PLANNING A HYDRAULIC HOIST ON A CAR TRAILER

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ABSTRACT

The objective of this project was to design a hydraulic hoist for a car trailer to lift a mass up to 500 kg by using 12 volts of direct current. The thesis project, commissioned by HAMK University of Applied Sciences, was started by gathering information on various hydraulics hoists which were already in use on car trailers.

By using morphological box analysis, the most suitable design pattern was chosen. The thesis work was carried out with the basic ideas of strength of materials which helped in the process of the beam design as well as selection of machine elements such as nuts and bolts. The entire design process was based on calculations, knowledge of strength of materials and machine elements. Knowledge of hydraulics was applied during the selection of the hydraulic cylinder. The design was assured from Finnish Traffic Regulations.

The total mass of the whole system was calculated and the total construction cost was estimated. During the project, the computer aided design tools were used for designing purposes to display the model in 3D along with engineering drawings.

Keywords   hydraulic hoist, morphological box, beam design, machine elements

Pages   42p. + appendices 15p

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1 INTRODUCTION

In modern days, the means of transportation have changed people's lives significantly. To balance between work life and personal life, people have started to use cars for individual purposes which gave the opportunity of personal control and autonomy to the people along with saving time. But the lack of having enough space in a car has always been a major drawback for people. Hence people's attraction towards towing vehicles started to increase day by day. The trailer towed by a car of own's has not only helped to move things from one place to another voluntarily but also has given more space and freedom to people to move with their belongings for example to go to summer camping or other recreational activities or simply just to move things out.

Generally, these car trailers can carry a mass from 500 kg to several tonnes depending upon their size and construction. For example, the load capacity of FARMI PRO 757T trailer is about 500 kg. Figure 1 illustrates a sample of FARMI PRO 757T trailer used in a car.

Figure 1 Car trailer (Nettivaraosa, 2016)

Today, there are many different varieties of car trailers in the market. Bicycle trailers, construction trailers, travel trailers, semi-trailers, motorcycle trailers, trailer winches, livestock trailers, boat trailers etc. are the most popular types of trailers (Trailer,2016.) These unpowered vehicles are entitled to carry large or heavier loads that do not simply fit into their cars. Hence, the need of a separate system for transporting the load from the ground to the trailer and vice versa was realised. The concept of hydraulic hoists was aroused to lift the mass from the ground and place it onto the trailer. These hydraulic systems or hoists can be

fixed within the trailer itself or they can be designed as mobile ones which are not attached to the trailer. A dump trailer is an example of a hydraulic system that is mounted on the trailer. Furthermore, hydraulic hoists used in car trailer can be operated by using alternating current or by direct current or even manually.

2 OBJECTIVE

The main objective of this thesis was to plot a hydraulic hoist for a car trailer to lift a mass up to 500 kg by using 12 volts of direct current. It was very important to consider also other parameters that converge the design process into a fruitful project. One of the important parameters was the height of the hoist whose construction was not to exceed the height of 2 m from the level of the trailer. The hydraulic hoist was to be constructed under the maximum authorized dimensions allowed for normal road traffic in Finland. Similarly, another concern to be taken into account was the mass of the complete hoist assembly which was aimed to be around 50 kg.

Figure 2 A hydraulic hoist by used in a trailer (TMS, 2016)

In addition to these, the hydraulic hoist was to be designed in such a way that it could be disassembled whenever they are necessary. In other words, the main goal of this thesis was to utilise various topics of mechanical engineering such as the basic concepts of mechanics, strength of materials and hydraulics in order to design a hydraulic hoist

that would works perfectly under the general principles to achieve the above mentioned objectives. Figure 2 is an example of a hydraulic hoist used on a car trailer.

3 MORPHOLOGICAL BOX ANALYSIS

Morphological box analysis is a method for exploring all the possible solutions to a multi-dimensional, non-quantified complex problem (Ritchey, 1998). As a problem-structuring and problem-solving technique, it was designed for multi-dimensional, non-quantifiable problems where causal modelling and simulation do not function well, or at all. Using the technique of cross-consistency assessment, the system allows for reduction by identifying the possible solutions that actually exist, eliminating the illogical solution combinations in a grid box rather than reducing the number of variables involved. (Morphological, 2016.)

This process was chosen to identify the most suitable one out of few possible design patterns. Each design patterns were named as A, B, C and D which were placed on the grid box for exploring the possible solutions. The design pattern A was obtained from Grabcad that was designed by David Dearing as diplayed in Figure 3. The design pattern B was obtained from Grabcad designed by Mateusz as showned in Figure 4. Similarly, as illustrated in Figure 5, the design pattern C was obtained from Grabcad designed by Rolando Venegas. The design pattern D was acquired from Zhejiang Shineda Machinery as demonstrated in Figure 6.

Figure 3 Design pattern A obtained from Grabcad (Dearing, 2016)

Figure 4 Design pattern B obtained from Grabcad (Mateusz, 2016)

Figure 5 Design pattern C obtained from Grabcad (Venegas, 2016)

Figure 6 Design pattern D acquired from Zhejiang Shineda Machinery
(Zhejiang, 2016)

All the possible patterns along with the design parameters such as size, weight, cost, design complexity, system compatibility, functionability and utility were inserted into a grid box to check the possible solutions.

System compatibility includes traffic safety, traffic law, proper alignment of the modified system such as folding when unnecessary or after use. The most easy and compatible one should be chosen which reduces design complexity along with design costs. On the other hand, design complexity denotes the level of difficulty during designing process. Simpler the design, better the result. Functionability refers to the ability to perform the tasks allocated to it. For example, ability to rotate 360 degree and capability to load and unload allocated mass to and from trailer. Reliability denotes the measure of consistency in terms of quality or performance of the design. Furthermore, the design also demands for smaller size and light weight which enhances lower the cost. A morphological box was created in order to find the most suitable solution out of them.

Table 1 shows a single driver input and clustered outputs in which the design pattern A acts as the only driver input and the other design parameters that decides the properties of that design pattern act as outputs.

Similarly tables 2, 3 and 4 denote the design patterns B, C and D as inputs of the configurations respectively with their variable outputs.

Table 1 Single driver input configuration for design pattern A (input = red, output = blue)

Patterns	Size	System Compatibility	Design Complexity	Cost	Functionability	Weight	Reliability
A	small	Easy	simple	low	Relatively best one	Light	Very Reliable
B	medium	Medium	medium	medium	Fairly functional	medium	Fairly reliable
C	large	Difficult	complex	higher	Poorly functional	Heavy	Poor one
D							

Table 2 Single driver input configuration for design pattern B

Patterns	Size	System Compatibility	Design Complexity	Cost	Functionability	Weight	Reliability
A	small	Easy	simple	low	Relatively best one	Light	Very Reliable
B	medium	Medium	medium	medium	Fairly functional	medium	Fairly Reliable
C	large	Difficult	complex	higher	Poorly functional	Heavy	Poor one
D							

Table 3 Single driver input configuration for design pattern C

Patterns	Size	System Compatibility	Design Complexity	Cost	Functionability	Weight	Reliability
A	small	Easy	simple	low	Relatively best one	Light	Very Reliable
B	medium	Medium	medium	medium	Fairly functional	medium	Fairly Reliable
C	large	Difficult	complex	higher	Poorly functional	Heavy	Poor one
D							

Table 4 Single driver input configuration for design pattern D

Patterns	Size	System Compatibility	Design Complexity	Cost	Functionability	Weight	Reliability
A	small	Easy	simple	low	Relatively best one	Light	Very Reliable
B	medium	Medium	medium	medium	Fairly functional	medium	Fairly Reliable
C	large	Difficult	complex	higher	Poorly functional	Heavy	Poor one
D							

Hence, pattern D was chosen as the required designed pattern since the output parameters were the best among all.

4   CALCULATIONS OF ANGLE OF CYLINDER

As the very first step, a geometrical layout of the hoist system had to be carried out which was more or less closer to the design pattern chosen from the morphological analysis. The dimensions of the layout were based upon logical the analytical interpretation of engineering laws as well as conventional methods with logical explanations.

Addressing the requirement of the design to minimize the stature of the hoist system under 2000 mm, the height of the beam was assigned to be 500 mm. Similarly, the total length of the boom was assumed to be 1000 mm. The beam and the boom now, altogether would shave a stature under 2000 mm. The minimization of the stature of the hoist system also led to minimize the design steps, complexity and the cost as well. For example, if the stature of the hoist system is much bigger than required, the design will need to take more complex steps such as foldability to keep it folded which would increase design and manufacturing cost as well as time.

A boom bracket was placed at 2/3rd of the total boom length from the edge of the boom, where the force is applied, in order to optimize mechanical advantage (Since the system is a perfect example of third class lever). Hence, the boom bracket was located at 330 mm from the beam. A beam bracket was placed at 100 mm from the base of the beam. The beam bracket helps to connect the hydraulic cylinder to the beam and the boom bracket help to connect the hydraulic cylinder to the boom as illustrated in Figure 10. The size of both brackets were assumed to be 50 mm.

After this assessment, the process of calculation of the angle of the cylinder was commenced. The significance of the calculation of the angle was to use it in the mechanical calculations and ultimately in the beam design process. There are mainly three conditions at which the angle of cylinder alters with the angle of boom. The boom attains different heights at different angles of the boom which consequently result in different angles of the cylinder. The three conditions are as follows:
When the boom attains the maximum height
When the boom is at a horizontal plane
When the boom attains a minimum height

4.1   Angle of cylinder(β) when the boom attains the maximum height

When the boom attains the maximum height, the angle made by the boom along with the horizontal plane was assumed to be 45°. This changes the angle of cylinder which is denoted by β. Figure 7 displays the angle of the cylinder when the boom attains the maximum height.
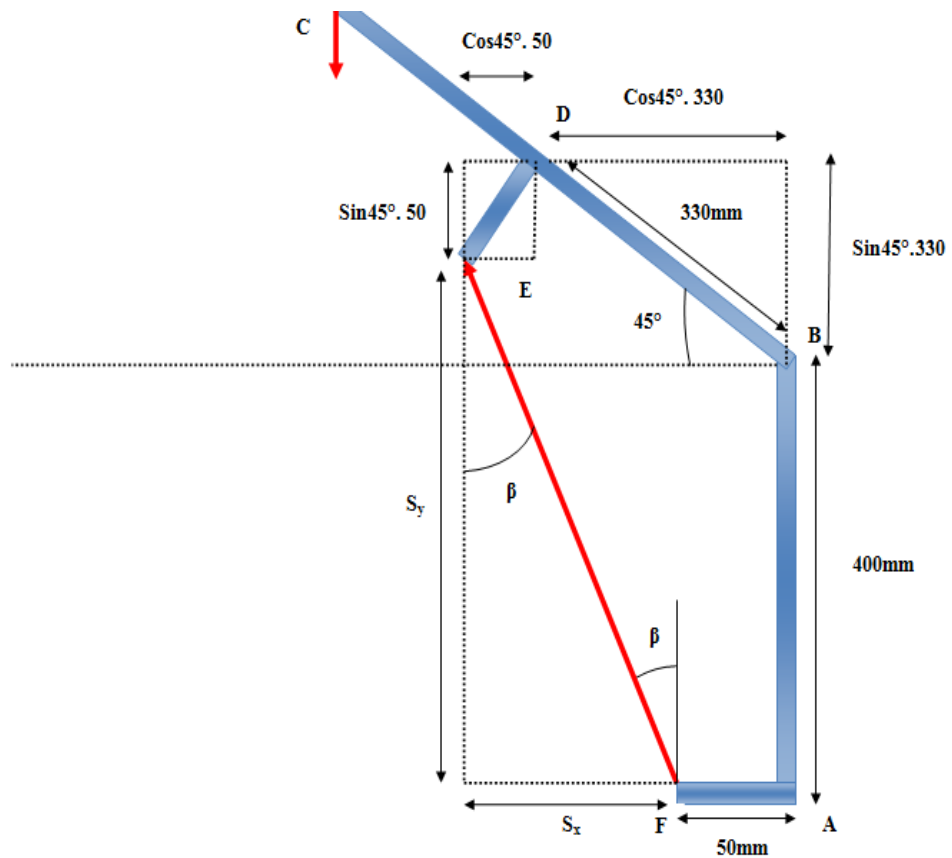
Figure 7 When the boom attains maximum height

The angle of cylinder (β) could be calculated by using trigonometry

S_x = Cos 45°. 330 mm + Cos 45° . 50 mm - 50 mm

S_x = 218.7 mm

S_y = 400 mm + Sin 45°. 330 mm - Sin 45° . 50 mm

S_y = 597.9 mm

Now,

$S = \sqrt{(S_x^2 + S_y^2)}$

S = 636.6 mm

$β = \tan^{-1}(S_x/S_y)$

β = 20.1°

Hence, the angle of the cylinder (β) was calculated to be 20.1° when the boom attained the maximum height.

4.2   Angle of cylinder(β) when the boom is at a horizontal plane

When the boom remains at the horizontal plane, the angle made by the boom along with the horizontal plane is 0°. This changes the angle of cylinder denoted by β. Figure 8 illustrates the angle of cylinder when the boom is at a horizonal plane.

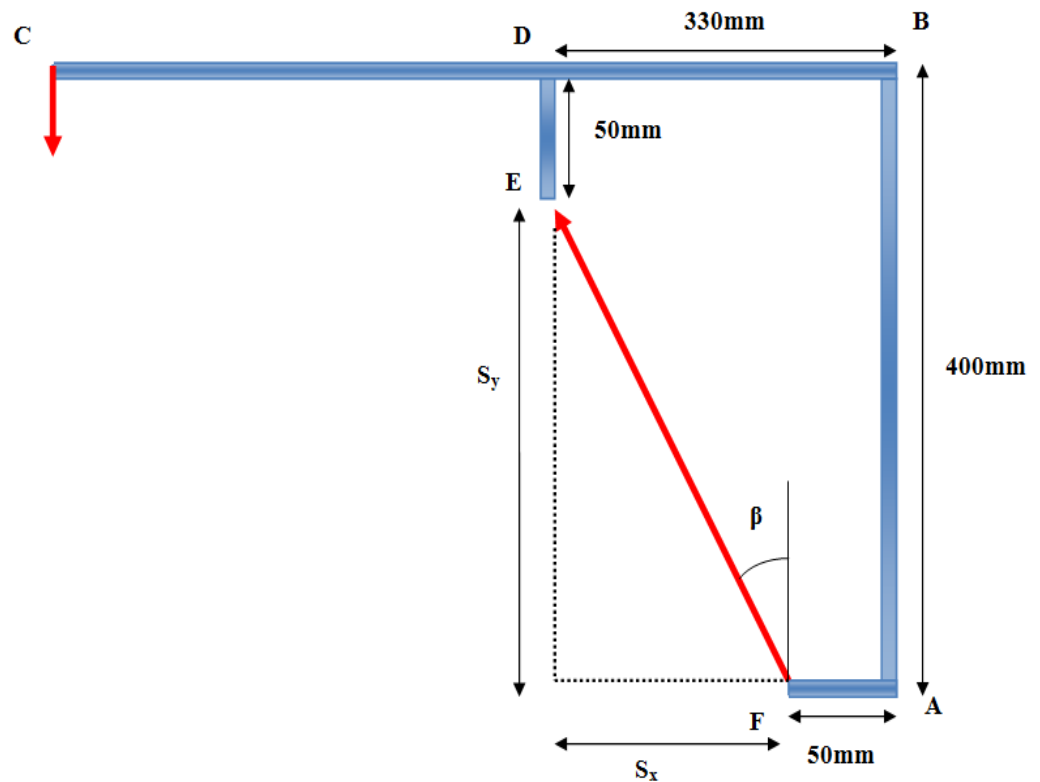Now, the angle of cylinder (β) could be calculated by using trigonometry.

Figure 8 When the boom is at horizontal plane

According to the figure,
Sx = 330 mm - 50 mm = 280 mm
Sy = 400 mm - 50 mm = 350 mm

Now,
$S = \sqrt{(S_x^2 + S_y^2)}$
S = 448.2 mm

$\beta = \tan^{-1}(S_x/S_y)$
β = 36°

Hence, the angle of the cylinder (β) was calculated to be 36° when the boom was at the horizontal plane.

4.3   Angle of cylinder(β) when the boom attains a minimum height

 When the boom attains the minimum height, the angle made by the boom along with the horizontal plane was assumed to be 20°. This brings different value to the angle of cylinder which is denoted by β. Figure 9 shows the angle of cylinder when the boom attains a minimum height. The angle of cylinder (β) could be calculated by using trigonometry.
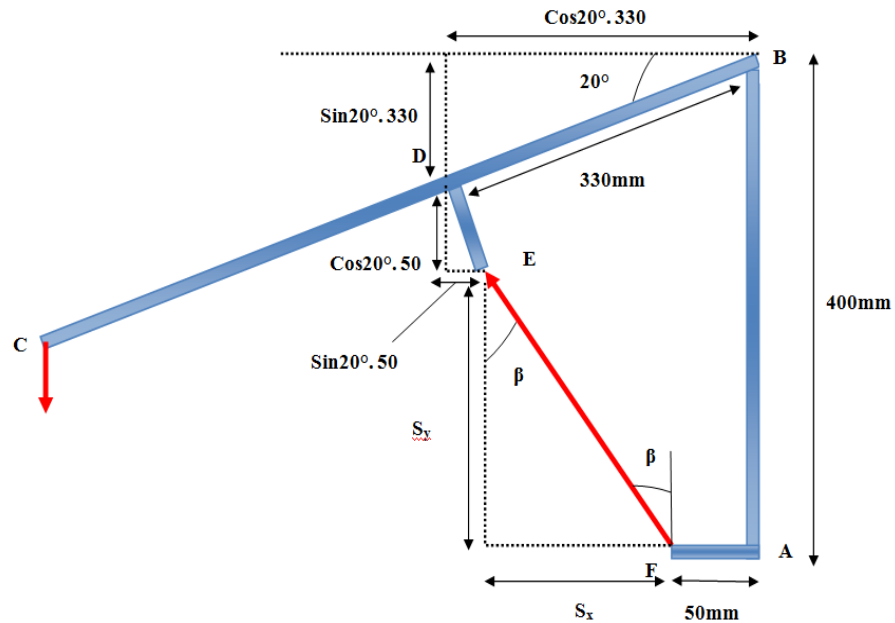
Figure 9 when the boom attains minimum height

Here,

S_x = Cos 20°. 330 mm - Sin 20° . 50 mm
S_x = 266 mm

S_y = 400 mm - Sin 20°. 330 mm - Sin 20° . 50 mm
S_y = 293.5 mm

Now,
$S = \sqrt{(S_x^2 + S_y^2)}$
S = 396.1 mm

$\beta = \tan^{-1}(S_x/S_y)$
β = 42 °

Hence, the angle of the cylinder (β) was calculated to be 42° when the boom attained the minimum height.

