Scissor Lift Table

Engineering Design I project

SLT Catalogue | 2022 - 2023

Engineering Design I (EDPT501)

Project

Industrial Scissor Lift Table

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INTRODUCTION

A scissor-lift table (SLT) refers to a vertical lifting device that can be raised or lowered to desired height, it is widely used in industrial applications and in our case, it could be used to store pallets in a warehouse's shelves. In this project, a Hand Operated Scissor Lift Table is designed to lift pallets with a 500 kg weight. The pallets are 600x400x500 mm^3 (LxWxH).

This design is a mechanical scissor lift. The table is extended using power screw. The power screw (also called lead screw) converts rotational motion into linear motion which therefore extends the linked folding supports connected in a crisscross X pattern. This is called a scissor mechanism or pantograph. Each link of the crosses is called scissor arm or scissor member. Other types of SLT uses hydraulic or pneumatic actuators.

Some technical details of our SLT:

- Lifting Capacity 500 KG
- Color Finish Yellow
- Construction Material Steel
- Control Hand Operated Platform
- Lifting Mechanism Lead Screw
- Max. Height 1070.92 mm
- Min. Height 295.34 mm
- Platform Length x Width 650 x 450
- Scissor arm length 600 mm

The main components are:

- Platform
- Scissor arms
- Power Screw
- Base
- Wheels

Note. At each section, the weight of each part will be ignored while calculating its stresses.

PLATFORM

- ***Platform sheet***

The dimensions chosen for the sheet metal are L = 650 mm and W = 450 mm. Material used is high tensile steel casting CS1030 (S_y = 850 MPa, S_{ut} = 1030 MPa) with a safety factor of 1.3.

We will calculate the bending at the longer side since the distance between the force and the reaction is bigger.

Taking into consideration that the load is 500 kg, we will take the force F to be 5000 N.

Step I Calculation of permissible stress

$$n = \frac{S_y}{\sigma_{all}} \rightarrow \sigma_{all} = \frac{S_y}{n} = \frac{850}{1.3} \approx 653.846 \text{ MPa}$$

Step II Calculation of bending moment and axial stress

F is acting at the center. Therefore, the reactions R_1 and R_2 are 2500 N each directed upwards.

$$M_c = R_2 * \frac{L}{2} = \frac{2500 * 650}{2} = 812.5 \text{ N.m}$$

$$\sigma_m = \frac{M_c y}{I} = \frac{M_c * \frac{t}{2}}{\frac{1}{12} * w * t^3} = \frac{812.5 * 10^3 * \frac{t}{2}}{\frac{1}{12} * 400 * t^3} = \frac{12187.5}{t^2} \text{ MPa}$$

Step II Calculation of thickness

$$\sigma = \frac{F}{A} + \frac{M_c y}{I} = \frac{5000}{650 * 400} + \frac{12187.5}{t^2} = 653.846$$

$$\rightarrow t = 4.137\ mm$$

Conclusion According to standards, we will take 5 mm as the thickness of the platform sheet.

- ***Platform frame***

Using sheet metal calculator (ρ = 7800 kg/m^3), the weight of the sheet is 11.4075 kg approximating it to 100 N. The platform frame will therefore experience 5100 N load.

The platform frame is divided into 4 parts, each part is a cold formed section. The material used is ASTM 572 Grade 42 steel (S_y = 290 MPa, S_{ut} = 415 MPa)

We will calculate the stresses at the longest part (L = 650 mm). The force applied at that part is $\frac{5100}{2}$ = 2550 N. Taking the force at the center, the reactions at the corners are therefore 1275 N each.

Step I Calculation of permissible stress

$$\mathrm{n} = \frac{\mathrm{S_y}}{\sigma_{\mathrm{all}}} \rightarrow \sigma_{\mathrm{all}} = \frac{\mathrm{S_y}}{\mathrm{n}} = \frac{290}{1.3} \approx 223.077 \text{ MPa}$$

Step II Calculation of bending moment

The axial stress is negligeable in comparison to bending stress.

$$M_c = R * \frac{L}{2} = \frac{1275 * 650}{2} = 414.375 \; N.m$$

$$\sigma_m = \frac{M_c y}{I} = \frac{M_c * \frac{H}{2}}{I} = \frac{414375 * \frac{H}{2}}{I}$$

Step III Calculation of area moment of inertia

We will take a random value of H to get an approximated value of I. Let H = 50 mm

$$\frac{414375 * \frac{H}{2}}{I} = 223.077 \; MPa \rightarrow I = \frac{414375 * \frac{50}{2}}{223.077} \approx 46438.6 \; mm^4$$

Checking the cold formed steel section tables, the nearest value is $I = 5.74 \; cm^4 = 5.74 * 10^4 \; mm^4$ for a standard cross section of: $H = 30, b = 25, t = 4$ and $r = 8$.

Step IV Recalculating bending moment

$$\sigma_m = \frac{M_c y}{I} = \frac{M_c * \frac{H}{2}}{I} = \frac{414375 * \frac{30}{2}}{57400} \approx 108.29 \; MPa$$

Step V Calculation of new safety factor

$$\mathrm{n} = \frac{\mathrm{S_y}}{\sigma_{\mathrm{m}}} = \frac{290}{108.29} \approx 2.678$$

Alternatively, we can calculate the new maximum load it can withstand with the old safety factor

$$\frac{M_c * \frac{50}{2}}{57400} = 223.077 \; MPa \rightarrow M_c \approx 512185 \; N.mm$$

$$\rightarrow R \approx 1575.95 \; N \rightarrow F \approx 3151.9 \; N$$

Therefore, in case of 1.3 safety factor, the platform can withstand up to 6303.81 N load (if we excluded the 100 N weight of the sheet, it can withstand up to a little more than 620 kg).

Conclusion According to standards, we will take $H = 30, b = 25, t = 4$ and $r = 8$ as the dimensions of the cold formed section of the platform.

