# Design and Simulation of Multipurpose Built-In Car Lifting Mechanism


by:

| | |
|---|---|
| **Hassan Ali Albuwaydi** | **391900538** |
| **Hamad Mohammed Alhashmy** | **391901224** |
| **Muthana Abdulla Al-Qater** | **392902680** |
| **Fahad Khaled Aldossary** | **401900196** |
| **Albaraa Mohammed Alshuaibi** | **401900913** |
| **Abdullah Ahmed Alqurashi Alzahrani** | **401902486** |


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**Jubail Industrial College**


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in

**MECHANICAL ENGINEERING**

**Semester 442, May 2023**

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**Semester 442, May 2023**

# JUBAIL INDUSTRIAL COLLEGE

**JUBAIL INDUSTRIAL CITY, SAUDI ARABIA.**

## DEPARTMENT OF MECHANICAL ENGINEERING

## CERTIFICATE OF COMPLETION

The Project entitled **"Design and Simulation of Multipurpose Built-In Car Lifting Mechanism",** has been submitted in partial fulfillment for the award of **Bachelor of Science Degree in Mechanical Engineering** by:

| | |
|---|---|
| Hassan Ali Albuwaydi | 391900538 |
| Hamad Mohammed Alhashmy | 391901224 |
| Muthana Abdulla Al-Qatar | 392902680 |
| Fahad Khaled Aldossary | 401900196 |
| Albaraa Mohammed Alshuaibi | 401900913 |
| Abdullah Ahmed Alqurashi Alzahrani | 401902486 |

under the guidance and supervision of Dr. Mohammed El-Shabasy.

The Project has been accepted by the Mechanical Engineering Department.

**First Examiner**

**Second Examiner**

**Supervisor**

**Chairperson**
**Mechanical Engineering Department**

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# Project Overview

In the first phase of the project, a preliminary design of multi-purpose built in car lifting mechanism. Our aim is to build a mechanism that can lift and move the car for several purposes, which can help the driver in many difficult situations. As result, built-in mechanism was eventually a leading factor to elevate and move the proposed targeted car. Preliminary calculation of the force analysis and stress analysis using both hand calculations and software such as SolidWorks and ANSYS was also done.

In the second phase of the project, we were able to change the mechanism of lifting and we were able thanks to Allah to contribute in an international competition for under graduate paper in IEOM. A double hydraulic scissor jack mechanism, which elevates the car and then move it forward or backward, has been considered as the project design, never the less, we faced problems during the semester and one of them was the difficulty to manufacture the mechanism. An improvement to be considered is to reduce the size of the mechanism beneath the car and make some source of adjustment in the sliding part using some type of silicon to avoid friction, dust and such minor components.

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# Nomenclatures

| Symbol | Quantity | Unit |
|---|---|---|
| **F** | **Force** | **N** |
| **m** | **Mass** | $Kg$ |
| **W** | **Weight** | $N$ |
| **g** | **Gravitational acceleration** | $m/s^2$ |
| **L** | **Length** | $meter$ |
| **M** | **Moment** | $\mathbf{N * m}$ |
| **θ** | **angle** | **degrees** |
| **A** | **Area** | $m^2$ |

# 1 INTRODUCTION

## 1.1 Motivation

Most of accidents occur on the road and those that happen off the road doesn't get mentioned in the study. Yet there is a lot of accidents that happen in different environments whether its mud, snow, and sand, where some people travel alone and get stuck in these difficult terrains, sometimes help can be reached within a call. However, most of the time people get stranded where there are no cell towers, and no recovery equipment.

Since most car users are not that skilled when it comes to going off road, and not everyone has the ability to push or fix the ground for the car to get the car unstuck. However, in some cases getting out of the car to get it out might be dangerous because of the environment or any outer dangers that the driver may face. Therefore, this project will aim at these three main points, skill, simplicity of mounting the mechanism and easy to operate.

## 1.2 Problem Statement

Many applications have been invented to get unstuck in all types of terrains under the name of recovery kits such as traction grips, jacks, winch and tire chains. However, they have many draw backs. In general, they require strength and power to apply. Every application is used in a certain scenario or a particular environment. Many applications are big in size due to that there will be loss in the space of the vehicle and they are limited to off-road use only. The disadvantage of the recovery kits are:

- Recovery kits are difficult to use.
- Each part is used in a particular scenario.
- You need to have the experience to know how and when to use them.
- Recovery kits take up space in your vehicle due to their size.
- The solutions are restricted to the off-road scenarios only.

## 1.3 Background

### 1.3.1 History

In the twentieth century the popularity of cars has risen, hence the car repairs were a demand. The mechanics back then have dug holes in the ground to conduct the necessary repairs. In 1925, the first car lift mechanism was invented by Peter lunati. It was a hydraulic mechanism that was inspired in a hydraulic barber's chair, in Figure1 [1]. He saw how easy it was to lift a body up and down. This mechanism has open doors to the car lifting in the modern days. in Figure 2 [2].

*Figure 1. Hydraulic barber chair.*

*Figure 2. First invention of car lifting mechanism.*

The modern car lifts were rooted into three types, stationary, portable and built-in. The stationary car lifting mechanisms has been developed since 1925. The latest stationary car lift mechanism is the shockwave.

In the field of portable car lifts, the mechanic found that it would be more convenient if the car lifts were easily moved around as per the car moves, in particular, Quickjack lifting mechanism.

Race cars had a demand on quick tire changing, hence mechanics have devolved a built-in car lifting mechanism which can give a quick recovery to change the car tires.

### 1.3.2 Definition

Car lifting is a procedure that elevate cars to allow mechanics to operate on cars by using various mechanisms. The car lifting contains a wide contact surface attached to links that are connected to a mechanism. The car lifting is well-known for its linearity in cars to do maintenance and repair.

The three types of car lifts are:

• Stationary car lift

• Portable car lifts

• Built-in car lifts

Stationary car lift is a big structure that lifts the car within many supports to the ground. There are many machines in the field. One of which is shockwave, it gave a new meaning of a quick fix, in Figure 3 [3]. Moreover, Two-post lift structure is used mainly in workshops, in Figure 4. Furthermore, stationary hydraulic scissor lift is used to lift massive trucks, in Figure 5 [5].


*Figure 3. Shockwave.*


*Figure 4. Two post lift mechanism.*


*Figure 5. Stationary Scissor lift.*

In order to maintain the car without relying on workshops, mechanics have developed portable car lifting device. For instance, although quickjack lifting mechanism is used in many workshops, many people use it for personal usage, as shown in Figure 6 [6]. In addition, Scissor power screwed jacks is the most convenient and cheap portable lifting device, as shown in Figure 7. Farm jacks, airbag lifting jacks are widely used for multi-functions particularly changing car tires due to how easily used, as shown in Figure 8. [7]


*Figure 6.  Quickjack lifting mechanism*

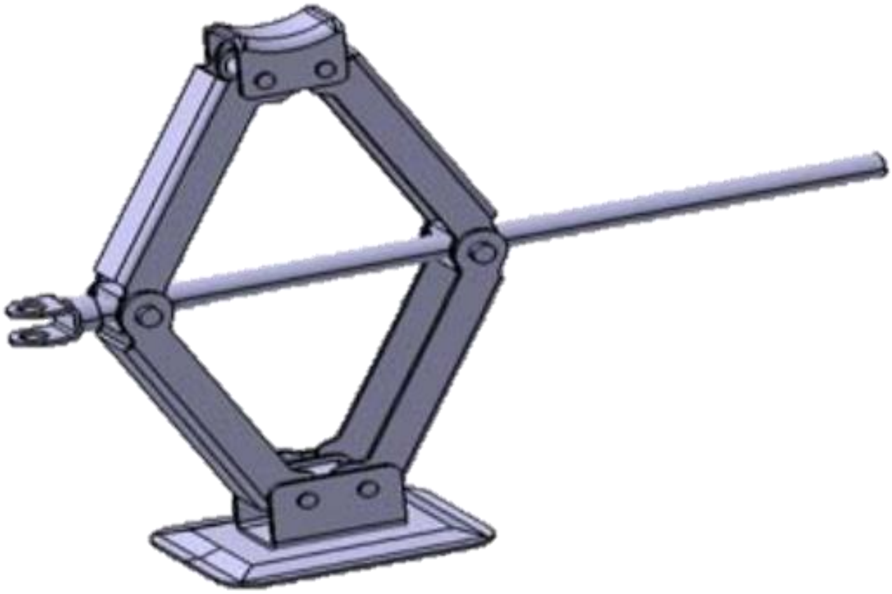


*Figure 7.  Scissor power screwed jacks.*

*Figure 8. Farm jack mechanism.*

Finally, in race cars speed plays a major part in races, including tire changing and repairs. Due to the demands of the mechanics to develop a device to lift cars faster, specialists have developed a built-in pneumatic system lifting mechanism that can lift the car faster with a hose that pumps air to the pneumatic lifting system, as shown in Figure 9. [8]

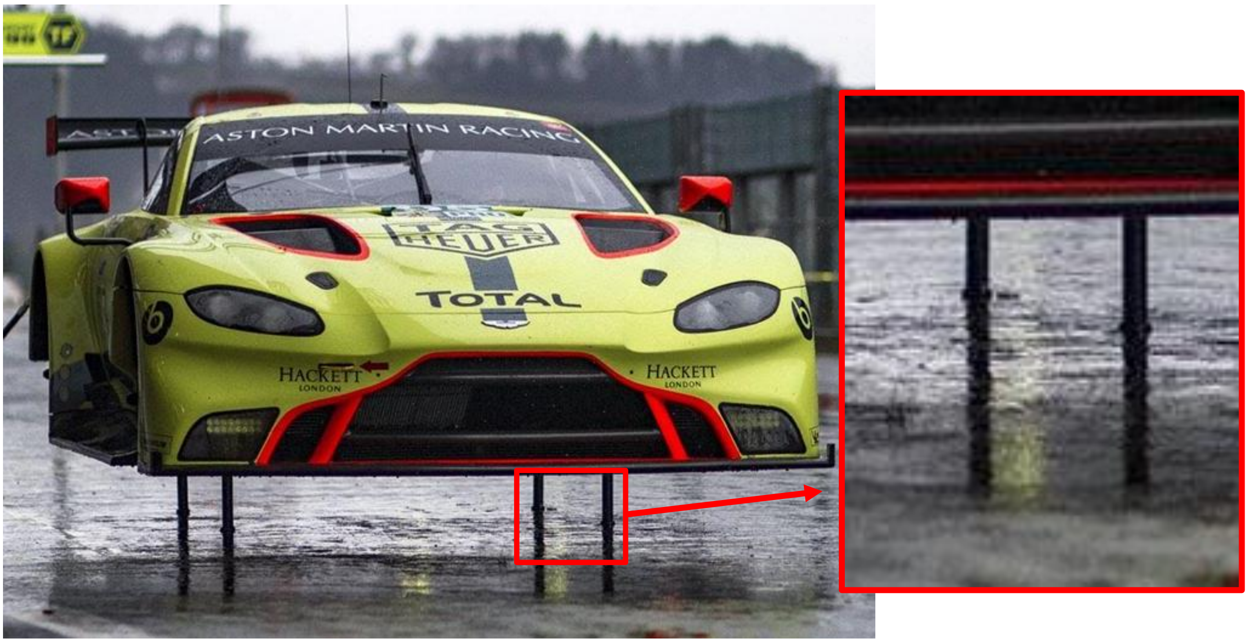



*Figure 9. Air powered race car lift.*

## 1.4 Objectives of the Current Work

The previous section showed what have been used to lift objects, and it is only used for that purpose. Therefore, this project will include many benefits and more than just lifting car and having other uses. It will follow several points strictly to design a lifting mechanism, which are:

- To design a mechanism that will lift the car.
- To design a mechanism that will move the car
- To design a built-in mechanism in the vehicles.
- To design a mechanism that will not alter the original design of the vehicle.
- To design a mechanism that will benefit from an already existing mechanism.

## 1.5 Methodology

The aim of this project is to design a mechanism that can lift the vehicle mainly. In addition, there will be another mechanism that will move the car forward or backward, and because not only lifting the car, but also for moving the car from its stuck position can be done, it can also be used for other purpose such as changing tires, doing general services for the car like changing oil, cleaning and changing parts.

Lifting the car can be done using multilabel mechanisms that can be considered as an alternative. Some comparison will be made that depends on some criteria such as, cost, maintenance, safety and endurance, upon which the applicable mechanism will be selected as the basic design for this project.

A kinematic analysis will be conducted to reach the required mobility of the mechanism and how many drivers the system will require to drive the mechanism will be determined. Then using the force analysis method, various factors such as factor of safety and predicting the failure points in the mechanism will be determined. Following the kinematic analysis, there will be a stress analysis that

will help in preventing the mechanical failures. Since this project will use various analyses mentioned above, some international standards must be selected to make this project consistent.

In order to compare the results of analytically calculated, SolidWorks stresses simulation model will be developed to preform stress analysis. Furthermore, SolidWorks can help in visualizing the motion and mobility of the model with the help of animations.

To conclude, the methodology involves the following steps:

- Define the aim of the project.
- Present the application of the project
- Show the alternative designs.
- Compare the alternatives based on constraints and selection criteria.
- Conduct kinematic analysis to test the mobility.
- Conduct the force analysis.
- Conduct the stress analysis and use the suitable theories of failure.

- Conduct simulation for the designed mechanism by using SolidWorks software.
- Conduct comparison between the results obtained from analytical calculations with the numerical calculations.
- Studying the motion and mobility of the design using animations.
- Manufacture a full-scale model and test the system.

# 2 PRELIMINARY DESIGN

## 2.1 Conceptual Design

In this project, the aim will be mainly to focus on a mechanism that will lift the vehicle. The mechanism that will meet all the required actions to move the car out of the stucking situation. In the following section, the alternatives will be discussed before selecting and deciding the perfect match for the project.

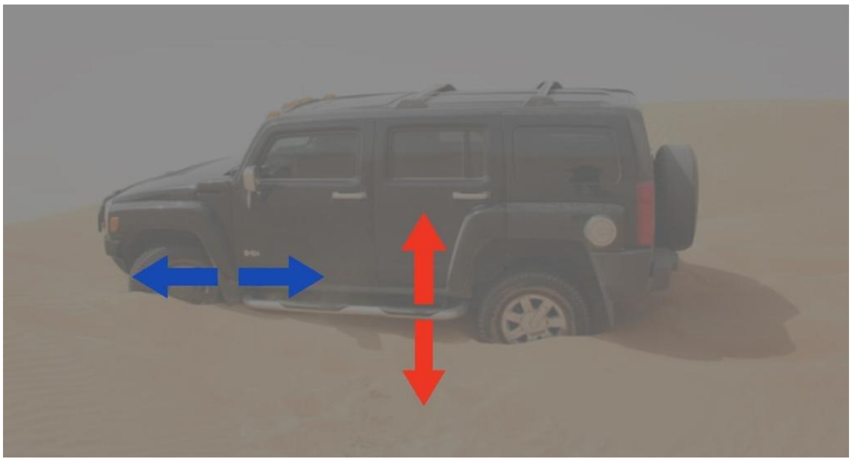


*Figure 10. Primary description for stacking situation.*

## 2.2 Alternatives for System Concepts

We have 5 different alternatives for the system layout: Double lead power screws mechanism, Hydraulic scissor jack and lead power screw jack, Lead power screw scissor jack and hydraulic system, Hydraulic scissor jack and hydraulic systems and Running board and hydraulic system.

### 2.2.1 Constraints and Selection Criteria

#### 2.2.1.1 Constraints

This part of the evaluation must be considered when comparing the main design and the alternatives as the following:

- Engineering standards must be considered when selecting or designing any of the components.
- The cost of manufacturing might be affordable and reasonable.
- The mechanism will mainly focus on big and heavy cars.
- Most of the selected materials should be affordable in the local market.
- The mechanism should fit in the space underneath the targeted vehicle.

#### 2.2.1.2 Selection Criteria

Based on constraints and the alternatives and the results obtained from searching and reading about the different types of lifting jack mechanisms the following have been decided:

- The source of driving power, due to the cost and manufacturing ability and maintenance.
- The type of mechanism they will be used, due to ease to operate, maintenance and installation difficulty.
- The system that will be designed will be a system similar to the running board, however, the system will be changed, it will be modified to meet the project's objectives.

### 2.2.2 Double lead power screws mechanism

The whole system will be driven by two lead power screws for the lifting as well as the movement, as shown in Figure 11. This system is affordable and easy to operate. The power screw will be powered by a motor. The rotary motion of the lead power screw will provide sufficient force to lift and move the vehicle. However, this system has many drawbacks. Two drawbacks will be discussed in brief. Firstly, the occurrence of failures of this system is feasible. Moreover, the periodic lubrication of the system is a major demand to reduce friction, as shown in Figure 12.

*Figure 11. Power screw scissor jack mechanism.*

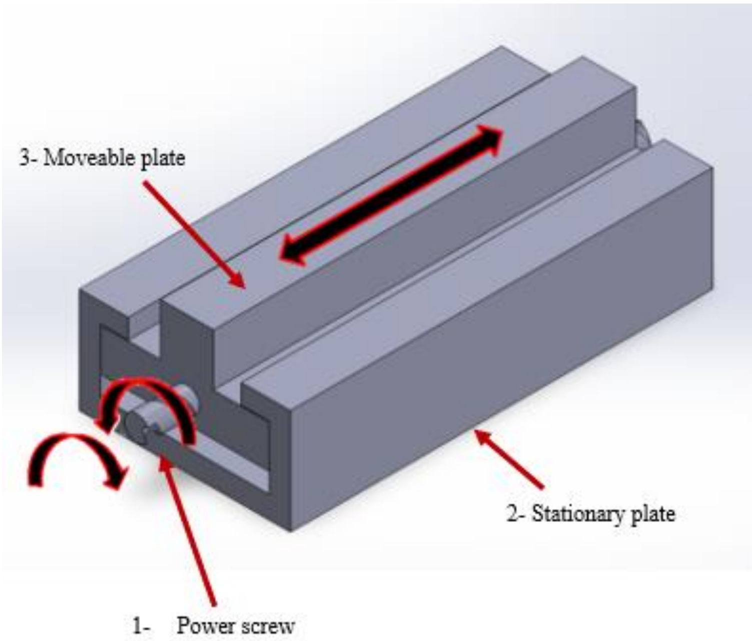



*Figure 12. Built-in Movable plate.*

1- Power screw, which is controlled by a motor, which move it clockwise and counter clockwise thus move it forward and backward

2- Stationary plate which is held under the car and it has a slot which the moveable plate moves in it.

3- Moveable plate which moves by the power screw, and it has the running board attached to it.

### 2.2.3 Hydraulic scissor jack and lead power screw jack

In this mechanism, the system is hybrid. The lifting is done by a hydraulic scissor jack, on the other hand, the forward-backward movement is done by a lead power screw mechanism. In fact, the addition of the hydraulic part is a good idea to be considered, as shown in Figure 13.

