Design and Analysis of Forklift for Confined warehouses

A Final Year Project Report

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ABSTRACT

The process of movement of materials (raw materials, finished or semi-finished materials) also known as material handling is of much importance in warehousing and different stages of production. The Forklift is widely used in these material handling processes by the industries as it is compact and is an alternative to manual lifting. Industries having warehouses with confined spaces cannot use most of the commercially available forklifts because of their size and operating radius. Design and development of smart, low cost, small operating radius forklifts is an area of constant interest. This project aims to locally design and develop a forklift that can meet the needs of local industries and warehouses who deal with small to medium sized packages.

ACKNOWLEDGMENTS

We are hugely thankful to our supervisor Dr Aamir Mubashar for his help during the process of making this project and all of its intricacies. We would also like to thank many of our fellow students who have helped us in doing a lot of research. Along the process, we came to know about many new things and found new and exciting interconnections among various fields of engineering with our own.

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ORIGINALITY REPORT

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PRIMARY SOURCES

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CHAPTER 1: INTRODUCTION

1.1. Motivation and Problem Statement:

The motivation for selecting our project was to provide an economical and compact forklift to address the issues of stability, operation, and cost having a mid-range capacity while targeting small industries.

1.1.1. Stability:

Firstly, there is an issue of stability encountered in the forklifts. There are chances of tipping over, as the center of gravity moves out of the stability triangle. This issue is encountered during usage of the forklift especially in a confined place and tall storage racks.

1.1.2. Ease of Operation:

Another fact is that for the operation of forklift, a skilled and trained person is required, it is difficult for a warehouse worker to operate the forklift. Therefore, another focal point of this design is the ease of operation.

1.1.3. Cost:

Cost of forklifts is currently high due to their import. Local design and development will pave way to cheaper manufacturing.

1.1.4. Mid-range Capacity:

This project is aimed at developing forklift for a mid-range load capacity of under 100kg. Such loads are typical of online trading stores such as Amazon, Alibaba, etc.

1.2. Aim and Objectives:

The aim of this project is to design and develop a smart, medium capacity, small operating radius and cost efficient forklift.

The project has following objectives:

1. Develop specifications for the forklift design based on industrial needs and review of available systems.
2. Design a feasible forklift based on specifications.
3. Analyze the design and perform iterations to finalize the design.
4. Develop prototype using 3D software.

CHAPTER 2: LITERATURE REVIEW

2.1. Introduction:

Forklift is a powered industrial truck that is used to lift and move materials over a short distance. Forklifts eliminate the need for many people to handle loading and unloading operations and the stacking of the goods on the racks. They fall under the category of lifting and transport machinery.

2.2. Basic Systems of Forklift:

Forklift consists of three systems, which are loading system, power or control system, and the driving system as shown in Figure 1.

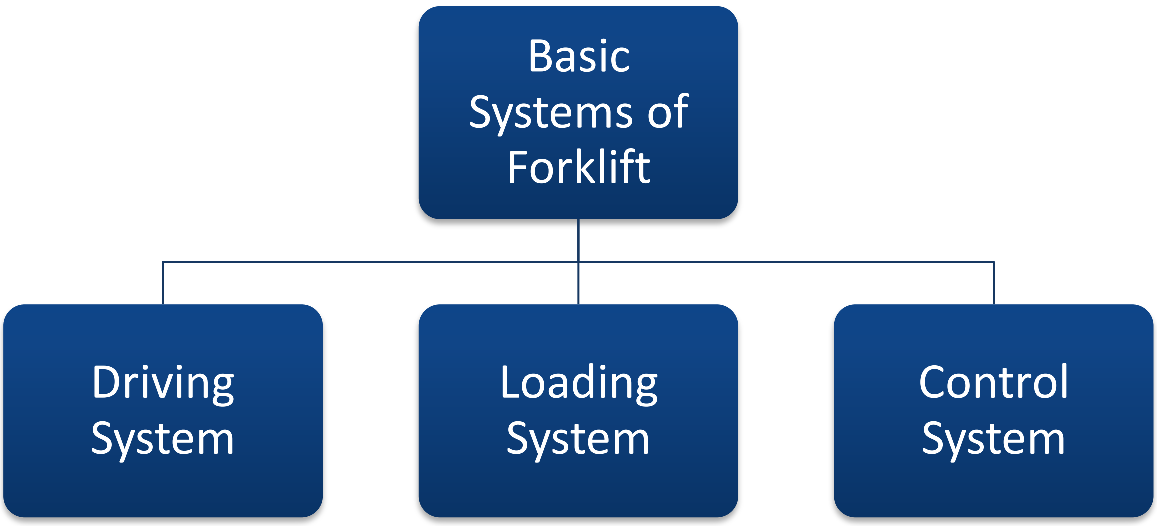


Figure 1: Basic Systems of Forklift

Driving System:

It is the system of the forklift that allows the movement of the forklift. One of the significant forklift subsystems is the steering system that rotates the driving wheel in the desired direction based on the steering input.

Loading System:

Loading system is comprised of the forks and the lifting chain to pick up and lift the material from a location and to move it to the other desired location.

Power (Control) System:

This system provides all the control and power to the other two systems. It is comprised of battery, motors, and sensors used for different purposes. Through battery, it provides power to the motors to rotate the tires to move the forklift body and to lift the materials by rotation of the lifting chain. Proper control of traction and lifting motors is essential for successful operation of an electric forklift.

2.3. Types of Forklifts

The main features that differentiate the forklifts are the following [3],

- Power Source
- Configuration

2.3.1 Categorization according to Power Sources

Batteries are used in the electric-forklifts, and there is a need to charge these batteries. For indoor or confine warehouses, such type of forklifts are used because they are environmentally friendly and do not exhaust the gases. One of the main advantages of this type of Forklift is that its operating cost is less than the other types. Based on lifting power, they can lift about 10000 to 15000 lbs. These forklifts have low acceleration, which makes them more stable, but one of the disadvantages is that their batteries take about sixteen hours for charging.

In the case of gas-powered forklifts, the fuel used is either natural gas or diesel. The capital investment in this forklift is less than the electric ones, but their maintenance cost is very high. These forklifts are mostly used for outdoor purposes because they can lift heavier weights as compared to the electric ones and their weight- lifting capacity lies from 15k to 35k pounds. This makes it more useful for heavy industries. Figure 2 shows a general comparison between electric and gas-powered forklifts, enlisting their advantages and disadvantages.

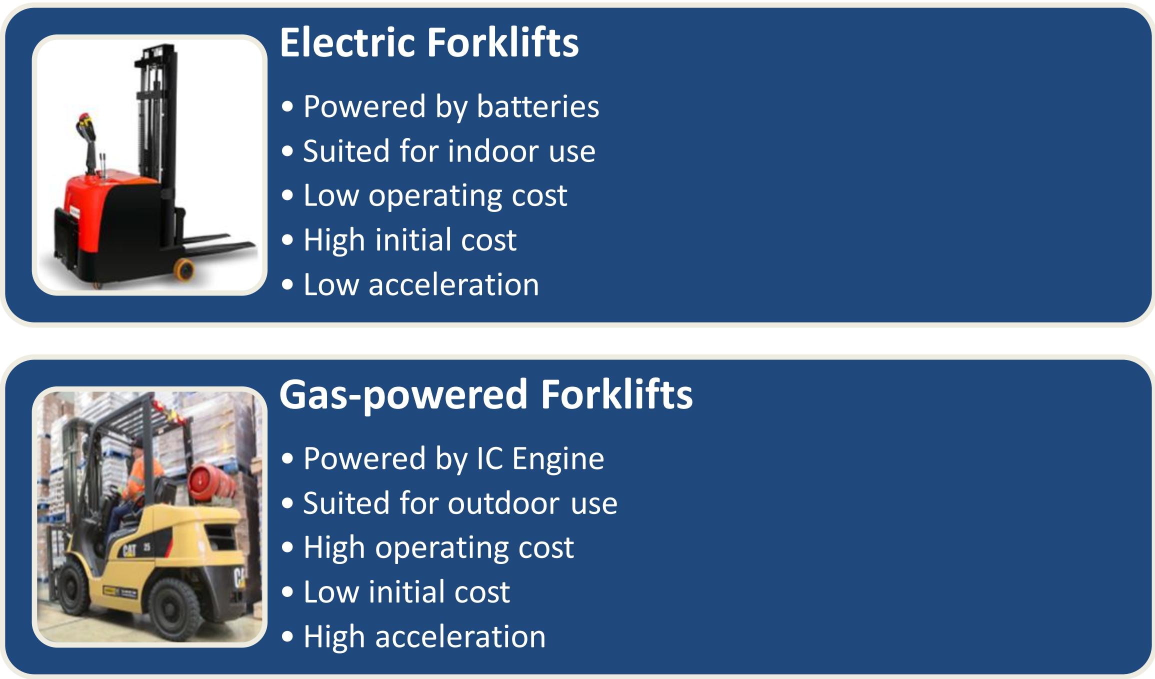


Figure 2: Comparison between Electric and Gas-powered Forklifts

2.3.2 Categorization according to Configuration

An overview of forklift types based on configuration is shown in Figure 3.

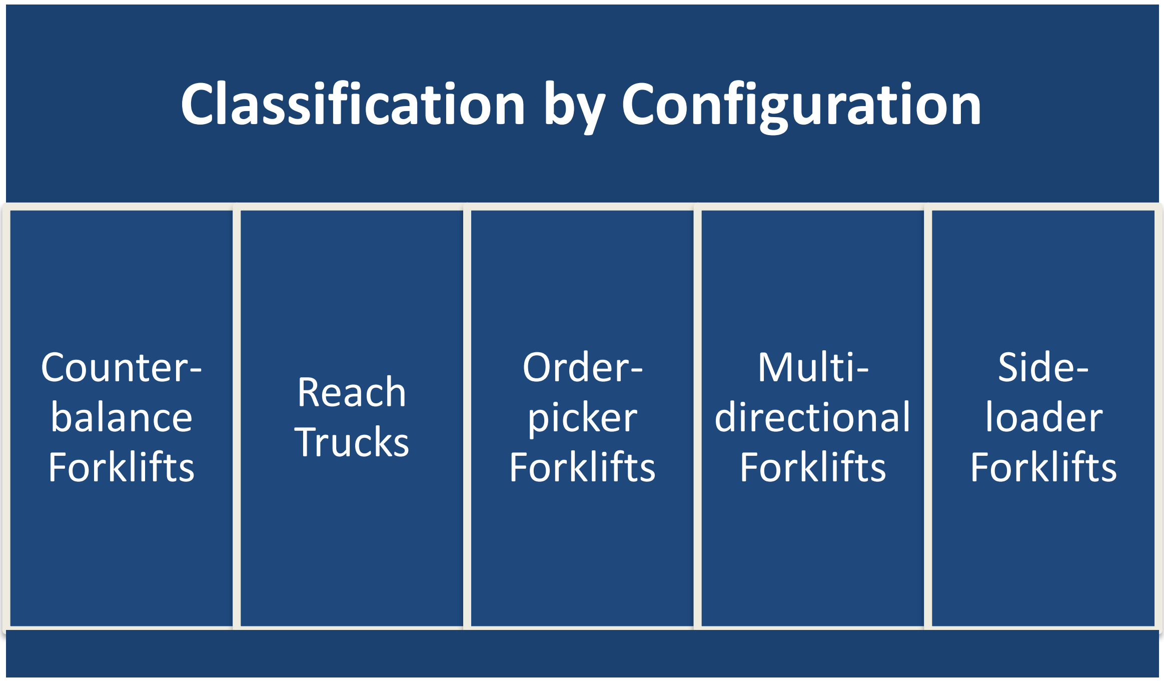


Figure 3: Forklift Classification according to Configuration

Counter-balance Forklifts:

This forklift type is very common. In these forklifts, a weight is placed at the rear end of the truck to avoid tipping over of forklift body, as shown in Figure 4.

Figure 4: Counter Balance Forklift

These forklifts come with both power sources, which means that they can either be electrically powered or can run by the gas or diesel. In electric-type, we can use the batteries as the counter-weight, and they can provide power. In some cases, there is a need for some extra weight for counter-balance. In these forklifts, there is no requirement of reach facility, and the fork lends itself to straightforward operations [6].

Counter-balance forklifts usually come in the form of two variants, three wheel as well as four-wheel machines. In case of three wheel forklifts, the main advantage is the maximum maneuverability making them useful for confined spaces [7], as can be seen in Figure 5.

Figure 5: 3-Wheel Counter Balance Forklift

Reach Truck Forklifts:

Based on design and layout, there is another forklift called reach truck. They are mostly used in the warehouses. They are well known for their maneuverability and provide the maximum lift height. They use the stabilizing legs for the extra stability purpose, and they have more reach capacity as compared to the other forklifts. In this design, batteries and stabilizing legs are enough for counterbalance. They are not suitable for outdoor working because of the uneven surfaces present outside. Nevertheless, it is beneficial for indoor applications [6]. An example of such a forklift is shown in Figure 6.

Figure 6: Reach Truck Forklift

In the **order picker forklift,** one of the main advantages is that operator can be lifted up with the picker. The use of this forklift is to carry heavyweight in small spaces as well as enough space for operator. Figure 7 represents the order picker forklift.

Figure 7: Order Picker Forklift

Multidirectional (four-way) Forklifts are very helpful in warehouses as they move in all directions, which help to reduce the time and as a result increases the productivity of the warehouse. Figure 8 shows a multidirectional forklift.

Figure 8: *Multi-Directional Forklift*

In Side-loaders, there is a stable structure/platform for lengthy loads. They are used to transport the long objects from the as seen in Figure 9. These forklifts can take large objects in the direction of travel.

Figure 9: Side Loader Forklift

Narrow aisle forklifts are very useful in confined warehouses since they have a compact structure and can operate efficiently in areas where space is limited.

These forklifts have the ability to rotate their forks without rotating the entire body. They can also lift their cab, which can increase the visibility of the driver and avoid accidents. Such a forklift is shown in Figure 10.

Figure 10: Narrow aisle Forklift

2.4. Stability:

Stability of the forklift is further classified into the following types.

- Lateral Stability
- Longitudinal Stability

Lateral stability means to ensure that the forklift does not tip sideways whereas longitudinal stability means that the forklift does not tip forward during the loading process.

Stability Triangle:

Vertices of the stability triangle are the center points of all three wheels. Combined center of gravity must lie within the stability triangle (safety zone) to ensure stability of the forklift.

As the forklift is loaded, the combined center of gravity is shifted forwards. Stability triangle and shifted CG are shown in figure 11.

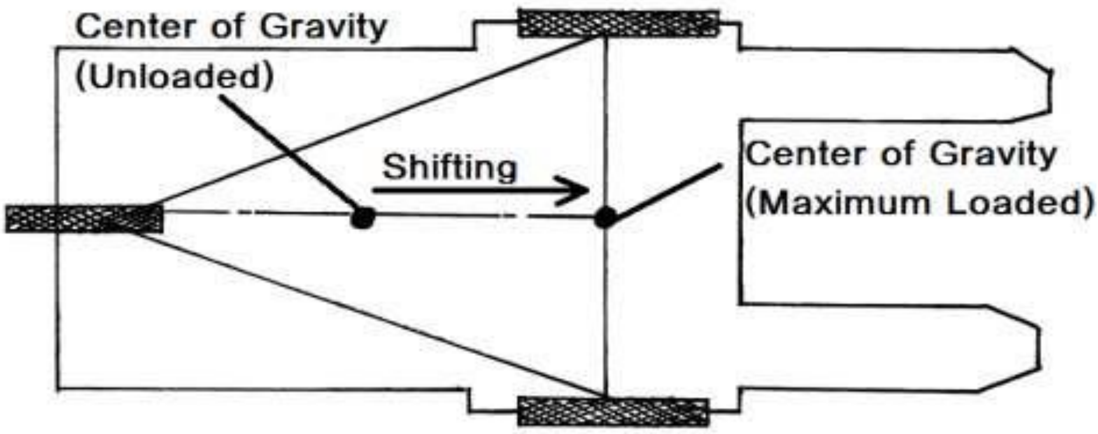


Figure 11: Stability triangle and shifting of C.G

Example of vertical shifting of CG is shown in Figure 12.

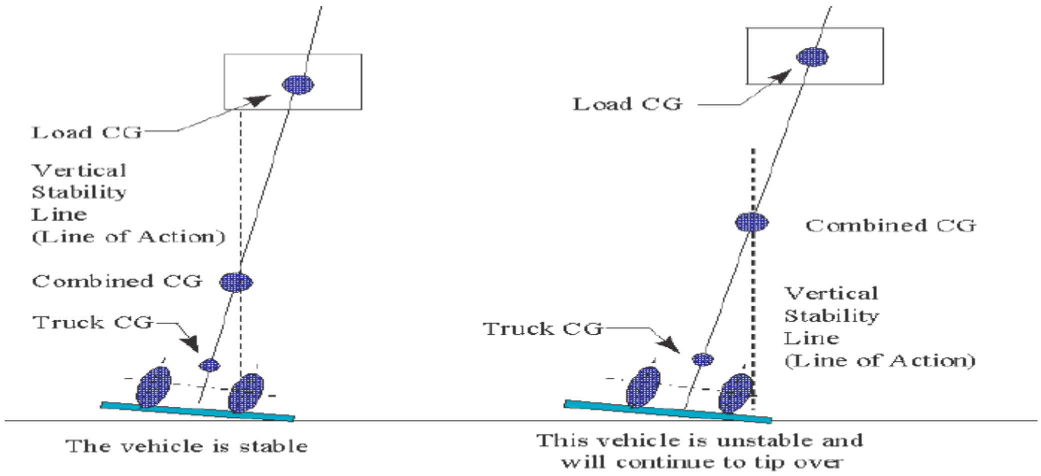


Figure 12: Vertical shifting of C.G

2.5. Benefits and Drawbacks of Existing Design

In the confined warehouses, the main issue is to deal with the restricted space, because most of the space is used for storage purposes. For this purpose, narrow aisle forklifts are used because they can quickly move in the narrow aisles. However, there are still issues linked with this type of forklift including visibility issues, cost and operating limitations. Maintenance of these forklifts require trained personal.