Note *The sheet is finished with a treaded surface.*

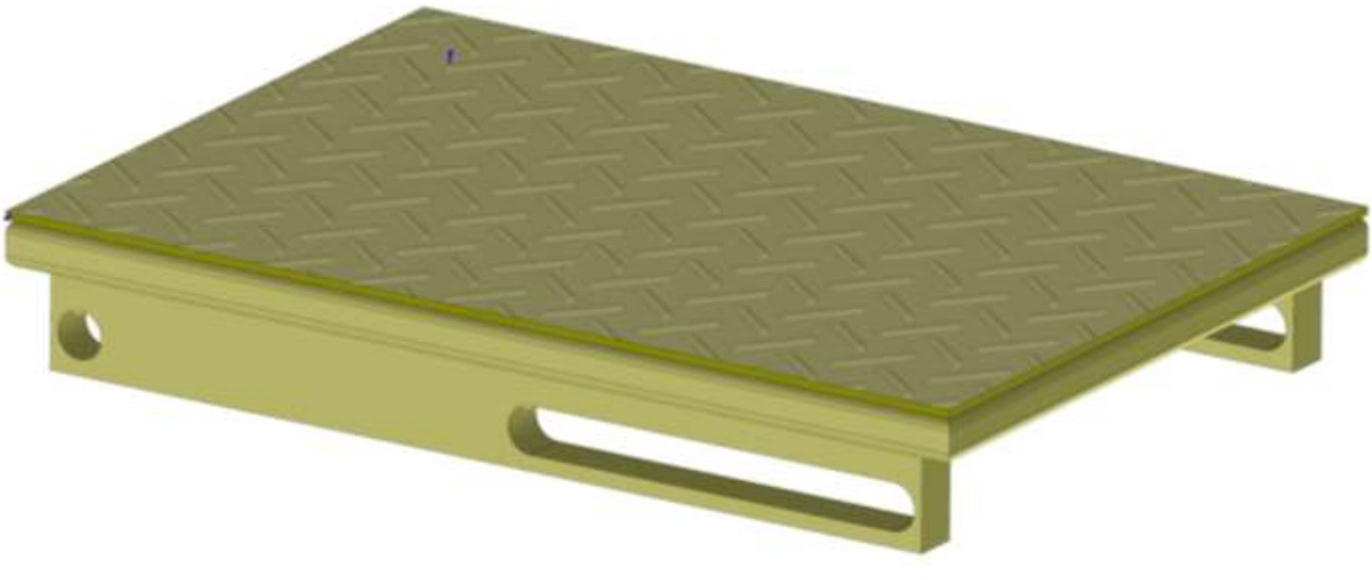


Figure 1 Top platform 3D Model

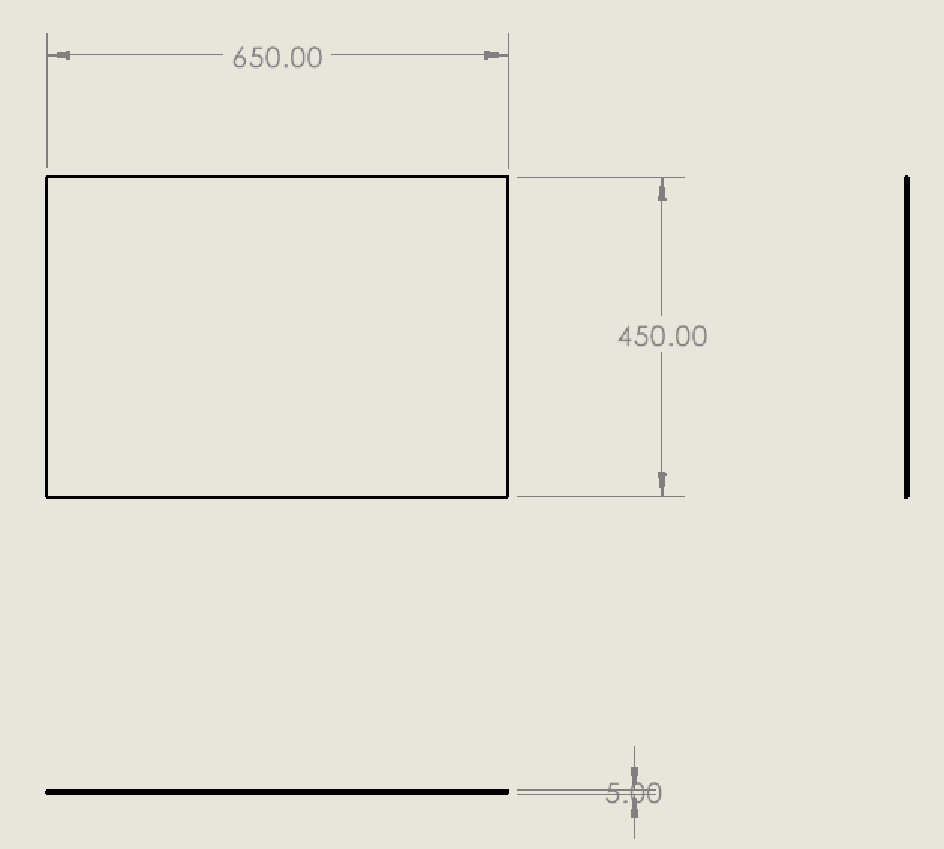


Figure 2 *Sheet 2D Drawing*

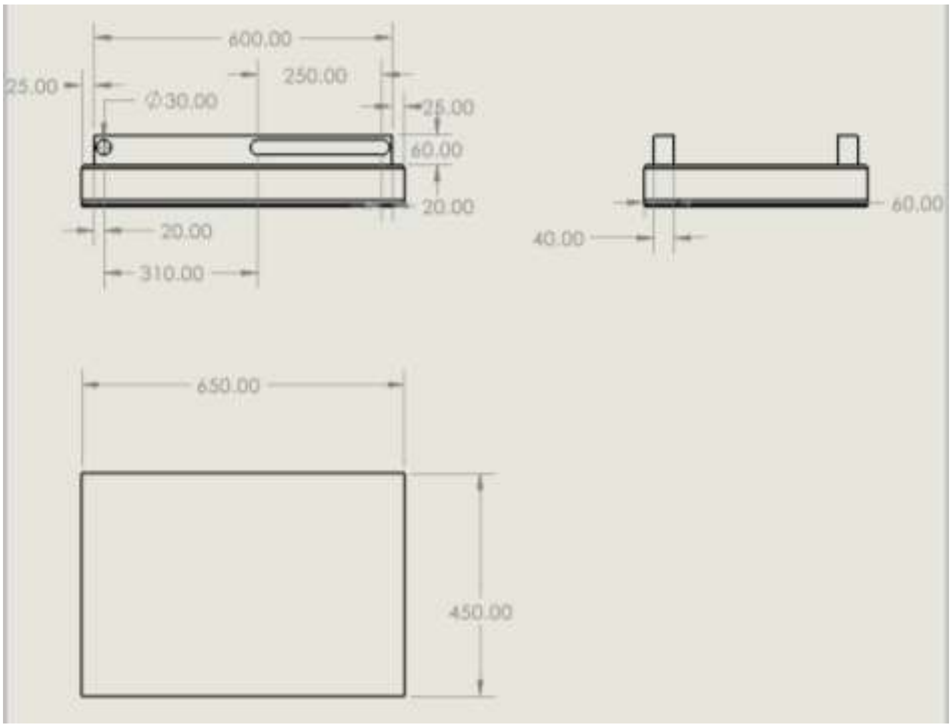


Figure 3 *Platform frame 2D drawing*

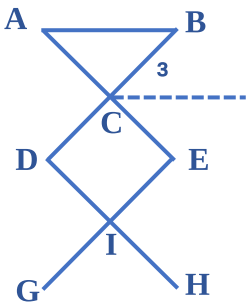
SCISSORS

The distance between each pin of the scissor arm is 280 mm:

$$AC = BC = DC = CE = DI = EI = IG = HI = 280\ mm$$

The load applied is the previous weight in addition to 100 N ($3.99 * (650 * 2 + 400 * 2 - 50 * 4) = 7.581\ kg$ rounded to 100 N) to account for the platform base divided by two:

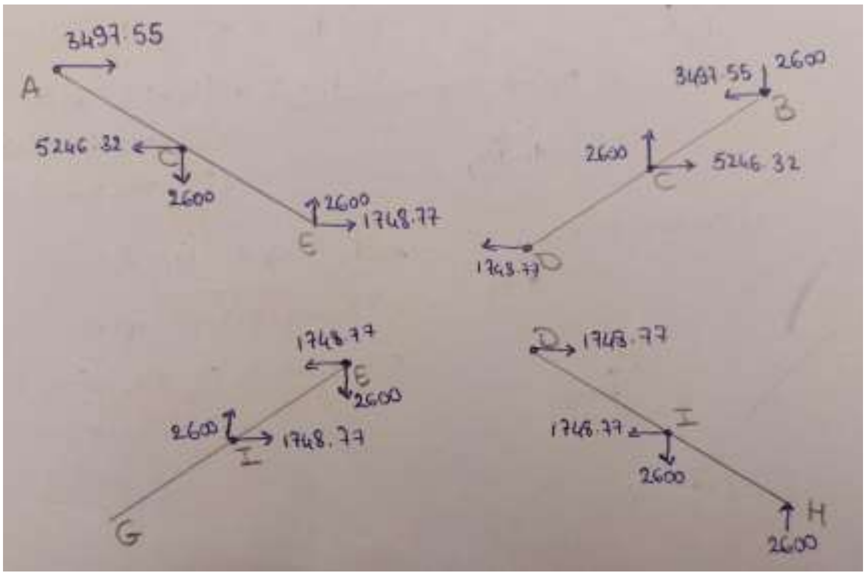
$$F = \frac{5100+100}{2} = 2600\ N$$



- ***Force Analysis of the chassis at maximum height***

In case of fully extended: $\theta = 56.075°, AB = 312.55\ mm.$

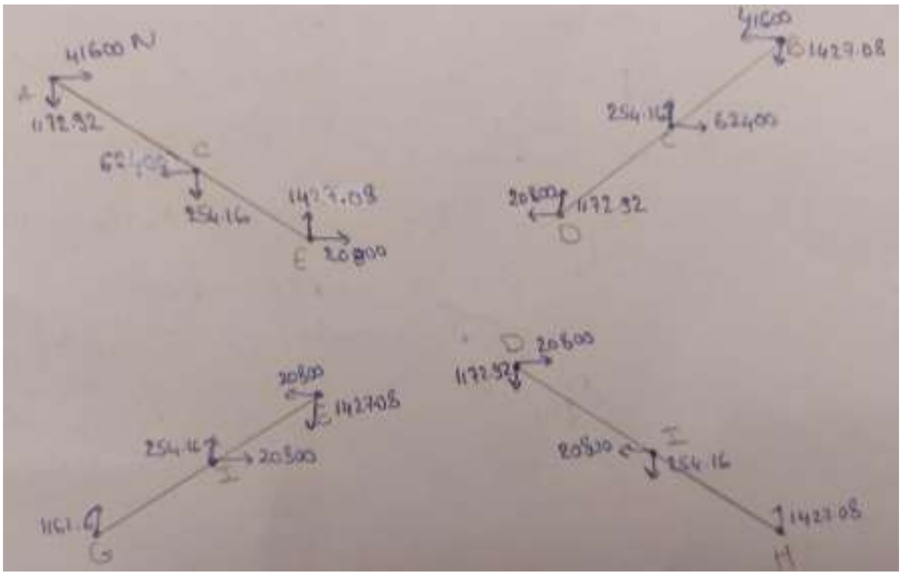
In this case the weight is applied directly on B (critical case).



- ***Force Analysis of the chassis at minimum height***

In case of fully compressed: $\theta = 7.125°, AB = 555.68\ mm.$

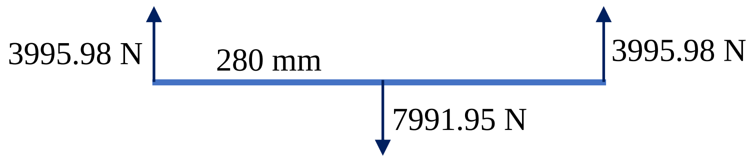
In this case the weight is applied at a distance $300 - 49.32 = 250.68\ mm$ from B and a distance of $305\ mm$ from A.



- ***Stress Analysis of the most critical member***

According to calculations, the most critical member is ACE at minimum height.