5   MECHANICS

When the boom carrying 5000N of load is at the horizontal plane, the maximum bending moment occurs. The reasons are quite simple; the force itself is maximum at that moment and the perpendicular distance becomes maximum when the boom is at the horizontal plane. The calculation of maximum bending moment in a beam helps in the beam design process to select the suitable size of the beam. In order to find out

the point where the maximum bending occurs, the shear force diagram and the bending moment diagram were needed to plot. So, all the reaction forces and unknown components were to be calculated. The model diagram was displayed in Figure 10 and the free body diagrams were illustrated in Figure 11, 12, 13, 14 and 15 respectively to calculate the unknown forces and reactions.
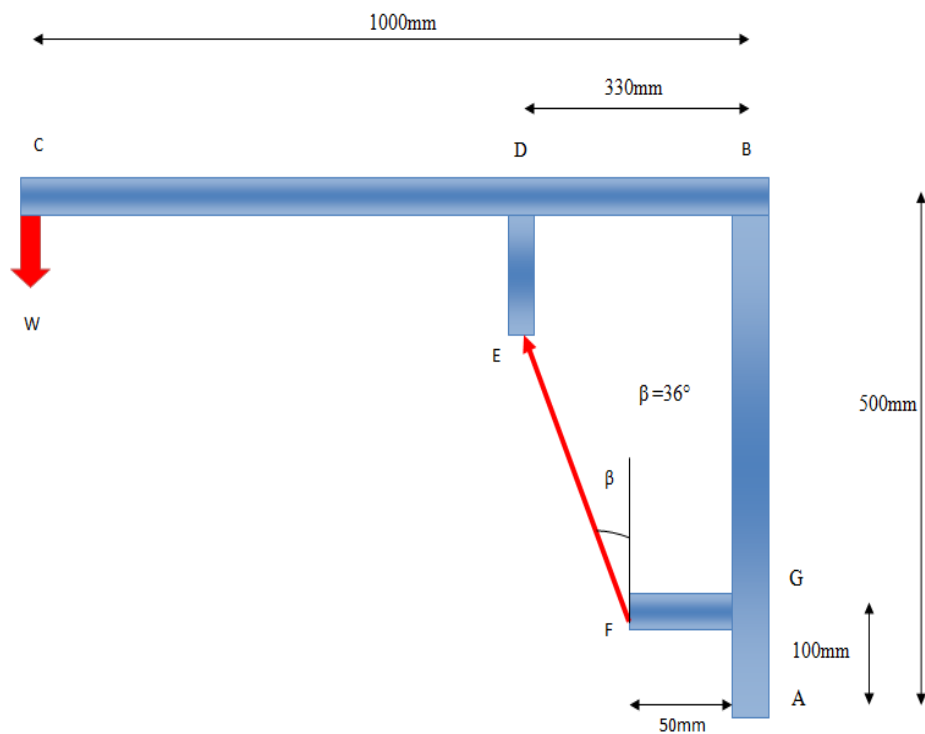

Figure 10 Model diagram when the maximum load acts
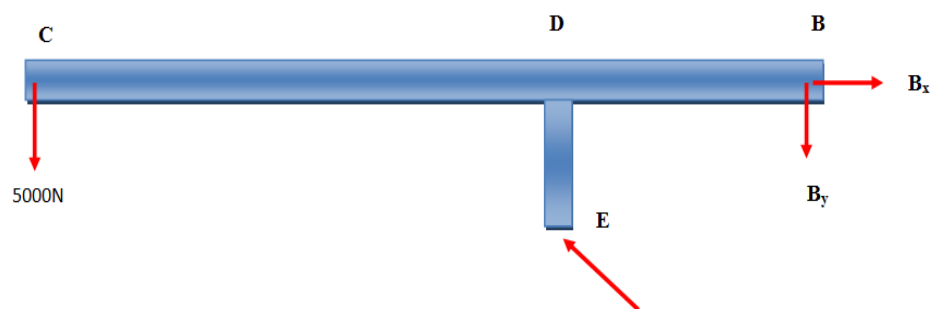
Figure 11 Free body diagram of the boom
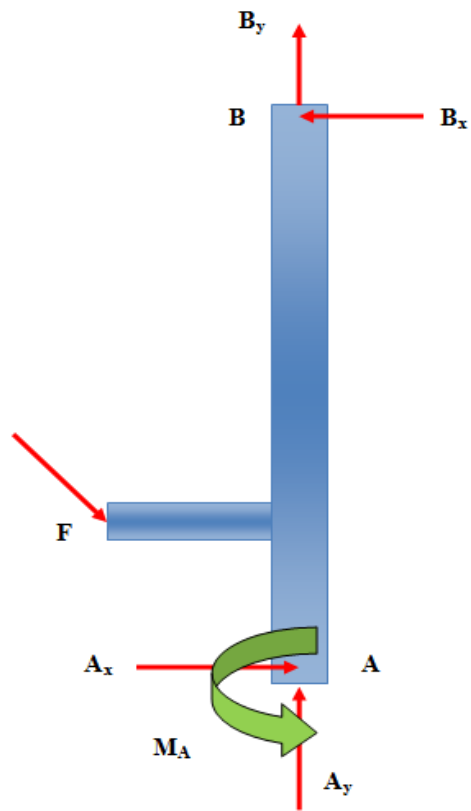
Figure 12 Free body diagram of the beam
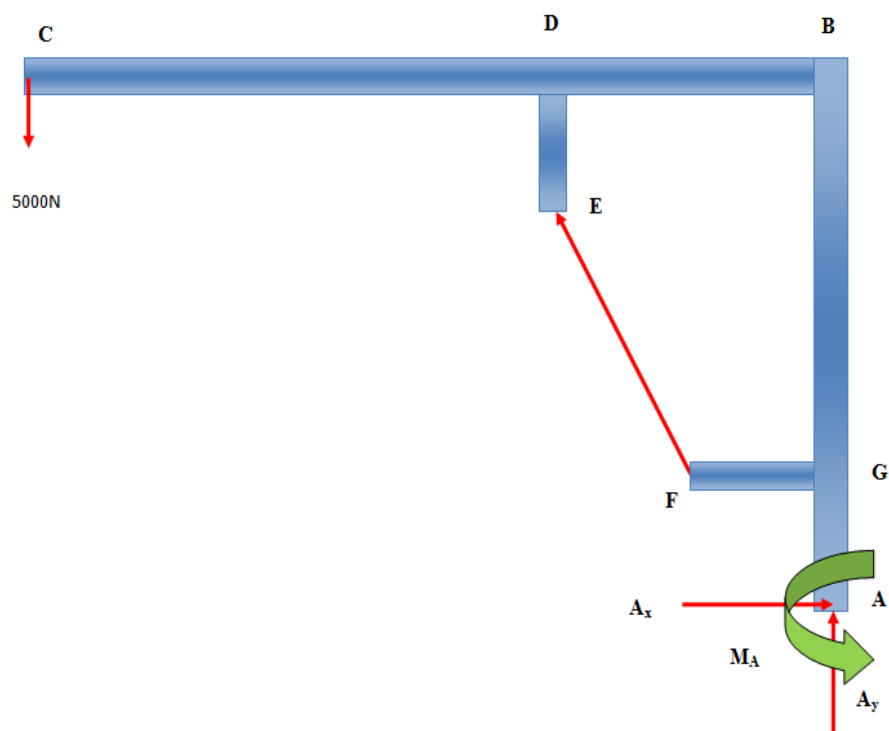
Figure 13 Free body diagram of the model
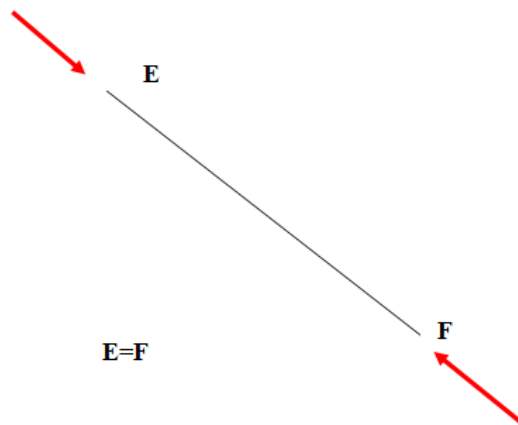
Figure 14 Free body diagram of cylinder

From the free body diagram of figure 13,

Σ Fx = 0 (1.1)

Ax = 0

Σ Fy = 0 (1.2)

Ay - 5000 = 0

Ay = 5000N

Σ M_A = M_A + 5000N . 1000 mm (1.3)

M_A= 5000000 Nmm

M_A= 5000 KNmm

From free body diagram of figure 11,
Σ M_B = 0
Σ M_B = 5000N.1000mm - E.sinβ .50mm - E.Cosβ .330mm
0 = 5000000Nmm - E.Sin36.50mm - E.Cos36.330mm
E = 16869N

Hence, the total force required for the hydraulic cylinder to move maximum mass was found to be 16,869 N. In fact, this helped to find the right size and type of the hydraulic cylinder in the cylinder selection process.
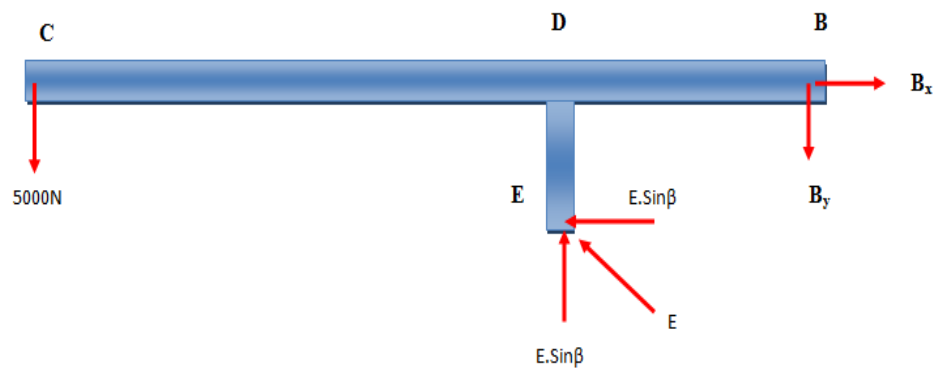
Figure 15 Detail of the free body diagram of the boom

Σ Fx = 0

 Bx - E.sinβ = 0

Bx = 9915 N

Σ Fy = 0

 Σ Fy = By + 5000N - E.cosβ

By = 8647 N

Hence, all the unknown reaction forces and unknown components were calculated and the plotted into shear force diagram and bending moment diagram to find out shear force and bending moment at each points.

The shear force diagram and bending moment diagram for beam AB was made and displayed in Figure 16. Similarly, the shear force diagram and bending moment diagram for boom BC was also plotted and displayed in the Figure 17.
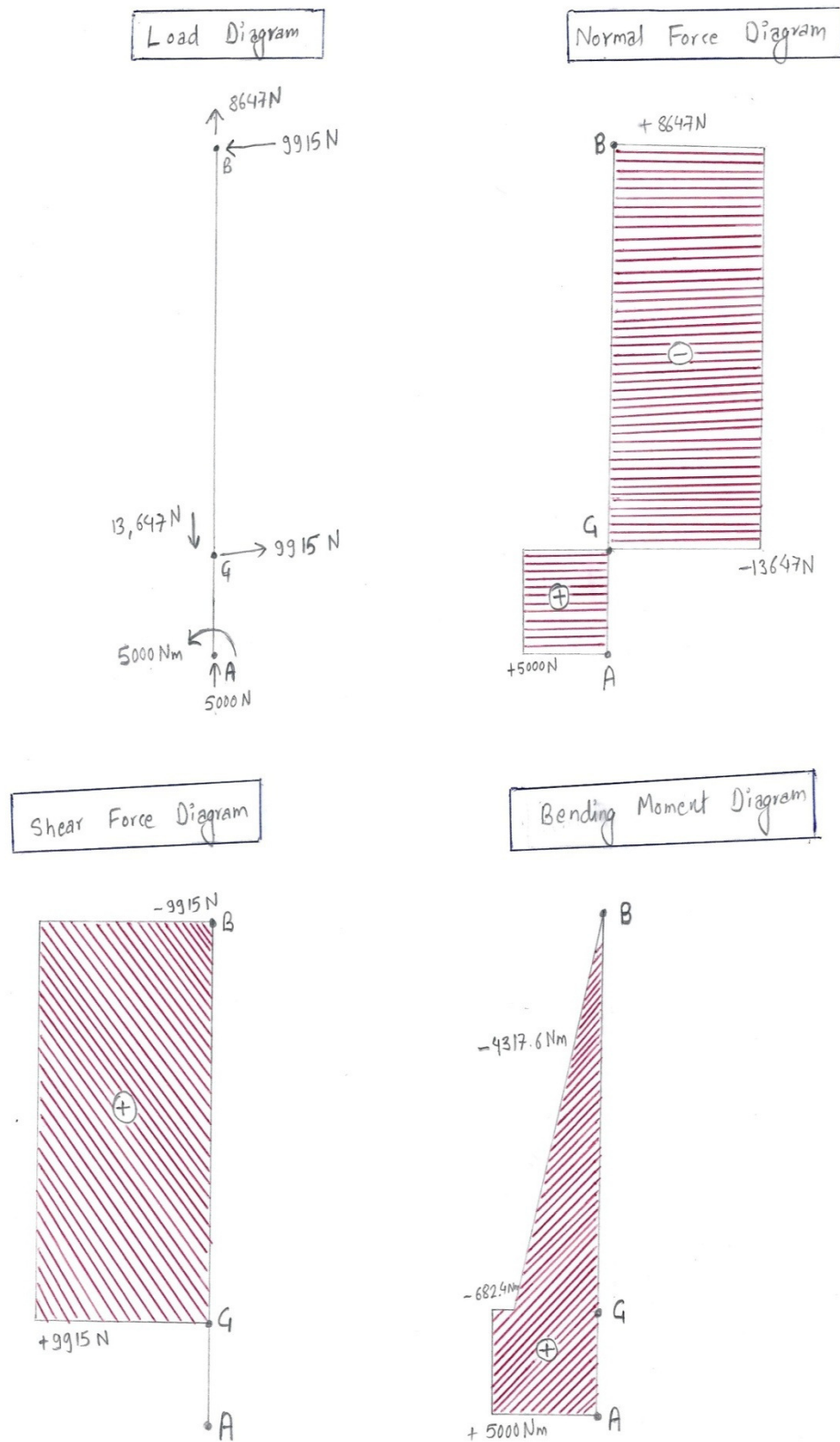
Figure 16 Shear force and bending moment diagram for the main beam
AB

Load Diagram

495.8 Nm

C   D ← 9915N   B → 9915 N

5000N   13647N   8647 N

Normal Force Diagram

C   D   B

+9915N

−9915 N

Shear Force Diagram

+5000N

C   (+)   D   B +8647N

(−)

− 13647N

Bending Moment Diagram

+3350 Nm
−495.8 Nm
− 2854.2 Nm

(+)

C   D   B
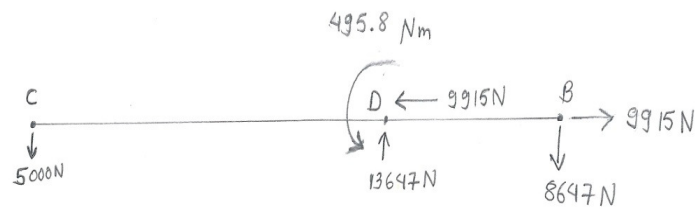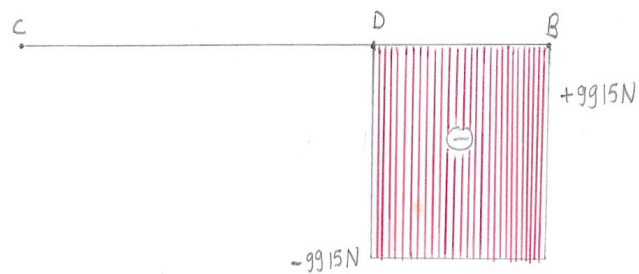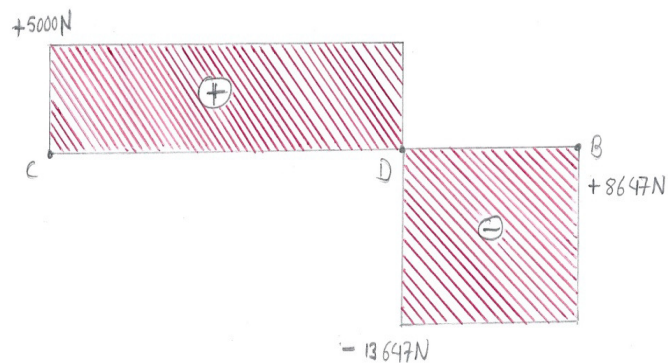
Figure 17 Shear force and bending moment diagram for the boom BC

From the bending moment diagram in the Figure 16, it was concluded that the maximum bending moment occurs between point A and G in the beam AB. The maximum bending moment was observed to be 5000 Nm.

