts. The value of the force applied in point A is the reaction calculated above in subchapter 5.1. The resultant moment in is calculated by equation (22) taking counterclockwise moments as positive. Distances are taken from Figure 3.

$$M = m_{load}g \cdot L_3 - R_{By} \cdot L_2 \ = -\ 359{,}6\ Nm \tag{22}$$

The resultant moment is a clockwise moment of a 359,6 Nm. Applying the Newton's laws to the free-body diagram in Figure 16, the following equations can be found.

$$\sum F_x = R_{Ay} - R_{Dx} \tag{23}$$

$$\sum F_y = R_{Dy} \tag{24}$$

$$\sum M_D = R_{Ax} \cdot L_4 - M \tag{25}$$

Considering equilibrium then equations (23), (24) and (25) are equal to zero. Is trivial to see that the value of the reaction support D in the direction of the vertical axis is also equal to zero. From the equation (23) the reaction in support D has the same value of the reaction in support A but with opposite direction.

$$R_{Dx} = -R_{Ay} = -3.716{,}9\ N \tag{26}$$

The support D is also subjected to a local moment which can be calculated by using equation (25).

$$\sum M_D = 476{,}7\ N \tag{27}$$

The results of the static analysis are resumed in Table 4.

***Table 4.*** *Reactions in the supports of the main column*

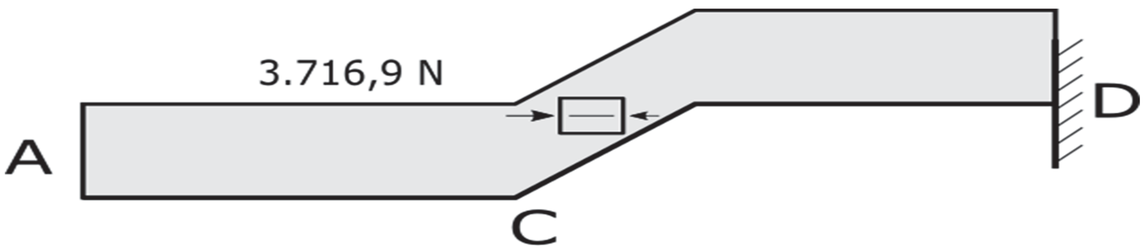
| Ref. | Force (N) | Direction |
|---|---|---|
| $R_{D,x}$ | 3.716,6 | Right |
| $R_{D,y}$ | 0 | - |
| **Ref.** | **Moment (Nm)** | **Direction** |
| $M_D$ | 476,7 | Clockwise |

Studying the forces previously calculated the diagram of stresses can be elaborated. The equation describing the normal forces all along the main column is:

$$From\ x = A\ to\ x = D: N_x = R_{Ay} \tag{28}$$

Thus, it becomes trivial to draw normal forces diagram which is shown in Figure 17.

***Figure 17.*** *Normal forces diagram of the main column*



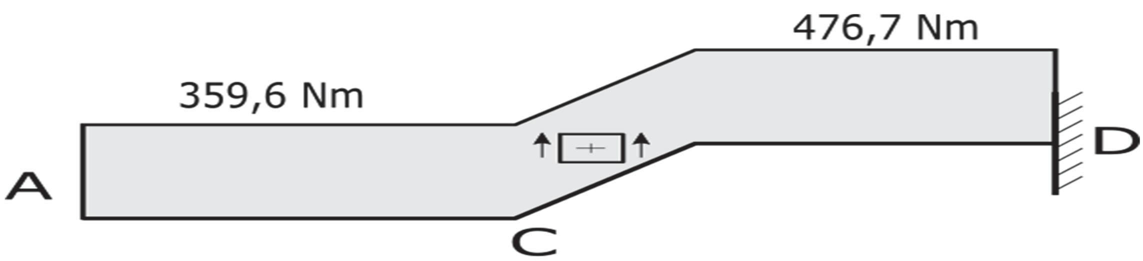

In order to draw the moment diagram in the main column the equations to find the moment in every part of the boom are:

$$From\ x = A\ to\ x = C: M = -359{,}6 \tag{29}$$

$$From\ x = C\ to\ x = D: M = -359{,}6 + R_{Ay} \cdot 0.225 \tag{30}$$

Therefore the moment diagram is shown in Figure 18.

***Figure 18.*** *Moment diagram of the main column*



In this case, in order check whether the size of the profile is correct or not, two cases are going to be studied. First of all, a normal procedures of bending verifications are going to be done. Nonetheless, noticing that there are considerable high normal forces the main column may also be subjected to a buckling phenomenon.

In order to check the safety of the profile when it is subjected to bending stresses equation (12) is used. Given that the profile is a square profile then the inertia can be estimated as:

$$I = \frac{h^3}{6} = 288.000\ mm^3 \tag{31}$$

Where:

- $h^3$: The outer size of the side of the square profile.

The value of the distance from the axis to the extreme fiber is again easy to find. The square is the half of the size of the outer side of the profile.

$$y_{max} = \frac{h}{2} = 60\ mm \tag{32}$$

Thus, the maximum stress due to bending in the main column is:

$$\sigma_{c,max} = \frac{M}{I} \cdot y_{max} = 74{,}91\ Mpa \tag{33}$$

The material used in the column is the same one used in the telescopic boom, the aluminum 6063-T4. As a result the safety factor when the column is subjected to bending is estimated in:

$$SF = \frac{\sigma_{tcolumn}}{\sigma_{b,max}} \cong 1{,}60 \tag{34}$$

To estimate the safety factor when the column is subjected to buckling the procedure followed is the same one according to steel structures. In this project the procedure is extracted from the open course of the University of Salamanca (Santo & Santillana, 2008).

In order to calculate the instability of the column due to the buckling the critic load should be estimated. Therefore, the equation that describes that critic load is:

$$N_{cr} = \frac{\pi^2 \cdot E \cdot I_{min}}{L_k^2} \tag{35}$$

Where:

- $N_{cr}$: Critical load.
- $E$: Young module of the material
- $I_{min}$: Inertia of the profile in the direction which has a minimum inertia.
- $L_K$: Is the characteristic length of buckling of the profile

The value Lk depends on which type of support do the column has. In this case support A is an articulated support (also called roller support) meanwhile the support D is a fixed support. That means that the parameter $\beta$ in equation (36) is equal to 0,7.

$$L_k = \beta \cdot L = 735mm \tag{36}$$

The minimum inertia of this profile is calculated by the following mathematical expression:

$$I = \frac{b \cdot h^3}{12} = \frac{h^4}{12} = 17.280.000\ mm^4 \tag{37}$$

Thus, the Euler buckling critical load can be estimated:

$$N_{cr} = 37.883{,}5\ N \tag{38}$$

The critical buckling stress is the Euler buckling load divided by the area of the profile.

$$\sigma_{cr} = \frac{N_{cr}}{A} = \frac{N_{cr}}{h^2} = 1.510{,}5\ Mpa \tag{39}$$

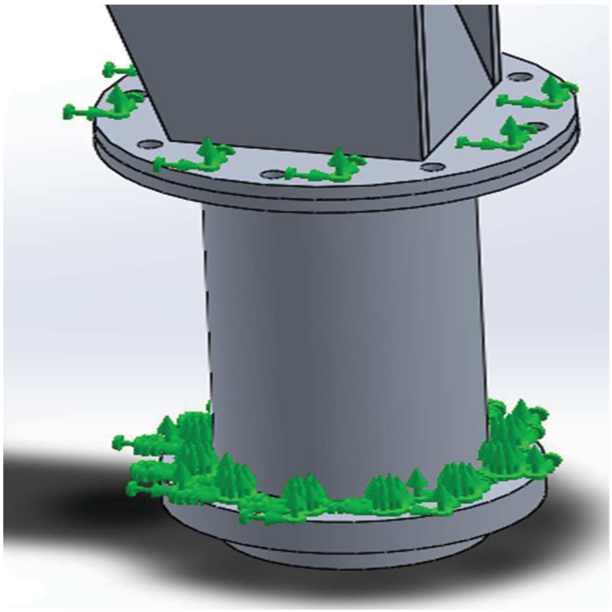
Since $\sigma_{cr} > \sigma_{c,max}$ yielding stress will be reached before buckling stress and therefore the main column will fail before because of bending rather than buckling. Concerning to the safety factor when the main column is subjected to buckling, it can be calculated as in equation (40).

$$SF = \frac{N_{cr}}{R_{Ay}} \cong 10{,}2 \tag{40}$$

As it is done in the previous subchapter in this case it will also be discussed the FEM analysis of the main column.

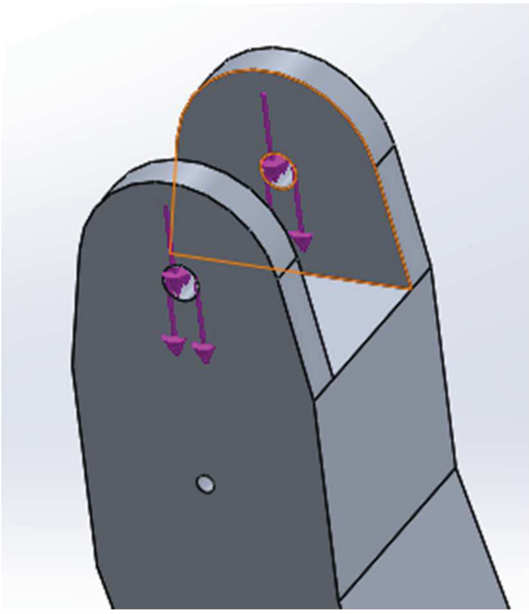
The fixed supports are modeled in such a way that the union between the two profiles act as a support (both are screwed to each other) and also the lower part of the column where it is supported by a slewing bearing.

***Figure 19.*** *Supports of the main column*



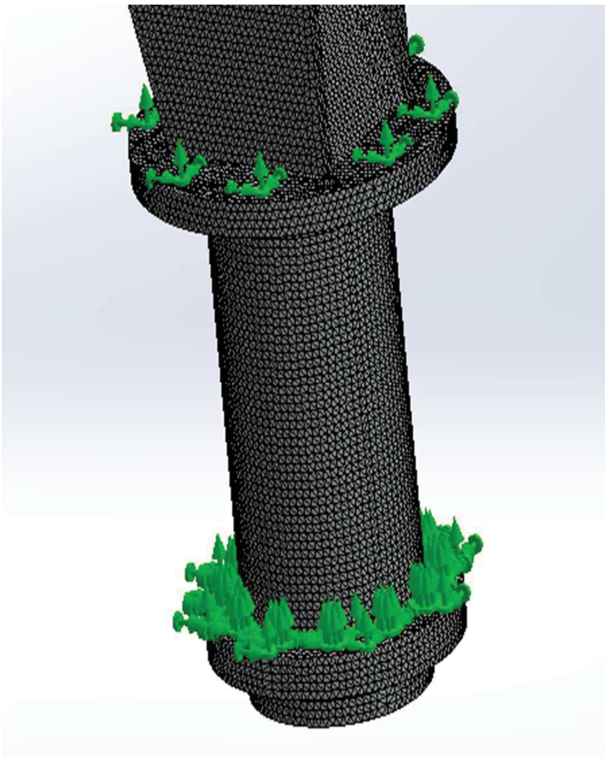
Gravity is also considered in this case so that the calculations take into account the weight of the material. The forces and moments are applied in the drilled holes that holds the pin where the column is connected to the telescopic boom as shown in Figure 20.

***Figure 20.*** *Forces on the main column*



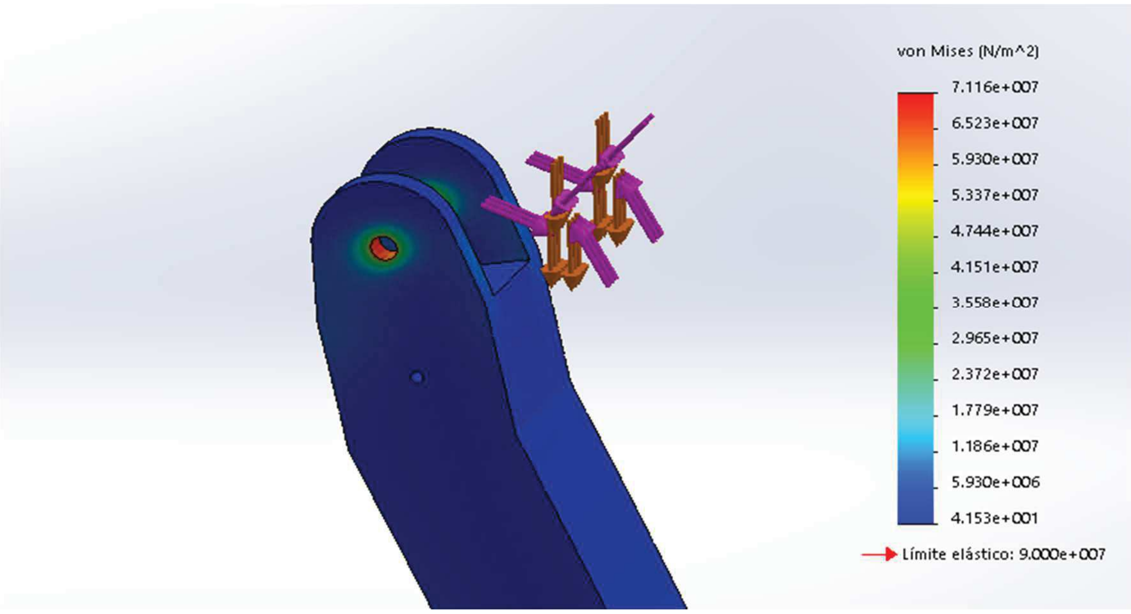
The mesh is generated following the same procedures that are described in subchapter 5.1.

*Figure 21. Mesh of the main column*



Having all the constraints defined it is possible to execute the calculations. The equivalent von Misses stress is shown in Figure 22. The maximum value is found where the load is applied given that it is a drilled hole and it suffers again from the concentrated tensions phenomenon. Its value is estimated in 71,16 Mpa.

*Figure 22. Von Misses stress in the main column*



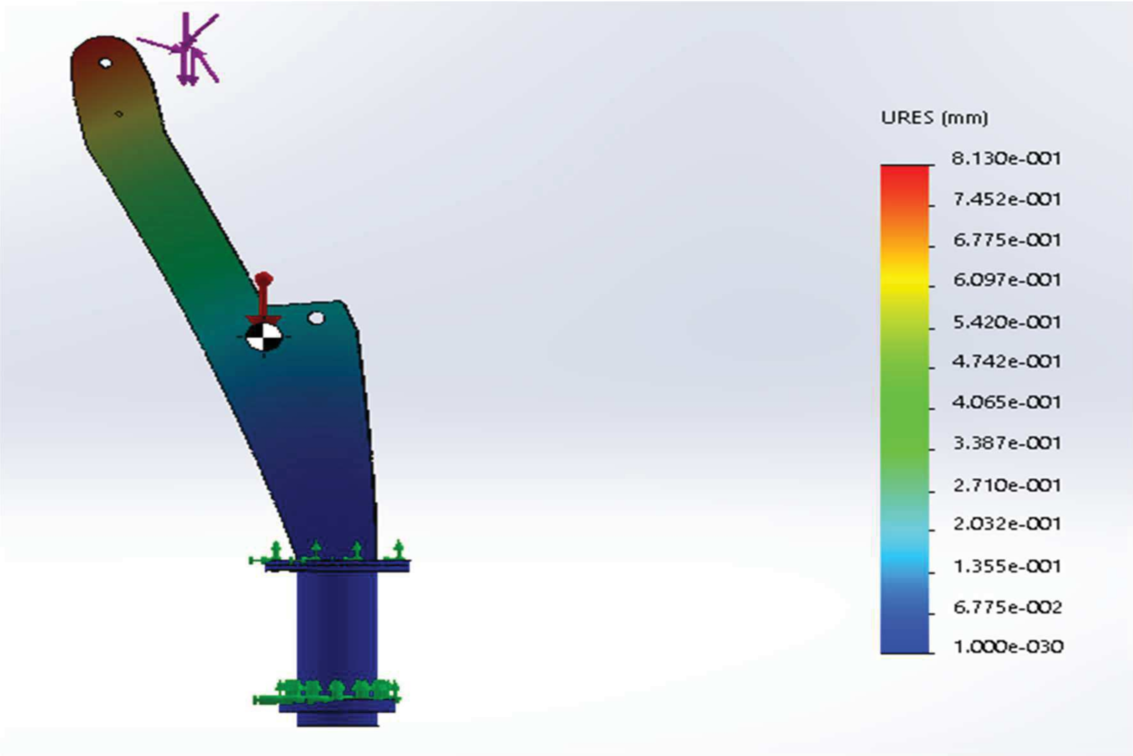

Thus, the safety factor following the values obtained in the FAE analysis is:

$$SF = \frac{\sigma_{tcolumn}}{\sigma_{b,max}} \cong 1{,}35 \tag{41}$$

In this case the value is close to the hand calculation which proves that it is a good method to use a starting point when designing machines.

The deformations calculated in the FEA analysis have low values, where the maximum is found at the top of the column. The deformation is 0,8mm its maximum point which is, as in the telescopic boom, under 1mm. For the purpose of this project it is considered to be feasible.

***Figure 23.*** *Deformation analysis of the main column*




In order to fix the telescopic boom to the column a cotter (pin) of a diameter of 10mm with a length of 140mm is used. The specification of the pin is the ISO 1234 -10x140 – St. The diameter of the hole passing through both parts is a M10 diameter hole. The cotter is subjected to both shear and crushing effects. The cotter is made of a steel with a tensile strength of 365,6 Mpa.

When the pin is subjected to shear stress the maximum allowable stress that the material can withstand considering a safety factor of 3 is:

$$\tau_{max} = \frac{\sigma_{tcolumn}}{2 \cdot C_s} = 60{,}93\ Mpa \tag{42}$$

Applying the maximum stress to the pin the area of the pin can be found.

$$A \geq \frac{F}{\tau_{max}} \geq 61mm^2 \tag{43}$$

Solving equation (43) the area of the pin must be higher than 61mm$^2$. Therefore the diameter of the pin can be calculated.

$$D_{cotter} = 2 \cdot \sqrt{\frac{A}{\pi}} = 8{,}8mm \tag{44}$$

Given that the diameter of the selected pin is 10mm it is assumed that it will not have any kind of problems against a shear failure.

Verifying the pin when is subjected to a crushing failure the maximum allowable stress that the material can withstand considering a safety factor of 3 is:

$$\tau_{max} = \frac{\sigma_{tcolumn}}{C_s} = 121{,}8\ Mpa \tag{45}$$

Applying the maximum stress to the pin the area of the pin can be found.

$$A \geq \frac{F}{\tau_{max}} \geq 30{,}51mm^2 \tag{46}$$

Solving equation (46) the area of the pin has to be higher than 30,51mm$^2$. Therefore the diameter of the pin can be calculated.

$$D_{cotter} = 2 \cdot \sqrt{\frac{A}{\pi}} = 6{,}22mm \tag{47}$$

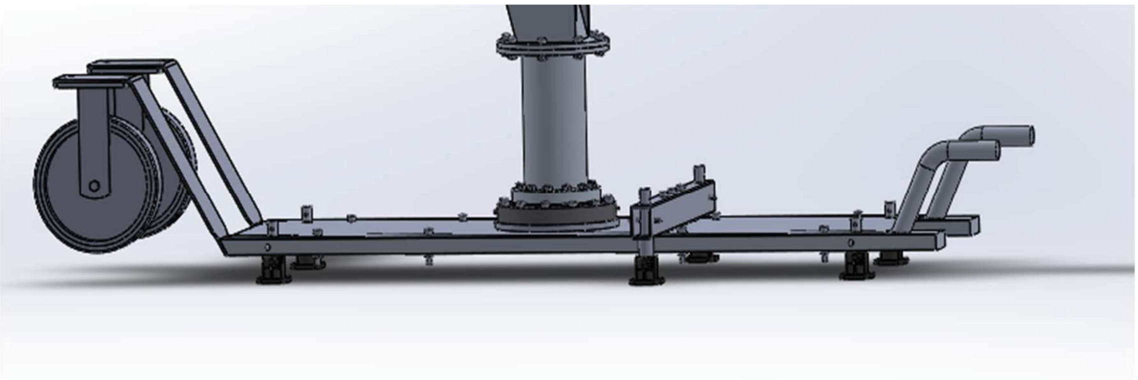
Given that the diameter of the selected pin is 10mm it is assumed that it will not have any kind of problems against a shear failure.

## 5.3 Base support

Every machine has a housing or any type of mechanical structure that serves as support of the whole machine. Various things are needed to be considered when it comes to design the support of a machine. The load that it has to withstand, where is the machine going to be placed and which kind of terrain it is going to be are among the questions that engineers would be likely to answer when they encounter this classical problem.

The final design of the base support is shown in Figure 24.The design of the base support of the portable crane consist in a plate screwed to 2 rectangular profiles of 85x30mm. The load that the crane will be lifting during the operation is relatively low so that is why the support is designed which small profiles as well as it helps reducing the weight of the whole machine. The profiles are attached to the base aluminum plate by 4xM10x55 ISO 4762 bolts.

***Figure 24.*** *Final design of the base support*

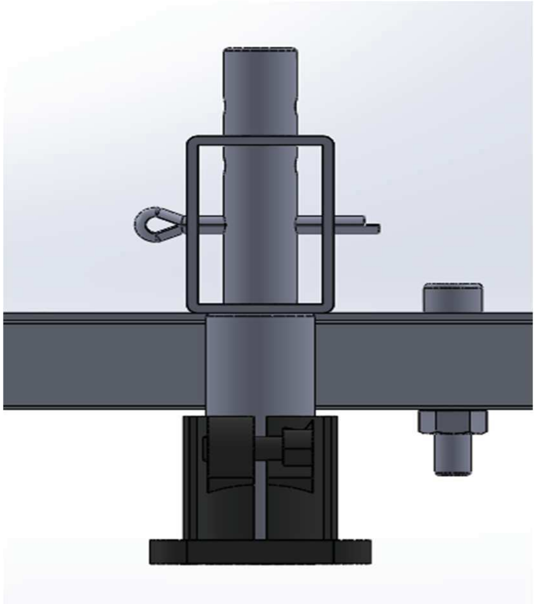


There are two extra features that had been added to the design of the base support. Firstly, at the front of the crane two handles made from aluminum rods are attached so that it eases to the worker to place the crane. To avoid to overcome excessive friction with the terrain two wheels of 250mm are attached to the rear part of the base support. The worker can incline upwards the crane by using the handles and then the wheels will be enabled to roll. The wheels are attached with 4xM8x20 ISO 4762 bolts to an aluminum plate that had been welded to the base support.

Secondly, two lateral legs are added so that the crane has two extra lateral supports. It will help to gain stability and to control any possible effect that had not been taken into account during the design of the portable crane unit. The lateral legs are attached to the base support by screwing 3xM10x80 ISO 4762 bolts.

In addition, adjustable supports might be added if the workers want to level the crane to the terrain. The supports can be attached by using a cotter passing through the supports and the rectangular profiles as shown in Figure 25. At the end of the aluminum cylinder that it is attached to the base support a clamped base (black piece in Figure 25) is added to have more surface making contact with the floor. The clamped base is tightened by means of a M8 bolt.

***Figure 25.*** *Attachment of the adjustable supports to the lateral legs*



The structural analysis is done by using the FEA software. The fixed supports are defined on the lower part of all the legs of the base support Figure 26.

***Figure 26.*** *Support simulation of the base support of the crane*

The force of gravity is considered so that the calculation takes into account the weight of the materials. The external forces calculated in the previous subchapters are defined in the hole where is used to hold the profile of the main column. To simplify the problem the values of the force are calculated applying the Newton's law to the whole crane as in equation (48). The overturning moment is not considered given that it is totally absorbed by the slewing bearing as it is explained in the next subchapter.

$$F_a = F_{cyl} - (m_{load} + m_{boom} + m_{winch} + m_{column})g = 2.258{,}4\ N \tag{48}$$

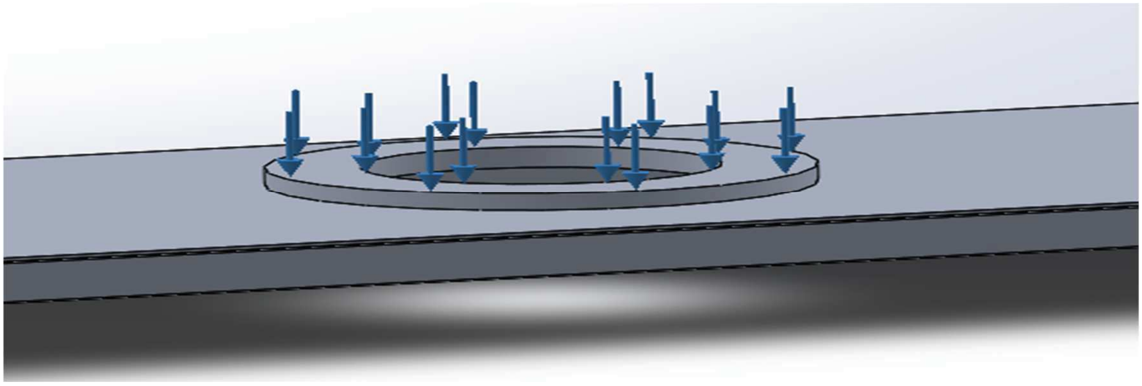
The values concerning the masses used to calculate the axial force are shown in Table 5.

