

# **Design and Simulation of Mechanical Scissor Screw Type Bus propeller shaft Lifter Machine**



Ambachew Asaye Reta

A thesis Submitted to the Department of Mechanical Engineering  
School of Mechanical, Chemical, and Materials Engineering  
Presented in Partial Fulfillment of the Requirement for the Degree of  
Master's in Automotive Engineering

Office of Graduate Studies  
Adama Science and Technology University

Dec. 2023  
Adama, Ethiopia

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**Advisor:** Siraj Kedir Busse (Ph.D.)

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## DECLARATION

I declare that this thesis entitled “Design and Simulation of a Mechanical Scissor Screw Type Bus propeller shaft Lifter Machine” is my work and has not been submitted to any university for a similar purpose. The references used in this thesis are duly recognized by proper citations.

Ambachew Asaye Reta

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I, the major advisor of this research thesis, hereby certify that I have closely advised/supervised the student while developing this thesis and read the draft thesis entitled “Design and Simulation of a Mechanical Scissor Screw Type Bus propeller shaft Lifter Machine” prepared under my guidance by Ambachew Asaye. Therefore, I recommend the submission of the thesis to the department for further review and evaluation.

Siraj Kediri Buse (Ph.D.)

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I/we hereby certify that the recommendation and suggestion given by the thesis review committee are appropriately incorporated into the final thesis entitled “Design and Simulation of a Mechanical Scissor Screw Type Bus propeller shaft Lifter Machine” by Ambachew Asaye.

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## **APPROVAL OF BOARD OF REVIEWERS**

We, the undersigned, members of the Board of Reviewers of the thesis open defense by Ambachew Asaye have read and evaluated the thesis entitled “Design and Simulation of a Mechanical Scissor Screw Type Bus propeller shaft Lifter Machine” and assessed the understanding of the candidate about the proposed research. This is, therefore, to certify that the thesis is accepted and we recommend the implementation of the thesis.

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Finally, approval and acceptance of the thesis are contingent upon submission of its final copy to the Office of Postgraduate Studies (OPGS) through the Department Graduate Council (DGC) and

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## **ACKNOWLEDGEMENT**

First of all, all praises, magnificence, and adoration are due to the almighty God alone who guides, directs, and sustain us throughout my study. Next, my heartfelt thanks Mechanical Engineering Department, and go to my adviser Siraj Kedir Busse (Ph.D.), for his tolerance, hospitality, and exceptional counsel during the paperwork, as well as his aid and time, which he offered more gracefully and with more goodwill than I imagined. He was generous with his ideas and energy, and I'm grateful to him. My heartfelt gratitude goes to my friends Kebebe Urgie and Aleyu Melese for their amazing and invaluable assistance. This paper would not have been possible without their assistance. I am extremely convenient as a result of them, and my sense of belonging is intact.

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## **ACRONYMS**

AI	Automotive Industry
BAI	Bishoftu Automotive Industry
CS	Compressive Strength
CFD	Computational Fluid Dynamics
CAD	Computer-Aided Design
CATIA	Computer-Aided Three-Dimensional Interactive Application

## ***ABSTRACT***

*Scissors are lifting heavy objects to reduce the risk of injuries in around the world. In the Bishoftu Automotive Industry, assembling the drive shaft on the engine of the bus can be done by one person, but in, the production of bus drive shaft is assembled by three technicians (i.e., one person for controlling the overhead crane, a second person for holding the drive shaft and the third person for connecting the drive shaft with the engine and bearings). The aim of this research is to design the scissor-type bus drive shaft lifter for bus production factories when the drive shaft is assembled and disassembled. The design of the scissor was concentrated on finding the optimal lifting height based on the weight of the bus drive shaft and minimizing labor costs. Focusing on the deflection in accordance with the ANSI standard, the calculation regarding the allowable maximum deflection must not be exceeded. To analyse the data, manual calculations, solid modelling, and computer simulations were involved using state-of-the-art CATIA V5 software. The results show that the deflection is still in the acceptable range. However, some manufacturers find that the allowable maximum deflection is too excessive. Based on the initial input data, the remaining dimension of the scissor lift is calculated, and it can lift a maximum height of 750 mm. According to the results observed from the ANSYS simulation, the maximum stress distribution over the body of the scissor lifter is 22.88 MPa, and the maximum total deflection is recorded at 3.24 \* mm. Therefore, the scissor lift can lift the weight of the drive shaft to carry and move horizontally by using four tires assembled on it and lifting the 80kg drive Shaft to the desired height. This screw-type scissor lift serves to give a mechanical advantage by changing rotational force to vertical force.*

***Keywords:*** *Aerial Scissor Lift, Lead Screw, mechanical drive*

# CHAPTER ONE

## 1. INTRODUCTION

### 1.1. Background and Justifications

An automotive jack is a device used to raise all or part of a vehicle into the air in order to facilitate repairs (Budynas, R. G., Nisbett, J. K., 2018). Most people are familiar with the basic car jack (manually operated) that is still included as standard equipment with most new cars. These days, a car jack is an important tool to have in our vehicle due to unknown upcoming event such as flat tire in our journey. Even so, people who like to rotate their tires themselves or who may install snow tires before the winter and remove them in the spring need to use a jack to perform the job. Furthermore, available jacks are typically large, heavy and also difficult to store, transport, carry or move into the proper position under an automobile. In addition, to the difficulties in assembling and setting up jacks, such jacks are generally not adapted to be readily disassembled and stored after automobile repairs have been completed (Tuffaha et al., 2019). Car jacks must be easy to use even for women or whoever had problem with the tire in the middle of nowhere. In light of such inherent disadvantages, commercial automobile repair and service stations are commonly equipped with large and hi-tech car lift, wherein such lifts are raised and lowered via electrically or hydraulically powered systems. However, due to their sheer size and high costs of purchasing and maintaining electrically-powered car lifts, such lifts are not available to the average car owner. Engineering is about making things simpler and effective (Budynas, R. G., Nisbett, J. K., 2018). Such electrical-powered portable jacks not only remove the arduous task of lifting an automobile via manually-operated jacks, but further decrease the time needed to repair the automobile. Such a feature can be especially advantageous when it is necessary to repair an automobile on the side of a roadway or under other hazardous conditions.

Available jacks present difficulties for the elderly, women and are especially disadvantageous under adverse weather conditions. They further require the operator to remain in prolonged bent or squatting position to operate the jack (Bazoune, 2017). Hydraulic systems that do not have the necessary hydraulic fluids will not function, which becomes a problem when a leak occurs. You must repair the leak so the hydraulic fluids can continue to produce flow; otherwise, the hydraulic system will begin to slow down. Areas that have leakage will also have hotter internal temperatures. This phenomenon can prove beneficial; since these temperatures can help the

operators of the hydraulic systems locate the leak, prevent leaks by using proper procedures and the correct materials, and by performing regular preventative maintenance. You must filter oils in hydraulic systems on a regular basis to ensure that the hydraulic fluid contains no broken particles, as well as to eliminate harmful damaging air pockets.

A mechanical jack is a device that lifts weights or heavy equipment and vehicles so that maintenance can be carried out underneath at the workplace or manufacturing setting. A jack screw is a type of jack that functions by turning a lead screw. It is commonly used to lift heavy loads to a height (Ezurike & Okwu, 2018).

A mechanical device that extends or places a platform is a screw-type scissor lift or mechanism, and also a mechanic with crisscross X-shaped folding supports was given the name scissor (Vaibhavbankar, 2017).

Applying force to one or more supports causes the cross pattern to elongate, resulting in extension or displacement motion. The drive shaft is loaded, moved, and attached at one of the bus factory's stations, as the name implies the scissors operate machinery by receiving a torque applied manually by the handle, which moves the frame that supports the driving shaft upward and downward (Zhang et al., 2015). Although this system is not new, it has been fine-tuned and updated to be capable of lifting loads. When the handle is spun, a square thread screw slides the pin connected to the nut horizontally and pushes the scissor link.

A mechanical lift is a scissor lift that operates on the screw jack concept. Mechanical scissor lifts have a wider range of applications than other types of lift systems due to their straightforward design and construction. Screw jacks are made of lead screws with square threads, and in today's cost-cutting economy, to must choose the most cost-effective industrial alternative with long-term benefits. Mechanical scissor lift tables provide ergonomic and productivity benefits across a wide range of industries, and the market has responded with multiple designs operated by varying technologies. The mechanical scissor lifter offers the following benefits over hydraulic, pneumatic, and electrical mechanisms: simple construction, portability over other mechanisms, low manufacturing cost, lifting big weights, and quick defect detection. no need for electricity (manual operation), and no external storage necessary (hydraulic and pneumatic requires a tank for fluid and air)(Kamble et al., 2018).

A scissor lift is the most cost-effective, dependable, and adaptable means of lifting weights it has minimal moving components that may simply require oil (Thorat et al., 2017). As a result, this lift

table was built to elevate the weight smoothly to any desired height, and a scissors mechanism is presented to address the shortcomings of the devices listed above. This mechanism has a power screw, and the scissors' top is coupled to a stable platform. This gadget will use the power supplied by a power screw to manually raise and lower a platform.

This thesis is aiming to design a scissor-type bus drive shaft lifter that uses in a bus maintenance garage when the drive shaft is assembled and disassembled. The design of the mechanical lift is a scissor lift type which works on the principle of screw jack. Because of simple design aspects and simplicity in construction, mechanical lift has a wider range of applications than other types of lift mechanisms. The screw jack is made by using a lead screw of square thread, as per today's condition of cost reduction.

## **1.2. Statement of the Problem**

The mechanical scissor lifter offers the following benefits over hydraulic, pneumatic, and electrical mechanisms: simple construction, portability over other mechanisms, low manufacturing cost, lifting big weights, and quick defect detection. No need for electricity, no external storage necessary (hydraulic and pneumatic requires tank for fluid and air).

The object of mechanical lifting scissor has been developed to cater to the needs of small and medium automobile garages, which are normally man powered with minimum skilled labour. In Most of garages the vehicle propeller shaft is lifted by using human power. This needs high man Power and skilled labour. In order to avoid all such disadvantages, the mechanical lifting jack has been designed in such a way that it can be used to lift the vehicle propeller shaft very smoothly. The operation is made simple so that even unskilled labour can use it with ease.

Assembling and dis assembling the drive shaft on the engine of the bus can be done by one person by using mechanical lifting but in the production factory, three workers are required (i.e., one person for controlling the overhead Crain, a second person for holding the drive shaft so that it does not fall, and the third person for connecting the drive shaft with the engine and bearings) involved to do their job.

## **1.3. Objectives**

### **1.3.1. General objective**

The general objective of this thesis is to design and simulate a mechanical scissor screw type Bus propeller shaft lifter machine

### **1.3.2. Specific objectives**

- To design a mechanical scissor screw type drive shaft lifter machine that can lift heavy loads.
- To Build a model of the new mechanical scissor & perform stress analysis to assess the machine's structural integrity
- To simulate the lifting machine using CATIAV5 software

### **1.4. Scope of the study**

The scope of the study consists of the design and modelling of individual components of the propeller shaft lifter machine, performing the stress, deflection and load carrying capacity of the machine that will be used for lifting the bus drive shaft during assembly. For analysis purpose the software packages such as; CATIA V5 and ANSYS, were used. From the analysis result, it can be concluded that the design within the safe range.

### **1.5. Significance of the study**

This design is a modern mechanism for lifting heavy loads. It has some advantages, such as:

- It also has a big economical advantage,
- Improve the time of manufacturing and commercialized.
- For the manufacturer, it is a new business venture since it is a new idea, and for the customer, it has a low initial cost compared to its benefits.
- To support the workers in the garage and bus assembly company, it needs to have a mini lifting mechanism.

### **1.6. Organization of the study**

The research is organized systematically, beginning with Chapter 1, an introduction that offers background information and the issue statement. The literature study portion in Chapter 2 addresses the pertinent concepts and principles linked to the design of the lifter machine. The methods section covers the procedures followed in Chapter 3 to design, simulate, and assess the machine. Chapter 4 presents the design and development of the mechanical scissor screw lifter machine. Chapter 5 gives the findings of the stress and deflection study. Finally, in Chapter 6, the conclusion section reviews the study's key themes and makes recommendations for further research.

## CHAPTER TWO

### 2. LITERATURE REVIEW

Lifting and displacement mechanisms of the drive shafts are difficult because of the weight of the drive shafts and the place where it assembled and disassembled on the bus. To support the workers in the bus assembly company, it needs to have a mini lifting mechanism which is called a mechanical scissor lifter machine. The concept generation process results in a set of product concepts from which make the final specification and design.

#### 2.1. Related study

Dhamak et al., 2015 describe that, scissor jack is a device which is used to lift heavy vehicles, partially or fully in the air for breakdown and maintenance. Scissor jack includes one power screw which is rotating, two fixed nuts, four links which is connected to nuts, eight pins used to fix that four links, two rings provided at screw end and load platform supported by upper two links. Since traditional jack that available in market involve plenty of variety like screw jack. (Patil & Kachave, 2015). No need for electricity, and no external storage is necessary (hydraulic and pneumatic requires tank for fluid and air)(Kamble et al., 2018). The concept of a motorized scissor lift that will get over the restrictions of needing to do things under the car at the top of the scissors is coupled to a table platform, and this mechanism includes a power screw. To manually raise or lower a platform, this gadget will utilize the power supplied by a power screw.

Mahmoud et al., 2017 scissor lift elevator is a vertical transportation cab which is raised and lowered from underneath, somewhat like a traditional hydraulic elevator, except that instead of a hydraulic cylinder the extendable mechanism is a folding lattice of crisscrossed beams similar to a pantograph. The entire mechanism extends upward when pressure is applied to the lowest members.

Michael-Adel, 2008 studied base identify a mechanical lift is a scissor lift that operates on the screw jack concept. Mechanical scissor lifts have a wider range of applications than other types of lift systems due to their straightforward design and construction. Mechanical lifts are screw jacks. Screw jacks are made of lead screws with square threads, and in today's cost-cutting economy, to must choose the most cost-effective industrial alternative with long-term benefits.

A lifting device is a system that allows a small force (effort) to overcome a large force or load (Chaturvedi, 2017a). There are practically hundreds of uses for lift tables in manufacturing,

warehousing, and distribution facilities. The addition of this device (lift table) makes the job faster, safer, and easier. Some typical applications include; machine feeding and off-loading, product assembly, inspection quality control repair, and feeding, and offloading conveyor levels. The commonest method for operating a scissors lift is the use of a power screw. According to (Ezurike & Okwu, 2018), power screws are devices that provide means for obtaining large mechanical advantage. Other peppers (Engineering et al., 2012) define the power screw as a device used in converting rotary motion into uniform longitudinal motion. The manually operated scissor lift is a device that makes use of a horizontally placed power screw to overcome a large load through less effort applied on the power screw, by turning the power screw with the aid of a ratchet handle on one side of the device.

Another constraint to be considered is the deflection of the unit. Deflection in a scissor lift can be defined as the resulting change in elevation of all parts of a scissor lift assembly, typically measured from the floor to the top of the platform deck, whenever a load is applied to or removed from the lift (Michael-Adel, 2008). The ANSI MH29.1 safety requirement for industrial scissor lifts states that all industrial scissor lifts will deflect under load. The industry standard goes on to outline the maximum allowable deflection base on platform size and the number of scissor mechanisms within the lift design (Engineering et al., 2012)

Bhalake et al., 2020 investigate scissor lift deflection in material handling applications where the lift must contact neighboring, fixed heights, particularly while transferring rolling weight. Scissor lift deflection becomes more critical in material handling applications where the lift must interface with adjoining, fixed elevations, especially when transferring the rolling load. In these cases, any difference in elevation between adjoining surfaces during material transfer must be minimized or if not eliminated. Before attempting to discuss how to limit scissor lift deflection, it is important to understand the contributing factors to a lift's total deflection. An open or raised scissor acts very much like a spring would apply a load and it compresses, remove a load, and expands. A component within the scissor lift has the potential to store or release energy when loaded and unloaded (and therefore deflects).