Similarly, the bending moment was found maximum at point D on the boom BC from the bending moment diagram in the figure 17. The maximum bending moment was noted to be 3350 Nm at point D.

In this way, the calculation of the bending moments were performed in order to move into beam design process. The beam must be designed in such a way that it could withstand the maximum bending moment.
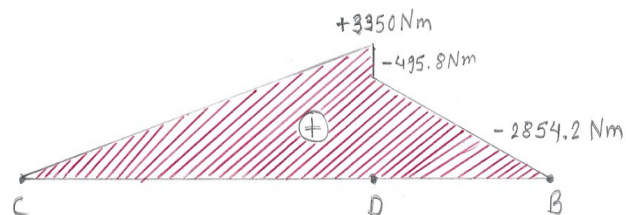
6 BEAM DESIGN AND CALCULATIONS

6.1 Beam design process

6.1.1 Conditions for the beam

There were some requirements that the beam needed to meet:
- It was to be strong enough to support the maximum weight of 500 kg when picked up by the boom.
- The beam needed to be designed in such a way that it would be able to rotate all around 360 degree.
- The mass of the beam should be minimized in order to justify the light weight design requirement.
- The height of the beam should be as minimum as possible so that the design part could be easily covered with the trailer lid. Otherwise the design would require more complex form such as folding for its proper management right after the use.
- The beam needed to be attached to some components such as brackets so that it would be easier to connect to the boom and the cylinder.
- The position of the brackets had to be designed so that its connection to the boom and the cylinder would work efficiently.
- The base plate had to be designed in such a way that it could give stability to the system and to be connected to the trailer with the help of the bolts.
- The size of the bolts had to be calculated so that it could handle the maximum bending moment.

6.1.2 Beam profile and material

There were several alternatives for the beam profile such as I-beam, hollow rectangular profile beam, hollow circular profile beam and so on. However, to provide rotational ability along with light weight requirement, the hollow circular profile beam was chosen as shown in Figure 18.
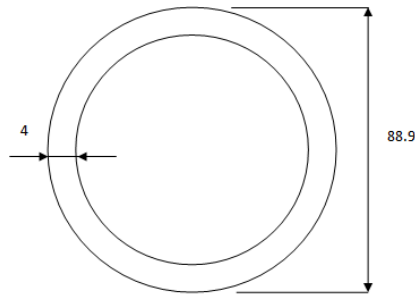
Figure 18 Hollow circular beam

The common S355 steel was chosen for the beam profile.

6.1.3 Beam design calculation

First of all, the calculation of the diameter of the circular beam was determined by using bending stress formula. The section modulus of the beam was calculated and the required beam size was selected by using Ruukki catalogue.

The maximum bending moment at the top of the circular beam occurs when the maximum load is applied on the system. So the maximum bending moment is 5000Nm. The yield strength of the structural steel S355 is 355 Mpa and the safety factor is 1.5.

M_{bMax} = 5000Nm

n = 1.5

σ_{max} = 355 Mpa = 355 Nmm^{-2}

Now,

$$n = \frac{\sigma_{max}}{\sigma_{all}} \text{ ,} \qquad\qquad (1.4)$$

$$\sigma_{all} = 355/1.5$$

We have,

$$\sigma = \frac{M_{bMax}}{W} \text{ ,} \qquad\qquad (1.5)$$

$$W = 2.1 \times 10^4 \text{ mm}^3$$

Where

      M_{bMax} = Maximum Bending Moment on the beam

      σ_{all} = Allowed stress

      σ = Tensile Stress

      W = Section Modulus

      σ_{max} = maximum stress that S355 steel can stand

      n = Safety factor

As shown in Table 5, the hollow circular beam profile with the diameter of 88.9 mm with the thickness of 4 mm was selected by using Rautaruukki catalogue.

Table 5 Cross section properties of circular hollow sections
(Rautaruukki, n.d.)

D mm	T mm	M kg/m	A mm² x 10²	A m²/m	I mm⁴ x 10⁴	W mm³ x 10³	W mm³ x 10³	i mm x 10	I mm⁴ x 10⁴	W mm³ x 10³
26.9	2.0	1.23	1.56	0.085	1.22	0.91	1.24	0.88	2.44	1.81
26.9	2.5	1.50	1.92	0.085	1.44	1.07	1.49	0.87	2.88	2.14
26.9	2.6	1.56	1.98	0.085	1.48	1.10	1.54	0.86	2.96	2.20
33.7	2.0	1.56	1.99	0.106	2.51	1.49	2.01	1.12	5.02	2.98
33.7	2.5	1.92	2.45	0.106	3.00	1.78	2.44	1.11	6.00	3.56
33.7	2.6	1.99	2.54	0.106	3.09	1.84	2.52	1.10	6.19	3.67
33.7	3.0	2.27	2.89	0.106	3.44	2.04	2.84	1.09	6.88	4.08
33.7	3.2	2.41	3.07	0.106	3.60	2.14	2.99	1.08	7.21	4.28
42.4	2.0	1.99	2.54	0.133	5.19	2.45	3.27	1.43	10.38	4.90
42.4	2.5	2.46	3.13	0.133	6.26	2.95	3.99	1.41	12.52	5.91
42.4	2.6	2.55	3.25	0.133	6.46	3.05	4.12	1.41	12.93	6.10
42.4	2.9	2.82	3.60	0.133	7.06	3.33	4.53	1.40	14.11	6.66
42.4	3.0	2.91	3.71	0.133	7.25	3.42	4.67	1.40	14.49	6.84
42.4	3.2	3.09	3.94	0.133	7.62	3.59	4.93	1.39	15.24	7.19
42.4	4.0	3.79	4.83	0.133	8.99	4.24	5.92	1.36	17.98	8.48
48.3	2.0	2.28	2.91	0.152	7.81	3.23	4.29	1.64	15.62	6.47
48.3	2.5	2.82	3.60	0.152	9.46	3.92	5.25	1.62	18.92	7.83
48.3	2.6	2.93	3.73	0.152	9.78	4.05	5.44	1.62	19.55	8.10
48.3	3.0	3.35	4.27	0.152	11.00	4.55	6.17	1.61	22.00	9.11
48.3	3.2	3.56	4.53	0.152	11.59	4.80	6.52	1.60	23.17	9.59
48.3	4.0	4.37	5.57	0.152	13.77	5.70	7.87	1.57	27.54	11.40
60.3	2.0	2.88	3.66	0.189	15.58	5.17	6.80	2.06	31.16	10.34
60.3	2.5	3.56	4.54	0.189	18.99	6.30	8.36	2.05	37.99	12.60
60.3	2.9	4.11	5.23	0.189	21.59	7.16	9.56	2.03	43.18	14.32
60.3	3.0	4.24	5.40	0.189	22.22	7.37	9.86	2.03	44.45	14.74
60.3	3.2	4.51	5.74	0.189	23.47	7.78	10.44	2.02	46.94	15.57
60.3	4.0	5.55	7.07	0.189	28.17	9.34	12.70	2.00	56.35	18.69
60.3	5.0	6.82	8.69	0.189	33.48	11.10	15.33	1.96	66.95	22.21
76.1	2.0	3.65	4.66	0.239	31.98	8.40	10.98	2.62	63.96	16.81
76.1	2.5	4.54	5.78	0.239	39.19	10.30	13.55	2.60	78.37	20.60
76.1	2.9	5.24	6.67	0.239	44.74	11.76	15.55	2.59	89.48	23.52
76.1	3.0	5.41	6.89	0.239	46.10	12.11	16.04	2.59	92.19	24.23
76.1	4.0	7.11	9.06	0.239	59.06	15.52	20.81	2.55	118.11	31.04
76.1	5.0	8.77	11.17	0.239	70.92	18.64	25.32	2.52	141.84	37.28
76.1	6.3	10.80	13.81	0.239	84.82	22.29	30.78	2.48	169.64	44.58
88.9	2.5	5.33	6.79	0.279	63.37	14.26	18.67	3.06	126.75	28.51
88.9	3.0	6.36	8.10	0.279	74.76	16.82	22.15	3.04	149.53	33.64
88.9	3.2	6.76	8.62	0.279	79.21	17.82	23.51	3.03	158.41	35.64
88.9	4.0	8.38	10.67	0.279	96.34	21.67	28.85	3.00	192.68	43.35
88.9	5.0	10.40	13.18	0.279	116.37	26.18	35.24	2.97	232.75	52.36

6.1.4   Beam design

The length of the beam was suitably chosen as 466mm which would be attached to the base plate during assembly which makes up total height 496 mm. The hollow circular beam profile of diameter 88.9 mm with the thickness of 4 mm was selected after the calculation, which was attached to the base plate.

Another circular hollow beam of diameter 96.9 mm with the thickness of 4 mm was chosen to be placed just outside the selected beam so that it could provide the ability to rotate all around 360°. The outer beam was provided with 6 holes which were symmetrically placed all around 360°. The holes were placed at 144 mm from the top of that beam. Each

hole had a diameter of 8mm making an angle of 60° to each other. The inner beam was provided with two holes in the same axis with a hole diameter of 8mm and the holes were 140mm from top of the beam. The external beam was mobile while inner beam was fixed along with the base plate. The holes of both beam should be coincided and locked with the help of keys.
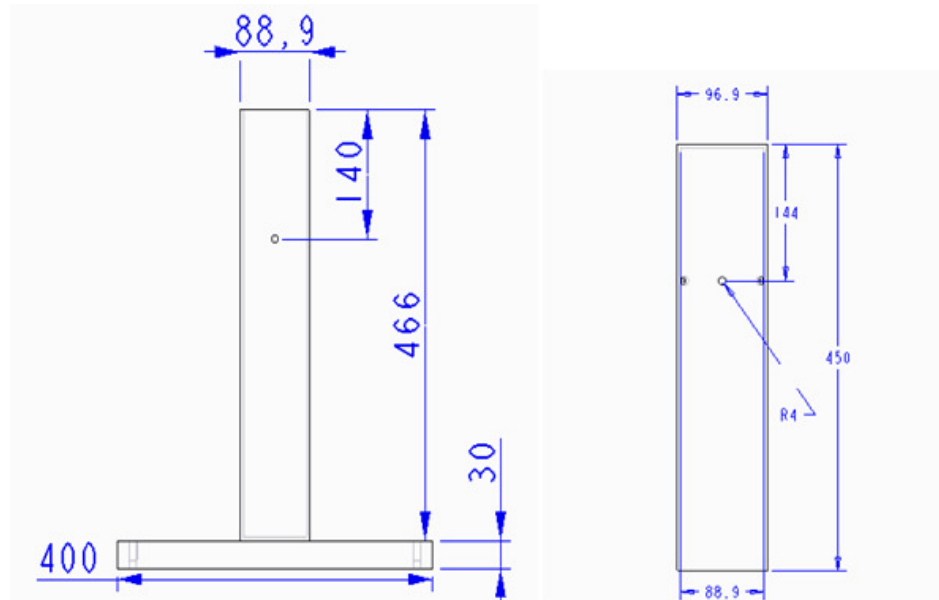

Figure 19 Inner beam with base plate (left) and outer beam (right)

The two brackets of identical size were placed on the external circular hollow beam. Upper beam bracket was rested at the tip of the beam whereas lower beam bracket was mounted at 340 mm from the top of the beam as illustrated in Figure 19.

The hollow circular beam was placed on the rectangular base plate whose dimensions were assigned in another topic 6.1.6.

6.1.5   Beam brackets

The two brackets placed on the outer beam were identical except the size and the position of the holes. The diameter of the hole of upper bracket was chosen to be 8 mm. The position of the hole was yet to be determined. The upper bracket was to be connected to the boom by using keys to lock upper beam bracket and the boom through the axis of the holes as shown in Figure 20. So the hole should be away from the beam so that boom would be able to attain the maximum height or minimum height freely without any disturbance. The position of the hole in the upper bracket had to be calculated.
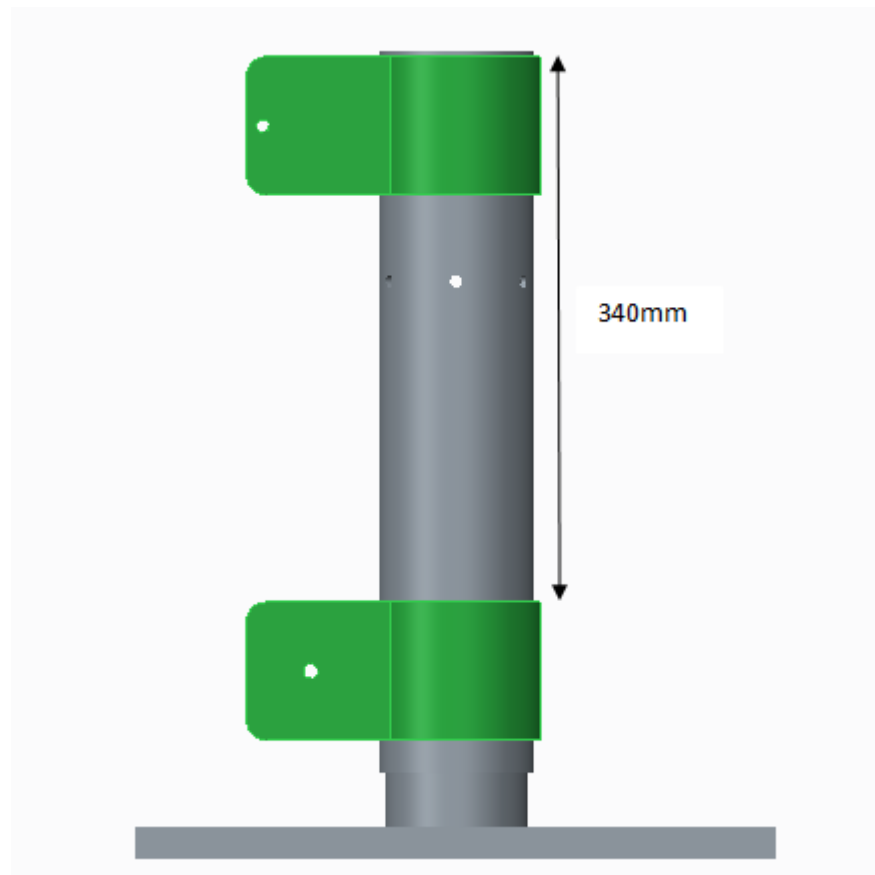
Figure 20 Upper beam bracket and lower beam bracket

First of all, the allowed shear stress was calculated as the material and the safety factor were known.

σ_{max} =355 Mpa = 355 Nmm^{-2}

n = 1.5

The allowed shear stress could be calculated as

$$\tau_{all} = \frac{0.6 \times \sigma_{max}}{n},$$
(1,6)

τ_{all} = 142 Mpa = 142 Nmm^{-2}

Now,

$$\tau = \frac{F}{4A}$$
(1.7)

The force F was determined by the resultant of the forces 9915N and 8647N which was found to be 13,156N. As shown in Figure 21, the area denoted by A was calculated by using following equation 1.8

A= l x t (1.8)

where,

l = distance between the hole and the edge of the bracket

t = thickness of the bracket
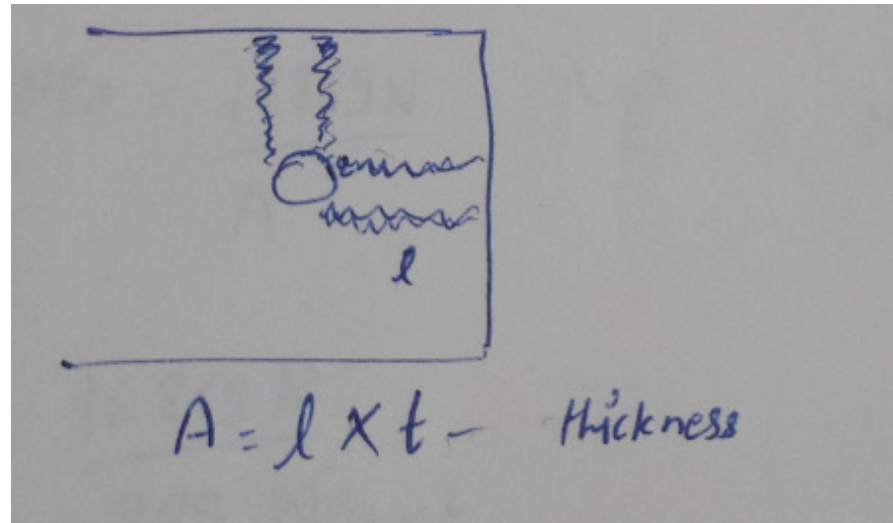
Figure 21 Edge of the upper beam bracket

Any one of the two parameters should be assumed and the calculation should be carried out in order to find the remaining parameter. The thickness of the bracket was assumed to be 4 mm and calculation was done and found the following result.

l = 6 mm

For more safety, the distance between hole and the edge of the bracket was chosen to be 8 mm.

Similarly, the lower beam bracket was provided with centrally located hole to give enough protection when maximum cylindrical force was exerted as it connected the beam to the cylinder. The maximum force needed to be handled by the hole is the maximum force applied by the cylinder to lift the maximum load which was 16,869 N

$$\tau_{all} = \frac{0.6 \times \sigma_{max}}{n}$$

τ_{all} = 142 Mpa = 142 Nmm^{-2}

Now,

$$\tau = \frac{F}{2A} \tag{1.9}$$

Since , A = $\pi d^2/4$ (1.10)

d = 8.7 mm

Hence the size of the hole was determined to be 9 mm.