Many of forklifts used in the industry are counterweighting forklifts. They consist of two forks, which are present at the front and used to elevate the load. These forks have some distance between them, which can be adjusted according to the load size. Another issue of a confined warehouse is how to use the vertical space. There are some forklifts available that can be used for this purpose like reach truck, but they are costly.

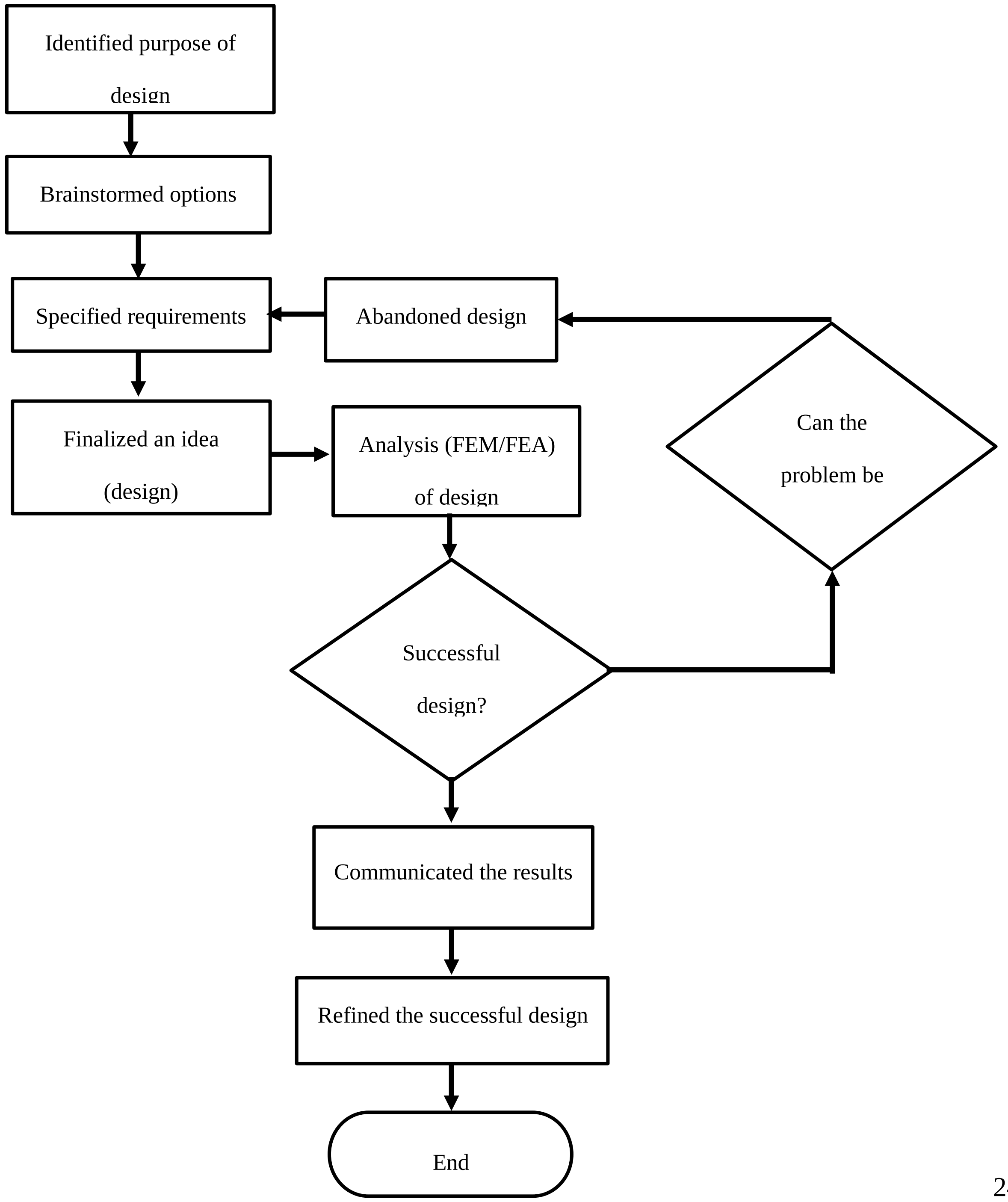
2.5. Summary

From the research, we conclude that there is a need of forklift that is less expensive, usable indoors and can be easily maintained and operated by the workers.

CHAPTER 3: METHODOLOGY AND DESIGN

3.1. Design Synthesis

To come up with the design of the forklift, a methodical and systematic design synthesis approach was adopted which is explained below with the help of a flow chart.



3.2. Design Parameters

After brainstorming different design ideas, each design was evaluated against a matrix of key design parameters, which resulted in the final design according to the requirements. The critical design parameters are shown in Figure 13.

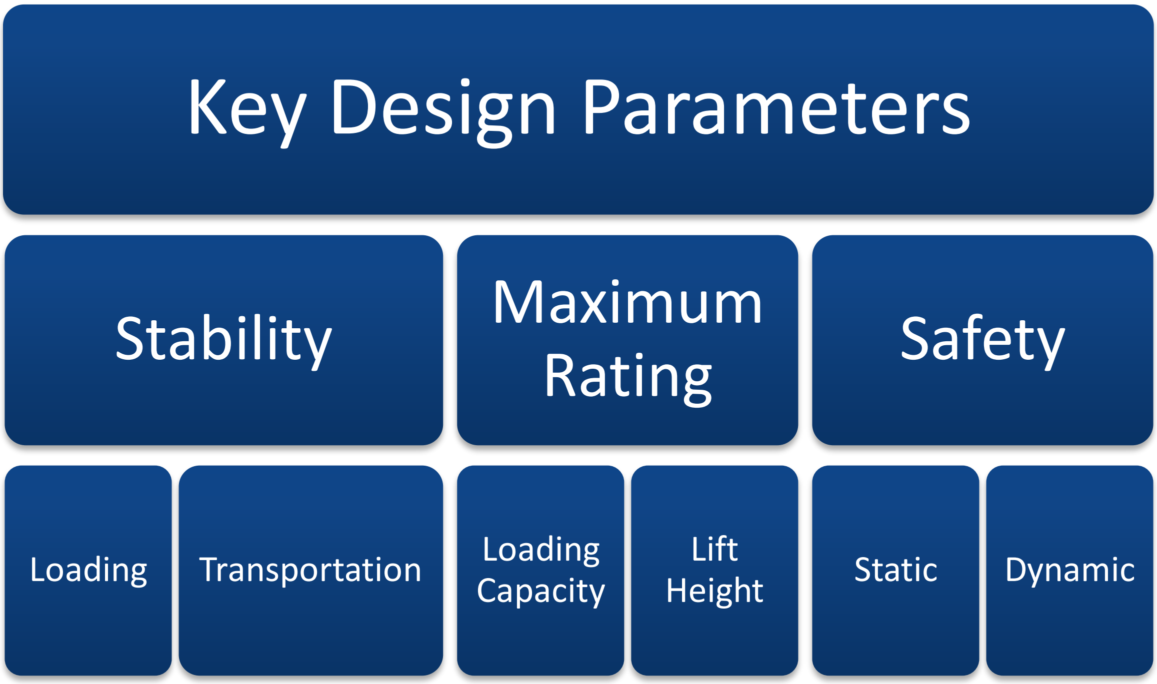


Figure 13: Key Design Parameters

Maximum rating is considered to be the top performance evaluation parameter for any forklift. It sets a limit on the maximum load and lift height a forklift can achieve. Other key design parameters include safety and stability, because without ensuring safety and stability a forklift cannot operate. If compromised, the life of the person operating the forklift can be in danger.

They are several other parameters to meet the requirements and demands of modern warehouses. These parameters are depicted in Figure 14.

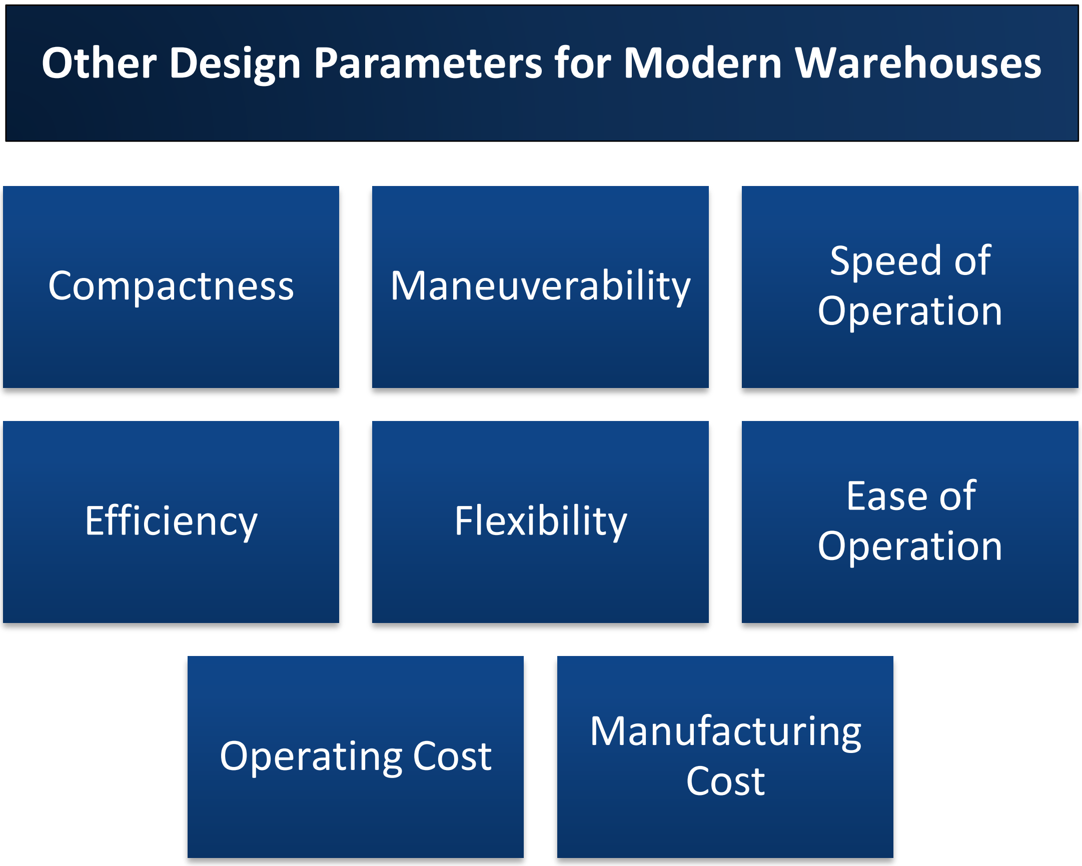


Figure 14: Other Design Parameters

3.3. Design Decisions

In this section, the key design decisions and the underlying requirements of modern warehouses which governed those decisions are discussed. Next section discusses the modern warehouse demands.

3.3.1. Modern Warehouse Demands

In modern warehouses, space is of paramount importance. More space is allocated for storage, and hence less space is available for the aisles as seen in Figure 15. As a result, there was a need for the development of narrow aisle forklifts that can operate with minimum aisle space. Therefore, while designing the forklift, the design should be highly compact and maneuverable so that it can take turns around sharp corners with very less turning room.

Figure 15: Modern Warehouse with minimum aisle space

The E-Commerce world moves at a swift and rapid pace. Therefore, there is a requirement to design a forklift through which we can undergo the loading/unloading and transportation of goods in minimal time.

Another important consideration is that in modern warehouses, the labor force is shrinking day by day. Therefore, there is a need to design a forklift with the least possible requirement of skilled labor. All these requirements are summarized in Figure 16.

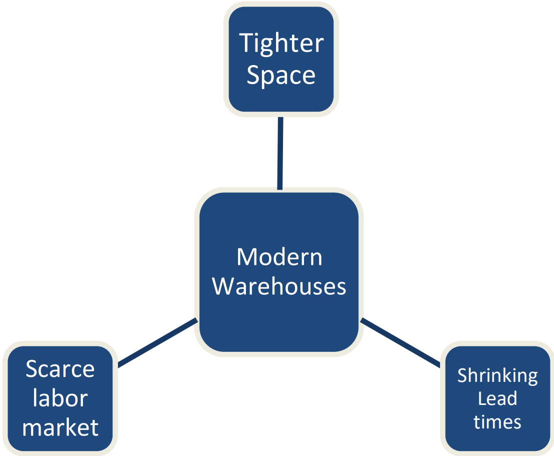


Figure 16: Modern Warehouse trends

3.3.2. Conceptual Design

The proposed design of forklift is an electric-powered forklift with two front pneumatic drive wheels and a solid-state rear wheel (swivel caster). The forklift will work with three DC motors. Two identical motors will be used to drive the front wheels and one stronger motor will be used for lifting purpose.

The forklift will consist of a vehicle body frame designed with a platform consisting of two motorized wheel mounts. The battery will be placed on the platform leaving enough space for a person to stand on and operate the forklift. The forklift will be steered with the help of a perpendicular handle. The lift mechanism will comprise of a large rotating chain mechanism to which the forks will be attached and will be powered by the lifting motor. To ensure the stability of the forklift, a locking mechanism will also be incorporated. It will be used to lock the forks carrying a load at a height slightly below the maximum limit.

The major systems of the proposed system design are shown in Figure 17.

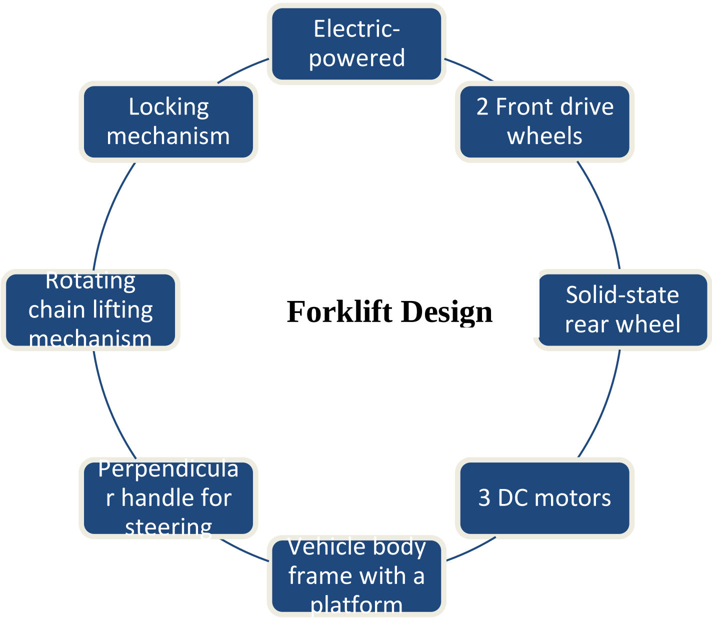


Figure 17: Design features of our forklift

3.3.3. Selection of Design Parameters

1. Keeping in mind that the forklift is to operate indoors; An electric-powered forklift design has chosen for the following reasons:
   - Electric-powered forklifts do not produce excessive noise and hence are suitable for the indoor quiet working environment.
   - They do not release any exhaust gases and hence are viable for closed working conditions.
   - They do not require refueling and thus have a lower operating cost than gas-powered ones.
   - The battery within an electric forklift negates the need for any counterbalance weight.

2. Two pneumatic front drive wheels ensure smooth motion of the forklift during transportation.
3. The single solid-state rear wheel gives maximum maneuverability. It allows the forklift to take turns across sharp corners with very less turning room, hence perfect for narrow aisles with limited space.
4. For a faster and more rapid time response during lifting, a sizeable rotating chain mechanism instead of a hydraulic cylinder system was chosen.
5. The locking mechanism ensures the stability of the forklift by locking the forks carrying a load at a height slightly below the maximum limit, thereby, lowering the combined center of gravity (CG).
6. Vehicle body frame with a perpendicular handle results in a compact design with maximum flexibility and efficiency.

3.4. Design Criteria

3.4.1. Main Components

The main components used in the forklift are given below.

- Material
- Carriage
- Forks
- Lifting Chain
- Shaft
- Motor
- Bearing

- Power Source
- Tires
- Steering

The details about these components are discussed below:

Material:

Material properties dictate if a material can endure different stresses or not. Based on commonly used materials in existing forklifts, a comparison between the material properties was carried out and is given below;

Parameters	Aluminium	Mild Steel	Stainless Steel
Strength	High	High	High
Ductility	High	High	High
Machinable	High	High	Medium
Malleable	Medium	High	Medium
Weight	Low	Medium	High
Cost	High	Low	High

Mild steel was selected because of its high strength, ductility and low cost and easy availability.

Forks:

Stainless steel forks with the following dimensions are used.

Figure 18: Fork

Length = 0.4m, Width = 0.1m, Thickness = 0.015m

Lifting Chain:

Lifting chain mechanism is used to lift and lower materials. Current design uses roller chain drive mechanism for lifting. The specifications of the lifting chain are given in Figure 19.