There are application-specific characteristics that may promote deflection, understanding these root causes helps to pin-point and apply effective measures to limit them where Leg deflection due to bending is a result of stress which is driven by the total weight supported by the legs, scissor leg length, and available leg cross section(Journal et al., 2019). Platform bending will increase as the

load center of gravity moves away from the center (unevenly distributed load) or at an edge (eccentric loading) of the platform, also as the scissor opens during the rising of the lift, the rollers could roll back towards the platform hinges and create an increasingly unsupported overhung portion of the platform assembly (Görkem Dengiz et al., 2018). Increasing platform strength via increased support structure material height does improve resistance to deflection but also contributes to an increased collapsed height of the lift.

For cases where the scissor lift is mounted to an elevated or portable frame, the potential for deflection increases. (Diriba Ejersa et al, 2022) studied to effectively resist deflection, the base frame must be rigidly supported from beneath to support the point loading created by the two scissor leg hinges. Scissor lifts are assembled with pins at all the hinge points and each pin has a running clearance of the outer diameter of the pin and the inner diameter of its clearance hole or bushing. More scissor pairs or pantographs, that are stacked on top of each other, the more pinned connections there are to accommodate movement, or deflection when compressing these running clearances under load.

Load placement also plays a large role in scissor lift deflection. Off-center load causes the scissor lift to deflect differently than with center or evenly distributed loads. (Engineering et al. 2012) studied the end load (in line with the scissor) is usually shared well between the two scissor leg pairs. Side load (perpendicular to the scissor) however is not shared as well between the scissor legs pairs and must be kept within acceptable design limits to prevent leg twist (unequal scissor leg deflection) which in addition to platform movement due to deflection, often results in poor roller tracking, unequal axle pin wear, and misalignment of cylinder mounts. As mentioned above, the degree of deflection is directly related to the change in system pressure and the change in component stress as a result of loading and unloading. Scissor lift typically experiences their highest system pressure and highest stress (and therefore the highest potential for deflection) with the first 20% of the total available vertical travel from the fully lowered position (Engineering et al. 2012)

Mechanical scissors lifts are used to raise objects, notably on construction sites. This is the most current advancement in scissors lift technology. The lift features a load screw that contracts when it is loose and expands when it is tight. A belt drive system is attached to the load screw. Since it transforms the input force from the wheel into the linear motion of the lift, the lead screw effectively completes the operation. This may assist to enhance the environment and tools used at

work. Numerous businesses all around the world have been utilizing various scissor lifts. Additionally, various mechanical options can increase portability and simplicity, and by offering the right gearing, input energy is decreased along with the cost. Another form of scissor lift that operates on the screw jack concept is the mechanical lift. Due to their easy design and construction, mechanical lifts are more versatile than other lift systems in terms of applications. A screw jack is an example of a mechanical lift. To create a screw jack, which is made of a lead screw with a square thread, it must find an industrial use that is both economical and beneficial in the long run. Therefore, it is necessary to rethink many items in the production system to reduce the cost of the product relative to the identical product. Hoist lifting, platform rising, and wheel assembly are among the industrial uses for a mechanical scissor lift.

The advantages of mechanical systems over other mechanisms such as hydraulic and pneumatic are their ease of design, portability, low manufacturing costs, capacity to lift large weights, simplicity of issue diagnostics, absence of a requirement for external storage, and manual operation. However, a general understanding of screws will show the loss caused by friction in the screw threads. As a result of friction losses, this device's efficiency is poor. Additionally, the machine requires manual power to operate, and a lot of energy is expended to get the desired outcome. Although it may be utilized practically everywhere in the nation, whether there is electricity available or not, its usefulness cannot be overstated. (Srinivasulu and Sainath 2019).



Figure: 2.1 Mechanical Scissors Lift

According to Thorat, et al., 2017, a scissor lift is a tool used to move large objects. One of its goals is to raise tables to a comfortable height. A mechanical scissor lift has fewer moving components and may merely need lubrication, making it one of the most affordable, reliable, and adaptable ways to lift goods. The weight is raised smoothly to any desired height by this lift table. A scissors mechanism is suggested as a way to lessen the shortcomings of the devices discussed above.

As stated in the work of (Wininger, Fred A., Zeng R., Johnson, G.S., Katz, M.L., Johson G.C., Bush, W.W., Jarboe, J.M., Coates, 2020) a scissor lift is a basic mechanical device used to raise an element or object from ground level to a set height to fulfill a certain activity with maximum load and lowest effort. This research describes the design and investigation of a mechanical scissor lift based on the screw concept. The design was built with the intention of the lift being mechanically operated, minimizing the total cost of the scissor lift. The mechanical lifts are extended through a rack and pinion system or power screw, both of which can convert rotational motion.

Arunkumar et al., 2021 in this scissor lift there is a table that is used to the mound, hoist, pass on and additionally move material in the middle at least two heights. The fundamental goal of the gadgets utilized for hoisting settle is to make the table movable to an ideal stature. In this venture, we advanced the heaviness of the base and top of the scissor lift by utilizing the I-outline structure and utilized regenerative technique to lessen weight in the pole. Scissor lift gives the most financially trustworthy and adaptable techniques for lifting loads. The edge is durable and sufficient with expansion in underlying respectability. A different-stature scissor lift is comprised of at least two leg sets. It is dissected and advanced utilizing Autodesk Fusion 360°.

Srinivasulu & Sainath, 2019 in the current project, a customized scissor lifting mechanism is being designed for lifting the three-tone load. The entire mechanism is designed using the CAD package. Simulations of the 2-DOF planar translational mechanisms and the docking procedures were carried out. Finally, the structural analysis is carried out for the structural strength of the links. The result observed from the simulation was 140.67Mpa of maximum equivalent stress at the links and it was suitable for the desired design.

Dengiz et al., 2018 the studies about the design and analysis of a scissor lift system were performed with a load-carrying capacity. The solid model and assembly of the system have been developed by the solid works program. The structural analysis of the system was also investigated with the help of the same program i.e., stress, deflection, and safety factor values of system elements were

obtained. As a result of the analyses, it was concluded that the system designed could safely be used during load/unload operations. the main aim of a scissor lift is to save the time required for building platforms or temporary lifts. Once the lift is constructed according to the requirement, much time is saved. The portability of the lift can enable the workers to use it in various working areas wherever needed. Considering the cost of the lift it is more suitable than making temporary platforms at construction sites with the help of bamboo sticks or other materials. The maintenance cost is almost negligible as only lubrication is required for only some components. The ultimate aim is to increase the production rate of the company through the efficient use of mechanical scissor lifts. has provided the design methodologies of the scissor lift. Various 3D software is used for modeling. The analysis is carried out using ANSYS software. A comparison of various materials and their results are given which leads to proper material selection to meet the system requirements. The analysis is carried out to meet the functional requirements of a particular system. The scissor lift can also be used to provide hydraulic drive to conventional mechanical fixtures for cylinder assembly in the industry. It is designed to solve a center matching problem caused by two conventional fixtures used for cylinder assembly. With the help of a scissor lift required raised height of the cylinder is achieved to complete the assembly process in the least time and effort by laborers is minimized. The selection of the materials is done according to their mechanical properties (Strength, stiffness, elasticity, hardness, and toughness) based on the particular working conditions.

The two main materials used for making good-quality lifters are Steel and Aluminum (Chaturvedi, 2017b). When selecting the material suitable for the construction of the Scissor lifter one has to consider the properties that will enable, it to function with no expected failure and at the same time the weight and ease of machining the product (Bhalake et al., 2020). Therefore, the main areas that can be classified in this case are the strength of the material, weight, ease, and cost of manufacturing. Aluminum's fatigue performance is half that of steel, which is an advantage steel has over aluminum in scissor-type drive shaft lifter life durability. Ramchandra D Patil, 2019 Depending on the characteristic states c-30 Steel is the most viable material selected for the manufacture of the scissor-type bus drive shaft lifter Components. For number 3, will use c -30 steel is chosen because of its relatively high strength and low cost.

## **2.2. Research Gap**

The exist papers are not designed for the vehicle to lift some parts of the vehicle to assemble and disassemble. As it is stated on the shortcomings of the existing scissor lift the currently available designed scissor lift are problems with weight, size height of lift, cost, installation construction to hold the propeller shaft and strength. Different researchers are tried to solve this problem by varying the thickness, size and changing the material. The Designs show promise at mitigating reduce worker fatigue and speed up work

As shown from the literature, a multi 'X' scissor lift was used to lift objects from the ground to some level of height using a mechanical scissor lifter. To lift much height by using a mechanical mechanism, much energy is expended to achieve the desired result, and not possible to perform a specific work with maximum load with minimum effort Automotive parts are mostly made of metal, which is a major reason for their large weight, and as such require devices of lifting and displacement of the same. In an automobile production and assembly facility, components have to be raised to certain heights which could be more convenient to the personnel working on it. When such a device is not available workers are often forced to bend from the waist to access the components which can lead to strains and major discomforts or even serious injuries that could affect productivity.

## CHAPTER THREE

### 3. MATERIAL AND METHODS

Bishoftu automotive manufacturing industry is located at Bishoftu city, around 47km far from Addis Ababa which is the capital city of Ethiopia. When came to Bishoftu city the company is get around 3km away from center of city or zuqala to the south.

#### 3.1. Material selection

The material selection for the proposed design typically considers Mechanical properties, affordable cost, and easily functional product to be designed. Here by considering the mechanical properties of the material with their high resistance to load, deformation, fracture, rust, and corrosion. c-30 steel material selected to use for designing all components of the proposed design. Depending on the characteristic states c-30 Steel is the most available material selected for the manufacture of the scissor-type bus drive shaft lifter components and this type of steel is chosen because of its relatively high strength and low cost. It has a tensile/compressive load capacity of  $500 N/mm^2$ , yield strength of  $300N/mm^2$ , tensile strength of  $139N/mm^2$ , Yield strength of  $310N/mm^2$ , Compressive strength  $604N/mm^2$ , Shear strength =  $192N/mm^2$ , and Allowable bearing pressure =  $17.5N/mm^2$ . The other components are manufactured from hot-rolled structural steel. It has a tensile strength of  $400N/mm^2$ , and yield strength of  $250N/mm^2$ . (Ramchandra D Patil, 2019).

##### 3.1.1. Load criteria and assumption

The load criteria and assumption depend on weight, length and width of the propeller shaft. Can include factors such as the maximum weight capacity, load distribution, operating conditions and safety factors. Assumptions, on the other hand, are the underlying beliefs or conditions that are taken for granted when designing or operating the lifter machine. This assumptions may include factors such as the material properties of the objects being lifted, the stability of the ground or support surface, the absence of external force or impacts, and the proper maintenance and operation of the machine.

The design of the scissors lifts is to lift to a height of 750mm and a carrying capacity of less than 90kg which is driven manually. The maximum extension, an “X” arrangement of the lift moves 0.75m, and the total number of tiers of scissors (combined) is 4, to lift a high of 0.7m. The base

platform dimensions are 700mm in length, 600mm in width, and 100mm in height from the ground.

Consider the above situation by making a scissor lift(Arunkumar et al., 2021).

- The total lift of the machine is 750mm
- The weight of the drive shaft is 80 kg
- The weight of the machine is 60kg
- Weight one side 80/4: 20kgs
- Factor of safety:3

### **3.1.2. Concept Generation**

- I. Design is collected from different sources (design book, related previous designs journals', design software and the internet)
- II. Calculating all distributed force, Model and analysis failures through factor safety
- III. Mathematical analysis of the lifting mechanism and full design of parts.

### **3.1.3. Research frameworks**

For this study, the first step is governing evidence and concepts about this thesis obtained from various academic sources such as design books, research journal articles related to this thesis objective, previous studies, a manual of the jack's related principle, and different types of websites. The second step of this study, experimental design based on the selected parameters that have been conducted and the process flow has indicated in Figure 3.1.

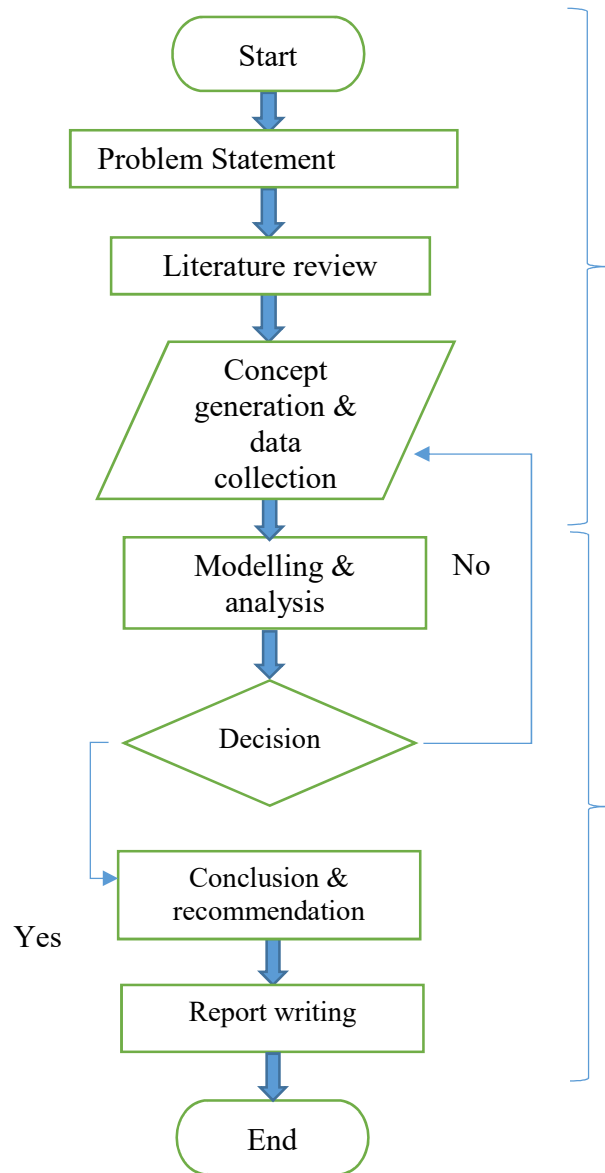


Figure 3.1. Research framework

## CHAPTER FOUR

### 4. DESIGN OF MECHANICAL SCISSOR LIFT

The mechanical scissor lifter offers the following benefits over hydraulic, pneumatic, and electrical mechanisms: simple construction, portability over other mechanisms, low manufacturing cost, lifting big weights, and quick defect detection. (Diagnosis), no need for electricity (manual operation), no external storage necessary (hydraulic and pneumatic requires tank for fluid and air) A scissor lift is the most cost-effective, dependable, and adaptable means of lifting weights since it has minimal moving components that may simply require oil. As a result, this lift table was built to elevate the weight smoothly to any desired height, and a scissors

Mechanical scissor lifts utilize a ball screw-driving system for making them exceptionally compact and easily able to adapt to any work environment. The manually operated scissor lift is a device that makes use of a horizontally placed power screw to overcome a large load through less effort applied on the power screw, by turning the power screw with the aid of a ratchet handle on one side of the device and capable of lifting an average load with little effort applied. One of the most important factors of a lift platform is stability. Another constraint to be considered is the deflection of the unit.

The major Advantages of Mechanical scissor lifts are:

- ❖ **Precision Positioning:** The ball-screw design makes it possible to perfectly position the lifts to your exact needs as well as to keep levels steady when loading or unloading the tables.
- ❖ **No Leak:** Using a dry cylinder, there's no oil leak and the work environment stays clean.
- ❖ **Lower Cost:** Costs less to purchase and maintain than hydraulic models.
- ❖ **Easy to operate:** It does not require well-trained personnel to operate it.

The scissor-type bus drive shaft lifter machine is used to move the drive shaft in the vertical direction to help the assembling process simple and easy. The drive shaft lifter is working mechanically and is operated manually. The force is applied to the screw and transferred to the structure link which is used to support the load and move the drive shaft up and down through the slot. To have a better scissor lifter it needs a proper design for most of the main components like a screw, link, pin, bar, nut, and load carry table.

### 4.1. The design procedure

It is a decision-making process (often iterative), in which the engineering sciences and mathematics is applied to convert resources optimally to meet a stated objective. Among the fundamental elements of the design process are the establishment of objectives and criteria, synthesis, analysis, construction, testing and evaluation.