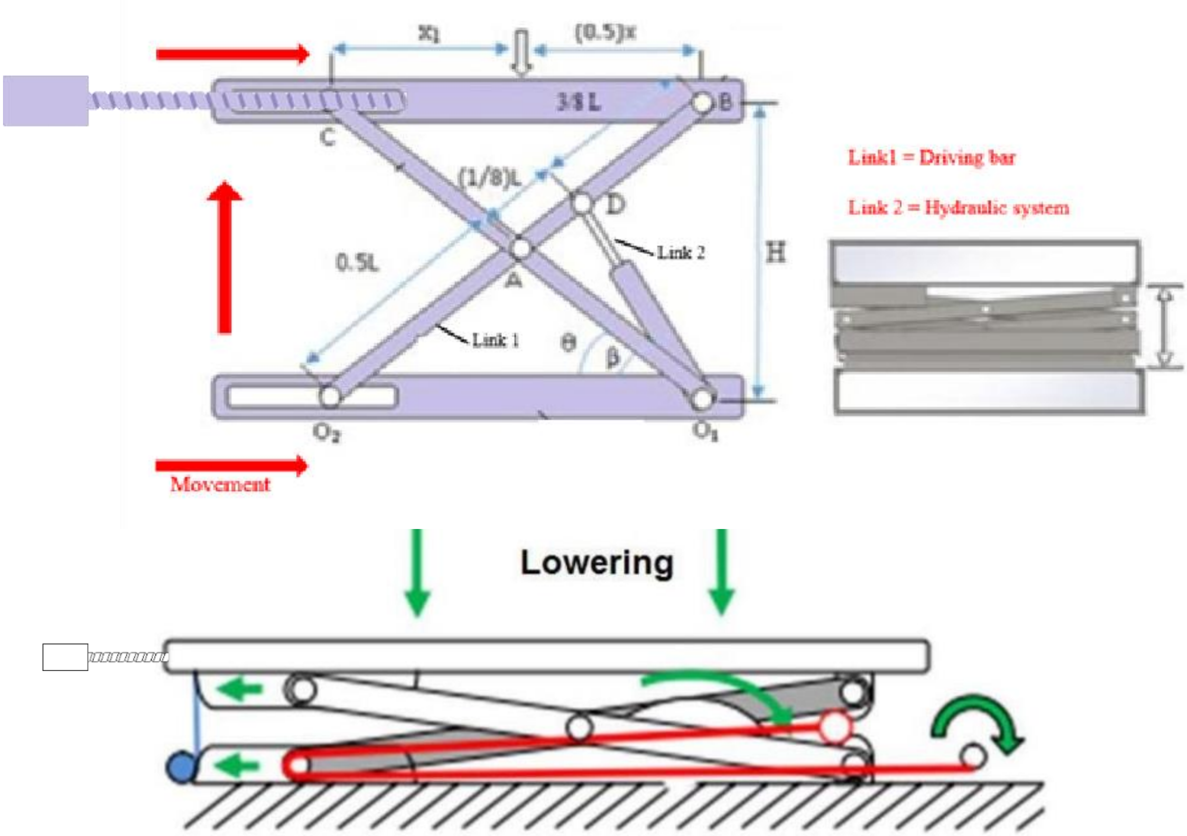



*Figure 13. Hydraulic lifting mechanism*

### 2.2.4 Lead power screw scissor jack and hydraulic system.

This mechanism will be altering the systems for each function of the previous alternatives. It will consist of two lead power screws mounted on the two chassis and the movement of the car will mainly be operated by a hydraulic system. This seems like an acceptable idea to be taken into consideration; however, the mechanism might start rusting hens it will not operate properly when desired, as shown in figure 14.

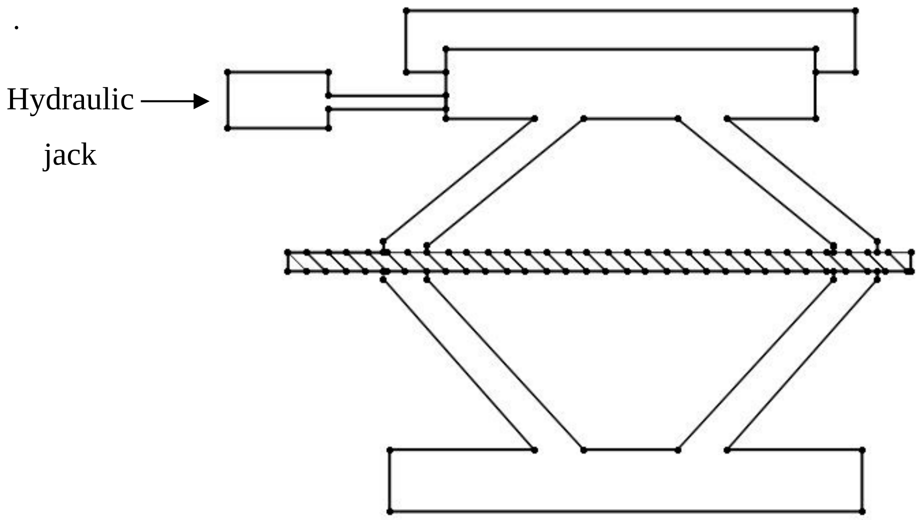



*Figure 14. Hydraulic lifting system.*

### 2.2.5 Hydraulic scissor jack and hydraulic systems.

The system will contain two hydraulic systems. The first mechanism will be mounted on the chassis to lift the car. The second one will function as in forward and backward moving mechanism. This system can be considered beneficial to cost to function ratio, as shown in Figures 15.

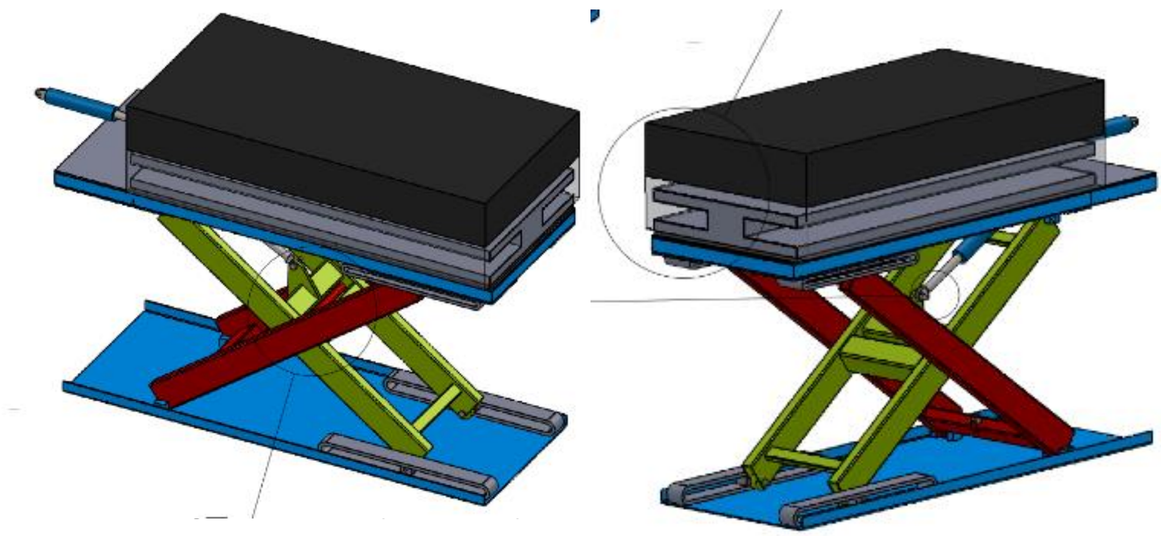


*Figure 15. Hydraulic scissor lift mechanism.*

### 2.2.7 Running Board System.

A running board is a narrow platform attached to the side of a vehicle, designed to provide a step for passengers. It is now less common on passenger vehicles, but still found on some trucks and SUVs. The running board is mounted on the sides of the vehicle due to less amount of weight needed to be lifted. Hence, in this project, the running board mechanism will be added to the chassis to lift the car. In addition, a hydraulic system will be added to the running board to go all the way to the ground and lift the car eventually. The movement of forward-backward will be operated by a hydraulic system. This will safe time of the design because the mechanism already exists on some cars, as shown in Figures 16 and 17.

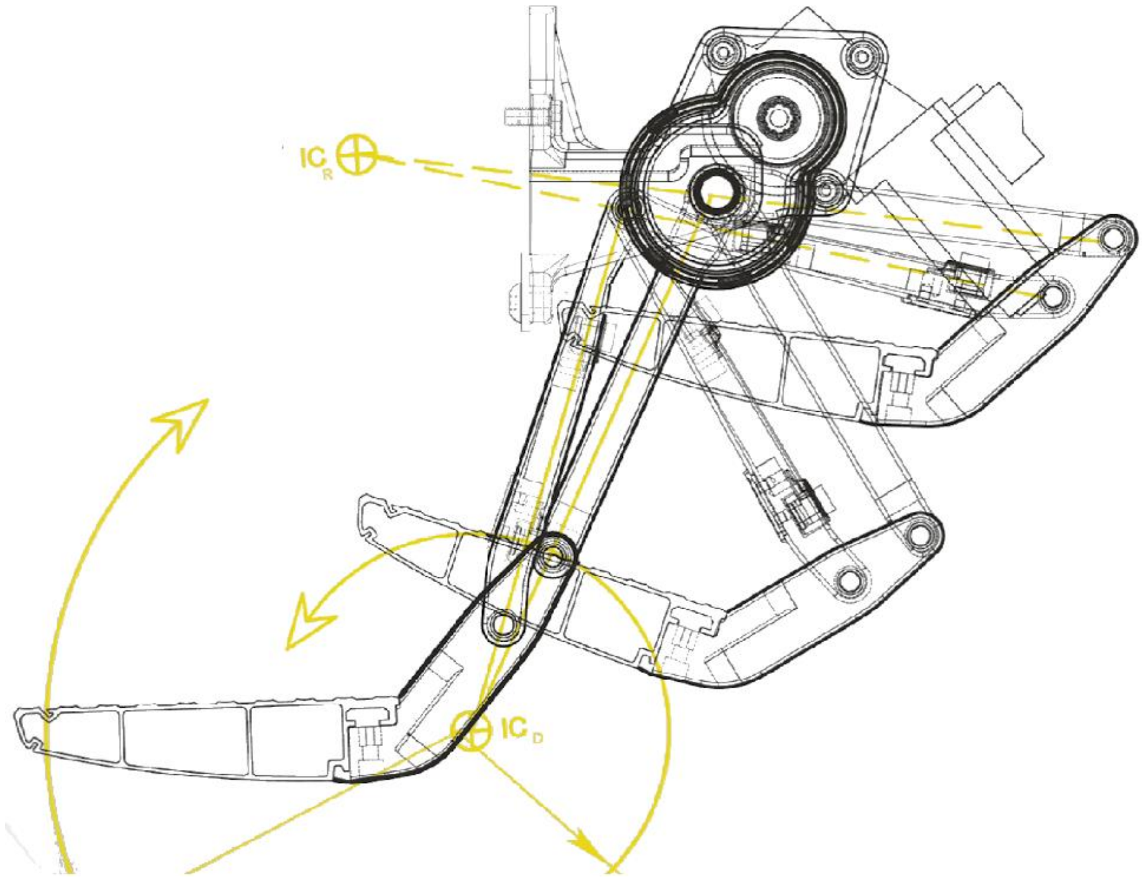



*Figure 16. Running Board mechanism.*

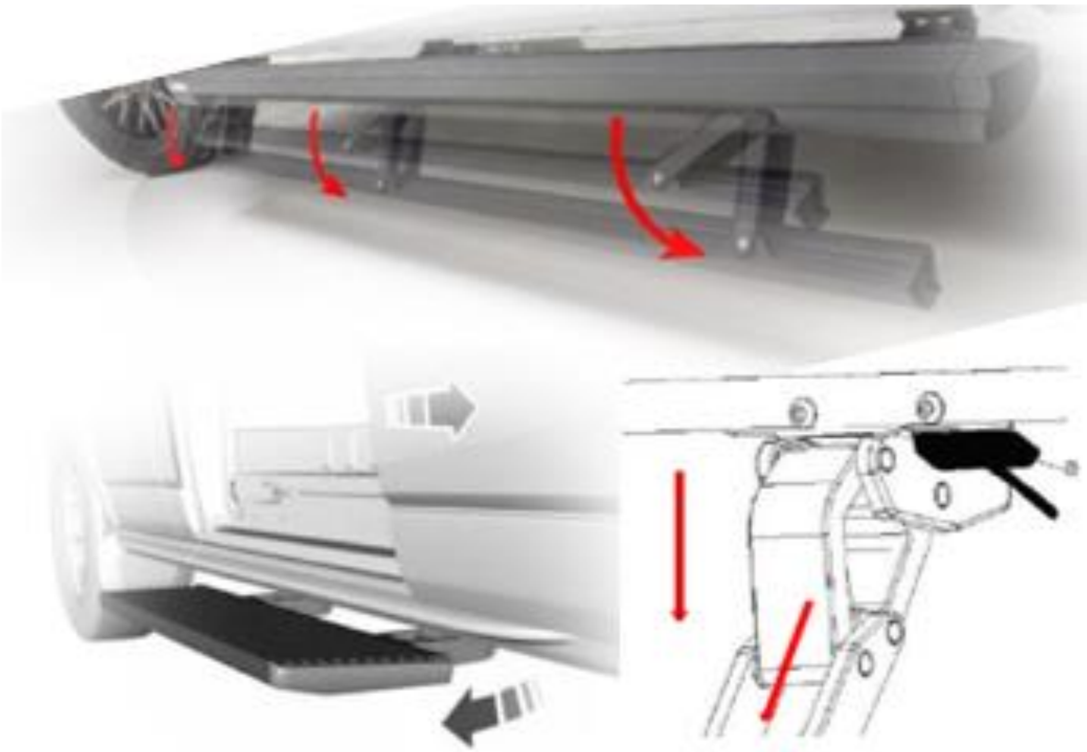


*Figure 17. Running board system.*

### 2.2.8 Decision Making: System Concept

Based upon the performed comparisons under the given constraints and selection criteria, the preferred mechanism will be the double hydraulic mechanism.

## 2.3 Alternatives of Components

In this section, the comparison will be made between the power input as well as the type of operating mechanism for the running-board. Then, all the suggested power suppliers will be compared eventually using a rating-based evaluation criterion. At last, the most preferred type will be chosen based on the final result evaluation.

### 2.3.1 Components Comparison Between Three Types of Driving Power Input

Table 1. Comparison between inside car battery and adding an external battery outside the car as a source of power

| Driving power input | Direct motor with coupling | Hydraulic jack system | Pneumatic system |
|---|---|---|---|
| **Initial cost** | Low (4) | Medium (3) | Low (4) |
| **Running cost** | Medium (3) | Medium (3) | Medium (3) |
| **Safety** | Low (2) | High (5) | Medium (2) |
| **Efficiency** | Medium (2) | High (5) | High (4) |
| **Endurance** | High (5) | High (5) | High (5) |
| **Maintenance** | High (2) | Low (4) | Medium (3) |
| **Total points** | 18 | **25** | **21** |

According to this comparison shown above in Table 1, the hydraulic system is suitable to be the driving input power, as shown in Figure 17.


| Direct motor with coupling | Hydraulic jack system | Pneumatic system |
|---|---|---|

*Figure 18. Driving power input.*

### 2.3.2 Comparison Between Operating Mechanism for the lifting and moving parts

Table 2 Comparison between five Types of running mechanisms

| Mechanism | Double lead screw | Hydraulic scissor jack and lead power screw jack. | Lead power screw scissor jack and hydraulic system. | Double hydraulic system mechanism | Running board and hydraulic system |
|---|---|---|---|---|---|
| **Maintenance** | High (1) | High (2) | Low (2) | Medium (3) | Medium (3) |
| **cost** | Medium (3) | Medium (2) | Medium (2) | High (2) | High (2) |
| **Difficulty to operate** | Low (3) | Medium (3) | Medium (3) | Low (5) | Low (5) |
| **Installation** | Medium (2) | Medium (3) | Medium (3) | Low (5) | Low (3) |
| **Complexity** | Medium (3) | Low (4) | Low (4) | High (2) | Medium (3) |
| **Total points count** | 12 | 14 | 14 | 17 | 16 |

Therefore, Double hydraulic system mechanism is more suitable for the selected criteria.

Specifications of selection:

Maintenance: 1 (poor), 5 (excellent)

Cost: 1 (high cost), 5 (low cost)

Difficulty to operate: 1(hard), 5(easy)

Simplicity of installation: 1(complicated), 5(simple)

Complexity of the mechanism: 1(complex), 5(simple)

### 2.3.3 Comparison Between two types of lower pair and higher pair joint

Table 3 Comparison between two types of joints

| **Operating mechanism** | **Slider joint scissor lift** | **Roller arm scissor lift** |
|---|---|---|
| **cost** | Medium (3) | **Low (4)** |
| **Safety** | High (4) | **High (4)** |
| **Stability** | Medium (3) | **Medium (2)** |
| **Efficiency** | High (5) | **Medium (3)** |
| **Maintenance** | Low (4) | **High (1)** |
| **Total points** | 19 | **14** |

So based on the comparison shown above in Table 2, the preferred joint will be the slider joint scissor lift, as shown in Figure 18.

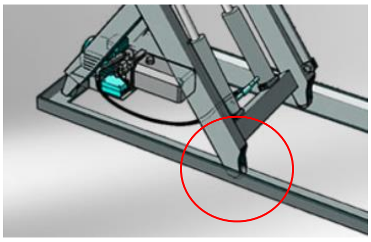
| | |
|---|---|
|  |  |
| Roller arm scissor lift | Roller arm scissor lift |

*Figure 19. Operating mechanism.*

### 2.3.4 Constraints

- Materials may not be available in local market.
- Time constrain we should finish manufacturing this project before the end of next semester.
- The engineering standards should be strictly followed when we select/design the components.

- Lack of resources to build this project upon.
- The cost of the design must be reasonable and under the budget if the manufacturing will take place.
- The available space underneath the car that will affect the size of the mechanism

#### 2.3.5 Selection Criteria

The selected car that will be used in our design will be a Ford F-150.
The source of power, due to its manufacturing and maintenance costs.
The type of the mechanism, based on a number of variables such as maintenance, endurance, and cost.

#### 2.3.6 Decision Making on System Components and Layout

After looking into the different mechanisms that have been previously presented and the comparison made between them under the certain selection criteria and constrains. It has been decided to choose the hydraulic scissor lift mechanism to drive out the lifting mechanism, a hydraulic system as a power source and a hydraulic system mechanism as the main part for forward and backward movements the car. The overall advantages and disadvantages met the desired requirement for the system. Moreover, the safety of the selected options has more advantages than the other alternatives. In addition, since this alternative uses a built-in mechanism that can lift passengers, afterward it can be modified and scaled up to lift the car.