6.1.6   Base Plate

A rectangular base plate layout with a length of 400 mm, width of 200 mm and thickness of 30 mm were assumed. It was quite normal to raise a question of capability of the base plate being constructed in a mere simple assumption. Therefore, an assurance of ability of the designed structure had to be made by calculating section modulus followed by calculation of maximum bending stress as shown in appendix 14 to ensure its ability to withstand under maximum bending moment. The dimensions were chosen slightly higher ones in order to give proper, balanced weight at the base and stability to the whole design structure. The reason behind choosing a rectangular shape was to occupy the minimum possible surface area of the trailer with an aim to fit it at one side of the trailer(minimizingany possible obstructions for goods being transported). Hence, to ensure the safety of the hoist system, the base plate was designed as displayed in Figure 22.

The base plate was attached and welded together with inner beam. In addition to it, the base plate was provided with 2 pairs of holes for bolts which could be bolted together with the base of the trailer and could be disassembled and taken away from the trailer to put down into the storage or warehouse. The size of the bolts required for the base plate was calculated in the following chapter 6.1.7
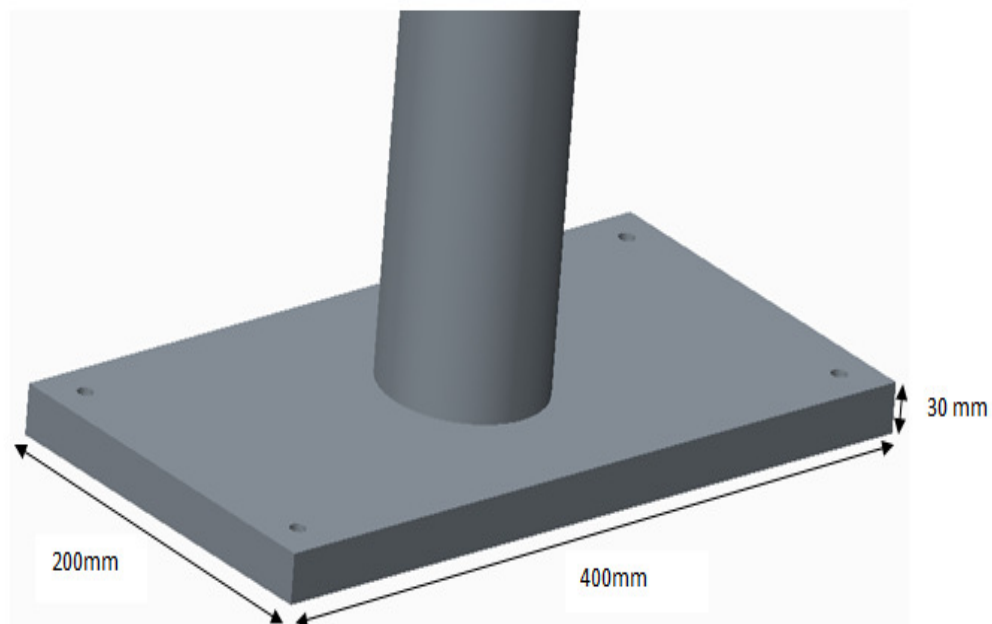
Figure 22 Simple layout of the base plate

6.1.7   Size of the base bolts

The base of the hydraulic hoist had to be fixed to the trailer temporarily so bolts were the most suitable elements for the purpose. The bolts had to be strong enough to hold the maximum moment taking place at that point of 5000 Nm. The number of bolts were to be determined which was decided to be 2 pairs as shown in Figure 23.
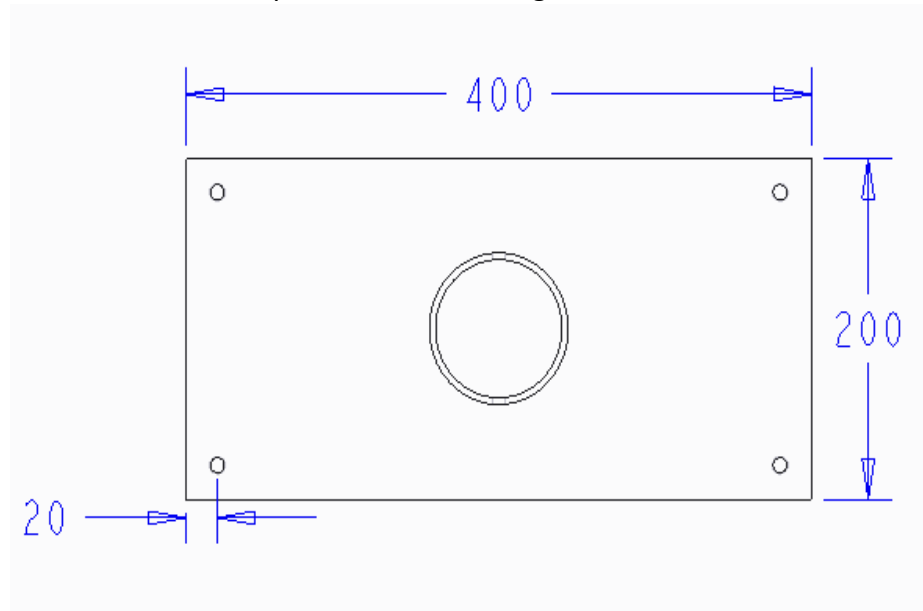
Figure 23 The bolts at the base of the beam

The distance between the bolt and the edge of the base (l) was assumed to be 20 mm.

2F.l = 5000   (1.11)

Hence, F = 12500 N

Now, the tensile stress (σ) for the steel S355 was figured out.
        σ = 355/n

σ = 236.6 Mpa

After that, the calculation of the required bolt diameter was carried out by using the formula of tensile stress as shown in equation

$$\sigma = \frac{F}{A}$$ (1.12)

Since , A = πd²/4

d = 8.2 mm

Hence, the required bolt size was determined to be 10 mm or M10 bolt.

6.2   Boom design process

6.2.1   Conditions for the boom

There were some requirements that the boom needed to meet:
- It was to be strong enough to support the maximum weight of 500kg.
- The boom needed to be designed in such a way that it would be able to increase or decrease its length according to the variation of the mass to be lifted.
- The mass of the boom should be minimized in order to justify the light weight design requirement.
- The boom needed to be attached to some component such as brackets, holes so that it would be easier to connect to the beam and the cylinder.
- The position of the bracket and holes had to be designed so that its connection to the boom and the cylinder worked out efficiently.
- The size and position of the hole and the bolt that held the mass was to be determined.

6.2.2   Boom profile and material

There were several alternatives for the beam profile such as an I-beam, a hollow rectangular profile beam, a hollow circular profile beam and so on. However, to maximize the bending moment of the beam through the y-axis from vertical forces and to minimize the mass, rectangular hollow section elongated at y-axis was chosen as illustrated in Figure 24.
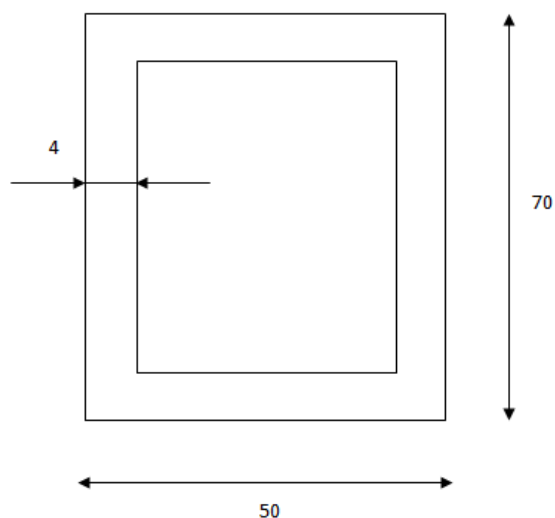
Figure 24 Rectangular hollow beam
The common S355 steel was chosen for the boom profile.

6.2.3 Boom design calculation

The maximum bending occurs at the point E where the force of the cylinder applies. when the maximum load is applied on it. So the maximum bending moment is 5000Nm. The yield strength of the structural steel S355 is 355 Mpa and the safety factor is 1.5.

M_{bMax} = 3350Nm

n = 1.5

σ_{max} = 355 Mpa = 355 Nmm^{-2}

Now,
$$n = \frac{\sigma_{max}}{\sigma_{all}}$$

σ_{all} = 355/1.5

We have,
$$\sigma = \frac{M_{bMax}}{W}$$

W = 1.42 x 10^4 mm^3

Where
M_{bMax} = Maximum Bending Moment on the beam
σ_{all} = Allowed stress
σ = tensile stress
W = Section Modulus
σ_{max} = maximum stress that S355 steel can stand
n = Safety factor

As mentioned in Table 6, the rectangular beam with the 70mm x 50mm with the thickness of 4 mm was selected by using Rautaruukki catalogue.

Table 6 Cross section properties of circular hollow sections
(Rautaruukki, n.d.)

Dimensions and cross-sectional properties. Rectangular hollow sections

M = Weight
A = Cross-section area
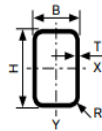A_u = External surface area
I = Moment of inertia
W = Section modulus

i = Radius of gyration
I_v = Torsion modulus
W_v = Section modulus in torsion
Theoretical density = 7.85 kg/dm³

The cross-sectional properties have been calculated by using nominal dimensions H. B and T and corner outer radius R.
R = 2.0 x T. when T ≤ 6.0 mm
R = 2.5 x T. when 6.0 mm < T ≤ 10.0 mm
R = 3.0 x T. when T > 10.0 mm

H x B mm	T mm	M kg/m	A mm²x10²	A_u m²/m	I_x mm⁴x10⁴	W_x m³x10³	W_px mm³x10³	i_x mm x10	I_y mm⁴x10⁴	W_y mm³x10³	W_py mm³x10³	i_y mm x10	I_yv mm⁴x10⁴	W_v mm³x10³
40 x 20	2.0	1.68	2.14	0.113	4.05	2.02	2.61	1.38	1.34	1.34	1.60	0.79	3.45	2.36
40 x 20	2.5	2.03	2.59	0.111	4.69	2.35	3.09	1.35	1.54	1.54	1.88	0.77	4.06	2.72
40 x 20	3.0	2.36	3.01	0.110	5.21	2.60	3.50	1.32	1.68	1.68	2.12	0.75	4.57	3.00
40 x 30	2.0	1.99	2.54	0.133	5.49	2.75	3.37	1.47	3.51	2.34	2.77	1.18	7.07	3.79
40 x 30	2.5	2.42	3.09	0.131	6.45	3.23	4.03	1.45	4.10	2.74	3.30	1.15	8.47	4.46
40 x 30	3.0	2.83	3.61	0.130	7.27	3.63	4.61	1.42	4.60	3.07	3.77	1.13	9.72	5.03
50 x 30	2.0	2.31	2.94	0.153	9.54	3.81	4.74	1.8	4.29	2.86	3.33	1.21	9.77	4.84
50 x 30	2.5	2.82	3.59	0.151	11.30	4.52	5.70	1.77	5.05	3.37	3.98	1.19	11.74	5.72
50 x 30	3.0	3.30	4.21	0.150	12.83	5.13	6.57	1.75	5.70	3.80	4.58	1.16	13.53	6.49
50 x 30	4.0	4.20	5.35	0.146	15.25	6.10	8.05	1.69	6.69	4.46	5.58	1.12	16.53	7.71
60 x 40	2.0	2.93	3.74	0.193	18.41	6.14	7.47	2.22	9.83	4.92	5.65	1.62	20.70	8.12
60 x 40	2.5	3.60	4.59	0.191	22.07	7.36	9.06	2.19	11.74	5.87	6.84	1.60	25.14	9.72
60 x 40	3.0	4.25	5.41	0.190	25.38	8.46	10.53	2.17	13.44	6.72	7.94	1.58	29.28	11.17
60 x 40	4.0	5.45	6.95	0.186	30.99	10.33	13.16	2.11	16.28	8.14	9.89	1.53	36.67	13.65
60 x 40	5.0	6.56	8.36	0.183	35.33	11.78	15.38	2.06	18.43	9.21	11.52	1.48	42.85	15.60
70 x 50	2.0	3.56	4.54	0.233	31.48	8.99	10.80	2.63	18.76	7.50	8.58	2.03	37.45	12.20
70 x 50	2.5	4.39	5.59	0.231	38.01	10.86	13.16	2.61	22.59	9.04	10.45	2.01	45.75	14.72
70 x 50	3.0	5.19	6.61	0.230	44.05	12.59	15.40	2.58	26.10	10.44	12.21	1.99	53.62	17.06
70 x 50	4.0	6.71	8.55	0.226	54.67	15.62	19.48	2.53	32.22	12.89	15.41	1.94	68.07	21.19
70 x 50	5.0	8.13	10.36	0.223	63.46	18.13	23.06	2.48	37.20	14.88	18.20	1.90	80.77	24.64
80 x 40	2.0	3.56	4.54	0.233	37.36	9.34	11.61	2.87	12.72	6.36	7.17	1.67	30.88	11.00
80 x 40	2.5	4.39	5.59	0.231	45.11	11.28	14.15	2.84	15.26	7.63	8.72	1.65	37.58	13.24

6.2.4 Boom design

The length of the boom changes with the variation of the load applied in the system. When the mass applied in the system was maximum, the length of the boom would be the minimum. The minimum possible length was chosen to be 1000 mm. On the other hand, the minimum load tends to demand the maximum length of the boom which was chosen to be 1670 mm. Since the length need was variable according to the load, the booms were designed in such a way that they could slide in or out easily whenever it required elongation or shortening of the boom length.

The hollow rectangular boom profile of 70mm x 50mm with the thickness of 4mm and length of 870mm was selected after the calculation, which was designed to slide under the outer hollow rectangular beam profile of 78mm x 58mm with the thickness of 4mm and length of 1000mm.

The inner hollow boom was provided with hole of diameter of 6 mm which was 40 mm from the edge of the boom. This hole connects the boom with the second boom which was to be locked with the help of span lock pin. The next hole at the other end of the boom is a threaded hole for a bolt to fasten in order to suspend the load on a chained hook. The threaded hole was assigned at the distance of 60 mm from the end of the boom. Figure 25 clearly shows the general outlines of an inner hollow beam.
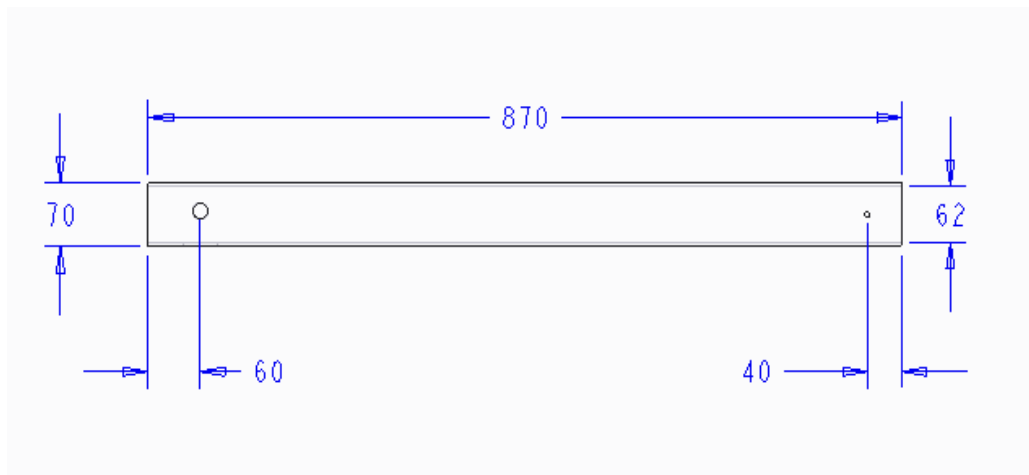
Figure 25 An illustration of inner hollow boom

On the other hand, the outer hollow boom is the one that is connected to the main beam with the help of upper beam bracket. A hole of 8 mm was assigned at 40 mm from the edge of the outer boom. Similarly, 60 mm away from the end of the boom, another hole of diameter 6 mm was assigned that connects outer and inner boom with the help of scaffolding lock. The details of outer hollow boom can be clarified through Figure 26.
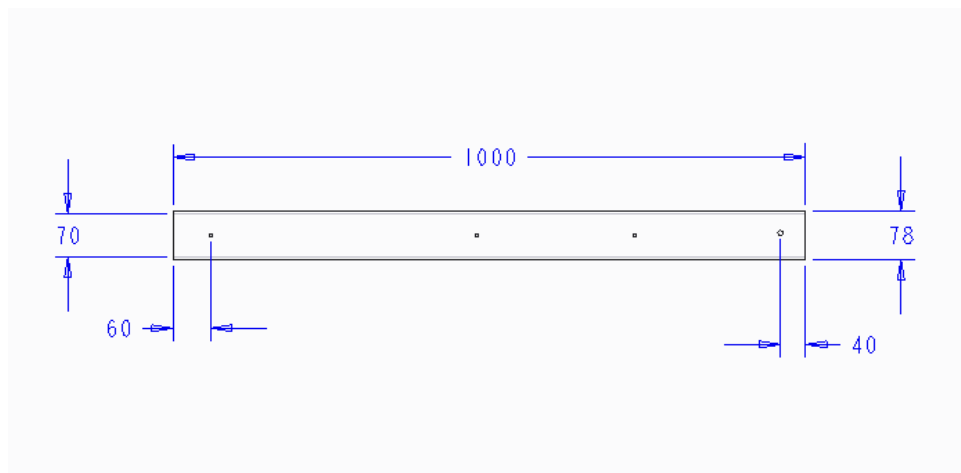

Figure 26 An illustration of outer hollow boom

During the assembly process, the holes of both booms were to be coincided and locked with the help of a span lock pin. There were two more holes with a diameter of 6 mm in the outer boom which were designed in order to shorten the length of the boom when the load was increased.
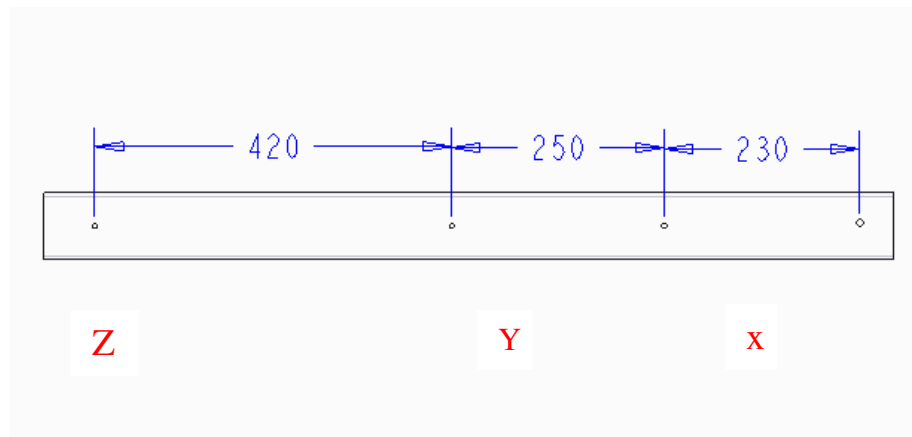
Figure 27 The selection of position of hole as per load

As seen in Figure 27, the positions X, Y and Z were attained for the holes in order to make it much simpler and easier to determine how the lengths were assigned to different loads. According to the calculations shown in Appendix 15, following results were obtained as shown in Table 7.