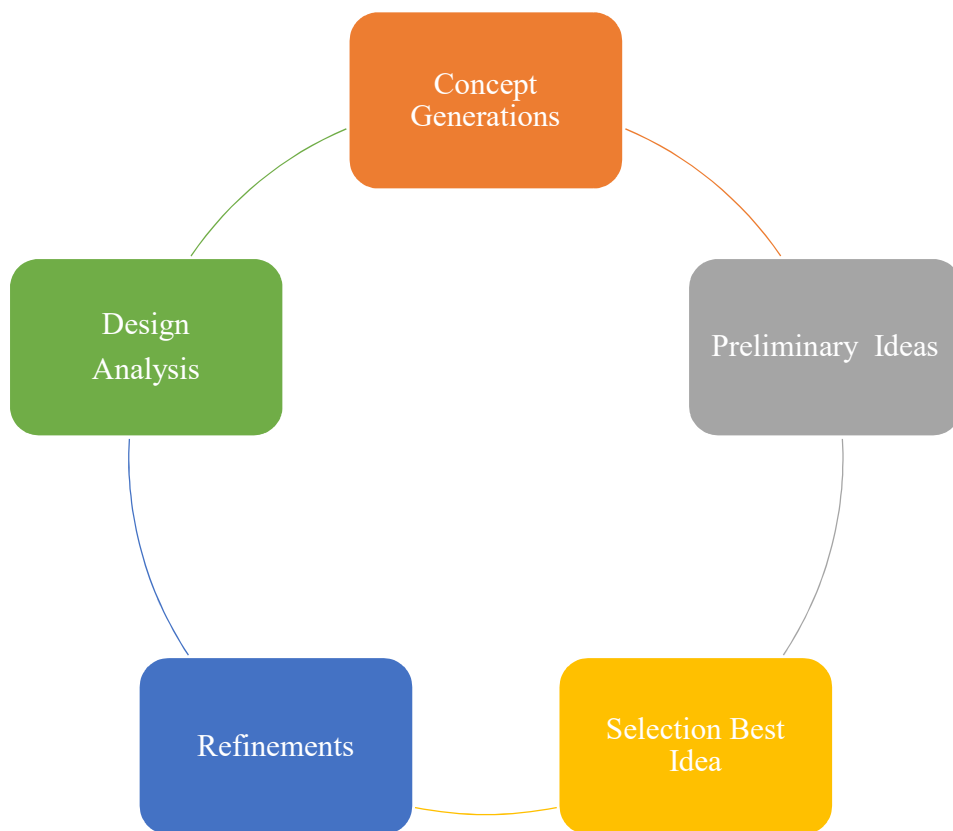


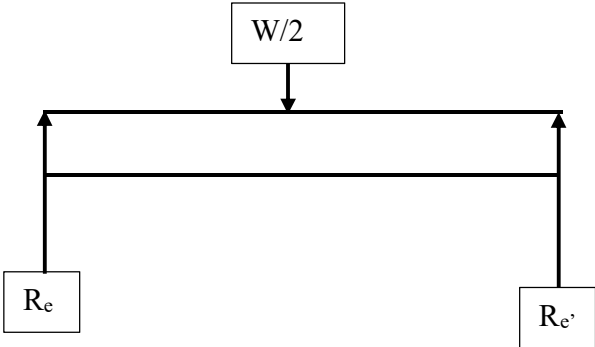
Figure 4:1 Design procedure

#### 4.1.1. Calculating all distributed load

To transform distributed loads into a single equivalent force and use composite parts to transform simple distributed loads into a single equivalent force. By calculate centers of mass and centroids using mathematical and mechanic’s concept. Mechanics is studied about ridge body and moving body because of the study improvement of uploading of vehicle for maintenance and assembly and other similar problems. By applied the concepts of particle and rigid body equilibrium to analyze structures including trusses, frames, and machines. Then friction, where the able to incorporate slipping/tipping, wedges, and belts.

Now will consider equivalent systems, distributed loads, centers of mass and centroids. A proper understanding of these concepts will make it possible for the designer is to apply their knowledge of statics and structures to complicated shapes that are subjected to complicated loadings.

**4.1.2. Force analysis in drive shaft lifter**



Reaction force during the free period

Reaction force during load acting

Figure 4.2. Side view of the shaft lifter

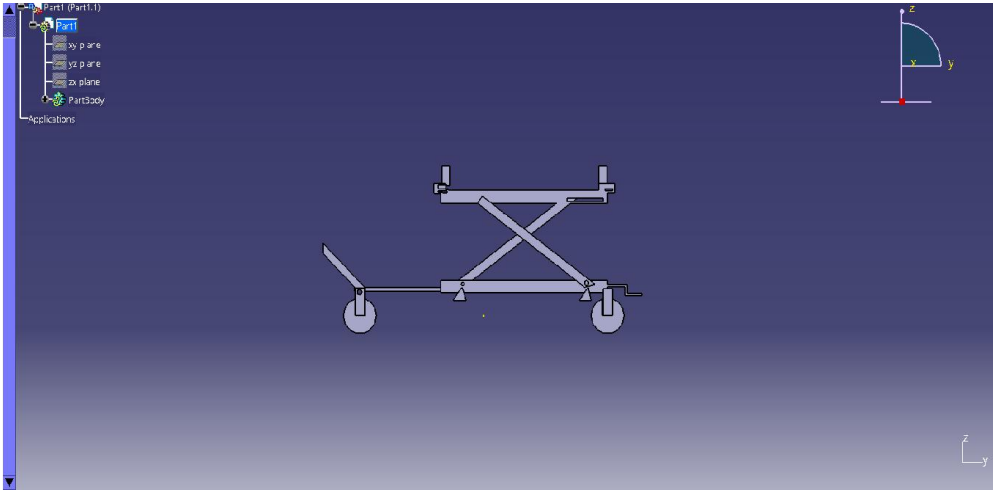


Figure 4.3. Free body diagram of scissor lift

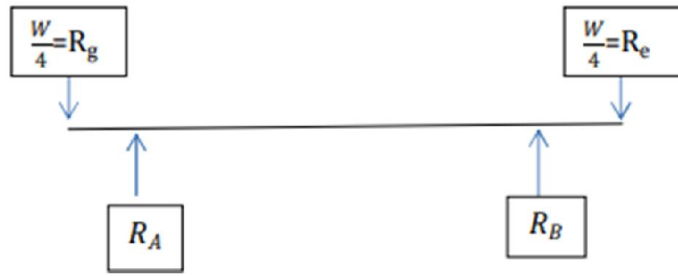


Figure 4.4. Free body diagram of scissors

According to Newton's second law formula

The summation of forces in the vertical direction which acts on each leg of the scissor lifter can be calculated (Gillespie, 2009)

$$\sum F_Y = 0 \text{ in equilibrium condition}$$

$$R_e' + R_e - \frac{w}{2}$$

\$R\_e\$ = reaction force during the free period

\$R\_e'\$ = reaction force during load acting (Id et al., 2019)

$$R_e' + R_e = \frac{w}{2} \dots \dots \dots (4.1)$$

$$\sum M_e = 0$$

\$M\_e\$ = momentum at e

$$\frac{wb}{4} R_e' b = 0$$

$$\sum F_Y = 0$$

$$R_e' = \frac{w}{4} R_A + R_D - R_g - R_e = 0$$

\$R\_g\$ = force at point g

\$R\_A\$ = reaction force of at point A

\$R\_D\$ = force at point D

$$R_A + R_D = \frac{w}{2} \dots \dots \dots (4.2)$$

From equation 1

$$R_e = \frac{w}{2} - R_e'$$

$$\sum M_A = 0$$

$$\frac{w}{2} - \frac{w}{4} - \frac{w}{4} R_D (S) + R_g a - R_e (s+a) = 0 \quad s = \text{distance}$$

$$R_D (s) + \frac{w}{4} a - \frac{w}{4} (s+a) = 0$$

$$R_D (s) + \frac{w}{4} (a-s-a) = 0$$

$$R_D (s) - \frac{w}{4} (s) = 0$$

$$R_D = \frac{w}{4} \frac{s}{s} = \frac{w}{4}$$

$$\text{From equation } 2R_A = W - R_D = \frac{w}{2} - \frac{w}{4} = \frac{w}{4}$$

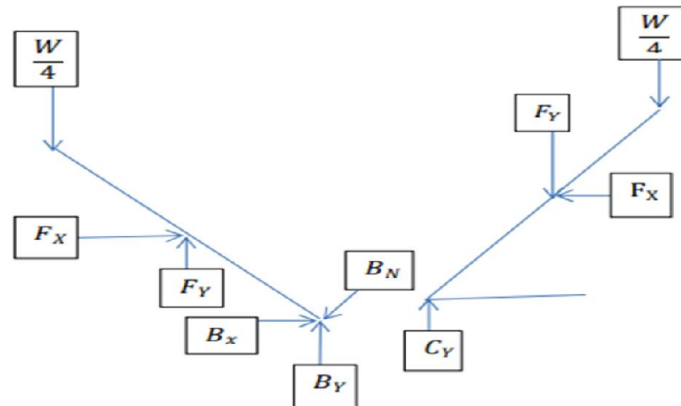


Figure 4.5. Force distributed

According to Newton's second law the resultant force acting on an object (Liu, 2019)

$$\sum F_x = 0, F_x + B_x = 0$$

$$F_x = -B_x \dots \dots \dots (4.3)$$

At equilibrium condition

$$\sum F_y = 0$$

$$B_y + F_y = \frac{w}{4} \dots \dots \dots (4.4)$$

$$\sum M_B = 0$$

$$\frac{L}{2} F_y \cos \theta + \frac{L}{2} F_x \sin \theta - \frac{w}{4} L \cos \theta = 0$$

$$F_y \cos \theta + F_x \frac{w}{2} \cos \theta = 0$$

$$F_y + F_x \tan \theta = \frac{w}{2} \dots \dots \dots (4.5)$$

For link CFD

$$\sum F_x = 0, C_x = -F_x \dots \dots \dots (4.6)$$

$$\sum M_C = 0$$

$$\frac{w}{4} L \cos \theta + F_y \frac{L}{2} \cos \theta - F_x \frac{L}{2} \sin \theta = 0$$

$$-F_y + F_x \tan \theta = \frac{w}{2} \dots \dots \dots (4.7)$$

$$\Sigma F_Y=0$$

$$C_Y - F_Y - \frac{w}{4} = 0$$

$$C_Y - F_Y = \frac{w}{4} \dots\dots\dots (4.8)$$

Solving equation 3&5

$$F_Y + F_X \tan \theta = \frac{w}{2} \dots\dots\dots 3$$

$$-F_Y + F_X \tan \theta = \frac{w}{2} \dots\dots\dots 5$$

$$2F_X \tan \theta = 2 \frac{w}{2}$$

$$F_X = \frac{w}{2 \tan \theta}$$

$$B_X = -F_X = - \frac{w}{2 \tan \theta}$$

$$F_Y = \frac{w}{2} - \frac{w}{2 \tan \theta} \tan \theta$$

$$F_Y = 0$$

$$B_Y + F_Y = \frac{w}{4}$$

$$B_Y = \frac{w}{4}$$

$$C_X = -F_X$$

$$C_X = - \frac{w}{2 \tan \theta}$$

The negative sign indicates the direction should be reversed, therefore

$$B_X = \frac{w}{2 \tan \theta} \quad \text{to the negative x-direction}$$

$$F_X = \frac{w}{2 \tan \theta}$$

$$F_Y = 0$$

$$B_Y = \frac{w}{4} \quad \text{to the positive y-direction}$$

$$C_X = \frac{w}{2 \tan \theta} \quad \text{to the positive x-direction}$$

$$F_Y = \frac{w}{4} \quad \text{to the positive y-direction}$$

The force exerted on the screw is

$$W_S = 2B_X = \frac{w}{\tan \theta}$$

To perform the analysis of the components to need to make the following assumptions

- a. The length of the link is 500mm.
- b. Maximum height the scissor lifts  $h_{max} = 400\text{mm}$

c. Maximum height the scissor lowers  $h_{min} = 100\text{mm}$

Calculating the minimum angle, the link makes from the horizontal when the scissor is at a lower Position

$$\theta_{min} = \sin^{-1}\left(\frac{h_{min}}{L}\right) = \sin^{-1}\left(\frac{100}{500}\right) = 11.54^\circ$$

Calculating the maximum angle, the link makes from the horizontal the scissors are at a higher position.

$$\theta_{max} = \sin^{-1}\left(\frac{h_{max}}{L}\right) = \sin^{-1}\left(\frac{400}{500}\right) = 53.13^\circ$$

Determine the maximum distance point B of the link slides horizontally to raise the load from 100mm to 400mm.

Let the maximum length to be slide  $s_{max} =$

$$s_{max} = L \cos \theta_{min} - L \cos \theta_{max}$$

$$s_{max} = L (\cos \theta_{min} - \cos \theta_{max})$$

$$s_{max} = 500(\cos 11.54 - \cos 53.13)$$

$$s_{max} = 189.89\text{mm} \quad \text{say } 190\text{mm}$$

#### 4.1.3. Design of the screw

The power screw is the ultimate component that takes up the load which is lifted or lowered by lift. This design can make the lift more compact and much more suitable for medium-scale work. In this case, along with lifting a bus drive shaft, they are also going to make an effort to move the drive shaft to a short distance by incorporating a roller mechanism. The complete design of the screw is designed, making both economically feasible and improving productivity. The calculations and diagrams of the design are shown below.



Figure 4.6. Screw

The force exerted on the screw (Patel et al., 2017)

$$(F_x) = \frac{w}{2 \tan \theta} \text{----- (4.9)}$$

Hence the axial force exerted on the screw is maximum when  $\theta$  is minimum.

Considering the force diagram,  $w_s = \frac{w}{2 \tan \theta_{min}} = 2 * 1469.29 N = 2938.58$

**Material selection for screw**

Select steel C-30 material because it is mostly available and economical. It has the The properties of this material are hardened steel and a good melting point.

$$\sigma_{tensile/compressive} = 500 N/mm^2$$

$$\sigma_{yield} = 300 N/mm^2$$

**Analysis**

Taking factor of safety 3 & service factor is 1.6 (based on the material property) and the factor of safety is taken as 3 to ensure the safety of the structure. To consider the factor of safety of less than 3 the dimensions will increase and the structure will become bulky hence it will not be portable to consider the factor of safety of more than 3 the dimension will be very small which will not be suitable for the application.

$$\tau_y = 0.5 * \sigma_{yield} = 300 N/mm^2 = 150 N/mm^2$$

$$\sigma_{av} = \frac{\sigma_c}{F.S} = \frac{500 N/mm^2}{3 * 1.6} = 104.2 N/mm^2$$

$$\tau_{av} = \frac{\tau_y}{F.S * k} = \frac{150 N/mm^2}{3 * 1.6} = 31.25 N/mm^2$$

$$\sigma_{av} = 2 * \frac{w_s}{\pi * d_c^2} \text{ or } 104.2 = 2 * \frac{1469.29}{\pi * d_c^2} \text{----- } d_c = 5.99 mm$$

$$d_c = 6 mm \quad d_c = \text{is core diameter}$$

The square thread power screw is chosen because of its high load-carrying capacity According to is fine series 4694-1968(reaffirmed 1996) we choose a standard core diameter

$$d_c = 12 mm$$

Were: -

$$d_c = \text{core diameter}$$

$$P = 2$$

$$P = \text{Pitch}$$

Nominal Diameter = pitch + core diameter ( Ezurike and Okwu 2018)

$$D = \frac{12+14}{2} = 13 \quad D = \text{mean diameter}$$

$$d_1 = 14 \text{mm} \quad d_1 = \text{nominal diameter}$$

$$H = 1.25$$

H = depth of thread

$$\tan \alpha = \frac{p}{\pi d_m} = \frac{2}{\pi * 13} = 0.04897$$

$$\alpha = 2.8^\circ$$

Assuming the average quality of materials and workmanship and average working conditions (Timoty n.d, 2018.) Take the coefficient of friction 0.18.,  $\mu = \tan \phi = 0.18$ ,  $\phi = \tan^{-1}(0.18) = 10.204$  To create this angle ( $\phi$ ). Now torque required to raise the load by overcoming friction becomes

$$T = \frac{w \tan(\alpha + \phi) * d_m}{P} = \frac{293.8 \tan(10.204 + 2.8) * 13}{2}$$

$T = 4411.165 \text{ N/mm}^2$  as it can be observed  $\tan \phi > \tan \alpha$  which means that the torque required to lower the load is positive, consequently the screw is self-locking. Checking maximum shear stresses induced in the screw

$$\tau_{max} = \frac{16T}{\pi d_c^3} = \frac{16 * 4411.165}{\pi * 12^3} = 13 \text{N/mm}^2$$

$$\sigma = \frac{F}{A} = \frac{2938.58 * 4}{\pi * 12^2} = 25.98 \text{N/mm}^2$$

Now checking the maximum principal stress (compressive) I have

$$\sigma_{max} = \frac{\sigma}{2} + \frac{1}{2} \sqrt{\sigma^2 + 4\tau^2} = \frac{25.38}{2} + \frac{1}{2} \sqrt{25.98^2 + 4 * 13^2}$$

$\sigma_{max} = 31.37 \text{N/mm}^2$  which is less than  $104.2 \text{N/mm}^2 (\sigma_{av})$  thus it is safe.