***Table 5.*** *Weight of the crane unit components*

| Ref. | Mass (kg) |
|---|---|
| $m_{load}$ | 110 |
| $m_{boom}$ | 15,1 |
| $m_{winch}$ | 60 |
| $m_{column}$ | 43 |

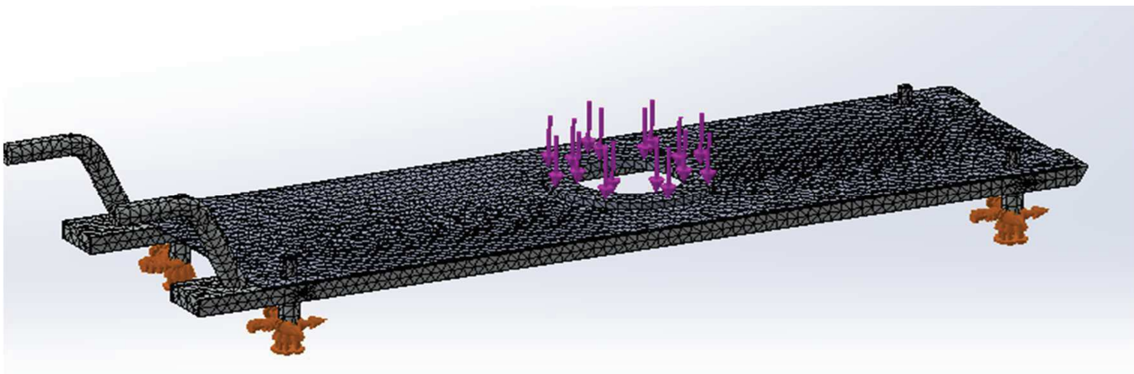
The force is defined where the slewing bearing is attached along with the column and the rest of the components of the crane.

***Figure 27.*** *Forces in the base support*



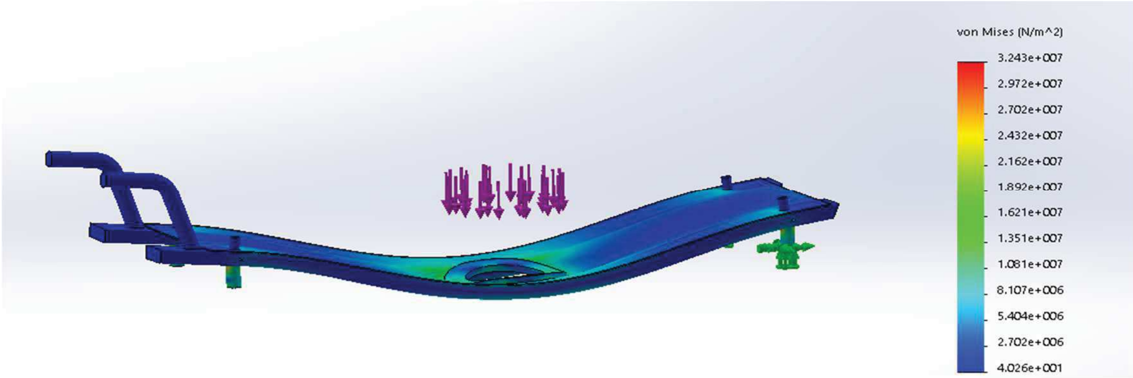
The whole mesh is defined following the same procedure described in the previous simulations. The size of the finite elements are set on 5mm in order to have enough accuracy in the results.

***Figure 28.*** *Mesh of the base support*



As a result of the simulation the maximum value of the von Misses stress is 32,43 Mpa as shown in Figure 29.

***Figure 29.*** *Von Misses stress in the base support*

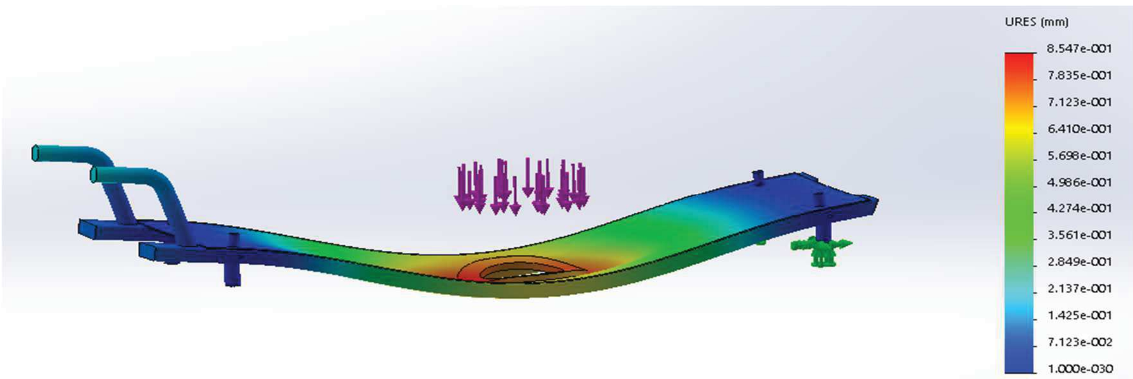



The safety factor can be calculated as previously done with other components.

$$SF = \frac{\sigma_{material}}{\sigma_{base,max}} \cong 3{,}70 \tag{49}$$

The deformation is mainly focused around where the slewing bearing is installed. The maximum value is estimated in 0,85mm which is considered to be acceptable. It is important to remark that this is not an easy simulation to run because of the effects that will absorb the slewing bearing. It is expected to improve in the reality given that the slewing bearing will work also as a support and it is oversized from the structural point of view as it is explained in subchapter 5.5.

***Figure 30****. Deformation in the base support*



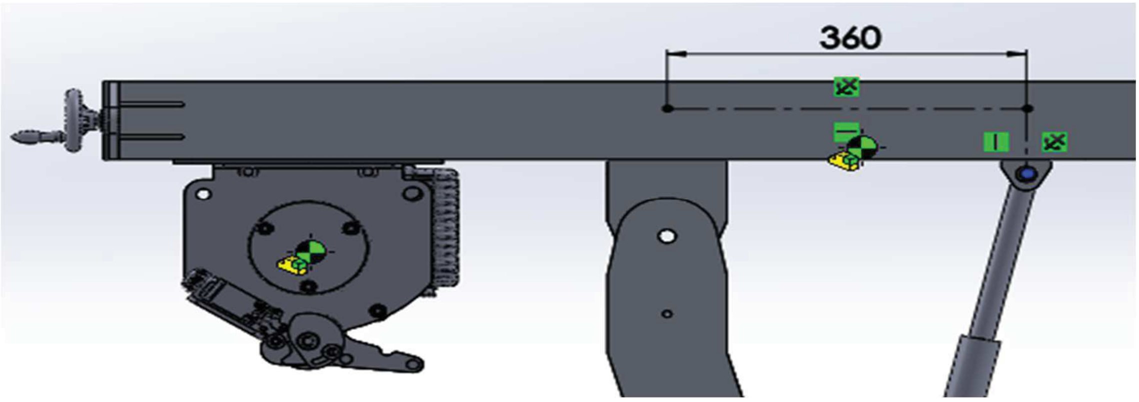

## 5.4 Hydraulic Cylinder

One of the main requirements for this project is the ability to vary the inclination of the telescopic boom. For the sake of this, a system with a hydraulic cylinder actioned by a hand pump is dimensioned and designed to be used.

The selection procedure used to select the hydraulic cylinder is extracted from the engineering support document of the manufacturer the German ROEMHELD (ROEMHELD, n.d.).

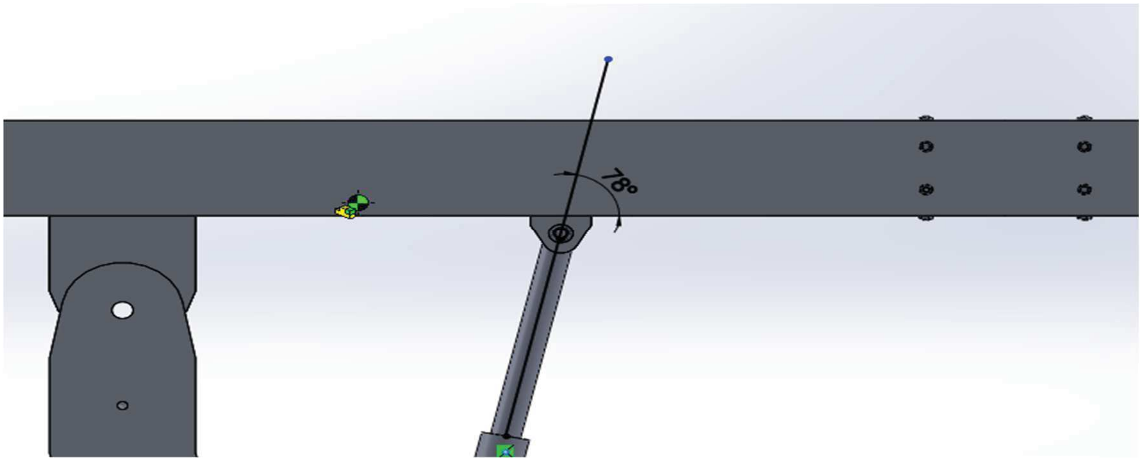
The hydraulic cylinder is attached by means of a precision tube to the telescopic boom. The distance from the attachment of the hydraulic cylinder and the joint between the main column and the telescopic boom is set in 360mm as it can be seen in Figure 31.

***Figure 31.*** *Installation of the hydraulic cylinder to the telescopic boom*




From the equation (4) the force that the hydraulic cylinder must be capable to exert in the vertical direction is 4.796 N. The angle between the hydraulic cylinder and the telescopic boom is measured directly from the CAD model as shown in Figure 32.

***Figure 32.*** *Angle of the hydraulic cylinder*




By use of basic trigonometry it is easy to find the total amount of force that the hydraulic cylinder must be able to produce.

$$F_{cyl} = \frac{F_y}{\cos\theta} = 23.068{,}7N \tag{50}$$

The guide of ROEMHELD proposes the equation to estimate the diameter of the piston rod of the cylinder.

$$d_{min}[cm] = \sqrt{\frac{F_{cyl}[kN] \cdot 400}{\pi \cdot p[bar]}} \quad (51)$$

Where:

- $d_{min}$: Minimum diameter of the piston.
- $p$: Pressure applied to the hydraulic cylinder

The pressure applied by the hand pump is usually set on 200 bar although bigger pressure can be exerted. Thus, the minimum diameter of the cylinder is:

$$d_{min}[cm] = 3{,}83\ cm \cong 38\ mm \quad (52)$$

The cylinder is selected from the catalogue of the manufacturer Bauer (Tilausavain, n.d.) which can be found in Finland to ease its purchase. In this case a stroke of 250mm is needed in order to vary the inclination of the boom as much as it is possible. For that reason a cylinder of 30mm of diameter of the piston rod and 50mm of bore diameter. The specification to purchase this cylinder is BPP50x30x250.

As it is calculated before the minimum diameter of the cylinder is satisfied so safety is assured. The cylinder can work as a double effect cylinder and its forces of traction and compression according to the guide of ROEMHELD can be estimated by:

$$F_{traction}[kN] = \frac{p[bar] \cdot \pi \cdot (D_{bore}^2[cm] - d_{rod}^2[cm])}{400} \quad (53)$$

$$F_{compression}[kN] = \frac{p[bar] \cdot \pi \cdot D_{bore}^2[cm]}{400} \quad (54)$$

As a result the value of the maximum forces that the selected cylinder can exert are:

$$F_{traction}[kN] = 25{,}1\ kN \quad (55)$$

$$F_{compression}[kN] = 39{,}3\ kN \quad (56)$$

The values that the cylinder can produce are higher by far of what is needed so it assumed as verified that its functioning it is going to be correct. The safety factor that the crane will have with the selected hydraulic cylinder is calculated in equation (57).

$$SF = \frac{F_{traction}}{F_{cyl}} \cong 1{,}1 \quad (57)$$

## 5.5 Crane rotation mechanism

An important specification in this project is to design a mechanism that enables the portable crane to rotate the boom. There are many different ways to design a system that allows to do that kind of motion. Based on the reliability because it is mostly use in every rotating crane and the easiness of the installation a mechanism using a slewing bearing is design.

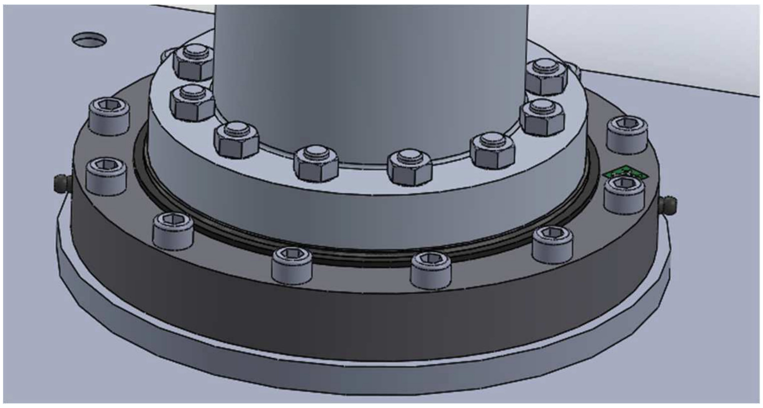
Generally, a slewing bearing consist in a special bearing with two races holding inside balls or rollers that enables the mechanism to rotate. Both races are attached with bolts so there is no need to think about what type of mounting it is going to be used as it would be when mounting angular contact bearings. In the slewing bearings both races have obviously different sizes but it is important to mention that some different configuration are found in the market, such as outer races with a spur gear machined in its surface.

The bearing is selected according the diameter of the main column which has to correspond to the diameter of the round tube profile of the main column. A 3D CAD model of the selected bearing is shown in Figure 33. None machined gear is needed given that the load is enough low to be moved by the workers by simply rotating the column.

***Figure 33.*** *3D model of the Schaeffler slewing bearing*

The attachment of the slewing bearing is done by 24 bolts whose dimensions are M8 as specified by the manufacturer (Figure 34).

***Figure 34.*** *Installation of the slewing bearing*



Thus, the slewing bearing is treated as a connection part between a fixed part (base support) and a moving part (main column). The bearing must be able to transmit the stress of the motion provoked by the rotation of the crane to the fixed part. It is very important to differentiate in the calculation the effect caused by the static load and the dynamic load because the latter will imply that the mechanism is subjected to a fatigue process.

According to the recommendations given by the manufacturer the forces in axial and radial direction as well as the overturning moment must be estimated. The axial force is easy to calculate because it is provoked by the vertical component of the force that the hydraulic cylinder exerts to the boom minus all the forces caused by the action on gravity on all the masses of the crane.

$$F_a = F_{cyl} - (m_{load} + m_{boom} + m_{winch} + m_{column})g = 2.258{,}4\ N \tag{58}$$

The only force in the radial direction of the bearing is the horizontal component of the hydraulic cylinder.

$$F_r = F_{cyl} \cdot tg(\theta) = 22.563\ N \tag{59}$$

The overturning moment that the slewing bearing is calculated as in equation (60).

$$\begin{aligned} M_{sb} = W_{load} \cdot L_1 + W_{boom} \cdot L_2 + F_{cyl} \cdot L_3 + F_{r,cyl} \cdot L_4 - W_{winch} \cdot L_5 \\ = 36.703{,}9\ Nm \end{aligned} \tag{60}$$

The distances used to calculate the value of the overturning moment are shown in Table 6.

***Table 6.*** *Distances to calculate the overturning moment*

| Ref. | Distance (mm) |
|---|---|
| $L_1$ | 1.240 |
| $L_2$ | 800 |
| $L_3$ | 360 |
| $L_4$ | 1.500 |
| $L_5$ | 550 |

The most important technical values concerning to the slewing bearing are extracted from the Schaeffler catalogue. The features of the bearing according to the manufacturer's technical data are listed in table.

***Table 7.*** *Basic features of the Schaeffler slewing bearing*

| Ref. | Description | Values |
|---|---|---|
| $d_i$ | Inner diameter | 124,5 mm |
| $D_a$ | Outer diameter | 234mm |
| $D_M$ | Rolling element pitch diameter | 180mm |
| M | Mass | 7kg |
| Fr | Maximum permissible radial load | 33,6kN |
| $C_a$ | Basic Dynamic load rating (axial) | 118kN |
| $C_{0a}$ | Basic static load rating (axial) | 179kN |
| $C_r$ | Basic dynamic load rating (radial) | 75kN |
| $C_{0r}$ | Basic static load rating (radial) | 88kN |

First of all the load eccentricity factor is calculated as indicated in the manufacturer guide.

$$\in = \frac{2.000 \cdot M_{sb}}{F_a \cdot D_m} = 185 \tag{61}$$

From the tables of the Schaeffler engineering manual the value of the Dynamic radial load factor ($K_f$) is assumed as approximately 25 (Schaeffler, 2017). The ratio between the radial and axial force is obtained by dividing them.

$$\frac{F_r}{F_a} = 9{,}99 \cong 10 \tag{62}$$

Therefore, following the manufacturer instructions the equivalent dynamic load is estimated.

$$P_{eq} = K_f \cdot F_a = 56.460\ N \tag{63}$$

The basic rating life of the bearing can be now calculated by using equation (67).

$$L_h = \frac{16.666}{2} \cdot \left(\frac{C_a}{P_{eq}}\right)^{\frac{10}{3}} = 97.031\ h \tag{64}$$

The total amount of hours estimated that the slewing bearing can be operational is considered to be acceptable given that the crane is not meant to be operating in a lot cycles.

## 5.6 Electric winch

The design of a worksite crane is always subjected to think about which kind of lifting device is going to be used. In many applications that worksite cranes are needed a system of pulleys that an experienced and qualified person for the purpose can operate in order to lift the load. That would require the design of a brake to assure the safety and also the reliability of the system to evade any possible scenario in which a failure due to either a mechanical system problem or a human mistake could happen.

By virtue of that possibility that option is discarded. A more reliable system should be used for this particular application. Therefore, in this project a commercial electric winch, model WT-180, of a manufacture named COMEUP is proposed to be the solution. Having a commercial product as a lifting device implies that a standardized device is installed in the crane and in case of failure or if by any reason de WT-180 is no longer in the market, any other commercial electric winches that fits the requirements could also work as well as a lifting device with minor adjustments to the final design of the crane.

The electric winch consist in a simple mechanism actioned by a 0,75kW per 4 poles induction brake motor which is connected to a Three-Phase Power and a frequency of 50Hz. The motor is connected to a drum by a gearbox which has 2 stages of planetary gears with a gear ratio of 36:1. That enables the electric motor to reduce the speed in order to gain enough torque to rotate a drum which has an outer diameter (including flange) of 185mm. This winding drum transmits power to the wire rope but it also serves as a reservoir given that the wire rope is stored in it by spoiling the cable.

Normally the wire is wrapped helically and when it reaches the end of the layer it automatically proceeds to start again in the first flange of the drum. The flanges of the drum are considered to be very important given that it is the only physical part of the drum that retains the wire. For example, if the unloading velocity is considerably high then, without any flanges or even with flanges with small dimensions, the wire could behave similarly to a rubber band and it could cause to jump over the flange edge.

The steel wire in this product has a diameter of 5mm. In this project, the winch is attached to the very end of the telescopic boom in order to gain weight in the rear part of the crane. That enables to reduce the amount of counterweight needed. The wire is conducted through all the length of the telescopic boom until the front part. At the end of the cable a hook is attached to the wire in order to attached the load and lift it. The total length of the steel wire of the WT-180 electric winch is 100m which according to the requirements is considered to be feasible.

The control of the winch is a low level control (24 VAC), which allows an operator to control the motion of the electric winch by a remote control. The remote control has 3 meters cord in order to avoid being close to the crane when it is completing its tasks. In case of accident or any possible harmful situation by pressing an emergency stop in the remote control a brake integrated in the motor stops the whole system and assures that the situation is ceased.

Calculations are not needed given that the winch is assured to be in order with the EN 14492-2. This directive assures that the working coefficient for the first rope layer on the drum is at least 5 and the ratio between the diameter of the drum and the diameter of the rope is at least 15.

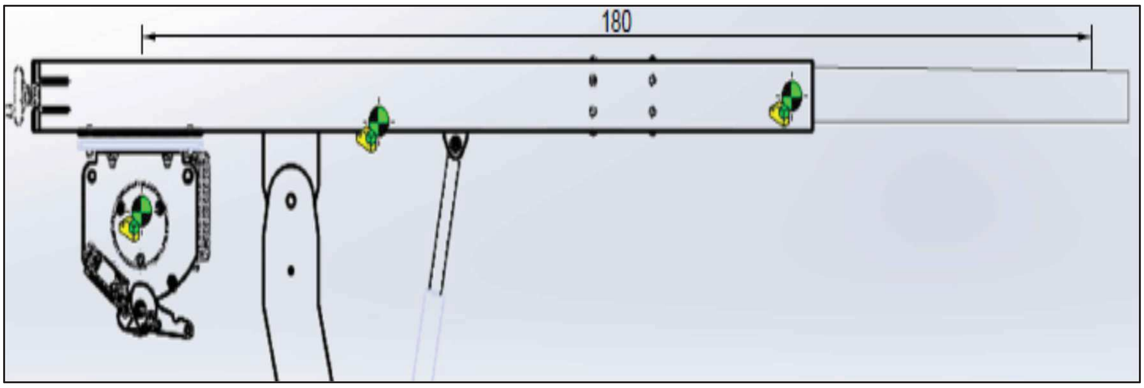
The lifting capability of the electric winch is shown in Table 8. which is extracted from the data given by the manufacturer (Comeup Industries Inc., n.d.)

***Table 8.*** *Lifting Capacity of the electric winch WT-180*

| Layer of Wire Rope | Lifting Capacity (kg) | Lifting Speed (m/s) | Total Wrapped Rope (m) |
|---|---|---|---|
| 1st | 252 | 17 | 14.5 |
| 2nd | 233.3 | 18.3 | 30.2 |
| 3rd | 217.2 | 19.7 | 47.1 |
| 4th | 203.2 | 21 | 65.1 |
| 5rd | 190.2 | 22.4 | 84.2 |
| 6th | 180 | 23 | 100 |

The exact position where the electric winch is installed in the telescopic boom is shown Figure 35. It can be seen that the distance between the center of mass of the electric winch and the load is set in 1,80m when the inner profile of the jig is completely gathered inside the outer profile.

***Figure 35.*** *Installation scheme of the electric winch*




The winch is attached to the boom by using bolts. The procedure to verify the joint of the pieces is extracted from the bibliography (Norton, 2006). The joint is verified against yielding and separation of both pieces.

First of all it is decided to use 4xM10x30 bolts of 8.8 class. The basic data used to calculate check whether the bolts are suitable or not is shown in Table 9.

***Table 9.*** *Resume of the data used to calculate the joint winch-telescopic boom*

| Ref. | Description | Values |
|---|---|---|
| P | External force | 1.765,8 N |
| Fi | Preload of the bolt | 27.300 N |
| $D_e$ | Diameter of the head of the bolt | 17 mm |
| $D_f$ | Diameter of the of bolt | 11,5 mm |
| $E_p$ | Young module of the pieces | 68900 N/mm² |
| $l_p$ | Length of the pieces | 9 mm |
| $A_t$ | Tensile-stress area of the bolt | 58 mm² |
| N | Number of bolts | 4 |
| $l_{thd}$ | Length of the threaded bolt | 30 mm |

The preload of the bolts is considered as 27.300N (Coral, 2016). The length of the threaded bolt is 30mm. The stiffness of the bolt can be estimated by using the theory of springs.

$$k_b = \frac{A_t \cdot E}{l_{thd}} = 0{,}406 \cdot 10^6 N/mm \qquad (65)$$

The stiffness of the pieces can be calculated using the equation (66).

$$k_p = \frac{\pi}{4} \cdot \frac{E_p}{l_p} \left[ \left( D_e + \frac{l_p}{10} \right)^2 - D_f^2 \right] = 1{,}131 \cdot 10^6 \; N/mm \tag{66}$$

The joint stiffness factor estimates the stiffness of the union between the bolt and the pieces.

$$C = \frac{k_b}{k_p + k_b} = 0{,}264 \tag{67}$$

The load provoked by the weight of the winch is divided in in two parts. A part of the load is felt by the bolt and the other part is felt by the material of the pieces.

$$P_b = C \cdot \frac{P}{N} = 116{,}54 \; N \tag{68}$$

$$P_m = (1 - C) \cdot \frac{P}{N} = 324{,}91 \; N \tag{69}$$

The resulting loads in both bolt and material are:

$$F_b = F_i + P_b = 27.416{,}54 \; N \tag{70}$$

$$F_m = F_i - P_m = 26.975{,}09 \; N \tag{71}$$

As indicated in the procedure no stress-concentration factor is used because it is a static calculation (Norton, 2006). The stress in the bolt is the bolt felt by the bolt divided by its tensile-stress area.

$$\sigma_b = \frac{F_b}{A_t} = 472{,}69 \; Mpa \tag{72}$$

The safety factor against a yielding failure is calculated in equation (73).

$$SF = \frac{\sigma_{b,material}}{\sigma_b} = 1{,}7 \tag{73}$$

The load that provokes the separation of the joint is found by using equation (74).

$$P_o = \frac{F_i}{1 - C} = 37.092 \; N \tag{74}$$

The safety factor against separation of the joint is calculated in equation (75).

$$SF = \frac{P_o}{F_m} = 1{,}38 \tag{75}$$

Thus, it is assumed as verified that the bolts are suitable for the desired application. Note that there is no calculation regarding the length of the bolt. The only important length is from the length between the top of the bolt that is making contact with the top piece of the joint to the start of the nut at the end of the joint. That is the only length that makes the joint to change its stiffness. Therefore, the only recommendation about the total length is that if the material of both bolt and nut are the same the minimum length of the bolt should be at least equal to 80% of the diameter of the bolt (Coral, 2016). That will assure that the nut does not loosen from the thread of the bolt.