Table 7 Position of the holes to determine the length according to the loads

Position	Load	Length
X	5000 N	1000 mm
Y	4000 N	1250 mm
Z	3000 N	1670 mm

For instance at position X, the span lock was done in order to attach both inner and outer boom to make the length of 1000 mm that lifts the mass of 5000N. On the contrary, the lock was assigned to be done at position Z to create the length of 1670 mm to lift 3000N load.

6.2.5 Boom bracket

The boom bracket is located at the lower side of the outer boom which helps to connect the cylinder to the boom. Basically, the cylinder was mounted on the boom bracket with the help of a bolt whose size was yet to be calculated. The boom bracket was located at 595 mm away from the tip of the boom as shown in Figure 29. It was the tip of the boom where the inner boom slided inside and created a required effective length according to the change in load in order to lift the load.
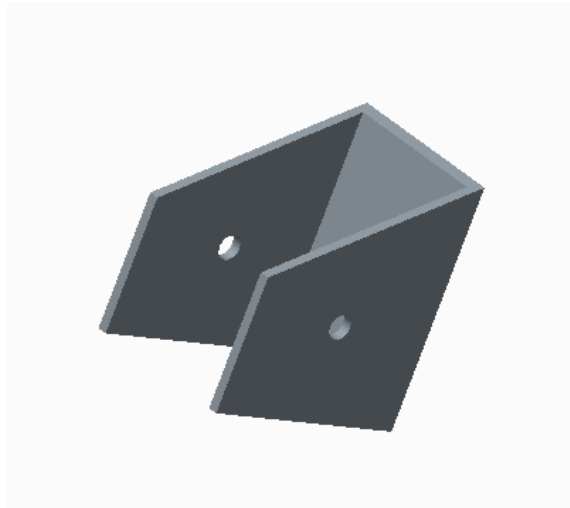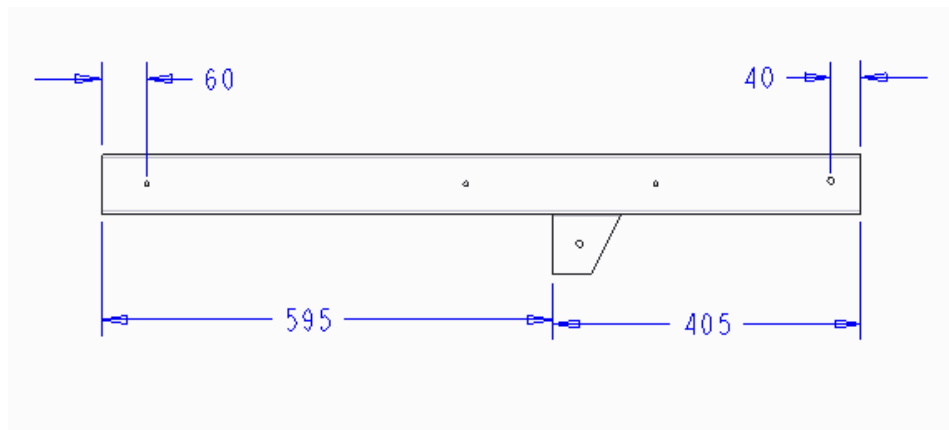
Figure 28 A general outline of boom bracket

Figure 29 The location of the boom bracket in the outer boom

The length of the boom bracket was 90 mm with a height of 80 mm and width of 66 mm which was assigned to coincide the width of the outer boom. As shown in Figure 28, the general shape of the boom bracket was right trapezoid in which two adjacent angles were right angled triangles. The slanted side was taken into account in order to place it at the tip of the cylinder so that the force applied from cylinder could easily be transmitted to the boom. The thickness of the boom bracket was 4 mm.

Similarly, the boom bracket was provided with a hole for a bolt to give enough protection when maximum cylindrical force was exerted as it connects the cylinder to the boom. Then, the diameter of the hole was calculated after the calculation of allowed shear stress (τ_{all}). The maximum force (F) needed to be handled by the hole was the maximum force applied by the cylinder to lift the maximum load which was 16,869 N.

$$\tau_{all} = \frac{0.6 \times \sigma_{max}}{n}$$

τ_{all} = 142 Mpa = 142 Nmm^{-2}

Now,

$$\tau = \frac{F}{2A}$$

Since , A = πd²/4

        d = 8.7 mm

        Hence, the size of the hole was determined to be 9 mm.
where,

        σ_{max} = maximum stress = 355 Mpa = 355 Nmm^{-2}
        n = safety factor = 1.5
        τ_{all} = allowed shear stress to the boom bracket material
        τ = shear stress to the boom bracket material
        A= surface area of the hole of the boom bracket
        d = diameter of the hole in the boom bracket

6.2.6   Bolt size at the tip of the boom

The bolt was placed at 60 mm away from the tip of the boom which was designed in order to suspend the load up to 5000N as illustrated in Figure 21. The bolt must be strong enough to resist any kind of deformation under the application of maximum load. For this purpose, the maximum bending(M_b) was calculated by using equation 21. The maximum load was 5000N. The perpendicular distance to the applied force was 23 mm as per illustrated in Figure 30.
The allowed tensile stress was calculated by dividing the maximum tensile stress by safety factor.
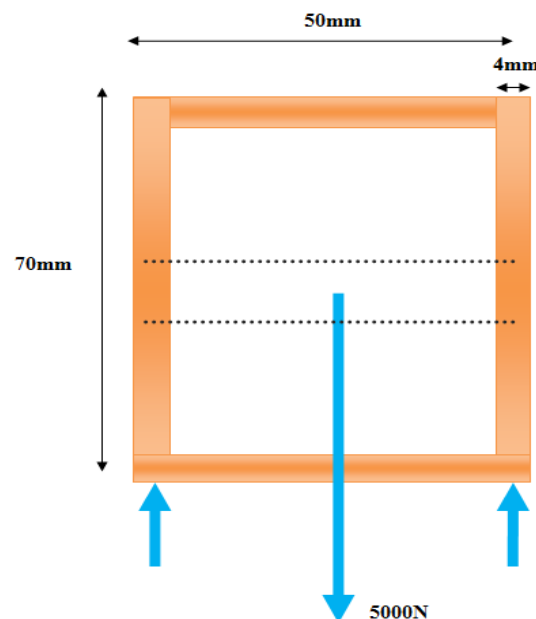
Figure 30 The maximum bending at the midpoint of the bolt

By using equation 12, the section modulus was calculated. The bolt here being used was a circular one hence the formula for section modulus for a circular was chosen as shown in equation 13. Therefore, the diameter of the required bolt size was determined.
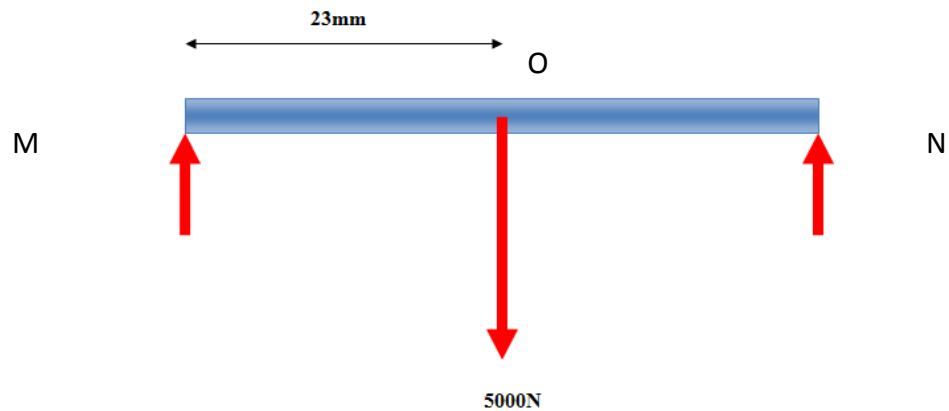
Figure 31 Free body diagram of the bolt under the load action

By using free body diagram as illustrated in Figure 31, the bolt MN acted as a beam on which the load was suspended and the side walls of the boom acted as supports which gave the reaction forces to overcome the suspended load,

M_b = F x d

M_b = 5000N x 23 mm

M_b = 115000 Nmm

Now,

σ_{max} = 355 Mpa = 355 Nmm^{-2}

n = 1.5

$$\sigma_b = \frac{M_b}{W} \hspace{4cm} (1.13)$$

$$\frac{355\ Nmm^{-2}}{1.5} = \frac{115000\ Nmm}{W}$$

$$W = \frac{\pi \times D^3}{32} \hspace{4cm} (1.14)$$

D = 17.04 mm ~ 18 mm

Hence, The bolt size was determined to be M18 with the diameter of 18mm.

Where,

      F = Applied force or load

      d = Perpendicular distance to the force

σ_b = allowed bending stress in the bolt
σ_{max} = maximum stress that steel S355 can stand
n = safety factor
W = Section modulus for circular object
D = Diameter of the bolt

7 HYDRAULIC CYLINDER SELECTION

The hydraulic cylinder selection process was one of the most significant part of this project because it was the one that determine main feature of the hoist as of how it was going to work when the whole project was completed. The weaknesses in determining the right cylinder size and type could lead to the failure of the whole design process. In addition to this, the right brand of suppliers and the one with good reputation in designing cylinders, were enlisted and researched thoroughly. The right size and right type fulfilling all the demanded features was to be chosen.

According to the objective of this thesis, an electro-hydraulic cylinder that was able to lift 500 kg of mass by using 12 V direct current. From the calculation, the maximum amount of force needed for the cylinder to counterpart it, when maximum bending moment occurs, was found to be 16.9 KN. Hence by using

$$\text{Let's say, } E = 16{,}9 \text{ KN} = 16{,}9.\,10^3 \text{ N}$$
$$P = 210 \text{ bar} = 21.10^6 \text{ Pa}$$
$$n = 0.85$$

Now,
$$E = PAn \qquad\qquad (1.15)$$

where n is a safety factor

$$A = \pi D^2/4$$

$$D = 34.8 = 35 \text{ mm}$$

Hence, the right size of the required hydraulic cylinder was found to be 35 mm.

There were various types of hydraulic cylinders found in the market. On the basis of features and fulfilling all the design requirements, an innovative new Compact Electro Hydraulic Actuator (EHA) from Parker Hannifin was decided to be chosen. The unique design of Parker's Compact EHA electro-hydraulic actuator integrates all components into a compact and tough monobloc housing inside which a high performance, double-acting hydraulic actuator, is driven via a reversible gear pump that is powered by an integrated 12 volt direct current electric motor. The pump is enclosed within the hydraulic fluid reservoir, with pressure relief and check valves also being incorporated within the system assembly as shown in Figure 32.
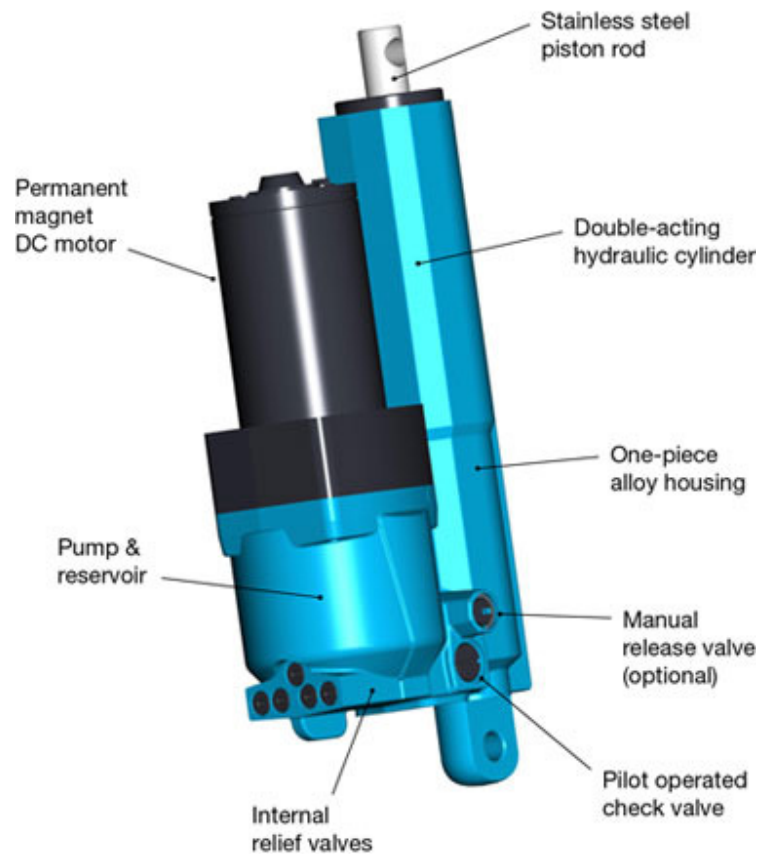
Figure 32: Compact electro-hydraulic actuator (Parker, 2013)

The hydraulic actuator had the ability to operate from the temperature range between -34°C to + 65°C which made it possible to work in the extreme winter days in Finland as well. (Parker, 2013)

Specifications
The main specifications of the chosen compact electro hydraulic actuator such as actuator, motor, pump, circuit, performance and general specifications are enlisted in the following:

    Actuator
        Type : Hydraulic, double-acting
        Bore sizes : 36.5mm
        Standard stroke lengths : 203mm
        Piston rod diameters : 19.1mm
        Standard mounting : 9.5 mm
    Motor
        Motor type : 12V DC, 560W (motor B)
        Leads – length 1.5m
        Leads – wire size : 12 gauge (for motors B & D)
        Connector type : Ring terminals, 6.6mm
    Pump
        Pump type : gear, reversible
        Pump capacities : .190 gear = .31cc/rev
        Fluid medium : Automatic transmission fluid (ATF)

Circuit

Sealed locking hydraulic circuit with integrated pump, motor, actuator and reservoir, relief, thermal, check and back pressure valves.

Performance

Maximum force – extend : 21.35kN

Maximum force – retract : 15.57kN

Maximum speed : 84mm/sec

General

Construction – body : anodized cast aluminium, one-piece

– piston rod : stainless steel

Operating temperature range : -34°C to +65°C

Sound Level < 70dBA

Weight: 5.5 Kg   (Parker,2013)

8   ASSEMBLY PROCESS

Before starting the assembly process, it was very important to figure out if all the necessary parts were accumulated or not. The assembly of the designed parts required many small parts such as pins, bolts, nuts, washers etc that would help to link or connect two designed parts. These parts looked generally simpler and did not need to spend too much time in choosing one. The cost price between the similar parts were not much different. Hence, it also implied there is no point of taking too long in choosing one out of available variables. But it was necessary to understand the importance of the part, its general dimensions and its functionability. These parts and their suppliers were described briefly below:

a) M18 bolt

The bolt M18 with the length of 80 mm was required to be allocated at the tip of the inner boom. The external width of the inner boom being 50 mm, there would be enough space for the nut, washer and head of the bolt itself. The main function of this bolt was to carry off the load up to 500 kg with the help of chain and hook. Hence, the M18 bolt with grade 8.8 made up of zinc material could be chosen which was supplied by supplier Fastenal, as illustrated in the Figure 33.

Figure 33: M18 bolt (left) and M10 bolt(right)

(Fastenal, 2016) (Screwfix, 2016)

b) M10 bolt

At the base of the beam, 4 bolts of size M10 with length of 60 mm were required in order to fix the base plate of the beam alongside the base of the trailer. The thickness of the base was 30 mm, so the bolts with 60 mm length would perfectly fit into it. Hence, the M10 bolts with grade 8.8 made up of stainless steel supplied by supplier Screwfix could be assigned for the assembly as in the Figure 33. (Screwfix, 2016.)

c) Clevis and cotter pins

Clevis pin is actually a cylindrical pin with a head at one end and one or more holes at the other end that runs through the diameter. A clevis pin is typically locked in place with a cotter pin to provide motion between components where great accuracy is not required. (Apexfasteners, 2016.) The assembly needed 3 clevis pins and 3 cotter pins of different sizes.