Chain diameter	Pitch	Outside width	Weight
• $d = 10\ mm$	• $p = 30\ mm$	• $b = 37\ mm$	• $w = 2.2\ \frac{kg}{m}$

Figure 19: Lifting chain specifications

Bearing:

Bearing is a mechanical component that is used for the smooth transmission of the movement between the components. SKF 6005 bearing with the specifications given in the Figure 20 was selected.

Dynamic load rating	Static load rating	Limiting speed
• $C_r = 11.9\ k\mathrm{N}$	• $C_{or} = 6.55\ k\mathrm{N}$	• $N_{max} = 9500\ rpm$

Figure 20: Bearing Specifications

Shaft:

Stainless steel shaft of 25mm diameter was used in the forklift.

Motor:

Two traction motors and one lifting motor was used. Motor specifications are included in the sections to come.

Carriage:

It is the part in the forklift to which forks are attached and it can move up and down through the mast rails. The carriage used in the design are given in Figure 21.

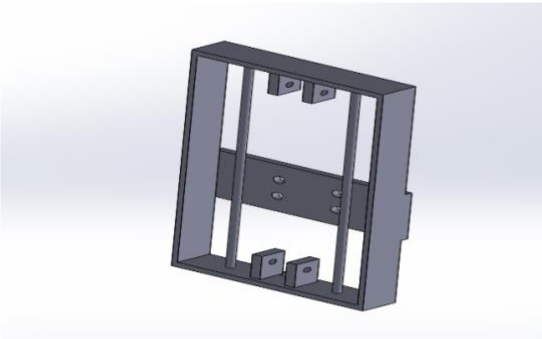


Figure 21: *Carriage*

Base:

It is the part to which different attachments like tires are attached and also counterweight is also on it. Base design is shown in Figure 22.

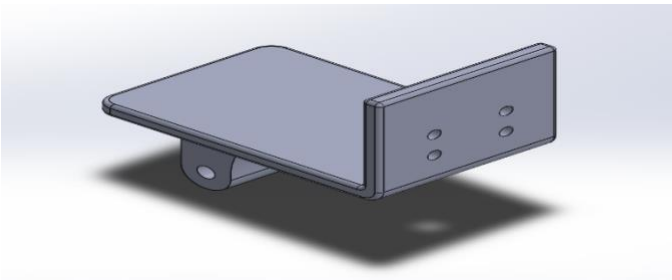


Figure 22: Base

Steering:

This part controls the movement of the forklift by its rotation and is manually operated. The steering part is shown in Figure 23.

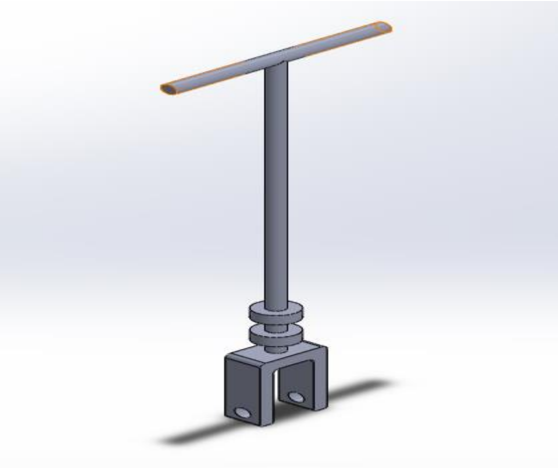


Figure 23: Steering

Power Source:

As our primary focus is on the electric-powered forklifts, so the different sources used to power up the Forklift with their benefits and drawbacks are mentioned below,

Source	Benefits	Drawbacks
Lead Acid Battery	Easily available in the market can be used for counterbalancing No noise Low maintenance cost	It is very costly It is challenging to dispose It is necessary to replace it after sometime
Fuel Cell	No emissions	Not durable

	Quick refueling for long-running times	Acquisition cost is very high Not too much efficient
Hybrid	New technology No noise and significant run times	Expensive than lead-acid batteries After four years need to replace the battery Complications in the disposal.

Tires:

Selection of tires is based on ground conditions. Since this is an indoor unit so solid tires were selected as shown in Figure 24.

Figure 24: Tires

3.4.2. Preliminary Designs:

Three preliminary designs were developed. Preliminary designs are described below;

Design 1

Design 1 uses four wheels and rope pulley system for lifting. This design takes more space, and has less maneuverability than the other designs. Thus, this design was not considered for design calculations. Figure 25 shows preliminary design 1.

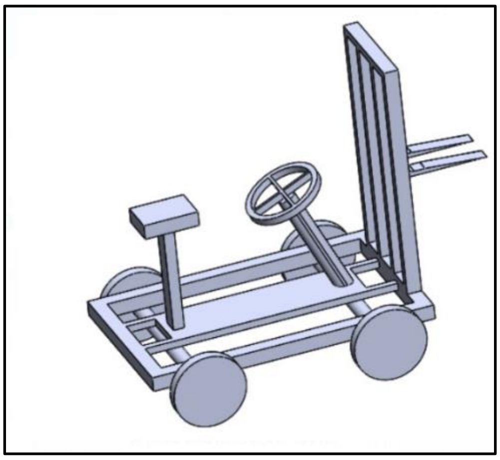


Figure 25: Design 1

Design 2

Design 2 uses two wheels and hydraulic pedalift system for lifting. Although the design is statically stable but lateral tipping over chances during transportation are more as compared to the other designs. This design was also not included in design calculations. Preliminary design 2 is shown in Figure 26.

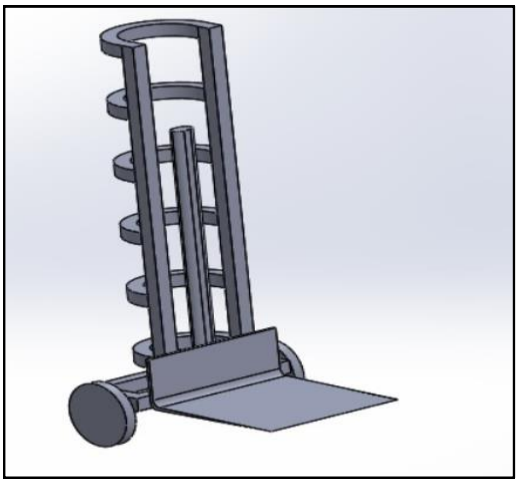


Figure 26: Design 2

Design 3

This design consists of three wheels which increased its maneuverability by decreasing its turning radius. It can lift the load by maintaining its stability and works best in confined warehouses.

The comparison matrix of above designs is given below;

	Design 1	Design 2	Design 3
Number of wheels	4	2	3
Lifting Mechanism	Rope pulley	Hydraulic pedalift	Rotating chain
Compactness of design	6	8	8
Maneuverability	5	7	8
Vehicle Stability	8	4	9

Total	19	19	25

From the above matrix, it was concluded that Design 3 is best for confined warehouses.

Final Design:

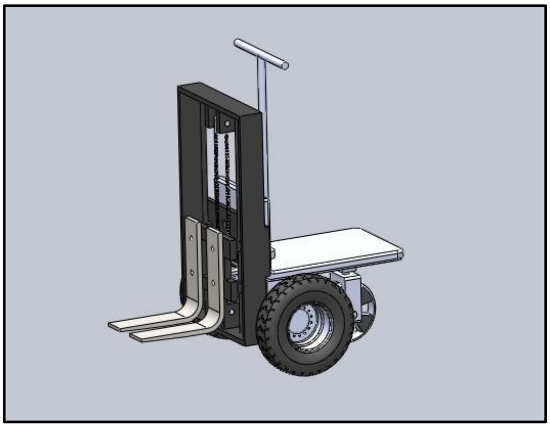


Figure 27: *Final Design*

The design was finalized and is shown in Figure 27. The detailed design of each component was then carried out.

3.5. Design Calculations:

An overview of detailed design process is shown in Figure 28.

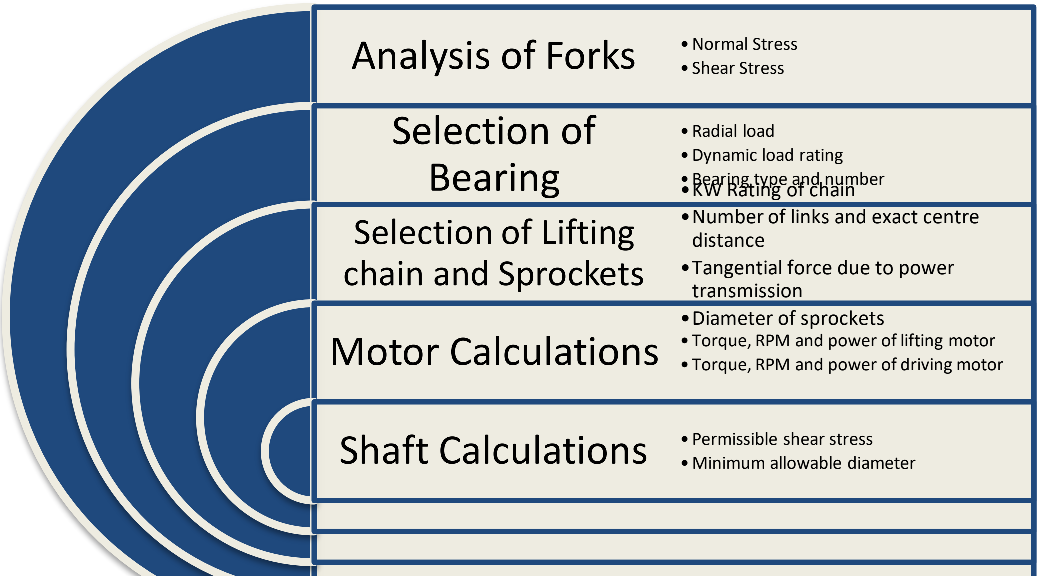


Figure 28: Design Calculation Summary

Details of design process of each component is given in the following sections;

3.5.1. Design of Forks

For the design of forks, two cases were considered. In the first case, consider forks as a cantilever beam with point loading. In the second case, the forks were considered a cantilever beam with a uniform distributed load. Maximum normal and shear stress were computed with the help of shear force and bending moment diagrams and compared with the yield strength of the material used to determine whether these stresses are within the permissible limit or not. The complete process is summarized in the Figure 29.

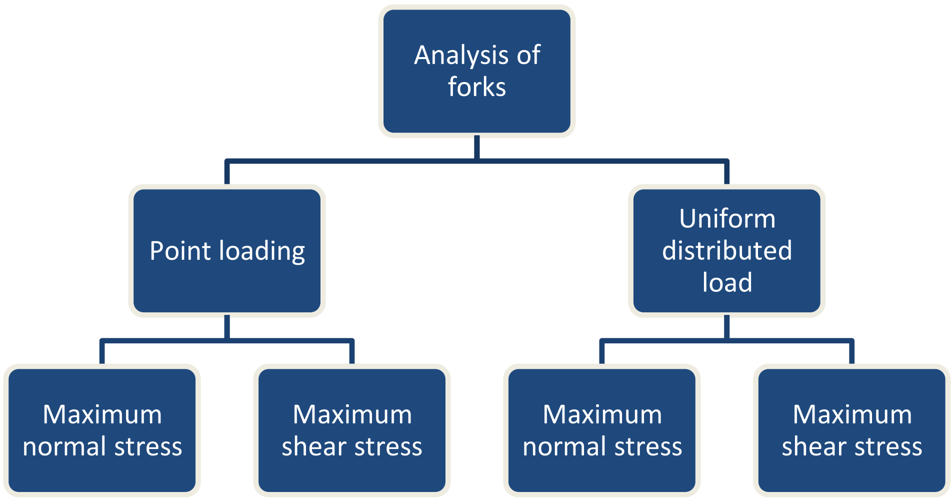


Figure 29: Analysis of Forks

Forks dimensions were assumed based on the requirements and stress calculations were done based on load.

<u>Dimensions of forks</u>

The dimensions of forks are given below:

$Length = 0.4\ m$

$Width = 0.1\ m$

$Thickness = 0.015\ m$

Case-1: By considering fork as a Cantilever Beam with Point Loading

According to the requirements, the maximum load to be lifted by the forklift is 20kg. Therefore, the maximum load on each fork will be 10kg. Hence for each fork,

$maximum\ load = m = 10\ kg$

$gravitational\ acceleration = g = 9.8067\ \frac{m}{s^2}$

$W = m\ .g = (10)(9.8067) = 98.067\ N$

The moment generated will be calculated as,

$Length\ of\ fork = l = 0.4\ m$

$M = W\ .l = (98.067)(0.4) = 39.227\ Nm$

Using sign convention,

$M_x = -39.227\ Nm$

<u>Free Body Diagram of Fork:</u>

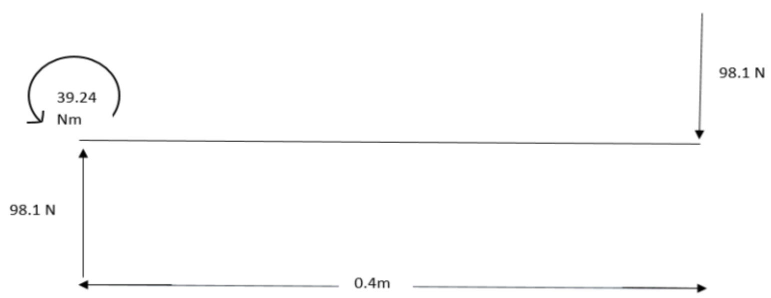
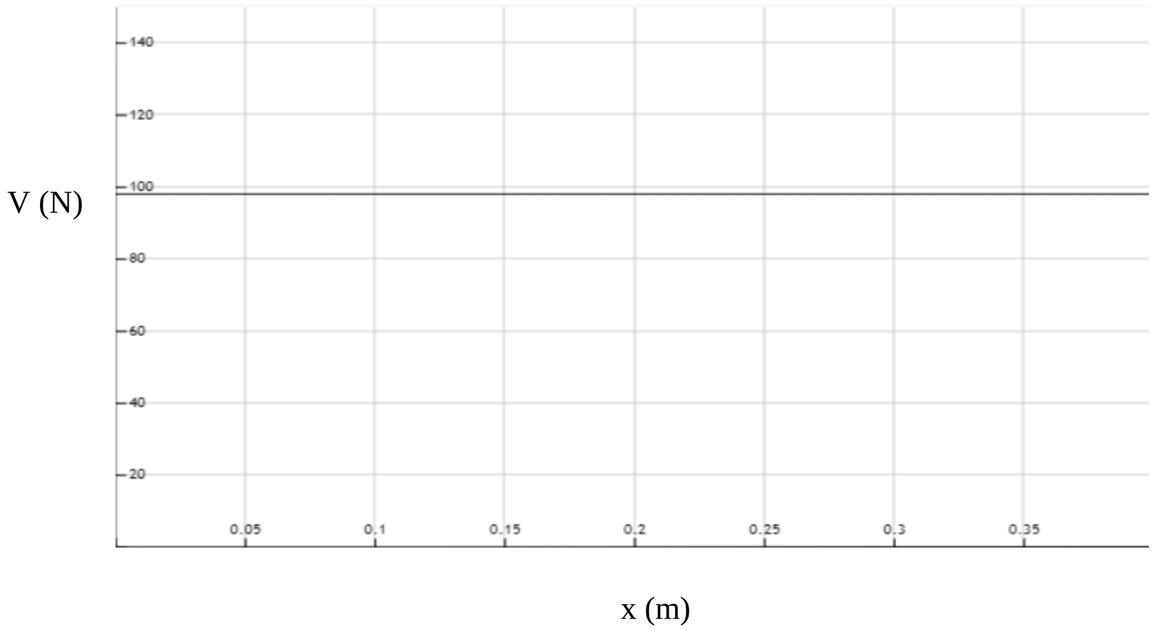


Figure 30: Free Body Diagram of Fork for Point Loading

To find ***Vmax and Mmax***, *the Shear For*ce and Bending Moment Diagrams were used.