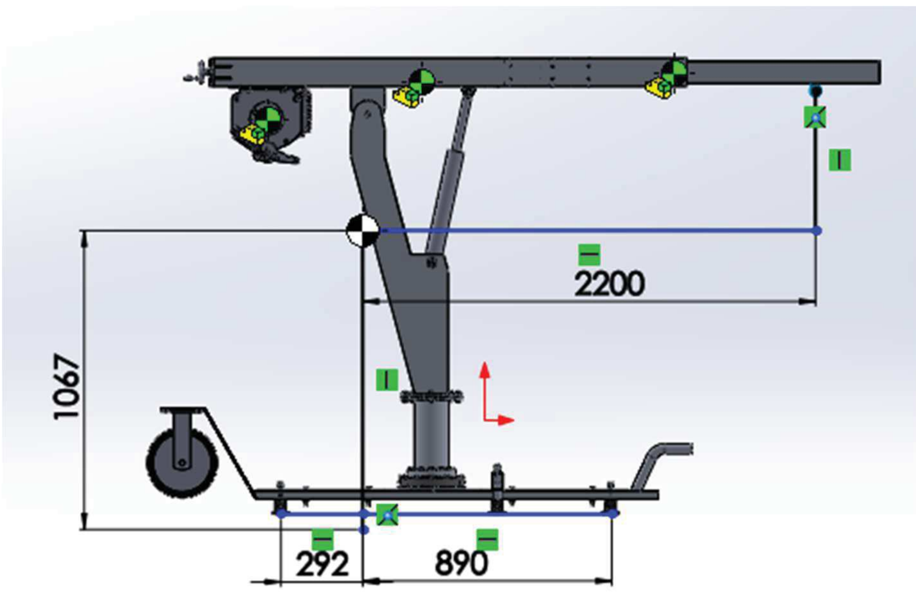
## 5.7 Counterweight

Cranes are lifting devices that are subjected to great amount of loads. That leads to complicate situations where the weight of the crane and its supports cannot withstand by itself the overturning moment of that the lifting loads applies to the crane. Due to that reason it is very common to find cranes using a countermeasure to avoid the machine to tip over: the counterweight.

A counterweight is a mass which is placed where at the opposite side where the overturning moment tends to tip over the crane. That enables to increase the moment in the opposite direction and maintain the unit stable with its supports making contact with the terrain.

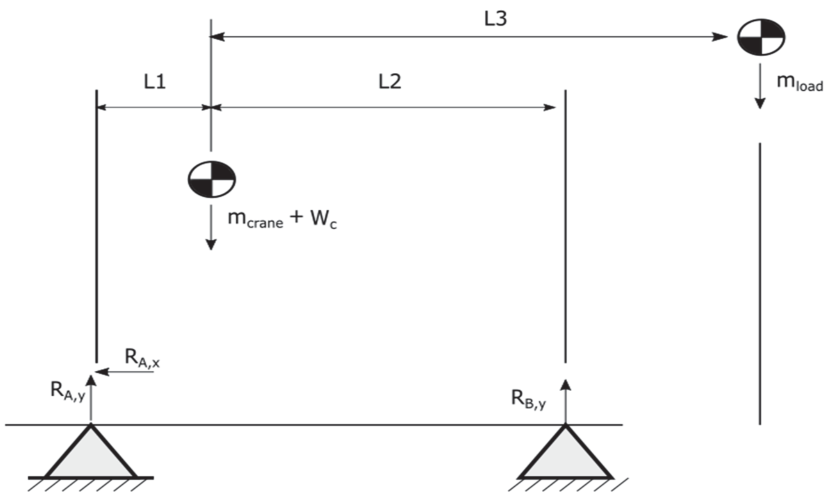
In this project the worse scenario is when the telescopic boom is totally extended. All measures are taken in the 3D CAD model as shown in Figure 36. To simplify the math, all measures are taken form the center of mass of the crane.

***Figure 36**. Distances to estimate the counterweight*



In order to calculate the amount of mass to be added to gain enough stability simple physics are used. The free-body diagram shows all the forces involved.

***Figure 37.*** *Free-body diagram of the counterweight problem*




The tip over will start to be produced when the vertical reaction in point A is equal to zero (just the instant before the dynamic problem starts). In order to simplify the calculation, it is considered that the center of gravity will not change its position due to the counterweight and therefore this load is applied at the same center of mass. By doing this, a worse scenario is considered given that the counterweight will change the position of the center of mass so that more stability will be gained. Assuming equilibrium and taking moments in B equal to zero.

$$(W_c + m_{crane} \cdot g) = m_{load} \cdot g \cdot (L_3 - L_2) \quad (76)$$

Solving the equation (76) the mass of the counterweight can be found.

$$W_c = 116{,}5\ kg \quad (77)$$

The amount of weight needed to avoid the tipping of the crane is at least 116,5kgs. As it can be seen in Figure 36 there is a space in the rear part of the crane, between the main column and the wheels. That is designed on purpose to place the counterweight. Given that the crane is going to be used inside of flooded mines it suggested to use plastic box which its weight is reduced and a person can carry it inside the mine without any problem. By simply adding water to the plastic box the desired amount of counterweight can be reached easily.

# 6. ASSEMBLY

This chapter explains how it is recommended to assembly the portable crane unit. It is not a complex machine to be mounted but it requires organization and concentration given that many different bolts, pins and so on are used to maintain the components all together and is easy to get confused even for an experienced worker.

The assembly procedure that is proposed here might not be the optimal and therefore there is not only one way to assembly the whole crane. It is on the hand of the responsible of the assembly of the crane to decide which path to follow based on its own judgement.

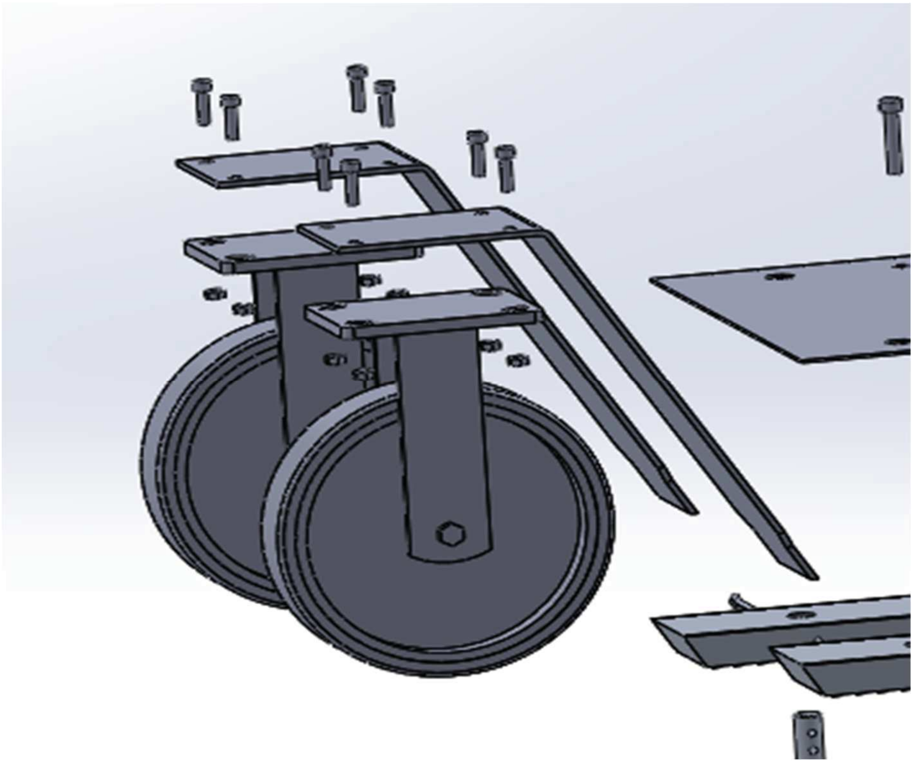
## 6.1 Assembly procedure

The crane consist in many different pieces that shall be mounted all together. An exploded view is shown in Figure 38. In order to make the discussion of the assembly easier the explanation it will be discussed using the main components described in this project as a reference.

***Figure 38.*** *Exploded view of the crane*

First of all, the base support is mounted. Two aluminum profiles of 85x30 are attached to the main aluminum plate by 8xM10x55 ISO 4762 bolts. Another profile of 40x60 which used as a lateral leg support is attached to the main aluminum plate using 3x M10x80. All supports are mounted in the same way as shown in Figure 25. A cotter is used to block the vertical motion of the supports while a clamped base with a M8 bolt is used to fix the support to the aluminum cylinder. Two aluminum plates are welded to the profiles of 40x60 so that the wheels can be attached to the base support by means of a 4xM8x30 ISO 4762 bolts. This is shown in Figure 39. Having done all this steps the base support is completely assembled.

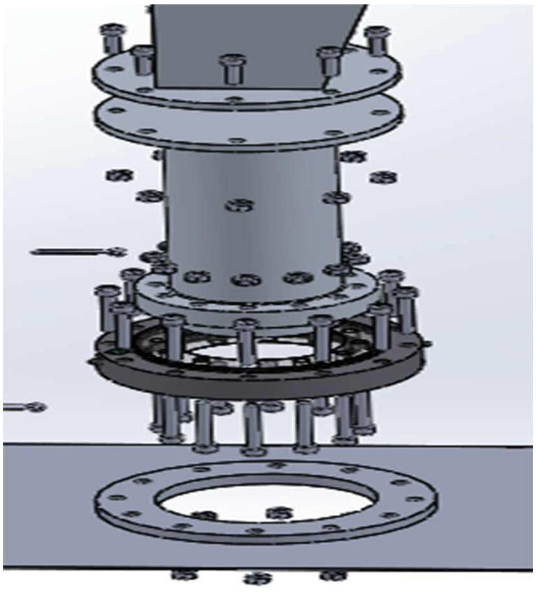
***Figure 39.*** *Assembly of the wheels*



It is considered to be easier to start by installing the slewing bearing to the main column rather than to the base support because at the time that the main column is placed into the base support, both components will be set up.

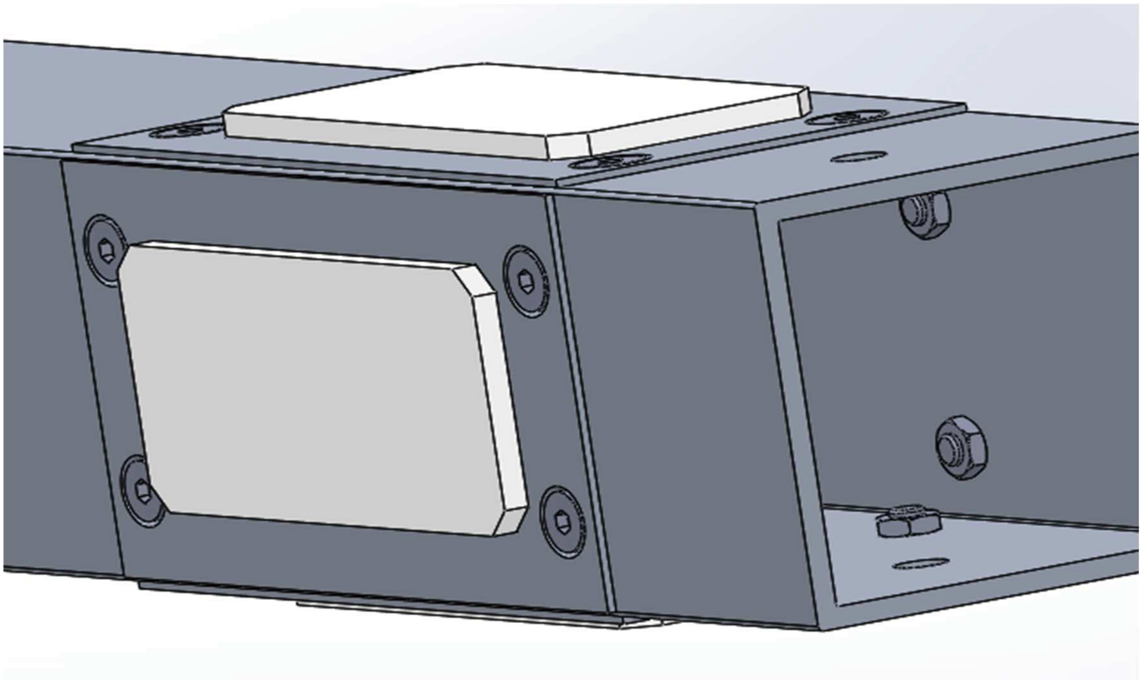
Hence, the inner race of the slewing bearing is attached to the main column rounded tube profile of diameter 125mm by means of 12xM10x65 (diameter and number of bolts indicated by the manufacturer) and using M10 nuts at the main column side. The square profile of 120x120 can now fixed to the rounded profile by means of 8xM10x30 and M10 nuts. The assembly of the main column is complete and now it can be placed into the machined hole to install the slewing bearing with also 12xM10x65 and M10 nuts at the main aluminum side (on its bottom). Figure 40 shows the exploded assembly in detail.

***Figure 40.*** *Assembly of the slewing bearing and the column*



The next step should be the assembly of the telescopic boom. First, of all the plastic plates are mounted on a small aluminum plate using 4xM6 ISO 10642 (Figure 41). Is important to use a bolt that its height of the head does not overcome the height of the plastic. Otherwise, an important failure of the system may occur. At the end, 8 assembled plastic plates should ready fixed to the booms. Four of them will be fixed to the inner profile and the rest to the outer profile.

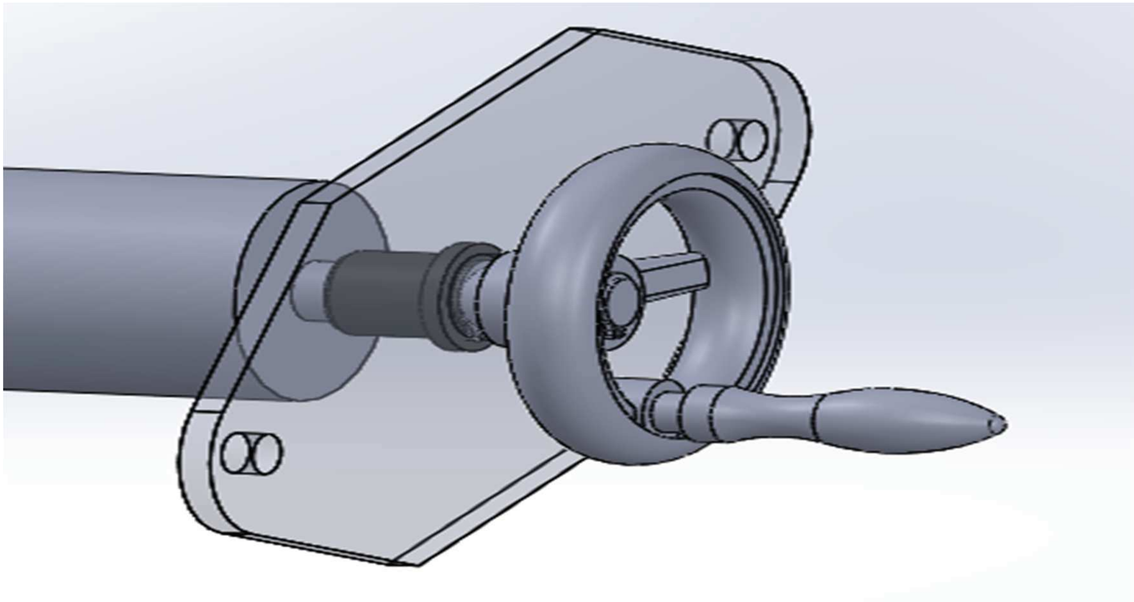
***Figure 41.*** *Plastic plates assembly to the telescopic boom*



A small pulley, where the end of the cable is held so that the hook is suspended vertically, is attached at the extreme of the inner profile of the telescopic boom by using 4xM4x15 ISO 4762 bolts.

It is now the turn of the power threaded screw mechanism. A threaded nut is attached to a machined aluminum cylinder 6xM4x30 bolts. At the opposite side of the nut a bushing with a retaining ring are mounted. Next to them a wheel handle blocked axially with another retaining ring is assembled as shown in Figure 42.

***Figure 42.*** *Wheel handle assembly*



The power threaded screw is introduced inside the nut and at the opposite side a bushing is used to avoid the direct contact of the power screw with the aluminum plate that is used as support. The bushing is also blocked with a retaining ring. Between the bushing and the wheel handle an aluminum plate is fixed to seal the telescopic boom and avoid any unwanted particle to enter where the mechanism is held. This whole assembly can be seen in Figure 43 so that the reader better understands these steps.

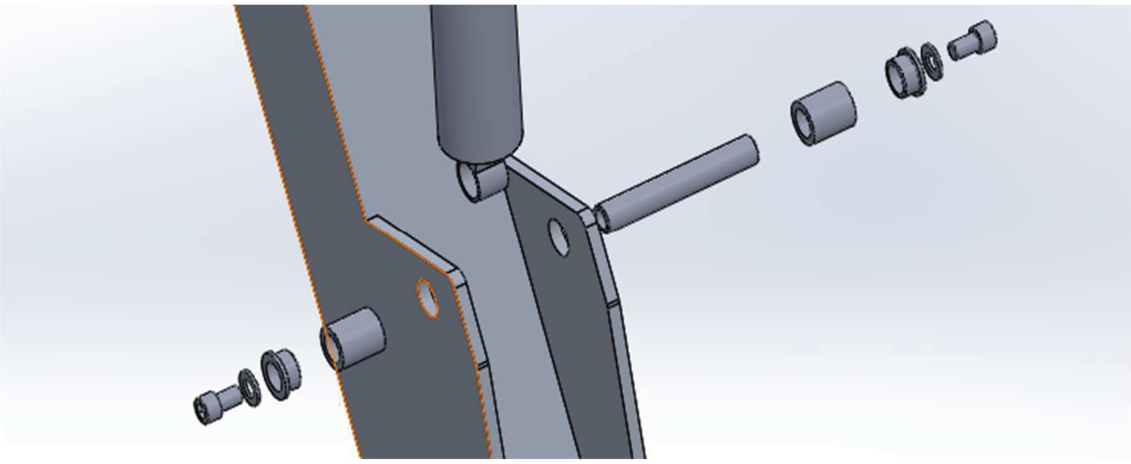
***Figure 43.*** *Power threaded screw assembly*

The whole power screw mechanism is introduced through the bigger profile of the telescopic boom and by the other side the plate shown in Figure 43 is attached to the inner profile by virtue of 2xM8x15 ISO 4762 bolts. So as to end the assembly of the telescopic boom the plate which is shown in Figure 42 is attached to the very end of the outer profile of the telescopic boom by using 2xM10x30 ISO 4762 bolts. To finalize the assembly of the telescopic boom a plate is attached to the extreme of the outer profile by using 4xM10x25 ISO 4762 bolts and M10 nuts. This plate will be used to fix the electric winch to the crane at the end of the process.

The telescopic boom is now ready to be assembled with the main column. The system to do that is a simple cotter of 10mm of diameter which is introduced through a drilled M10 hole and then the extreme of the pin is bended so that the components are absolutely blocked axially. As an alternative rounded pins with a proper axial blocking (such washer plus a retaining ring in both sides) could be used.

The hydraulic cylinder can be installed to the main column by using a precision tube with two separators to block the cylinder axially and two bushing at both sides to avoid a metallic contact between the tube and the holes of the main column. To block the bushing a washer with an M12x20 ISO 4762 is used on both sides (Figure 44).

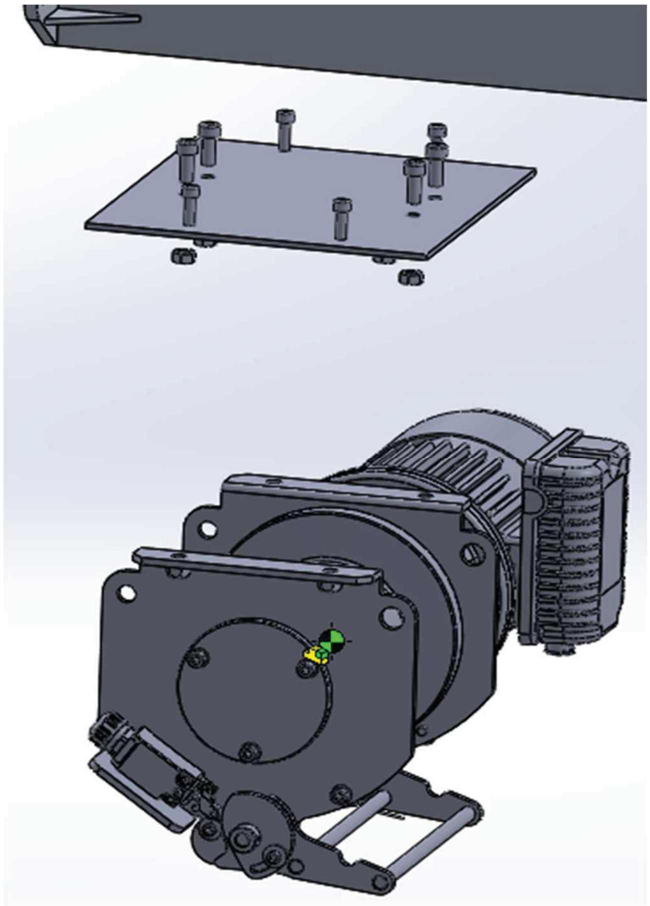
***Figure 44.*** *Assembly of the hydraulic cylinder*



The same strategy is used to attach to hydraulic cylinder to the telescopic boom but instead of using a washer and a bolt, a retaining ring is used.

The last part of the assembly is to fix the electric winch to the telescopic boom. For this purpose the manufacture has a support structure where, in this case, by using 4xM8x30 the electric winch is completely attached to the structure (Figure 45).

***Figure 45.*** *Assembly of the electric winch*



At this point the crane is totally assembled and ready to be used. Before starting the desired lifting operations it might be a good idea to reassure the reliability and the safety of the crane. For instance, verifying that the slewing bearing is lubricated, that the plastic plates do not show excessive wear. A must would be to inspect visually the crane and test the proper functioning with minor loads.

# 7. RELIABILITY OF THE PORTABLE CRANE UNIT

During the development of this chapter the reliability of the portable crane unit will be discussed. For that reason an important engineering tool "*Failure mode effects analysis (FMEA)*" is used. This tool is commonly used in design task as well as in administration and quality task.

The FMEA is an analysis that is used to identify, evaluate and prevent possible failures or effects that might be found in a product or process. It is considered to be of a great importance because it allows the final user of the design to reduce or even completely avoid future problems.

## 7.1 Failure mode effects analysis of the crane (FMEA)

First of all, a briefly explanation of how a FMEA analysis work is discussed in this sub-chapter. Principally, there are two types of failure mode analysis that can be differentiate:

- FMEA of products or designs
  Generally, the analysis is done to investigate preventively the design of the products or services, including components, subsystems, etc.
- FMEA of processes
  The analysis is focused on analyzing the possible failures on the process (materials, team work, execution, methodology, environments, sales, etc.) and its influence to the final result.

In this project the FMEA is based on the design of the crane. In order to complete the FMEA three concepts have to be taken into account (E. McDermott, J. Mikulak, & R. Beauregard, 2009).

- Severity
- Occurrence
- Detection

This three concepts will be evaluated as shown in the tables below according to a common keys (E. McDermott et al., 2009).

*Table 10. Guidelines of rating the severity of the failure. FMEA*

| SEVERITY | | |
|---|---|---|
| **CRITERIA** | **IMPORTANCE OF THE FAILURE** | **RATING** |
| None | No relevant effect on reliability or safety | 1 |
| Very Minor | Failure not readily apparent to the customer | 2 |
| Minor | Failure noticed by the customer but unlikely have a warrant complaint | 3 |
| Very low | Failure that can be easily overcome by the customer | 4 |
| Low | Failure likely to cause isolated customer complaints | 5 |
| Moderate | High degree of dissatisfaction and numerous complaints | 6 |
| High | Customer percepts a safety issue. High degree of dissatisfaction. Re-call for business reasons might be possible. | 7 |
| Very High | Failure could lead to adverse reaction for customer. Failure possible lead to recall. | 8 |
| Extremely High | Failure could lead to injury to the customer. | 9 |
| Dangerously High | Failure could lead to death or permanent injury to the customer | 10 |

The evaluation of occurrence is briefly discussed in Table 11. In this case the probability or the frequency of the failure occurring is subjectively evaluated.

*Table 11. Guidelines of rating the occurrence of the failure. FMEA*

| OCCURRENCE | | |
|---|---|---|
| **CRITERIA** | **IMPORTANCE OF THE FAILURE** | **RATING** |
| Remote | One occurrence in greater than five years | 1 |
| | Once occurrence every three to five years | 2 |
| Low | Once occurrence every one to three years | 3 |
| | Once occurrence per year | 4 |
| | Once occurrence every six months to one year | 5 |
| Moderate | Once occurrence every three months | 6 |
| | Once occurrence every month | 7 |
| High | One occurrence every month or a probability of 5 occurrences in 100 units | 8 |
| | Once occurrence every three to four days | 9 |
| Very high | More than one occurrence per day | 10 |

To quantify the detection Table 12 explains all the possible choices.

*Table 12. Guidelines of rating the detection of the failure. FMEA*

| DETECTION | | |
|---|---|---|
| **CRITERIA** | **IMPORTANCE OF THE FAILURE** | **RATING** |
| Almost certain | The defect is obvious and detected with an ordinary inspection | 1 |
| Very High | All product is 100% automatically inspected | 2 |
| High | An effective statistical process control (SPC) program with process capabilities greater than 1.33 | 3 |
| Moderately High | SPC is used and there is immediate reaction to out-of-control conditions | 4 |
| Moderate | SPC is used in the process and product is final inspected off-line | 5 |
| Low | Product is 100% manually inspected using mistake-proofing gauges | 6 |
| Very Low | Product is 100% manually inspected | 7 |
| Remote | Product is accepter based on no defects in a sample | 8 |
| Very Remote | Product is sampled, inspected and released based on acceptable quality level sampling plans. | 9 |
| Absolute uncertainty | Product is not inspected or the defect cannot be detected | 10 |

Having quantified all the three concepts by multiplying the value assigned of each of them a value will be obtained. That value is named risk priority number (RPN). Its maximum value corresponds to 100. If the value is overcame by a possible failure then a plan to fix it or reduce its potential failure must be elaborated and executed.