At upper beam bracket, the clevis pin of 8 mm diameter with length of 76 mm, could be chosen. The clevis pin had a hole of 2.6 mm diameter that fits with cotter pin of size 2 mm x 25.4 mm together perfectly. Both clevis pin and cotter pin were made up of stainless steel. (Tasman, 2016.) Figure 34 displays the clevis pins and cotter pins produced by manufacturer Tasman Industries.

Figure 34: clevis pins (left) and cotter pins (right)

(Tasman, 2016)

Likewise, the clevis pins that have diameter of 9 mm with length of 76 mm, made up of stainless steel could be chosen to be placed at lower beam bracket and boom bracket. The clevis pins having holes of 2.6 mm diameter could be a great fit when cotter pins of 2 mm x 25.4 mm is placed along.

d) Span lock pins

Two span pins of diameter 8 mm but of variable lengths could be assigned for locking the circular movement of the beams and the linear moment of the booms. Since the external diameter of the external beam was 97.5 mm, the effective length of chosen span lock pin at beam should always be greater than 97.5 mm but it was not necessarily be precise one. Similarly, the effective length of the span lock pin at boom should be greater than 58 mm. An illustration of a span lock pin is displayed in the Figure 35.

Figure 35: Span lock pin (Scaffold, 2016)

e) Hook and chain

The clevis grab chain hook of 8 mm made up of grade 70 alloy steel, supplied by GS products, had a breaking load of 9,000 kg. This hook could easily be attached to the chain of 8 mm by simply removing the clevis pin. GS products also supplied chains of 8 mm diameter grade 80 steel that had 2,000 kg working load limit. This chain was approved for overhead lifting as well as an excellent choice for towing and lifting. The grade 80 chain was suitable for using in a temperature range of - 40 to 200 C. These two products fitted well if they had been chosen for the design project. The clevis grab chain hook and grade 80 chain are displayed in the Figure 36. (GS, 2016.)

Figure 36 : Clevis grab chain hook (left) and chain (right) (GS, 2016)

All of the above mentioned parts were not necessarily be bought online. Those part which were available in the market in Finland, could be ordered easily which would save both shipping cost as well as time.

Before starting the assembly process, there was a very important task yet to be done regarding foundation plate. A foundation plate was needed to

support the base plate of the beam and bolted together to give stability to the hoist system. The foundation plate did not require to be in a precise dimension but a rectangular shaped or symmetrical shaped slightly larger and thicker than base plate of the beam would work since the foundation plate was welded firmly to the frame of the trailer. 4 thread holes for M10 bolts were to be made on the foundation plate as well as trailer base (usually made up of wood) coinciding the position of holes as on the base of the beam so that they could be fastened together firmly. The exact location on the trailer where the hoist had to be installed was determined to be at one of the side about halfway through the tail so that there would still be enough space in the trailer.

Figure 37 : An illustration of support legs (Tangshan, 2016)

Secondly, when the arm of hydraulic hoist system fetched the mass from same side of the trailer where the hoist itself was installed, there might arise a situation that the trailer would bend down or even somersault. To prevent this situation, an additional support or a stabilizer leg could be attached so that it could stand even even the maximum load was applied. A pair of stabilizer legs manufactured by Tangshan Rongcheng were shown in Figure 37. The pattern of the leg might vary due to availability of variety of trailers. This support should be able to get disassembled and kept at the storage when it was not necessary. Another alternative to counter this problem was using a wooden log just as a leg to create balance which would be much cheaper, easier and time saving. Now, the assembly process could be commenced since all the prerequisites were ready. The general assembly processes of this hydraulic hoist project were discussed stepwise below:

1. The base of the inner beam should be placed on the base of the trailer below which the foundation plate was welded in such a way that the threaded holes of all three plates would coincide each others.
2. M10 bolts should be aligned and fastened tightly so that the inner beam would stand firmly at the side of the trailer.
3. The outer beam should be laid upside down on the lower beam giving so that the holes of both beams coincides. It gave a possibility to rotate the outer beam around 360 degree.
4. A span lock pin should be used to lock rotation of the outer beam. The pin could easily be removed whenever the outer beam was needed to be rotated and locked again.
5. Initially, lower beam bracket should be passed through the outer beam followed by upper beam bracket that would stay at the top. The lower beam bracket should be 340 mm from the top as illustrated in Figure 20.
6. The boom bracket should be welded at the base of the outer boom in such a way that the distance between the bracket and tip of the boom was 595 mm as shown in Figure 29.
7. The holes of outer boom bracket should be aligned to the holes of the upper beam bracket so that clevis pins and cotter pins were used to lock them.
8. The inner boom should be slided inside the outer boom which possessed three pairs of holes. According to the desired length of the boom, the holes of the both booms should be coincided and locked by using span lock pin.
9. The clevis hook and chain should be coupled together.
10. The hook and chain should be suspended on the M18 bolt that would get passed through the tip of the inner boom and kept fastened tightly.
11. The Compact Electro Hydraulic Actuator (EHA) should be mounted on the upper part at the boom bracket and lower part at the lower beam bracket. Both were mounted and fixed with the help of clevis pins and cotter pins.
12. Hence, the hydraulic hoist was ready for use. Initially, the testing should be done without using mass to check if all the functions were carried out promptly.

9   TOTAL MASS OF THE ASSEMBLY

The design requirement of this project demands the total mass of the whole system including the hydraulic cylinder to be around 50 kg. Therefore, the mass of the designed system without hydraulic cylinder, bolts and lock pins was obtained from Creo Parametric 2.0 which was found to be 44 kg as shown in Figure 38.
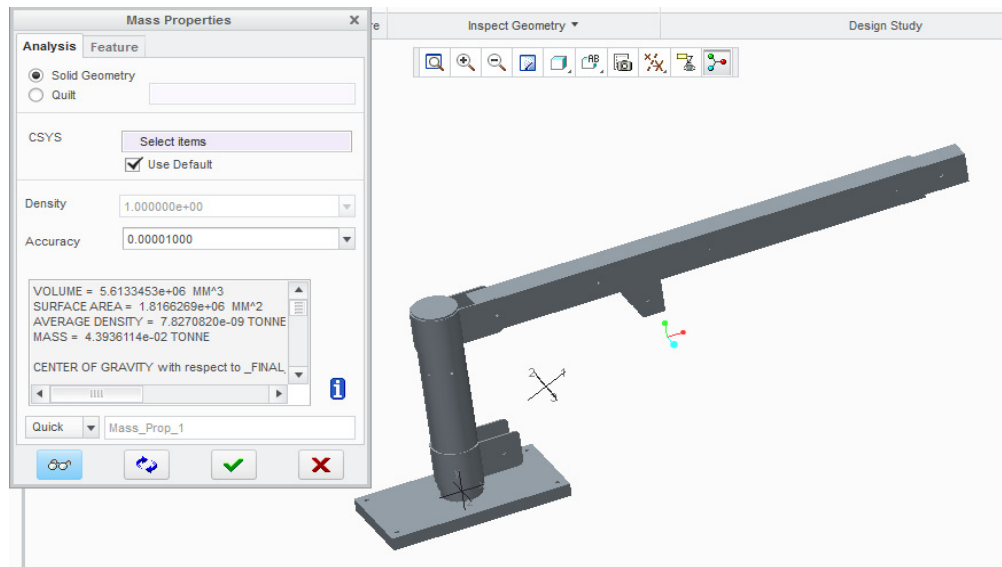
Figure 38 Mass of the Assembly without cylinder, bolts and lock pins

Similarly, the mass of the chosen hydraulic cylinder was found to be 5.5 kg. On the other hand, the mass of bolts, nuts, washers, clevis pins, cotter pins, span lock pins along the chain and clevis hook were found to be around 2.9 kg according to the Table 8. Hence, the total mass of the assembly was found to be 52.4 kg which was not far away from the target.

Table 8 Details on mass and price of bolts, cotter pins, lock pins, chain and hook (Fastenal, 2016)   (Screwfix, 2016) (Tasman, 2016)   (Scaffold, 2016) (GS, 2016)

Material	Type	Size	Location	Quantity	Mass	Price
Bolt/ nut	M18 x 80	18 mm	Tip of boom	1 pc	252 gm	2.5 €
Bolts/nuts	M10 x 60	10 mm	The base plate	4pcs	166 gm	5 €
Pin	Clevis & cotter	8 mm	U. beam bracket	1 pc	140 gm	4 €
Pin	Clevis & cotter	9 mm	L. beam bracket	1 pc	150 gm	4 €
Pin	Clevis & cotter	9 mm	Boom bracket	1 pc	150 gm	4 €
Lock pin	Span lock pin	8 mm	Beam locking	1 pc	120 gm	3 €
Lock pin	Span lock pin	8 mm	Boom locking	1 pc	120 gm	3 €
Chain	Grade 80	8 mm	Tip of the boom	1 pc	1400 gm	7 €
Hook	Clevis hook	8 mm	Tip of the chain	1 pc	360 gm	2.5 €
TOTAL					**2.9 kg**	**35 €**

10 ESTIMATION OF TOTAL COSTS

Every project starts with a budget according to the scale of the project and its nature. The estimation of the budget is done during the project planning stage that take place prior to the project process. But this hoist project was carried out without an exact sum of budget. That does not imply that the budget was

limitless but the aim of the project was to conduct the process by minimizing the budget but without compromising the quality of the final product.

The price of the design work of the parts of this project included the price of the raw materials and the manufacturing costs. The price of steel (structural sections and beams) was found to be 545 Euros per tonne as of July 2016 (MEPS, 2016). The total mass of the designed parts was around 44 kg. If 60 kg of structural steel was ordered , keeping in mind that there would be some parts wasted during the manufacturing processes, hence, it would still cost 32 Euros only. Not to forget that, the manufacturing costs in Finland are among the highest in Europe. It requires few hours of manufacturing processes and finishing processes including painting of the parts in order to prevent the product against rust. The cost was estimated to be around 350 Euros.

Most importantly, the decision of choosing of compact electro hydraulic actuator (EHA) increased the price of the whole project significantly since the actuator alone cost around 600 Euros. EHC was chosen because most of other actuators either did not qualify for the design requirements or they were very unreliable.

On the other hand, the total cost of the bolts, the nuts, the clevis, the cotter pins, the lock pins, the hook and the chain altogether was around 35 Euros. Hence, the whole project cost was found to be EUR 1017.

11 CONCLUSION

The design project to plot hydraulic hoist was carried out successfully. The designed hydraulic hoist was able to meet all the technical requirements which were targeted to it. The design has no restrictions for the road traffic regulations in Finland (RAC 2016). The total mass of the hoist system does not exceed the limit. Stated although the price seemed to be a bit higher than expected, the design project deserves an appreciation especially for having taken the bold step to decide and choose to use one of the newest innovations EHA from Parkers Hannifin Oy. It was done so that one will not simply be able to point a finger at the quality of the assigned product.

This project was a good learning experience for the author who is learning and trying to test and implement the ideas he has learned in the classroom. It was a golden opportunity provided by Mr Antti Simpura, one of the most experienced lecturers in the Mechanical Department of HAMK University of Applied Sciences to understand how design projects are usually carried out in the field of engineering.

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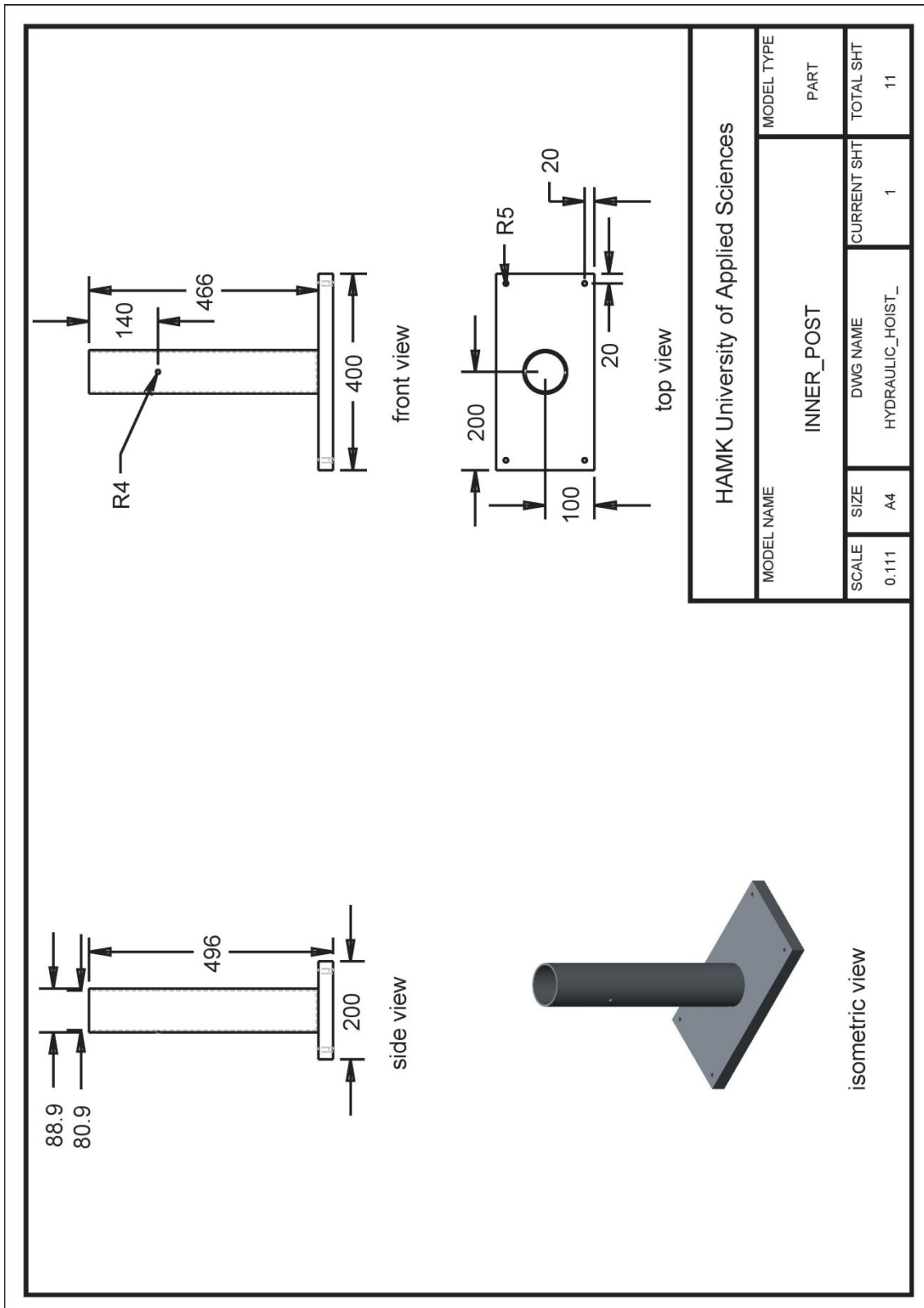
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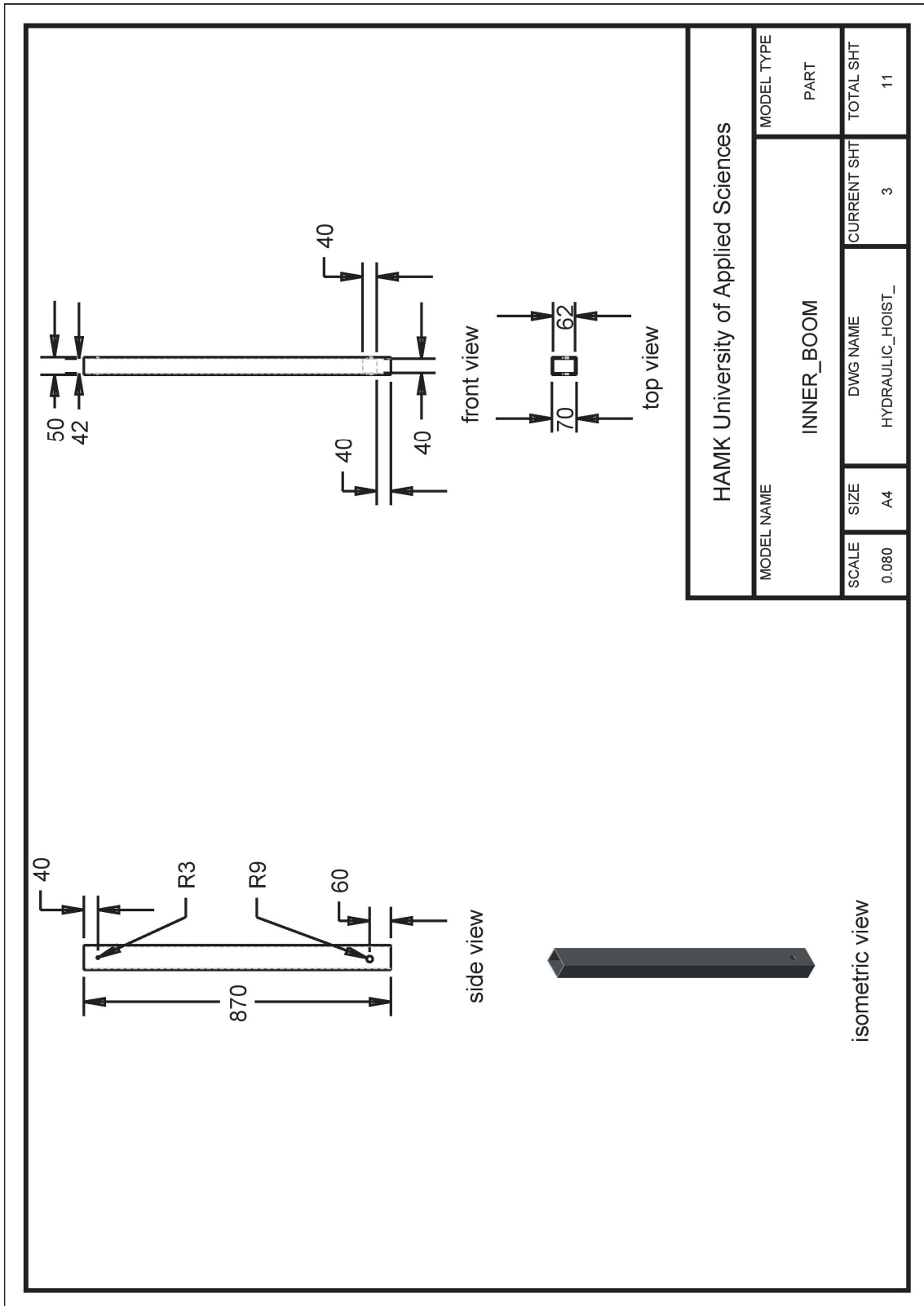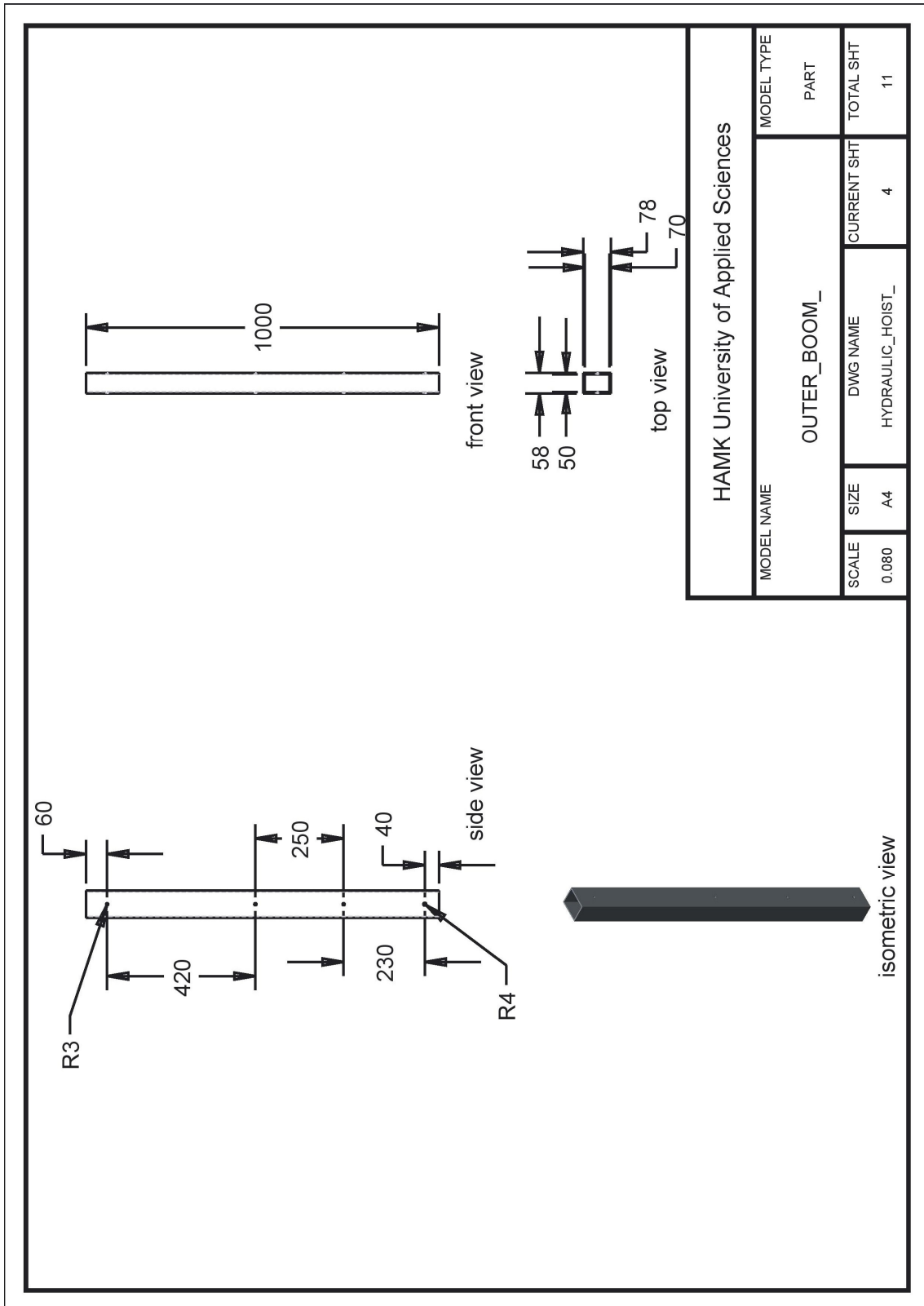
TECHNICAL DRAWING OF THE INNER BEAM/POST