## 2.4 Design Standards

In the following section, the selected configuration of the two design alternatives is designed based on the American Society of Testing and Materials (ASTM), the American Society of Mechanical Engineering (ASME) and ECISS (European Committee for Iron and Steel Standa-rdization). Those standards are followed on the level of the used von mises stress theory of failure, the recommend factor of safety and the mechanical properties of the selected materials.

### 2.4.1 Hydraulic system standards

ISO 5598, Fluid power systems and components

ISO 6162-1, Hydraulic fluid power — Flange connectors with split or one-piece flange clamps and metric or inch screws — Part 1: Flange connectors for use at pressures of 3,5 MPa (35 bar) to 35 MPa (350 bar), DN 13 to DN 127

ISO 10763, Hydraulic fluid power — Plain-end, seamless and welded precision steel tubes — Dimensions and nominal working pressures.

### 2.4.2 Four-bar linkage and running board standards

2.5.2 ASTM A240 for 904L Duplex Stainless Steel.

Table 4. Physical properties of duplex stainless steel.

| Physical Property | Value |
|---|---|
| Density | 7.805 g/cm³ |
| Thermal Expansion | 13.7 x$10^{-6}$ /K |
| Modulus of Elasticity | 200 GPa |
| Thermal Conductivity | 19.0 W/m.K |
| Electrical Resistivity | 0.85 x$10^{-6}$ Ω .m |

Table 5. Mechanical properties of duplex stainless steel.

| Mechanical Property | Value |
|---|---|
| Proof Stress | 450 Min MPa |
| Tensile Strength | 620 Min MPa |
| Elongation A50 mm | 25 Min % |
| Hardness Brinell | 290 Max HB |

Table 6. Minimum weld-metal properties

| Class | Min. Tensile Strength | Min. Yield Strength | Percent Elongation |
|---|---|---|---|
| E60XX | 427 MPa | 345 MPa | 17-25 |
| E70XX | 482 MPa | 393 MPa | 22 |
| E80XX | 551 MPa | 462 MPa | 19 |
| E90XX | 620 MPa | 531 MPa | 14-17 |
| E100XX | 689 MPa | 600 MPa | 13-16 |
| E120XX | 827 MPa | 737 MPa | 14 |

# 3 DETAILED DESIGN

The following subsections contain all kinematic and force analysis.

## 3.1 Kinematic Analysis

To reach the desired goal, we decided to separate the two drivers and focus on a single driver which is the lifting driver. The degree of freedom for the lower mechanism must equal one, which is one movement upward or downward, and for that, one input will control the movement. Thus, the degree of freedom at Equ.1 (mobility), as shown in Figure 20, will be calculated in the following step:

$$M = 3(L-1) - 2J_1 - J_2 \; [\,Equ.1\,]$$

$while$:

$M$: $Mobility$

$L$: $Number\ of\ links$

$J_1$: $Number\ of\ full\ joints$

$J_2$: $Number\ of\ half\ joints$

$$L = 6$$

$$J_1 = 7$$

$$J_2 = 0$$

$$M = 3(L-1) - 2J_1 - J_2 \; [\,Equ.1\,]$$

$$M = 3\,(6 - 1) - 2 \times 7 - 0$$

$$M = 1$$

So, the mobility of the mechanisms is,

M= 1 (constrained mechanism)

Thus, the assumption made above is right.

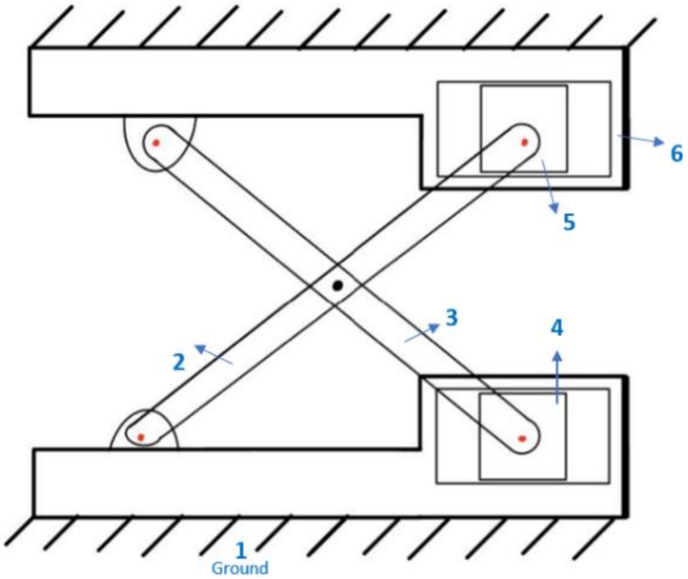



*Figure 20. Kinematic Chain of the proposed design*

Notice that the lower part of the mechanism which is causing the backward and the forward motion is the only considered mechanism in the above mobility analysis.

## 3.2 Mechanism Analysis

Figure 21 shows the main component of the proposed mechanism, and Figure 22 shows a simplified representation of the hydraulic system used in it for lifting and moving. In Figure 23, we calculated the height of chassis of Ford F-150 from the ground. We went with the height to be 3000 Kg, the chosen value of the car was way more than the original weight of the car to be in the safe side considering the worst-case scenario while calculating the factor of safety in stress analysis, the following symbols convey information about figures and forces, allowing us to describe complex ideas in detail.

F: Represent the weight of the car.

A: Is the location where the weight of the car acts on at the left side of the plate.

B: Is the location where the fixed pin at the top at 0.25m from left side of the plate.

C: Is the location where the pin in the slider at the top at 0.75m from left side of the plate.

D: Is the location where the fixed pin at the bottom at 0.25m from left side of the plate.

E: Is the location where the pin in the slider at the bottom at 0.75m from left side of the plate.

P: Is the location of the middle pin of the scissor where the hydraulic act on.

L: Represents the length of links (BE) and (CD).

FN: Represent the normal force due to the friction on the lower plate.

FP: Represent the force due to the hydraulic.

FR: The reaction force of the bottom plate to the force F at location A.

MR: The moment due to the shifting of the force F at location A to the mid-point.

$\theta$max: The maximum angle between the hydraulic and the plate.

RBy: The reaction of the fixed pin at vertical direction at point B.

RBx: The reaction of the fixed pin at horizontal direction at point B.

Rcy: The reaction of the pin in the slider at vertical direction at point C.

Rcx: The reaction of the pin in the slider at horizontal direction at point C.

RDy: Is the reaction of the fixed pin at vertical direction at point D.

RDx: Is the reaction of the fixed pin at horizontal direction at point D.

REy: Is the reaction of the pin in the slider at vertical direction at point E.

REx: Is the reaction of the pin in the slider at horizontal direction at point E.

RPy: Is the reaction of the fixed pin at vertical direction at point P.

RPx: Is the reaction of the fixed pin at horizontal direction at point P.

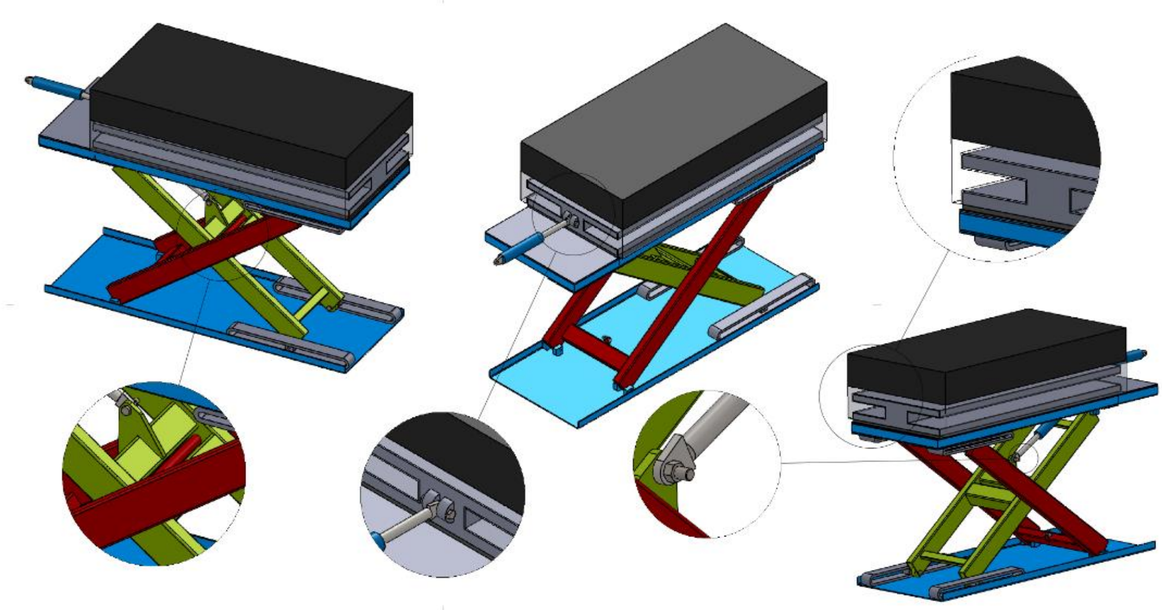


*Figure 21. Hydraulic scissor lift mechanism.*

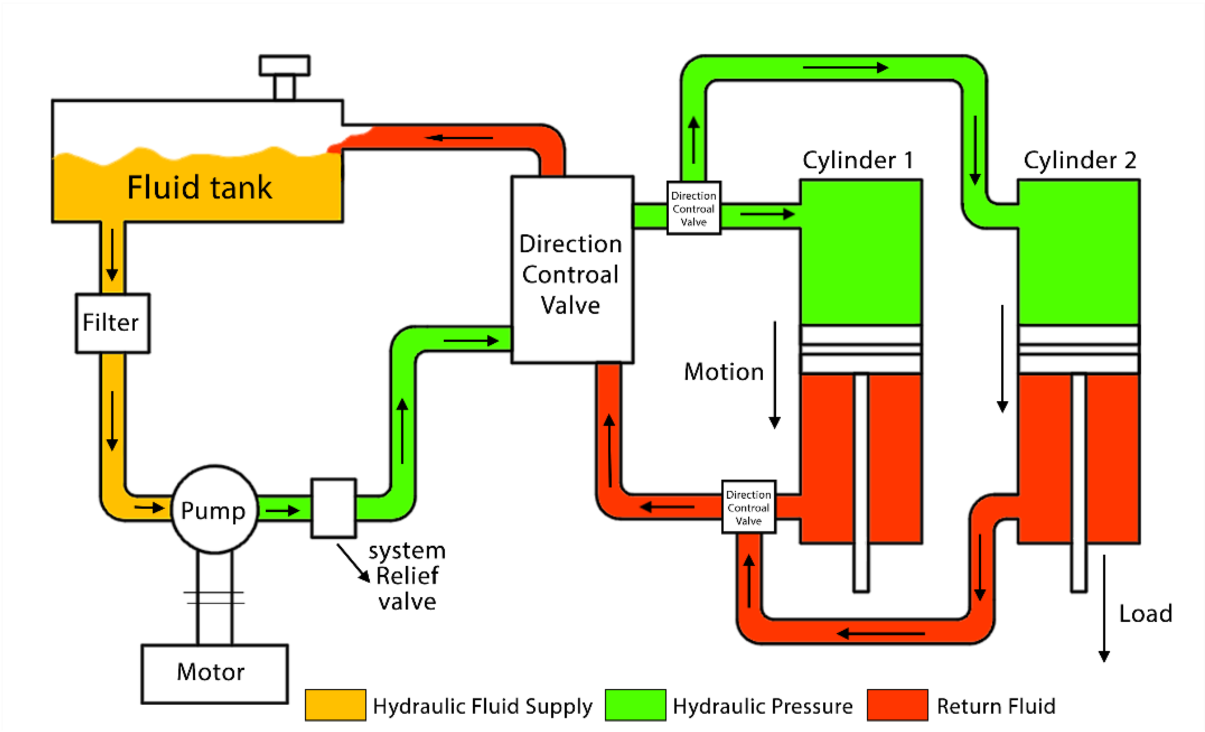



*Figure 22. Hydraulic System cycle.*

From the force analysis we found that the hydraulic force applied on the scissor to be 6.5 KN. And the length of the piston arm is 610.32 mm which means that it has a stroke of 610.32 × 2 = 1220.64 mm. With a bore diameter of 4 in the pressure in the piston should be P $= \frac{F_p}{A_p} = \frac{6500}{\pi\frac{(4\times25.4)^2}{4}} = 0.802\ Mpa$ Which equals to 116.32 psi. So we select a hydraulic circuit based on these specifications. A bore piston diameter of 4 in and pressure of 116.32 psi and a crank of 48.1 in.

### 3.2.1 Force of whole mechanism

In calculating the forces on the mechanism, we considered the worst-case scenario which is the weight will be on the end of the lifting mechanism. In addition, each link will lift one-eighth of the car since we have eight arms, so in each side of the mechanism there will be 2/8 from the car weight, and to be more in the safe side we considered that force to be 3/8 instead of 2/8 of car's weight.

$\Sigma F_y = 0\ \uparrow +$

$-F + F_R = 0$

$F_R = F$

$F_R = \left(\frac{3}{8}\right) \times 3000\ \times 9.81$

$F_R = 11.04\ KN\ \uparrow$

$+\circlearrowleft\ \Sigma\, M_A = 0$

$-(0.5 \times 11.04) + M_R = 0$

$M_R = 5.525\ kN$

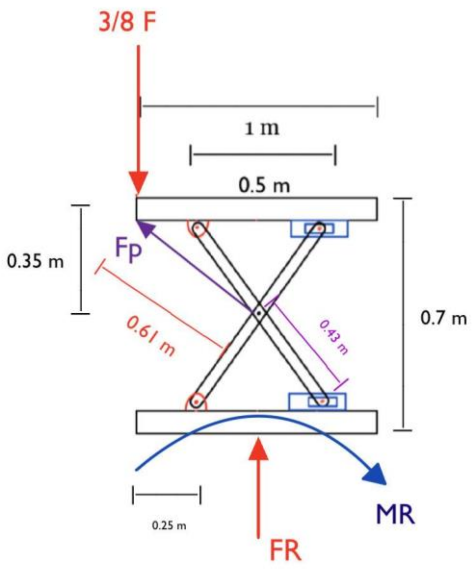



*Figure 23. FBD of whole mechanism*

#### 3.2.2 Force of upper plate

$+\circlearrowright$ ΣM at point C = 0

$R_B^y \times 0.5 + Fp \sin\theta \times 0.75 - F \times 0.75 = 0$

$$R_B^y = \frac{F \times 0.75 - Fp \sin\theta \times 0.75}{0.5}$$

$$R_B^y = \frac{11.04 \times 0.75 - Fp \sin 35° \times 0.75}{0.5}$$

$$R_B^y = \frac{8.28 - 0.43 \times Fp}{0.5}$$

$$R_B^y = 16.56 - 0.86 \times Fp \quad [Equ.2]$$

$+\circlearrowright$ ΣM at point B = 0

$Fp \sin\theta \times 0.25 - R_C^y \times 0.75 - F \times 0.25$
$= 0$

$$R_C^y = \frac{Fp \sin\theta \times 0.25 - F \times 0.25}{0.75}$$

$$R_C^y = \frac{Fp \sin 35° \times 0.25 - 11.04 \times 0.25}{0.75}$$

$$R_C^y = 0.1434\, Fp - 3.68 \quad [Equ.3]$$

$\Sigma Fy = 0 \uparrow +$

$-F + Fp + R_C^y + R_B^y = 0$

$-F + Fp + (0.1434 \times Fp - 3.68)$
$+ (16.56 - 0.86 \times Fp) = 0$

$Fp\ (1 + 0.1434 - 0.86)$
$= 11.04 - 16.56 + 3.68$

$$Fp = \frac{11.04 - 16.56 + 3.68}{0.2834}$$

$Fp = 6.5\ kN \downarrow$

Substitute FP in Equ.2 & Equ.3

$R_B^y = 16.56 - 0.86 \times (-6.5) \quad [Equ.2]$

$R_B^y = 22.15\ kN \uparrow$

$R_C^y = 0.1434 \times (-6.5) - 3.68 \quad [Equ.3]$

$R_C^y = 4.61\ kN$

$\Sigma F_X = 0 +\!\rightarrow$

$R_C^x + R_B^x - Fp \cos\theta_{max} = 0\ [Equ.4]$

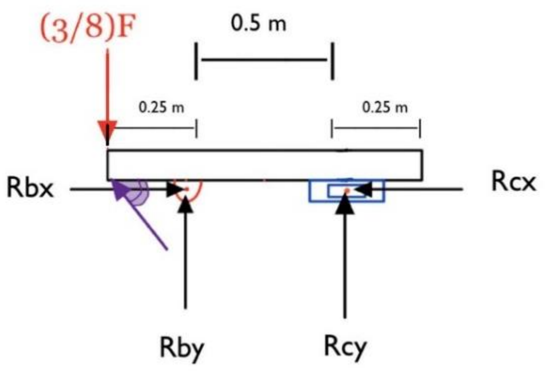



*Figure 24. FBD of upper plate*

### 3.2.3 Force of lower plate

It's important to mention that the dynamic viscosity between two alloy steel metals will not proceed a value of 0.2, nevertheless, we also considered here the worst-case scenario and made $\mu = 0.33\frac{F}{N}$ to be more on the safe side.

