Design and Manufacturing a Service Lift For The Blue Nile Club Boats.

A thesis submitted in partial fulfillment of the requirements for the degree of B.Sc. in Mechanical Engineering

Presented by:

Esmael Alfadil Esmael. Salah Eldein Hussein.

Guttiba Abdalatif Alseid.

Supervised by:

Dr. Dina Mohamed Bilal.

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Dedication:

To my mother, father, brothers and sisters who suffered very much till I have finished successfully my task. To my teachers who lead and lighten my way through these very long studies by gathering very good vocabulary and education.
Abstract:

In order to lift or load the visitor’s goods, fuel and engines of the boats to the Blue Nile bank the workers need to exert extreme efforts in handling petroleum material and food and beverages, so from here the idea of the design and manufacture service lift came up.

The service lift main component are. Truss and a sliding car on the truss which connected to the truss by guide rails and the car is driven by a powered motor, a gearbox and a drum. The whole mechanism is controlled to establish safety operation and maximum efficiency.

All the mechanisms is designed and found out that it cost approximately 150,000SDG.
المستخلص:

تتلاخص المشكلة في عدم وجود خدمات لكل المرافق الترفيهية التي تقع على ضفاف النيل حيث يستخدم العمال في ذلك سلم، ونجد أن السلم لايستمر كثيرا حتي ينهار وكما ان الأمر فيه مضيعة للمزمن والجهد، ومن هنا اتت فكرة المشروع بانشاء مصعد خدمات للمساعدة في ذلك.

تم تصنيع المصعد وهو يتكون من جملون وب عربة تسير راسيا من اعلى الي اسفل والعكس حاملة الاغراض أو الوقود.
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List of symbols and abbreviations:

RPM: revolve per minute.

KN: kilo newton.

Q&T: Quenched and tempered.

SDG: Sudanese geneah. (= 0.147$ in bank/0.098$ in black market)
CHAPTER ONE
INTRODUCTION
1. **CHAPTER ONE: INTRODUCTION**

1.1 **Preface:**

The BLUE NILE is a river originating at Lake Tana in Ethiopia. With the White Nile, the river becomes one of the two major tributaries of the Nile. The upper course of the river is called the Abbay in Ethiopia, where many regard it as holy. Some Ethiopians have long identified the Blue Nile as the River Gihon mentioned as flowing out of the Garden of Eden in Genesis 2 and "encircling the entire land of Kush". (1)

According to materials published by the Ethiopian Central Statistical Agency, the Blue Nile has a total length of 1,450 kilometers (900 mil), of which 800 km (500 mil) are inside Ethiopia. The Blue Nile flows generally south from Lake Tana and then west across Ethiopia and northwest into Sudan. Within 30 kilometers (19 mil) of its source at Lake Tana, the river enters a canyon about 400 kilometers (250 mil) long. This gorge is a tremendous obstacle for travel and communication from the north half of Ethiopia to the southern half. The power of the Blue Nile may best be appreciated at Tis Issat Falls, which are 45 meters (148 ft) high, located about 40 kilometers (25 mil) downstream of Lake Tana. (1)

Although there are several feeder streams that flow into Lake Tana, the sacred source of the river is generally considered to be a small spring at Gish Abbai, situated at an altitude of approximately 2,744 meters (9,003 ft). This stream, known as the Lesser Abay, flows north into Lake Tana. Other affluents of this lake include, in clockwise order from Gorgora, the Magech, the Northern Gumara, the Reb, the Southern Gumara, and the Kilte. (2)

Lake Tana’s outflow then flows some 30 kilometers before plunging over the Tis Issat Falls. The river then loops across northwest Ethiopia through a series of deep valleys and canyons into Sudan, by which point it is only known as the Blue Nile. (2)

After flowing past Er Roseires inside Sudan, and receiving the Dinder on its right bank at Dinder, the Blue Nile joins the White Nile at Khartoum and, as the River Nile, flows through Egypt to the Mediterranean Sea at Alexandria.
The Blue Nile is so-called because during flood times, the water current is so high that the river turns almost black (in the local Sudanese language, the word for black is also used for the color blue).