The FMEA obtained for the application of this project is shown in Table 13. It is confirmed that it is of a high importance to have a proper control of the plastic plates installed inside the telescopic boom. That is because is the only way in the mechanism to avoid to contact between two metallic parts which would produce excessive wear of the aluminum profiles, noise over the regulations and it is likely to provoke a failure with further consequences.

Although it will hardly occur, it is also remarkable that the slewing bearing must be lubricated during the operation. It is one of the components that has a preventive maintenance to be carried on. Fortunately, it is not an expensive component so that it can be easily replaced if it is considered suitable.

***Table 13.*** *Failure mode and effects analysis of the crane*

**FAILURE MODE AND EFFECTS ANALYSIS**

| | | | | | |
|---|---|---|---|---|---|
| Item: | Portable Crane Unit | Responsibility: | Bastian Bravo Gutiérrez | FMEA number: | 1 |
| Model: | | Prepared by: | Bastian Bravo Gutiérrez | Page : | 1 of 1 |
| Core Team: | Bastian Bravo Gutiérrez | | | FMEA Date (Orig): | 18/07/2017 Rev: 1 |

| Process Function | Potential Failure Mode | Potential Effect(s) of Failure | Sev | Potential Cause(s)/ Mechanism(s) of Failure | Occur | Current Process Controls | Detec | RPN | Recommended Action(s) | Responsibility and Target Completion Date | Action Results | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | | | | | Actions Taken | Sev | Occ | Det | RPN |
| Materials | Incorrect materials. Materials do not match with the specification | Possible structural failure | 9 | Provider made an error selecting the materials | 2 | Inspection at the reception of the components | 1 | 18 | | | | | | | |
| | Material is damaged | Assembly is not possible | 9 | The provider or the manufacturer has not detected the failure | 1 | Inspection at the reception of the components | 1 | 9 | | | | | | | |
| Power Screw | Self-locking | Inner profile will not stay locked | 7 | Manufacturing problem of the threads | 2 | Testing the component after manufacturing | 6 | 84 | | | | | | | |
| | Defect or failure in a bushing | Excessive wear | 7 | The bushing has completed is lifecycle | 1 | 100% visual inspection | 6 | 42 | | | | | | | |

| | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Telescopic boom | Wear or defect on the plastic plates | Incorrect functioning of the mechanism | 4 | The provider has not detected the failure | 4 | 100% visual ins-pection | 6 | 96 | | | | | | | |
| Hydraulic cy-linder system | Sealing failures | Possible leak of fluid | 3 | Provider has not detected the failure | 4 | Inspection after operation | 6 | 72 | | | | | | | |
| | Incorrect func-tioning of the cylinder | Possible broken hand pump | 9 | Provider has not detected the failure or incorrect as-sembly of the sys-tem | 1 | Testing before operation | 2 | 18 | | | | | | | |
| Main column | The attache-ment to the tel-escopic boom is damaged | The cotter is under a suffering from a non-expected overstress situation. | 9 | Overloaded crane | 1 | 100% visual ins-pection | 6 | 54 | | | | | | | |
| Slewing bea-ring | Lack of lubrica-tion | Incorrect component set up | 8 | The responsible for the mainte-nance forgot to lu-bricate the bearing during its assem-bly | 1 | 100% visual ins-pection | 6 | 48 | | | | | | | |
| | The bearing does not rotate correctly | Not enough quality sur-face where the bearing is mounted | 4 | Incorrect machi-ning | 3 | Testing before operation | 2 | 24 | | | | | | | |

| | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Base support | The base is not stable | Supports are not properly in-stalled | 6 | Incorrect assembly | 4 | Testing before operation | 2 | 48 | | | | | | | | |
| | Break of the aluminum plates that hold the wheels | The weld-ing of the plates is not cor-rectly exe-cuted | 1 | Lack of tempera-ture in druing the welding or need of a bigger seam weld | 4 | 100% visual ins-pection | 6 | 24 | | | | | | | | |

# 8. CONCLUSIONS

From the beginning of this project is has represented a challenge to the author. As a result of that it is concluded that the goal of this project is achieved. A customized crane can be manufactured to be used for the desired application inside the mines.

It is very important to remark that the most important part of this project is the design itself. The design understood as the whole procedure where the engineer takes into account all the variables, specifications and requirements given from the beginning until the end of the design. The design of a mechanical machines requires of an iterative process where at least needs four points to be checked before the final design is considered as ended:

- Defining specifications and requirements
- Design of the desired machine
- Verification and validation of the design by means of both hand calculations and FEA analysis. Redesign if needed.
- Elaborate final blueprints to start manufacturing

That is why it is precisely a work that is made by engineers. The advanced knowledge about physics, math and mechanical engineering systems is considered to be a requirement for someone that starts a design. However, it might be not enough. A design involves an absolutely well-organized project where it is easy to get lost. Throughout the elaboration of this work it is essential to work in concordance with the new needs that it might appear as long as the project advances and that usually means that the requirements are changed or even that new ones emerge.

Regarding to the mechanical engineering aspects of the project, all the requirements have been satisfied:

- The portable crane has been designed mostly in aluminum 6063 T4 which allows the final design to have a lower weight compared to other materials such as the steel. The structural analysis has been shown that the material is good enough to be considered as suitable for the application.
- The crane has been designed so that most pieces can be carried and assembled in the working place. The different components of the crane are attached to each other by means of bolts, pins, cotters and so on.
- The selected electric winch allows to lift the desired load and its cable has a length of 90 meters.
- The total height of the crane is at least 1,5 meters and it is achieved basically by a proper design of the main column. Note that adjusting the design to other different

height might result easy with the 3D CAD model and recalculating all the structural parts as explained in the project.

- The need of an adjustable boom has been designed with a telescopic boom which can be easily control by a person applying force to a handle. It is a reliable mechanism and it does not need any expertise from the worker to operate it.
- The crane has a slewing bearing attached to the main column which allows the crane to rotate the telescopic boom. It also enables to withstand the forces and the overturning moment that provokes the load and the masses of the components of the crane
- There is no mechanism that has an electronic device rather than the electric winch. Nonetheless the control of the winch does not compromise the safety and reliability of the winch. The mechanism used in the project are purely based in mechanical parts and all of them have been verified to be safety enough to be considered suitable.

# 9. BIBLIOGRAPHY

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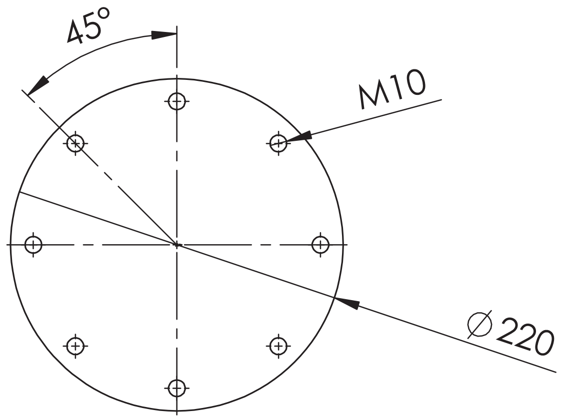
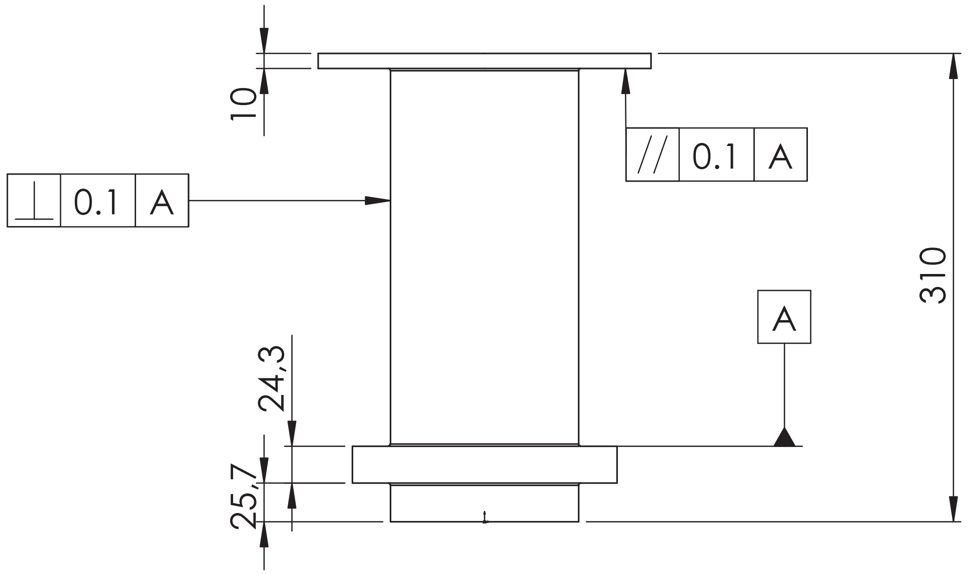
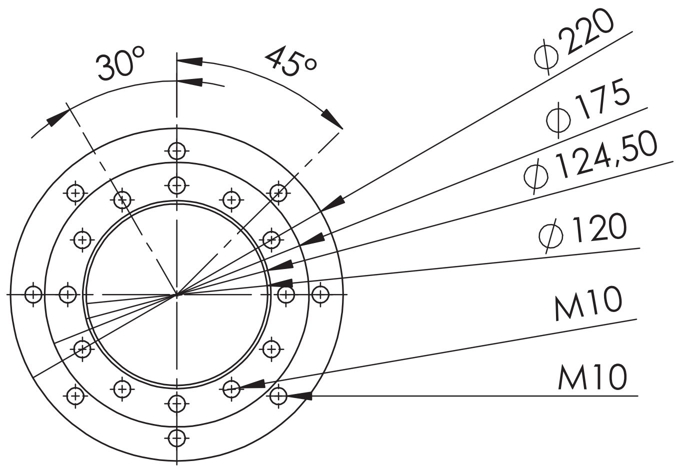
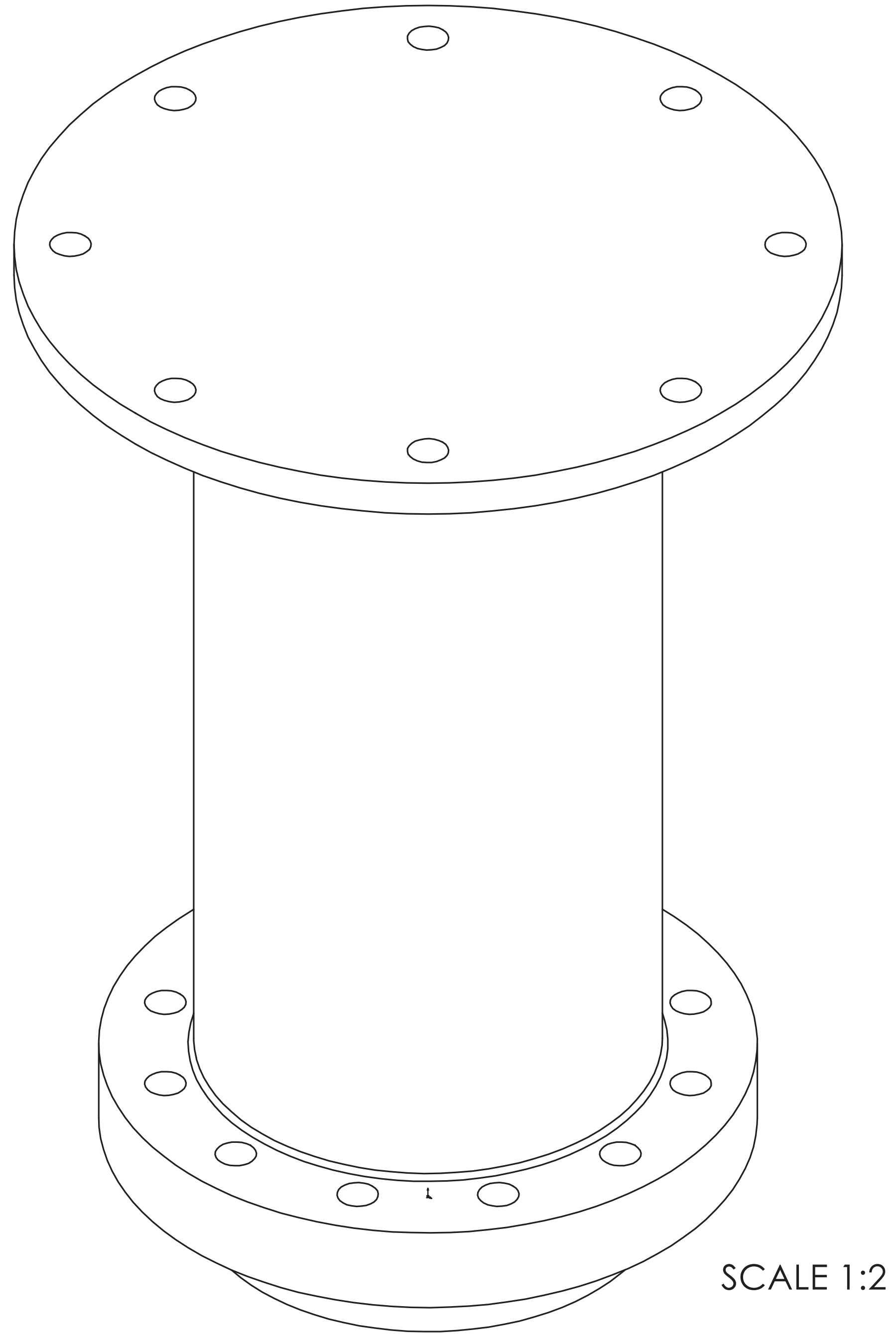
# APPENDIX A: BLUEPRINTS



SCALE 1:2

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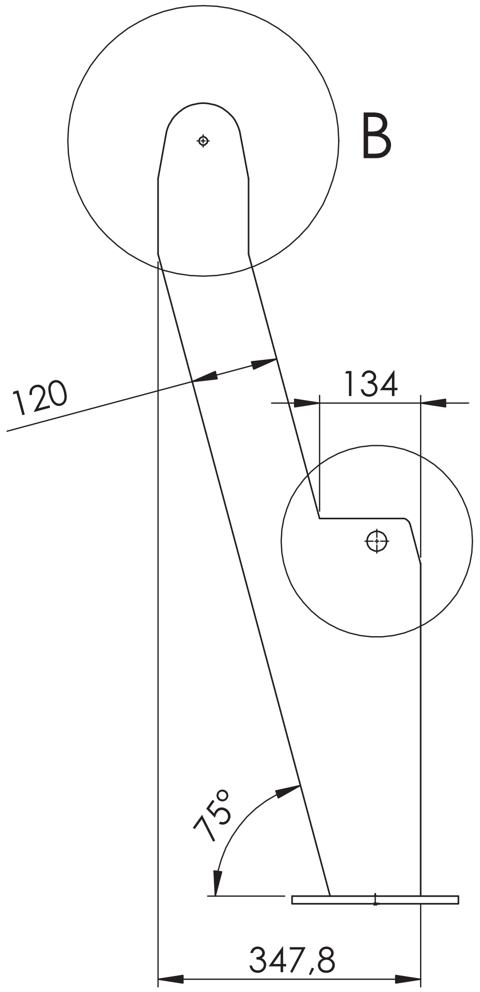


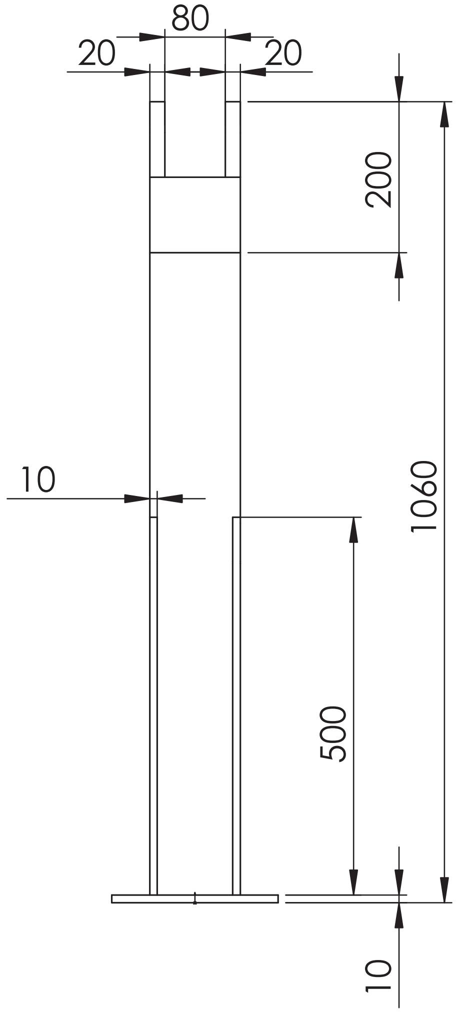


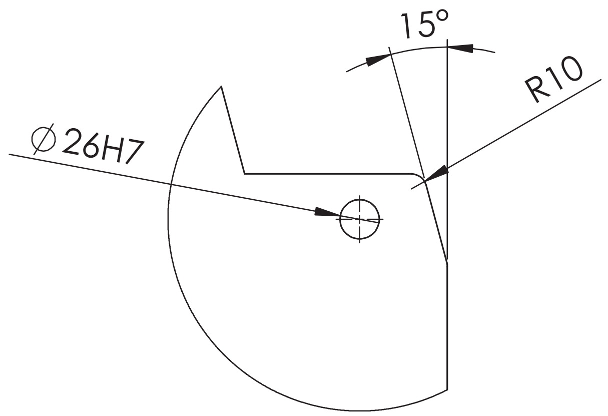


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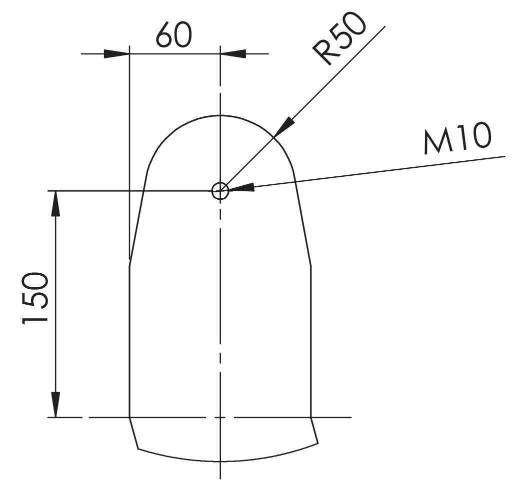


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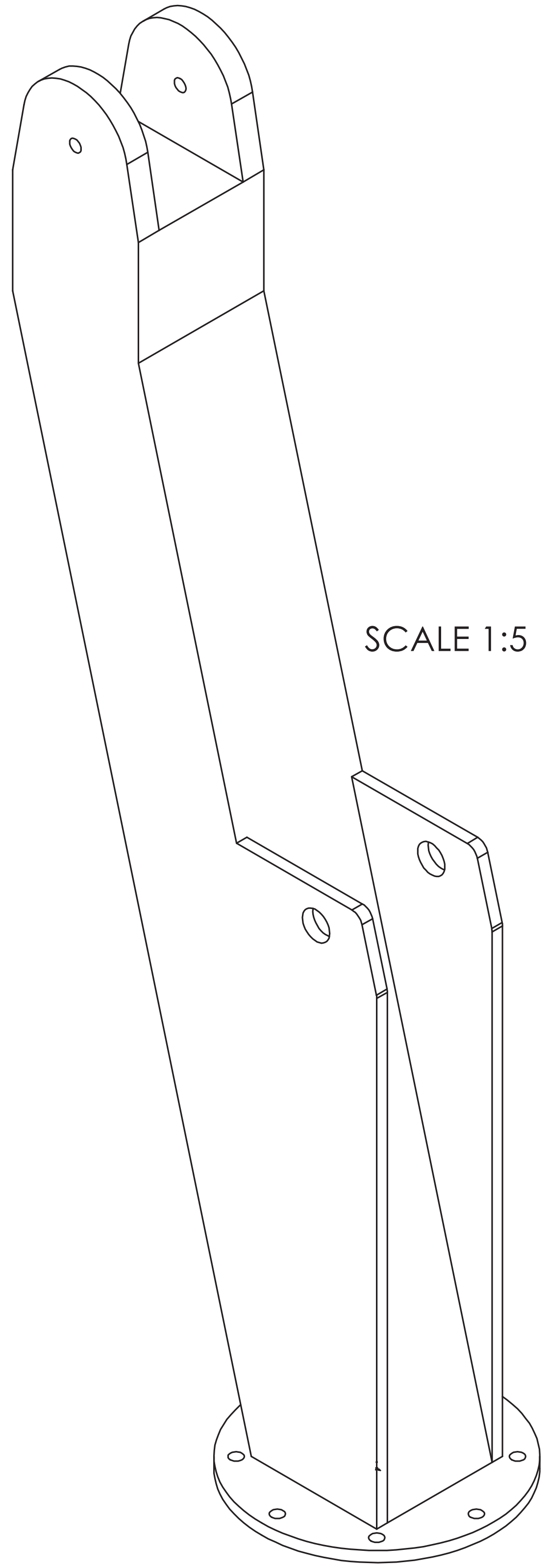

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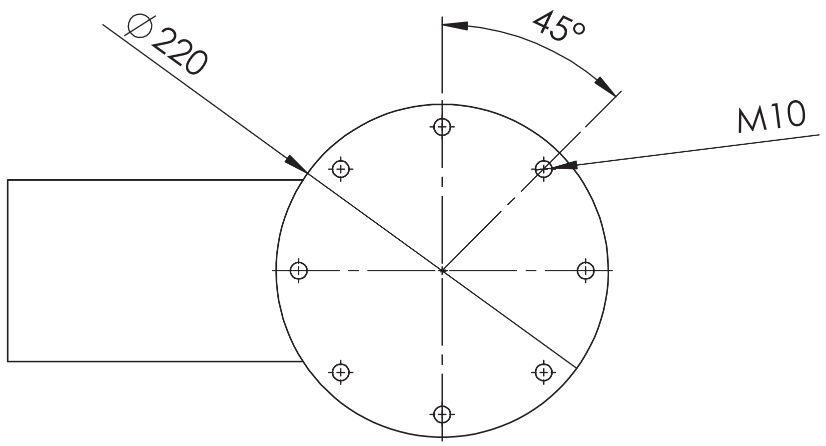


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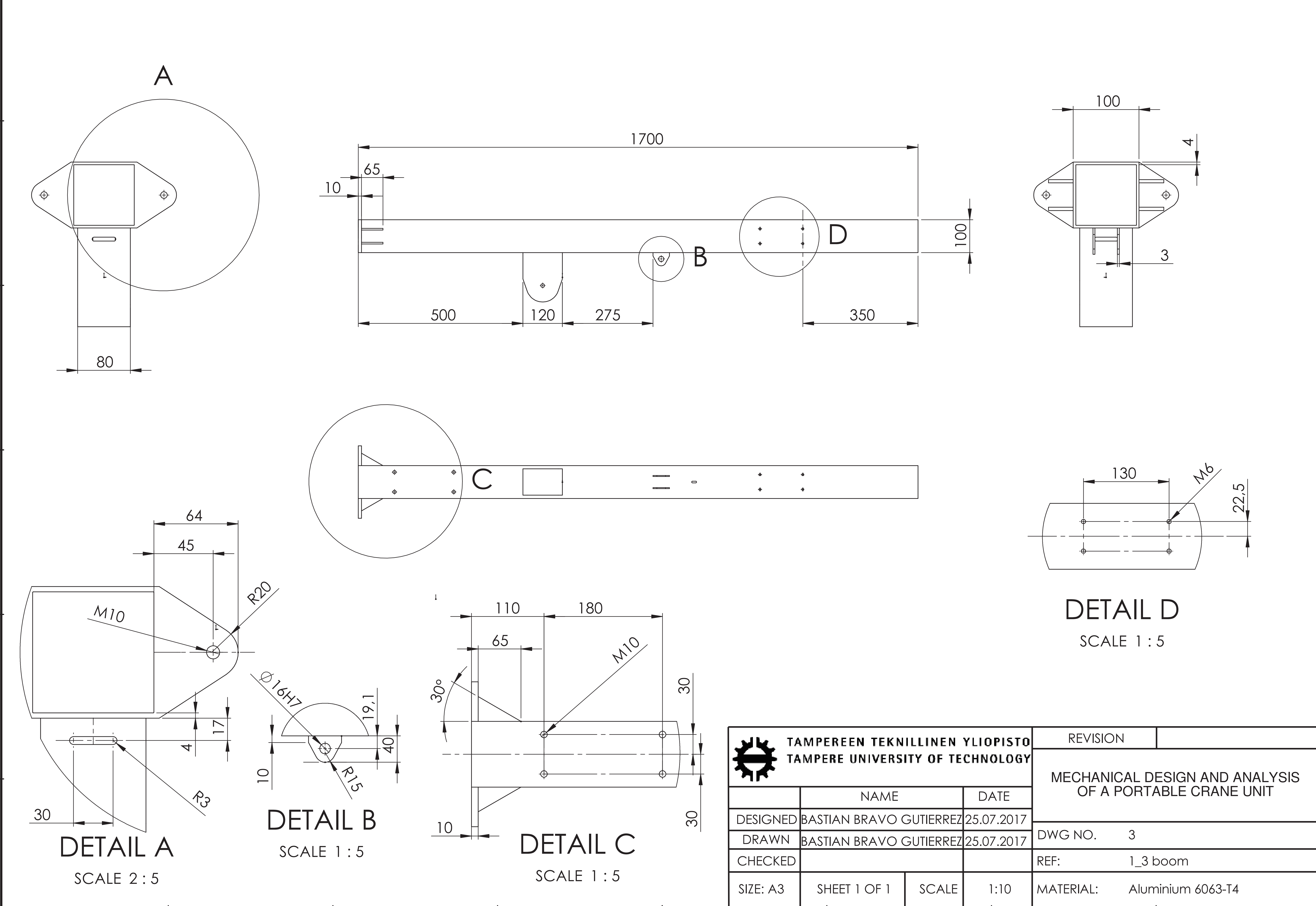