$+\circlearrowright\ \Sigma M_D = 0$

$(-F_R \times 0.25) - (F_N \times 0.5) = 0$

$(-11.04 \times 0.25) - (F_N \times 0.5) = 0$

$F_N = 5.52\ kN\ \downarrow$

$\Sigma \mathrm{Fy} = 0 \uparrow +$

$R_d^y = -F_N + F_R$

$R_d^y = -5.52 + 11.04$

$R_d^y = 5.52\ kN\ \downarrow$

$Hence\ \mu F_N = 0.33 \times 5.52 = 1.825\ kN$

$\Sigma \mathrm{Fx} = 0\ \rightarrow +$

$R_d^x = \mu F_N = 1.825\ kN\ \rightarrow$

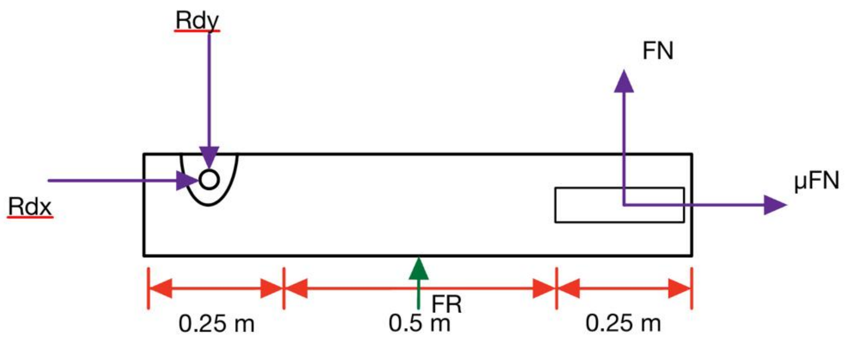



*Figure 25. FBD of lower plate*

### 5.2.1 Force of links

$\Sigma \mathrm{Fy} = 0 + \uparrow$

$R_e^y = R_C^y + R_d^y$

$R_e^y = 4.6121 + 16.57.$

$R_e^y = 21.18\ kN\ \uparrow$

$+\circlearrowright\ \Sigma \mathrm{M\ at\ point\ C} = 0$

$0.35 \times R_p^x + 0.7 \times R_d^x = 0$

$$R_p^x = -\frac{0.7}{0.3} \times 3.06$$

$R_p^x = 7.14\ kN\ \leftarrow$

$\Sigma \mathrm{Fx} = 0 \rightarrow +$

$R_C^x = R_d^x + R_p^x$

$R_C^x = 3.06 - 7.14$

$R_C^x = 4.08\ \mathrm{kN}\ \leftarrow$

substitute $R_C^x$ in equ. 4

$R_C^x + R_B^x - Fp\cos\theta_{max} = 0\ [equ. 4]$

$-4.08 + R_B^x - (6.5 \times \cos 35\,°) = 0$

$R_B^x = (6.5 \times \cos 35°) + 4.08$

$R_B^x = \ 9.40 \text{ kN } \rightarrow$

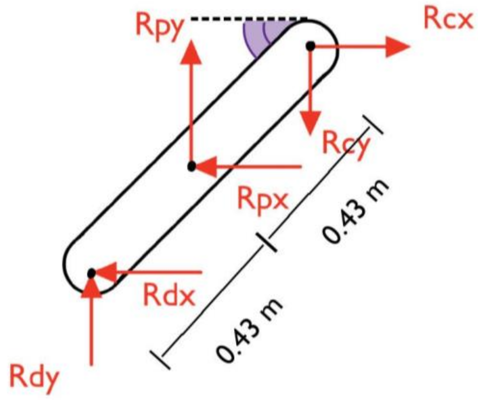



*Figure 26. FBD of right link of Jack scissor*

## 3.3 Numerical Results

$I$: *mass moment of inertia*

$t_1$: *thikness of the lower plate*

$w_1$: *width of the lower plate*

$\sigma_b$: *bending stress*

$\sigma_N$: *Normal stress*

$\tau_{xy}$: *shear stress*

$S_y$: *yeild strength*

$R_{t2}$: *magnitude of forces on point d*

$d_2$: *diameter of the shaft on point d*

$R_{t6}$: *magnitude of forces on point c*

$d_2$: *diameter of the shaft on point c*

$w_2$: *width of the pin on point 2*

$t_2$: *thikness of the first arm pin*

$w_3$: *width of the pin on point 3*

$t_3$: *thikness of the first arm pin sectional viwe*

$d_P$: *diameter of the rod inside first pin*

$w_3$: *width of the pin on point 3*

$K_t$: *correction factor in axial load due to notch*

$d_5$: *diameter of the rod on second pin arm*

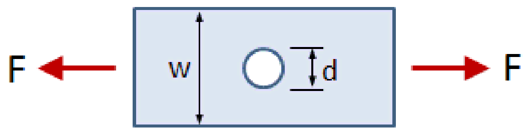
$t_6$: *thikness of second first arm pin sectional viwe*

$w_6$: *width of the pin on point 6*

$w_4$: *width of the pin on point 4*

$t_5$: *double the thikness of second first arm pin sectional viwe*

$$@\ 4140\ Alloy\ steel \rightarrow \quad nf = 3\,, \quad s_y = 620.42\ MPa$$




- w = bar width
- d = hole diameter
- t = bar thickness
- F = applied force (tensile or compressive)

**Figure A–15–1**

Bar in tension or simple compression with a transverse hole. $\sigma_0 = F/A$, where $A = (w - d)t$ and $t$ is the thickness.

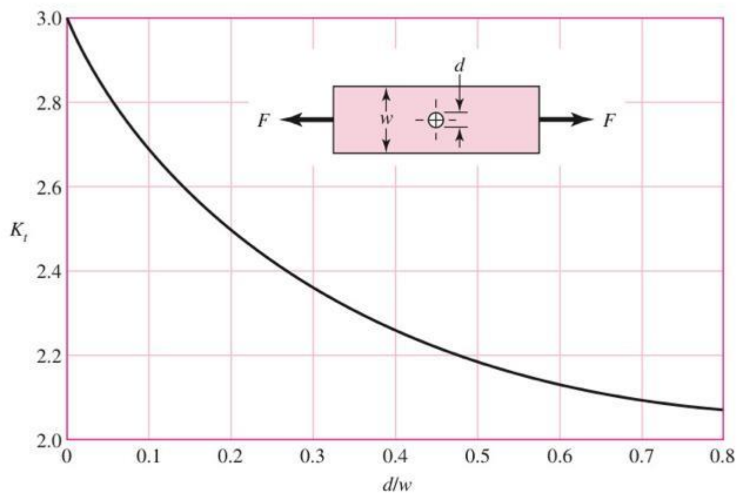
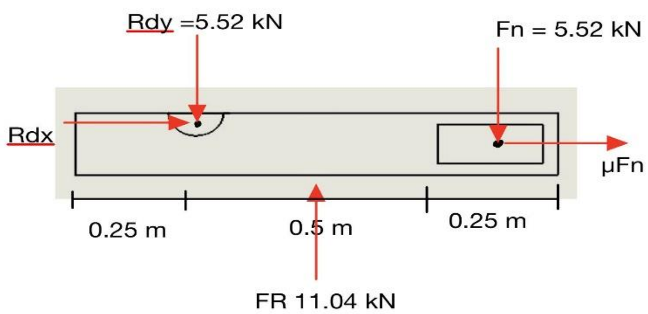
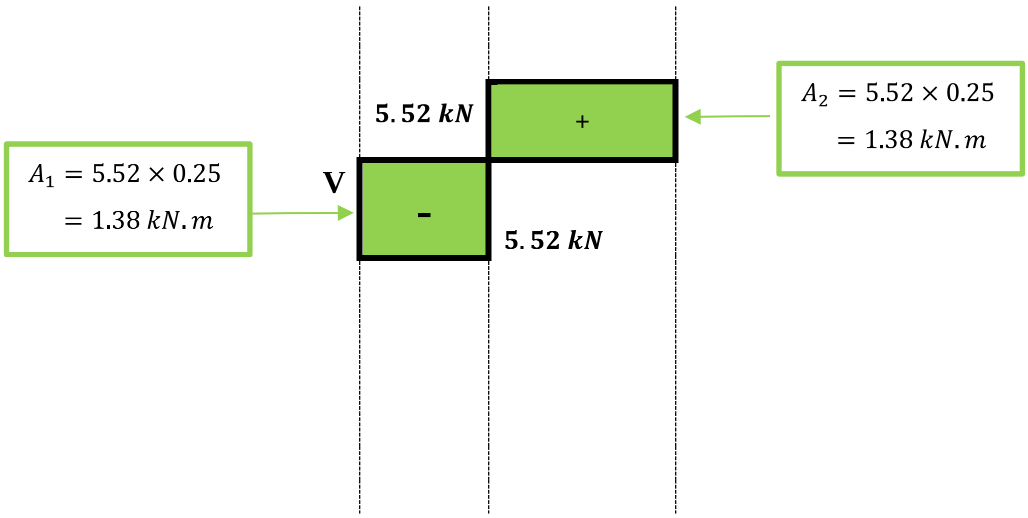



*Figure 27. Bar in tension or simple compression with a transverse hole.*

## 3.1 Stress Analysis at Lower Plate



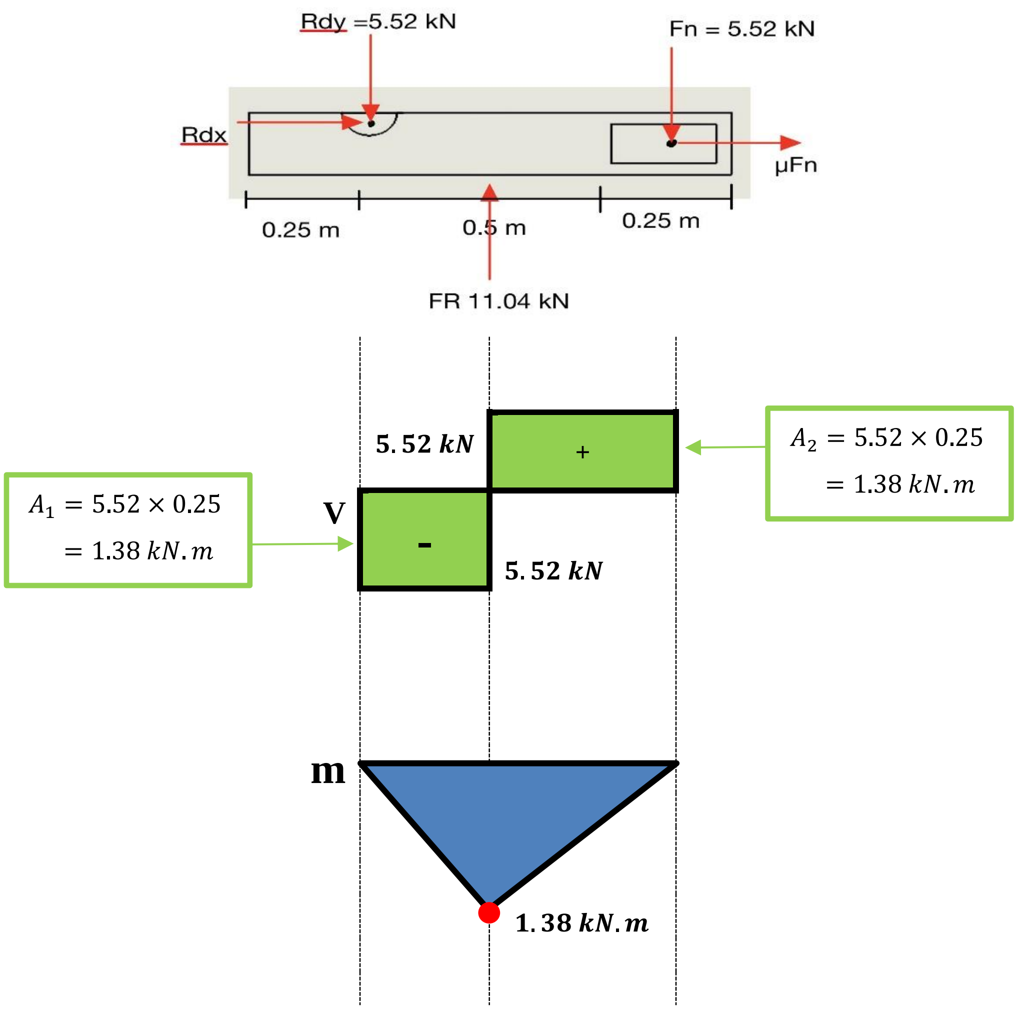



*Figure 28. Shear Force and Bending Moment Diagram at Lower Plate*

$$I = \frac{W \times t_1^3}{12} = \frac{300 \times t_1^3}{12} = 25\ t_1^3$$

$$\sigma_b = \frac{M \times {}^{t_1}/_2}{I} = \frac{1.38 \times 10^6 \times {}^{t_1}/_2}{125 t_1^3} = 55200\ t_1^{-2} \quad {}^{N}/_{mm^2}$$

$$\sigma_N = \frac{R_y}{A} = \frac{R_y}{w \times t_1} = \frac{5.52 \times 10^3}{300 \times t_1} = 18.4\ t_1^{-1} \quad {}^{N}/_{mm^2}$$

$$\tau = \frac{3V}{2A} = \frac{(3 \times 5.58 \times 1000)}{600 \times t_1} = 27.9\ t_1^{-1} \quad {}^{N}/_{mm^2}$$

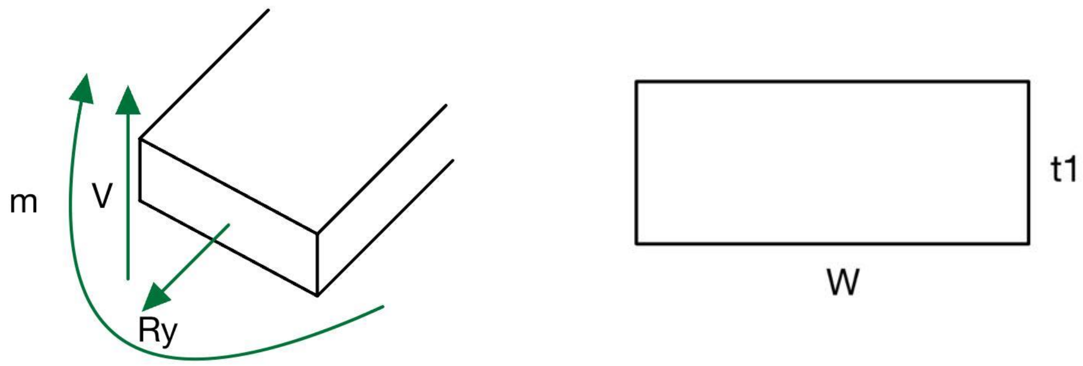



*Figure 29. Stress Analysis on Lower plate*

### 3.1.1 Stress Analysis at Rectangular Shape

$$\sigma_v = \sqrt{\sigma_x^2 - \sigma_x\sigma_y + \sigma_y^2 + 3\tau_{xy}^2}\,, \quad nf = \frac{s_y}{\sigma_v} \rightarrow \sigma_v = \frac{s_y}{nf}$$

Case 1: At the Edge

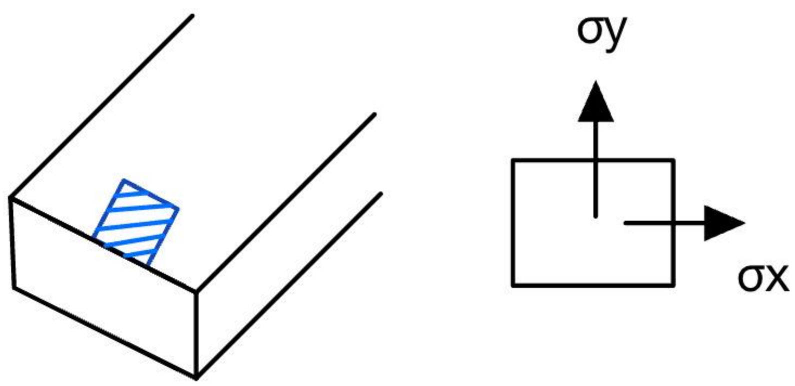



*Figure 30. Element study on lower plate*

$\tau_{xy} = 0\,, \qquad \sigma_y = 0\,, \qquad \sigma_x = \sigma_b + \sigma_N = 55200\, t_1^{-2} + 18.4\, t_1^{-1}$

$$\frac{s_y}{nf} = \sqrt{\sigma_x^2 - \sigma_x\sigma_y + \sigma_y^2 + 3\tau_{xy}^2}$$

$$\frac{620.42}{3} = \sqrt{(55200\, t_1^{-2} + 18.4\, t_1^{-1})^2 - 0 + 0 + 0}$$

$t_1 = 16.38\, mm \approx 20\, mm$

Case 2: At the middle

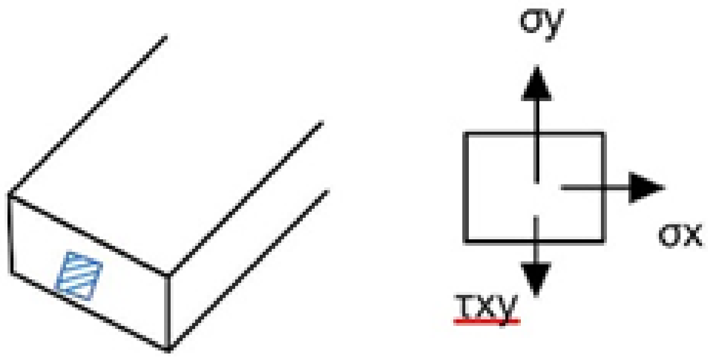



*Figure 31. Shear on the*

$$\tau_{xy} = \frac{3V}{2A} = 27.9\, t_1^{-1}\,, \qquad \sigma_y = 0\,, \qquad \sigma_x = \sigma_N = 18.4\, t_1^{-1}$$

$$\frac{s_y}{nf} = \sqrt{\sigma_x^2 - \sigma_x\sigma_y + \sigma_y^2 + 3\tau_{xy}^2}$$

$$\frac{620.42}{3} = \sqrt{(18.4\, t_1^{-1})^2 - 0 + 0 + 3(27.09)^3}$$

$$t_1 = 0.25\, mm$$

$$Hence, the\ thickness\ of\ the\ palte\ is = \ 20\ mm$$

## 3.2 Stress Analysis at Roller Arm

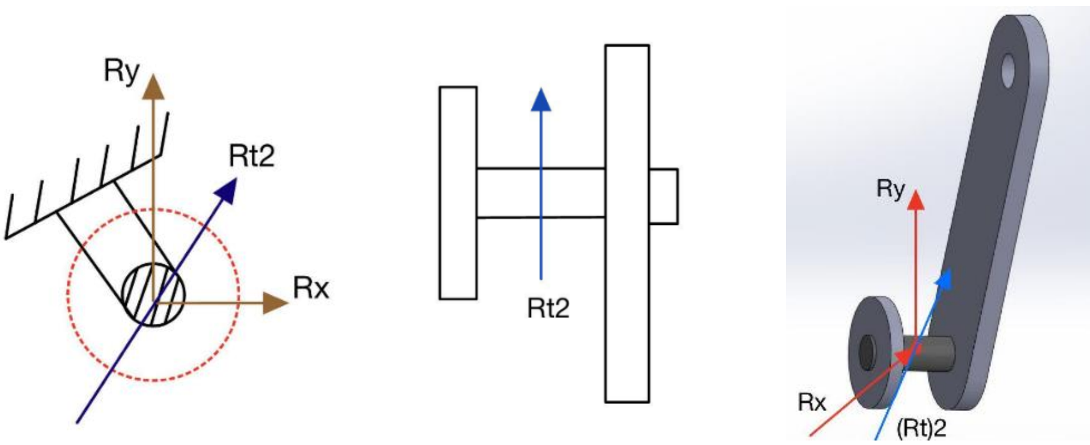



*Figure 32. Shear on Roller Arm*

$(R_t)_2 = \sqrt{R_y^2 + R_x^2} = \sqrt{16.75^2 + 1.825^2} = 16.67\ kN$

it can be noticed that it will cause shearing first, so we can calculate d

$(R_t)_2$will cause direct shear + bending moment.