Finally, checking the maximum shear stress induced in the screw we have

$$\tau_{max} = \frac{1}{2} \sqrt{\sigma^2 + 4\tau^2} = \frac{1}{2} \sqrt{25.98^2 + 4 * 13^2} = 18.38 \text{N/mm}^2, \text{ which is less than } 31.25 \text{N/mm}^2,$$

which is less than  $31.25 \text{N/mm}^2 (\tau_{av})$ , thus the design is safe.

### The efficiency of the screw

$\eta = \frac{T_o}{T}$ , where  $T_o$  = torque to lift the load without friction

$$\eta = \frac{w \tan \alpha d_m}{2T} = \frac{2938.58 * 13 * \tan 28}{2 * 4411.165} = 0.212$$

$\eta = 21.2\%$ , which is less than 50%, resulting self – looking screw.

Arm length of the handle (L) =100mm

Average human force=250N

To determine the amount of force required to apply torque, 4411.165N.mm through a radius of 150mm.

$$F_{applied} = \frac{T}{L} = \frac{4411.165 \text{ N.mm}}{100} = 44.112 \text{ N}$$

This is much less than the average human force therefore it only requires one man to do the job.

#### 4.1.4. Design of nut screw

To have good stability and also to prevent the undesirable movement of the screw, screw nut designing is essential. During the modeling of the components, the material selection is carried out simultaneously based on the design considerations related to loads. The stress and strain concentrations, and deformation, have been found by applying the drive shaft load. The material of the nut is assumed to be Grey Cast Iron.



Figure 4.7. Screw Nut

#### Material selection for nut

The material for this nut is Grey Cast Iron Type A which is available in the factory with the

Following mechanical property

Yield strength=310N/  $mm^2$

Tensile strength=139 N/  $mm^2$

Compressive strength=604 N/  $mm^2$

Shear strength=192 N/  $mm^2$

#### Analysis

Assuming a factor of safety 3

The permissible shear stress becomes

$$\sigma_{av} = \frac{\sigma_y}{F.S} = \frac{310 \text{ N/mm}^2}{3} = 103.33 \text{ N/mm}^2$$

$$\tau_{av} = \frac{\tau_y}{F \cdot S} = 192 \text{ N/mm}^2$$

Let,

N = number of threads in contact with the screw

H = height of the nut

T = thickness of the screw

$p_b$  = Bearing pressure

Assuming that the load W is distributed uniformly over the cross-sectional area of the nut, therefore bearing pressure between the threads (Chaturvedi, 2017b)

$$p_b = \frac{w_s}{\frac{\pi}{4}(d_o^2 - d_c^2)n}$$

Then rearranging

$$N \frac{4 \cdot w_s}{\pi(d_o^2 - d_c^2)p_b} = \frac{4 \cdot 2938.58}{\pi \cdot (14^2 - 12^2) \cdot 17.5} = 4.11 \quad \text{say 5 teeth}$$

The minimum height of the nut becomes

$H = n \cdot p$ , where p is the pitch

$$H = 3 \cdot 5 \cdot 2 = 30 \text{ mm}$$

Width of the nut (b) =  $1.5 \cdot d_o = 1.5 \cdot 14 = 21 \text{ mm}$

Check for shear failure of screw nut threads

$$\tau = \frac{w_s}{n \cdot \pi \cdot d_c \cdot p/2} = \frac{2938.58}{5 \cdot \pi \cdot 12 \cdot 2/2} = \frac{13.36 \text{ N}}{\text{mm}^2} \text{ Which is less than } 64 \text{ N/mm}^2 (\tau_{av})$$

Therefore, the design is safe.

Check for shear failure of screw threads

$$\tau = \frac{w_s}{n \cdot \pi \cdot d_c \cdot p/2} = \frac{2938.58}{5 \cdot \pi \cdot 12 \cdot 2/2} = 15.636 \text{ N/mm}^2 \text{ which is less than } 31.2536 \text{ N/mm}^2 (\tau_{av})$$

Therefore, the design is safe.

#### 4.1.5. Design of the link

This component is subjected to buckling load and bending load tending to break or cause bending of the components. Hence based on strength, stiffness, plasticity, and hardness. A recommended material is structural steel.



Figure 4.8. Link

$$\text{Load acting on the link (p)} = \frac{w}{4} = \frac{600}{4} = 150\text{N}$$

Width of the link (w) = 3\*t where t=thickness of the link

The cross-sectional area of link (A) = 3\*t<sup>2</sup>

$$\text{Moment of inertia (I)} = \frac{1}{12} * t * t^3 = \frac{1}{4} t^4$$

$$\text{Radius of gyration (r)} = \sqrt{\frac{I}{A}} = \sqrt{\frac{1/4 t^4}{3t^2}} = 0.28867t$$

Length = 0.5m

#### Material selection for link

To use structural steel (ASTMA36) with the following properties

$$\sigma_{t/c} = 400 \text{ N/mm}^2$$

$$\sigma_y = 250 \text{ N/mm}^2,$$

$$E = 200 \text{ Gpa}$$

#### Analysis

Taking the factor of safety (N) = 3, P<sub>cr</sub> = N\*P<sub>a</sub> = 3\*150 = 450N since for buckling of the link in the vertical. In the plane, the ends are considered as hinged, therefore the equivalent length of the link.

$$L = 500 \text{ mm}$$

#### According to Rankin's formula, buckling load (W<sub>cr</sub>),

Let = k\*L = 1.0\*500mm = 500mm, where k=1.0 for pin ends.

$$C_c = \sqrt{\frac{2E\pi^2}{\sigma_y}} = \sqrt{\frac{2\pi^2(200*10^3)}{250}} = 125.65$$

Assume;  $\frac{L_e}{r} > C_o$

$$P_{Cr} = \frac{\pi^2 EI}{(kL)^2}, \quad \text{Solving for I}$$

$$I = \frac{P_{Cr}(kL)^2}{\pi^2 E} = \frac{450(0.5)^2}{\pi^2 * 200 * 10^9} = 5.6993 * 10^{-11} \text{ m}^4 \text{ but}$$

$$I = \frac{1}{4} t^4 = 5.6993 * 10^{-11} m^4,$$

$$t^4 = 2.27973 * 10^{-10} m^4$$

$$t = 3.8857 * 10^{-3} m \quad \text{say } t = 4 \text{ mm}$$

$$\frac{Le}{r} = \frac{0.5}{0.28867 * 4 * 10^{-3}} = 433.02$$

Since,  $\frac{Le}{r} = 433.02 > 125.66 = Cc$  the thickness  $t = 4 \text{ mm}$  is acceptable for resisting buckling.

$$t = 4 \text{ mm}$$

$$\text{Width} = 3t = 3 * 4 = 12 \text{ mm}$$

The link is also subjected to bending

Taking link AFB

$A_N =$  Force normal to the link at point A

$F_N =$  Force normal to the link at point F

$B_N =$  Force normal to the link at point B

$$A_N = \frac{w}{4} \cos \theta = 150 \cos 11.54 = 146.97 \text{ N}$$

$$F_N = F_x \sin \theta = 1469.29 * \sin 11.54 = 293.93 \text{ N}$$

$$B_N - B_Y \cos \theta - B_X \sin \theta = 0$$

$$B_N = B_Y \cos \theta + B_X \sin \theta = 150 \cos 11.54 + 1469.29 \sin 11.54 = 440.9 \text{ N}$$

From the bending moment

$$/M_{max}/ = 36742.5 \text{ Nmm}$$

$$S = \frac{/M_{Max}/}{\sigma_{av}} = \frac{36742.5 \text{ Nmm}}{83.33 \text{ N/mm}^2} = 440.93 \text{ mm}^3$$

$$\text{But, } S = \frac{I}{t/2} = \frac{\frac{1}{4} t^4}{\frac{1}{2} t} = \frac{1}{2} t^3 = 440.93 \text{ mm}^3$$

$$t = 9.6 \text{ mm} \quad \text{say } t = 10 \text{ mm}$$

$$\text{Since } w = 3t = 3 * 10 = 30 \text{ mm}$$

Therefore, the cross-section of the link is safe.

#### 4.1.6. Design of pin

Base, links, and other components are connected by using pin connections where these pins are made of steel material which is having a cylindrical cross-section. The roller guide bars are designed such that the roller is subjected to minimal wear. It is designed as a round bar which will

have minimum wear during the operation of the mechanism. The pins are inserted after the press-fitted bush in the bore everywhere to ease the maintenance of the scissor lift as and when required.



Figure 4.9. Pin

### Material selection for pin

To use the same material used for the link is hot-formed structural steel which has the following properties

$$\sigma_{t/c} = 400 \text{ N/mm}^2$$

$$\sigma_y = 250 \text{ N/mm}^2$$

$$E = 200 \times 10^9 \text{ N/m}^2$$

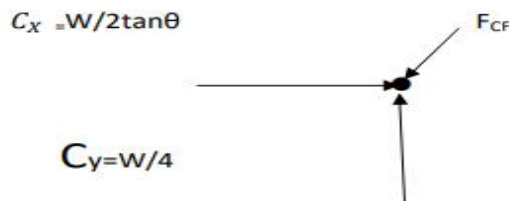
### Analysis

Taking factor of safety (F.S) = 3

$$\sigma_{av} = 250/3 = 83.33 \text{ N/mm}^2, \quad \tau_y = 250/2 = 125 \text{ N/mm}^2$$

$$= \tau_{av} = \tau_y / \text{F.S} = 125 \text{ N/mm}^2 / 3 = 41.67 \text{ N/mm}^2$$

To take point C



$$\Sigma F = 0$$

$$F_{CF} \cos \theta + C_x = 0$$

$$F_{CF} = C_x / \cos \theta = 1469 / \cos 11.54 = 1499.6 \text{ N}$$

Since the pin is in double shear

$$\tau_{av} = F_{CF} / 2A = 2F_{CF} / \pi * D^2 \quad = D = (2F_{CF} / \pi * \tau_{av})^{1/2} = (2 * 1499.6 / \pi * 41.67)^{1/2} =$$

$$4.786 \text{ Mm}$$

$$\text{Say } D = 5 \text{ mm}$$

The pin also subjected to bending

$\sigma_b = 16F_{CF} (t_2 + 3g) / 3 * \pi * d^3$  where  $g$ =clearance b/n the link and the bracket

$d^3 = 16F_{CF} (t + 3g) / 3 * \pi * \sigma_b$   $t$ =thickness of the link

$= 16 * 1499.6 * (10 + 3 * 0.5) / 3 * \pi * 83.33$

$d$ =pin diameter

$d = 7.06\text{mm}$

$\sigma_b$  =bending stress

Taking standard value  $d = 10\text{mm}$

To select a thread with pitch= $1.5\text{mm}$

And  $d = 10\text{mm}$

Depth of thread = $0.920\text{mm}$

At this point check the crushing stress between the  $10\text{mm}$  thick link and  $10\text{mm}$  diameter Pin

$\sigma_b F_{CF} / t * d = 1499.6 / 10 * 10 = 14.996 \text{ N/mm}^2$  which is much less than the allowable stress.

Therefore, the design is safe.

#### **4.1.7. Design of the bar which supports the weight of the drive shaft**

##### **Material selection**

To choose hot-formed structural steel with the following properties

$\sigma_{tensile} = 400 \text{ N/mm}^2$ ,

$\sigma_{yield} = 250 \text{ N/mm}^2$

##### **Analysis**

Taking factor of safety 3

$\sigma_{av} = \sigma_y / 3 = 250 \text{ N/mm}^2 / 3 = 83.33 \text{ N/mm}^2$

The bar that supports the load is considered a simply supported bar and the load is A concentrated load is applied at the center of the bar.

Where  $P = W / 2 = 600 / 2 = 300\text{N}$

The shear force diagram and bending moment diagram are shown below.

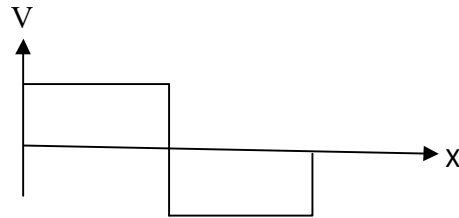


Figure 4.10. Shear force diagram

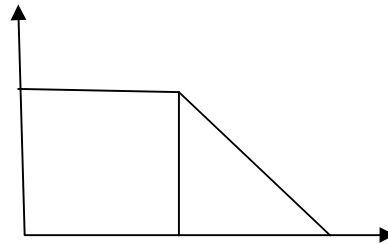


Figure 4.11. Bending moment diagram

From the shear force diagram or bending moment diagram we can determine the maximum Bending moment.

$$\text{i.e., } M_{Max} = PL/4$$

Also, know that

$$\sigma_{av} = M_{Max} / S = \sigma_{av} \text{ where } S = 1/r$$

r = radius

$$S = M_{Max} / \sigma_{alu} = PL/4 \sigma_{au} = (300 \cdot 600) / 4 \cdot 83.33 = 540.02 \text{ mm}^3$$

S = section modulus

$$S = 0.54 \cdot 10^3 \text{ mm}^3$$

From a table to selected a hollow rectangular cross-section bar that has a section modulus (s) =  $1.34 \cdot 10^3 \text{ mm}^3$  and the following dimensions

Where width (B) = 40mm

t=2mm height (H) =20mm

#### 4.1.8. Selection of ball bearings

The bearings chosen for the screw shaft are single-row deep groove ball bearings for machines used intermittently whose break down would not have serious consequences, the life of the bearing is taken 4000 hours ( $L_H$ )

Now to have calculated the dynamic load factor first to choose the bearing.

$$w_r \text{ (Radial load)} = 150\text{N}$$

$w_a$  (Axial load) = 1469.29N

An average human can apply a force = 250N and a power of = 100W

$$P_{in} = F \cdot r \cdot w_p$$

$$w_p = 100/250 \cdot 0.15 = 2.7 \text{ rad/sec}, N = 25.5 \text{ rpm}$$

$$\begin{aligned} \text{Life of the bearing in revolution (L)} &= 60 \cdot N \cdot L_H \\ &= 60 \cdot 25.5 \cdot 4000 \\ &= 6120000 \text{ rev} \end{aligned}$$

$w_a/W_r = 1469.29/150 = 9.8$ , taking  $w_a/C_o = 0.5$ , from tables

$$X = 0.56, y = 1.0$$

Now dynamic equivalent load factor  $W = X \cdot v \cdot W_r \cdot y \cdot W_A$ , taking  $v$  as 1.

$$\begin{aligned} W &= 0.56 \cdot 1 \cdot 150 \cdot 1.0 \cdot 1469.29 \\ &= 1553.3 \text{ N} \end{aligned}$$

Service factor  $K_s$  for ball bearing is 1, so bearing should be selected for 1553.3N

Now to know that basic dynamic load rating is

$$C = W (L/10^6)^{1/K} = 1553.3 (6120000/10^6)^{1/3}$$

$$C = 2.841 \text{ KN}$$

Matching the above value with values from the Indian catalog. I have chosen bearing number 200.