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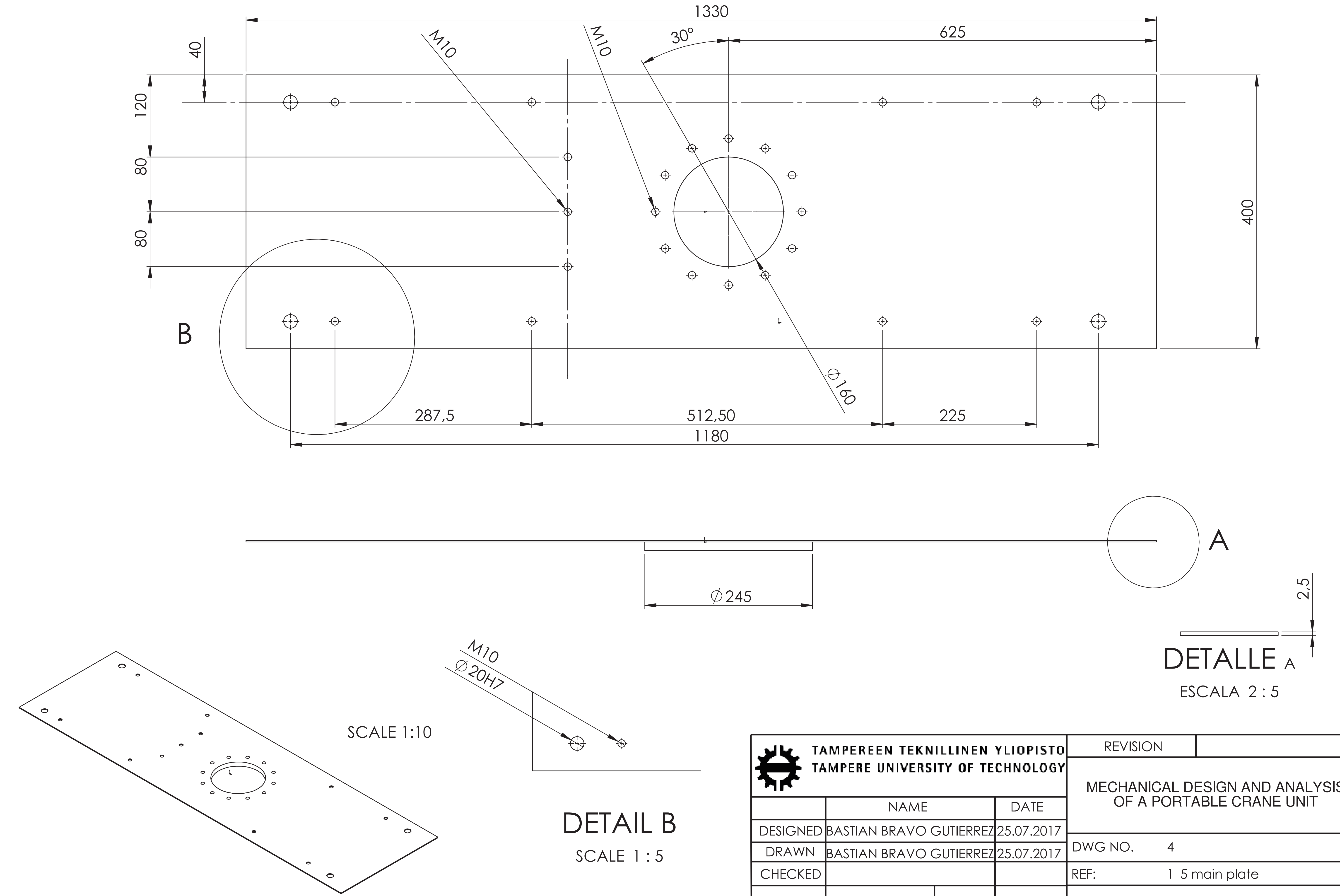


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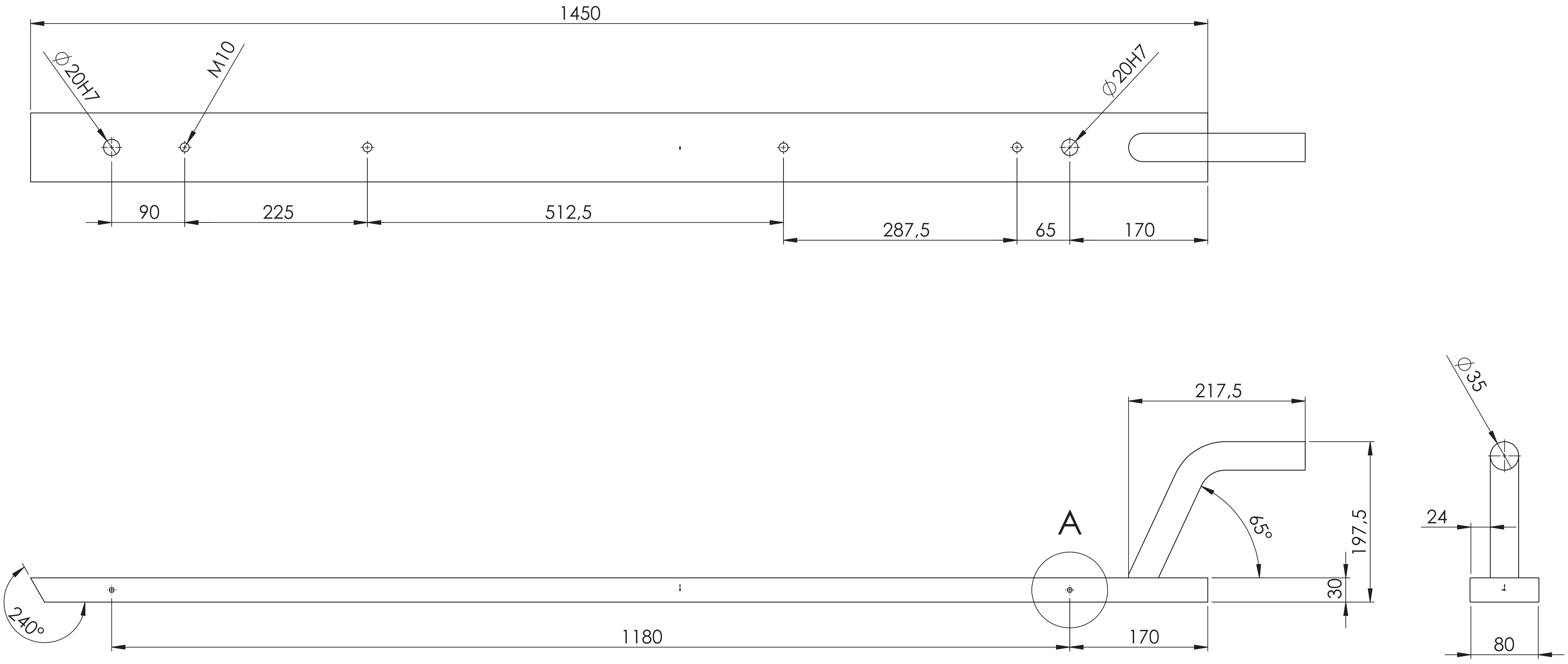


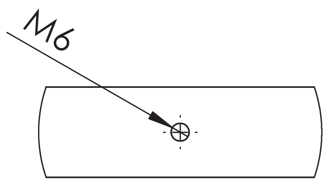


DETAIL A

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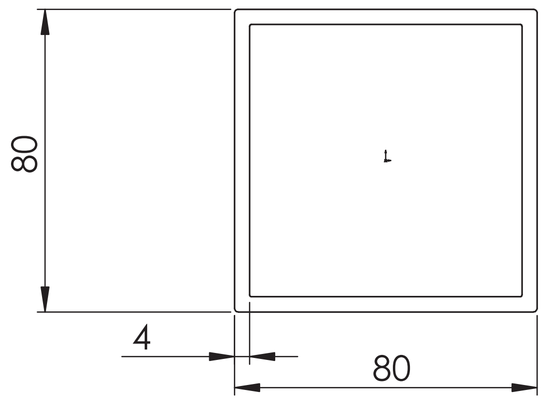


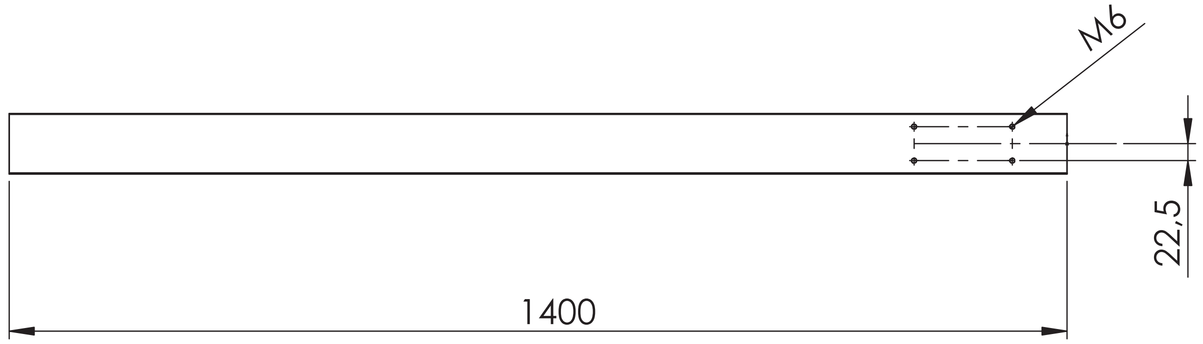


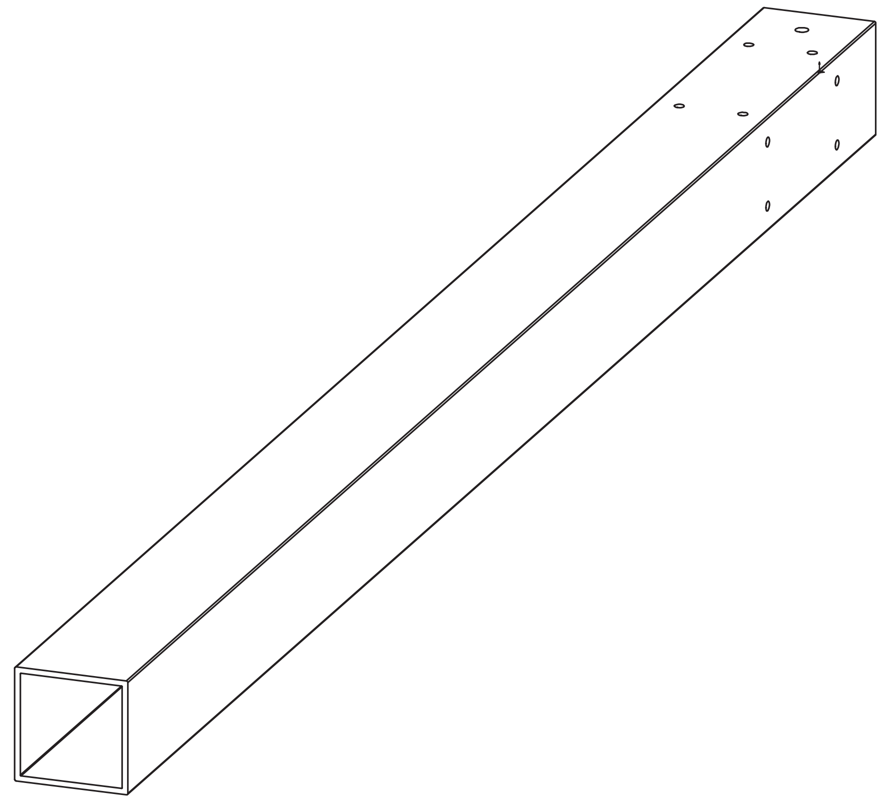

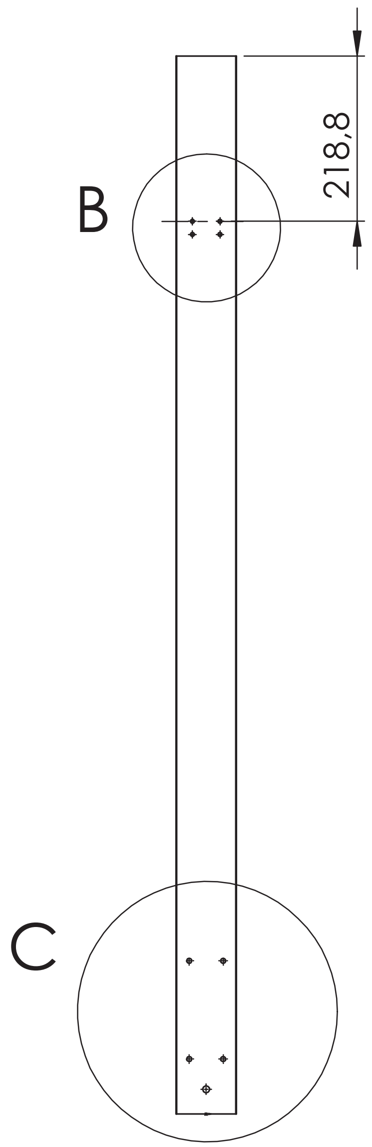


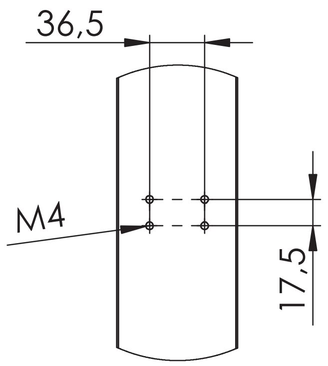


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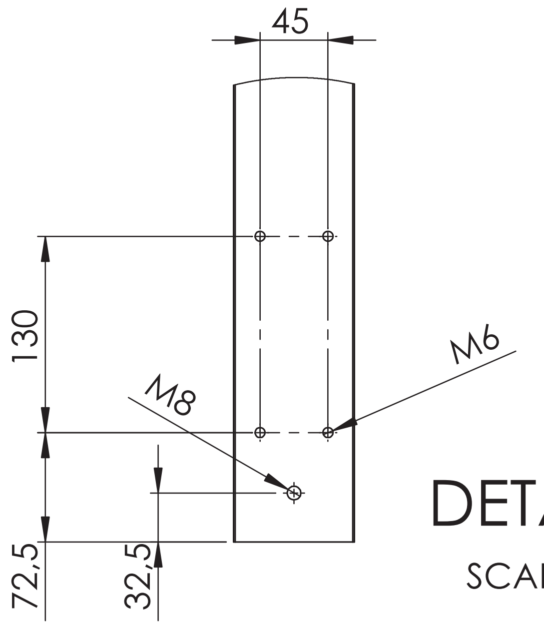


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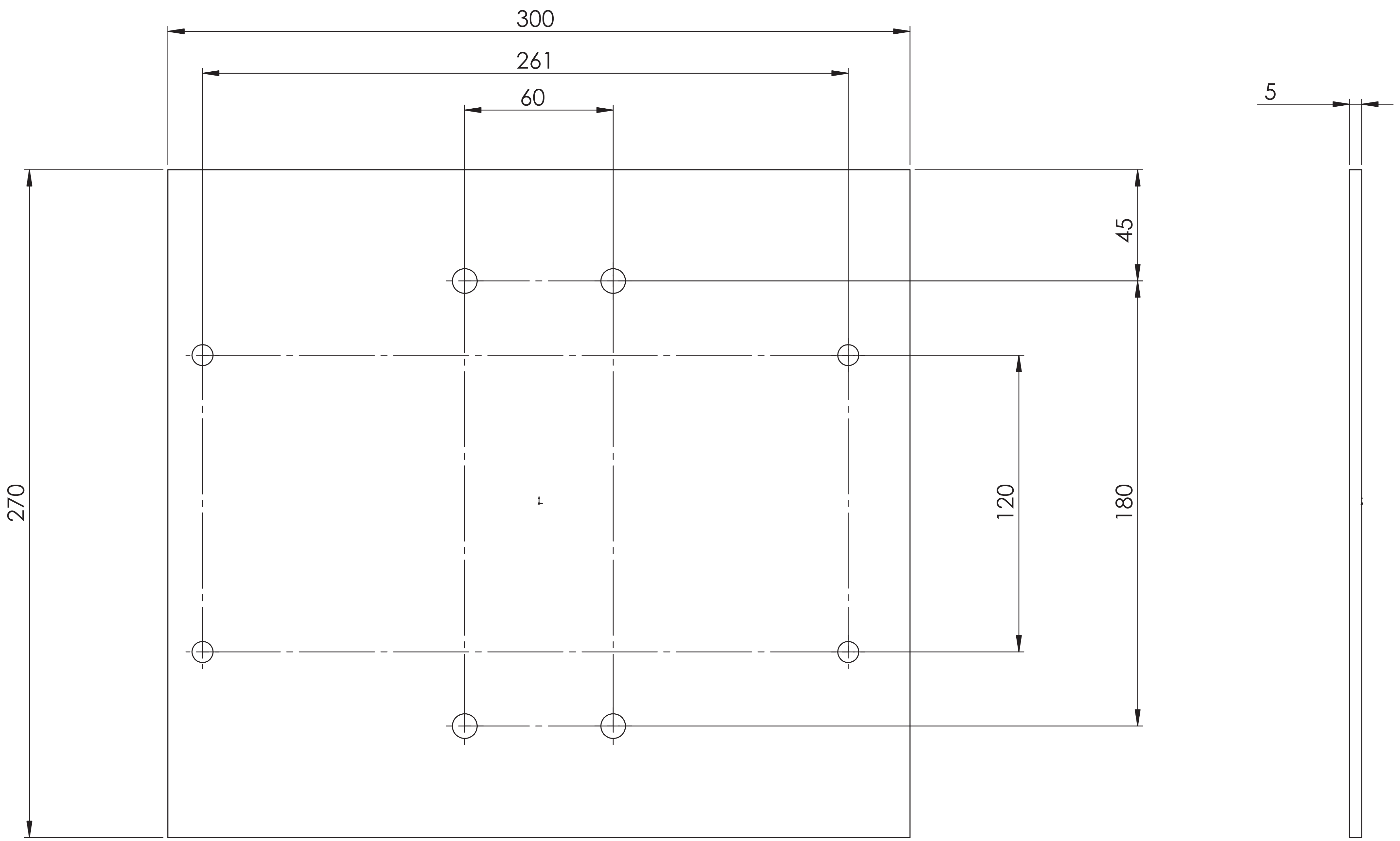


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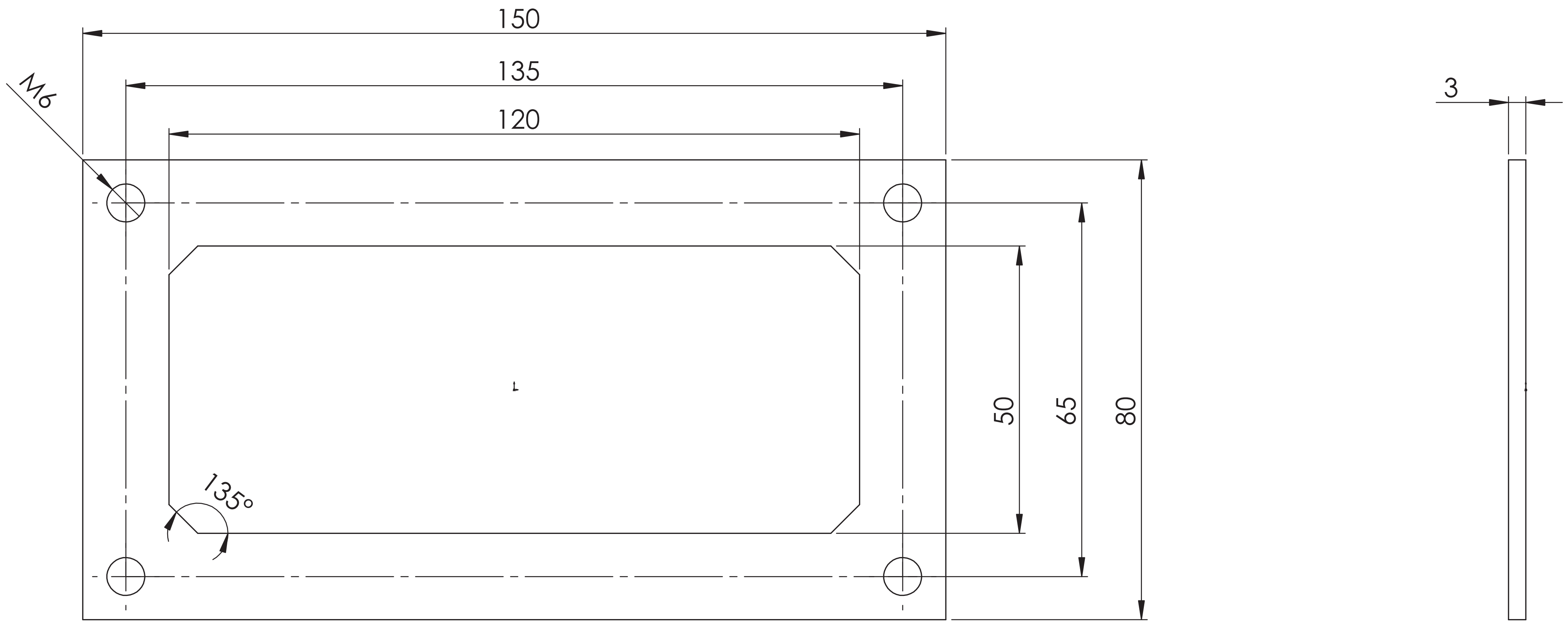


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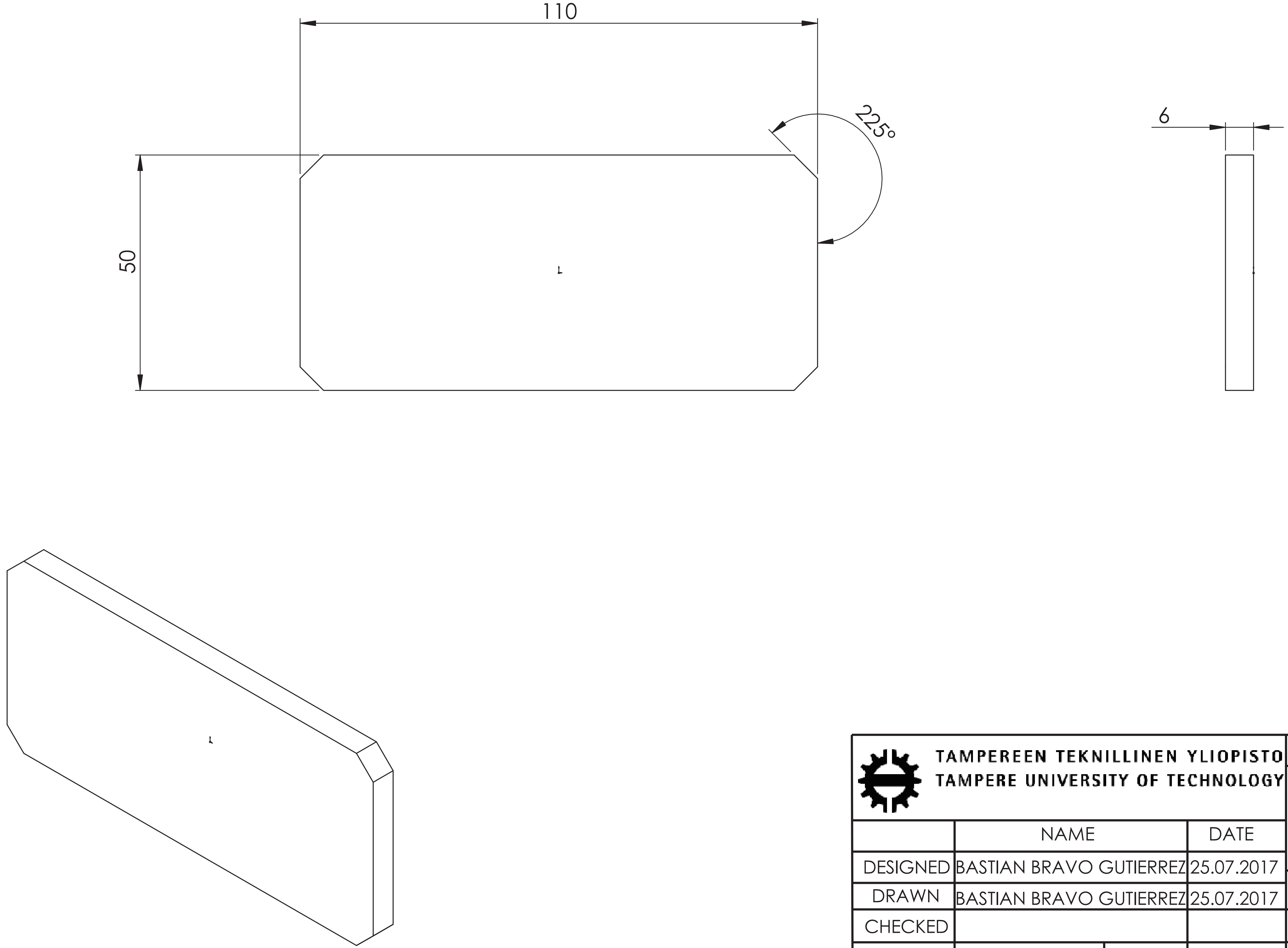


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| | NAME | DATE |
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| DRAWN | BASTIAN BRAVO GUTIERREZ | 25.07.2017 |
| CHECKED | | |