$I_{direct} = \frac{F}{A} = \frac{(R_t)_2}{\frac{\pi}{4} d_2^2}$

$use\ VON\ mises\ to\ get\ d_2$

$u = (R_t)_2 \times x$

$\sigma_b = \frac{My}{I} = \frac{M\left(\frac{d_2}{2}\right)}{\frac{\pi}{64} d_2^4} = \sigma_x$

$\sigma_b = \frac{(R_t)_2 \times 2 \times 10}{\frac{\pi}{64} d_2^4}$

$(R_t)_2 = 16.67\ kN$

$\sigma_{Prime} = \sqrt{\sigma_x^2 - \sigma_x \sigma_y + \sigma_y^2 + 3\tau_{xy}^2} \rightarrow \sigma_y$

$\sigma_b = \sigma_x\ 6.792 \times 10^6 (d_2^4)^{-1}\ MPa$

$\tau_{direct} = \tau_{xy} = 21.22 \times 10^3 (d_2^2)^{-1} MPa$

$\sigma_{Prime} = \sqrt{(6.792 \times 10^6 \times (d_2^4)^{-1})^2 - 0 + 0 + 3(21.22 \times 10^3 (d_2^2)^{-1})^2}$

$\therefore d_2 = 15.12\ mm$

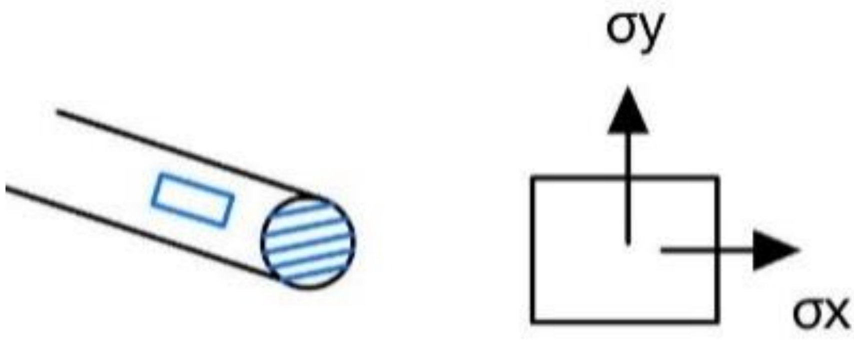



*Figure 33. Element steady on Dimeter of Pin*

## 3.3 Stress Analysis at Second Roller Arm

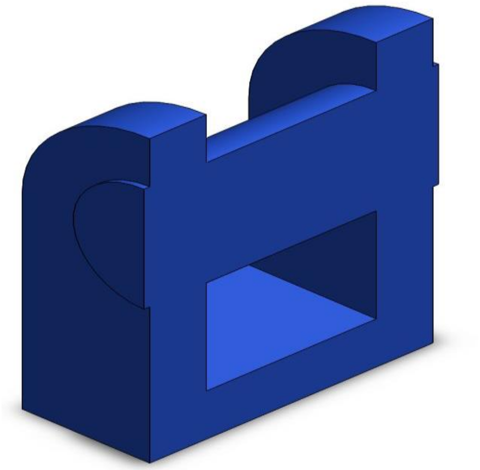


*Figure 34. Shear on Second Roller Arm*

$$(R_t)_6 = \sqrt{R_c^{y^2} + R_c^{x^2}}$$

$$(R_t)_6 = \sqrt{4.08^2 + 4.61^2} = 6.156\text{kN}$$

$$\tau_{\text{direct}} = \frac{F}{A} = \frac{(R_t)_6}{\frac{\pi}{4}d_\sigma^2} = \frac{6.156 \times 10^3}{\frac{\pi}{4}d_6^2} = 7838.063 d_\sigma^{-2} = \tau_{xy}$$

$$M = (R_t)_6 \cdot x$$

$$\sigma_{\text{bend}} = \frac{My}{I} = \frac{\frac{Md_6}{2}}{\frac{\pi}{64}d_6^4} = \sigma_x$$

$$\sigma_{\text{bend}} = \frac{(R_t)_6 \cdot 20}{\frac{\pi}{32} \cdot d_\sigma^3} = \frac{6.156 \times 10^3 \cdot 20}{\frac{\pi}{32} \cdot d_6^3} = 1254090.022 d_6^{-3} = \sigma_x$$

$$\sigma' = \sqrt{(1254090.022 d_6^{-3})^2 - 0 + 0 + 3(7838.063 d_6^{-2})^2} = \frac{s_y}{n_f} = \frac{620.42}{3}$$

$$d_6 = 18.4 \text{ mm}$$

$$\sigma_{\text{bearing}} = \frac{(R_t)_6}{\text{Aprojected}} = \frac{(R_t)_6}{tw \times d_6} = \frac{s_y}{n_y}$$

$$\frac{6.156 \times 10^3}{tw \times 18.4} = \frac{620.42}{3}$$

$$tw = 1.62 \text{ mm}$$

## 3.4 Stress analysis on the first pin arm

We separate the two bodies to get $d_P$ first and then t2 value.

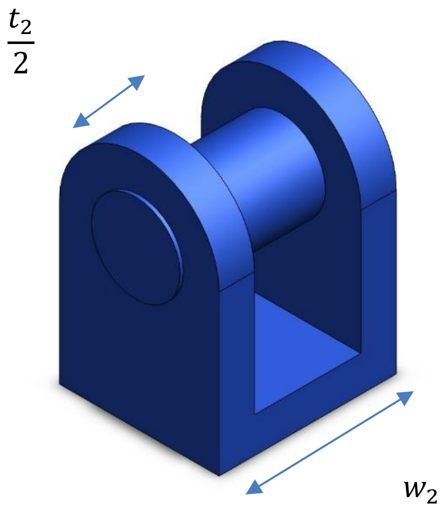



*Figure 35. Shear on the first pin arm*

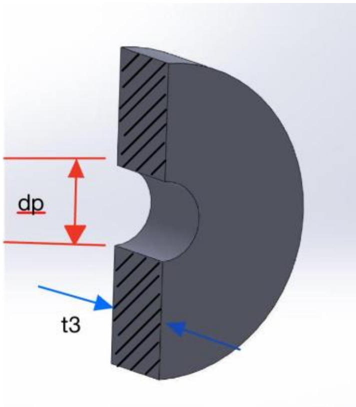



*Figure 36. Sectional side view of the pin slot*

$$\tau_{\text{Direct}} = \tau_{\text{Double}} = \frac{(Rt)_3}{2 \times \frac{\pi}{4} d_p^2}$$

$$M = (Rt)_3 \times L_2$$

$$\sigma_{6en} = \frac{My}{I}$$

$$\frac{M\left(\frac{dp}{2}\right)}{\frac{\pi}{64} d_p^4} = 759.93\text{t}_2$$

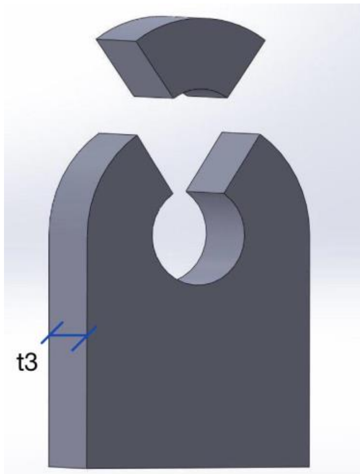



*Figure 37. Another cause of failure due to tension*

$$\tau_{\text{Direct}} = \frac{\frac{(Rt)_3}{2}}{\frac{\pi}{4} d_p^2} = \frac{Ssy}{n_f} \rightarrow \frac{\frac{6.158 \times 10^3}{2}}{\frac{\pi}{4} \times d_p^2} = \frac{620.42}{3}$$

$$\therefore d_p = 4.354 \text{ mm}$$

$$\sigma_{bearing} = \frac{(Rt)_3}{A_{\text{projected}}} = \frac{(Rt)_3}{dp \times t_3} = \frac{S_y}{nf}$$

$$\frac{6.158 \times 10^3}{4.354 \times t_3} = \frac{620.42}{3}$$

$t_3 = 6.839$ mm

$\sigma_{\text{bearing}} = 206.80$ MPa

$$\sigma_{\text{tension}} = \frac{(R_t)_3}{(w_3 - d_\rho) \times t_3} = \frac{Sy}{nf}$$

$$= \frac{(6.158 \times 10^3)}{(w_3 - 4.354) \times 6.839} = \frac{620.42}{3}$$

$\therefore w_3 = 8.70$ mm

Another cause of failure: the distance will almost be equal to the radius of the circle.

First, we will assume a correction factor which will be $K_t$ to be equal to 2.5, and then we will recalculate the factor to get the exact value of it.

$$\tau_{2D} = \frac{\frac{(Rt)_3}{2}}{2 \times t_3 \times \frac{w_3}{2}} = \frac{S_y}{n_f} \longrightarrow \frac{\frac{6.158 \times 10^3}{2}}{2 \times 6.839 \times \frac{w_3}{2}} = \frac{620.42}{3}$$

$$= \frac{450.212}{w_3} = 206.81 \longrightarrow w_3 = 2.177 \text{ mm}$$

$$\sigma_t = \frac{(R_t)_3}{(w_2 - d_p)t_2} \times k_t = \frac{S_y}{n_f}$$

$$\frac{6.158 \times 10^3}{(w_2 - 4.354) \times 13.678} \times 2.5 = \frac{620.42}{3}$$

$\therefore w_2 = 9.794$ mm

$$\frac{d_p}{w} = \frac{4.354}{9.794} = .45$$
$$k_t = \frac{d_p}{w_2} = \frac{4.354}{9.794} = 0.47$$
$$\therefore k_t = 2.2$$
$$w_t = 9.25$$

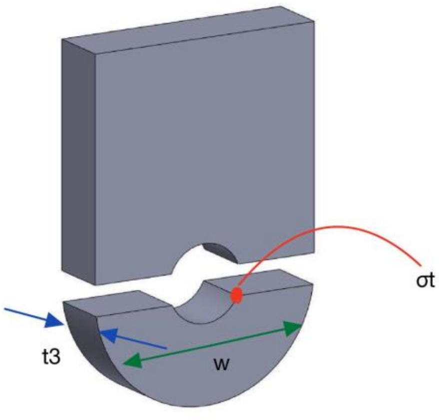



*Figure 38. bending stress*

## 3.5 Stress analysis on the second pin arm

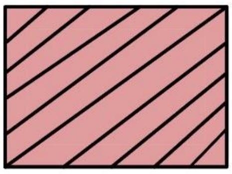



*Figure 39. Shear on the second pin arm*

$R_B^y = 22.15\text{kN}$

$R_B^x = 9.4\text{kN}$

$R_{t5} = \sqrt{R_B^{x^2} + R_B^{y^2}}$

$(R_t)_5 = \sqrt{9.4^2 + 22.15^2} = 24.1\text{kN}$

$$\tau_{\text{Direct}} = \tau_{\text{Double}} = \frac{(R_t)_5}{\frac{2\pi}{4} d_p^2} = \frac{S_{S_y}}{\eta_f} \to \frac{24.1 \times 10^3}{\frac{\pi}{2} dp^2} = \frac{310.21}{3}$$

$M = (R_t)_5 \times t_5$

$d_P = 12.2 \text{ mm}$

$$\sigma_{\text{bend}} = \frac{\text{My}}{I} = \frac{(R_t)_5 \cdot t_5 \times \frac{d\rho}{2}}{\frac{\pi}{64} dp^4} = \frac{24.1 \times 10^3 \cdot t_5 \times \frac{12.2}{2}}{\frac{\pi}{64} \cdot (12.2)^4} = 135.188 t_5$$

$$\sigma_{\text{bering}} = \frac{(R_t)_5}{A_{\text{ projection}}} = \frac{(R_t)_5}{d\rho \times t_6} = \frac{s_y}{y_y}$$

$$\frac{24.1 \times 10^3}{12.2 \times 66} = \frac{620.42}{3}$$

$t_6 = 9.552 \text{ mm}$

$$\sigma_{\text{tension}} = \frac{(R_t)_5}{(w_5 - d_\rho)_{xy}} = \frac{s_y}{n_f}$$

$$= \frac{24.1 \times 10^3}{(65 - 12.2) \times 9.552} = \frac{620.42}{3}$$

$w_5 = 24.4$ mm

$$(\&\tau_{2D} = \frac{\frac{(R_t)_5}{2}}{2 \times t_6 \times \frac{w_5}{2}} = \frac{s_{5y}}{\eta_8} \longrightarrow \frac{\frac{24.1\times10^3}{2}}{2 \times 9.552 \times \frac{w_5}{2}} = \frac{310.21}{3}$$

$w_5 = 12.2$ mm

$k_t = 2.5$

$$\sigma_{6r} = \frac{(R+)_5}{dp \times t_5}\ 1\ t_5 = 2t_6$$

$\therefore t_5 = 2 \times 9.552 = 19.104$ mm

$$\sigma_t = \frac{(R)_5}{(w_4 - d_\rho)t_5} \times k_t = \frac{\mathrm{S}_y}{n_f}$$

$$\frac{24.1 \times 10^3}{(w_4 - 12.2).\,19.104} \times 2.5 = \frac{620.42}{3}$$

$\therefore w_4 = 27.45$ mm

$k_t = 2.5$ and $\frac{d_p}{\omega_4} = 0.2$

$$\frac{dp}{W_4} = \frac{12.2}{27.45} = 0.44.$$

$$k_4 = 2.2\ \frac{d\rho}{\omega_4} = 0.47$$

$$w_4 = 25.62 \text{ mm},\ \frac{dp}{w_4} = \frac{12.2}{25.62} = 0.476$$

## 3.6 Two fixed ends rod

The distance between the two ends is equal to 30 cm as shown before in the force analysis.

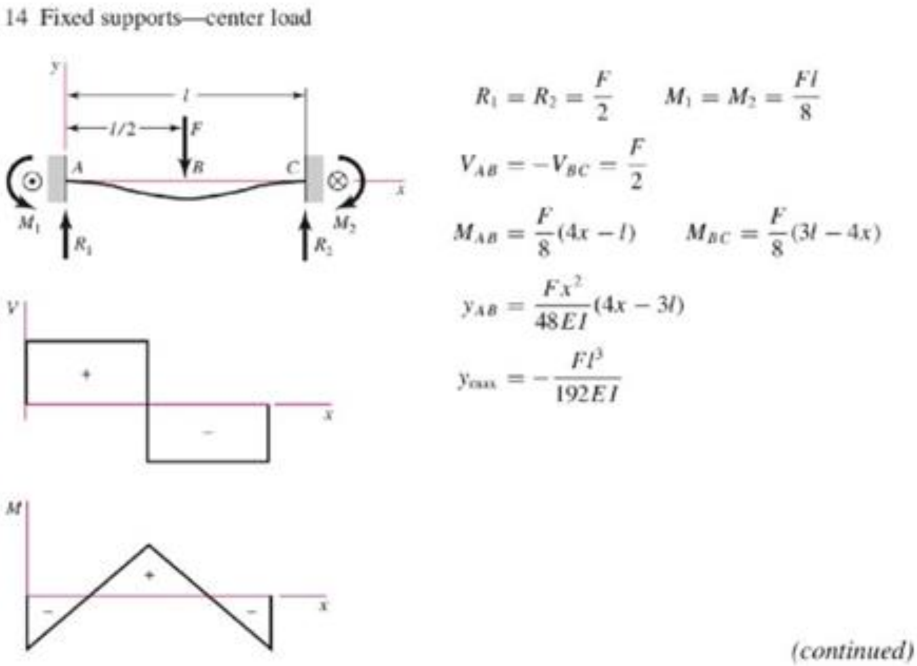



*Figure 40. Case of study of fixed rod*

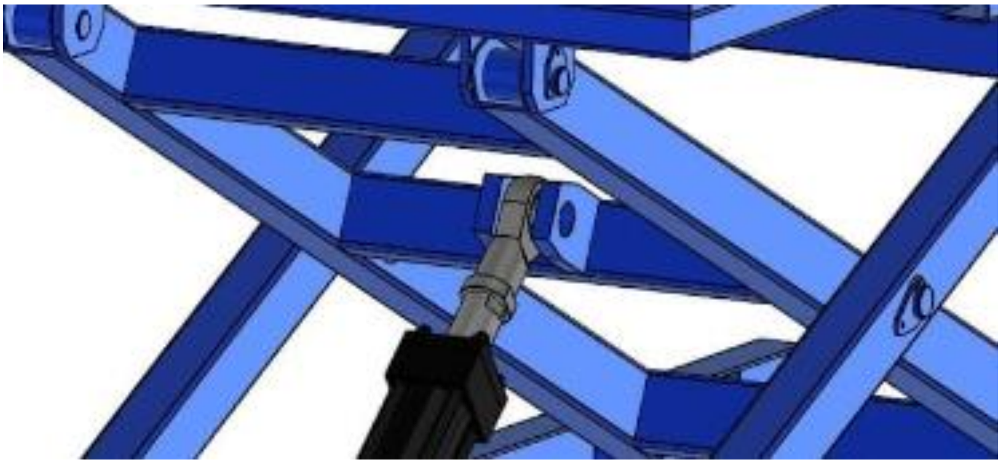


*Figure 41. Two ends fixed rod fixed rod*

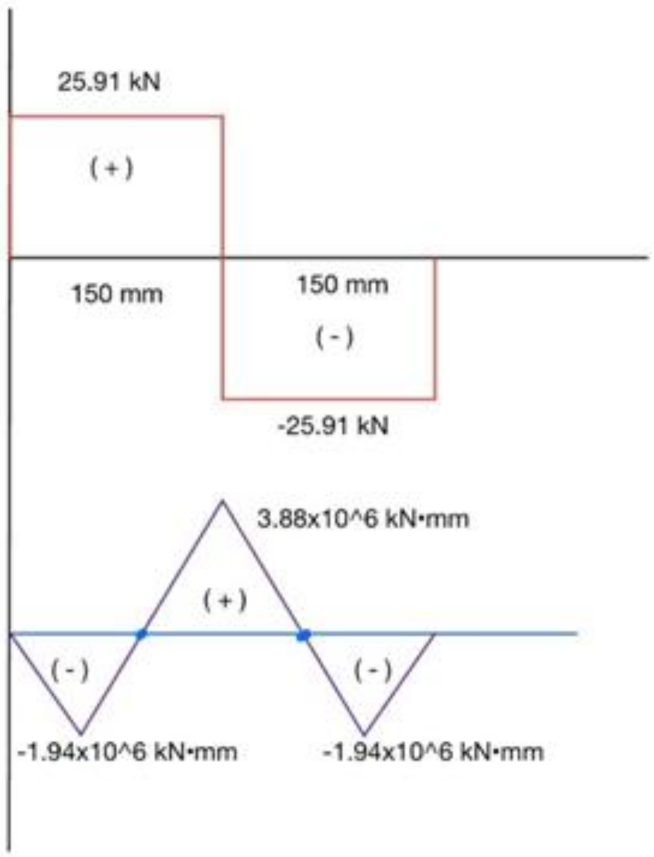



*Figure 42. SFD and BMDs for the middle rod*

$$\sigma_x = \frac{My}{I} \quad \Rightarrow M = \frac{F_p}{8}(4x - l)$$

$$M = \frac{6.51}{8}(4(150) - 300)$$

$$M = 243.75 \times 10^3 \text{ N.mm}$$

$$\sigma_x = \frac{243.75 \times 10^3 \times \frac{dsh}{2}}{\frac{\pi}{64} d_{sh}^4}$$

$$= \frac{2.48 \times 10^6}{d_{sh}^3}$$

$$\tau_{xy} = \frac{F}{A} = \frac{(R_t)_4}{\frac{\pi}{4} d_{sh}^2} = \frac{25.91 \times 10^3}{\frac{\pi}{4} d_{sh}{}^2} = \frac{32.99 \times 10^3}{d_{sh^2}^2}$$

$$\sigma = \sqrt{\sigma_x^2 - \sigma_x\sigma_y + \sigma_y^2 + 3\tau_{xy}^2} \longrightarrow \sigma_y = 0 = \frac{S_y}{nf}$$

$$\sqrt{\left(\frac{2.48\times10^6}{d_{sh}^3}\right)^2+3\left(\frac{32.99\times10^3}{d_{sh^2}}\right)^2}=\frac{620.42}{3}$$