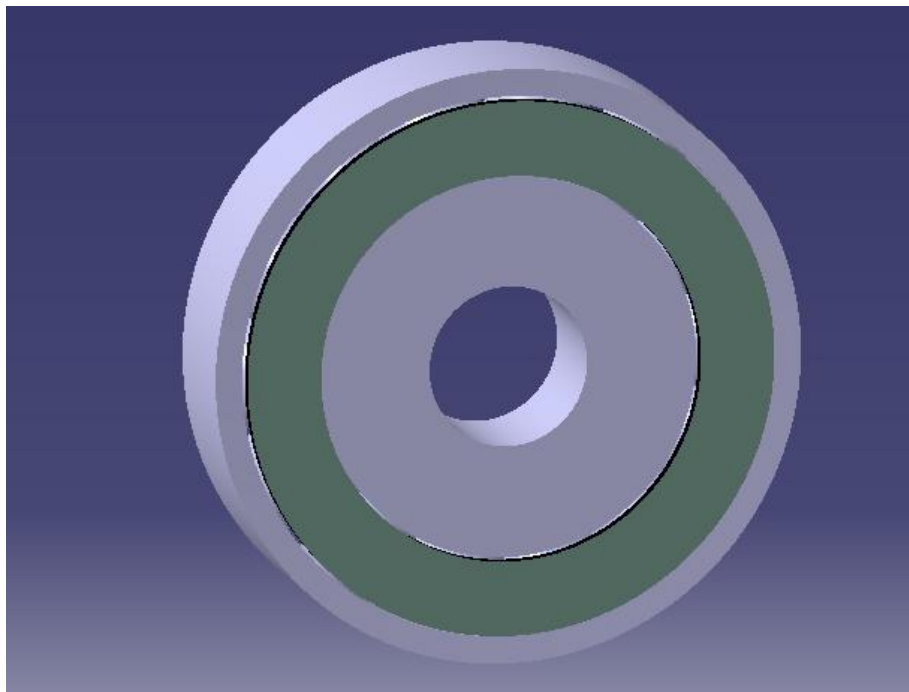


Figure 4.12. Bearing

#### 4.1.9. Design of the base

The base is made up of rectangular cross-section beams that can support the whole load applied to it. These beams are welded together so that they can support the load applied to them and transfer the load to the wheels.

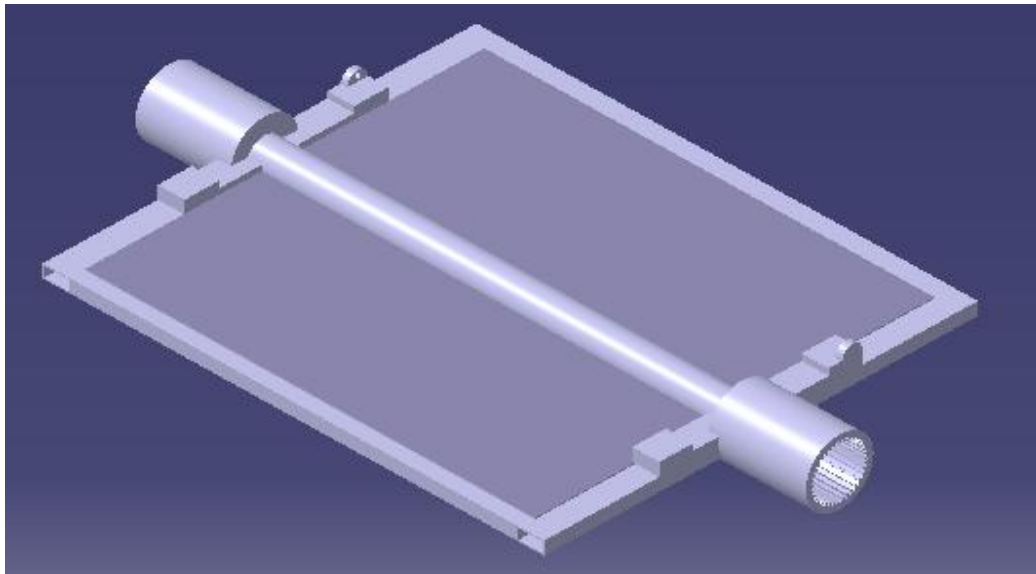


Figure 4.13. Base

#### Material selection

Select hot-formed structural steel b/c it is economical and widely available in the industry. It has the following properties,

$$\sigma_{tensile}=400\text{N/mm}^2, \sigma_{yielded}=250\text{N/mm}^2, \text{density } (\rho)=7860\text{kg/m}^3$$

#### Analysis

Taking factor of safety 3

$$\sigma_{av}=\sigma_y/3=250\text{N/mm}^2/3=83.33\text{N/mm}^2$$

The bar is considered a simply supported bar and the loads (I.e., Weight of the lifter and vertical load on the bracket due to the weight of the drive shaft) are uniformly distributed over 700mm length

$$L=1000\text{mm}$$

From CATIA v5 measurement loads found that the volume of the lifter above the base

$$(v_s)=0.003\text{m}^3$$

$$\text{Mass of lifter } (m_2)=\rho * v_s=7860\text{kg/m}^3*0.003\text{m}^3=23.6\text{kg}$$

$$\text{Weight } (w_2)=m*g=23.6\text{kg}*10\text{m/s}^2=236\text{N}$$

Weight transferred to the bracket ( $w_b$ ) =  $236\text{N}/2 = 118\text{N}$  ↓

$C_Y = B_Y = W/4 = 150\text{N}$

Total vertical force ( $F$ ) =  $w_b + C_Y + B_Y = 118 + 150 + 150 = 418\text{N}$  Distributed load ( $W$ ) =  $F / 700\text{mm} = 418\text{N}/700\text{mm} = 0.597\text{N/mm}$

**From statics**

$R_c = 146.3\text{N}$ ,  $R_a = 271.7\text{N}$

The shear force and bending moment diagrams are

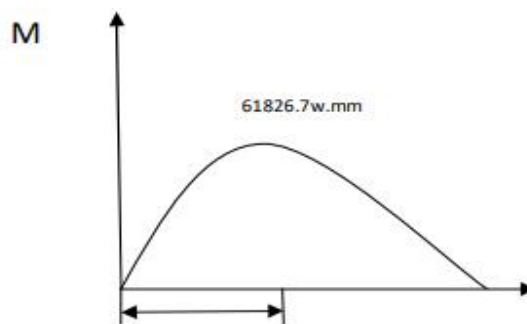


Figure 4.14. Bending moment diagrams

The maximum bending moment occurs at point d

$M_{max} = 61826.7\text{Nmm}$ , it also knows that  $\sigma_{av} = M_{max} / S$

$S = M_{max} / \sigma_{av} = 61826.7\text{Nmm} / 83.33\text{N/mm}^2 = 741.95\text{mm}^3$   $S = 0.74195 \cdot 10^3\text{mm}^3$

From a table select a section modulus ( $s$ ) =  $1.34 \cdot 10^3\text{mm}^3$  which is close to the section modulus to have calculated, from the same table to get the following dimensions for the select section modulus.

Were

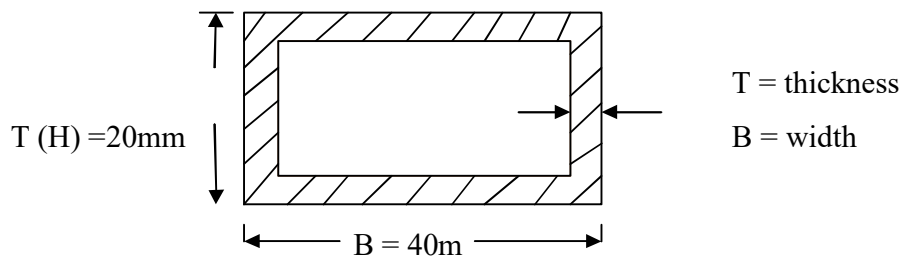


Figure 4.15. Section module

**4.1.10. Welding design of the base bracket**

The bracket is attached to the base by a long fillet weld subjected to transverse and load parallel weld. Let

$S$  = size of weld in mm

t= throat thickness in mm

l=length of the weld

$$\tau_{max}=25\text{N/mm}^2$$

Design of the weld considering only load parallel to the weld

$$p_x=C_x=B_x=1469.29\text{N}$$

$$E=30\text{mm}$$

$$L=350\text{mm}$$

$$A=2t*L=2*0.707*S*350=494.9*S$$

$$\tau=p_x/A=1469.29/494.9*S=2.99/S*\text{N/mm}^2$$

$$M=P_x*e=1469.29*30=44078.7\text{N/mm}$$

Section modulus of the weld through the throat

$$Z=(S*L^2)/4.242=(S*350^2)/4.242=28877.9*S \quad Z= \text{shear modulus}$$

$$\sigma_b = \frac{M}{Z} = \frac{44078.7}{28877.9*S} = 1.53/S \text{ N/ mm}^2$$

To know that maximum shear stress ( $\tau_{max}$ )

$$2*S=1/2(\sigma_b^2+4 \tau^2)^{1/2}=1/2((1.53/S)^2+4(2.99/S)^2)^{1/2}=3.1/S$$

S=0.123mm, which is the minimum size of the weld

Therefore, we selected, S = 2mm

## 4.2. Selection of wheel

The wheels chosen for this machine are not the ordinary types that are made up of rubber filled with air. Their minimized contact area helps the operator to push them with little effort. The wheels are available in Bishoftu Automotive Industry. Also, the wheels should be able to raise the base from the ground 150 mm.



Figure 4.16. Mechanical shaft lifter wheel

### **Model and Analysis Failures through Factor Safety**

**Safety Factor**, is used to provide a design margin over the theoretical design capacity to allow for uncertainty in the design process. Factor of safety is recommended by the conditions over which the designer has no control, that is to account for the uncertainties involved in the design process and the difference between the safety factor and design factor (design safety factor) is as follows: The safety factor, or yield stress, is how much the designed part actually will be able to withstand (first usage from above). The design factor, or working stress, is what the item is required to be able to withstand (second usage). The design factor is defined for an application (generally provided in advance and often set by regulatory building codes or policy) and is not an actual calculation, the safety factor is a ratio of maximum strength to intended load for the actual item that was designed.

$$\text{Factor of safety} = \frac{\text{Yield stress}}{\text{Working stress}}$$

- ✓ The design load is the maximum load the part should ever see in service
- ✓ By this definition, a structure with an FOS of exactly 1 will support only the design load and no more. Any additional load will cause the structure to fail. A structure with an FOS of 2 will fail at twice the design load.

### **4.3. Manufacturing methods**

The screw-driven scissor-type propeller shaft lifter is designed by observing the material and spare components available in the automotive industry. For example, the screw and the bending are difficult to manufacture due to bureaucracy problems. But there are used and functional screws and bearing available in the industry. The links and pins can be manufactured on the convectional machinery. The brackets are manufactured from a 5mm thick plate.

### **Engineering drawings and specifications**

Engineering drawings and specifications are essential for the detailed design and manufacturing of the mechanical scissor screw type drive shaft lifter machine. These drawings provide precise and standardized representations of the components, dimensions, tolerances, materials, and other critical details.

### 3D assembly drawing

3D assembly drawing focus on each component of machine and provide specific directions of the parts to be assemble. These drawings allow for accurate manufacturing and ensure proper fit and functionality of the parts.



Figure 4.17. Assembly drawings

## CHAPTER FIVE

### 5. RESULT AND DISCUSSION

The transition from developing the scissor lift model in CATIA V5 to importing it into ANSYS Workbench represents a crucial phase in the analysis and validation of the design. CATIA V5 is widely used for creating detailed 3D models of mechanical components, while ANSYS Workbench specializes in simulation and finite element analysis. This seamless integration streamlines the process, as it eliminates the need for manual model reconstruction in the simulation software, ensuring that the design representation remains consistent. Upon importing the model into ANSYS Workbench, the first critical step is to assign materials to the model's components. Material properties, such as tensile strength, yield strength, and elasticity, play a fundamental role in determining the lift's structural behavior under the applied load.

The subsequent phase involves meshing, wherein the complex 3D model is discretized into smaller elements. Meshing is a vital step in the finite element analysis process, as it divides the model into finite elements, allowing ANSYS to perform calculations at these discrete points. Proper meshing is essential for obtaining accurate results, as it affects the precision of the analysis. Once meshing is complete, the focus shifts to performing a strength analysis to assess the lift's performance under the specified mean load. The values of total deformation (0.36mm), equivalent elastic stress (22.88MPa), and equivalent elastic strain  $3.24 \times 10^{-1}$  mm represent critical output parameters obtained from the analysis. These values serve as crucial indicators of the lift's structural integrity, helping to determine if the design meets the desired performance criteria and if any design modifications are necessary to ensure safety and functionality.

The combination of CATIA V5 and ANSYS Workbench presents a robust methodology for designing and validating complex mechanical systems. This process enables engineers to seamlessly transition from the design phase to simulation and analysis, ensuring that the designed scissor lift meets the specified performance criteria. The analysis results, including total deformation, equivalent elastic stress, and equivalent elastic strain, provide valuable insights into the lift's behavior under the mean load, ultimately guiding design improvements and assuring its compatibility and reliability in real-world applications.

## 5.1. Mesh Generation

In order to conduct a comprehensive analysis of the scissor lifting system using the finite element technique, a step-wise approach was adopted to simplify the simulation process. Initially, non-critical components of the system, such as the platform and base plate, were removed from the analysis. By eliminating these components, the simulation could be executed more efficiently, significantly reducing the computational time and resources required. This approach allowed for a focused examination of the essential elements that influence the system's mechanical behavior. It is worth noting that these non-critical parts, though essential for the complete system's functionality, do not directly impact the critical aspects of load-bearing and structural stability, which were the primary focus of the analysis.

For the analysis, the system was securely fastened at four specific locations, which were attached to the base plate at an angle of  $45^\circ$ . These anchor points were strategically chosen to represent real-world conditions where the lifting system would be subjected to both vertical and lateral loads. By securing the system in this manner, it was possible to study the system's response to forces acting at different angles, thereby providing valuable insights into its overall stability and load-bearing capacity.

To accurately represent the mechanical conditions, the weights applied to the platform and its maximum load capacity were considered. The platform's own weight and its maximum load capacity were determined to be 236 N and 418 N, respectively. These weights were then assigned to the four attachment points on the platform during the simulation. This ensured that the finite element analysis accounted for the true operational conditions, where the lifting system would be subjected to varying loads within the specified weight range. The results of this analysis, which simulated realistic loading scenarios, were pivotal in evaluating the system's structural integrity and efficiency.

The approach of removing non-critical components and focusing on the essential structural elements during the finite element analysis allowed for a more efficient and targeted investigation of the scissor lifting system. By securing the system at strategic angles, simulating realistic loading conditions, and considering the platform's weight and maximum load capacity, the analysis provided valuable insights into the system's performance under real-world operational conditions. This step-wise approach was essential for understanding the system's mechanical behavior, ensuring its stability, and optimizing its load-bearing capacity.

The integration of the scissor lift model into ANSYS Workbench, following its initial creation using CATIA V5, constitutes a crucial phase in this study. By directly importing the model into ANSYS, we establish a seamless workflow for subsequent analysis. The first step in this process is the assignment of appropriate material properties to the model, ensuring that the structural components are represented accurately in the simulation. Once the material assignment is complete, the mesh generation process begins. Generating a mesh, with a focus on achieving a high-density and high-quality structure, is essential to accurately represent the geometry in the finite element analysis. This ensures that the model captures the complexities of the scissor lift's structure, as even minor variations can significantly impact the results of the subsequent strength study.

The primary focus of this strength study is to assess the scissor lift's performance under a mean load of 80kg, a critical aspect for its intended application. To evaluate the lift's compatibility, we look into key metrics, including Total deformation, Equivalent elastic stress, and Equivalent elastic strain. These parameters provide insights into how the lift responds to the applied load, whether it deforms within acceptable limits, and whether the material experiences stress and strain levels within the safe operating range. The accuracy and reliability of these results rely on the mesh structure's quality, emphasizing the importance of high-density meshing. The combination of accurate material properties, precise meshing, and rigorous analysis using ANSYS Workbench allows for a comprehensive evaluation of the scissor lift's performance, aiding in its optimization and further development.