After projecting the forces parallel and perpendicular to the axis of the member



$$M = F * d = 3995.98 * 280 = 1118.8744\ N.m$$

$$\sigma_m = \frac{My}{I} = \frac{M * \frac{H}{2}}{I} = \frac{1118874.4 * \frac{H}{2}}{I}$$

Taking a cross section of the following standard dimensions

$$H = 40, b = 24, t = 4, r = 8 \text{ and } I_x = 116800\ mm^4$$

$$\sigma_m = \frac{1118874.4 * \frac{40}{2}}{116800} \approx 191.59$$

$$\mathrm{n} = \frac{\mathrm{S_y}}{\mathrm{\sigma_m}} = \frac{290}{191.59} \approx 1.514$$

Conclusion According to standards, we will take $H = 40, b = 24, t = 4$ and $r = 8$ as the dimensions of the cold formed section of the scissors.

Calculation of scissors' weight

$$W = 4.49\ kg/m$$

$$W_{scissors} = 4.49 * 0.6 * 8 = 21.552\ kg$$

$$W_{support} = 4.49 * 2 * (0.282 + 0.198) = 3.42138$$

$$W_{scissor\ mecha} = 21.552 + 3.42138 = 24.97338\ kg \approx 250\ N$$

The total weight so far is $5200 + 250 = 5450\ N$

Note *Please refer to attached documents for detailed calculations.*

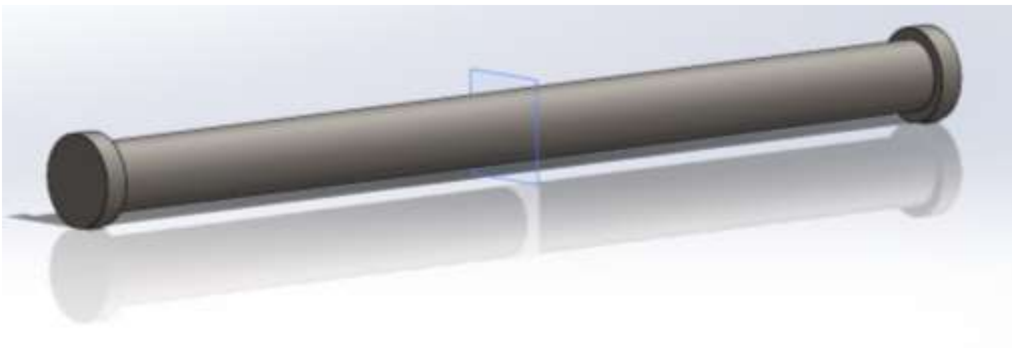


Figure 4 *Bottom tube connecting inner and outer scissor members*

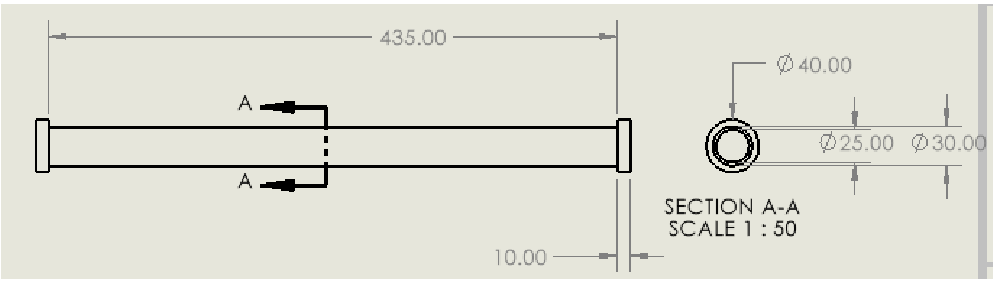


Figure 5 *Bottom tube 2D drawing*

Figure 7 *Inner Scissor arms 3D model*

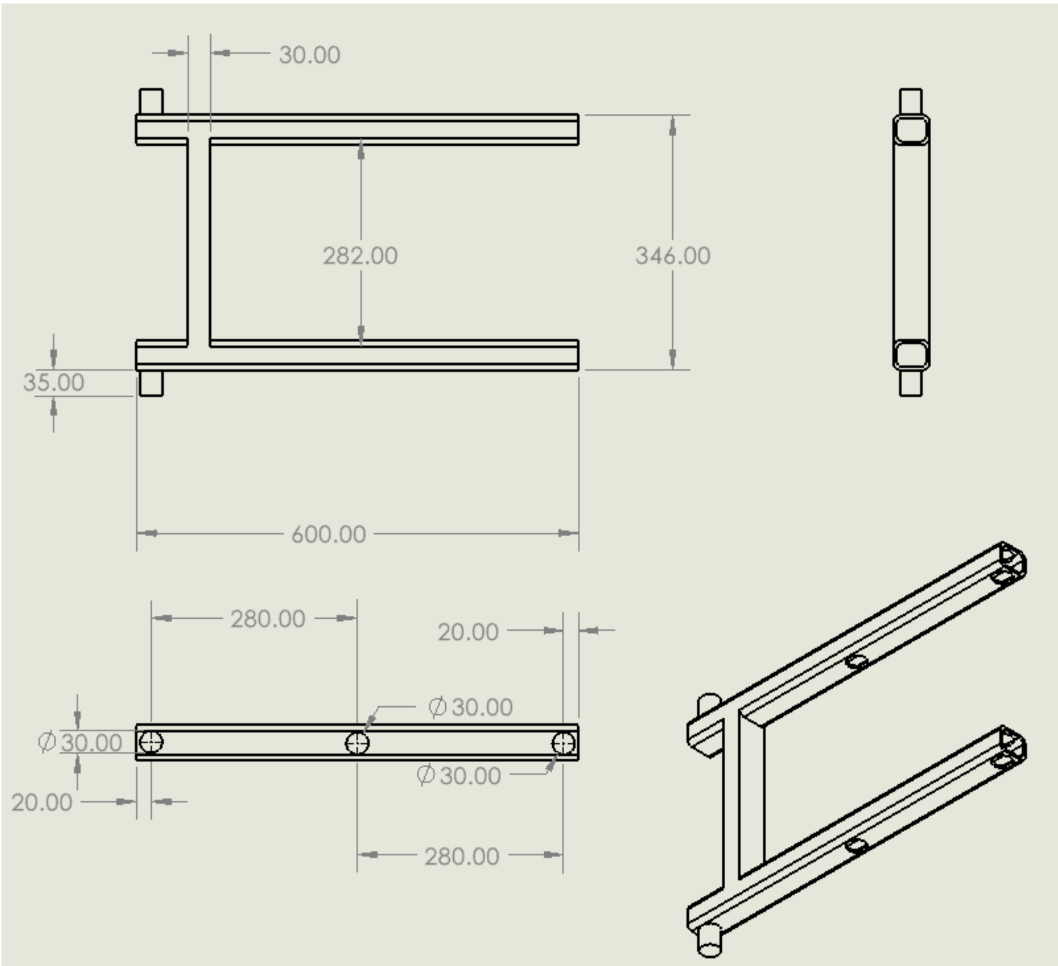


Figure 6 *Inner Scissor arms 2D drawing*

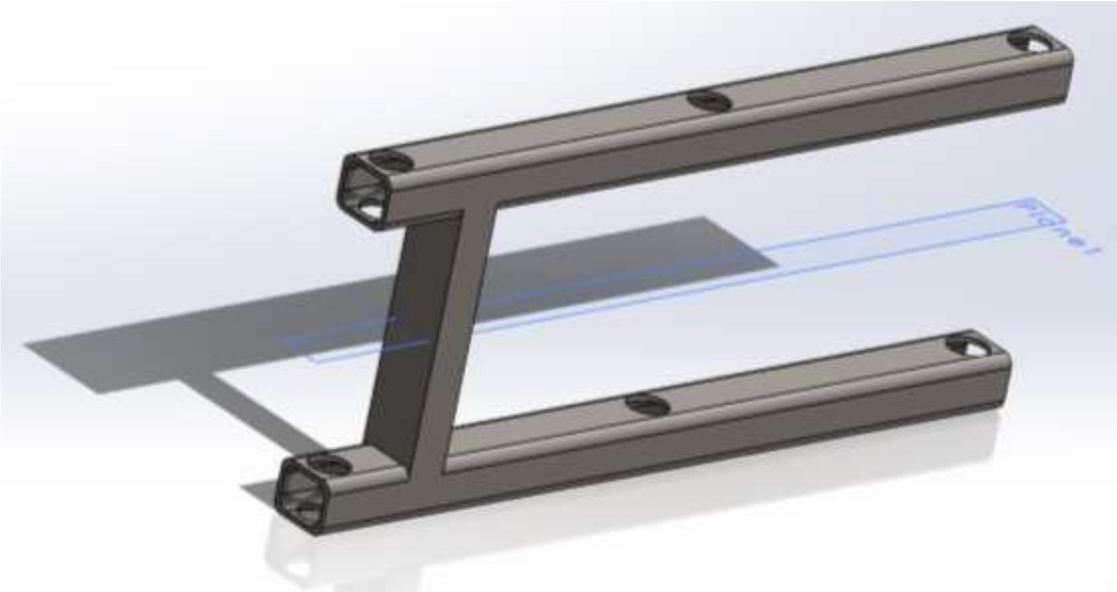


Figure 9 Outer Scissor arms 3D model

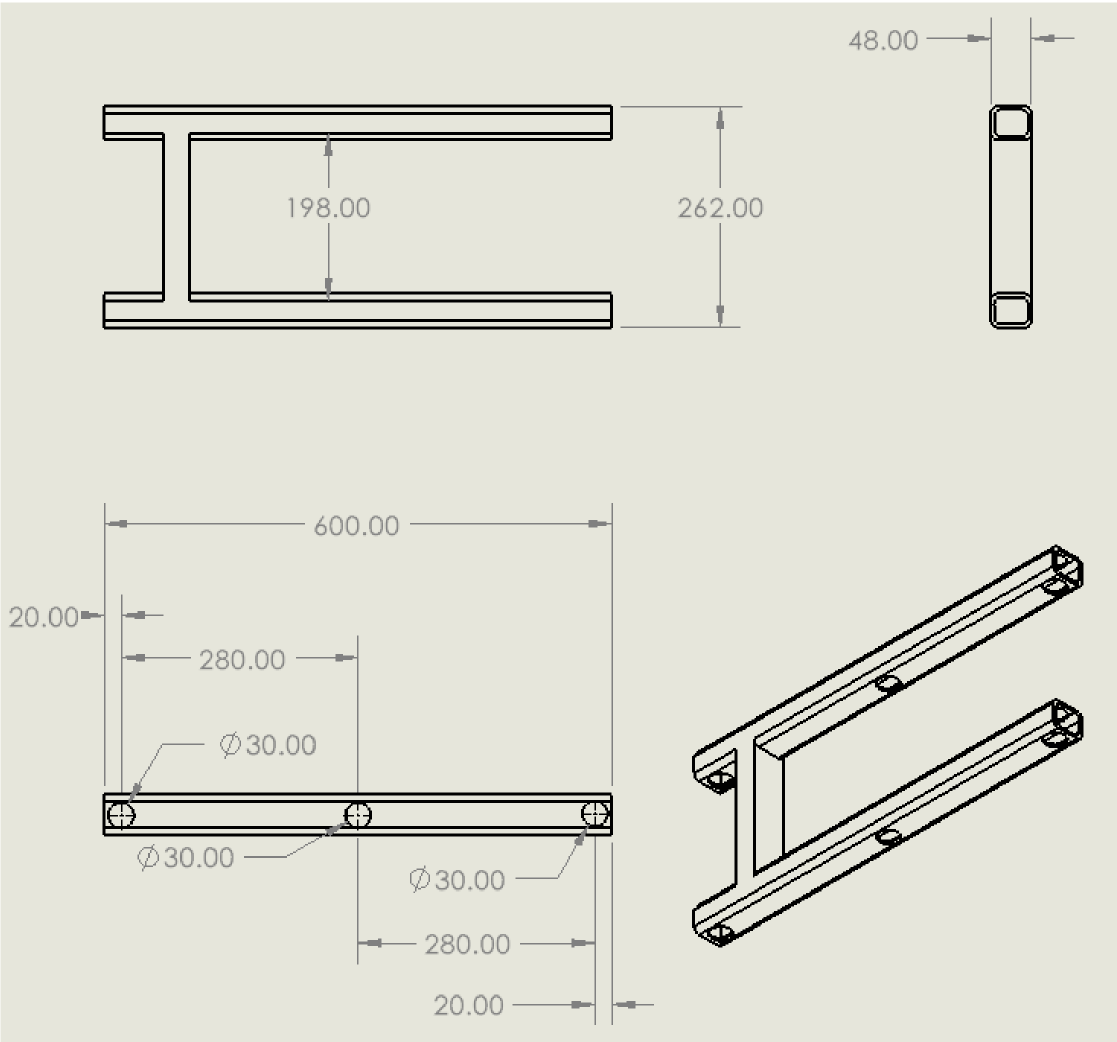


Figure 8 Outer Scissor arms 2D drawing

LEAD SCREW

The material used for the power screw is structural steel with $S_y = 220$ MPa.

Taking the force to be 5500 N and coefficient of friction to be 0.14 (assuming the nut is also steel).

- ***22x5 Metric Square thread***

$$d_m = d - \frac{p}{2} = 22 - \frac{5}{2} = 19.5\ mm$$

$$d_r = d - p = 22 - 5 = 17\ mm$$

$$l = np = 1 * 5 = 5\ mm$$

Step I Calculating the torque required to turn the screw against the load