<u>Shear Force Diagram</u>



Hence from the shear force diagram,

$V_{max} = 98.1\ N$

<u>Calculation of Maximum Deflection y_{max}:</u>

The centroid moment of inertia of the cross-section is given by,

$$I = \frac{1}{12} bh^3 = \frac{1}{12}(0.1)(0.015)^3$$

$$I = 2.8125\ .10^{-8}\ m^4$$

For mild steel the modulus of elasticity is;

$$E = 215\ .10^9\ Pa$$

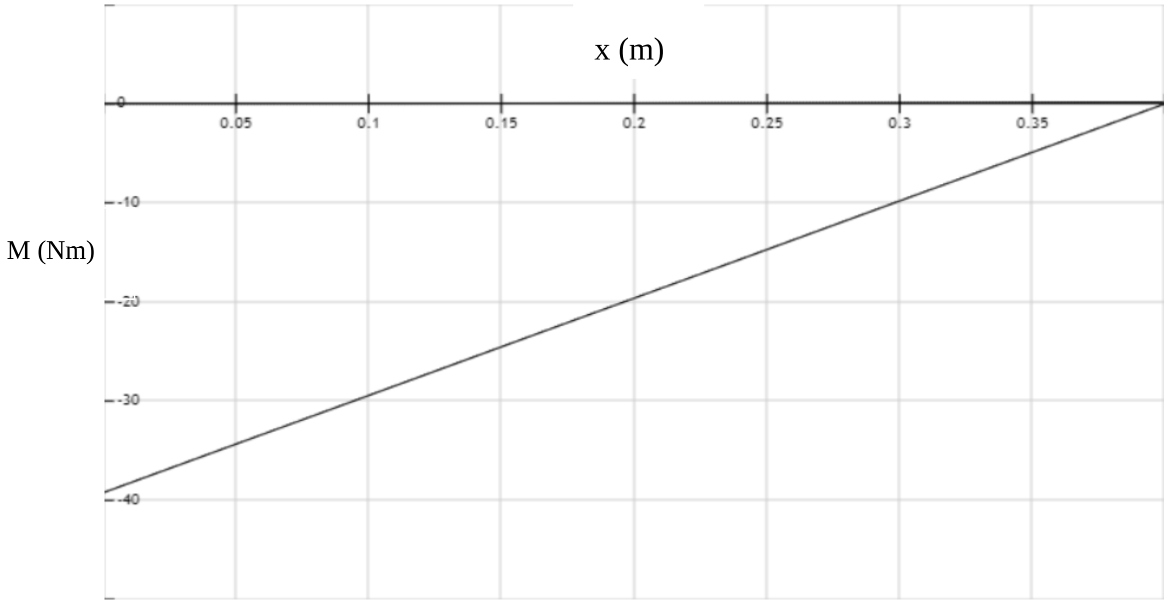
Therefore, the maximum deflection is computed as;

$$y_{max} = \frac{Wl^3}{3EI} = \frac{(98.067)(0.4)^3}{3(215.10^9)(2.8125.10^{-8})} = 3.46.10^{-4}\ m$$

Calculation of Maximum Bending (Normal) Stress:

To calculate the maximum bending stress, find M_{max} which is obtained from the bending moment diagram.

Bending Moment Diagram



Hence from the Bending Moment Diagram

$|M_{max}| = 39.24\ Nm$

Also,

$c = perpendicular\ distance\ to\ neutral\ axis = \frac{0.015}{2} = 0.0075m$

Therefore, the bending stress can be calculated as,

$$\sigma_b = \frac{Mc}{I} = \frac{(39.24)(0.0075)}{(2.8125.\ 10^{-8})} = 1.046 * 10^7\ Pa$$

$\sigma_b = 10.46\ MPa$

For Mild steel, the yield stress,

$\sigma_Y = 250MPa$

Clearly,

$\sigma_b < \sigma_Y$

It means that the design is feasible since maximum bending stress is less than the yield stress, and the material will not fail.

Calculations of Maximum Shear stress

From shear flow formula;

$$\tau_{max} = \frac{V_{max}Q_{max}}{It}$$

Where shear flow Q is maximum at the neutral axis and its maximum value is given by,

$$Q_{max} = \frac{(0.1)(0.015)^2}{(2)(4)} = 2.8125.10^{-6}m^3$$

Substituting values in shear flow formula;

$$\tau_{max} = \frac{(98.1)(2.8125.10^{-6})}{(2.8125.10^{-8})(0.1)} = 9.81.10^4Pa$$

Clearly,

$\tau_{max} < \sigma_Y$

It means that the design is feasible since maximum shear stress is less than the yield stress, and the material will not fail.

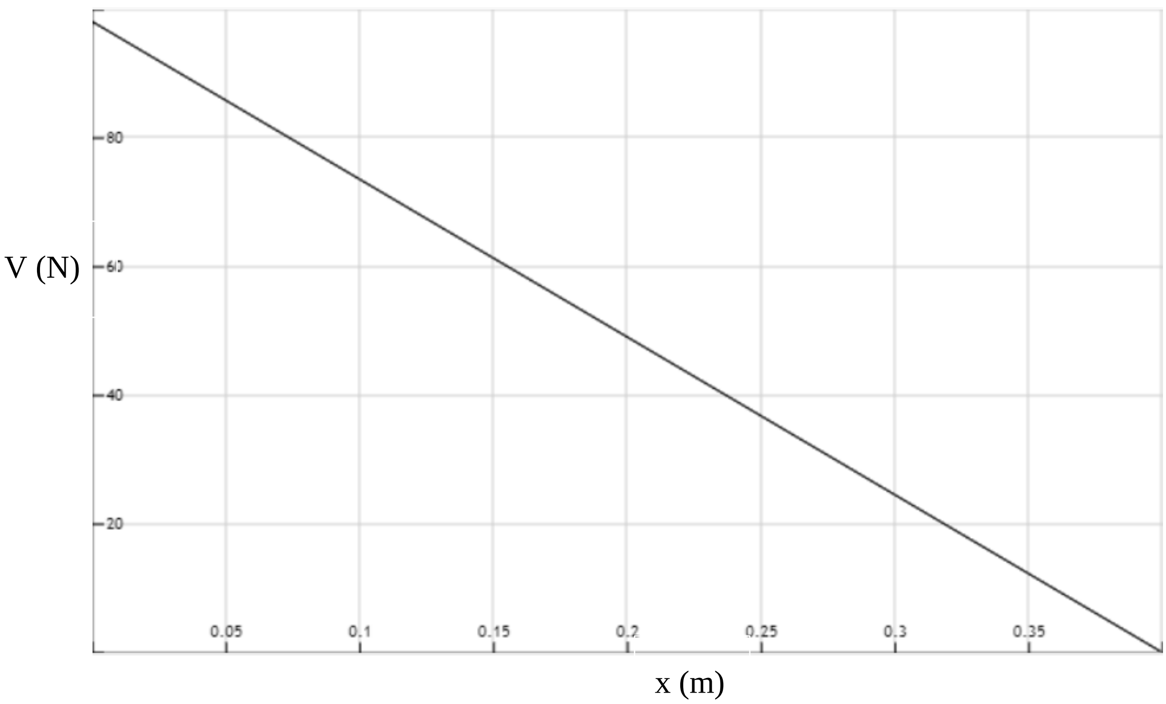
Case-2: By considering fork as a cantilever beam with uniform distributed load

The load on one fork will be 10kg, Thus. the intensity of distributed load on the fork will be

$$w = \frac{(10)(9.81)}{0.4}$$

$$load\ intensity = w = 245.25N/m$$

<u>Shear Force Diagram</u>



From the shear force diagram

$$V_{max} = 98.1$$

<u>Calculation of Maximum Deflection</u>

The maximum deflection is computed as,

$$y_{max} = \frac{Wl^4}{8EI}$$

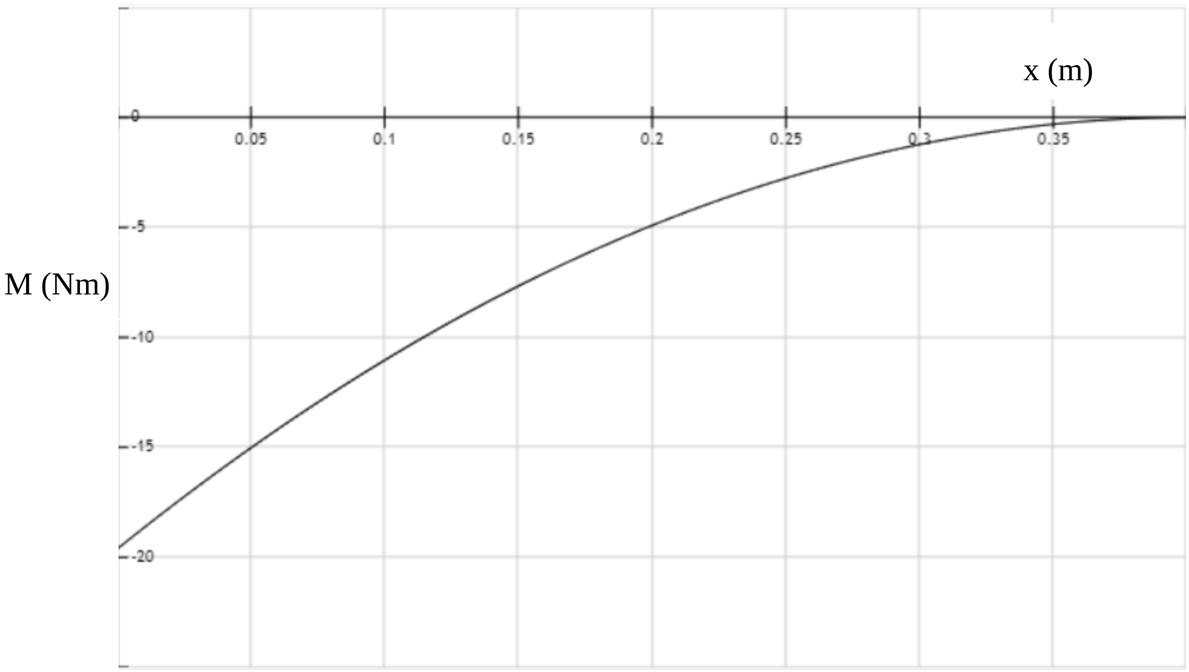
$$y_{max} = \frac{(98.1)(0.4)^4}{8(215.10^9)(2.8125.10^{-8})} = 5.19.10^{-5}m$$

<u>Bending Moment Diagram</u>

To obtain bending moment;

$$\sum M_x = -98.1x + 245.25x\left(\frac{x}{2}\right) + M_x + 19.62 = 0$$

$$\boldsymbol{M_x = (-122.625x^2 + 98.1x - 19.62\ )\ Nm}$$



Hence from the bending moment diagram,

$$|M_{max}| = 19.62Nm$$

<u>Calculation of Maximum Bending (Normal) Stress:</u>

The maximum bending stress can be calculated as,

$$\sigma_b = \frac{M_{max}c}{I} = \frac{(19.62)(0.0075)}{(2.8125.10^{-8})} = 5.232.10^6\ Pa$$

$$\sigma_b = 5.232\ MPa$$

Since for Mild steel, the yield stress

$$\sigma_Y = 250MPa$$

Clearly,

$$\sigma_b < \sigma_Y$$

It means that design is feasible since maximum bending stress is less than the yield stress.

Therefore the material will not fail.

Calculation of Maximum Shear Stress

From shear flow formula;

$$\tau_{max} = \frac{V_{max}Q_{max}}{It}$$

Where shear flow Q is maximum at the neutral axis and its maximum value is given by;

$$\mathrm{Q_{max}} = \frac{(0.1)(0.015)^2}{(2)(4)} = 2.8125.10^{-6}\mathrm{m}^3$$

$$\tau_{max} = \frac{(98.1)(2.8125.10^{-6})}{(2.8125.10^{-8})(0.1)} = 9.81.10^4 Pa$$

Clearly,

$$\tau_{max} < \sigma_Y$$

It means that design is feasible since maximum shear stress is less than the yield stress; therefore, the material will not fail.

3.5.2. Motor Calculations

For the final design, two identical motors are required which will be used to drive the front wheels and one motor will be used to lift the forks. This section shows step-by-step calculations for the rotational speed, torque, and power requirements for each of the motor.

3.5.2.1 Selection of driving/traction motor

The step-by-step process for the selection of driving/traction motor is given below.

Step 1: Determination of the total weight and traction requirements

Forklift mass is estimated for the selection of motors.

$$kerb\ mass\ of\ forklift = m_{forklift} = 94\ kg$$

The maximum load to be carried by the forklift is;

$$total\ load\ lifted = m_L = 20\ kg$$

$$mass\ of\ operator = m_{operator} = 75\ kg$$

Similar sized electric forklifts usually have a maximum speed in the range of 0.4 to 0.8 m/s. Therefore, the nominal speed of the forklift is;

$$nominal\ speed = v_N = 0.5\frac{m}{s}$$

Usually, the surface floor of warehouses is flat, but this design assumes a maximum slope incline;

$$maximum\ slope\ incline = k = 10\%$$

$angle\ of\ incline = \propto = \tan^{-1}(0.1) = 5.71°$

In this calculation, acceleration from $v = 0$ is not considered and it is assumed that the forklift is already in motion. Furthermore, the drag force acting on the forklift was neglected, as the forklift will be operating in closed working environments with nominal speed.

The traction force can be calculated as,

$F_{tr} = F_R + F_{grad}$

$Force\ due\ to\ rolling\ friction = F_R = \mu_R N$

Where,

$coefficient\ of\ rolling\ friction = \mu_R = 0.013$

$Normal\ force = N = m_t g$

$total\ mass = m_t = m_{forklift} + m_{operator} + m_L$

$m_t = 94 + 75 + 20 = 189\ kg$

$N = (189)(9.81) = 1854.1\ N$

$F_R = (0.013)(1854.1) = 24.1\ N$

$Force\ due\ to\ gradient = F_{grad} = m_t g \sin(\propto)$

$F_{grad} = (189)(9.81)\sin(5.71°) = 184.47\ N$

$F_{tr} = 24.1 + 184.47 = 208.57\ N$

Step 2: Calculation of the traction wheel rotation speed

$wheel\ diameter = D_w = 40.64\ cm = 0.4064m$

The front traction wheel rotation speed can be calculated as,

$$wheel\ rotation\ speed = N_T = \frac{60v_N}{\pi D_w}$$

$$N_T = \frac{(60)(0.5)}{(\pi)(0.4064)} = 23.5rpm$$

<u>Step 3: Calculation of the traction motor power and torque</u>

The total power required to operate the forklift at the nominal speed is calculated as;

$$total\ power = P_{forklift} = F_{tr}v_N$$

$$P_{forklift} = (208.57)(0.5) = 104.3\ W$$

The power provided per traction motor,

$$P_T = \frac{P_{forklift}}{2} = \frac{104.3}{2} = 52.15\ W$$

The corresponding torque per traction wheel is given as,

$$T_T = \left(\frac{1}{2}\right)\left(\frac{D_w F_{tr}}{2}\right)$$

$$T_T = \frac{(0.4064)(208.57)}{(4)} = 21.2\ Nm$$

<u>Step 4: Requirements for traction motor</u>

All the essential parameters for the selection of traction motor have been computed. Requirements for the traction motor (x2) are:

Nominal rotational speed	Torque	Power
• $N_T = 23.5\ rpm$	• $T_T = 21.2\ Nm$	• $P_T = 52.15\ W$

Step 5: Selection of traction motor

According to the requirements, the motor selected from the datasheet is LST10 24VDC 120 RPM Brushed Motor.

Characteristic Curve of selected Motor

The characteristic curve of LST10 24V, DC motor obtained from datasheet is shown in the Figure 31.

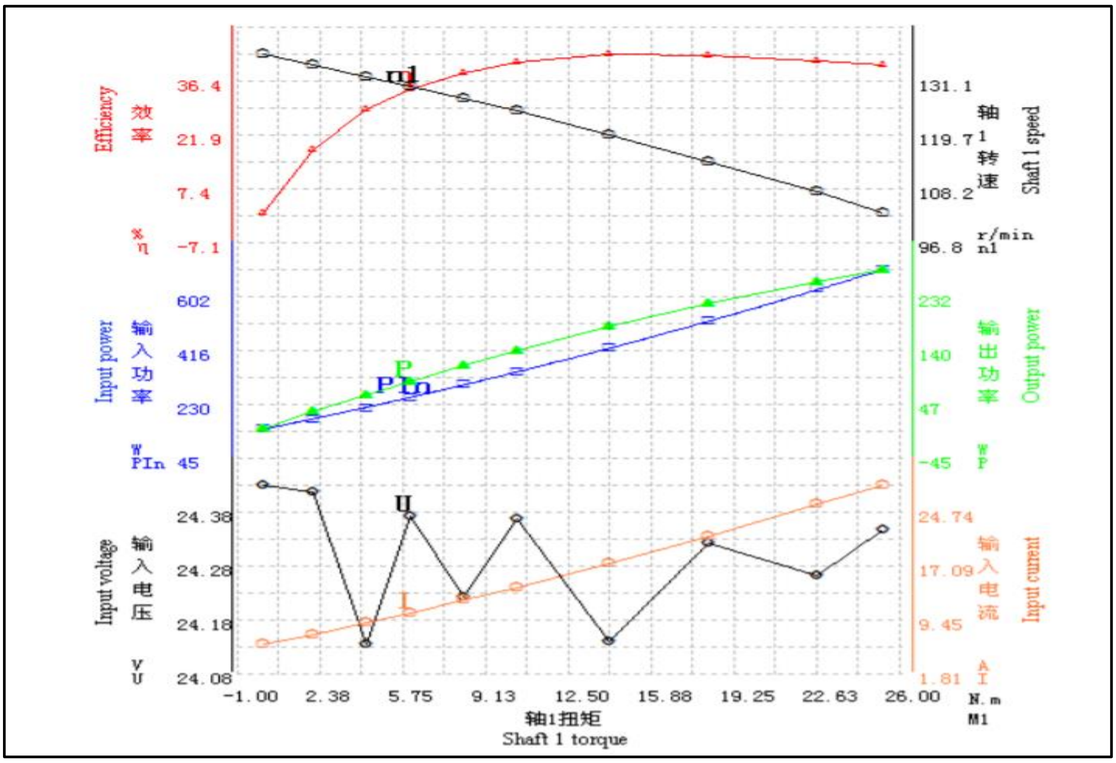


Figure 31: Characteristic Curve of LST10 Motor

The essential results from the characteristic curve are summarized in the Table below:

Table: Motor Characteristics

Item	Current (A)	Output speed (rpm)	Output torque (N.m)	Efficiency(%)	Input power (w)	Output power (W)
Max. efficiency(%)	18.577	138.6	14.2	45.4	453.6	206
Max. output power(w)	25.841	118.2	21.3	42.1	626.1	263.8
Rated output speed(rpm	17.08	119.1	14.08	42.1	412	215

The requirement for rotational speed is 23.5 rpm, therefore the motor needed to geared down. To compute the reduction ratio, identify the motor working point on the characteristic curve corresponding to the output torque required ($T_T = 21.2\ Nm$).