Figure 5.1, which depicts the mesh structure of the scissor lifting system after loading, showcases the intricate details of the model. The automatic generation of this mesh at high density and quality is a testament to the advanced capabilities of ANSYS Workbench. The figures, 361,027 nodes and 176,394 elements, attest to the fine resolution achieved, particularly for the base part of the scissor lift. These numbers exemplify the depth of detail captured in the simulation, enabling a thorough analysis of the scissor lift's response to the applied load. However, the discussion doesn't end here; the outcomes of this study will depend on how well these elements interact and represent the real-world behavior of the scissor lift, underlining the importance of mesh quality and model accuracy in obtaining meaningful results.

In general, the integration of the scissor lift model into ANSYS Workbench, followed by the assignment of material properties, mesh generation, and strength study, constitutes a critical part

of this research. The quality of the mesh and the accuracy of material properties are paramount in achieving reliable simulation results. The evaluation of Total deformation, Equivalent elastic stress, and Equivalent elastic strain under an 80kg mean load is essential to understand the scissor lift's performance and its compatibility with intended applications. The highly detailed mesh structure, comprising over 361,000 nodes and 176,000 elements, demonstrates the precision of the simulation and its ability to provide valuable insights into the scissor lift's structural behavior under load.

**Table 5.1 Mesh data**

Object Name	Mesh
State	Solved
<b>Display</b>	
Display Style	Use Geometry Setting
<b>Defaults</b>	
Physics Preference	Mechanical
Element Order	Program Controlled
Element Size	40.0 mm
<b>Sizing</b>	
Use Adaptive Sizing	Yes
Resolution	0
Mesh Defeaturing	Yes
Defeature Size	Default
Transition	Fast
Span Angle Center	Coarse
Initial Size Seed	Assembly
Bounding Box Diagonal	1934.9 mm
Average Surface Area	1401.2 mm <sup>2</sup>
Minimum Edge Length	1.0057e-003 mm
<b>Quality</b>	
Check Mesh Quality	Yes, Errors

Error Limits	Aggressive Mechanical
Target Element Quality	Default (5.e-002)
Smoothing	Medium
Mesh Metric	Skewness
Min	1.3057e-010
Max	1.
Average	0.61455
Standard Deviation	0.28616
<b>Inflation</b>	
Use Automatic Inflation	None
Inflation Option	Smooth Transition
Transition Ratio	0.272
Maximum Layers	5
Growth Rate	1.2
Inflation Algorithm	Pre
View Advanced Options	No
<b>Advanced</b>	
Number of CPUs for Parallel Part Meshing	Program Controlled
Straight Sided Elements	No
Rigid Body Behavior	Dimensionally Reduced
Triangle Surface Mesher	Program Controlled
Topology Checking	Yes
Pinch Tolerance	5.e-002 mm
Generate Pinch on Refresh	Yes
<b>Statistics</b>	
Nodes	361027
Elements	176394

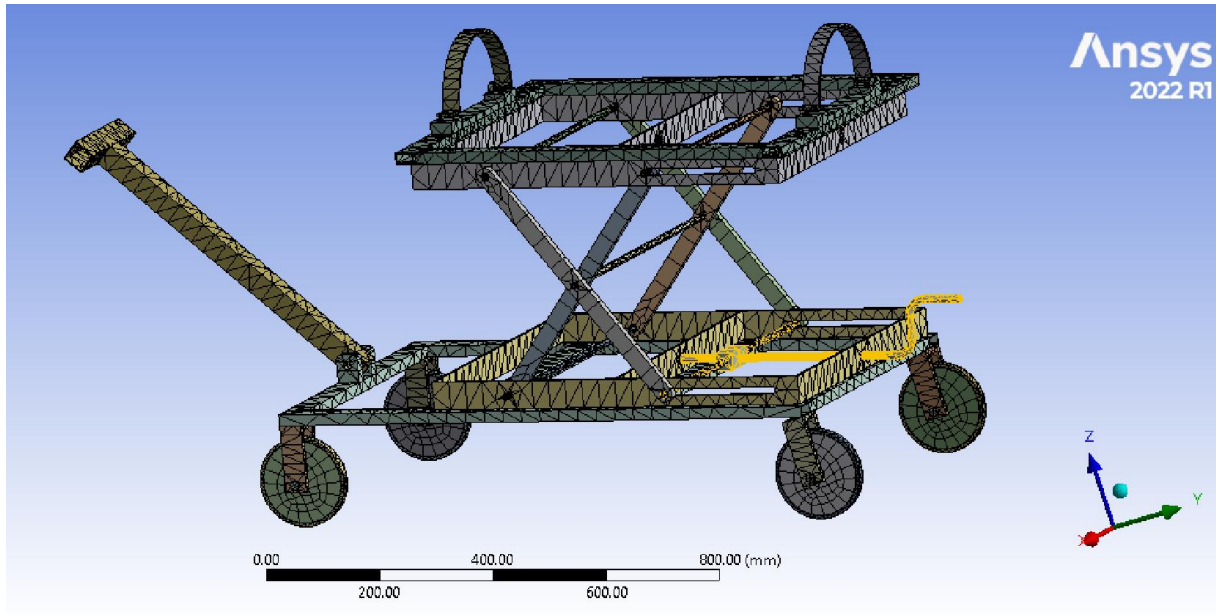


Figure: 5.1. Geometry and Mesh of the scissor lifter model

Generally, the overall geometry of this lifting machine enables the operator not only to carry the load but also to transport it. The simplicity of the components and good lubrication makes it have a long lifetime and easy to maintain. The engineers and propeller shaft assemblies believe that this machine greatly helps the shop in transporting and mounting activity saving labor and time with a small amount of applied load on the hand lever it can lift the propeller shaft easily. Since there is only one overhead crane. Since the machine is operated manually it is a major help when electricity is not available.

## 5.2. Stress Distribution

In this study, the ANSYS simulation package was employed as a powerful tool to comprehensively analyze the stress distribution and deformation in the scissor-lifting system. The primary aim was to assess the structural integrity and mechanical performance of the scissor screw type drive shaft lifter machine. The analysis focused on the stress distribution within the system under various operational conditions. The von Mises stress equivalent was chosen as a measure to evaluate the maximum stress in the machine. The utilization of ANSYS software allowed for a detailed and accurate evaluation of these parameters.

The stress distribution analysis within the scissor-lifting system played a pivotal role in evaluating its structural integrity and safety. As shown in Figure 5.2, the maximum recorded stress on the mechanical scissor lifter was measured at 22.88MPa. This numerical value, however, should be

interpreted contextually, especially when compared to the minimum stress recorded, which 16.357MPa was. This comparison reveals that the scissor structure experiences varying levels of stress across its components, with some areas being more heavily loaded than others.

The stress distribution within the scissor mechanism is not uniform but rather concentrated in specific regions, as evident from the simulation results. This information is instrumental in identifying potential weak points or points of high mechanical loading within the scissor structure. These concentrated stress zones may warrant further investigation and possibly necessitate design optimization or reinforcement. Such findings are critical in ensuring that the scissor screw type drive shaft lifter machine operates safely and reliably in practical applications.

In the design and engineering of lifting machinery, understanding the stress distribution is essential. It allows for targeted improvements, such as strengthening specific components or optimizing the design to mitigate areas of high stress concentration. By focusing on these findings, the scissor screw type drive shaft lifter machine can be refined to enhance its performance, extend its lifespan, and ensure its safety under various operating conditions. This iterative process of analysis and design refinement is a fundamental aspect of engineering, ensuring that the final product meets the necessary standards of reliability and efficiency.

The findings of this stress distribution analysis serve as a valuable foundation for discussions related to the structural integrity and safety of the scissor screw type drive shaft lifter machine. It provides engineers and designers with insights into how the machine performs under load, enabling them to make informed decisions regarding material selection, structural modifications, or operational limitations. Furthermore, this data guides efforts in enhancing the machine's efficiency, minimizing the risk of structural failure, and ultimately ensuring its successful implementation in various industrial applications.

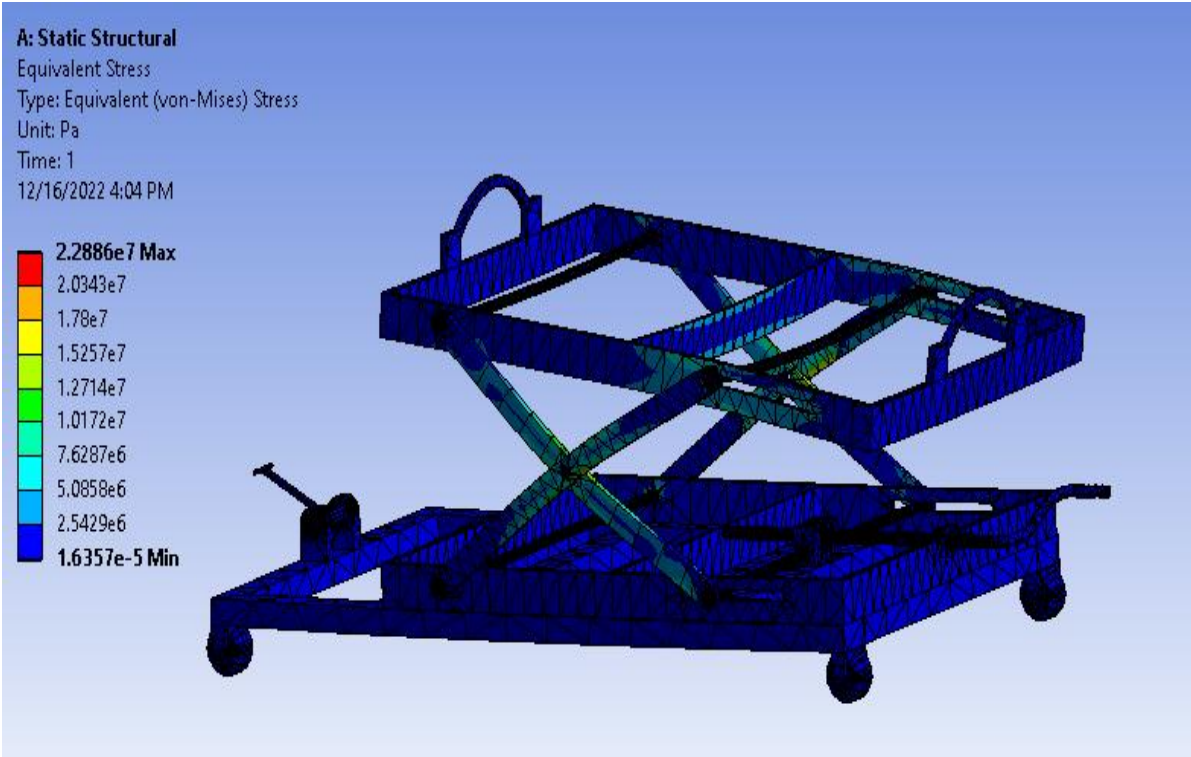


Figure 5.2. Stress distribution in Pa

**Table 5.2. Stress distribution data**

Object Name	<i>Force</i>	<i>Fixed Support</i>
State	Fully Defined	
<b>Scope</b>		
Scoping Method	Geometry Selection	
Geometry	2 Faces	4 Faces
<b>Definition</b>		
Type	Force	Fixed Support
Define By	Vector	
Applied By	Surface Effect	
Magnitude	22.88 N (ramped)	
Direction	Defined	
Suppressed	No	

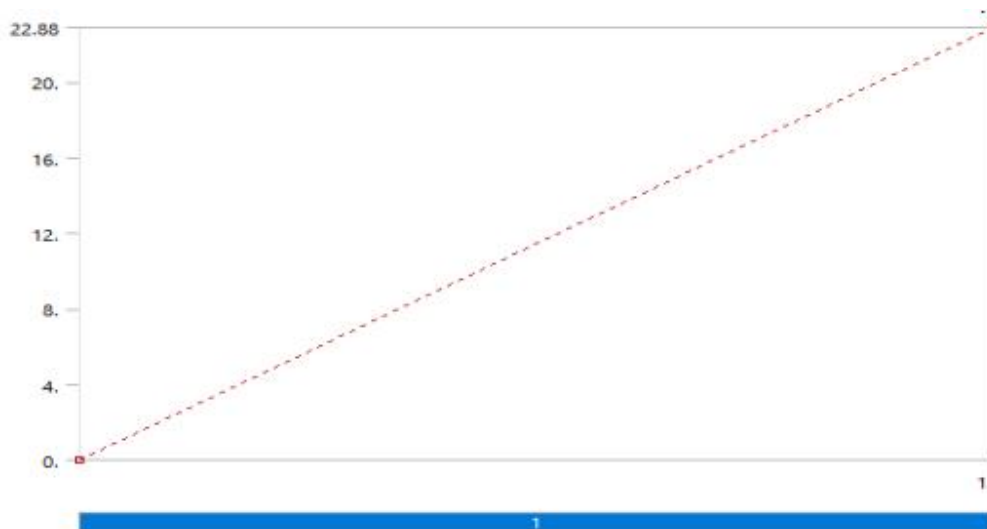


Figure: 5.3. Load vs stress diagram

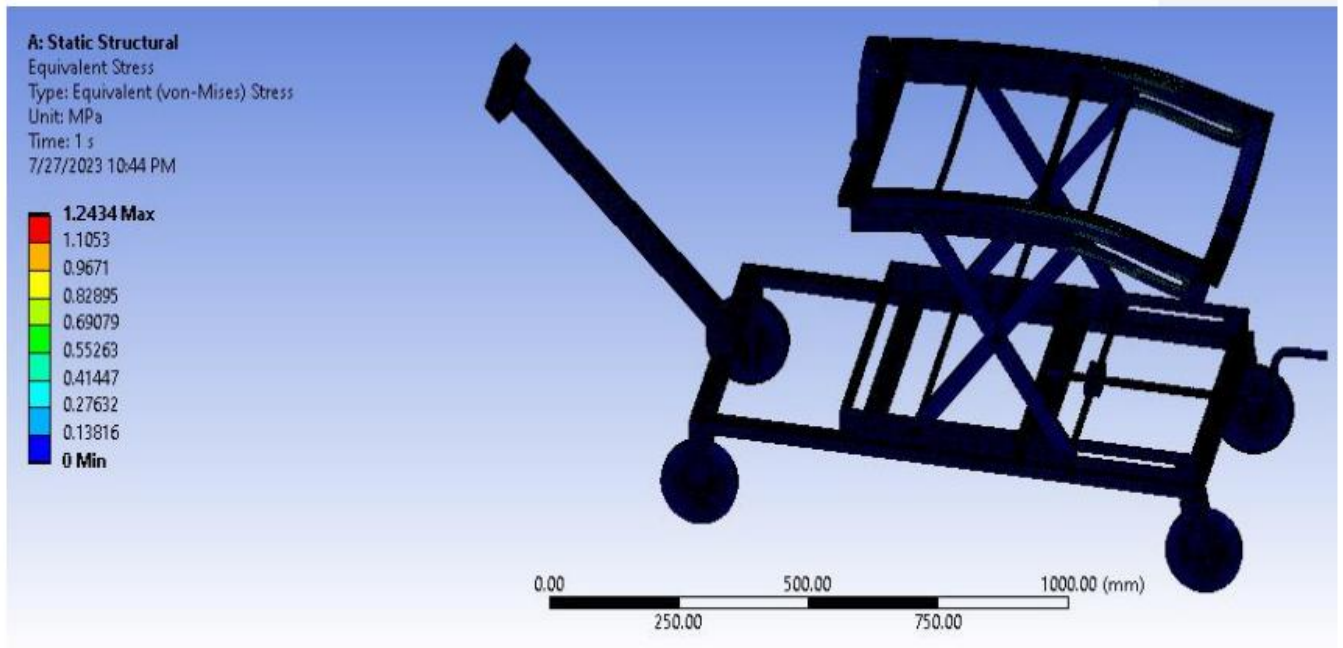


Figure: 5.4. Stress distribution in Mpa

**Table 5.3. Stress distribution data**

Object Name	<i>Force</i>	<i>Fixed Support</i>
State	Fully Defined	
<b>Scope</b>		
Scoping Method	Geometry Selection	
Geometry	2 Faces	4 Faces
<b>Definition</b>		
Type	Force	Fixed Support
Define By	Vector	
Applied By	Surface Effect	
Magnitude	50 N (ramped)	
Direction	Defined	
Suppressed	No	

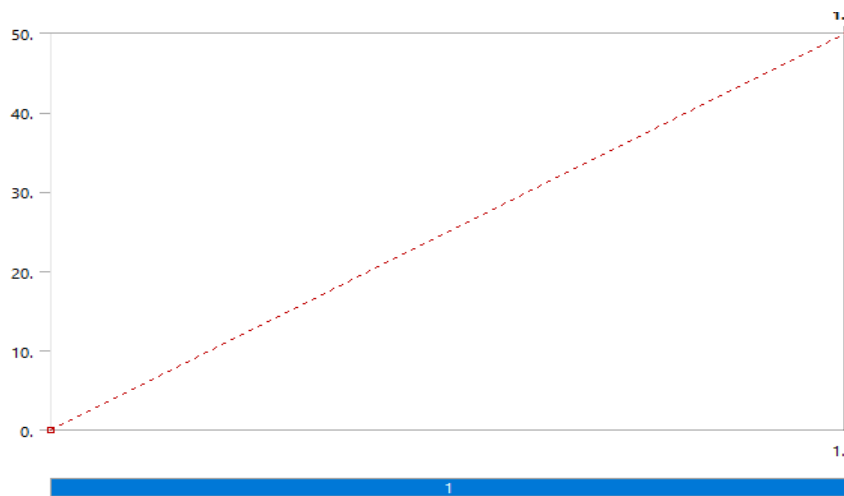


Figure: 5.5. Load vs stress diagram

### **5.3. Structural Deformation**

Deformation, in the context of structural engineering, is a fundamental parameter used to assess how a structural component or element responds to the application of external loads. It is a measure of the extent to which the shape of the structure changes when subjected to these loads. This deformation can manifest as changes in distance or angle, and its visibility is contingent on multiple factors, including the intensity of the applied load, the geometry of the component, and the material properties. Deformation, whether visible to the naked eye or not, plays a crucial role in understanding a structure's behavior under various loading conditions.