$$T_R = \frac{F * d_m}{2} * \frac{l + \pi f d_m}{\pi d_m - fl} = \frac{5500 * 19.5 * (5 + \pi * 0.14 * 19.5)}{2 * (\pi * 19.5 - 0.14 * 5)} \approx 12.022\ N.m$$

The maximum wrist flexion torque is 11.9 ± 2.9 Nm in the supination position.[1] Therefore, the human wrist could apply this torque to raise the load.

Step II Calculating the load-lowering torque

$$T_L = \frac{F * d_m}{2} * \frac{\pi f d_m - l}{\pi d_m + fl} = \frac{5500 * 19.5 * (\pi * 0.14 * 19.5 - 5)}{2 * (\pi * 19.5 + 0.14 * 5)} \approx 3.095\ N.m$$

Step III Calculating the shear stress τ due to torsional moment T_R at the outside of the screw body

$$\tau = \frac{16T_r}{\pi d_r^3} = \frac{16 * 11.803 * 10^3}{\pi * 17^3} \approx 12.462\ MPa$$

Step IV Calculating the axial normal stress σ

$$\sigma = -\frac{4F}{\pi d_r^2} = -\frac{4 * 5500}{\pi * 17^2} \approx -24.23\ MPa$$

Step V Calculating the thread root bending stress σ_b with one thread carrying $0.38F$

$$\sigma_b = -\frac{6 * 0.38F}{\pi d_r * p} = -\frac{6 * 0.38 * 5400}{\pi * 17 * 5} \approx 46.96\ MPa$$

Step V Calculating von Misses effective stress σ_{eff}

$$\sigma_{eff} = \frac{1}{\sqrt{2}}\sqrt{(\sigma_x - \sigma_y)^2 + (\sigma_y - \sigma_z)^2 + (\sigma_z - \sigma_x)^2 + 6(\tau_{xy}^2 + \tau_{yz}^2 + \tau_{zx}^2)}$$

$$\sigma_{eff} = \frac{1}{\sqrt{2}}\sqrt{46.96^2 + 24.23^2 + (-24.23 - 46.95)^2 + 6(12.462)^2}$$

[1] Measurement of wrist flexion and extension torques in different forearm positions - PMC (nih.gov)

$$\sigma_{eff} \approx 66.3\ MPa$$

Step V Calculation of safety factor

$$n = \frac{S_y}{\sigma_m} = \frac{220}{66.3} \approx 3.32$$

The power screw is therefore safe

Conclusion The lead screw used is 22x5 Metric square thread (single thread).

Figure 10 *Power screw 3D model*

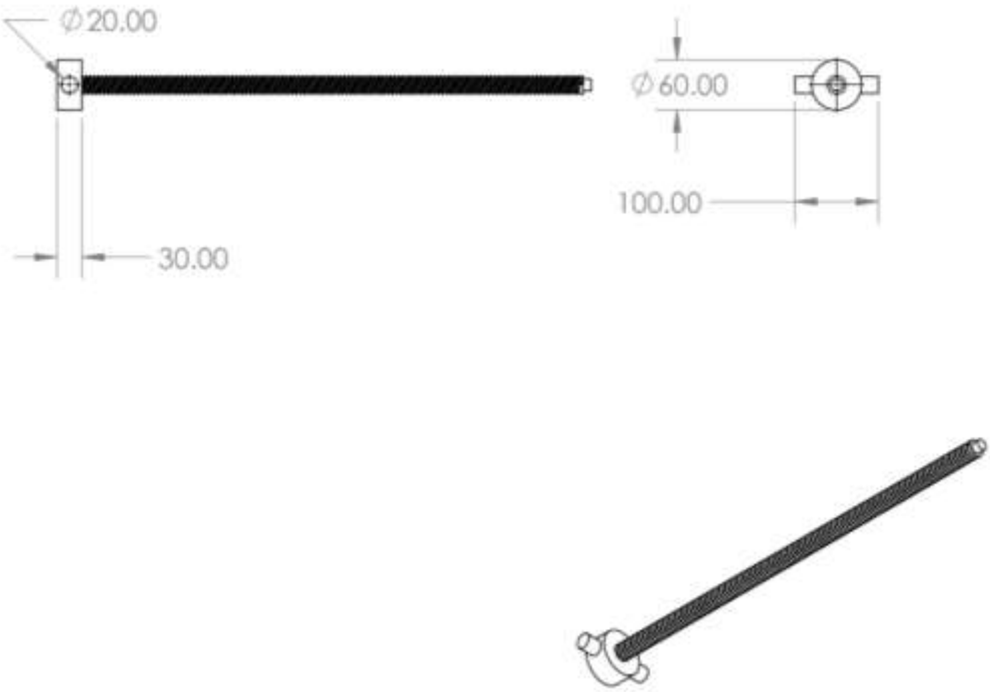


Figure 11 *Power screw 2D drawing*

For the screw support, one side is threaded to control the motion of the power screw while the other is not threaded.