As shown in the table above,

$$output\ speed\ corresponding\ to\ T_T = 118.2\ rpm$$

Therefore,

$$reduction\ ratio = R = \frac{23.5}{118.2}$$

$$R \approx \frac{1}{5}$$

From the Table;

$$motor\ efficiency = \eta_{motor} = 42.1\ \%$$

And from the datasheet;

$$gearbox\ efficiency = \eta_G = 70\ \%$$

Taking these efficiencies into account required motor power is,

$$P_{motor} = \frac{P_T}{\eta_{motor}\eta_G}$$

$$P_{motor} = \frac{52.15}{(0.421)(0.7)} = 176.96\,W$$

The output power 263.8 W is greater than 176.96 W, This implies that the selection of the motor is correct and it will provide the necessary traction power.

3.5.2.2 Selection of lifter motor

The step by step process for the selection of lifter motor is given below.

Step 1: Determination of lifting requirements

Friction is ignored during this calculation.

According to the requirements, the maximum load to be carried by our Forklift is;

$maximum\ load\ lifted = m_L = 20\ kg$

$mass\ of\ forks = m_{forks} = 2(3.85) = 7.7\ kg$

$total\ mass\ lifted = m_T = m_{forks} + m_L = 7.7 + 20 = 27.7\ kg$

Also, the nominal load lifting speed for the forklift;

$load\ lifting\ speed = v_L = 0.2\frac{m}{s}$

Step 2: Calculation of the lifter motor rotation speed

From manufacturer's catalogue,

$pitch\ diameter\ of\ sprocket = D_p = 48.82\ mm$

The nominal rotation speed is calculated as,

$$nominal\ rotation\ speed = N_L = \frac{60v_L}{\pi D_p}$$

$$N_L = \frac{(60)(0.2)}{(\pi)(0.04882)} = 78.24\ rpm$$

Step 3: Calculation of the lifter motor power and torque

The force required to lift the fork and the load is,

$$F_L = m_T g = (27.7)(9.81) = 271.74\ N$$

The power required for lifting is,

$$lifting\ power = P_L = F_L v_L$$

$$P_L = (271.74)(0.2) = 54.35\ W$$

The corresponding torque is,

$$T_L = \left(\frac{D_P}{2}\right)(F_L)$$

$$T_L = \frac{(0.04882)(271.74)}{(2)} = 6.63\ Nm$$

Step 4: Requirements for lifter motor

All the essential parameters for the selection of lifter motor have been computed. Requirements for the lifter motor are:

Nominal rotational speed	Torque	Power
• $N_L = 78.24\ rpm$	• $T_L = 6.63\ Nm$	• $P_L = 54.35\ W$

Step 5: Selection of lifter motor

According to the requirements mentioned above, the motor selected from the datasheet is brushed 12V DC motor, model CL-53 as shown in Figure 30.

Figure 32: Motor

The motor has a rated power of 100W at an output speed of 80rpm.

Circuit Diagram of Lifting Motor:

The wiring diagram of lifting motor is shown in Figure 33.

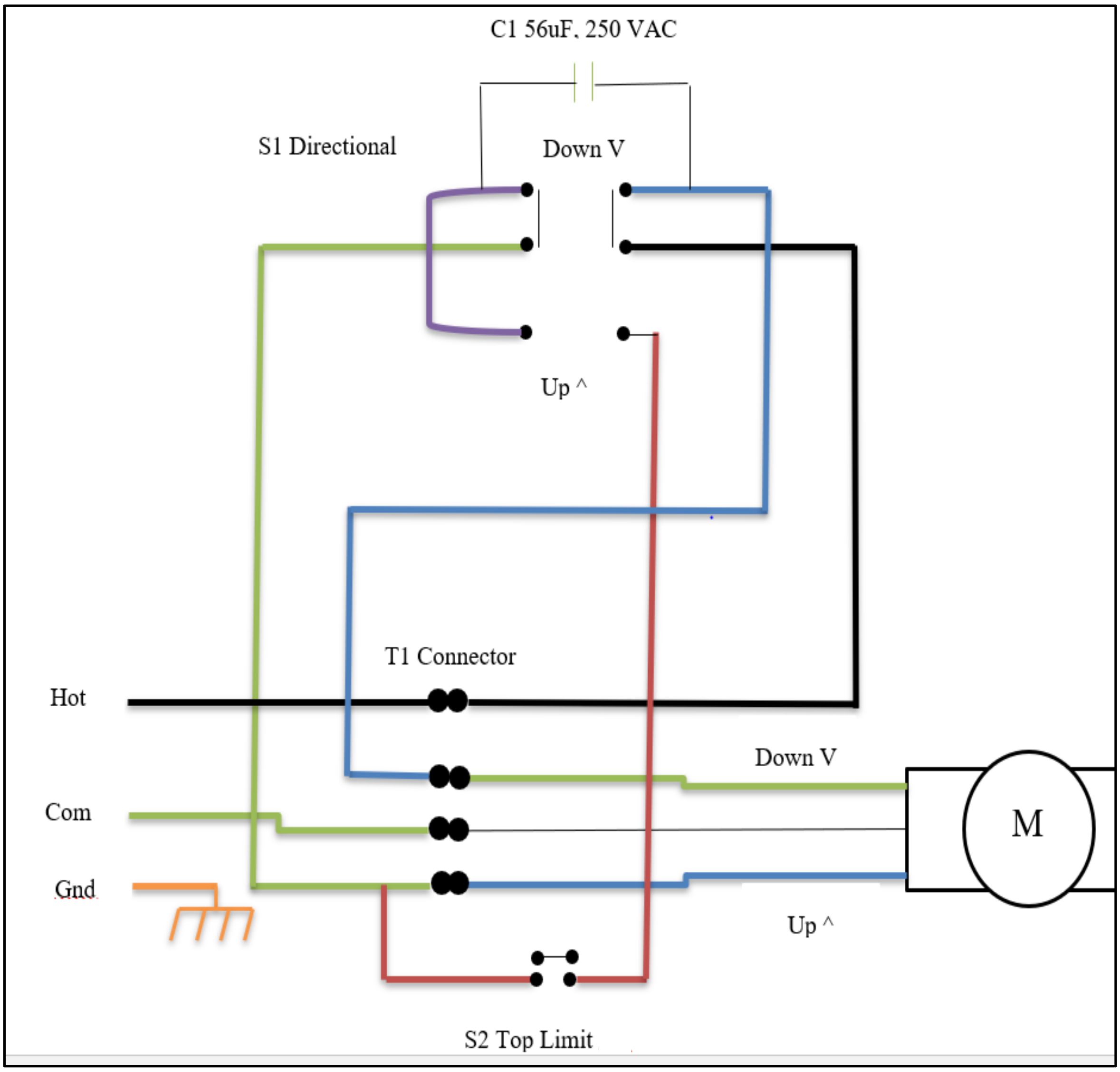


Figure 33: Circuit/Wiring Diagram of Lifting Motor

3.5.3. Design of Lifting Chain and Sprocket

Power required for lifting is;

$$P_L = 54.35\ W$$

The kilo-watt rating of the chain can be calculated using;

$$KW\ Rating\ of\ chain = \frac{(KW\ to\ be\ transmit)(K_s)}{(K_1)(K_2)}$$

$K_s = Service\ factor = 1$

$K_1 = multiple\ strand\ factor = 1$

$K_s = tooth\ correction\ factor = 1$

$$W\ rating\ of\ chain = \frac{(54.35.10^{-3})(1)}{(1)(0.72)} = 75.5.10^{-3}\ KW$$

According to the requirements, we select a Grade 80 alloy steel chain with the following specifications was selected.

Chain diameter	Pitch	Outside width	Weight
• $d = 10\ mm$	• $p = 30\ mm$	• $b = 37\ mm$	• $w = 2.2\ \frac{kg}{m}$

For the driving sprocket,

$pitch\ diameter = d_p = 48.82\ mm$

$number\ of\ teeth = z_1 = 16$

Usually, the center distance between sprockets is in the range of 30p to 50p. Thus;

$centre\ distance\ between\ sprockets = a = 45p = (45)(30) = 1350\ mm$

Now we can compute the number of links in the chain,

$$Number\ of\ links = L_n = \left(\frac{2a}{p}\right) + \left(\frac{z_1 + z_2}{2}\right) + \left(\frac{z_2 - z_1}{2\pi}\right)^2 \left(\frac{p}{a}\right)$$

$$L_n = \left(\frac{2(1350)}{30}\right) + \left(\frac{16+16}{2}\right) = 106\ links$$

Exact center distance,

$$a = \frac{p}{4}\left\{L_n - \left(\frac{z_1+z_2}{2}\right) + \sqrt{\left[L_n - \left(\frac{z_1+z_2}{2}\right)\right]^2 - 8\left(\frac{z_2-z_1}{2\pi}\right)^2}\right\}$$

Simplifying and plugging in values we get,

$$a = \frac{2(30)}{4}\left[106 - \left(\frac{16+16}{2}\right)\right]$$

$$a = 1350\ mm$$

This means that the initial assumption was correct

$$length\ of\ chain = L = L_n p$$

$$L = (106)(30) = 3180\ mm$$

The average velocity of the chain,

$$v = \frac{\pi D_p N}{60}$$

$$nominal\ rotation\ speed\ for\ lifting = N = 78.24\ rpm$$

$$v = \frac{\pi(48.82)(78.24)}{60} = 200\ {}^{mm}/_{s}$$

Tangential force due to power transmission,

$$F_t = \frac{102P}{v} = \frac{(102)(75.5.10^{-3})}{(200.10^{-3})} = 38.5\ kgf = 377.73\ N$$

Tension due to sagging of chain,

$F_s = (k)(w)(a)$

Where,

$k = coefficient\ of\ slag\ (for\ vertical) = 1$

$w = weight\ of\ chain$

$a = central\ distance$

$F_s = (1)(2.2)(1350.10^{-3}) = 2.97\ kgf = 29.14\ N$

The chain selected above is acceptable based on the requirements since the load for which it is tested (test load) is higher than the load it has to bear in our design.

3.5.4. Design of Shaft

Mild steel was selected as the material of the shaft.

For mild steel,

$\sigma_Y = 250\ MPa$

According to the requirements, a factor of safety was chosen;

$Factor\ of\ safety = FOS = 1.5$

Therefore;

$$\tau_{permissible} = \frac{\sigma_Y}{FOS} = \frac{250}{1.5} = 166.67\ MPa$$

Furthermore, the solid circular shaft is subjected to a bending moment of 200 Nm and a twisting moment of 150Nm based on the bending moment diagram and torsional moment diagram. Two different theories were used to find a suitable diameter for the shaft.

Maximum principal stress theory (MPST)

It is generally used for brittle materials in which failure occurs by brittle fracture.

In this case;

$bending\ moment = M = 200\ Nm$

And,

$torsional\ moment = T = 150\ Nm$

Therefore, using MPST, the diameter of the shaft can be calculated as,

$$d_{shaft} = \left[\frac{16}{\pi\tau_{permissible}}\left(M + \sqrt{M^2 + T^2}\right)\right]^{\frac{1}{3}}$$

$$d_{shaft} = \left[\frac{16}{\pi(166.67.10^6)}\left(200 + \sqrt{200^2 + 150^2}\right)\right]^{\frac{1}{3}} = 23.962\ mm$$

Maximum shear stress theory (MSST)

It is usually used for ductile materials.

Using MSST, the diameter of the shaft can be calculated as,

$$d_{shaft} = \left[\frac{16}{\pi\tau_{permissible}}\left(\sqrt{M^2 + T^2}\right)\right]^{\frac{1}{3}}$$

$$d_{shaft} = \left[\frac{16}{\pi(166.67.10^6)}\left(\sqrt{200^2 + 150^2}\right)\right]^{\frac{1}{3}} = 19.698\ mm$$

After these calculations, we decided to use a shaft of 25mm diameter.

3.5.5 Selection of Bearing

For the front shaft;

$diameter\ of\ shaft = d = 25\ mm$

$Axial\ load = F_a = 0\ N$

Normally 60 % of the weight of the forklift is supported by the front shaft.

$m_t = 189\ Kg$

Also the centripetal force will contribute to the radial load.

$nominal\ rotation\ speed = N = 23.5\ rpm$

Therefore the radial load can be computed as,

$Radial\ load = F_r = 0.6(m_t g) + mr\omega^2$

$$F_r = (0.6)(189)(9.81) + (1.8)(0.0125)\left(\frac{2\pi(23.5)}{60}\right)^2 = 1112.6\ N = 1.1126\ kN$$

Since there is only radial load, single row deep groove ball bearing was used.

For ball bearing,

$a = 3$

As there is inner ring rotation, therefore;

$v = 1$

$equivalent\ load = F_e = vF_r = 1.1126\ kN$

For 90% reliability, hence

$reliability\ factor = k_r = 1$

$load\ factor = k_a = 1$

According to the requirements,

$desired\ bearing\ life = 10{,}000\ hours$

$desired\ life\ in\ cycles = L_D = (10{,}000)(N)(60)$

$L_d = (10{,}000)(23.5)(60) = 14.1\ .10^6\ cycles$

For SKF Bearing,

$L_{10} = 10^6\ cycles$

The required dynamic load rating for the bearing,

$$Required\ dynamic\ load\ rating = C_{req} = F_e \left(\frac{L_D}{L_{10}}\right)^{\frac{1}{a}}$$

$$C_{req} = 1.1126 \left(\frac{14.1\ .10^6}{10^6}\right)^{\frac{1}{3}} = 2.7\ kN$$

Selected bearing is SKF 6005. For ball bearing 6005, C_{10} value is greater than C_{req}.

3.6. Stability Analysis

In this section, the stability analysis of the forklift design has performed to ensure that it meets the stability required for safe operation. The analysis tools used are summarized in the Figure 34.

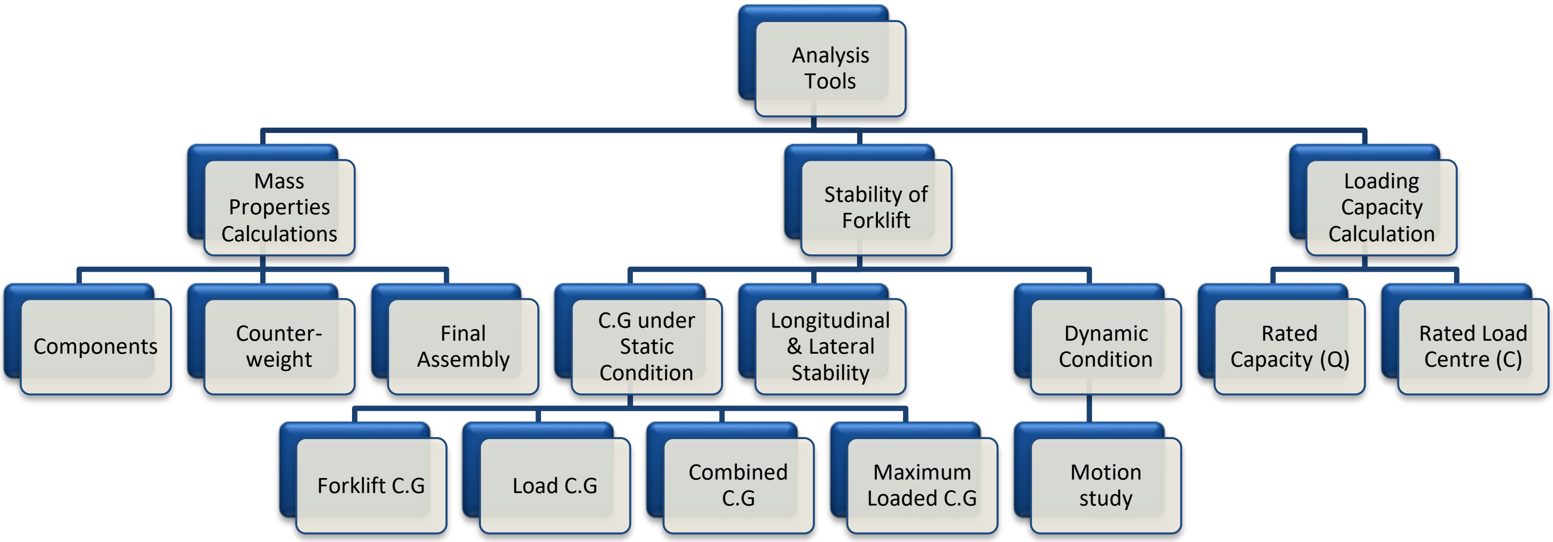


Figure 34: Stability Analysis

Mass Properties Calculations

To compute the mass properties of the design, SolidWorks (CAD system). Every individual component was assigned material properties, and then by using the mass properties calculations tool the principal moments, axes of inertia, volume, mass, and center of mass of the final assembly were obtained. These measurements were necessary for the stability analysis of the forklift. The calculated values are given below;

```
Mass properties of base assembly
     Configuration: Default
     Coordinate system: -- default --

Mass = 123730.15 grams

Volume = 38684660.42 cubic millimeters

Surface area = 5217056.98  square millimeters

Center of mass: ( millimeters )
        X = 3631.09
        Y = -1048.70
        Z = -5642.63

Principal axes of inertia and principal moments of inertia: ( grams *  square millimeters )
Taken at the center of mass.
         Ix = ( 0.30,  0.95,  0.02)      Px = 6819872428.71
         Iy = (-0.68,  0.23, -0.70)      Py = 14603651577.05
         Iz = (-0.67,  0.19,  0.72)      Pz = 15459381332.25

Moments of inertia: ( grams *  square millimeters )
Taken at the center of mass and aligned with the output coordinate system.
        Lxx = 14274842259.93    Lxy = 2353532026.80     Lxz = 466830335.66
        Lyx = 2353532026.80     Lyy = 7568013768.42     Lyz = 57646313.25
        Lzx = 466830335.66      Lzy = 57646313.25       Lzz = 15040049309.66

Moments of inertia: ( grams *  square millimeters )
Taken at the output coordinate system.
        Ixx = 4089826006820.69  Ixy = -468802380075.21  Ixz = -2534623559242.15
        Iyx = -468802380075.21  Iyy = 5578398512944.72  Iyz = 732223699493.48
        Izx = -2534623559242.15 Izy = 732223699493.48   Izz = 1782470914312.01
```

Stability of the Forklift

To evaluate the stability of the design, stability analysis was divided into three categories. First of all, the longitudinal stability of the design was analyzed to ensure that the forklift does not tip forward. In this analysis, the rated capacity of the forklift was computed. Furthermore, a static and dynamic stability analysis was performed. For the evaluation of static stability, the concept of center of gravity and stability triangle was used. To ensure that our forklift is stable under dynamic conditions, the motion study of the design was performed.