In the specific case of the mechanical scissor lifter, a simulation was conducted using the ANSYS package to analyze the deformation it undergoes when subjected to its intended working conditions. The results, as depicted in Figure 5.6, revealed that the total deformation of the scissor lifter exhibited a maximum value of 0.00036mm. This small magnitude of deformation suggests that the structure is relatively stiff and rigid when subjected to the applied loads. Such limited deformation is desirable in many engineering applications as it indicates that the machine retains its shape and structural integrity under typical operating conditions, ensuring stability and functionality. However, it's essential to interpret this result in the context of the machine's specific requirements and safety margins.

The observed deformation value of 0.00036mm raises questions about the adequacy of the structural design, the choice of materials, and the safety margins built into the machine. Further discussion and analysis are warranted to determine if this deformation is within acceptable limits for the machine's intended use. Additionally, consideration should be given to the implications of deformation under various operational scenarios, such as maximum load capacity or dynamic loading. This information is crucial for validating the design's suitability, ensuring the machine's longevity, and making any necessary improvements or optimizations.

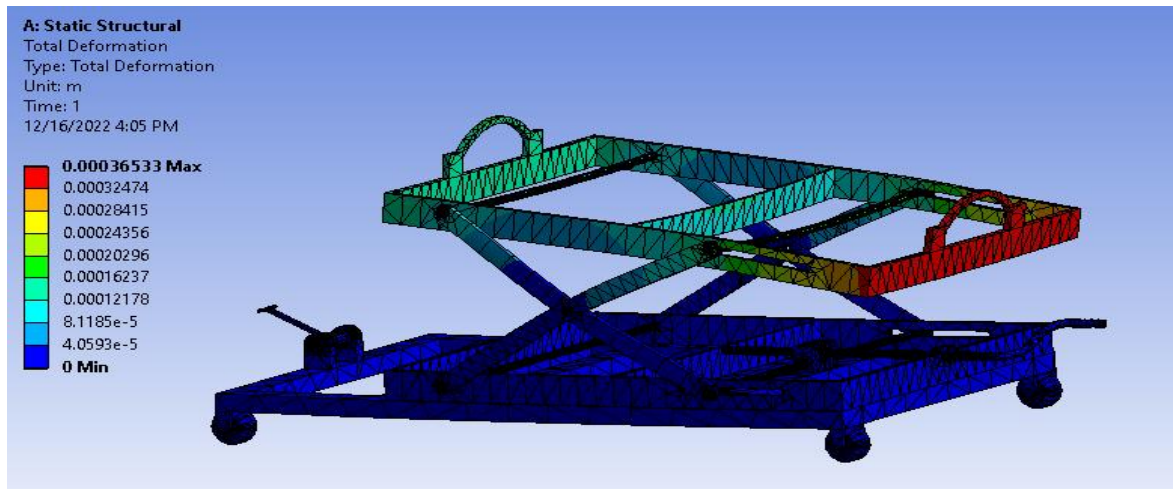


Figure: 5.6. Deformation

Deformation in the table or bed of the scissor mechanism is a critical issue to address, primarily due to the direct loading of the drive shaft and the distribution of its weight across this component. This deformation is a consequence of the mechanical stress and strain experienced by the scissor table during operation. The scissor screw type drive shaft lifter machine is designed to lift heavy loads, and the drive shaft is a significant component with substantial mass. As a result, the stress concentration on the table can lead to deformation.

The deformation of the scissor table can be a significant concern in the machine's performance and longevity. It not only affects the stability and reliability of the lifting process but also poses potential safety risks, especially in applications where precise positioning is crucial. During the simulation of the machine's performance, it is essential to closely analyze and assess the degree of deformation that occurs. This information can be used to refine the design and possibly introduce structural enhancements, such as reinforcing the table or optimizing the materials used. Discussion and analysis of the deformation are essential aspects of the simulation phase, as they provide valuable insights into the machine's behavior under various conditions.

Addressing the deformation in the scissor table can lead to several practical considerations in the design and simulation process. It might involve optimizing the dimensions and materials of the table to better withstand the loading conditions. Furthermore, the simulation results can be used to determine the maximum allowable weight that the machine can lift without causing excessive deformation. This information is crucial for ensuring the machine's safety and efficiency.

## CHAPTER SIX

### 6. CONCLUSION AND RECOMMENDATIONS

#### 6.1. Conclusion

Over the years, scissor mechanisms have played a pivotal role in various industries by providing an effective means of lifting heavy objects beyond the capacity of human effort. The mechanical scissor, a particular type of scissor designed for such tasks, has proven invaluable in numerous work sites, including the Bishoftu Automotive Industry. This study has addressed the fundamental need for a well-designed and safe mechanical scissor, exploring the intricacies of its design and the safety aspects concerning each of its components. It is clear that the design and analysis conducted have resulted in a machine that meets the required safety standards, offers considerable carrying capacity, and exhibits impressive lifting capabilities.

The design of each component of the mechanical scissor is a significant achievement, as it contributes to the overall integrity and reliability of the machine. The fact that each part of the scissor has undergone rigorous design analysis to ensure its safety is a testament to the commitment to creating a dependable and robust lifting system. This meticulous attention to design and safety demonstrates the commitment to not only meeting industry standards but also surpassing them. The impressive performance metrics of the mechanical scissor, with a maximum carrying capacity of 80 kg and a lifting height greater than 750 mm, indicate that the machine is well-suited for a wide range of lifting operations in the Bishoftu Automotive Industry. The observed maximum stress deformation at 22.88 MPa and maximum deflection of  $3.24 * 10^{-1}$  mm provide critical insights into the machine's behavior under load, which is essential for maintaining operational safety and efficiency.

In general, this study has made a valuable contribution to the field of mechanical engineering and the practical needs of the Bishoftu Automotive Industry. The well-designed mechanical scissor not only meets but exceeds the requirements for lifting heavy objects safely. It stands as a testament to the power of meticulous design, thorough analysis, and a commitment to quality. As the industry continues to evolve and expand, this research serves as a foundation for the development of even more advanced and efficient mechanical lifting solutions.

## **6.2. Recommendation**

Design of the lifting mechanism on a mechanical scissor can never be ending process based on different users' basic requirements. Every design can play the vital role of outcome from previous design, as well as a reference to inform the next version. The analysis performed in this study based on some assumptions and restrictions. However, complete structural analysis, understanding behavior of the mechanical scissor is attained considering any possible details. Therefore, the following points are recommended for further future work extensions and elaborations on this research area.

### **Future work**

- Investigate the use of advanced materials and composites to enhance the scissor's strength-to-weight ratio and overall performance.
- Research the potential for incorporating automation and control systems to improve the scissor's precision and efficiency in lifting operations.
- Explore safety mechanisms and fail-safes to minimize risks associated with the operation of the mechanical scissor in industrial settings.
- Investigate ergonomic design considerations to enhance the usability and comfort for operators during prolonged usage.
- Evaluate the economic feasibility of the mechanical scissor, considering both initial costs and long-term operational savings.

Explore the adaptability of the mechanical scissor design for use in various industries and applications, thereby expanding its versatility and market potential.

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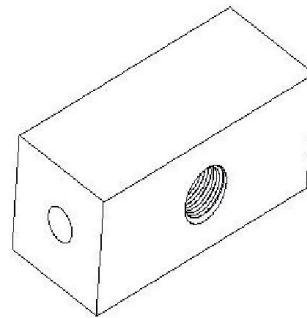
## APPENDIX

Part drawing of components

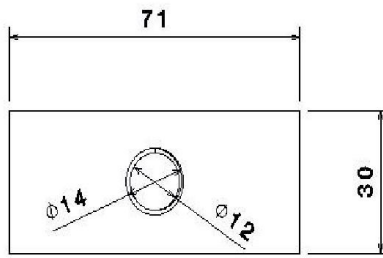
All dimensions are in millimeter (mm)



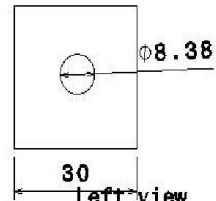
Bottom view  
Scale: 1:1



Isometric view  
Scale: 1:1

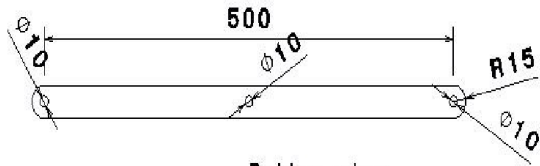


Front view  
Scale: 1:1

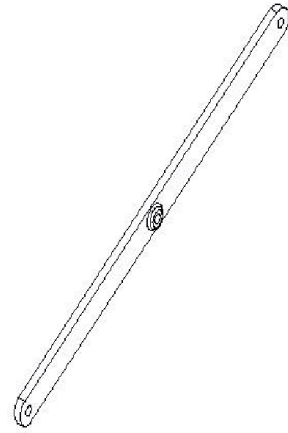


Left view  
Scale: 1:1

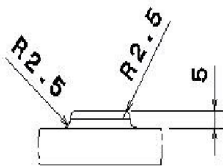
Figure A1, Nut of screw



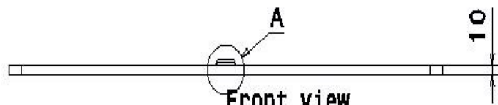
Bottom view  
Scale: 1:5



Isometric view  
Scale: 1:5



Detail A  
Scale: 2:3



Front view  
Scale: 1:5



Left view  
Scale: 1:5

Figure A2, Link

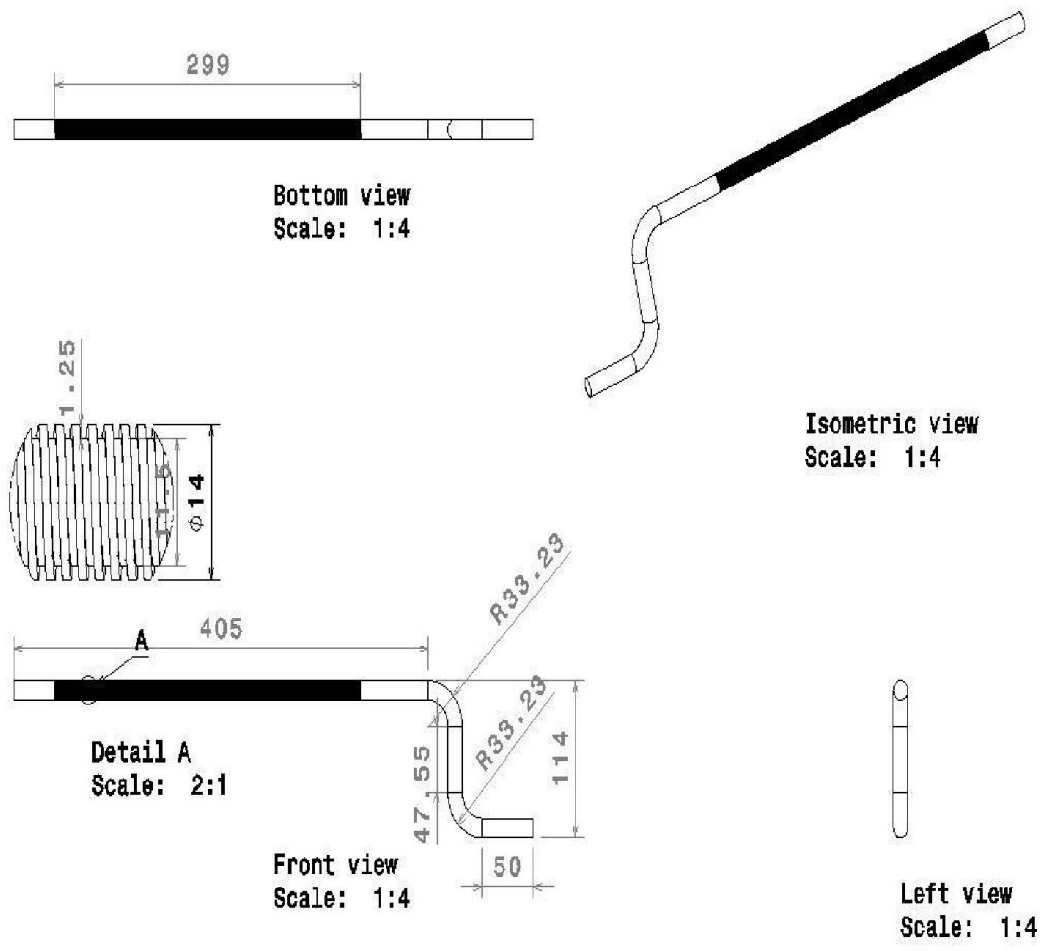


Figure A3, Screw with lever

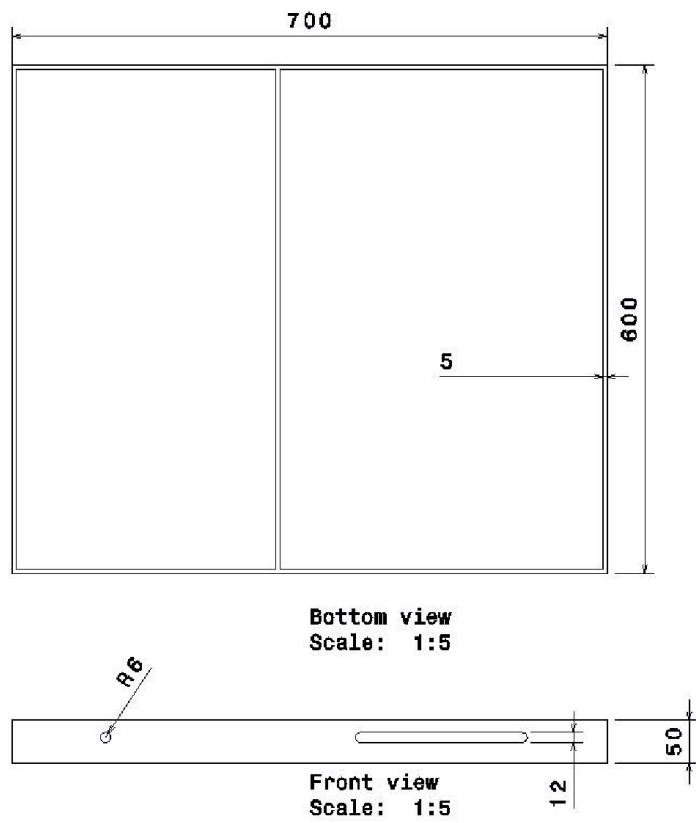
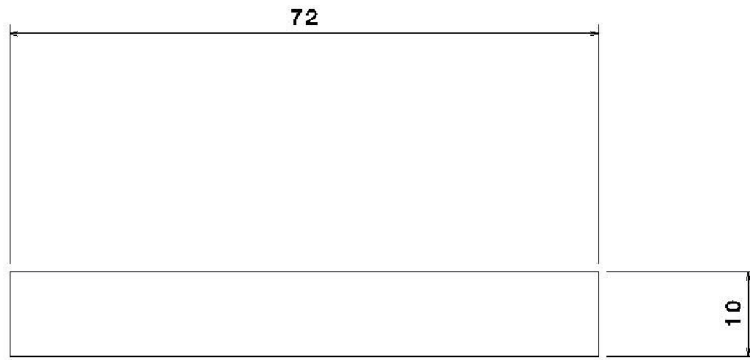