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|---|---|---|---|
| SIZE: A3 | SHEET 1 OF 1 | SCALE | 1:1 |

REVISION

MECHANICAL DESIGN AND ANALYSIS OF A PORTABLE CRANE UNIT

| | |
|---|---|
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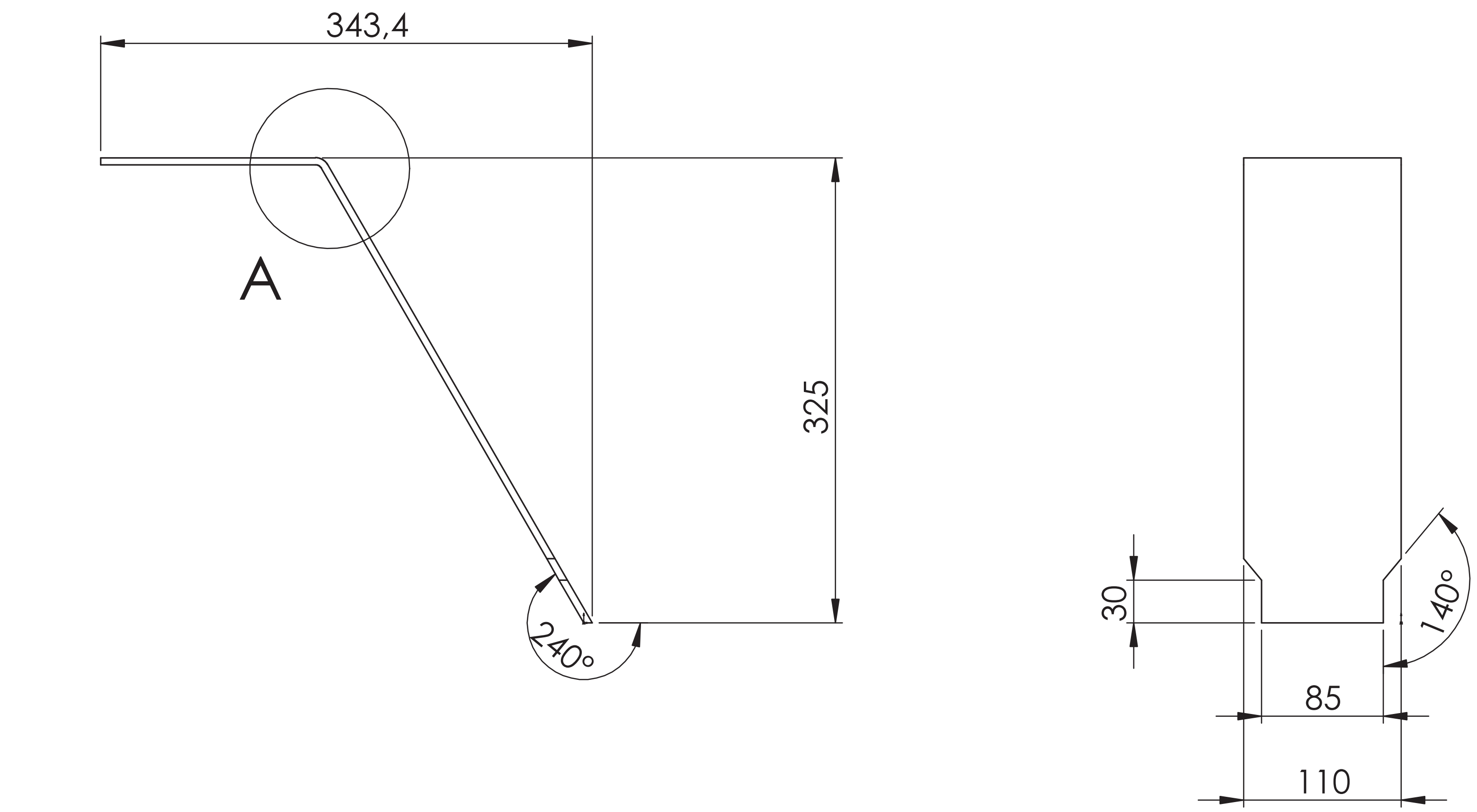
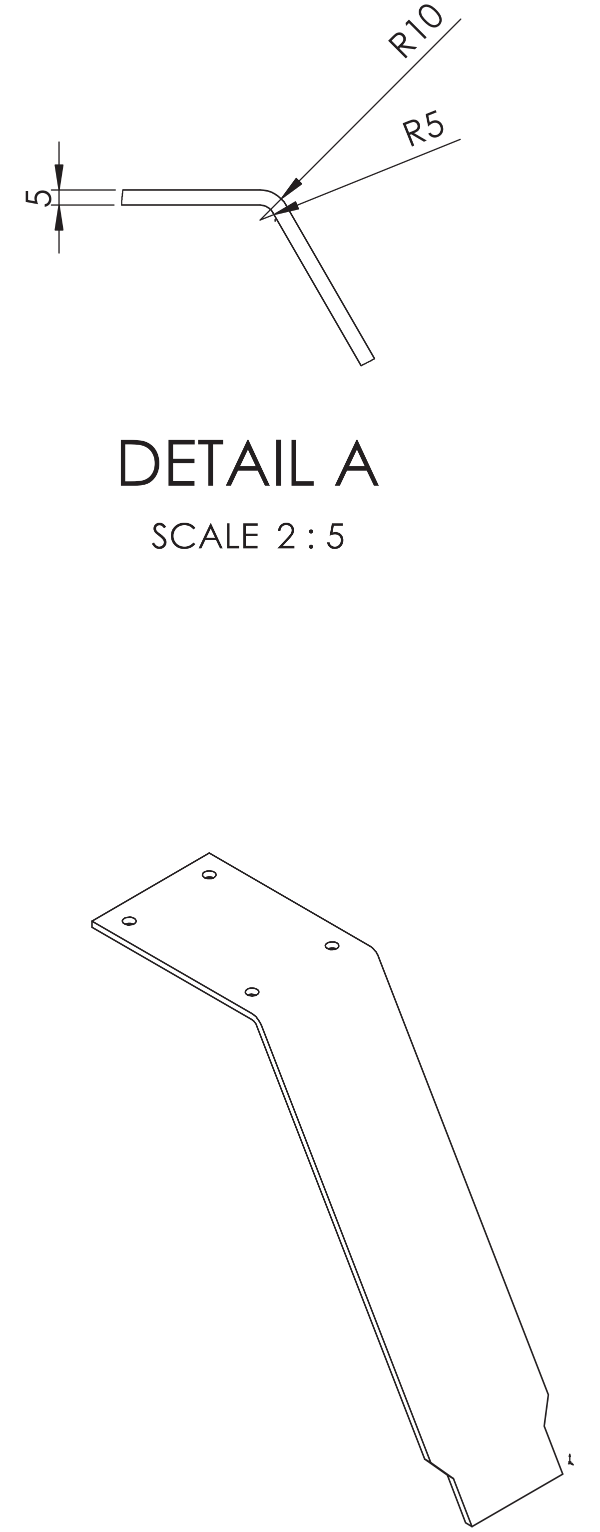
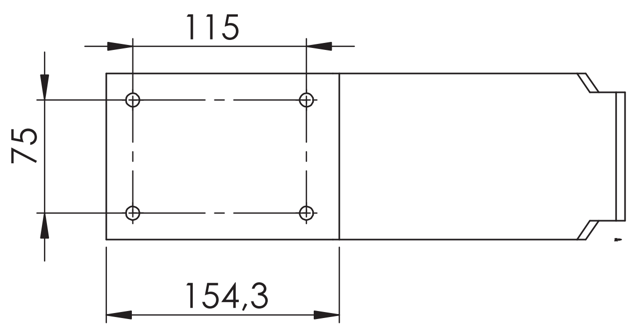
| MATERIAL: | PET (plastic) |

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| TAMPEREEN TEKNILLINEN YLIOPISTO<br>TAMPERE UNIVERSITY OF TECHNOLOGY | | | | REVISION | |
| | NAME | | DATE | MECHANICAL DESIGN AND ANALYSIS OF A PORTABLE CRANE UNIT | |
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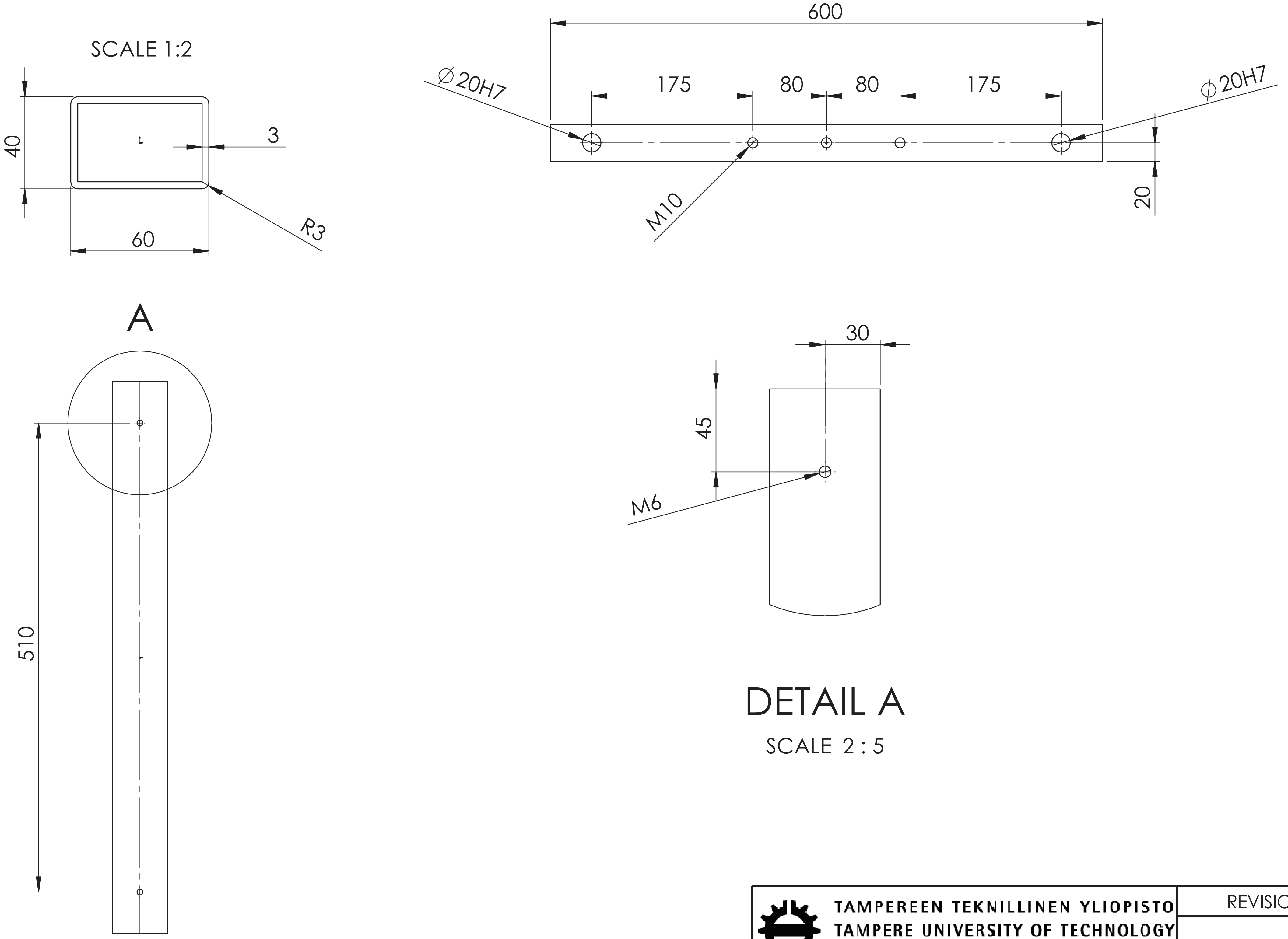


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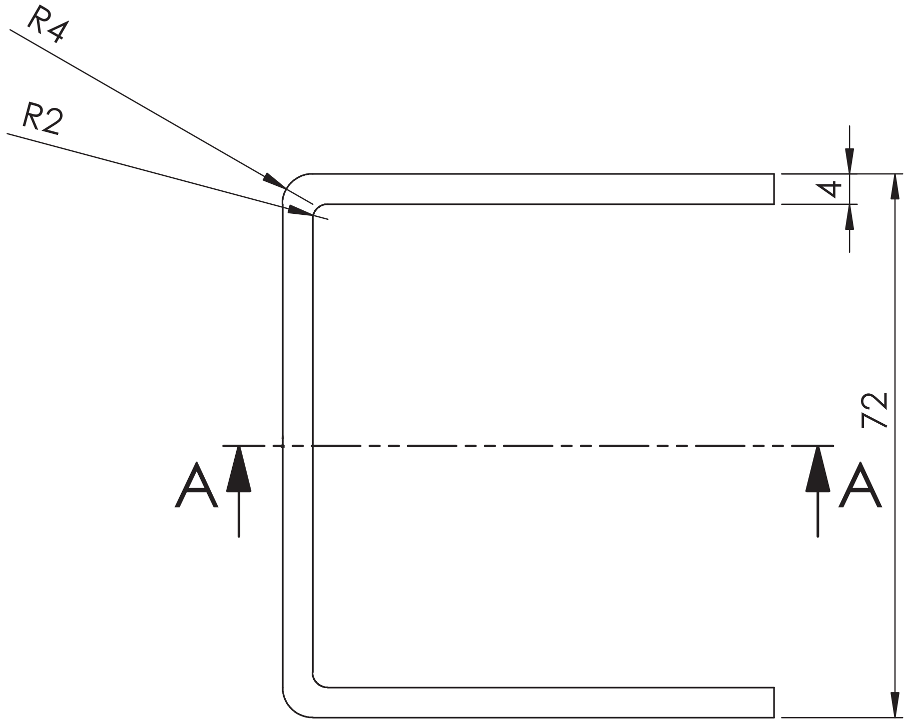
| | | |
|---|---|---|
| | NAME | DATE |
| DESIGNED | BASTIAN BRAVO GUTIERREZ | 25.07.2017 |
| DRAWN | BASTIAN BRAVO GUTIERREZ | 25.07.2017 |
| CHECKED | | |

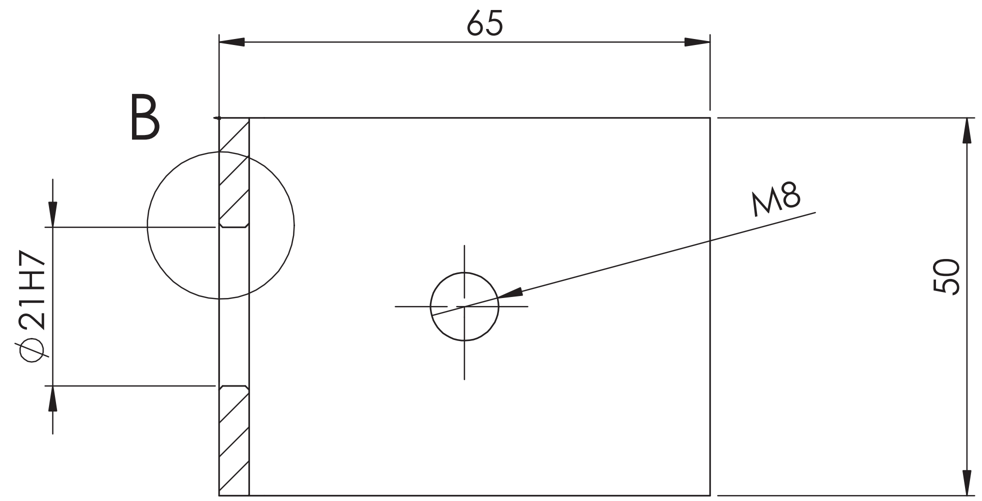
| | | | |
|---|---|---|---|
| SIZE: A3 | SHEET 1 OF 1 | SCALE | 1:5 |

REVISION

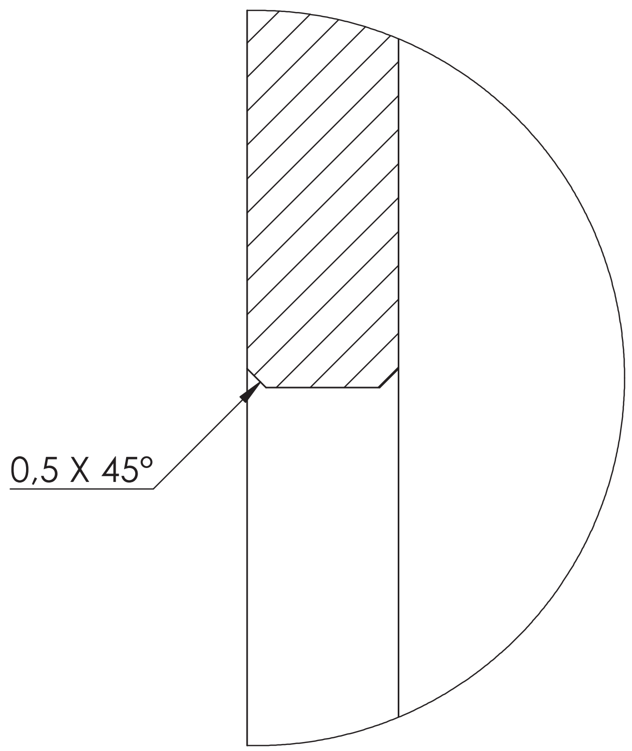
MECHANICAL DESIGN AND ANALYSIS OF A PORTABLE CRANE UNIT

| | |
|---|---|
| DWG NO. | 11 |
| REF: | 1_19 boom lateral legs |
| MATERIAL: | Aluminium 6063-T4 |



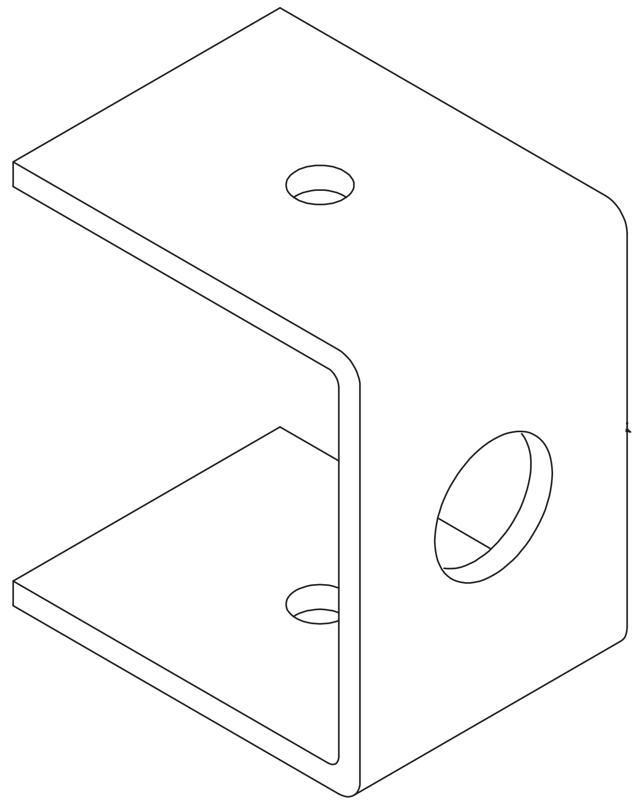




SECTION A-A




DETAIL B

SCALE 5 : 1



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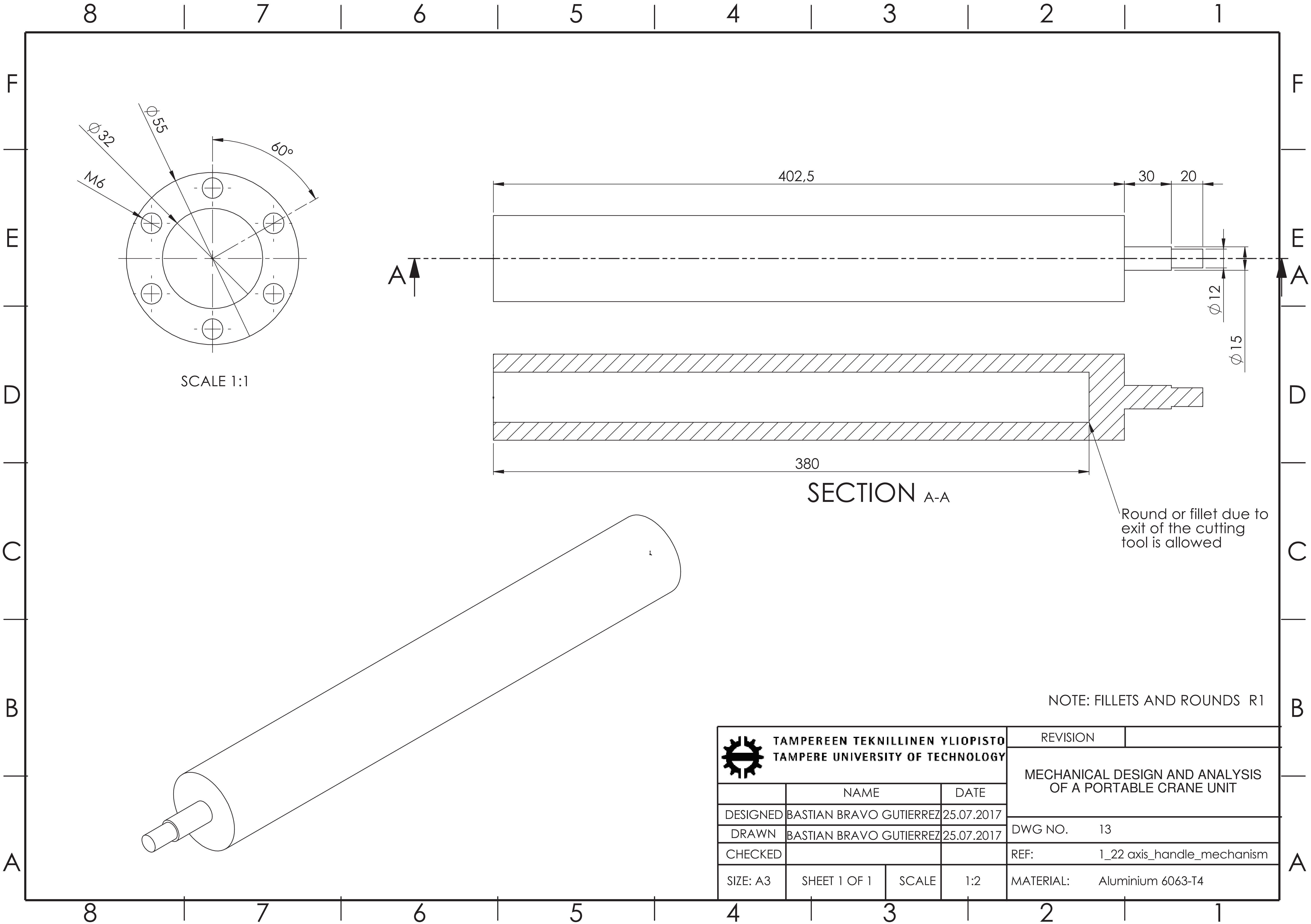
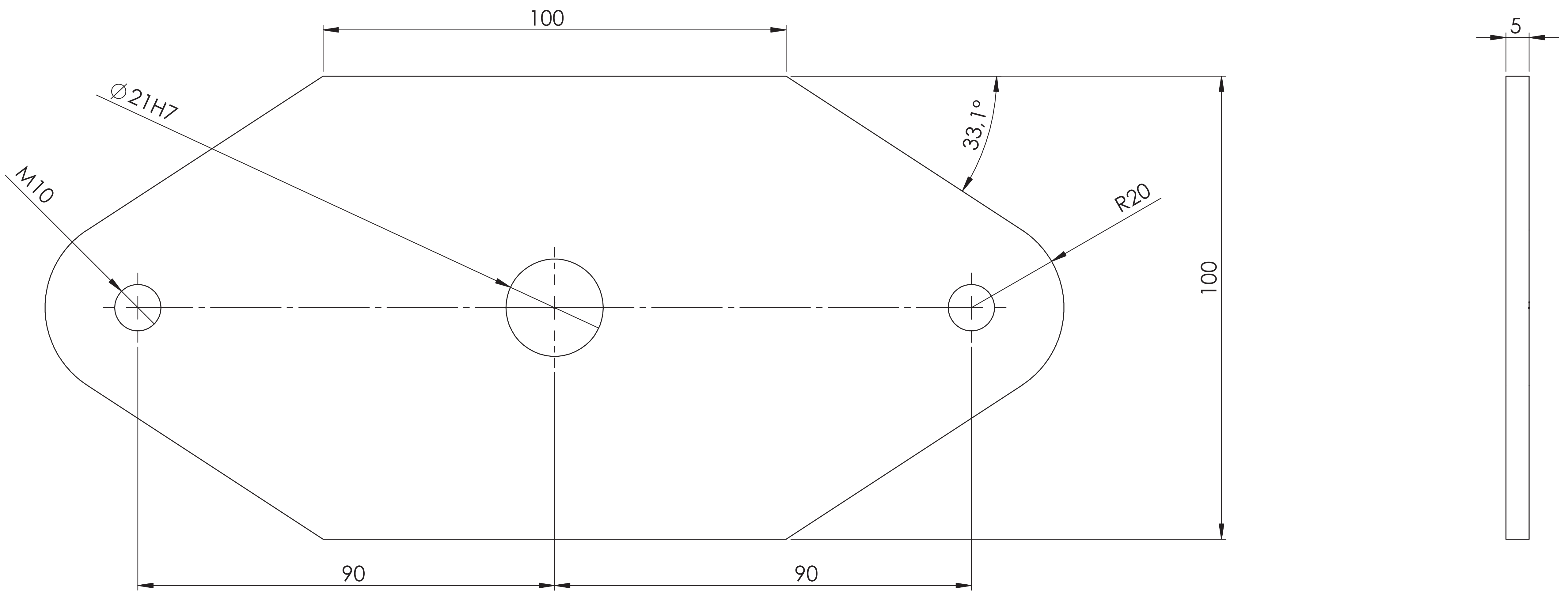
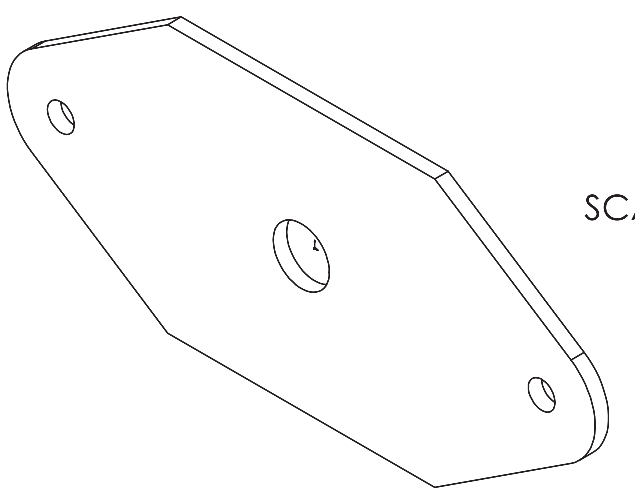
|---|---|---|---|---|
| TAMPEREEN TEKNILLINEN YLIOPISTO<br>TAMPERE UNIVERSITY OF TECHNOLOGY | | | | REVISION |
| | NAME | | DATE | MECHANICAL DESIGN AND ANALYSIS OF A PORTABLE CRANE UNIT |
| DESIGNED | BASTIAN BRAVO GUTIERREZ | | 25.07.2017 | |
| DRAWN | BASTIAN BRAVO GUTIERREZ | | 25.07.2017 | DWG NO. 12 |
| CHECKED | | | | REF: 1_20 plate screw |
| SIZE: A3 | SHEET 1 OF 1 | SCALE | 1:1 | MATERIAL: Aluminium 6063-T4 |