TECHNICAL DRAWING OF THE OUTER BEAM/POST

front view

450

R4

R44.45
R48.45

top view

side view

306

isometric view

HAMK University of Applied Sciences

MODEL NAME

OUTER_POST

MODEL TYPE

PART

TOTAL SHT

11

CURRENT SHT

2

DWG NAME

HYDRAULIC_HOIST_

SCALE

0.160

SIZE

A4
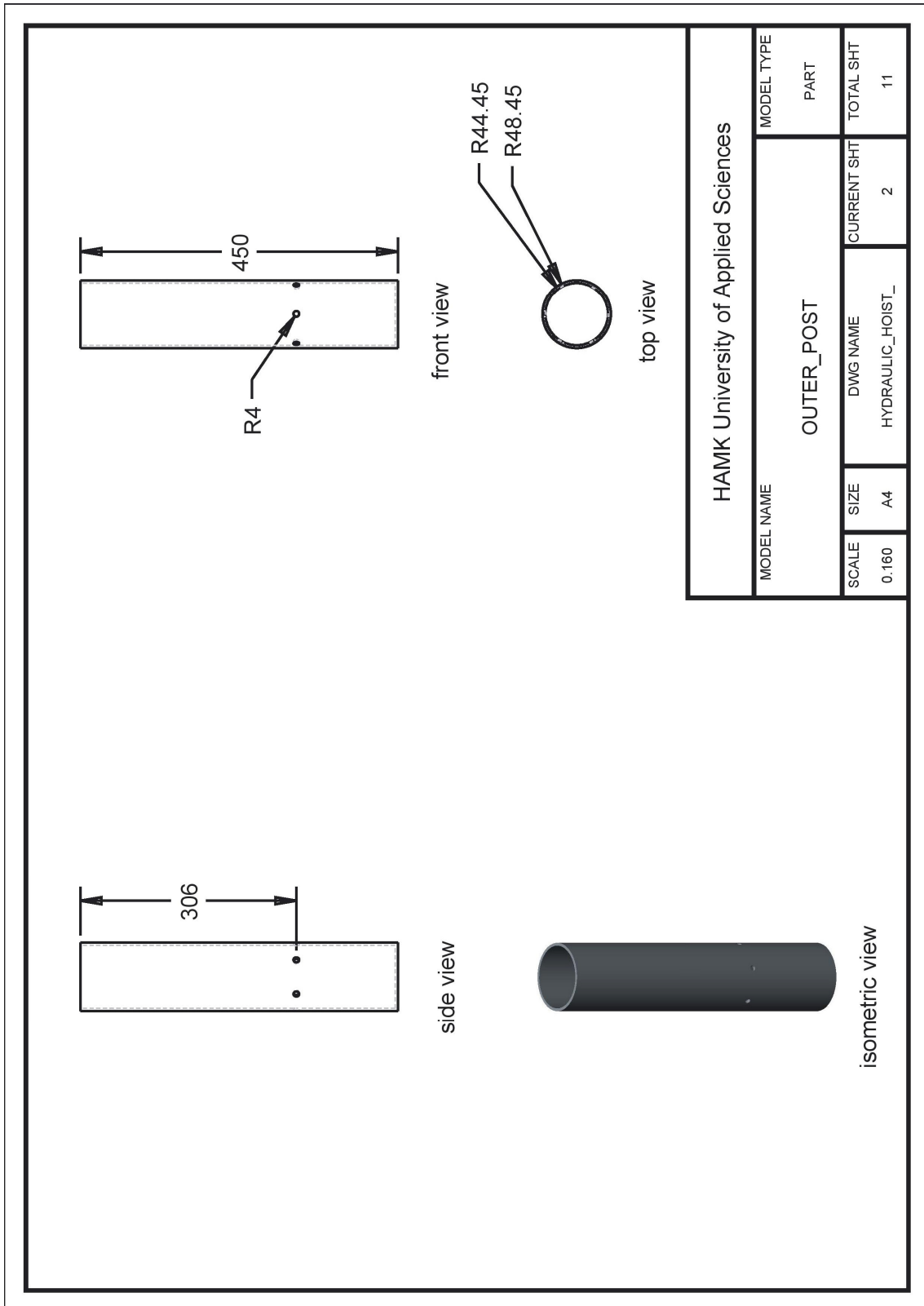
TECHNICAL DRAWING OF THE INNER BOOM

Appendix 4

TECHNICAL DRAWING OF THE OUTER BOOM

TECHNICAL DRAWING OF THE UPPER BEAM BRACKET

front view

R12

43

10

90

R4

top view

R52.45
R48.45

side view

86

66
58

isometric view

HAMK University of Applied Sciences

MODEL NAME			MODEL TYPE	
UPPER_BEAM_BRACKET			PART	
		DWG NAME	CURRENT SHT	TOTAL SHT
		HYDRAULIC_HOIST_	5	11
SCALE	SIZE			
0.400	A4			
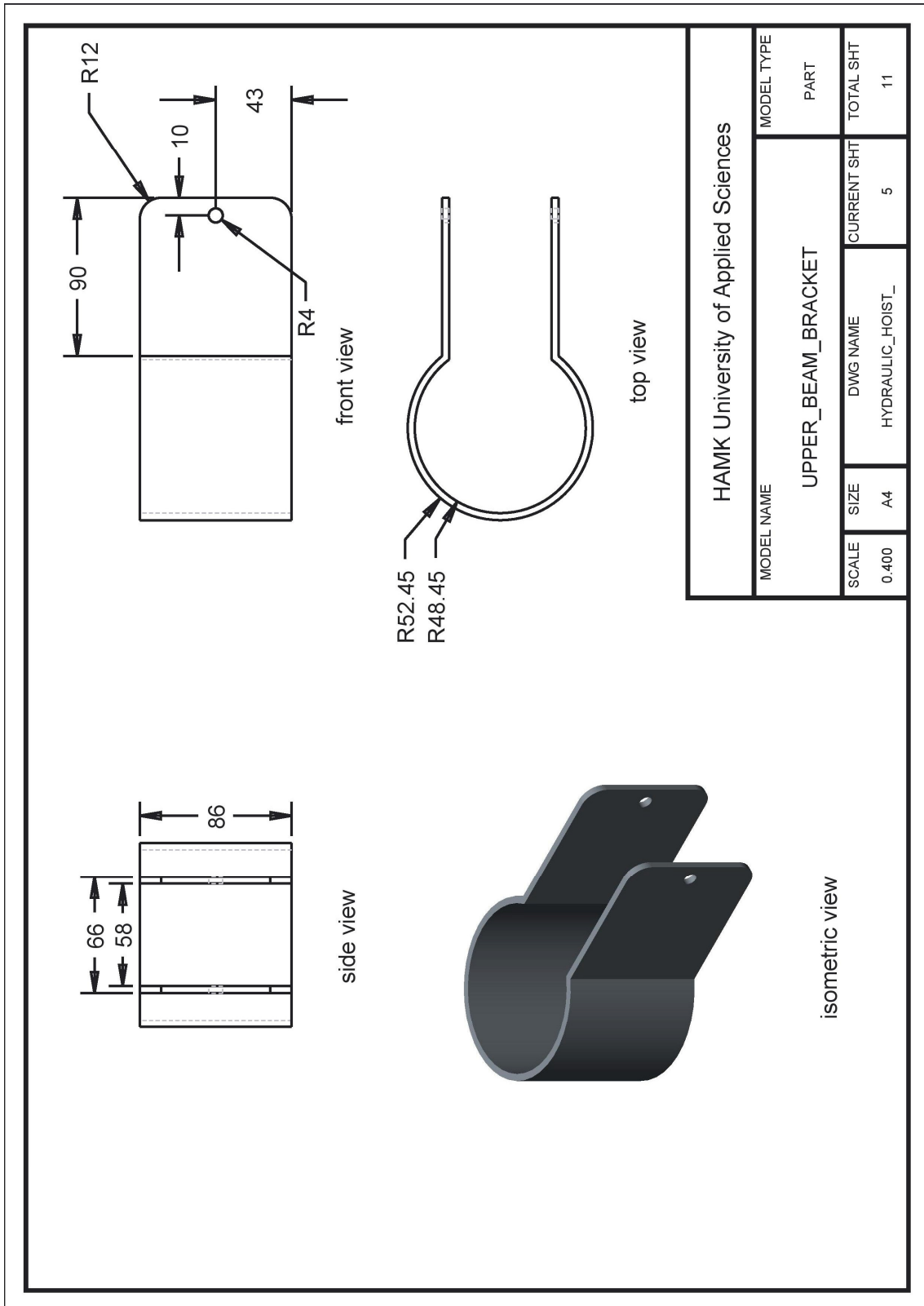
TECHNICAL DRAWING OF THELOWER BEAM BRACKET

R48.45
R52.45

R4.5

90

40

43

R12

front view

top view

86

66

58

side view

isometric view

HAMK University of Applied Sciences

MODEL NAME				MODEL TYPE	
LOWER_BEAM_BRACKET				PART	
			DWG NAME	CURRENT SHT	TOTAL SHT
SCALE	SIZE		HYDRAULIC_HOIST_	6	11
0.400	A4				