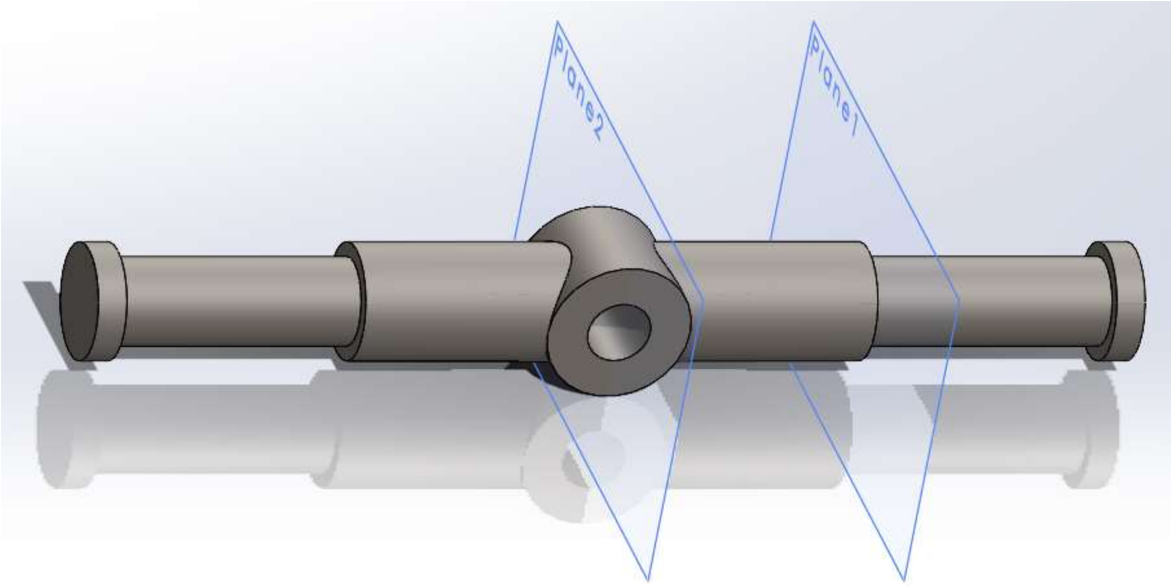


Figure 12 *Screw Support 3D model*

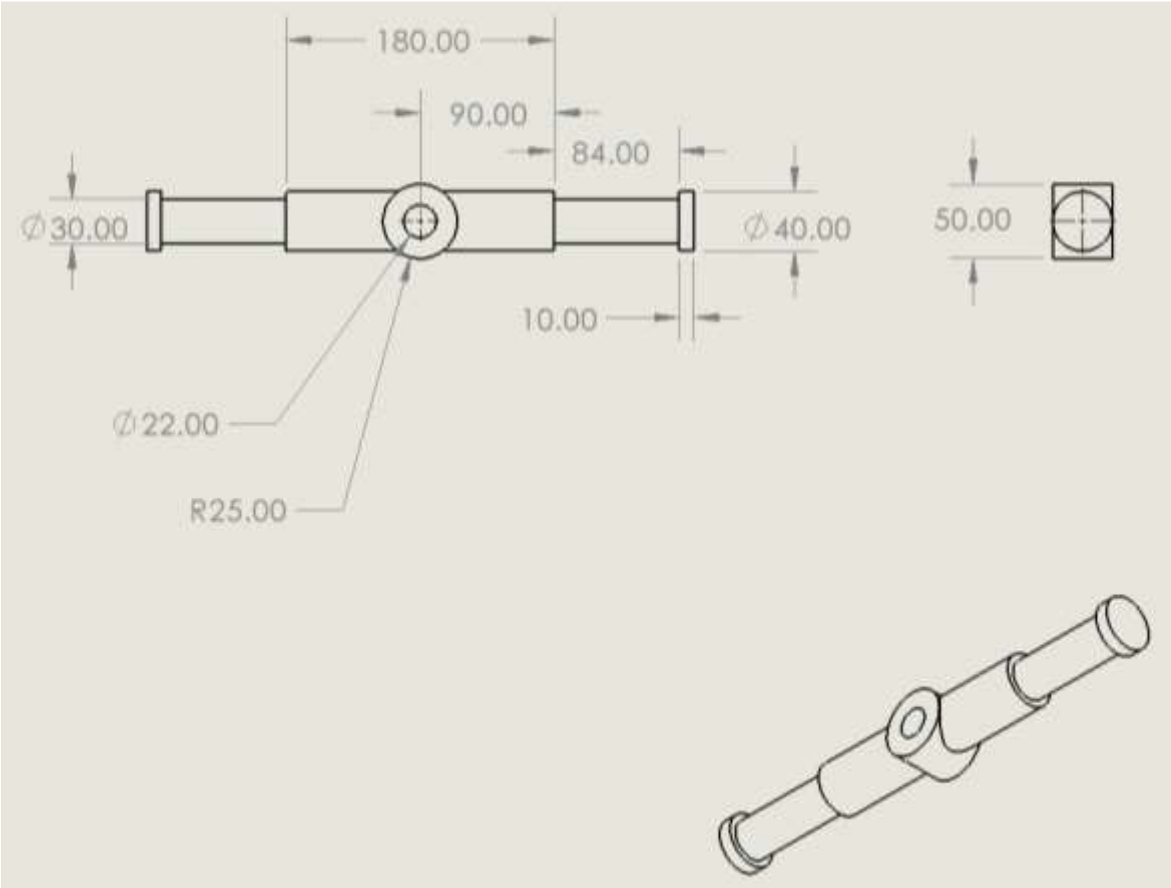


Figure 13 *Screw Support 2D drawing*

For the screw support, one side is threaded to control the motion of the power screw while the other is not threaded.