NOTE: FILLETS AND ROUNDS R1

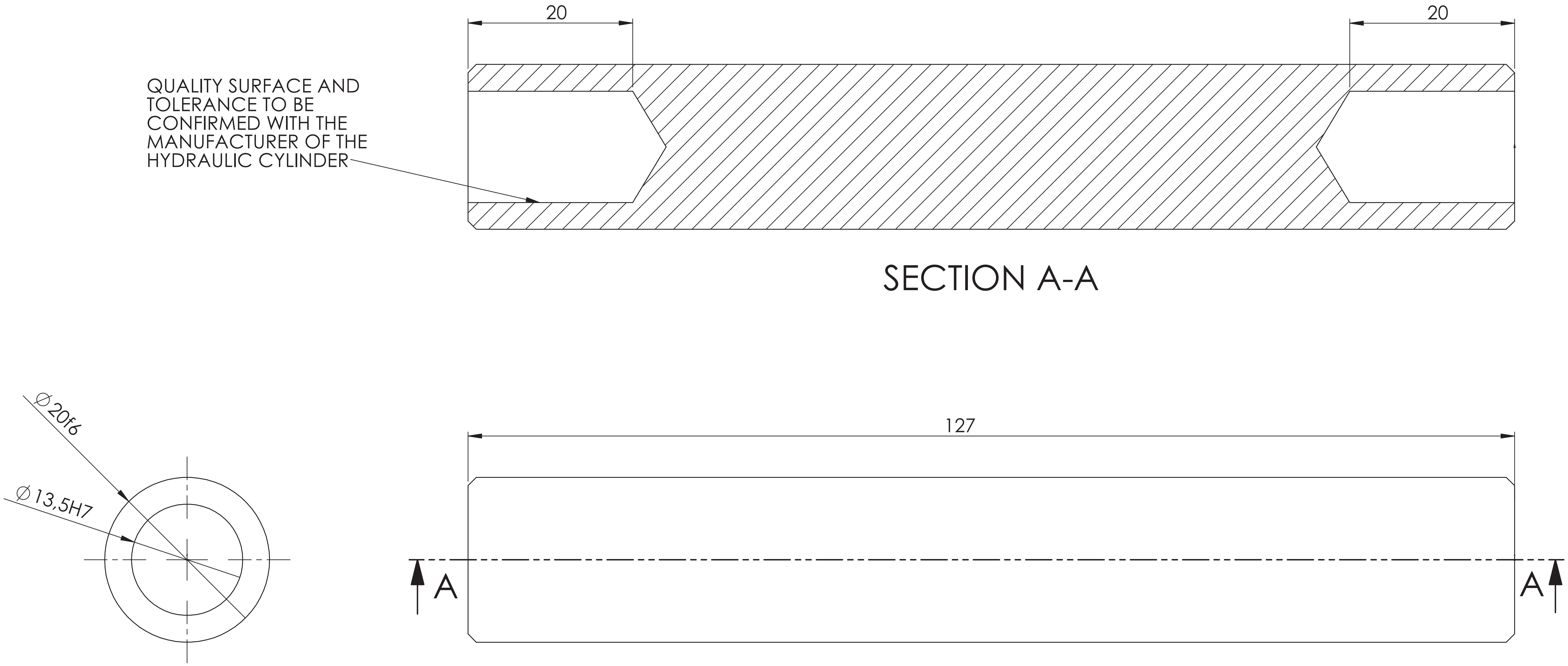
| TAMPEREEN TEKNILLINEN YLIOPISTO<br>TAMPERE UNIVERSITY OF TECHNOLOGY | | | REVISION | |
|---|---|---|---|---|
| | NAME | DATE | MECHANICAL DESIGN AND ANALYSIS OF A PORTABLE CRANE UNIT | |
| DESIGNED | BASTIAN BRAVO GUTIERREZ | 25.07.2017 | | |
| DRAWN | BASTIAN BRAVO GUTIERREZ | 25.07.2017 | DWG NO. | 13 |
| CHECKED | | | REF: | 1_22 axis_handle_mechanism |
| SIZE: A3 | SHEET 1 OF 1 | SCALE 1:2 | MATERIAL: | Aluminium 6063-T4 |



SCALE 1:2

| | | | | |
|---|---|---|---|---|
| TAMPEREEN TEKNILLINEN YLIOPISTO<br>TAMPERE UNIVERSITY OF TECHNOLOGY | | | | REVISION |
| | NAME | | DATE | MECHANICAL DESIGN AND ANALYSIS OF A PORTABLE CRANE UNIT |
| DESIGNED | BASTIAN BRAVO GUTIERREZ | | 25.07.2017 | |
| DRAWN | BASTIAN BRAVO GUTIERREZ | | 25.07.2017 | DWG NO. 14 |
| CHECKED | | | | REF: 1_23 cover_al_plate |
| SIZE: A3 | SHEET 1 OF 1 | SCALE | 1:1 | MATERIAL: Aluminium 6063-T4 |

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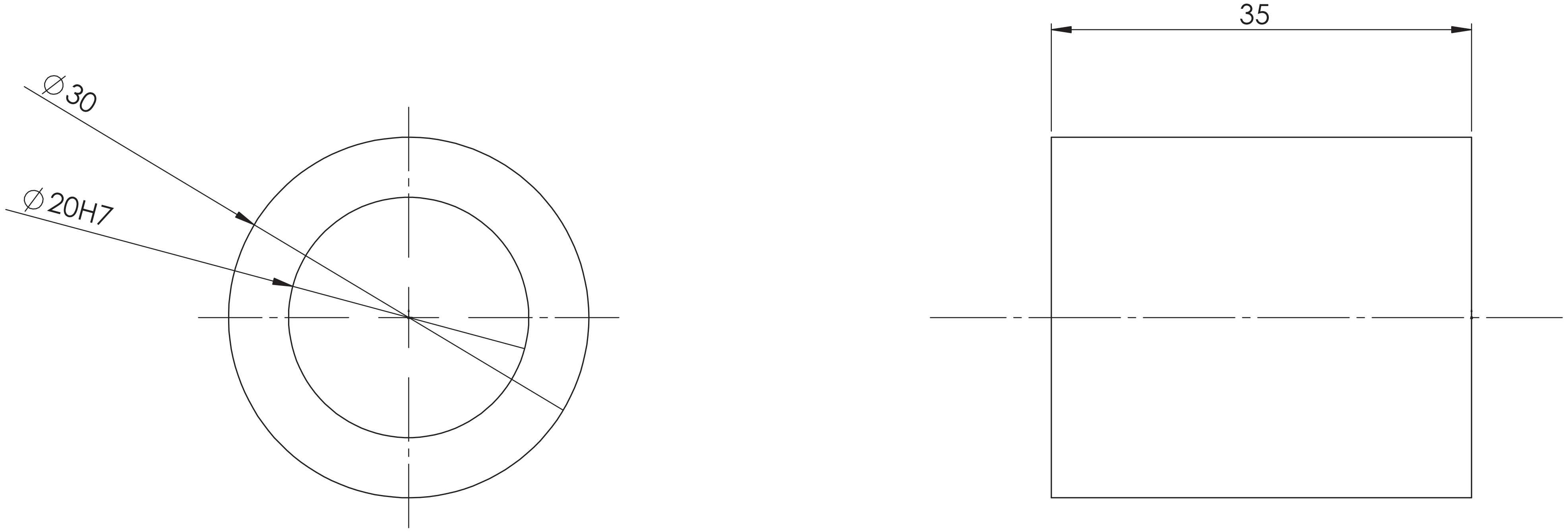
|---|---|---|---|---|
| TAMPEREEN TEKNILLINEN YLIOPISTO<br>TAMPERE UNIVERSITY OF TECHNOLOGY | | | REVISION | |
| | NAME | DATE | MECHANICAL DESIGN AND ANALYSIS OF A PORTABLE CRANE UNIT | |
| DESIGNED | BASTIAN BRAVO GUTIERREZ | 25.07.2017 | | |
| DRAWN | BASTIAN BRAVO GUTIERREZ | 25.07.2017 | DWG NO. | 15 |
| CHECKED | | | REF: | 1_24 lower_union_hydcyl |
| SIZE: A3 | SHEET 1 OF 1 | SCALE 2:1 | MATERIAL: | Aluminium 6063-T4 |

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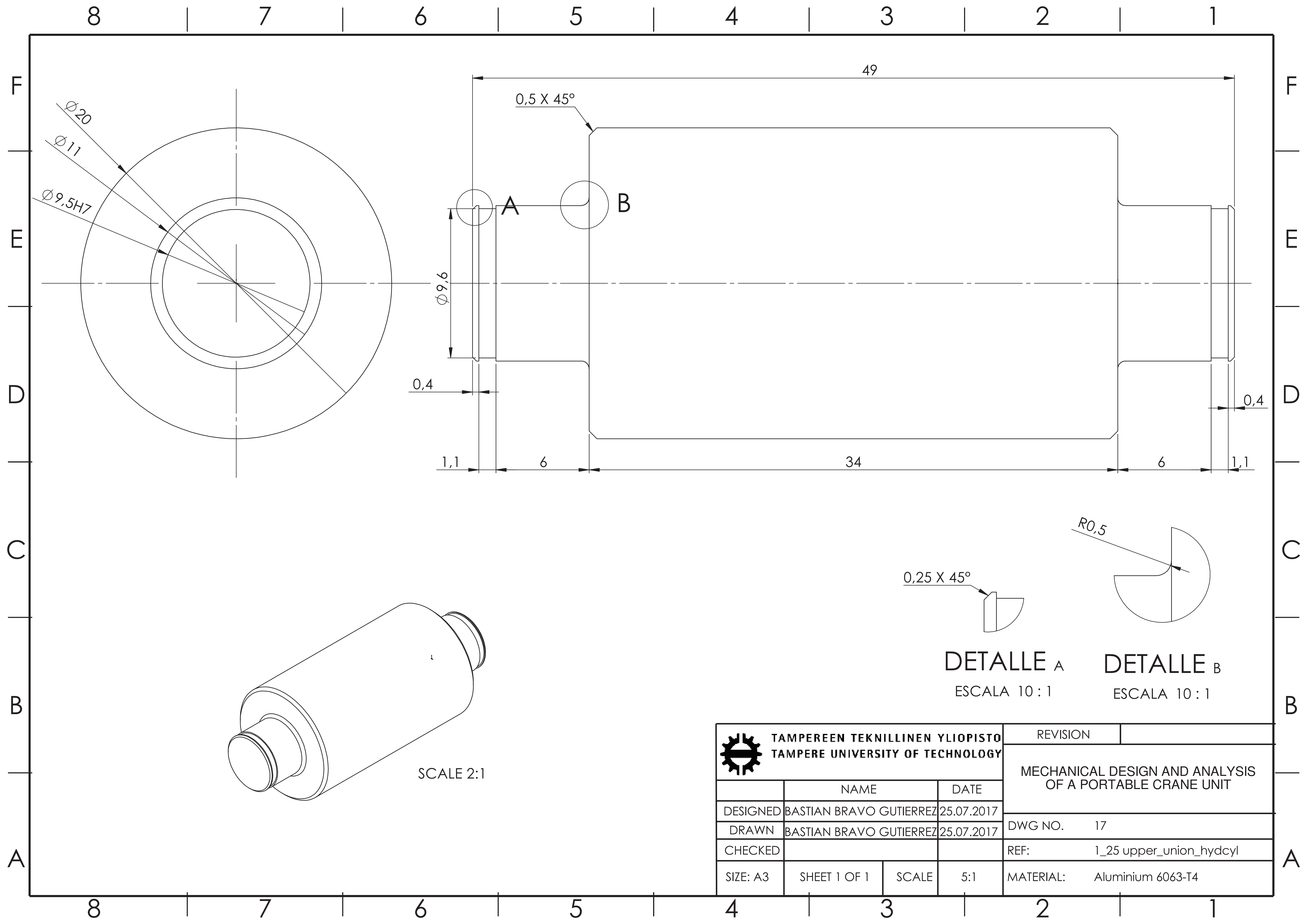
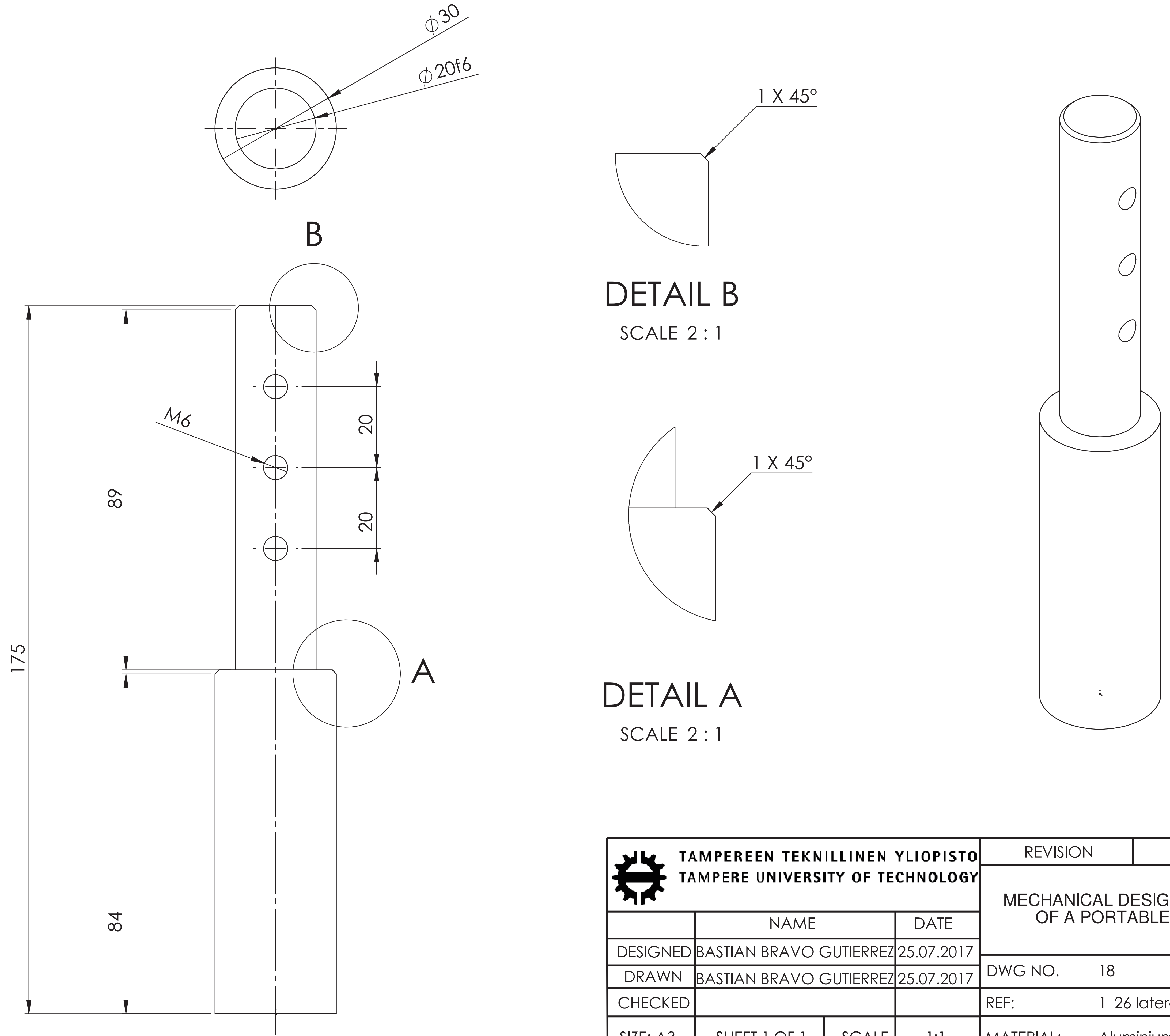
|---|---|---|---|
| TAMPEREEN TEKNILLINEN YLIOPISTO<br>TAMPERE UNIVERSITY OF TECHNOLOGY | | | REVISION |
| | NAME | DATE | MECHANICAL DESIGN AND ANALYSIS OF A PORTABLE CRANE UNIT |
| DESIGNED | BASTIAN BRAVO GUTIERREZ | 25.07.2017 | |
| DRAWN | BASTIAN BRAVO GUTIERREZ | 25.07.2017 | DWG NO. 16 |
| CHECKED | | | REF: 1_25 mech axial retainer |
| SIZE: A3 | SHEET 1 OF 1 | SCALE 2:1 | MATERIAL: Aluminium 6063-T4 |

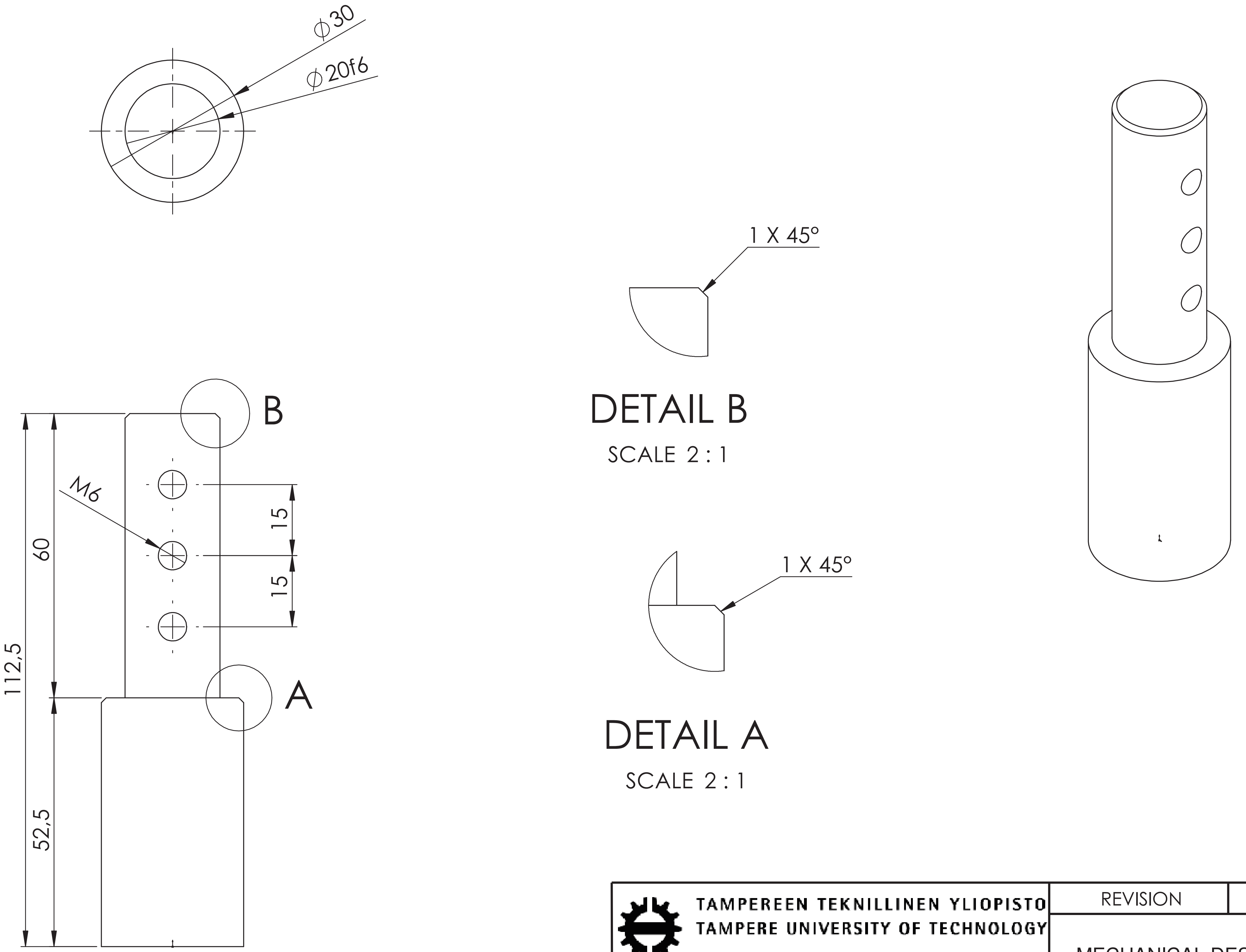



| TAMPEREEN TEKNILLINEN YLIOPISTO<br>TAMPERE UNIVERSITY OF TECHNOLOGY | | | | REVISION | |
|---|---|---|---|---|---|
| | NAME | | DATE | MECHANICAL DESIGN AND ANALYSIS OF A PORTABLE CRANE UNIT | |
| DESIGNED | BASTIAN BRAVO GUTIERREZ | | 25.07.2017 | | |
| DRAWN | BASTIAN BRAVO GUTIERREZ | | 25.07.2017 | DWG NO. | 17 |
| CHECKED | | | | REF: | 1_25 upper_union_hydcyl |
| SIZE: A3 | SHEET 1 OF 1 | SCALE | 5:1 | MATERIAL: | Aluminium 6063-T4 |




TAMPEREEN TEKNILLINEN YLIOPISTO
TAMPERE UNIVERSITY OF TECHNOLOGY

| | | | | |
|---|---|---|---|---|
| | NAME | | DATE | REVISION |
| DESIGNED | BASTIAN BRAVO GUTIERREZ | | 25.07.2017 | MECHANICAL DESIGN AND ANALYSIS OF A PORTABLE CRANE UNIT |
| DRAWN | BASTIAN BRAVO GUTIERREZ | | 25.07.2017 | DWG NO. 18 |
| CHECKED | | | | REF: 1_26 lateral legs support |
| SIZE: A3 | SHEET 1 OF 1 | SCALE | 1:1 | MATERIAL: Aluminium 6063-T4 |