Longitudinal Stability

Longitudinal stability refers to the ability of the forklift to resist tipping forward. If excess weight beyond the rated capacity is placed, a tipping moment is produced that can cause the forklift to tip forward. To ensure the longitudinal stability of the forklift the counterweight is an essential component since it counteracts and balances the tipping moment.

As shown in Figure 35, the front wheels act as the fulcrum.

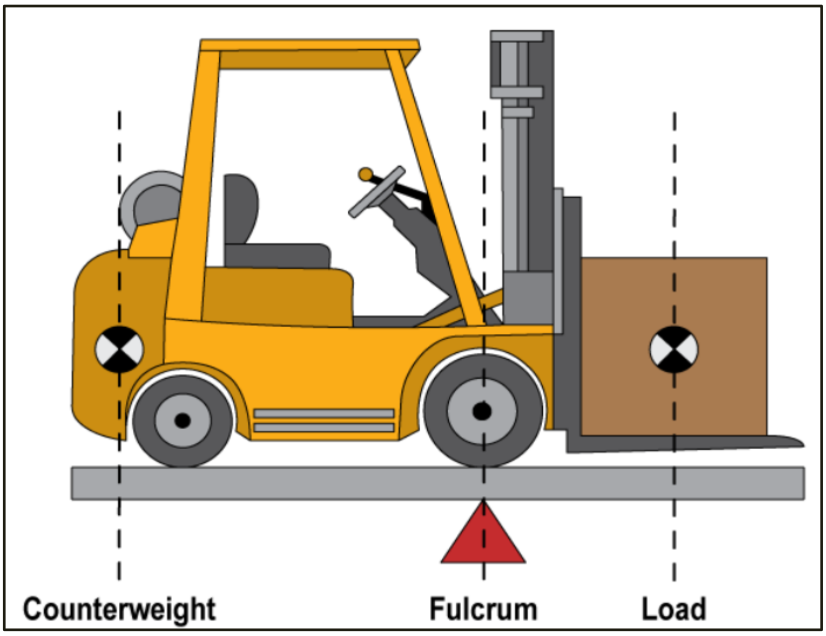


Figure 35: Forklift Longitudinal Stability

For this design,

$$Load\ distance = Front\ axle\ centre\ to\ the\ fork\ surface = X = 0.25\ m$$

$$Rated\ Load\ Centre = C = 0.2\ m$$

$$distance\ between\ fulcrum\ and\ load = C + X = 0.2 + 0.25 = 0.45\ m$$

$$distance\ between\ fulcrum\ and\ counterweight = 0.65\ m$$

For the design, the battery acts as the counterweight, therefore

$$counterweight = weight\ of\ battery = 35\ Kg$$

The rated capacity (Q) for the forklift,

$Forward\ tipping\ Moment = (Q)(0.45)$

$Counter\ Moment = (35)(0.65) = 22.75\ Kg\ .m$

For balance point,

$Forward\ tipping\ Moment = Counter\ Moment$

$(Q)(0.45) = (22.75)$

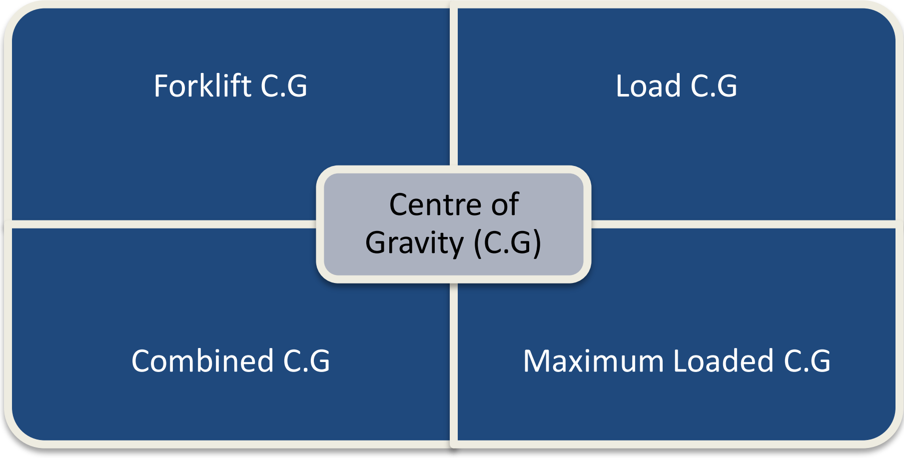
Therefore,

$$Q = \frac{22.75}{0.45} = 50.56\ Kg$$

Hence the rated capacity of the forklift is approximately 50.6 Kg. Here it is important to point out that the weight of the operator is not included in this analysis. The weight of the operator will also add to the counter moment. In the coming sub-sections, the maximum load-carrying capacity of the forklift using stability triangle considerations is computed.

Centre of Gravity under Static Condition

The combined center of gravity must lie within the stability triangle for the forklift to be statically stable. The stability triangle for the design is obtained by connecting the center points of all three wheels, i.e. two front wheels and one rear wheel. It is also necessary to consider different CGs for complete analysis depicted in the Figure be. SolidWorks CAD system was used to find the exact location of the center of gravity (CG).



Stability Triangle of our Forklift

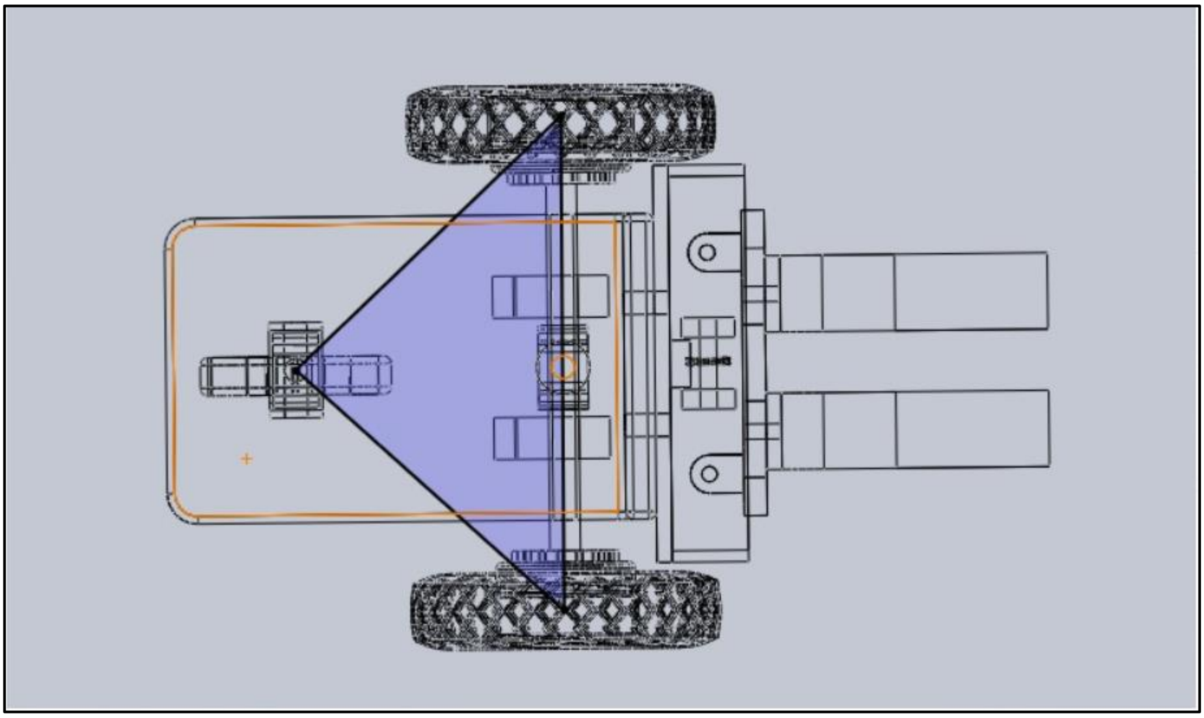


Figure 36: Stability Triangle (top view)

In Figure 36, the stability triangle is visible in the top view of the forklift obtained using SolidWorks. As indicated, the vertices of the triangle are the center points of the wheels. It constitutes an isosceles triangle with a 1m bottom and two 0.7m sides.

<u>Centre of Gravity of our Forklift</u>

First of all, the calculated center of gravity of unloaded forklift is present within the stability triangle. It is shown in the Figure 37.

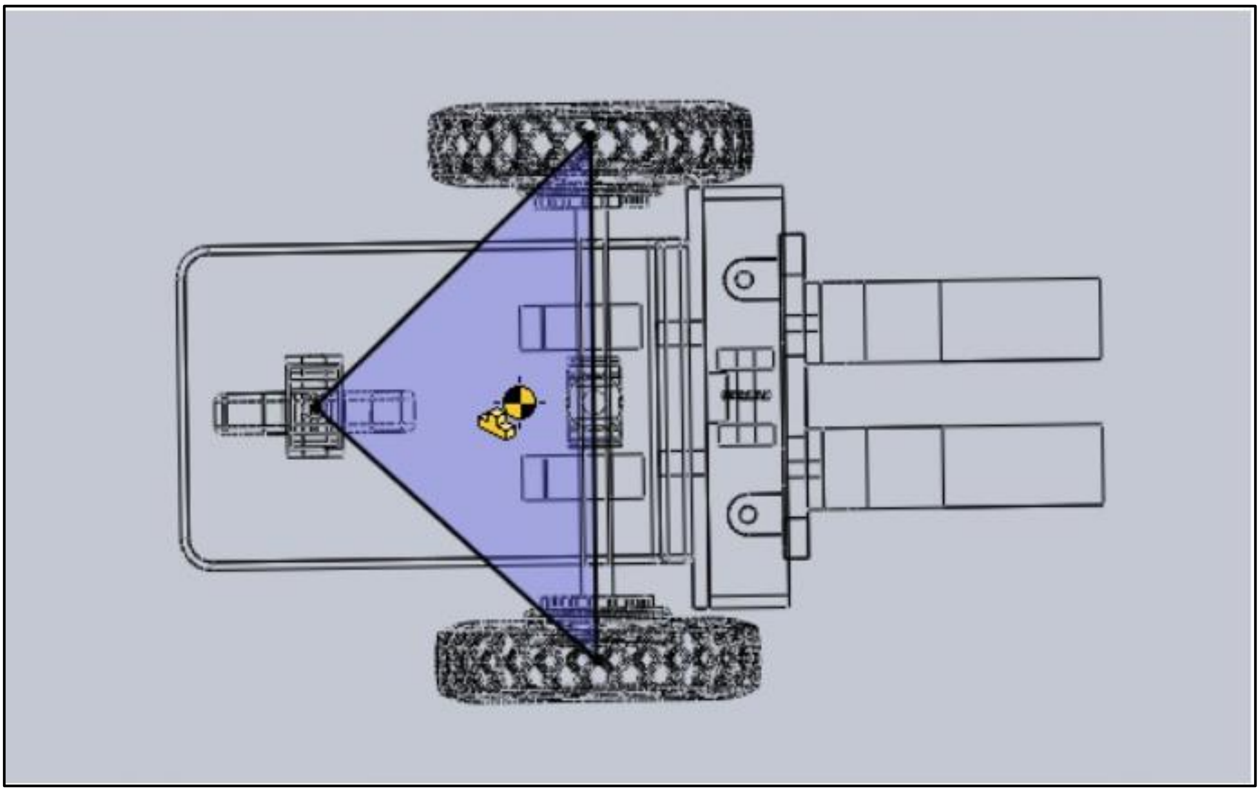


Figure 37: Forklift C.G (Unloaded)

Combined Centre of gravity and Maximum Loading Capacity Calculation

Combined center of gravity is considered under loaded conditions. Under no load condition, the forklift center of gravity and combined center of gravity are coincident. When the forklift is loaded, the combined center of gravity shifts towards the front axle. The load can be added until the combined center of gravity reaches the base of the stability triangle, i.e. endpoint of the safety zone. This method was used to calculate the maximum load-carrying capacity.

Load was incrementally increased to find the maximum load that keeps the center of gravity with in the stability triangle. In the considered design, it came out to be 50 kg. The location of center of gravity under maximum loading is shown in Figure 38.

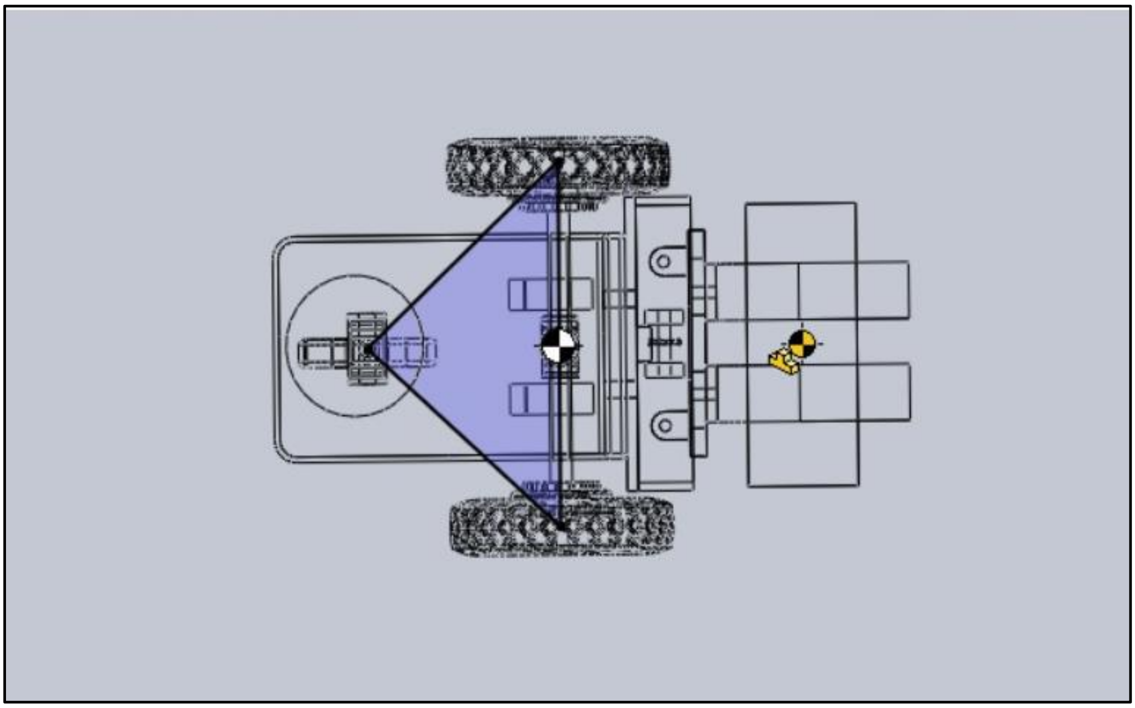


Figure 38: Maximum Loaded Centre of Gravity

Load Centre and Capacity Calculations

The load center is the distance between the fork surface and the load center of gravity. The rated load center is a particular load center for which the forklift can carry maximum load (rated capacity Q).

$Rated\ Load\ Centre = C = 0.2\ m$

If the load center of gravity moves away, the capacity of the forklift to carry load reduces.

For the forklift, the capacity at different load centers was calculated. The load distance (X) is measured from the center of the front axle to the fork surface and is fixed.

$Load\ distance = X = 0.25\ m$

$Rated\ Capacity = Q = 50\ Kg$

$$Capacity\ for\ load\ centre\ of\ 0.25\ m = \frac{(50)(0.2 + 0.25)}{(0.25 + 0.25)} = 45\ kg$$

$$Capacity\ for\ load\ centre\ of\ 0.3\ m = \frac{(50)(0.2 + 0.25)}{(0.3 + 0.25)} = 40.91\ kg$$

$$Capacity\ for\ load\ centre\ of\ 0.35\ m = \frac{(50)(0.2 + 0.25)}{(0.35 + 0.25)} = 37.5\ kg$$

Summary of results is given below;

Load Centre	Capacity (kg)
0.2m	50kg
0.25m	45kg
0.3m	40.91kg
0.35m	37.5kg

The load capacity chart of the forklift design is shown in Figure 39.