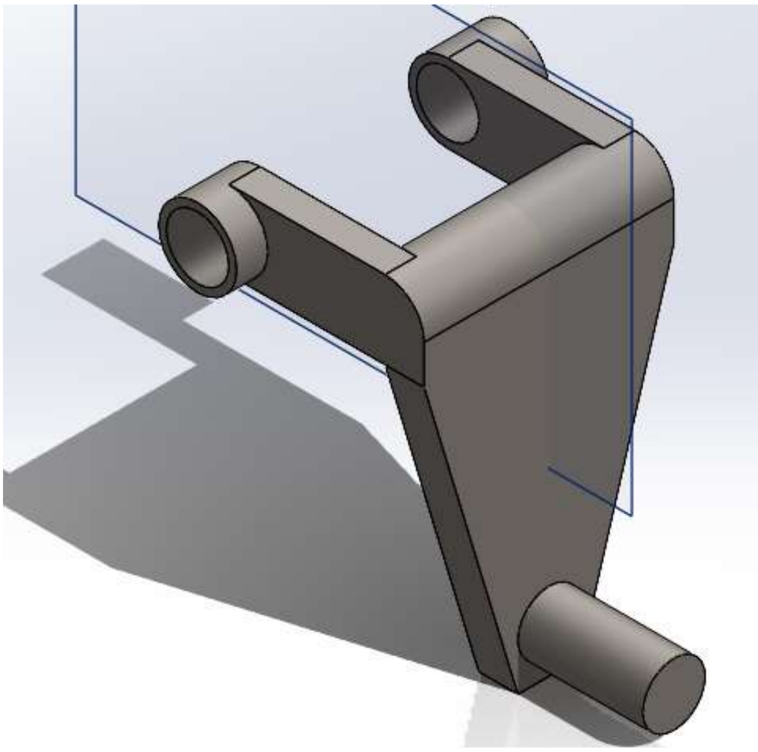


***Figure 15** Handle 3D model*

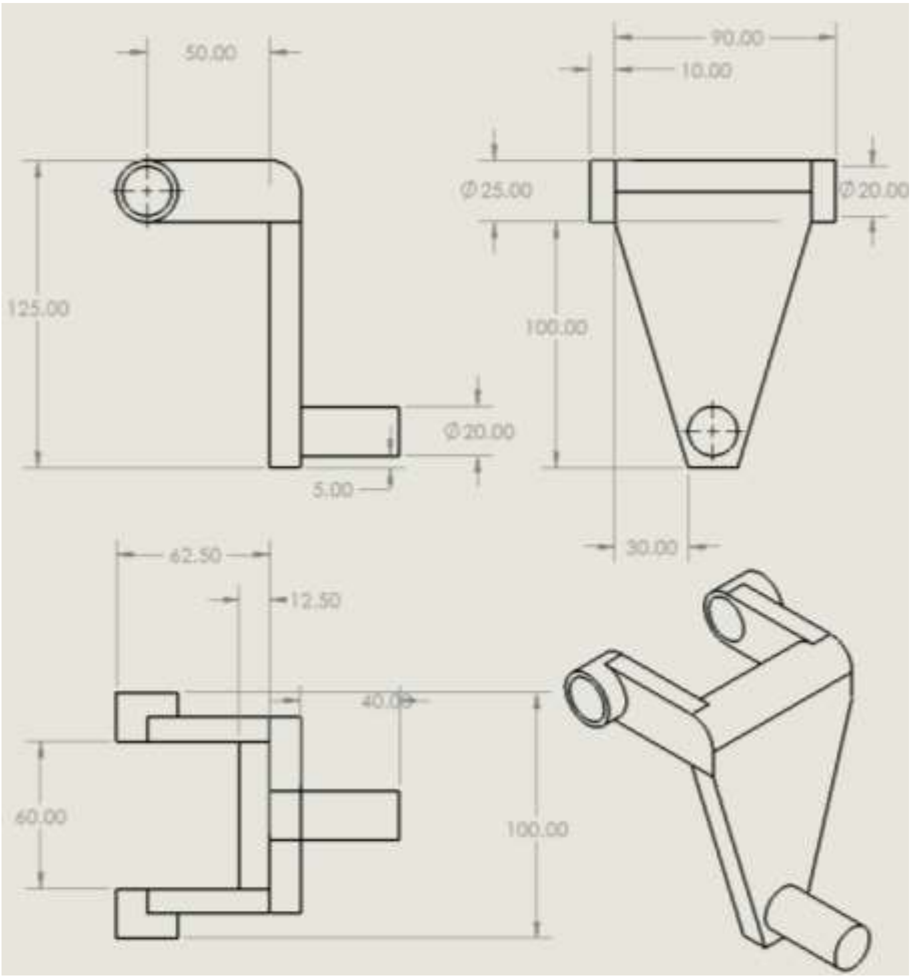


***Figure 14** Handle 2D drawing*

PINS

The force applied on the most critical pin (pin C) is $\sqrt{62400^2 + 254.16^2} \approx 62400\ N$

- ***8.8 Grade Carbon Steel***

$$S_y = 640\ MPa, S_{sy} = \frac{640}{2} = 320\ MPa$$

Calculating shear stress and safety factor for d = 6 mm

$$A = \frac{\pi}{4} d^2 = \frac{\pi}{4} * 6^2 = 9\pi$$

$$\tau = \frac{F}{A} = \frac{62400}{9\pi} \approx 2206\ MPa$$

$$n = \frac{S_{sy}}{\tau} = \frac{320}{2206} \approx 0.145 \text{ Rejected}$$

Calculating shear stress and safety factor for d = 16 mm

$$A = \frac{\pi}{4} d^2 = \frac{\pi}{4} * 16^2 = 64\pi$$

$$\tau = \frac{F}{A} = \frac{62400}{25\pi} \approx 310.352\ MPa$$

$$n = \frac{S_{sy}}{\tau} = \frac{320}{310.352} \approx 1.031 \text{ Rejected}$$

Calculating shear stress and safety factor for d = 30 mm

$$A = \frac{\pi}{4} d^2 = \frac{\pi}{4} * 30^2 = 225\pi$$

$$\tau = \frac{F}{A} = \frac{62400}{25\pi} \approx 88.278\ MPa$$

$$n = \frac{S_{sy}}{\tau} = \frac{320}{88.278} \approx 3.625$$

Conclusion According to standards, we will take 30 mm as diameter of the pin.

Note a snap ring will be used to retain the pins in their places.

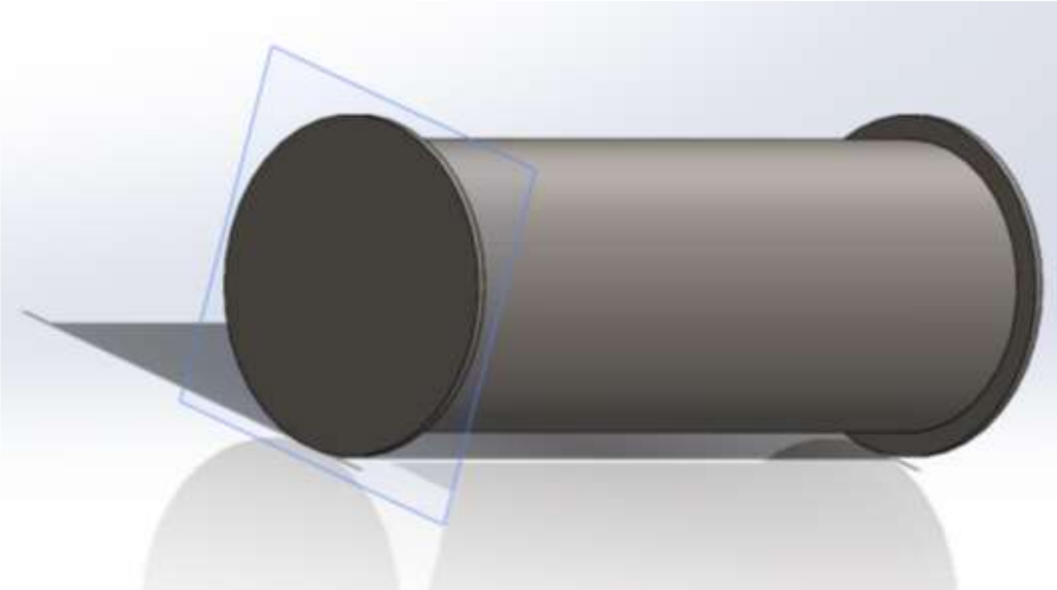


Figure 16 *Pin 3D model*

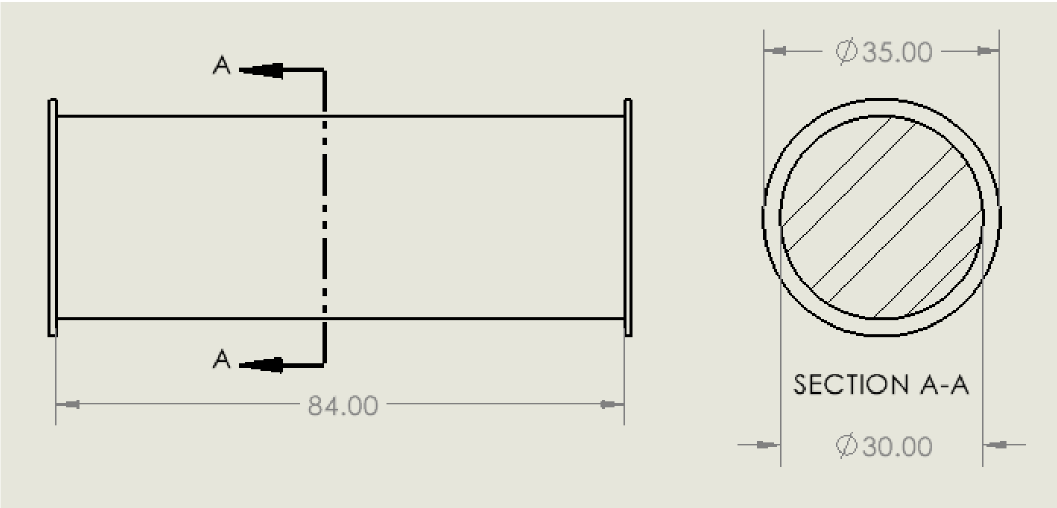


Figure 17 *Pin 2D drawing*

BASE FRAME

The material used is ASTM 572 Grade 42 steel (S_y = 290 MPa, S_{ut} = 415 MPa)

We will calculate the stresses at the longest part (L = 650 mm). The force applied at that part is $\frac{5500}{2}$ = 2750 N (adding another 50 kN to account for pins and power screw). Taking the force at the center, the reactions at the corners are therefore 1375 N each.

For a standard cross section of: $H = 30, b = 25, t = 4, r = 8$ and $I = 57400\ mm^4$.

Step I Calculation of bending moment

$$M = F * d = 1375 * 325 = 446875\ N.mm$$

$$\sigma_m = \frac{M_c y}{I} = \frac{M_c * \frac{H}{2}}{I} = \frac{446875 * \frac{30}{2}}{57400} = 116.78\ MPa$$

Step II Calculation of safety factor

$$\mathrm{n} = \frac{\mathrm{S_y}}{\sigma_{\mathrm{all}}} = \frac{290}{116.78} \approx 2.483\ \mathrm{MPa}$$

Step III Calculation of weight

$$W_{scissors} = 4.49 * (2 * 0.65 + 2 * 0.45 - 4 * 0.05) = 8.98\ kg \approx 100\ N$$

The total weight up till now is 5600 N.

Conclusion According to standards we will take $H = 30, b = 25, t = 4$ and $r = 8$ as the dimensions of the cold formed section of the base.