The flow of the Blue Nile reaches maximum volume in the rainy season (from June to September), when it supplies about two thirds of the water of the Nile proper. The Blue Nile was a major source of the annual Nile floods that contributed to the fertility of the Nile Valley and the consequent rise of ancient Egyptian civilization and Egyptian Mythology. With the completion in 1970 of the Aswan High Dam in Egypt, the Nile floods ended. (2)

1.2 **Objective:**

The objective is to design a service lift at the Blue Nile bank for servicing boats.

Study the feasibility of the project.

1.3 **Methodology:**

- Previous studies.
- Collecting data by gathering from web sites and reliable books.
- Industrial and marketing survey to find optimum manufacture and suitable cost.
- The use of SOLID WORKS and AUTO CAD for detailed 2D and 3D drawing.

1.4 **Result expected:**

- Facilitating services at Blue Nile bank for servicing boats.
- Develop the Tourism in Sudan.
- Economic, safety and high quality design.
1.5 **Restriction to the study:**

- Designing problem due to complexity of design.
- Financial support to provide the desirable product.
- Difficulties in finding the data and limitations of resources.
- Confidentiality problem in companies.
CHAPTER TWO
LITERATURE REVIEW
2. CHAPTER TWO: LITERATURE REVIEW

2.1 Background:

2.1.1 Pre-industrial era:

The earliest known reference to an elevator is in the works of the Roman architect Vitruvius, who reported that Archimedes (c. 287 BC – c. 212 BC) built his first elevator probably in 236 BC. Some sources from later historical periods mention elevators as cabs on a hemp rope powered by hand or by animals. It is supposed that elevators of this type were installed in the Sinai monastery of Egypt. (3)

In 1000, the Book of Secrets by al-Muradi in Islamic Spain described the use of an elevator-like lifting device, in order to raise a large battering ram to destroy a fortress.

In the 17th century the prototypes of elevators were located in the palace buildings of England and France. Louis XV of France had a so-called 'flying chair' built for one of his mistresses at the Chateaude Versailles in 1743. (3)

Ancient and medieval elevators used drive systems based on hoists or winders. The invention of a system based on the screwdrive was perhaps the most important step in elevator technology since ancient times, leading to the creation of modern passenger elevators. The first screw drive elevator was built by IvanKulibin and installed in WinterPalace in 1793. Several years later another of Kulibin's elevators was installed in Arkhangels koye near Moscow. (3)

2.1.2 Industrial era:

The development of elevators was led by the need for movement of raw materials including coal and lumber from hillsides.

The technology developed by these industries and the introduction of steel beam construction worked together to provide the passenger and freight elevators in use today. (4)

Starting in the coal mines, by the mid-19th century elevators were operated with steam power and were used for moving goods in bulk in mines and factories.
These steam driven devices were soon being applied to a diverse set of purposes - in 1823, two architects working in London, Burton and Hormer, built and operated a novel tourist attraction, which they called the "ascending room". It elevated paying customers to a considerable height in the centre of London, allowing them a magnificent panoramic view of the city centre. (4)

Early, crude steam-driven elevators were refined in the ensuing decade; in 1835 an innovative elevator called the "Teagle" was developed by the company Frost and Stutt in England. The elevator was belt-driven and used a counterweight for extra power.

The hydraulic crane was invented by Sir William Armstrong in 1846, primarily for use at the Tyneside docks for loading cargo. These quickly supplanted the earlier steam driven elevators: exploiting Pascal's law, they provided a much greater force. A water pump supplied a variable level of water pressure to a plunger encased inside a vertical cylinder, allowing the level of the platform (carrying a heavy load) to be raised and lowered. Counterweights and balances were also used to increase the lifting power of the apparatus. (4)

Henry Waterman of New York is credited with inventing the "standing rope control" for an elevator in 1850.

In 1845, the Neapolitan architect Gaetano Genovese realized in the Royal Palace of Caserta the "Flying Chair", an elevator ahead of its time, covered with chestnut wood outside and with maple wood inside, accompanied by a light, two benches and hand signal to the plan, and can be activated from the outside, without any effort on the part of the occupants. The traction was ensured by a motor mechanic with a complex of toothed wheels, and was equipped with a system to prevent the consequences of the splitting of the strings: the jaggies in iron plates along the walls between which is inserted a beam would be placed under railroad the cab and pushed outwards by a spring system in steel. Elisha Otis demonstrating his safety system, Crystal Palace, 1853. (4)

In 1852, Elisha Otis introduced the safety elevator, which prevented the fall of the cab if the cable broke. The design of the Otis safety elevator is somewhat similar to one type still used today.