TECHNICAL DRAWING OF THE BOOM BRACKET

front view

top view

side view

isometric view

R4.5

	MODEL TYPE		
	PART		
HAMK University of Applied Sciences	CURRENT SHT	TOTAL SHT	
	7	11	
BOOM_BRACKET	DWG NAME		
	HYDRAULIC_HOIST_		
MODEL NAME	SIZE	A4	
SCALE	0.500		
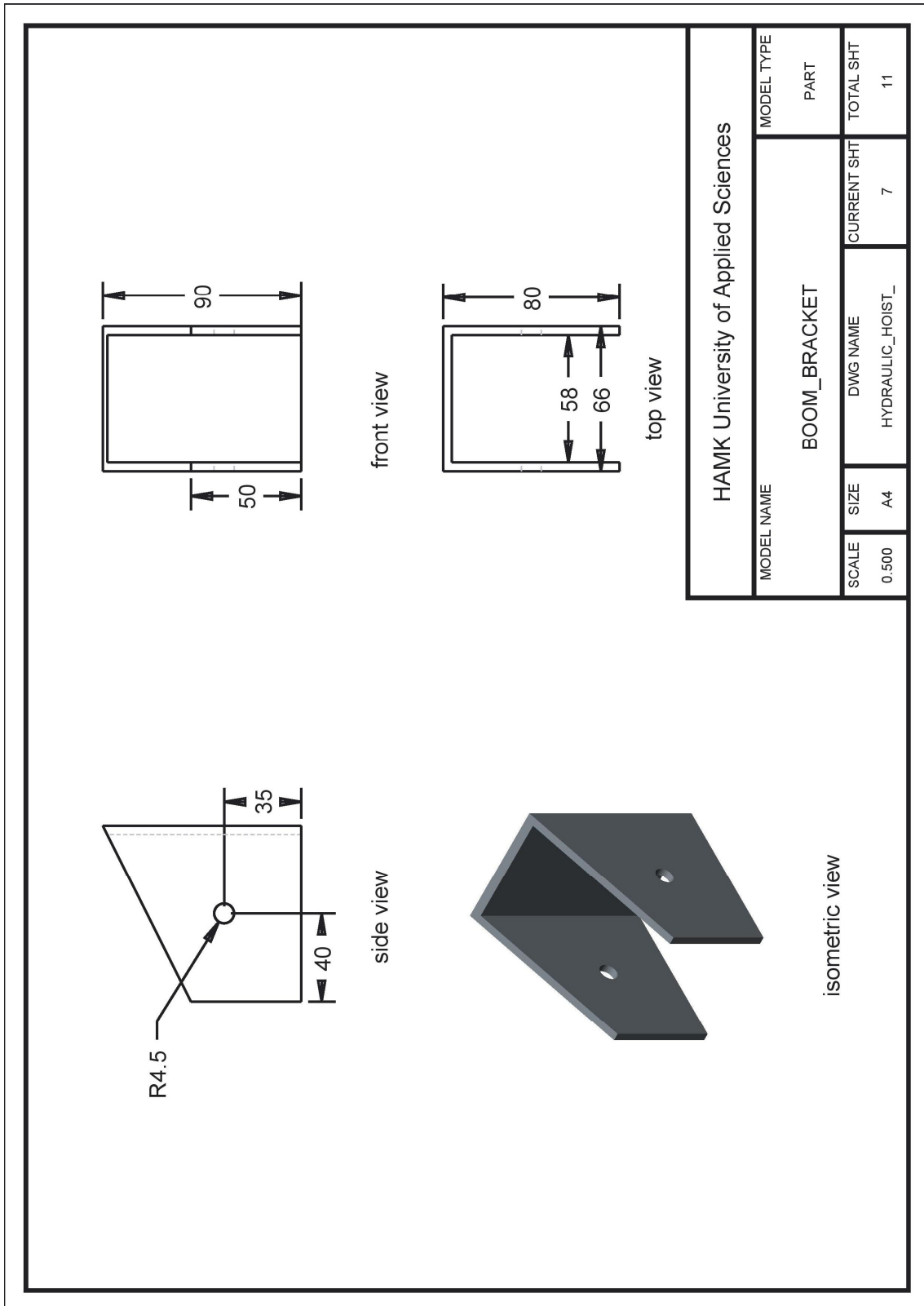
TECHNICAL DRAWING OF THE FINAL ASSEMBLY - ISOMETRIC VIEW
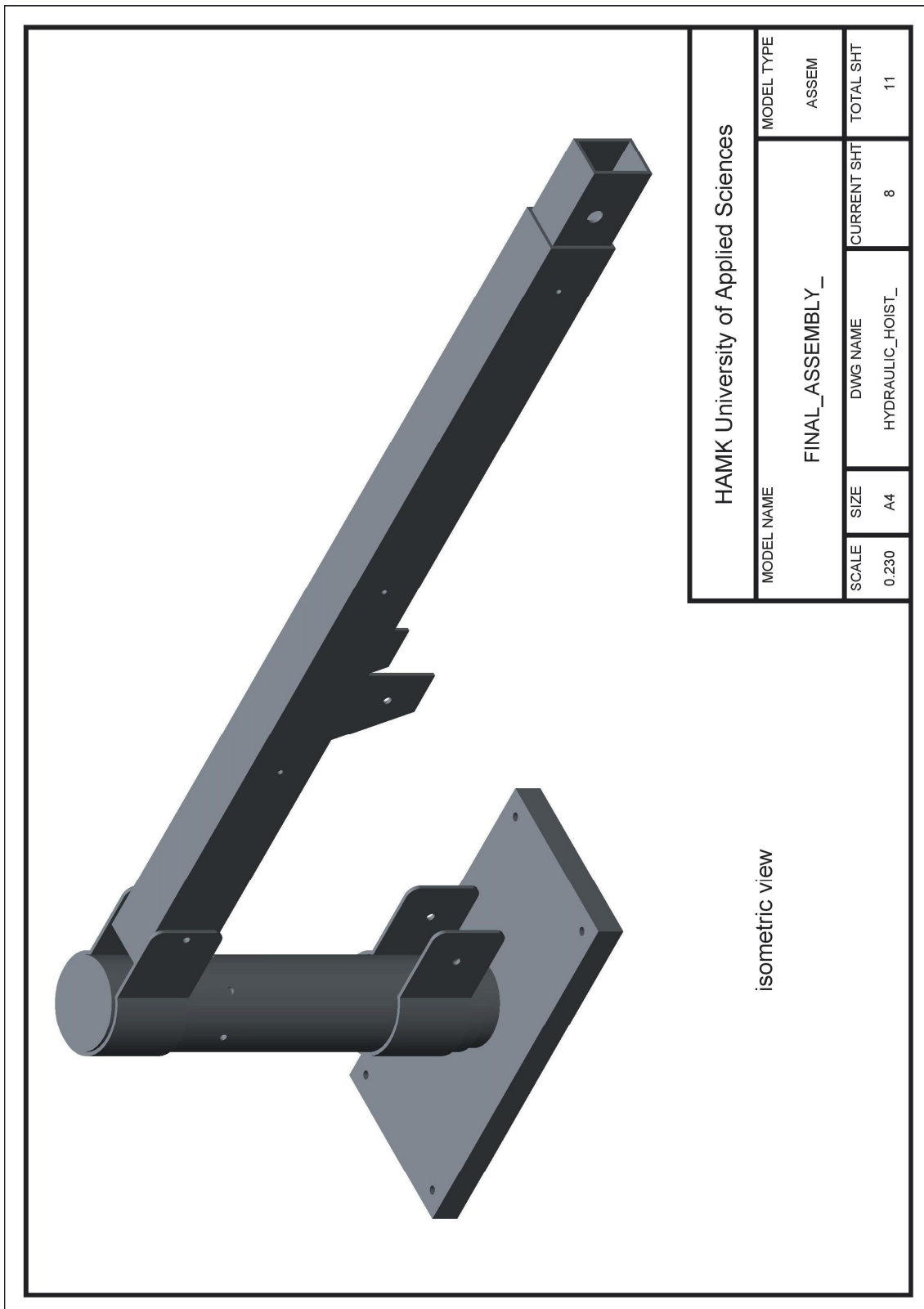
isometric view

	MODEL TYPE	
HAMK University of Applied Sciences	ASSEM	
	CURRENT SHT	TOTAL SHT
FINAL_ASSEMBLY_	8	11
MODEL NAME	DWG NAME	
	HYDRAULIC_HOIST_	
SCALE	SIZE	
0.230	A4	

TECHNICAL DRAWING OF THE FINAL ASSEMBLY - FRONT VIEW
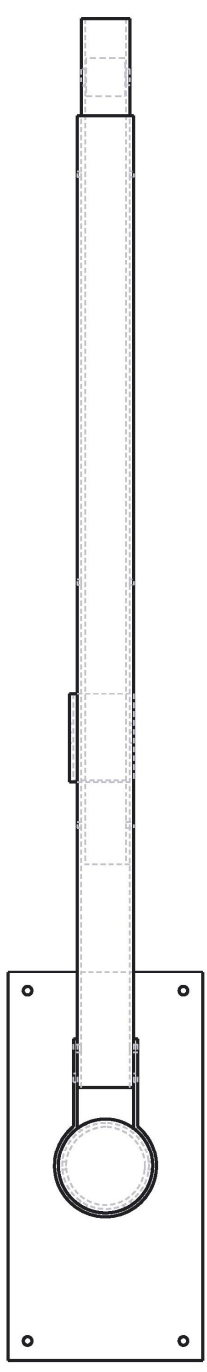
front view

MODEL NAME	HAMK University of Applied Sciences		MODEL TYPE	ASSEM
	FINAL_ASSEMBLY_		CURRENT SHT	9
		DWG NAME HYDRAULIC_HOIST_		TOTAL SHT 11
SCALE 0.180	SIZE A4			

TECHNICAL DRAWING OF THE FINAL ASSEMBLY - SIDE VIEW

side view

HAMK University of Applied Sciences

FINAL_ASSEMBLY_

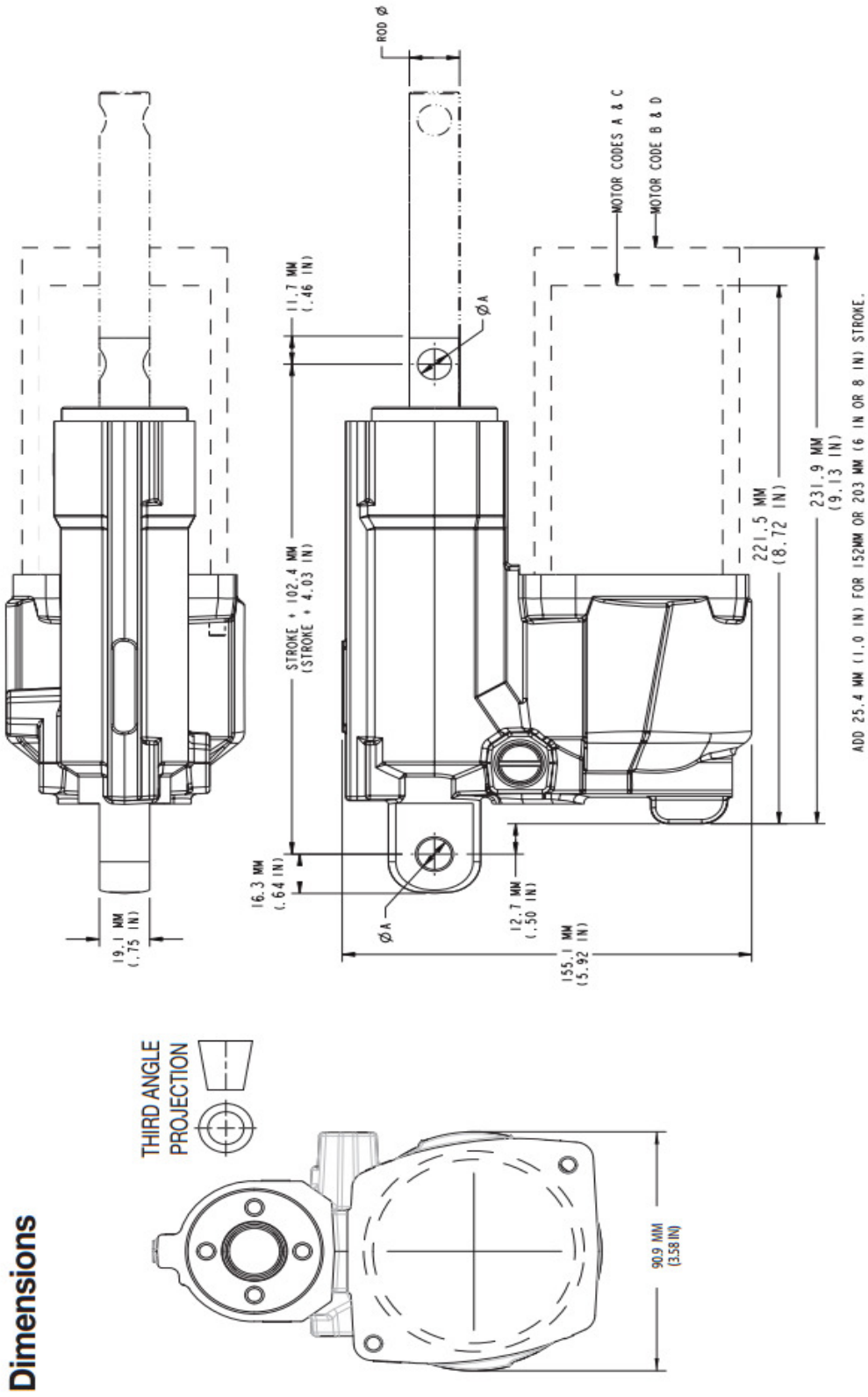| MODEL NAME			MODEL TYPE	
			ASSEM	
		DWG NAME	CURRENT SHT	TOTAL SHT
		HYDRAULIC_HOIST_	10	11
SCALE	SIZE			
0.300	A4			

TECHNICAL DRAWING OF THE FINAL ASSEMBLY - TOP VIEW

top view

HAMK University of Applied Sciences

FINAL_ASSEMBLY_

MODEL NAME				MODEL TYPE
				ASSEM
SIZE	DWG NAME		CURRENT SHT	TOTAL SHT
A4	HYDRAULIC_HOIST_		11	11
SCALE				
0.180				

TECHNICAL DRAWING OF THE ELECTRO-HYDRAULIC ACTUATOR

PERFORMANCE DATA OF THE ELECTRO-HYDRAULIC ACTUATOR

Electro-Hydraulic Actuators
Compact EHA
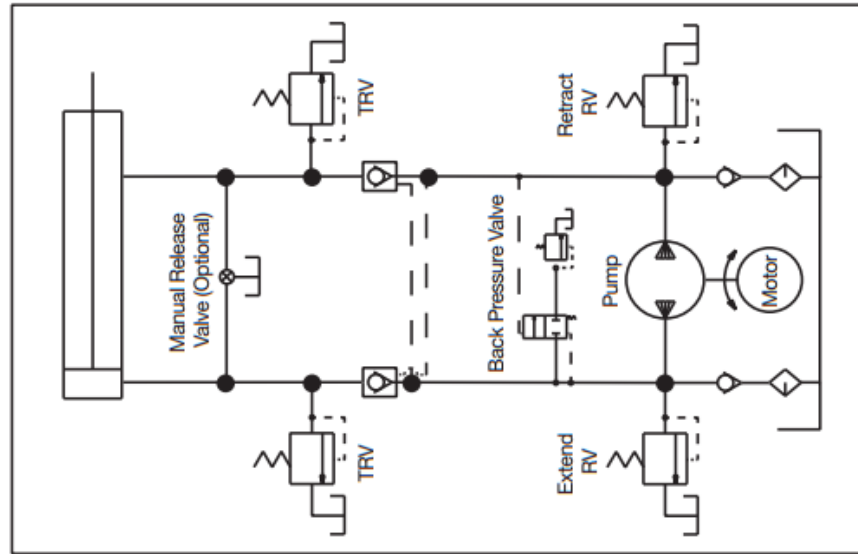
Suggested Diagram for Wiring

Ground

CR-1　CR-2

Extend　Retract

SPDT Centre Off Toggle Switch

12V DC +
24V DC +

Blue　CR-1　Motor　Green　CR-2

CR-1　CR-2
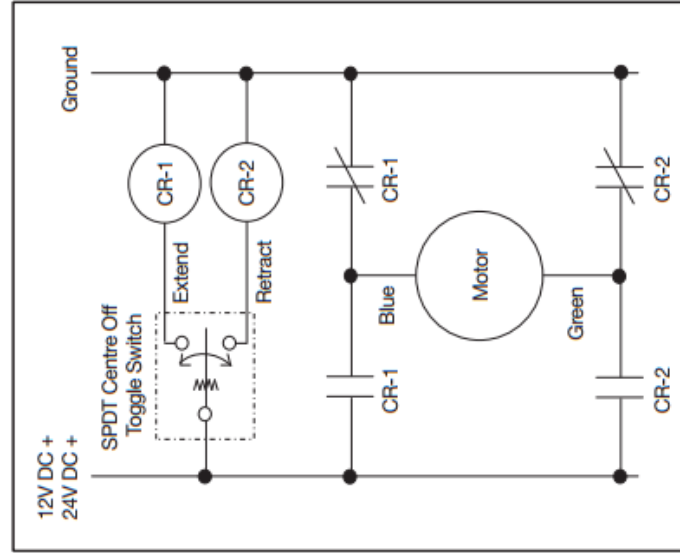
Function	Positive	Ground
Extend	Blue	Green
Retract	Green	Blue

Catalogue HY07-1310/UK
Performance Data

Hydraulic Schematic

TRV

Retract RV

Manual Release Valve (Optional)

Back Pressure Valve

Pump　Motor

TRV

Extend RV

CALCULATION OF THE BASE PLATE

Given,

b = 200 mm
h = 30 mm
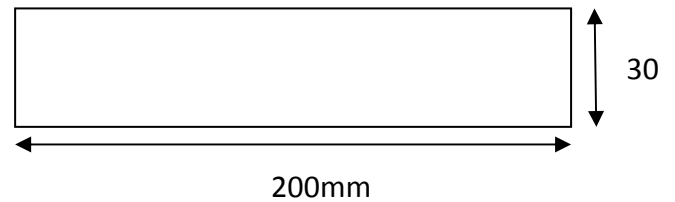
Calculating section modulus (Z) for the base plate,

$Z = bh^2/6$

$Z = 30 \times 10^3 \text{ mm}^3$

Then,

Bending Stress(σ_{max}) is calculated since the maximum bending moment occurs at the base of the beam is 5000 Nm,

$M_{bMax} = 5000 \text{ Nm} = 5000 \times 10^3 \text{ Nmm}$

We have,

$\sigma_{max} = M_{bMax}/Z$

$\sigma_{max} = 5000 \times 10^3 \text{ Nmm}/ 30 \times 10^3 \text{ mm}^3$

$\sigma_{max} = 166.67 \text{ Nmm}^2$

Now, since the material used for base plate is S355 steel, allowed stress (σ_{all}) is calculated,

$\sigma_{all} = 355/n$

$\sigma_{all} = 355/1.5$

$\sigma_{all} = 236.67 \text{ Nmm}^2$

Hence,

$\sigma_{max} < \sigma_{all}$

The designed base structure could handle the maximum bending moment since the maximum bending stress taking place at the base plate was smaller than the allowed bending stress of the material.

DETERMINATION OF BOOM LENGTHS WITH VARIABLE LOADS

Here,

In the given diagram,

At a particular condition,

Load at point C (F_1) = 5000 N

Length of BC (L_1) = 1000 mm

Moment is calculated,

$$\text{Moment (M)} = F_1 \times L_1$$

M = 5000,000Nmm

When the load is decreased, the length BC must be increased and vice versa.
 Hence when,

F_2 = 4000 N

L_2 = ?

M = $F_2 \times L_2$ (from law of moments)

$L_2 = M/F_2$

L_2 = 5000,000Nmm/4000N
    = 1250 mm

Similarly,

when F_3 = 3000N

$L_3 = M/ F_3$

L_3 = 5000000Nmm/3000N

L_3 = 1670 mm
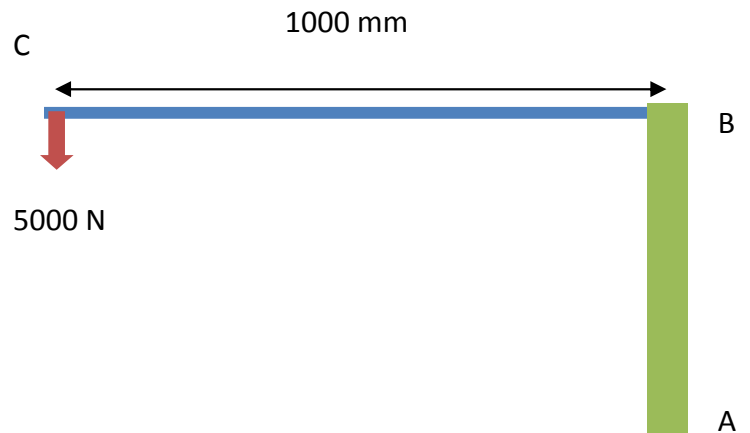
Position	Load	Length
X	5000 N	1000 mm
Y	4000 N	1250 mm
Z	3000 N	1670 mm

 Hence, the positions X, Y and Z are the points on beam BC which connects two booms to give variable lengths in order to give efficient result with respect to variable loads. Load and length relationship is illustrated clearly in the table above.