Figure 18 *Base 3D model*

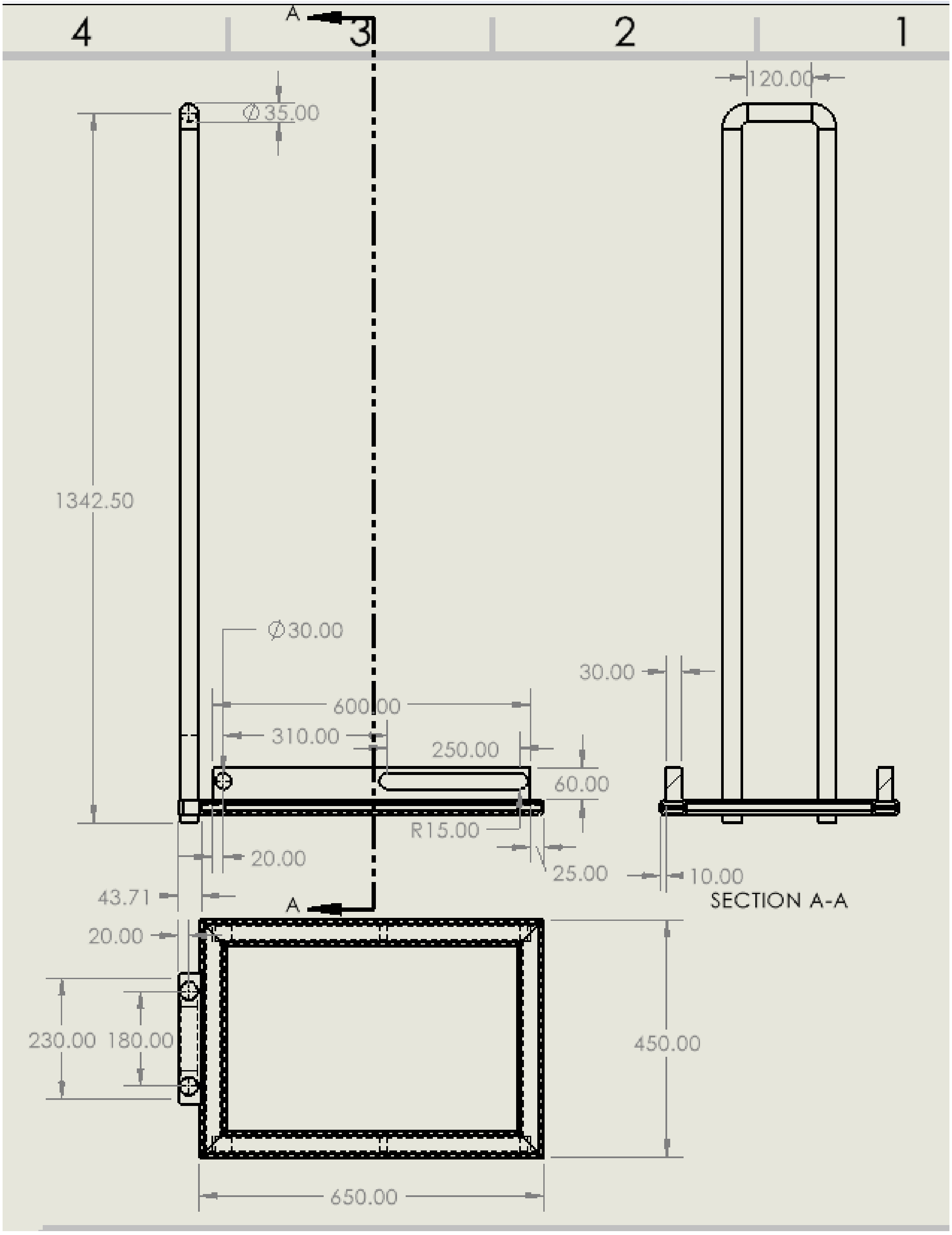


Figure 19 Base 2D drawing

WELDING

The welding operators should match the strength of the filler metal to the lower strength of the base material.

- ***Shear stress in the weld at the frame***

$$l = \frac{50}{\cos 45} = 50\sqrt{2}\ mm$$

$$h = 30\ mm$$

$$P = \frac{total\ weight}{4} = \frac{5450}{4} = 1362.5\ N$$

$$\tau_{max} = \frac{P}{0.707hl} = \frac{1362.5}{0.707 * 30 * 50\sqrt{2}}$$

$$\tau_{max} = 0.90847\ MPa$$

Conclusion The maximum stress is negligible. The table will not fail in regards to welding at the frame.

- ***Shear stress in the weld at the rail***

$$l = 600\ mm$$

$$h = 5\ mm$$

$$W = \frac{total\ weight}{2} = \frac{5450}{2} = 2725\ N$$

As It is double welded the force is again divided by 2

$$P = \frac{W}{2} = \frac{2725}{2} = 1362.5\ N$$

$$\tau_{max} = \frac{P}{0.707hl} = \frac{1362.5}{0.707 * 5 * 600}$$

$$\tau_{max} = 0.6424\ MPa$$

Conclusion The maximum stress is negligible. The table will not fail in regards to welding at the rails.

WHEELS

The total weight up till now was 5600 N divided on the 4 wheels, we get:

$$force\ per\ wheel = \frac{5600}{4} = 1400\ N$$

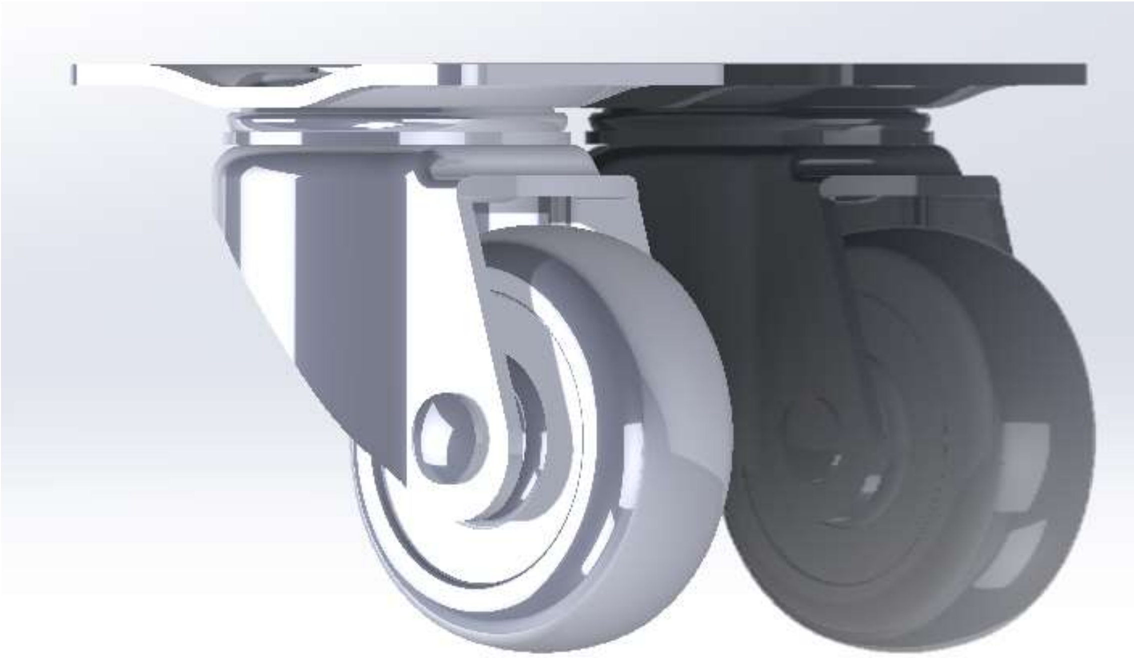


Figure 20 *Wheels 3D model*

FINAL ASSEMBLY

__Figure 21__ Full assembly 3D model

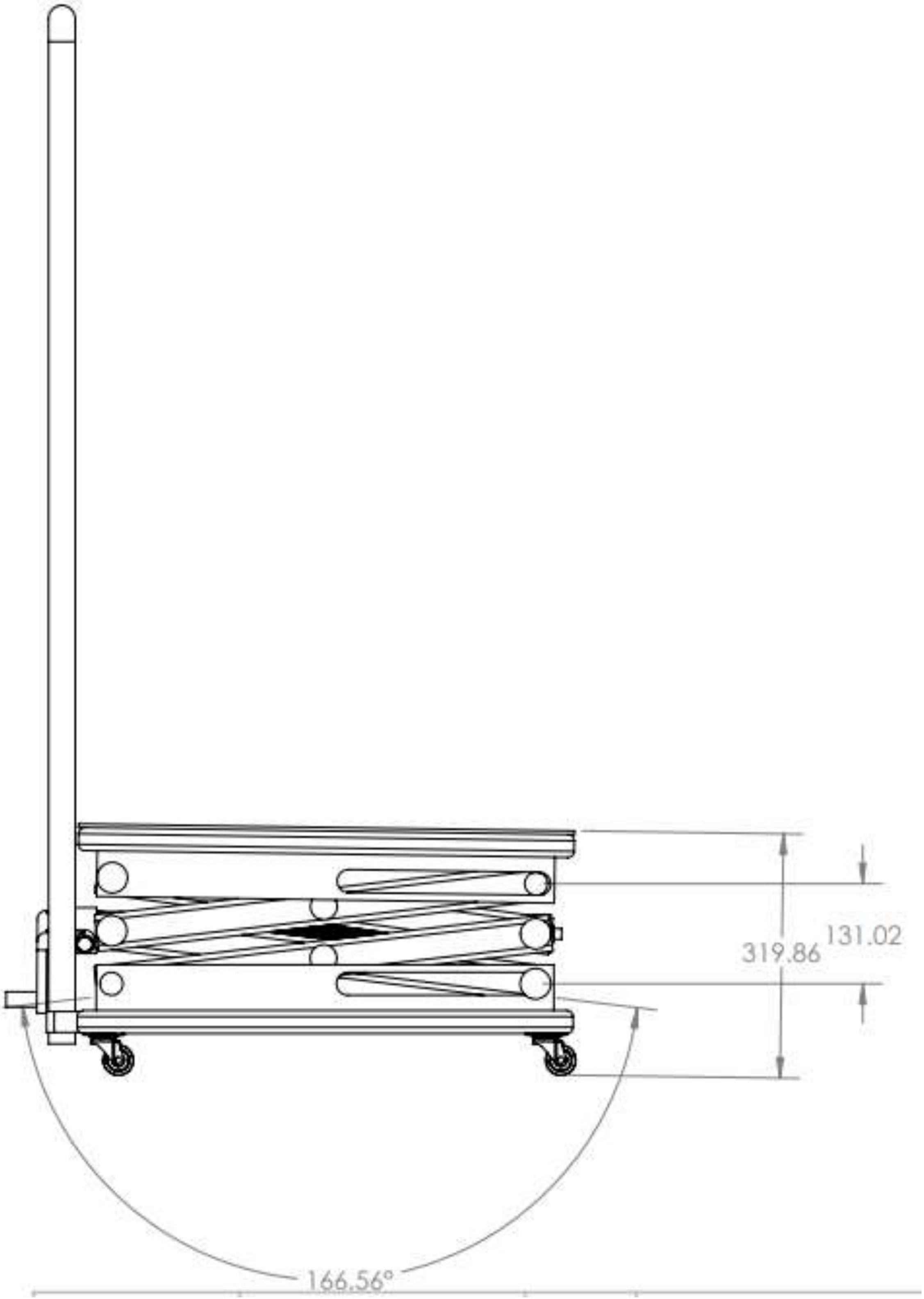


Figure 22 Fully compressed

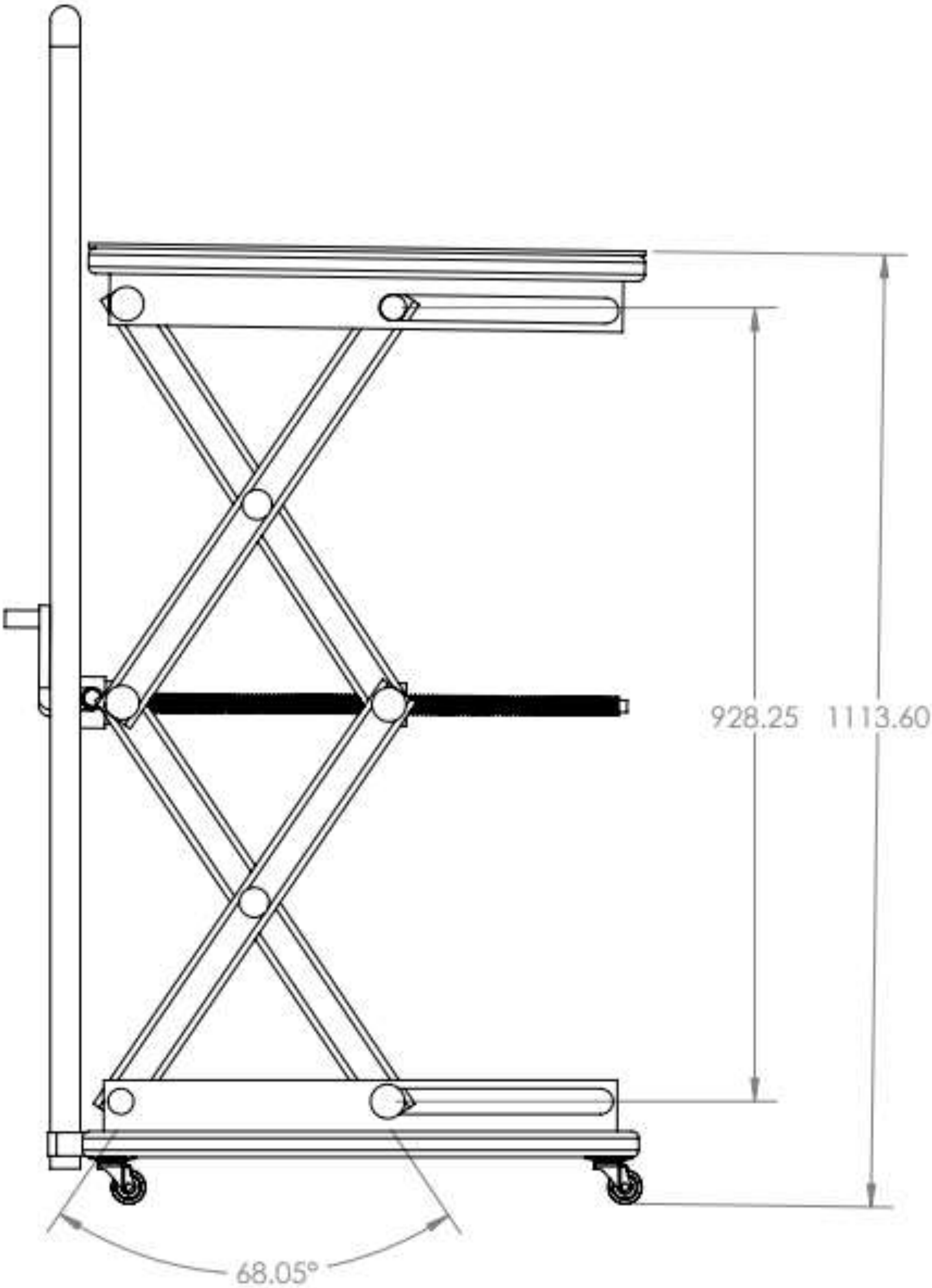
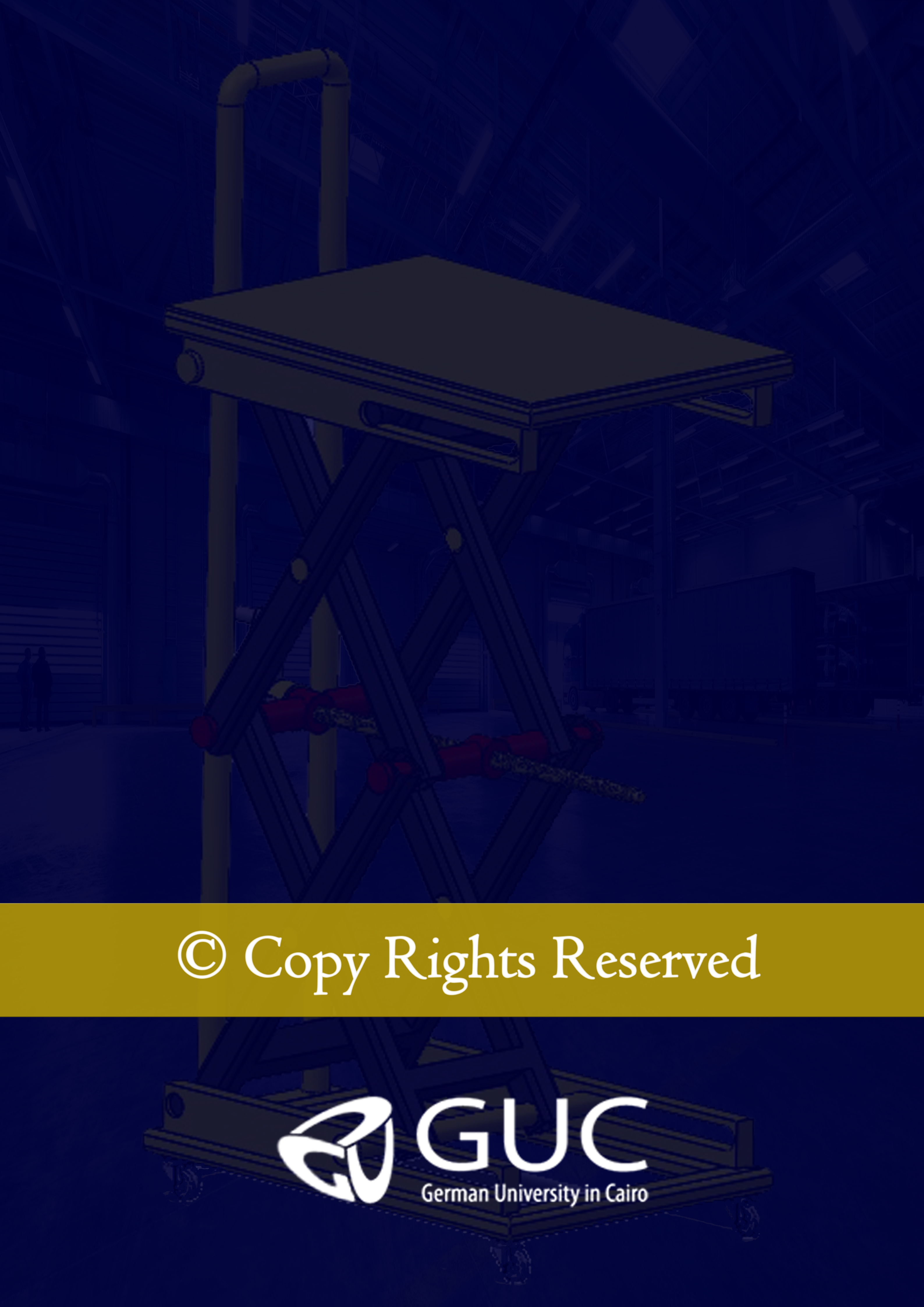


Figure 23 *Fully extended*

Figure 24 *Motion Study*



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