A governor device engages knurled roller(s), locking the elevator to its guides should the elevator descend at excessive speed. He demonstrated it at the New York exposition in the Crystal Palace in a dramatic, death-defying presentation in 1854, and the first such passenger elevator was installed at 488 Broadway in New York City on March 23, 1857.
Elisha Otis's elevator patent drawing, 15 January 1861.

The first elevator shaft preceded the first elevator by four years. Construction for Peter Cooper's Cooper Union Foundation building in New York began in 1853. An elevator shaft was included in the design, because Cooper was confident that a safe passenger elevator would soon be invented. The shaft was cylindrical because Cooper thought it was the most efficient design.

Later, Otis designed a special elevator for the building. Today the Otis Elevator Company, now a subsidiary of United Technologies Corporation, is the world's largest manufacturer of vertical transport systems.

The Equitable Life Building completed in 1870 in New York City was the first office building to have passenger elevators. (5)

The first electric elevator was built by Werner von Siemens in 1880 in Germany. The inventor Anton Freissler developed the ideas of von Siemens and built up a successful enterprise in Austria-Hungary. The safety and speed of electric elevators were significantly enhanced by Frank Sprague who added floor control, automatic elevators, acceleration control of cars, and safeties. His elevator ran faster and with larger loads than hydraulic or steam elevators, and 584 electric elevators were installed before Sprague sold his company to the Otis Elevator Company in 1895. Sprague also developed the idea and technology for multiple elevators in a single shaft.

In 1882, when hydraulic power was a well-established technology, a company later named the London Hydraulic Power Company was formed. It constructed a network of high-pressure mains on both sides of the Thames which, ultimately, extended to 184 miles and powered some 8,000 machines, predominantly elevators (lifts) and cranes.

In 1874, J.W. Meaker patented a method which permitted elevator doors to open and close safely. In 1887, American Inventor Alexander Miles of Duluth, Minnesota patented an elevator with automatic doors that would close off the elevator shaft. In 2000, the first vacuum elevator was offered commercially in Argentina. (6)

2.2 **Piles foundation:**

Pile foundation is required when the soil bearing capacity is not sufficient for the structure to withstand. This is due to the soil condition or the order of bottom layers, type of loads on foundations, conditions at site and operational conditions.
Many factors prevent the selection of surface foundation as a suitable foundation such as the nature of soil and intensity of loads, we use the piles when the soil have low bearing capacity or in building in water like bridges and dams.

A pile foundation consists of two components: Pile cap and single or group of piles. Piles transfers the loads from structures to the hard strata, rocks or soil with high bearing capacity. These are long and slender members whose length can be more than 15m.

Piles can be made from concrete, wood or steel depending on the requirements. These piles are then driven, drilled or jacked into the ground and connected to pile caps. Pile foundation are classified based on:

1- Material of pile construction.
2- Type of soil.
3- Load transmitting characteristic of piles.

As other types of foundations, the purpose of pile foundations is:

– To transmit the buildings loads to the foundations and the ground soil layers whether these loads vertical or inclined.

– To install loose cohesion less soil through displacement and vibration.

– To control the settlements; which can be accompanied by surface foundations.

– To increase the factor of safety for heavy loads buildings.

The selection of type of pile foundation is based on site investigation report. Site investigation report suggests the need of pile foundation,

Type of pile foundation to be used, and depth of pile foundation to be provided. The cost analysis of various options for use of pile foundation should be carried out before selection of pile foundation types.
Began Robert Ross Industrial Disposal because he saw an opportunity to meet the hazardous waste management needs of companies in northern Ohio. In 1958, the company built one of the first hazardous waste incinerators in the U.S. The first full-scale, municipally operated incineration facility in the U.S. was the Arnold O. Chant land Resource Recovery Plant, built in 1975 and located in Ames, Iowa. This plant is still in operation and produces refuse-derived fuel that is sent to local power plants for fuel. The first commercially successful incineration plant in the U.S. was built in Saugus, Massachusetts in October 1975 by Wheelabrator Technologies, and is still in operation today.