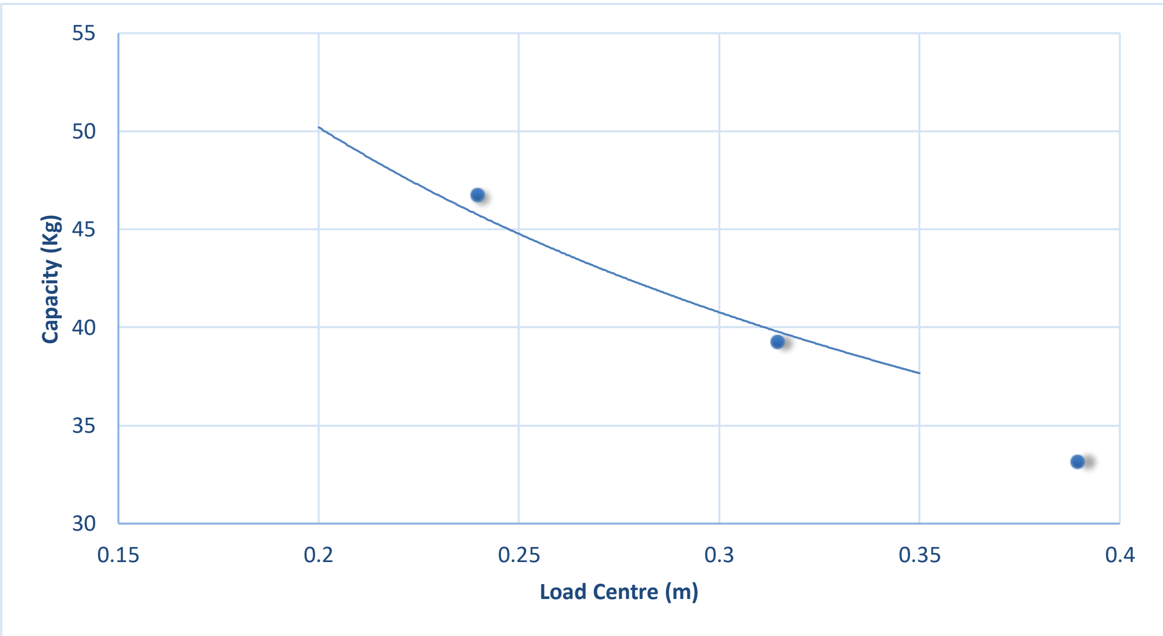


Figure 39: Load Capacity chart

3.7. Economic Analysis:

Following are the significant expenses that are projected to be incurred in the project during the development of forklift prototype:

Cost Break Down		
Expenditure Head	**Description**	**Cost Incurred**
Materials (Mechanical)	Base Frame, Support Frame, Front Drive pneumatic wheels, Rear wheel (solid-state), G80 Lifting Chain, Sprockets, Cast Iron Shaft, Forks, Deep Groove Ball Bearing (6002), Bearing 6005 (Front Wheels)	PKR 34000
Materials (Electrical)	Driving Motor (Wheelchair Motor, Brand ZD Motor), Lifting Motor (Windshield Wiper Motor), Rechargeable VRLA Deep Cycle Lead Acid Gel Battery (Brand Sendon)	PKR 35700

Labor Charges	includes machining and support staff cost	PKR 5000
Logistics and other Support services	For project transportation and other Miscellaneous Expenditure	PKR 5000
Total Cost		**PKR 79700**

Cost saving calculations because of the reduction in labor are as below;

Cost Saving Analysis		
Title/Cost Head	**Unit**	**Quantity**
Labor picking loads manually around 80kg	Number	4
Labor Required for operating Forklift to handle this load	Number	1
Daily Wage of one worker	PKR	500
Labor Cost Saving per day	PKR	1500
Total cost saving per year		**PKR 500,000**

Pay Back Period		
Title/Cost Head	**Unit**	**Quantity**
Initial Investment	PKR	90,000
Cost Saving Per Year	PKR	500,000
Net Cash Inflow in Year 1	PKR	400,000
Pay Back Period		**0.2 years**

Cost saving calculations in comparison with fuel powered forklift of the same size,

Cost Saving Analysis		
Title/Cost Head	**Unit**	**Quantity**
Hourly Fuel Consumption of Forklift	PKR	100
Usage per Day	Hours	6
Cost Saving per day	PKR	600
Total Cost Saving per year		**PKR 210,000**

Pay Back Period		
Title/Cost Head	**Unit**	**Quantity**
Initial Investment	PKR	90,000
Cost Saving Per Year	PKR	210,000
Net Cash Inflow in Year 1	PKR	120,000
Pay Back Period		**0.4 years**

CHAPTER 4: ANALYSIS

4.1. Dynamic Stability Analysis:

We performed a motion study in Solidworks to investigate the dynamic stability of our Forklift. Motion Analysis was performed for worst-case scenarios, i.e. Forklift operating at maximum loading capacity and maximum lift height.

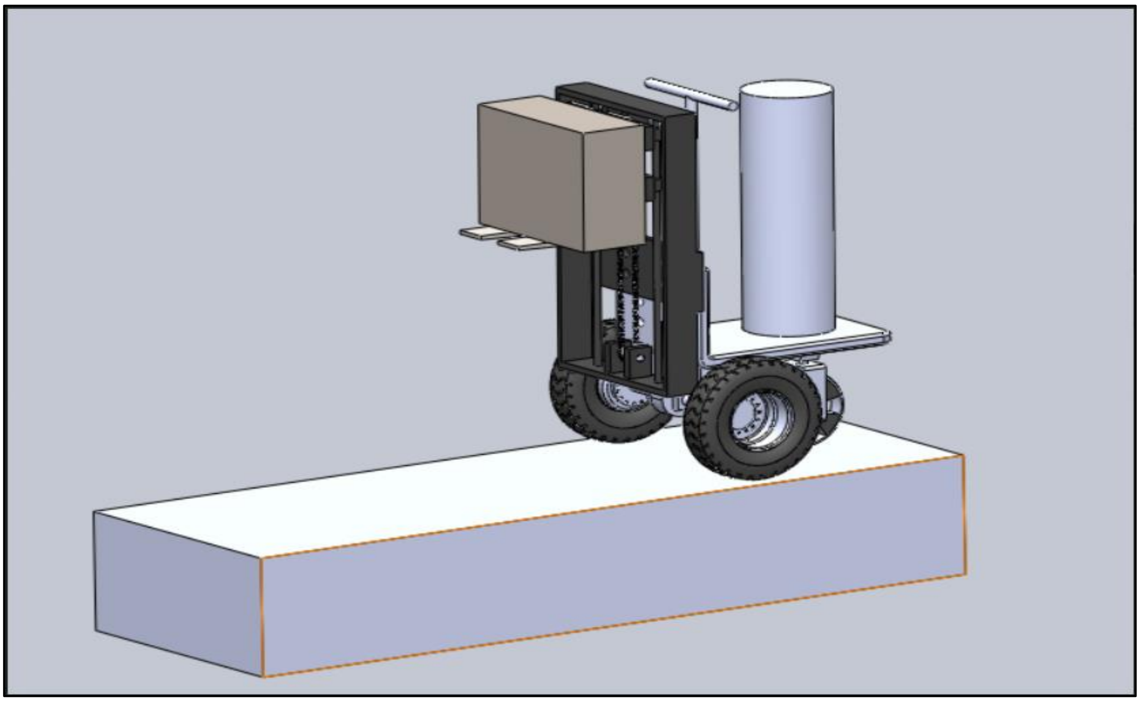


Figure 40: Motion Study

In the Figure 40, the cylinder represents the combined weight of the operator and counterweight.

The motion analysis considers the interaction of different components while undergoing different motions associated with the model. The successful motion study analysis proves that the model is safe and stable for the associated motions.

4.2. FEM/FEA Analysis

In this section, stress analysis of forklift parts was carried out. The software we used for this purpose is SolidWorks 2018.

The analysis is divided into the following four categories.

- Forks Static Analysis
- Support frame fork assembly Static Test
- Support frame fork assembly Drop Test
- Forks Vibration Analysis

4.2.1. Analysis of Fork:

In the static analysis, the maximum loading capacity for this forklift is 20 kg or 200 N. Forklift consists of two forks. The structure of the two forks is symmetrical, thus, both forks will share the load equally so each fork would carry 100 N load.

Figure 41 shows the model obtained because of meshing on the fork. The normal force is applied at the top horizontal of the fork and the fork is fixed as its boundary condition.

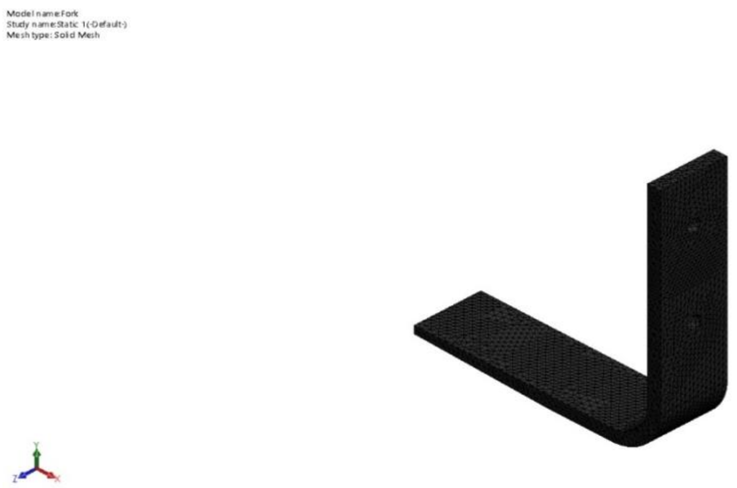
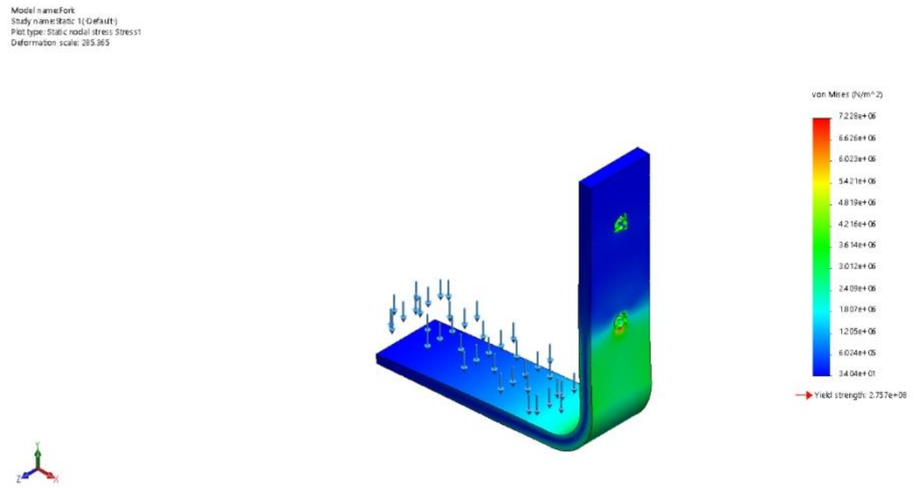


Figure 41: Meshed Model of Fork

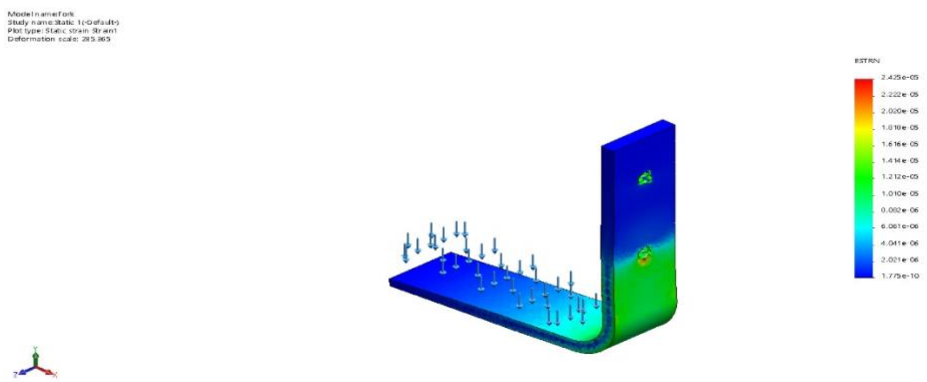
Name	Type	Min	Max
Stress1	VON: von Mises Stress	3.404e+01 N/m^2 Node: 12902	7.228e+06 N/m^2 Node: 7575



Fork-Static 1-Stress-Stress1

Figure 42: Stress Analysis of Fork

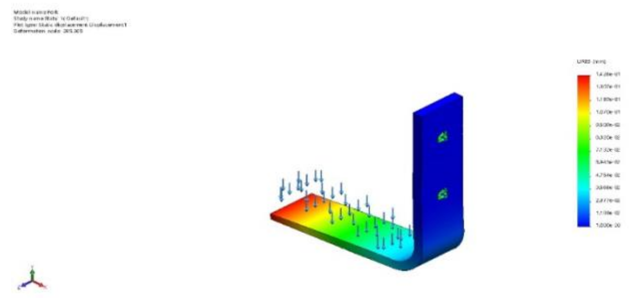
Name	Type	Min	Max
Strain1	ESTRN: Equivalent Strain	1.775e-10 Element: 18753	2.425e-05 Element: 6422



Fork-Static 1-Strain-Strain1

Figure 43: Strain Analysis of Fork

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0.000e+00 mm Node: 488	1.426e-01 mm Node: 473

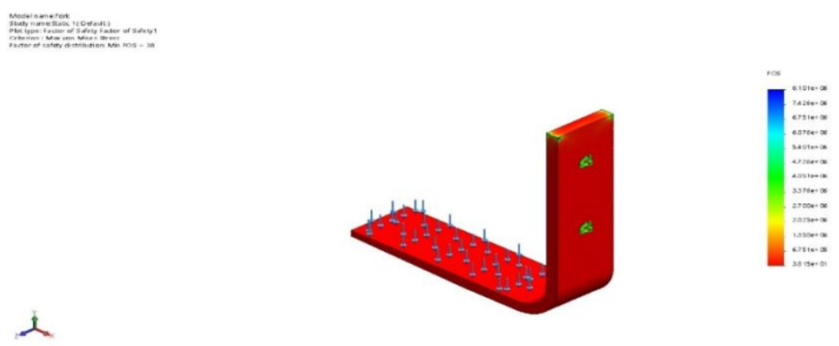


Fork-Static 1-Displacement-Displacement1

Figure 44: Displacement Analysis of Fork

Figures 42, 43 and 44 show stress, strain and displacement analysis. The maximum stress occurs at the fixture location and its value is 7.228 MPa. The maximum strain is also at the same location with a magnitude of 2.425×10^{5}. The end of the fork experiences a maximum displacement of 0.142 mm.

Name	Type	Min	Max
Factor of Safety1	Max von Mises Stress	3.815e+01 Node: 7575	8.101e+06 Node: 12902



Fork-Static 1-Factor of Safety-Factor of Safety1

Figure 45: Factor of Safety of Fork

The analysis performed to check the factor of safety of the fork is shown in Figure 45. The minimum FOS is 38, which shows that the design is safe.

4.2.2. Analysis of Fork Assembly:

4.2.2.1. Fork Support Frame static

Figure 46 shows that the forks were attached to carrier holding both forks, and the load was applied on the support assembly during the analysis.

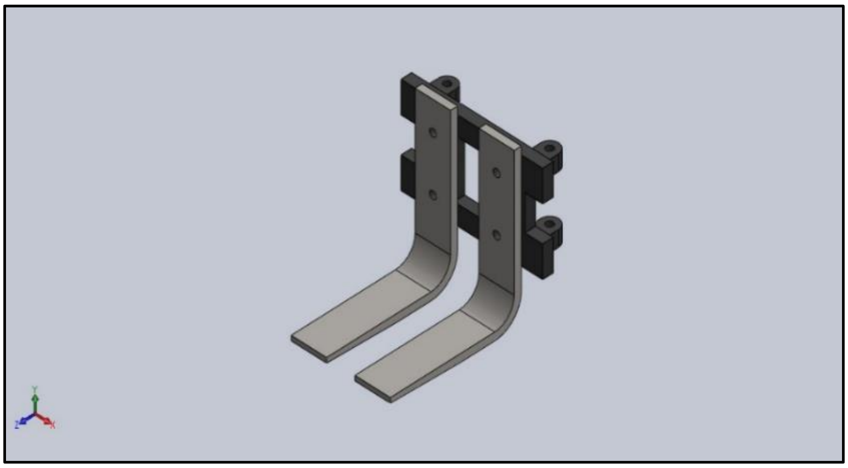
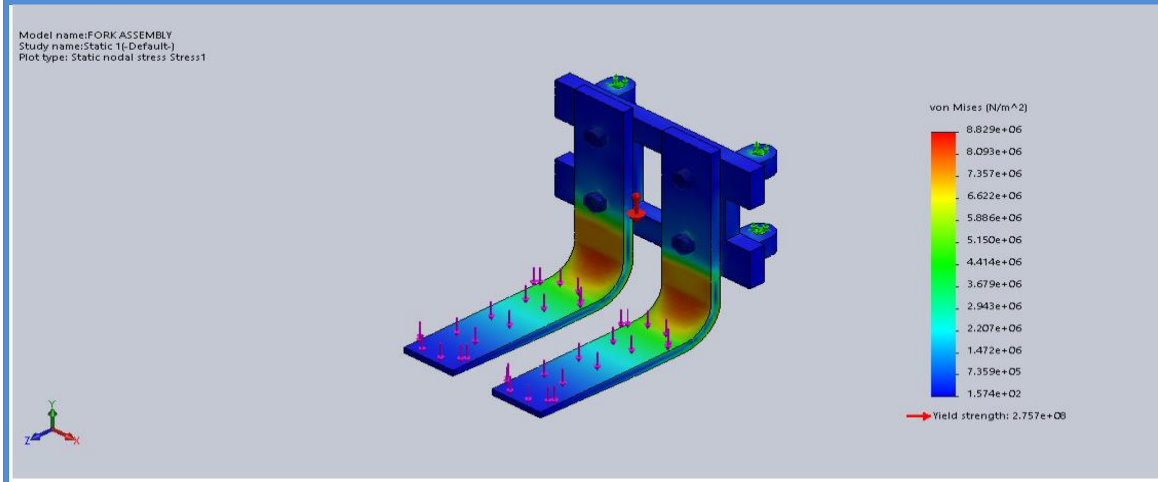


Figure 46: Support Frame

Figure 47, 48 and 49 show result of stress, strain and displacement, respectively. The maximum stress is in the curved path of the forks as shown by red color in figure 43 and has a value of 8.6 MPa. Maximum strain is observed at the same location in figure 44 and has magnitude of 3.31×10^5. Maximum displacement is at the free ends of the forks, which displace by 0.3 mm, as shown in Figure 49.