Figure AP4, Bottom bracket



Front view  
Scale: 2:1

Figure A5, Handle pin

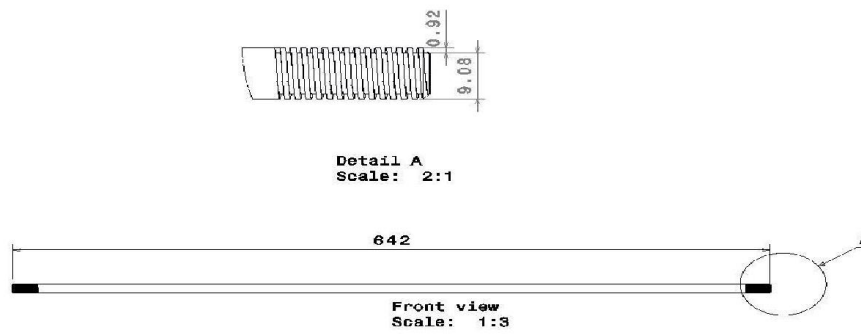


Figure A6, Pin between two links

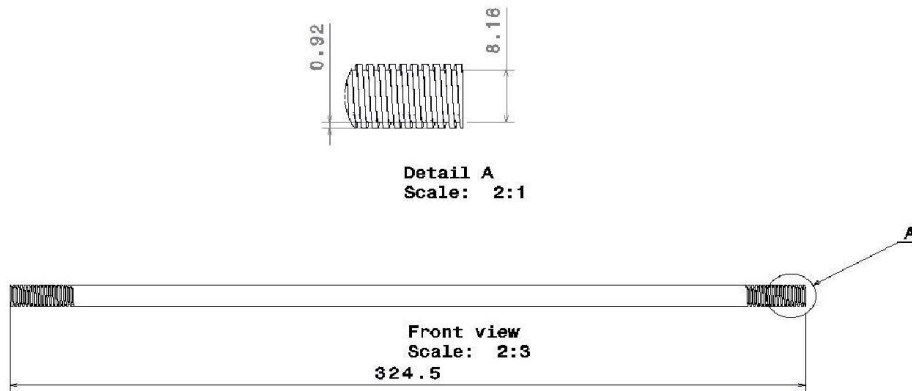


Figure A7, Slider pin connect link to screw nut

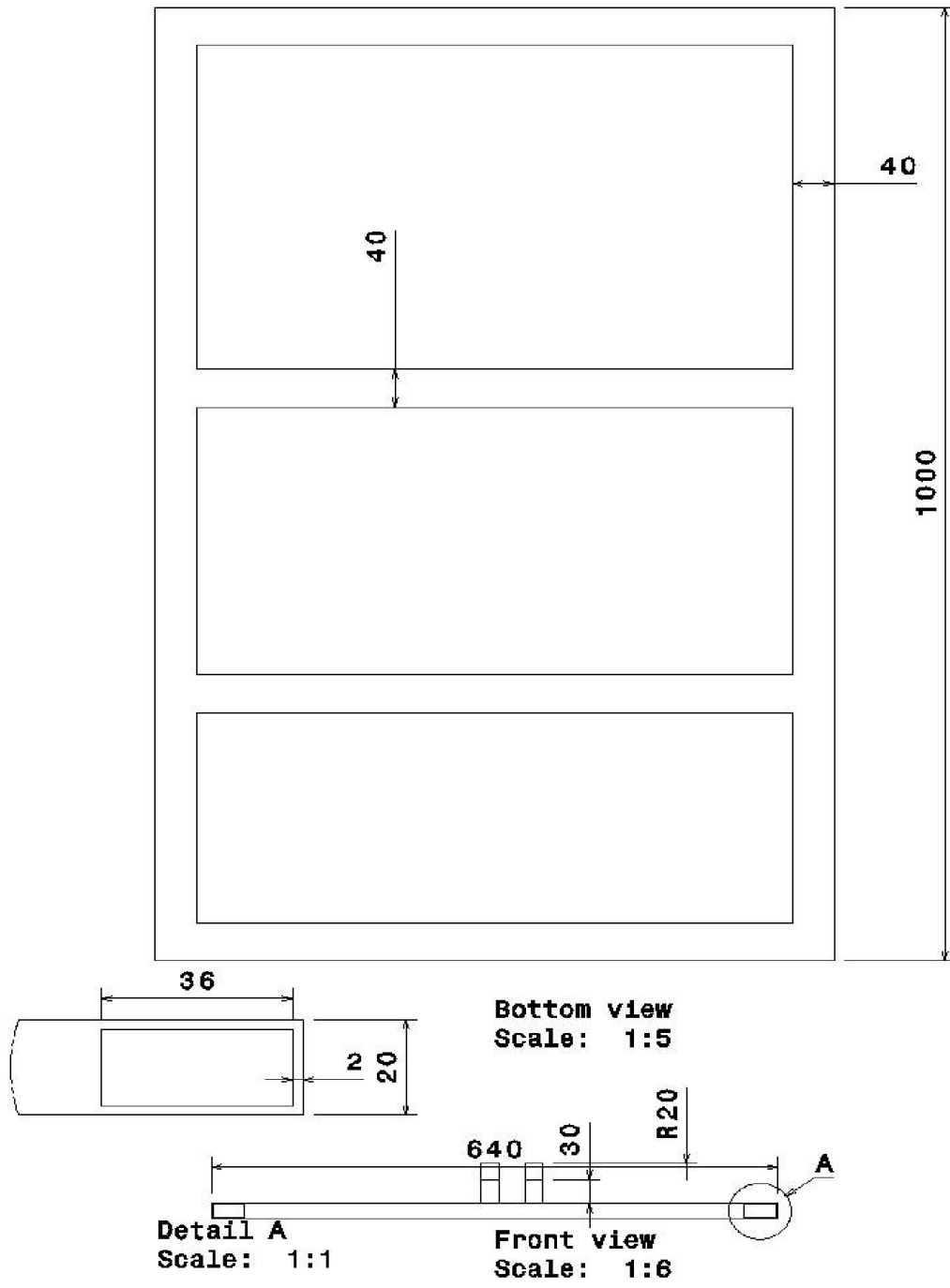


Figure A8, Base

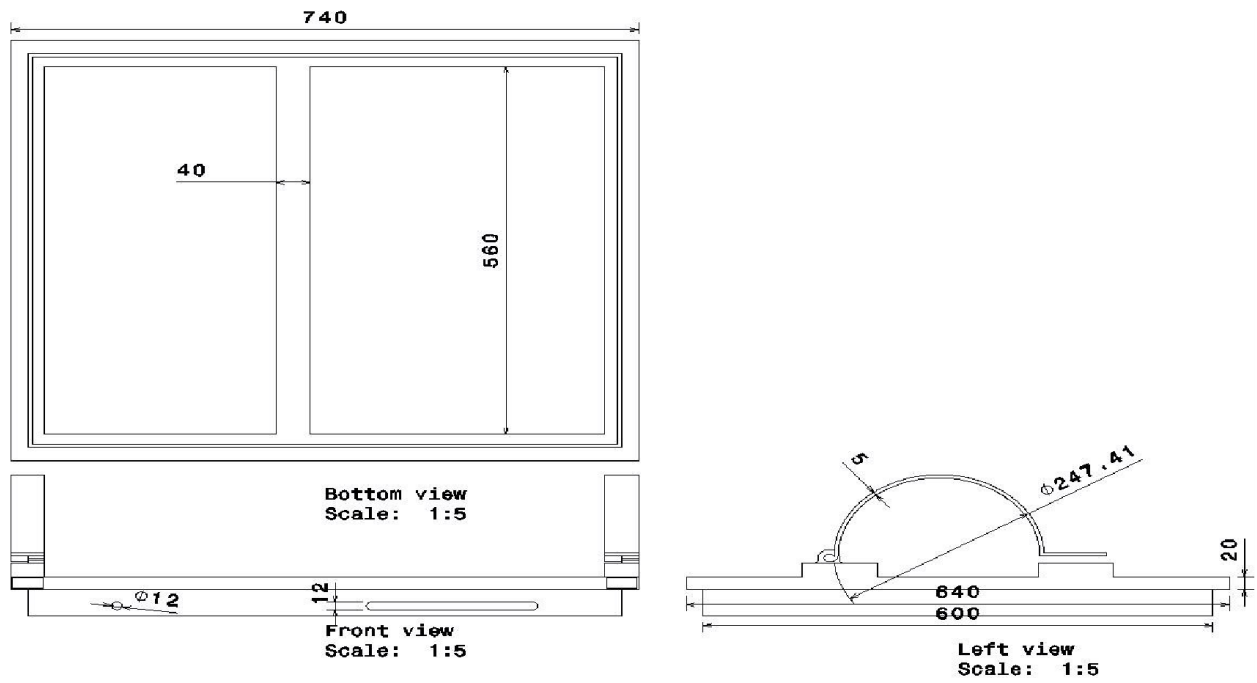


Figure A9, Top frame assembled with top bracket and clamp

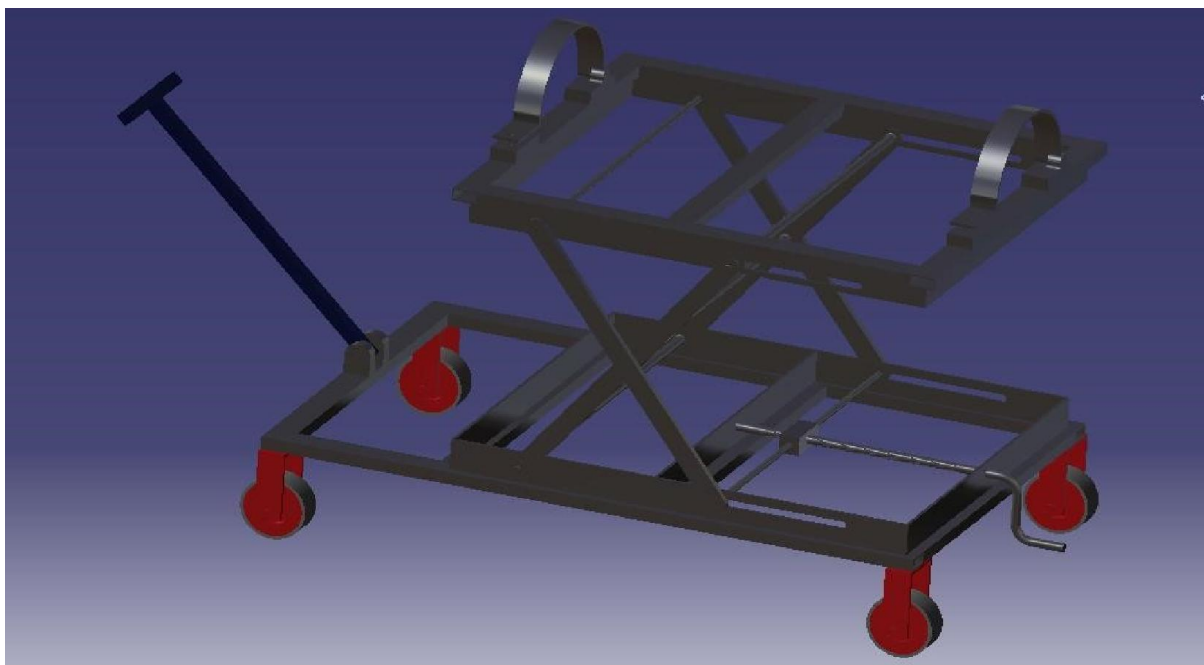


Figure A10, Assembled 3d, maximum lifting distance

**TABLE A1. Structural Steel > S-N Curve**

Alternating Stress MPa	Cycles	Mean Stress MPa
3999	10	0
2827	20	0
1896	50	0
1413	100	0
1069	200	0
441	2000	0
262	10000	0
214	20000	0
138	1.e+005	0
114	2.e+005	0
86.2	1.e+006	0

**TABLEA2. Structural Steel > Strain-Life Parameters**

Strength Coefficient MPa	Strength Exponent	Ductility Coefficient	Ductility Exponent	Cyclic Strength Coefficient MPa	Cyclic Strain Hardening Exponent
920	-0.106	0.213	-0.47	1000	0.2

**TABLEA3. Structural Steel > Isotropic Elasticity**

Young's Modulus MPa	Poisson's Ratio	Bulk Modulus MPa	Shear Modulus MPa	Temperature C
2.e+005	0.3	1.6667e+005	76923	

**TABLE A4. Structural Steel > Isotropic Relative Permeability**

Relative Permeability
10000



11	Wheel	4
12	Handle	1
13	clamp	2

**Table A6. Summary result of the designed components**

No.	Components	Dimensions	Manually calculated results	Assembly Method	Manufacturing method
1.	Screw (square threaded)	L=190mm d <sub>c</sub> =12 mm d <sub>o</sub> =14 mm p = 2 mm	$\tau_{max}=18.38\text{N/mm}^2$ $\sigma_{max}=104.2\text{N/mm}^2$	Passes through the teeth of the nut and bearings	Machining on lathe
2.	Top frame	Width = 600 mm Length = 700 mm		Pined on top of the link	By welding 20x40 hollow rectangular cross-section beams together
3.	Pin	D = 10 mm		Fitted between the link & bracket	
4.	Screw nut	H = 30mm n = 5 teeth b = 71 mm	$\tau = 13.36 \text{ N/mm}^2$	Mounted on the screw and pins are mounted to the side of the nut	Machining
5.	Link	t = 10mm w = 30mm L = 500mm	$\sigma = 73.48 \text{ N/mm}^2$	Connected to the top and bottom bracket by pin	molding

6.	Base frame	L = 1000 mm t = 20 mm w =600mm		The wheel and bracket are welded on it	By welding 20 x 40 mm cross-section hollow rectangular beams together
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**Bill of Materials (BOM):**

The BOM is a comprehensive list that itemizes all the components, materials, and quantities required to build the lifting machine. It provides a detailed breakdown of each part, including part numbers, descriptions, dimensions, and quantities. The BOM facilitates accurate procurement and helps ensure that all necessary components are accounted for during the manufacturing process.

**Table A7. Cost of the scissor lift**

No.	Parts	Material	Qty.	(Birr)
1.	Link	Structural Steel	4	5600
2.	Screw	Steel C-30	1	4900
3.	Screw Nut	Grey Cast Iron	1	7000
4.	pin	Structural Steel	5	4000
5.	Screw handle	Structural Steel	1	900
6.	Top frame	Structural Steel	1	9000
7.	Bottom bracket	Structural Steel	1	5000
8.	Nut	Stainless Steel	10	3100
9.	Base frame	Structural Steel	1	8900
10.	Handle pin	Structural Steel	1	1000
11.	Wheel	Rubber Filled	4	8100
12.	Handle	Structural Steel	1	1200
13.	clamp	Aluminium	2	1300
<b>Total</b>			33	60,000

- Machining Cost (Drilling, Welding, Grinding, Gas Cutting Cost, Punching) = 2,500birr
- Labour cost = 3,500

If to purchase lift machineries (required specification) from the wholesale market then doing assembly process, price will come about 64,300birr. While assembly process supposed to machining process involved, it will take some additional amount for this process. So total cost  
Total cost = Material Cost + Machining Cost+ Labour cost = 66,000