$\therefore d_{sh} = 23.91$ mm

## 3.7 Stress analysis for middle scissor lift rod welding

The distance between the two ands of the middle rod is equal to 30 cm.

$$\therefore d_{sh} = 154.95 \text{ mm}$$

$$(R_\text{t})_4 = \sqrt{R_x^2 + {R_y}^2}$$

$$= \sqrt{24.91^2 + 7.14^2}$$

$$= 25.91\text{kN}$$

$$\therefore\ \tau_{xy} = \frac{R_{t4}}{2\pi r_\omega \times .707 \times h}, \text{r}_\omega = 11.955 \text{ mm}$$

$$= \frac{487.88}{h}$$

$$\sigma_\text{b} = \frac{My}{I} = \frac{R_\text{t4}\times(150)}{I_u \times 0.707 \times h} = \frac{R_\text{t4}\times 150}{\pi r^3 \times .707 \times h}$$

$$= \frac{1024.09}{h}$$

$$\sigma^r = \sqrt{\left(\frac{1024.09}{h}\right)^2 + 3\left(\frac{487.88}{h}\right)^2} = \frac{462.42}{3}$$

$$h = 6.420 \text{ mm}$$

## 3.8 Stress analysis at the second arm pin welding

$$\tau_{xy} = \frac{(Rt)_2}{2\pi r_w \times 0.707h}$$

$$(\tau_{xy}) = \frac{(R_t)_2}{2\pi r_w \times 0.707 \times h} = \frac{16.67\times10^3}{2\pi\left(\frac{15.12}{2}\right)\times 0.707\times h}$$

$$d_2 = 15.12 \text{ mm}, = \frac{313.89}{\text{h}} = (\tau_{xy})$$

$$\sigma_b = \frac{My}{I} = \frac{MR}{I} = \frac{MR}{I\times0.707\times h} = 0$$

$$\left(\tau_{xy}\right) = \frac{313.89}{h}, \sigma_{\text{bend}} = 0$$
$$\therefore \sigma_y = 0$$

$$\therefore \sigma' = \sqrt{(0)^2 + 3\left(\frac{313.89}{h}\right)^2} = \frac{462}{3} = \frac{(Sy)_w}{(nf)_w}$$
$$\therefore h = 2.63 \text{ mm}$$

## 3.9 Stress Analysis at Welding for Rolling Rod

We assumed that the distance between the slider and the joint is equal to 2cm.

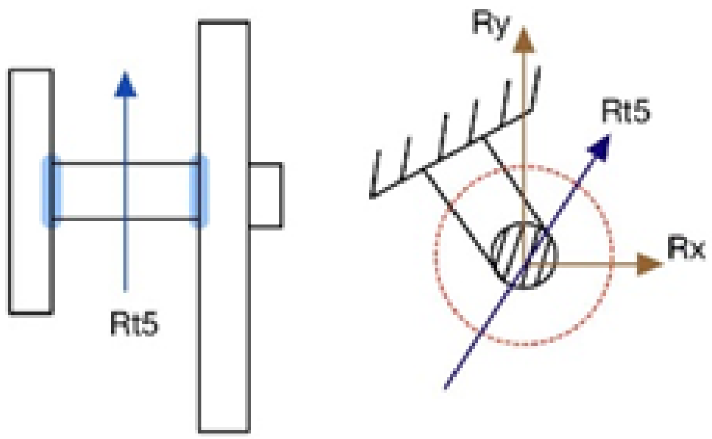



*Figure 43. Shear on the*

$$Rt_5 = \sqrt{R_{Bx} + R_B y}$$
$$R_{t5} = 24.06\text{kN}$$
$$\therefore \left(\tau_{xy}\right) = \frac{R_{t5}}{2\pi r_\omega \times .707 \times h}, r_\omega = 15.12 \text{ mm}$$
$$= \frac{716.43}{h}$$
$$\sigma_{\text{b}} = \frac{My}{I} = \frac{R_{t_5} \times (20)}{I_{\text{u}} \times 0.707 \times h} = \frac{Rt_5 \times 20}{\pi r^3 \times .707 \times h}$$
$$\therefore \frac{501.41}{h}$$
$$\sigma^r = \sqrt{\left(\frac{501.41}{h}\right)^2 + 3\left(\frac{716.43}{h}\right)^2} = \frac{462.42}{3}$$
$$h = 6.47 \text{ mm}$$

# 4 SYSTEM SIMULATION AND MANUFACTURING

In this design project there will be manufacturing of a real model, and simulation of the whole system will be done using solid works and ANSYS programs.

## 4.1 System Specifications and CAD Modelling

The components of the mechanism are divided into seven major components: the upper plate that connects to the vehicle body, the middle link, which is a massless link, the lower plate that descends to the ground when the mechanism is actuated to raise the vehicle, two split links, and two hydraulic, one of them is utilized for elevating and lowering the mechanism, while the other is used to move it forward and backward.

- Upper plate

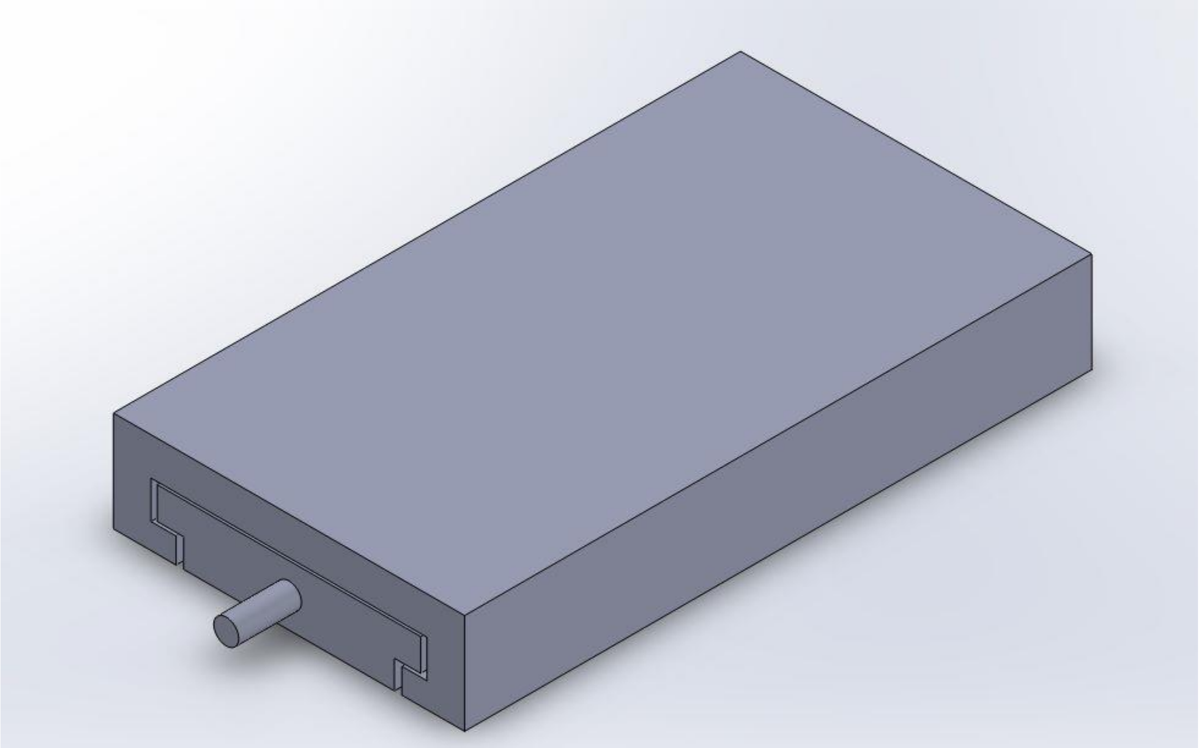


*Figure 44. Upper plate drawing*

- Lower plate

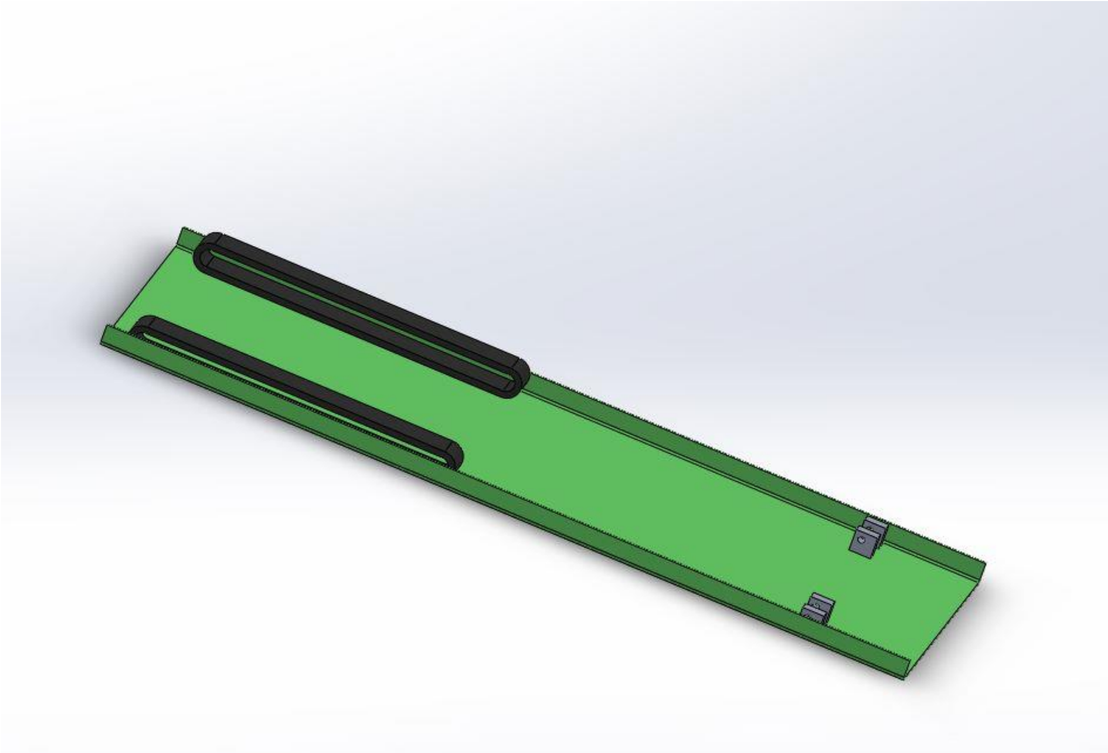


*Figure 45 . Lower plate drawing*

- Middle link

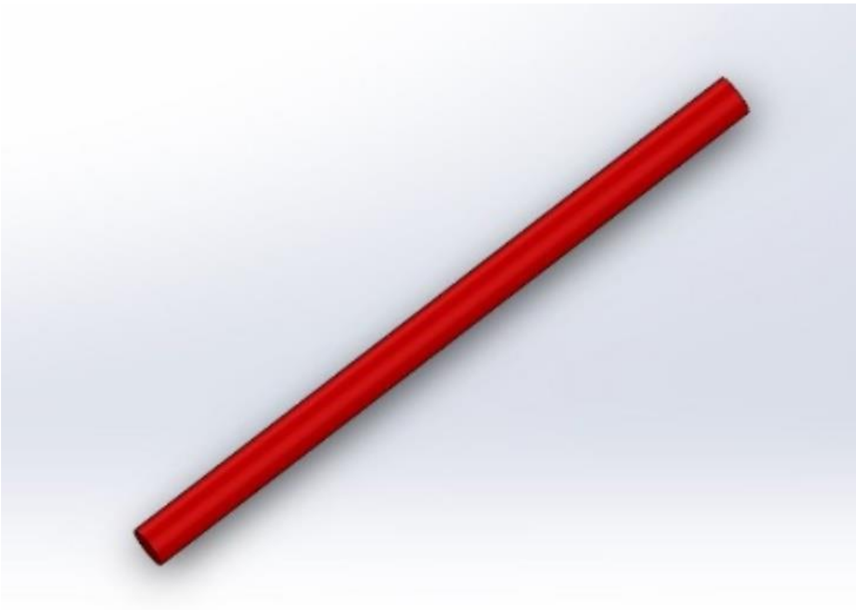


*Figure 46 . Middle link drawing*

- Two split links

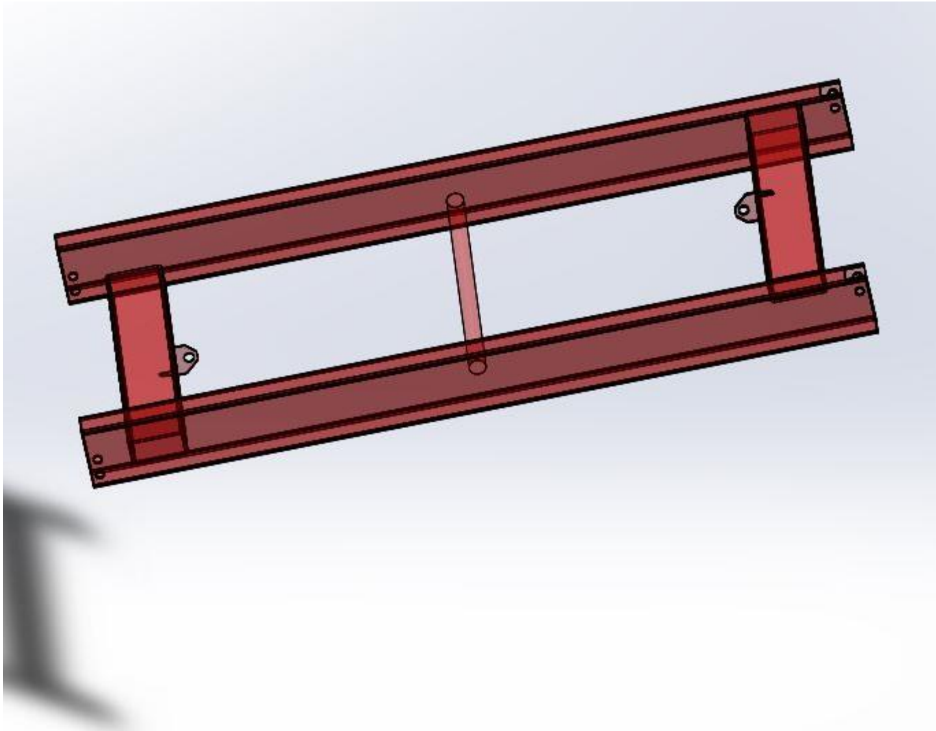


*Figure 47 . First split link drawing*

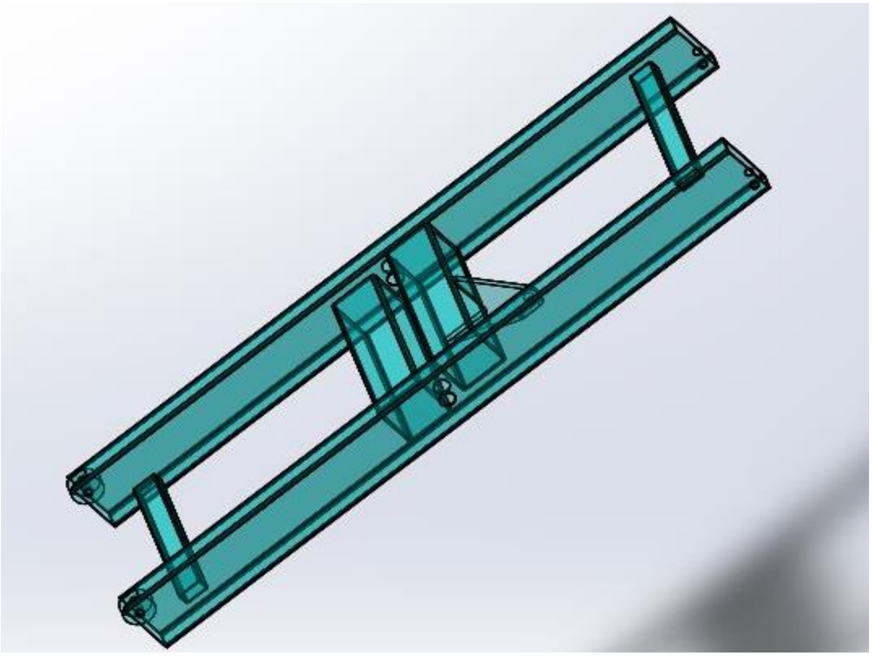


*Figure 48 . Second split link drawing*

- Hydraulic

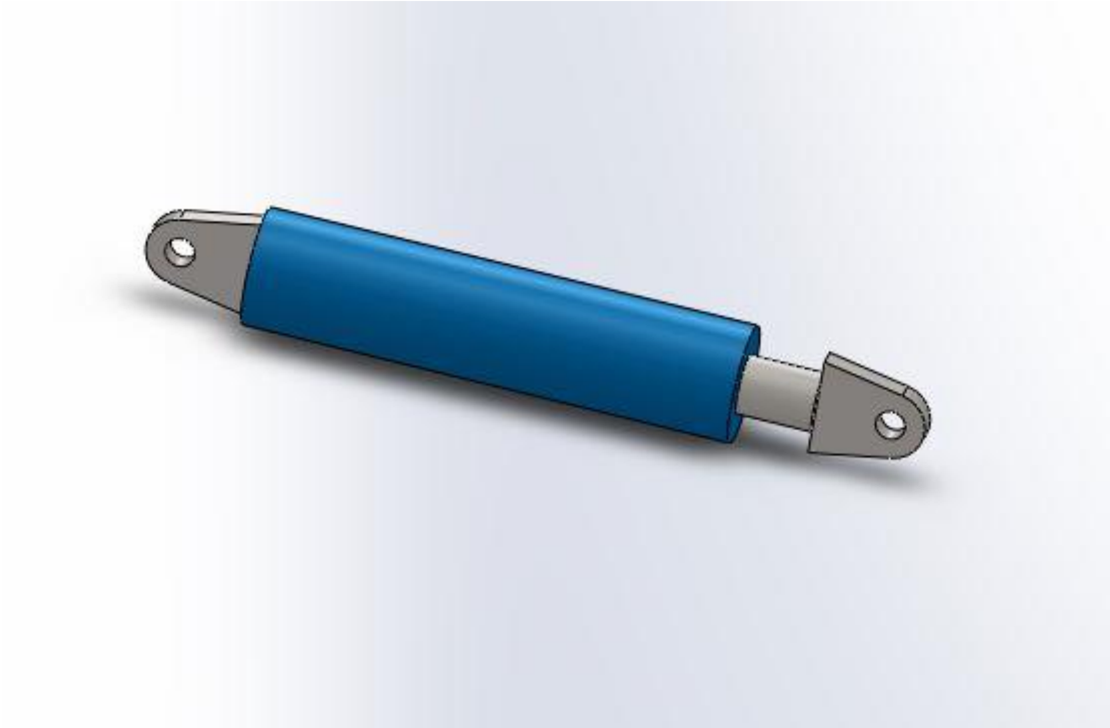


*Figure 49 . Hydraulic drawing*

- Connecting screws

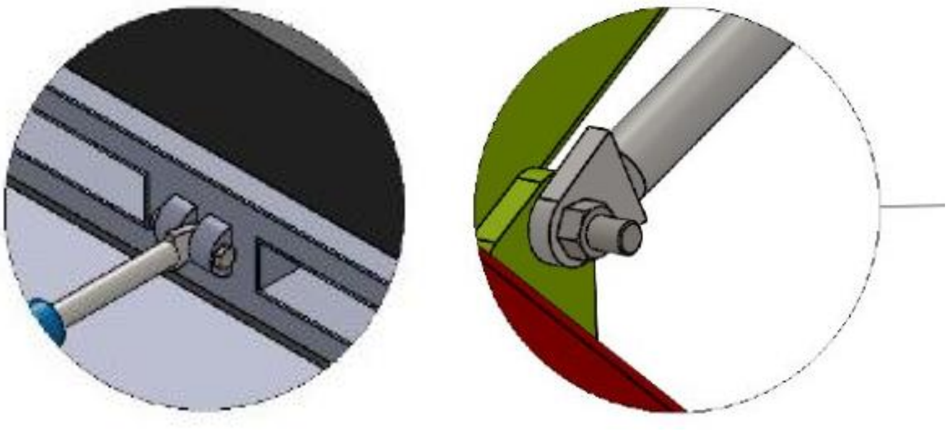


*Figure 50 . Connecting screws drawing*

## 4.2 Simulation Procedure and Settings

Simulations are conducted for each major component of the mechanism, followed by assembling them to ensure the proper functionality of the complete mechanism, as illustrated in the figure below.