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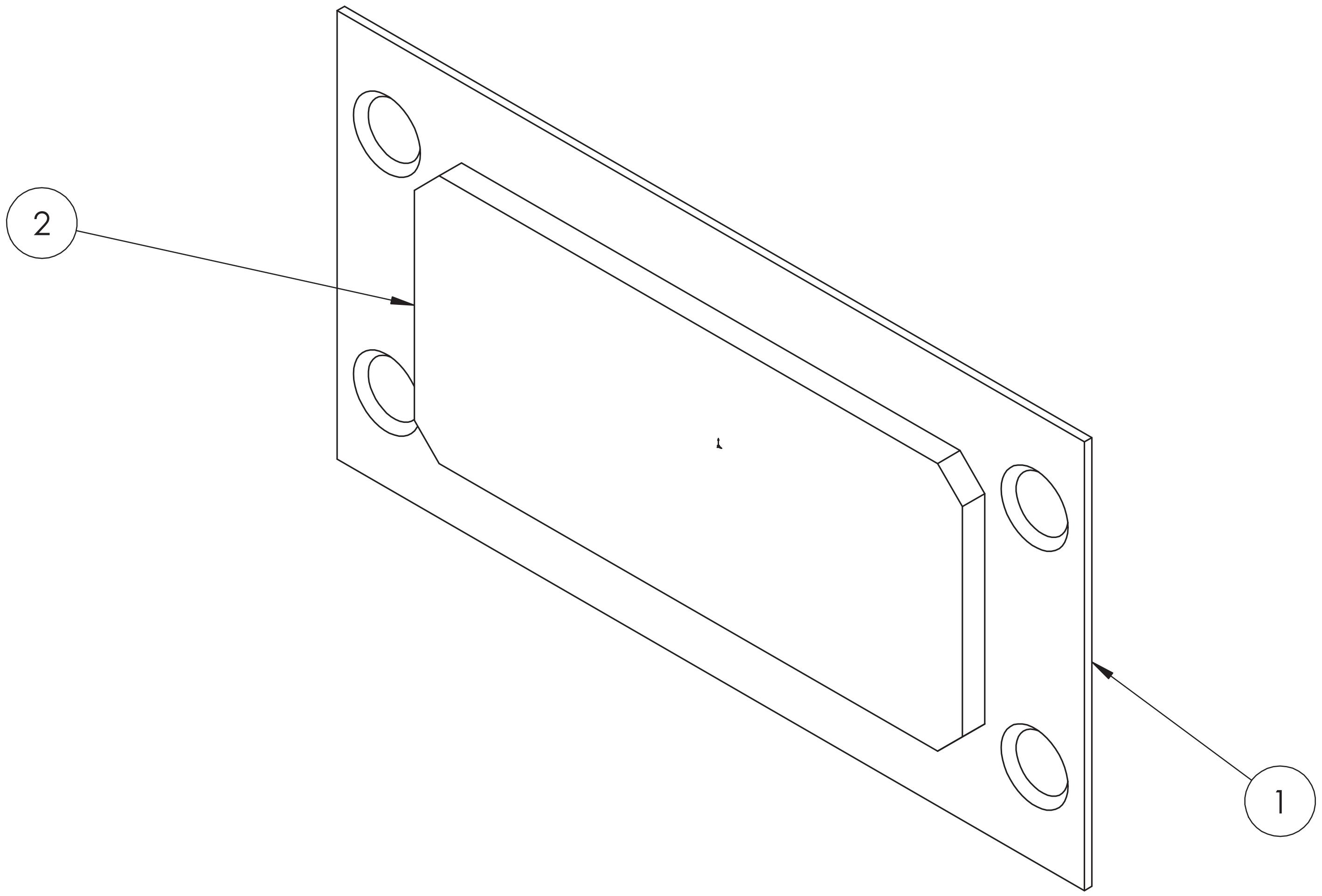
|---|---|---|---|---|---|
| TAMPEREEN TEKNILLINEN YLIOPISTO<br>TAMPERE UNIVERSITY OF TECHNOLOGY | | | | REVISION | |
| | NAME | | DATE | MECHANICAL DESIGN AND ANALYSIS OF A PORTABLE CRANE UNIT | |
| DESIGNED | BASTIAN BRAVO GUTIERREZ | | 25.07.2017 | | |
| DRAWN | BASTIAN BRAVO GUTIERREZ | | 25.07.2017 | DWG NO. | 20 |
| CHECKED | | | | REF: | 1_27 base leg support |
| SIZE: A3 | SHEET 1 OF 1 | SCALE | 1:1 | MATERIAL: | Aluminium 6063-T4 |




| N.º OF THE ELEMENT | REF | COMMENTS | QTY |
|---|---|---|---|
| 1 | 1_13 plate_teleboom | | 1 |
| 2 | 1_14 PET Sheet | | 1 |

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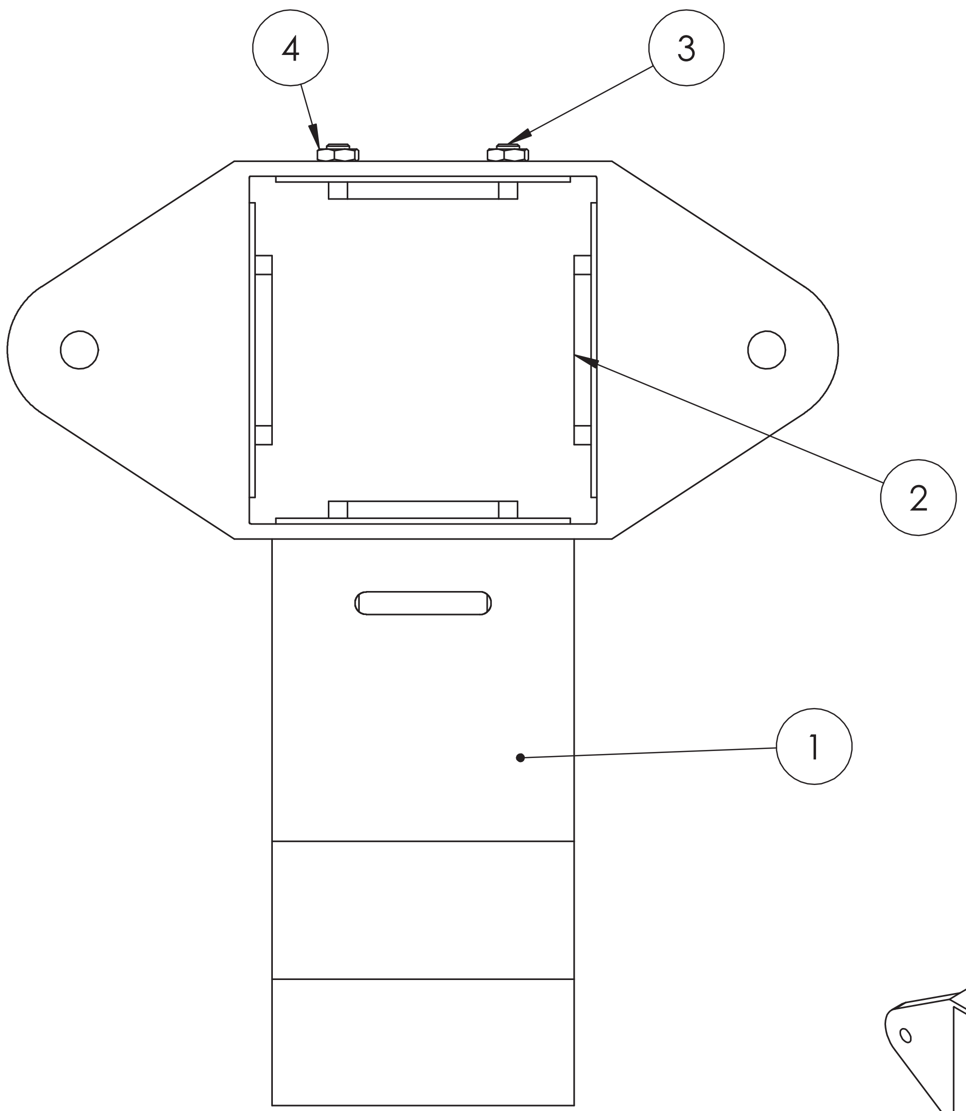
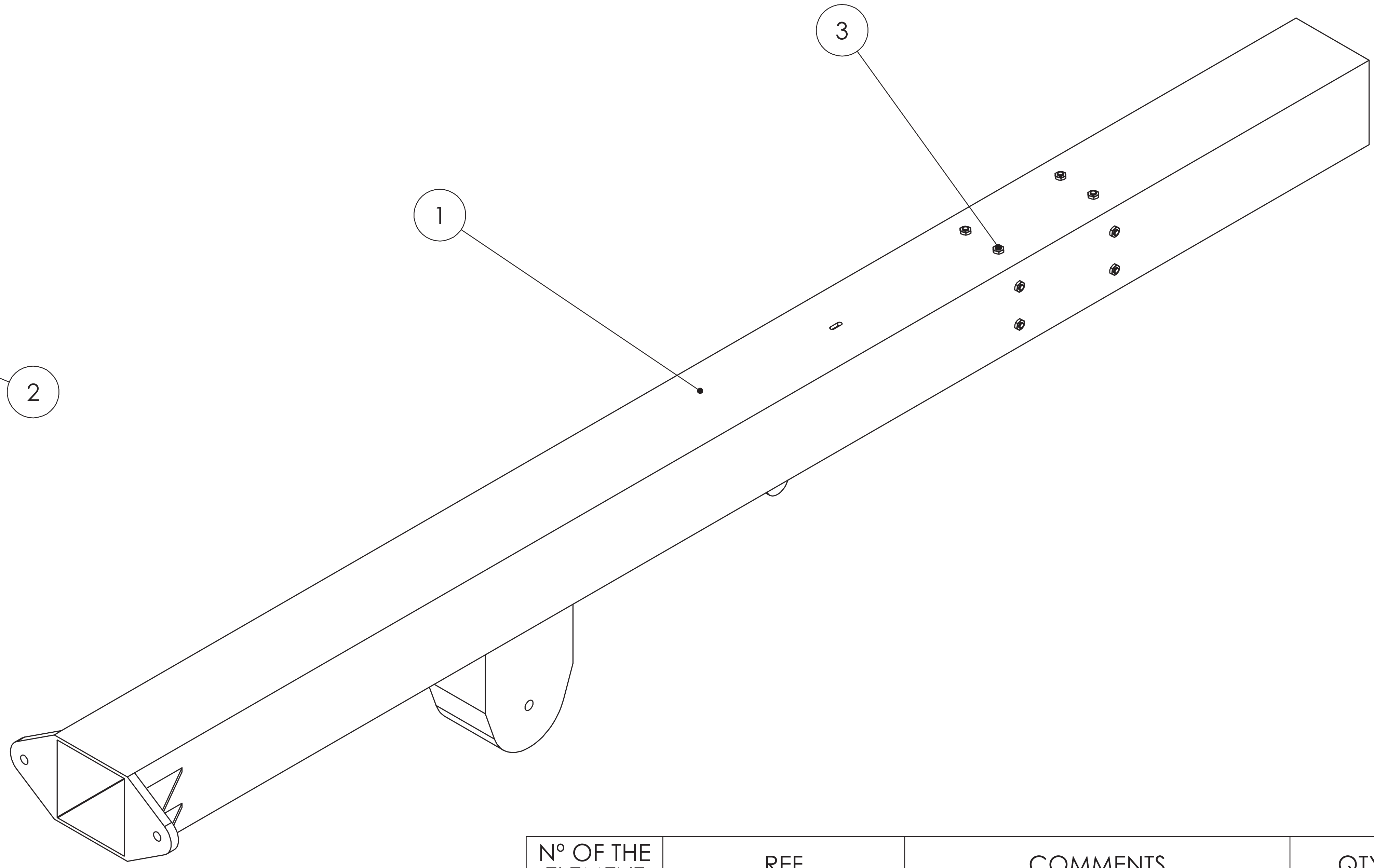
| DESIGNED | BASTIAN BRAVO GUTIERREZ | 25.07.2017 |
| DRAWN | BASTIAN BRAVO GUTIERREZ | 25.07.2017 |
| CHECKED | | |
| SIZE: A3 | SHEET 1 OF 1 | SCALE - |

REVISION

MECHANICAL DESIGN AND ANALYSIS OF A PORTABLE CRANE UNIT

| | |
|---|---|
| DWG NO. | 20 |
| REF: | PET SUBASS |
| MATERIAL: | - |




| N° OF THE ELEMENT | REF | COMMENTS | QTY |
|---|---|---|---|
| 1 | 1_3 boom | | 1 |
| 2 | PET Subassembly | | 4 |
| 3 | ISO 10642 - M6 x 10 - 10 | | 16 |
| 4 | thin nut ISO - 4035 - M6 - C | | 16 |

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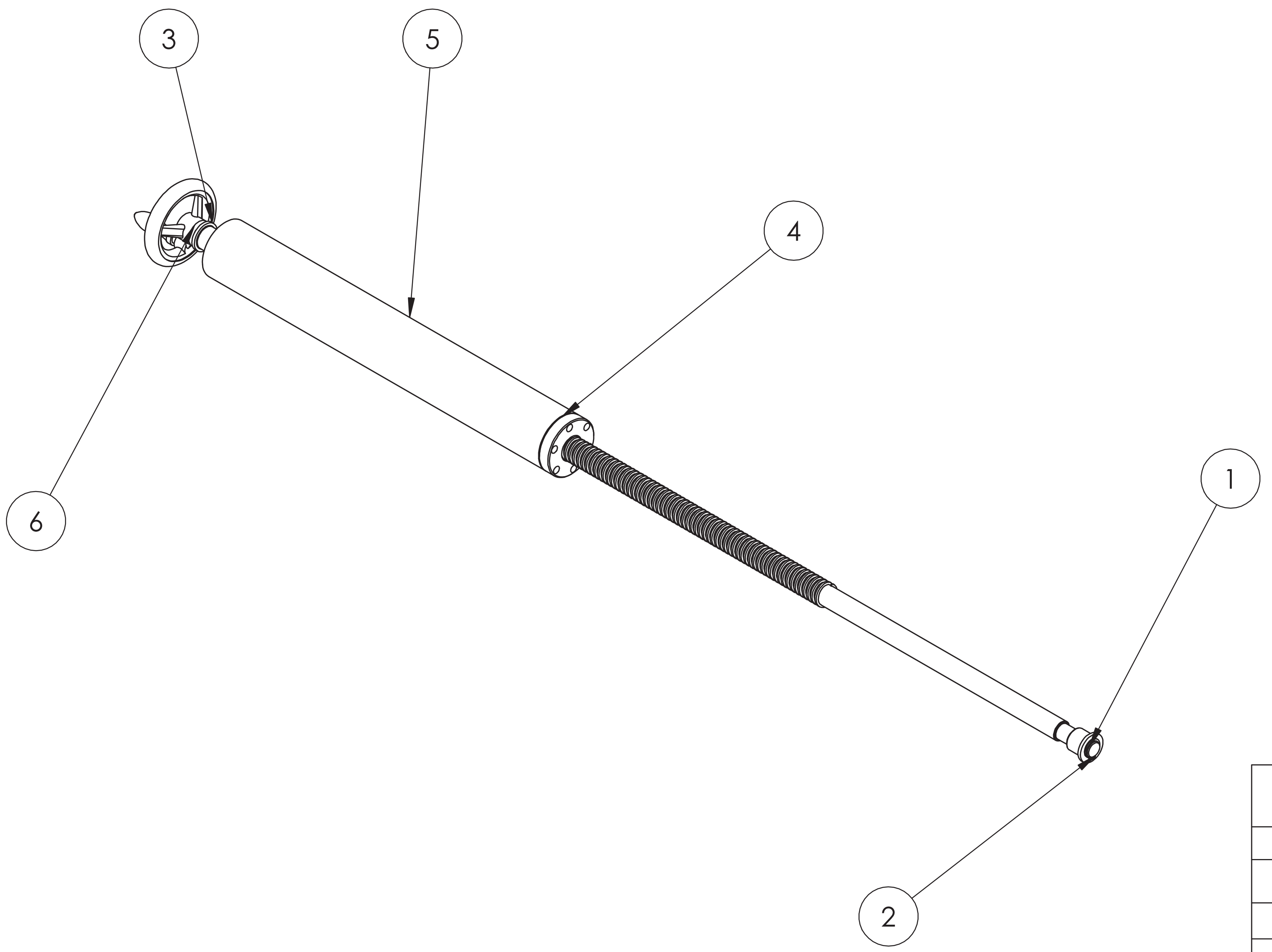
| | NAME | DATE |
|---|---|---|
| DESIGNED | BASTIAN BRAVO GUTIERREZ | 25.07.2017 |
| DRAWN | BASTIAN BRAVO GUTIERREZ | 25.07.2017 |
| CHECKED | | |

SIZE: A3 | SHEET 1 OF 1 | SCALE - 

REVISION

MECHANICAL DESIGN AND ANALYSIS OF A PORTABLE CRANE UNIT

| | |
|---|---|
| DWG NO. | 21 |
| REF: | TELESC_MAIN BOOM_SUBASS |
| MATERIAL: | - |




| N.º OF ELEMENT | REF | COMMENT | QTY |
|---|---|---|---|
| 1 | 1_17 Power screw_tr24x5 din103 | | 1 |
| 2 | Circlip DIN 471 15 x 1 | | 2 |
| 3 | Igus_Bushing_MFM1521-20 | | 2 |
| 4 | trapezoidal flanged nut Tr24x5 | | 1 |
| 5 | 1_22 axis_handle mechanism | | 1 |
| 6 | haldern_eh_handwheel | | 1 |
| 7 | Circlip_DIN 471 - 12 x 1 | | 1 |

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TAMPERE UNIVERSITY OF TECHNOLOGY

| | NAME | DATE |
|---|---|---|
| DESIGNED | BASTIAN BRAVO GUTIERREZ | 25.07.2017 |
| DRAWN | BASTIAN BRAVO GUTIERREZ | 25.07.2017 |
| CHECKED | | |

SIZE: A3 | SHEET 1 OF 1 | SCALE -

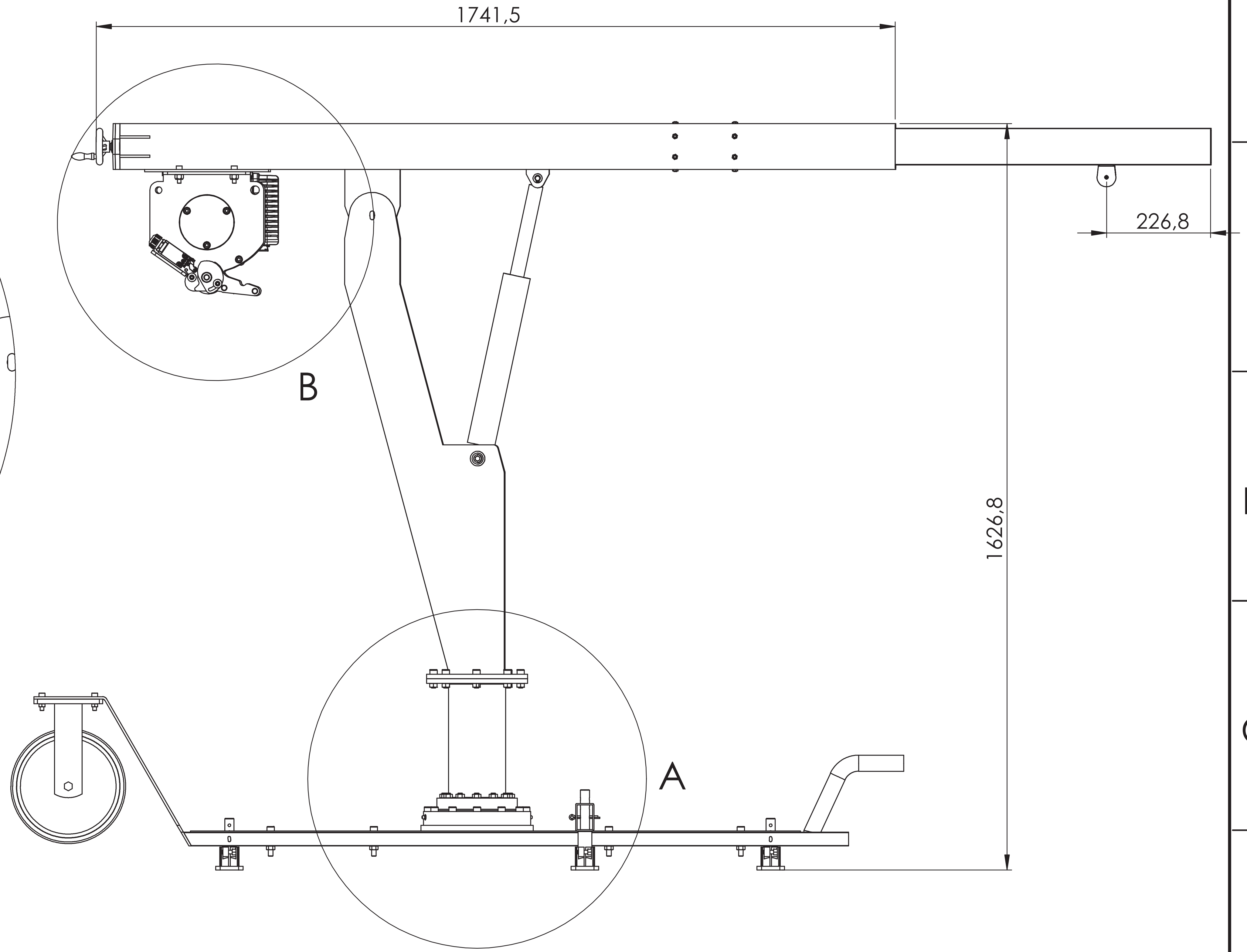
REVISION

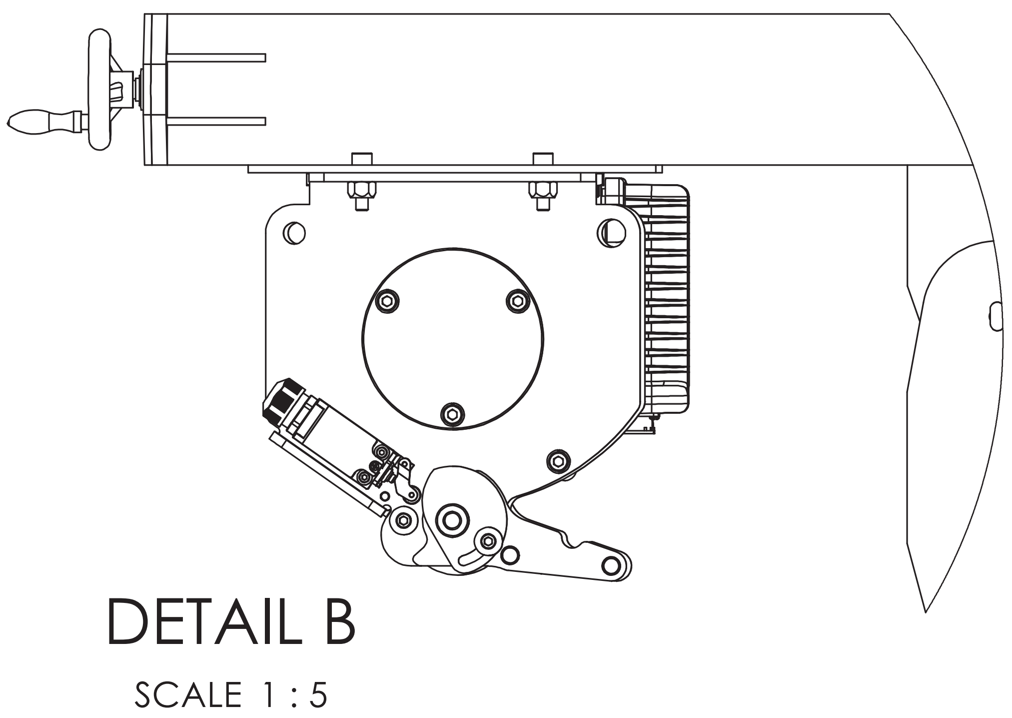
MECHANICAL DESIGN AND ANALYSIS OF A PORTABLE CRANE UNIT

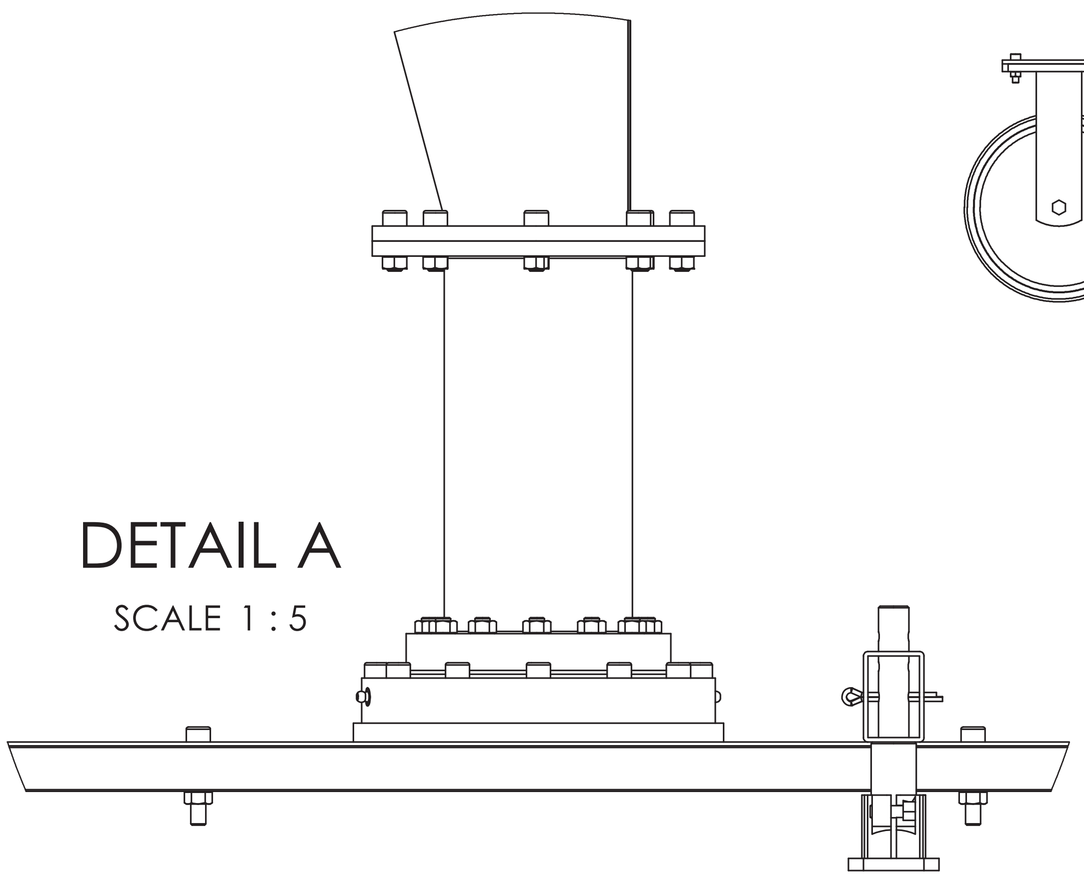
DWG NO. 22

REF: Power screw Subass

MATERIAL: -

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|---|---|---|---|---|
| TAMPEREEN TEKNILLINEN YLIOPISTO<br>TAMPERE UNIVERSITY OF TECHNOLOGY | | | | REVISION |
| | | | | MECHANICAL DESIGN AND ANALYSIS OF A PORTABLE CRANE UNIT |
| | NAME | | DATE | |
| DESIGNED | BASTIAN BRAVO GUTIERREZ | | 25.07.2017 | |
| DRAWN | BASTIAN BRAVO GUTIERREZ | | 25.07.2017 | DWG NO. 23 |
| CHECKED | | | | REF: PORTABLE CRANE UNIT |
| SIZE: A3 | SHEET 1 OF 1 | SCALE | 1:10 | MATERIAL: - |