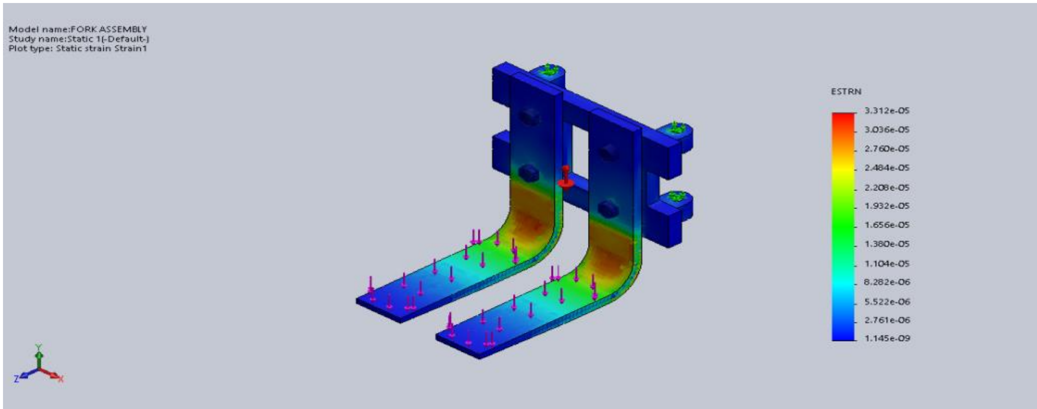
Name	Type	Min	Max
Stress1	VON: von Mises Stress	1.574e+02 N/m^2 Node: 47580	8.829e+06 N/m^2 Node: 71622



FORK ASSEMBLY-Static 1-Stress-Stress1

Figure 47: Fork Assembly Stress Analysis

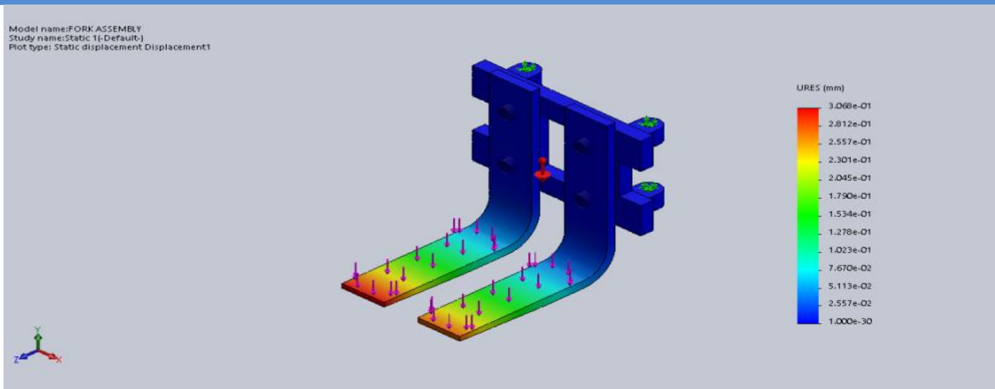
Name	Type	Min	Max
Strain1	ESTRN: Equivalent Strain	1.145e-09 Element: 51168	3.312e-05 Element: 37309



FORK ASSEMBLY-Static 1-Strain-Strain1

Figure 48: Fork Assembly Stress Analysis

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0.000e+00 mm Node: 2028	3.068e-01 mm Node: 91201



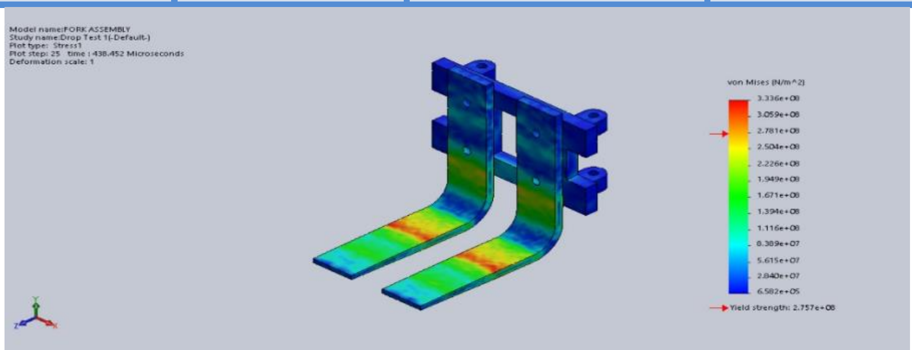
FORK ASSEMBLY-Static 1-Displacement-Displacement1

Figure 49: Fork Assembly Displacement Analysis

4.2.2.2. Fork Support Frame dynamic (Drop test)

This test is necessary to perform on the support frame assembly, as there are chances of a sudden drop out of the forks from the rail due to heavy load and movement of the fork on the rails. The analysis is performed by using the SolidWorks software, and the results obtained are shown in Figures 50, 51 and 52.

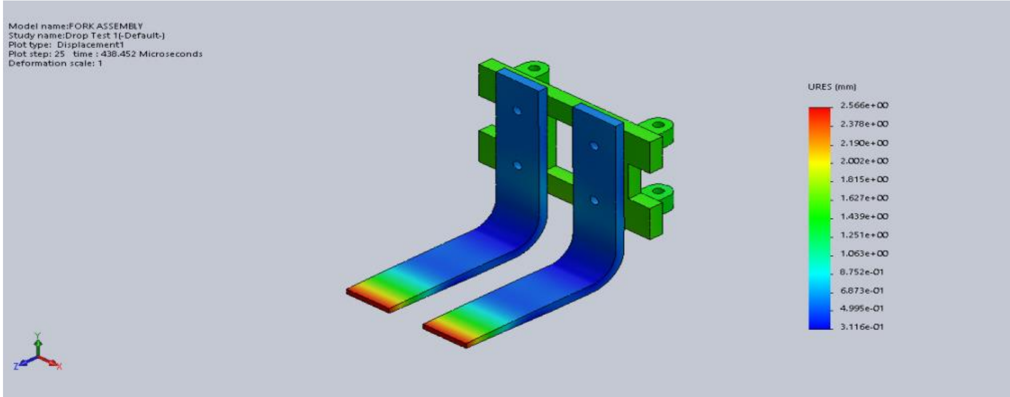
Name	Type	Min	Max
Stress1	VON: von Mises Stress	6.582e+05 N/m^2 Node: 30387	3.336e+08 N/m^2 Node: 47487



FORK ASSEMBLY-Drop Test 1-Stress-Stress1

Figure 50 : Fork Assembly Drop Test Stress

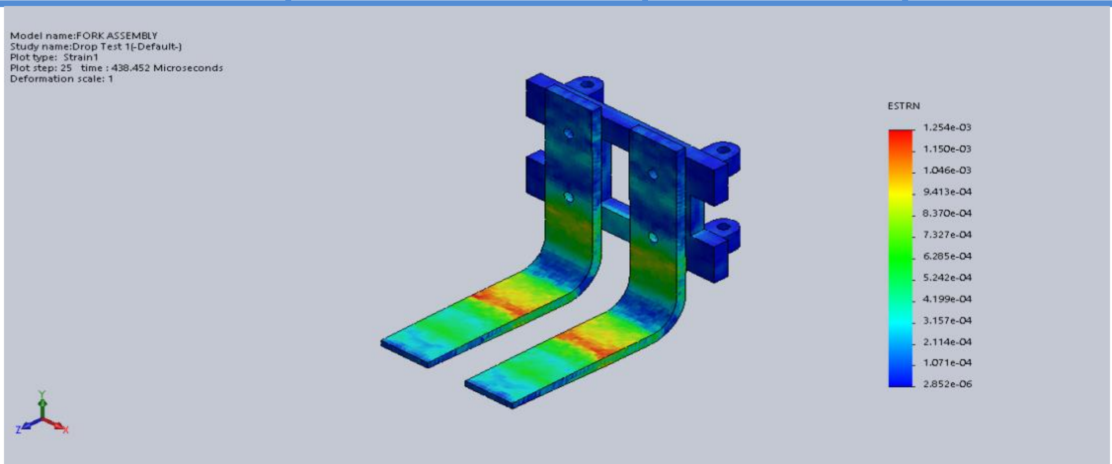
Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	3.116e-01 mm Node: 59269	2.566e+00 mm Node: 47352



FORK ASSEMBLY-Drop Test 1-Displacement-Displacement1

Figure 51: *Fork Assembly Drop Test Displacement*

Name	Type	Min	Max
Strain1	ESTRN: Equivalent Strain	2.852e-06 Element: 11742	1.254e-03 Element: 57471



FORK ASSEMBLY-Drop Test 1-Strain-Strain1

Figure 52: *Fork Assembly Drop Test Strain*

The above results give us the maximum values of stress, strain and displacement under the applied load during the drop test. Red portion faces the maximum stress.

4.3 Forks Vibration Analysis:

Forks are the part of the forklift that are highly susceptible to vibrations. The amplitude of vibration increases drastically upon resonance, therefore it is mandatory to ensure that resonance does not take place. According to the 1996 British Occupational Hygiene Society research dissertation “the most critical vibration for forklift truck is z-axis vibration with a frequency of no more than 20 Hz”. Thus, it was required to investigate the normal modes of vibration for forks, to ensure that we avoid resonance since it can lead to failure.

<u>Modes of Vibration for Forks</u>

Four modes of vibration for forks are depicted in Figure 53.

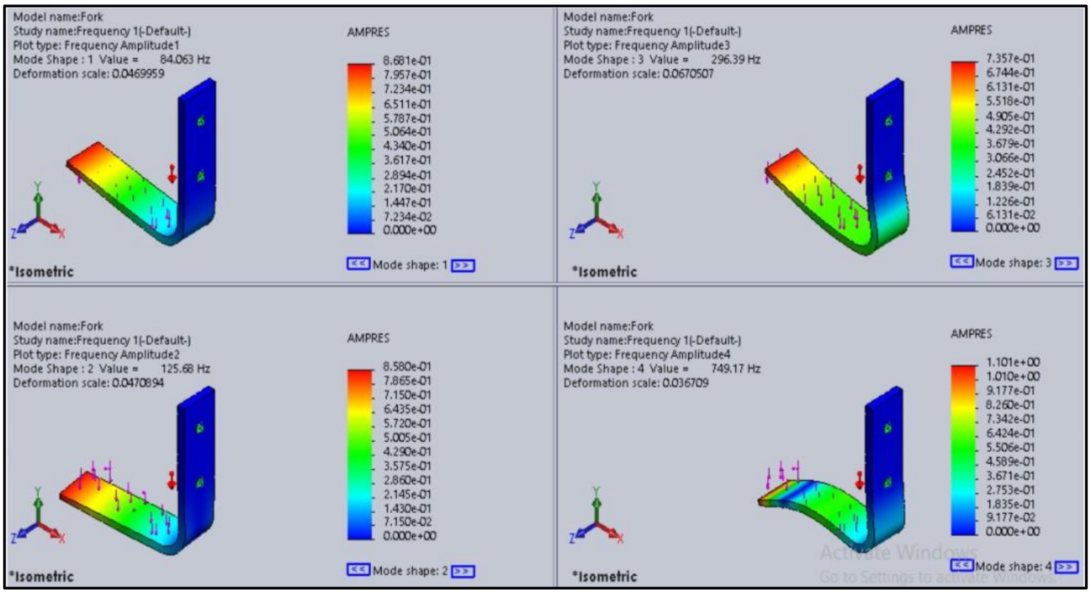


Figure 53: Modes of Vibration (Forks)

The table below summarizes the results obtained for the four modes of vibration.

Frequency Number	Rad/sec	Hertz	Seconds
1	528.18	84.063	0.011896
2	789.69	125.68	0.0079565
3	1862.3	296.39	0.0033739
4	4707.2	749.17	0.0013348

It is clear from the above table that the normal mode of vibration for the forks has a frequency of approximately 84 Hz which is well above the 20 Hz.

4.4. Finalized Design:

Following table summarizes all design parameters of the finalized forklift design.

GENERAL	1	**Operator Type**	Sit/Stand	The operator stands on the platform
	2	**Power Type/Mode**	Electric	Electric – 24 Volt
	3	**Capacity**	Rated Capacity (Q)	50 Kg
	4	**Load Centre**	Rated Load Centre (C)	0.2 m
	5	**Optional Capabilities**	Capability at 0.3 m Load Centre (Q_2)	40.91 Kg
	6	**Load Distance**	Front Axle Centre to the Fork Surface (X)	0.25 m

	7	**Model Configuration**	Wheel Base Designation	Small WB
	8	**Wheels**	Number, Front/Rear	2/1
	9	**Tire Type**	Front	Pneumatic, Recycled Thermoplastic Rubber
	10	**Tire Type**	Rear	Swivel caster, Polypropylene
	11	**Ground Clearance**	Centre of Wheelbase	115 mm
	12	**Turning Radius**	Minimum Turning Radius	0.75 m
	13	**Lifting Mechanism**	Rotating Chain Drive Mechanism	Roller chain with sprockets
	14	**Material**	Base Frame/Platform	Mild Steel
	15	**Material**	Support Frame/Vehicle Body Frame	Mild Steel
	16	**Material**	Forks	Mild Steel
DIMENSIONS	17	**Forks**	(Length x Width x Thickness	(0.4 x 0.1 x 0.015) m

	18	**Overall Dimensions**	Frame Width		0.75 m
	19		Base Width (with wheels)		1 m
	20		Frame Height		1.5 m
	21	**Wheel Base**			0.7 m
	22	**Track**			0.85 m
	23	**Tire Size**	Front	Diameter	0.4064 m
	24			Tread	0.1016 m
	25		Rear	Diameter	0.1524 m
	26			Tread	0.05 m
	27	**Lifting Chain**	Length		2.75 m
	28	**Drive Shaft (Front Wheels)**	Diameter		25 mm
WT.	29	**Counterweight**	Battery		35 kg
	30	**Total Weight**	Forklift		124 Kg

PERFORMANCE	31	**Maximum Travel Speed**	Loaded/Unloaded	0.5/0.7 m/s
	32	**Maximum Lifting Speed**	Loaded/Unloaded	0.2/0.3 m/s
	33	**Maximum Lowering Speed**	Loaded/Unloaded	0.35/0.3 m/s
	34	**Maximum Load Carrying Capacity**	Rated Capacity (Q)	50 Kg
	35	**Lift**	Maximum Lift Height	1.1 m
	36		Maximum Extended Height	1.2 m
	37	**Steering**		Perpendicular handle
	38	**Drive Motor**	Maximum Efficiency	45 %
ELECTRIC	39	**Battery**	Battery Type	Lead Acid, Rechargeable, Deep Cycle
	40		Volts / Max Ampere Hours	48V / 200 AH
	41		Weight	35 Kg

	41	**Electric Motors**	Drive/Traction Motor	Type	Brushed DC
	42			Model	LST-10
	43			Voltage	24 V
	44			Rating	250 W
	45			Rated Output Speed	120 rpm
	46			Number/Quantity	2
	47		Lifter/Hoist Motor	Type	Brushed DC
	48			Model	CL-5344
	49			Voltage	12 V
	50			Rating	100 W
	51			Rated Output Speed	80 rpm
	52			Number/Quantity	1

CHAPTER 5: CONCLUSIONS

Following are the main conclusions of this project;

1. A feasible design of small, medium load carrying electric forklift capable of working in tighter spaces was produced.
2. Determination of stability triangle and maintaining CG of the forklift within the stability triangle during operation is a mandatory requirement for successful design.
3. Maximum stress and strain occur in the bent section of the forks.
4. Maximum displacement of forks is at their free ends.
5. Natural frequency of the forks action should be above 20 hertz that was 84 hertz in current design.
6. Motion analysis provided validation of the finalized design by considering interaction of components.

Future Recommendations

In future, it is recommended to incorporate a System of Active Stability (SAS) into the design. The system will compose of a series of sensors that will detect instability by monitoring key forklift operations. When the system detects instability, the Active Control Rear Stabilizer locks the rear wheel, thereby ensuring lateral stability. Through the incorporation of this system in the design, prevention of lateral tipping overs is possible.

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