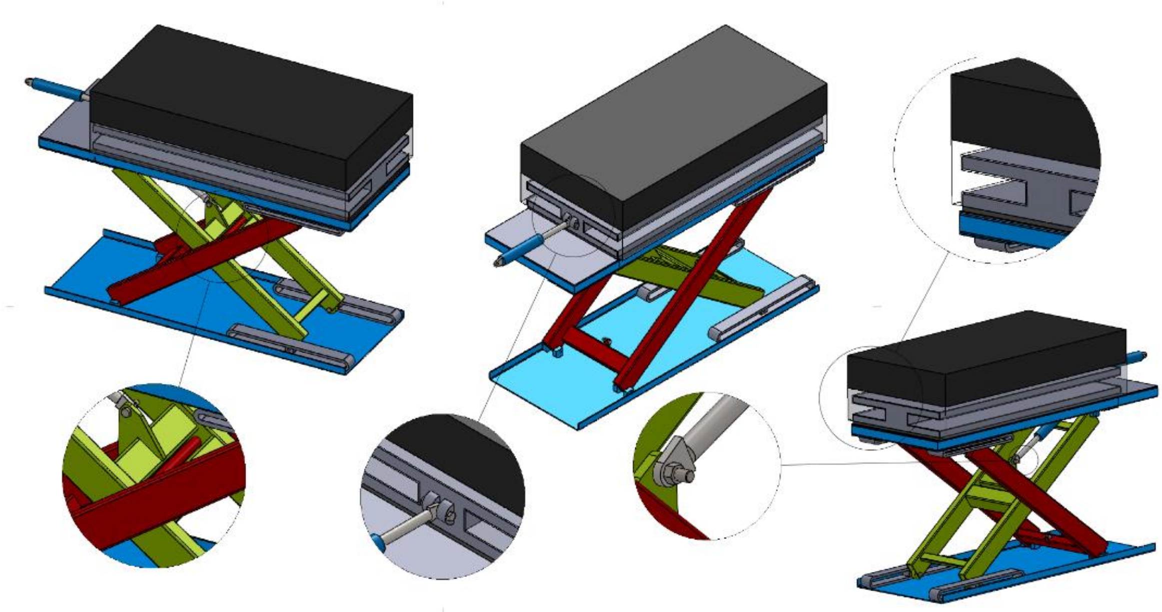


*Figure 21 . Hydraulic scissor lift mechanism.*

## 4.3 Simulation Results

The following figure shows the stress variation in each part of the mechanisms and in some combined parts using ANSYS for the validation purpose. In the following tables, the mechanical properties of the adopted materials are listed down. The stresses at the critical points of the mechanism are also listed below.

### 4.3.1 Material Selection

The choice of material was 4140 Alloy Steel due to some points that can fit perfectly to our aim in the project. The requirements were the common use of the material and the previous experience.

Table 7. Property of the 4140 Alloy Steel

| Property | Elastic modulus | Poisson's ratio | Shear Modulus | Mass density | Tensile strength | Yield strength | Thermal conductivity |
|---|---|---|---|---|---|---|---|
| value | 210 GPa | 0.28 | 79 GPa | 7700 $g/m^3$ | 723.8256 MPa | 620.422 MPa | 50 W/(m.K) |

### 4.3.2 Simulation Result of Links

In this section the critical points of the links stress, strain, deformation and life cycles were obtained as shown in the Table 8 and Figures 27-29.

Table 8. Critical Results of links

| Property | Minimum | Maximum | Average |
|---|---|---|---|
| Stress [MPa] | $1.1147 \times 10^{-12}$ | 235.31 | 19.924 |
| Strain [mm/mm] | $1.7682 \times 10^{-17}$ | $1.4519 \times 10^{-3}$ | $1.0416 \times 10^{-4}$ |
| Deformation [mm] | 0 | 1.2249 | 0.34166 |
| Life [cycles] | 14448 | $1.0 \times 10^{6}$ | $9.9676 \times 10^{5}$ |

The table shows the critical results of links, which are the minimum, maximum, and average values of stress, strain, deformation, and life for a set of links.

Stress is the force per unit area that is applied to the link. The minimum stress observed is $1.1147 \times 10^{-12}$ MPa, while the maximum stress recorded is 235.31 MPa. On average, the stress value is 19.924 MPa.

Strain is the measure of deformation or elongation experienced by the link relative to its original length. The minimum strain observed is $1.7682 \times 10^{-17}$ mm/mm, and the maximum strain is $1.4519 \times 10^{-3}$ mm/mm. The average strain is $1.0416 \times 10^{-4}$ mm/mm.

Deformation is the total deformation or displacement experienced by the link, measured in millimeters (mm). The deformation ranges from a minimum of 0 mm to a maximum of 1.2249 mm. The average deformation is 0.34166 mm.

Life is the number of cycles or repetitions of loading that the link can withstand before failure. It is given in cycles. The minimum life observed is 14448 cycles, while the maximum life is $1.0 \times 10^6$ cycles. The average life is $9.9676 \times 10^5$ cycles.

These values provide information about the mechanical behavior of the links, including their stress, strain, deformation, and expected lifespan under loading conditions.

The links in this study have a good mechanical behavior. They are able to withstand a significant amount of stress, strain, and deformation before they fail. The links also have a long life under loading conditions. These results suggest that the links are a suitable material for a variety of applications.

Here are some additional details about the properties of the links:

The stress values in the table show that the links are able to withstand a significant amount of force before failure. The average stress that the links can withstand is 19.924 MPa. This means that the links can be used in applications where they will be subjected to high levels of stress.

The strain values in the table show that the links can deform significantly before they fail. The average strain that the links can withstand is $1.0416 \times 10^{-4}$ mm/mm. This means that the links can be used in applications where they will be subjected to a lot of bending or stretching.

The deformation values in the table show that the links can move a significant distance before they fail. The average deformation that the links can withstand is 0.34166 mm. This means that the links can be used in applications where they will be subjected to a lot of movement.

The life values in the table show that the links have a long lifespan. The average life of the links is $9.9676 \times 10^5$ cycles. This means that the links can be used in applications where they will be subjected to a lot of wear and tear.

Overall, the links in this study have a good mechanical behavior. They are able to withstand a significant amount of stress, strain, deformation, and wear and tear before they fail. These results suggest that the links are a suitable material for a variety of applications.

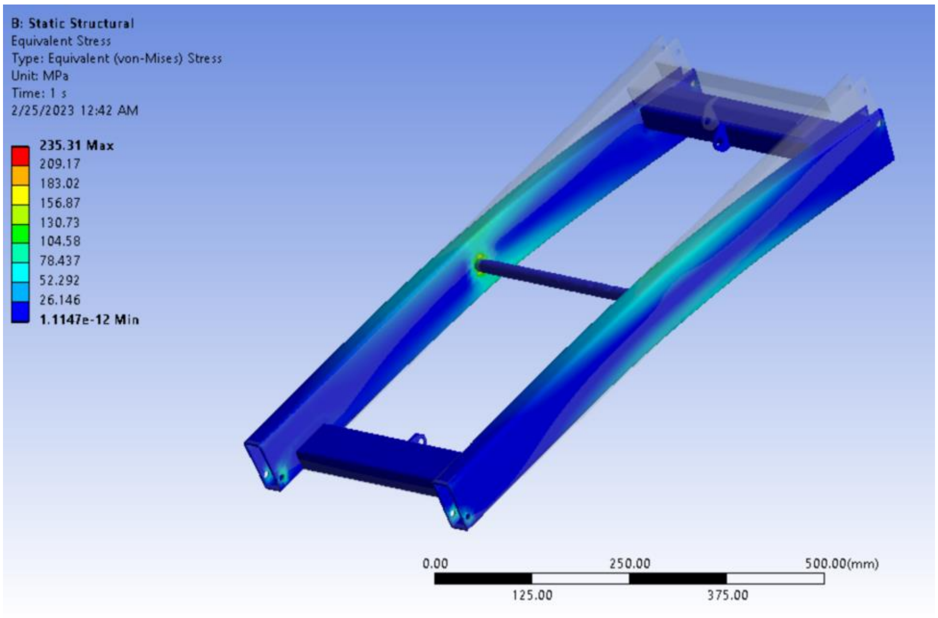



*Figure 51. Equivalent Stress of link.*

*Figure 52. Equivalent Elastic Strain of link.*

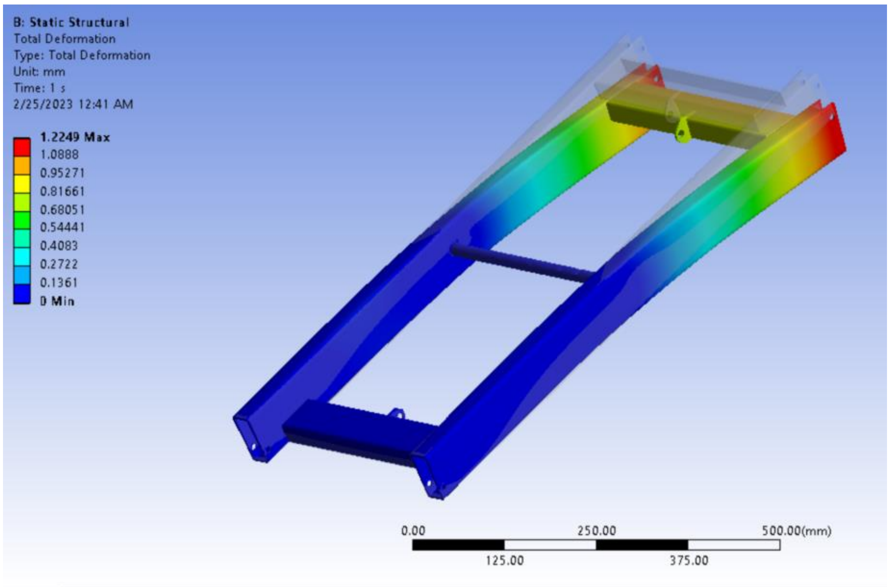



*Figure 53. Total Deformation of link*

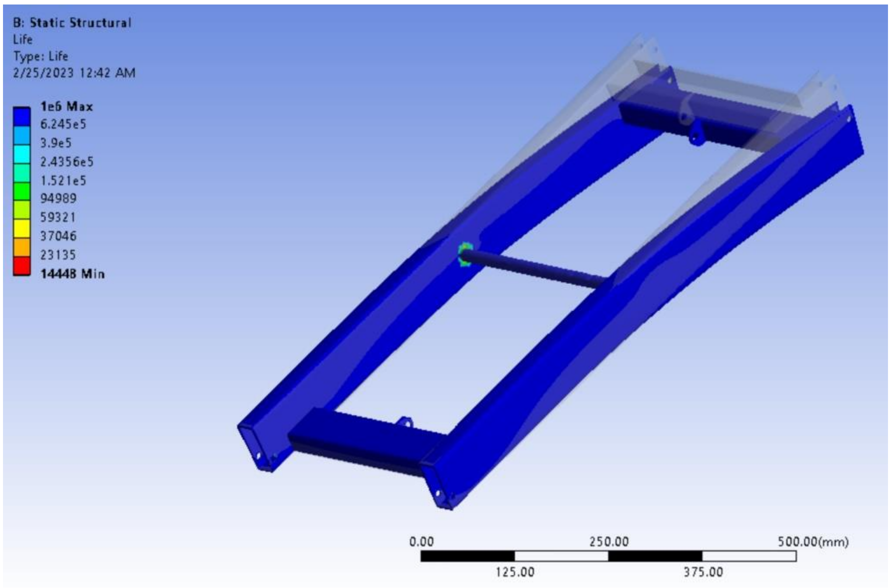



*Figure 54 . Life cycles of link.*

### 4.3.3 Simulation Result of Upper Plate

In this section the critical points of upper plate stress, strain, deformation, and life cycles were obtained as shown in the Table 9 and Figures 31-34.

Table 9. Final Result of Upper Plate

| Property | Minimum | Maximum | Average |
|---|---|---|---|
| Stress [MPa] | $3.3734 \times 10 - 2$ | 115.7 | 8.0471 |
| Strain [mm/mm] | $2.9468 \times 10^{-7}$ | $6.8942 \times 10^{-4}$ | $5.0576 \times 10^{-5}$ |
| Deformation [mm] | 0 | 0.5091 | $6.693 \times 10^{-2}$ |
| Life [cycles] | $1.8953 \times 10^{5}$ | $1.0 \times 10^{6}$ | $9.9998 \times 10^{5}$ |

The table shows the results of a study on the mechanical behavior of an upper plate. The properties that were measured include stress, strain, deformation, and life.

Stress is the force per unit area that is applied to the upper plate. The minimum stress that the upper plate can withstand is $3.3734 \times 10^{-2}$ MPa, while the maximum stress that it can withstand is 115.7 MPa. The average stress that the upper plate can withstand is 8.0471 MPa.

Strain is the measure of deformation or elongation experienced by the upper plate relative to its original length. The minimum strain that the upper plate can experience is $2.9468 \times 10^{-7}$

mm/mm, while the maximum strain that it can experience is $6.8942 \times 10^{-4}$ mm/mm. The average strain that the upper plate can experience is $5.0576 \times 10^{-5}$ mm/mm.

Deformation is the total deformation or displacement experienced by the upper plate. The minimum deformation that the upper plate can experience is 0 mm, while the maximum deformation that it can experience is 0.5091 mm. The average deformation that the upper plate can experience is $6.693 \times 10^{-2}$ mm.

Life is the number of cycles or repetitions of loading that the upper plate can withstand before failure. The minimum life that the upper plate can withstand is $1.8953 \times 10^{5}$ cycles, while the maximum life that it can withstand is $1.0 \times 10^{6}$ cycles. The average life that the upper plate can withstand is $9.9998 \times 10^{5}$ cycles.

The values in the table show that the upper plate is able to withstand a significant amount of stress, strain, and deformation before it fails. The upper plate also has a long life under loading conditions. These results suggest that the upper plate is a suitable material for a variety of applications.

Here are some additional details about the properties of the upper plate:

The stress that the upper plate can withstand is determined by its material properties, such as its strength and toughness.

The strain that the upper plate can experience is determined by its ductility.

The deformation that the upper plate can experience is determined by its stiffness.

The life of the upper plate is determined by its fatigue strength.

The mechanical behavior of the upper plate is important for its design and application. The values in the table can be used to design the upper plate to withstand the expected loads and stresses. The values can also be used to predict the lifespan of the upper plate under different loading conditions.

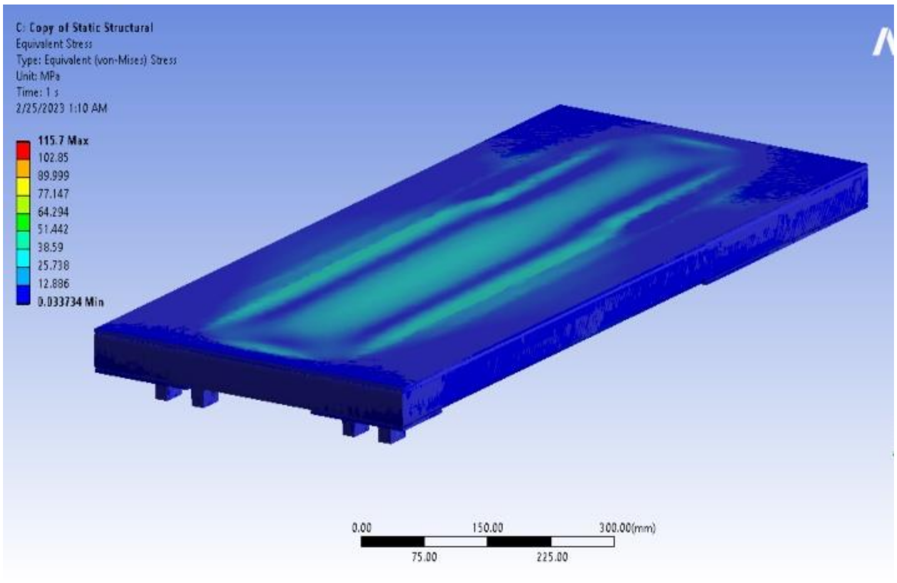



*Figure 55. Equivalent Stress of Upper Plate.*

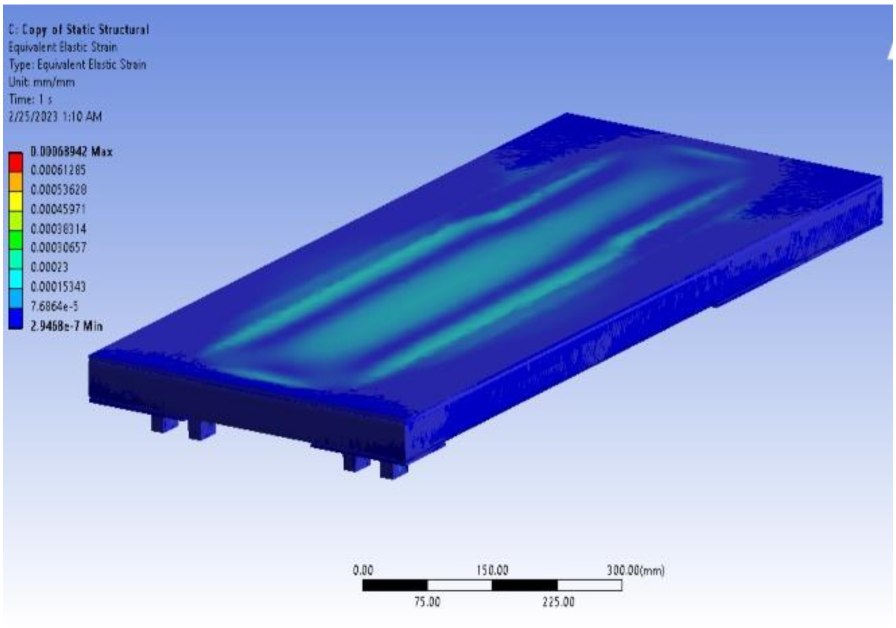



*Figure 56. Equivalent Elastic Strain of Upper Plate.*

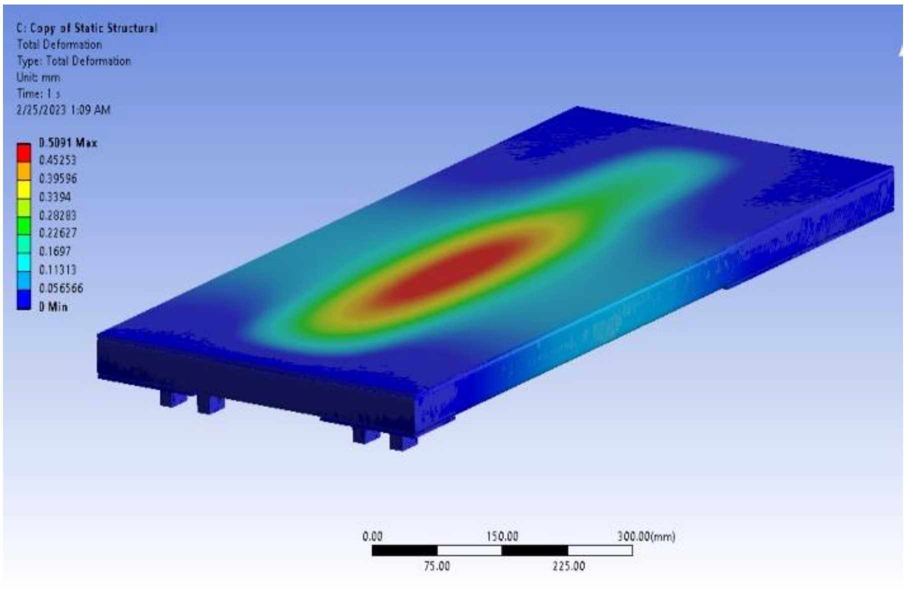



*Figure 57. Total Deformation of Upper Plate.*

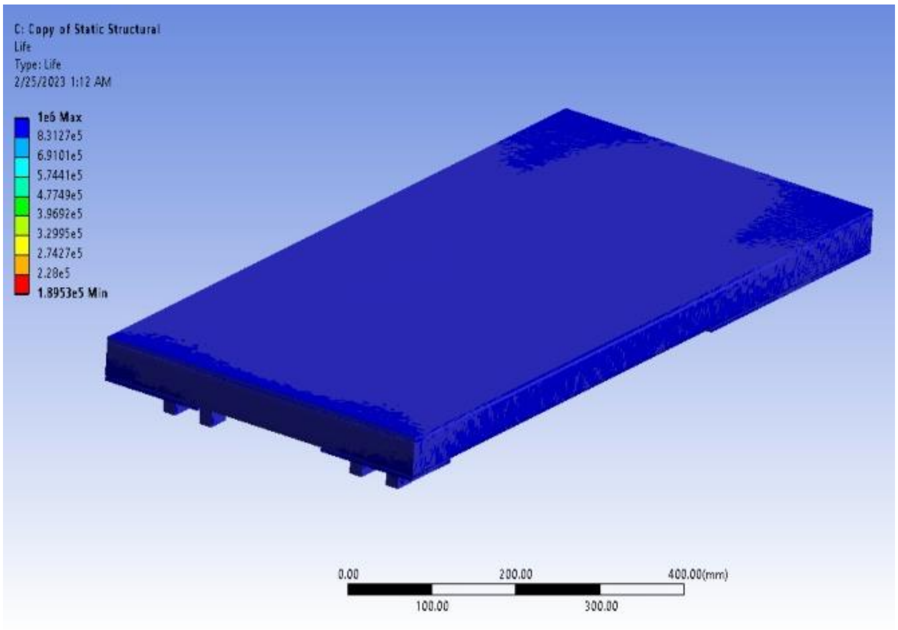



*Figure 58. Life cycles of Upper Plate.*

## 4.4 Manufacturing Procedure

The car lifting mechanism consists of seven major components, each with its own unique manufacturing process. The top plate is made by cutting and shaping a metal sheet to match the vehicle's body dimensions and then attaching it to the body using bolts or welding. The middle link is a massless connector made by selecting a metal rod and cutting and shaping it to specifications. The bottom plate is designed to lower to the ground when the mechanism is activated and is made by cutting and shaping a metal sheet to match the middle link's dimensions, followed by drilling holes for attaching the lead screw unit. The mechanism also includes two split links made by selecting suitable metal rods, cutting them to appropriate dimensions, and shaping them to fit design specifications. The hydraulic units are purchased from shops, selected for suitability, and mounted onto the upper and lower plates using bolts.

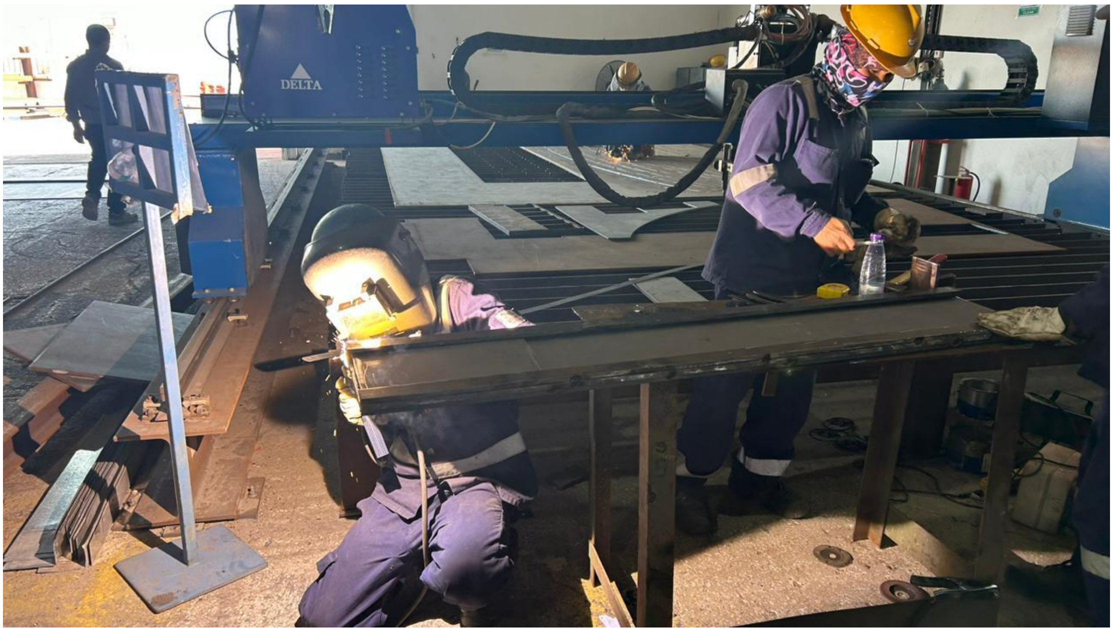



*Figure 59. Image 1 of the real mechanism during manufacturing*

Once all components are manufactured, they are assembled by connecting the top plate to the vehicle's body, followed by connecting the middle link and split links to the upper and lower plates, attaching the lead screw units, and testing the system for proper functionality, load capacity, and durability.

The manufacturing process requires precision and careful attention to detail to ensure that the components fit together correctly, and that the mechanism operates safely and smoothly.

*Figure 60. Image 2 of the real mechanism during manufacturing*

*Figure 61. Image 3 of the real mechanism during manufacturing*

## 4.5 Manufacturing Costs

To construct our project, which is the Design and Simulation of a Multipurpose Built-In Car Lifting Mechanism, we need to acquire two lead power screw systems, metal sheets for the upper and lower plates, and metal beams for the two links.

To bring this lifting mechanism to life, we require a set of source materials and components. The table presented below showcases the specific materials and their respective estimated prices and quantities ordered. By computing the anticipated cost for each item, we can arrive at the total expense of the project.

Table 10. Manufacturing Costs

| Type | Price SAR (include Vat) | Quantity | Subtotal SAR |
|---|---|---|---|
| Power screw lift | 349.5 | 2 | 699 |
| cast iron 10 mm plat | 500 | 2 | 1000 |
| cast iron 10 mm beam | 479.25 | 4 | 1917 |
| Screw | 150 | 1 | 150 |
| Wielding | 1000 | - | 1000 |
| Blots | 7 | 12 | 84 |
| Laber | 2000 | - | 2000 |
| **Total amount to be paid in SAR (including VAT)** | | | 6850 |

# 5 SYSTEM TESTING AND EVALUATION

## 5.1 Description of Final System Design

First of all, all components for the designed system, which are five parts, which are the lower plate, the two arms of the scissor lift, the upper plate and the movable plate will be discussed in the following points. The whole mechanism was decided to be designed with alloy steel as manufacturing material, but we chose the material to be carbon steel because it is less in price and more affordable in local market. The lower plate was manufactured to be holding the scissor lift by simple plate that is connected with scissor by welding and that plate is secured to the plate using nuts. The upper side of the scissor lift is welded using a plate that is also welded to the upper plate of the mechanism. The upper plate is manufactured to have some source of path for the movement of the moving plate. Finally, the whole mechanism is welded to the car chassis so the car van move after the motor starts to rotate. The whole mechanism will be operated using a power lead screw system. The lead screw was selected and not designed because of the time period. At the end, we tried our best to get a manufactured system similar to the designed system that we have in here, the manufactured system needs to be adjusted and finalized. The evaluation of the final system was incredibly as expected to raise the car and move it after all and that justifies that the designed system is doing its purpose eventually.

## 5.2 System Testing Procedure

After manufacturing the system, we ran some tests to make sure that the mechanism is sufficiently working. We first concentrated a force on the top plate and then we started the motor to rotate the lead screw to make the upward and downward movement. Then, when the scissor lift reached its maximum ability, we started the other motor that moved the car forward and backward. The mechanism was able to elevate the weight that was on top of it and then move it to the preferred direction we decide. The test was run on the workshop that we manufactured the mechanism in and the results were more than sufficient.

## 5.3 Discussion

In the provided project the target is to design a system that eventually lift a car with a weight of 3000 kg maximum. So based on the best result that we can get form our calculations the design should be considered. With the designed mechanism, a hydraulic system will operate the whole mechanism and it will be connected to the car battery to get the needed power. Whenever the mechanism should be used, this hydraulic jack, regarding they mechanism that was manufactured, will operate the mechanism to move for the preferred situation depending on what chassis should be considered and after placing the system the hydraulic will give an efficient amount of force to lift up the car and move it backward and forwards.

## 5.4 System Limitation Based on Design and Fabrication

The mechanism's actual model may not function exactly as intended, and data may not match will theoretical calculations. The followings are some constraints brought on by budget and design.

- The availability of materials.
- The budget of the project limits if there will be manufacturing.
- The mechanism can operate on heavy cars only.
- Legislation

Legal actions should be taken before building the mechanism beneath the car, federal and local levels must be in consideration.

## 5.5 Conclusion

Based on the results of the kinematic and the stress analysis, the proposed mechanism is practically applicable and could be easily and cost-wise manufactured and installed on certain categories of safari cars. The proposed mechanism is a multi-purpose mechanism that could be safely and easily used in emergencies and maintenance situations. with the selection of better material, a smaller models could be manufactured for a smaller car for the same purposes without any changes in the car chassis and affecting the structural integrity of the car.

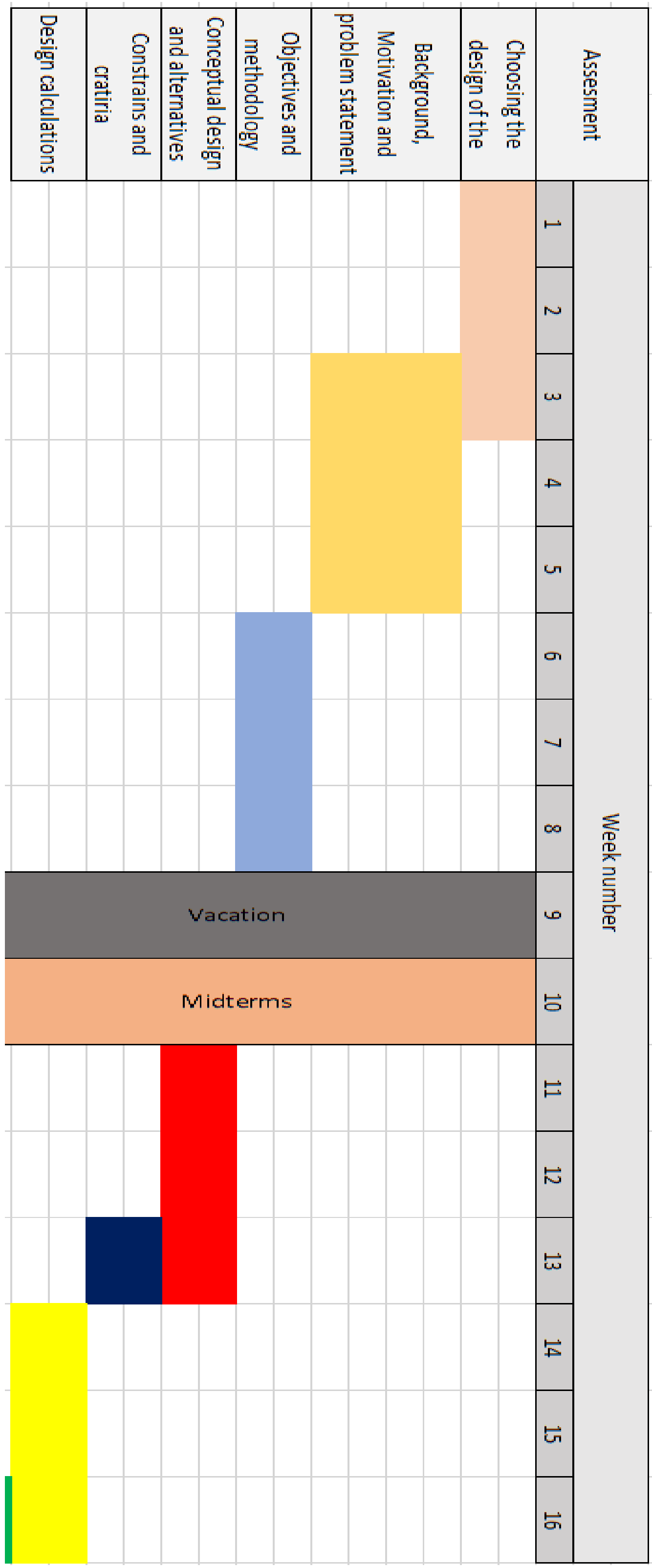



*Figure 62. Gantt chart for this semester*

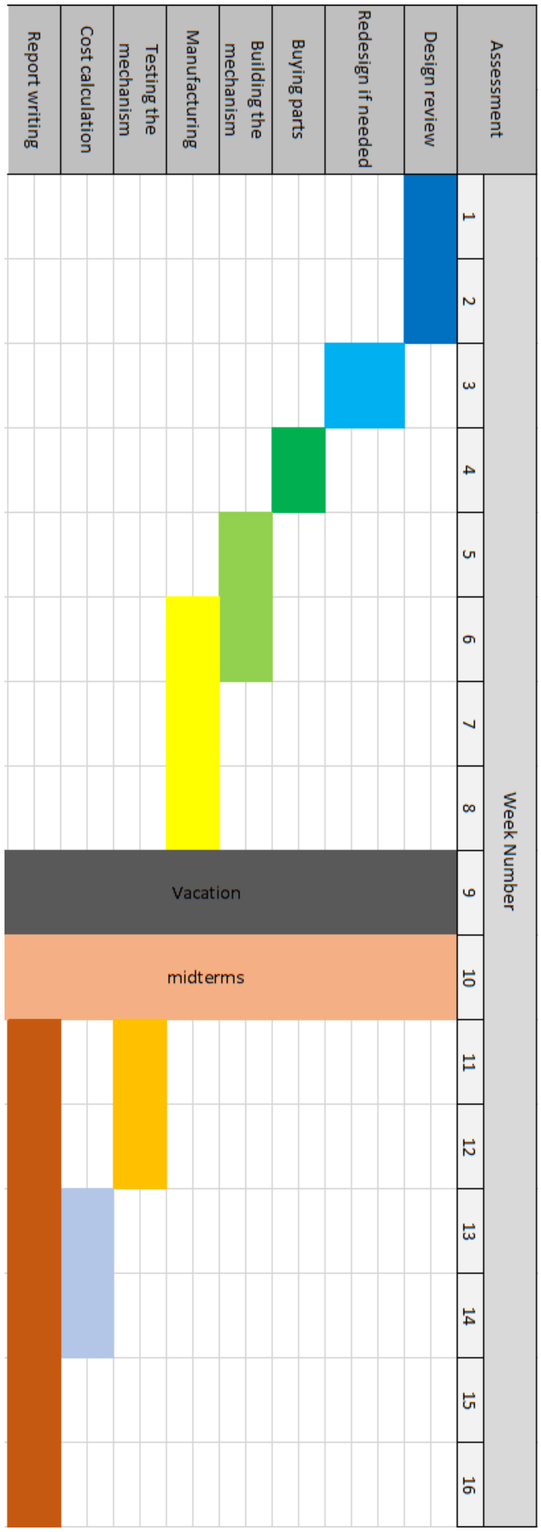



*Figure 63. Gantt chart for previous semester*

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