There are several environmental or waste management corporations that transport ultimately to an incinerator or cement kiln treatment center. Currently (2009), there are three main businesses that incinerate waste: Clean Harbors, WTI-Heritage, and Ross Incineration Services. Clean Harbors has acquired many of the smaller, independently run facilities, accumulating 5–7 incinerators in the process across the U.S. WTI-Heritage has one incinerator, located in the southeastern corner of Ohio (across the Ohio River from West Virginia).

Several old generation incinerators have been closed; of the 186 MSW incinerators in 1990, only 89 remained by 2007, and of the 6200 medical waste incinerators in 1988, only 115 remained in 2003. No new incinerators were built between 1996 and 2007. The main reasons for lack of activity have been:

There has been renewed interest in incineration and other waste-to-energy technologies in the U.S. and Canada. In the U.S., incineration was granted qualification for renewable energy production tax credits in 2004. Projects to add capacity to existing plants are underway, and municipalities are once again evaluating the option of building incineration plants rather than continue land filling municipal wastes.

However, many of these projects have faced continued political opposition in spite of renewed arguments for the greenhouse gas benefits of incineration and improved air pollution control and ash recycling. (7)
CHAPTER THREE
DESIGN
3. CHAPTER THREE: DESIGN

3.1 Design of gearbox:
See Appendix C Fig 8.

3.1.1 Gear sets:
Gears are toothed cylindrical wheel used for transmitting mechanical power from one rotating shaft to another.

Spur Gears
First of all we should know the typical material used for gears and pinion:

Table 1: pinion and gear material:

<table>
<thead>
<tr>
<th>Gear Material</th>
<th>Pinion Material</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cast iron</td>
<td>Cast iron</td>
</tr>
<tr>
<td>Cast iron</td>
<td>Carbon steel</td>
</tr>
<tr>
<td>Cast iron</td>
<td>Alloy steel</td>
</tr>
<tr>
<td>Alloy steel</td>
<td>Alloy steel</td>
</tr>
<tr>
<td>Alloy steel</td>
<td>Case Hardened steel</td>
</tr>
</tbody>
</table>

Table 2; Useful Ranges of Gears Ratios:

<table>
<thead>
<tr>
<th>Gear</th>
<th>Ratio range</th>
<th>Pitch linear velocity</th>
<th>Efficiency</th>
</tr>
</thead>
<tbody>
<tr>
<td>Spur</td>
<td>1:1 - 1:5</td>
<td>25 m/s</td>
<td>98%-99%</td>
</tr>
</tbody>
</table>
Set 1:

Power transmitted = 2926.76 Kw

Pinion speed = 1440 r.p.m.

Reduction value (i) = 4.5

Assumption:

Table 3; Material to be used:

<table>
<thead>
<tr>
<th>Pinion</th>
<th>Gear</th>
</tr>
</thead>
<tbody>
<tr>
<td>Non-alloy steel</td>
<td>Non-alloy steel</td>
</tr>
<tr>
<td>Plain low-carbon steel</td>
<td>Plain low-carbon steel</td>
</tr>
</tbody>
</table>

C1030 Oil-Quenched and tempered plain carbon steel mainly classified as high-carbon steel.

From Table 4 table of materials:

<table>
<thead>
<tr>
<th>Material</th>
<th>Yield strength</th>
<th>BHN</th>
</tr>
</thead>
<tbody>
<tr>
<td>C1030 Q&amp;T</td>
<td>84 ksi</td>
<td>302</td>
</tr>
</tbody>
</table>

Sum of BHN (as we used the same material for pinion & gear) = 285+285 = 570

From table: BHN = 604 (equivalent K = 2.756MPa @ $\phi = 20^\circ$)

Assumption:

$\phi = 20^\circ$ Full depth involute

Module (m) = 2 mm

Pinion teeth $Z_p = 16$

Equivalent Lewis factor ($Y$) = 0.292

Face width ($b$) = $(3-4)\pi m$
\[ = 3\pi \times 2 = 18.85 \text{ mm (say 19)} \]

\[ Z_g = Z_p \times i = 72 \text{ (say 72)} \]

Diameters:

\[ D_p = m \times Z_p = 2 \times 16 = 32 \text{ mm} \]

\[ D_g = m \times Z_g = 2 \times 72 = 144 \text{ mm} \]

\[ \omega = \frac{2\pi N}{60} = \frac{2\pi \times 1440}{60} = 150.8 \text{ rad/s} \]

\[ T = \frac{\text{Power}}{\omega} = \frac{2926.76}{150.8} = 19.4 \text{ N.m} \]

Now we should test pinion strength:

Mainly, gears experience two principle types of stress bending stress at the root of the teeth due to transmitted load and contact stresses on the flank of the teeth due to repeated impact.

**First: Static check (Lewis check):**

\[ F_s = \sigma_a \times m \times b \times y = \frac{\text{yield stress} \times 6.89 \times m \times b \times y}{\text{safety factor}} = \frac{84 \times 6.89 \times 2 \times 0.292 \times 19}{3} \]

\[ = 2140.64 \text{ N} \]

(Safety factor is assumed to be 3)
\[ F_t = \frac{\text{Power}}{\text{velocity}} \quad , \quad v = \frac{\omega \cdot D}{2} = \frac{150.8 \cdot 32}{2 \cdot 1000} = 2.4128 \, m/s \]

\[ F_t = \frac{2926.76}{2.4128} = 1213.04 N \]

\[ 1.25 < \frac{F_s}{F_t} = \frac{2140.64}{1213.04} = 1.76 < 1.85 \]

\[ : \text{The set is statically safe, thus no bending or shear failure will occur.} \]

**Second: Dynamic check (wear check):**

As well as failure occurs due to bending stresses in gears, the failure also can occur due to wear on the surface of gear and it should be considered:

\[ Q = \frac{2i}{1 + i} = \frac{2 \cdot 4.5}{1 + 4.5} = 1.636 \]

\[ c_v = \frac{6 + v}{6} \quad (\text{for steel}) \]

\[ c_v = \frac{6 + 2.4128}{6} = 1.402 \]

\[ F_d = c_v \cdot F_t = 1.402 \cdot 1213.04 = 1700.68 \, N \]
\[ F_w = k \times Q \times D_p \times b = 2.756 \times 1.636 \times 32 \times 19 = 2741.36 \text{ N} \]

\[
1.25 < \frac{F_w}{F_d} = \frac{2741.36}{1700.68} = 1.61 < 1.85
\]

\[ \therefore \text{The set will resist wear and it would be safe.} \]

**Summary:**

Table 5; summary of set1:

<table>
<thead>
<tr>
<th>Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Module</td>
<td>2mm</td>
</tr>
<tr>
<td>Reduction ratio</td>
<td>4.5</td>
</tr>
<tr>
<td>Pinion teeth</td>
<td>16</td>
</tr>
<tr>
<td>Gear teeth</td>
<td>72</td>
</tr>
<tr>
<td>Pinion diameter</td>
<td>32 mm</td>
</tr>
<tr>
<td>Gear diameter</td>
<td>144 mm</td>
</tr>
<tr>
<td>Power</td>
<td>2926.76Watt</td>
</tr>
<tr>
<td>N</td>
<td>1440 rpm</td>
</tr>
<tr>
<td>Pinion addendum</td>
<td>36 mm</td>
</tr>
<tr>
<td>Gear addendum</td>
<td>148mm</td>
</tr>
<tr>
<td>Pressure angle ((\varphi))</td>
<td>20°</td>
</tr>
<tr>
<td>Set width</td>
<td>19 mm</td>
</tr>
<tr>
<td>Check 1</td>
<td>1.76</td>
</tr>
<tr>
<td>Check 2</td>
<td>1.61</td>
</tr>
</tbody>
</table>
Set 2:

Assumption:

\[ i = 4.00 \]

\[ N = 320 \text{ rpm} \]

From table of materials:

<table>
<thead>
<tr>
<th>Material</th>
<th>Yield strength</th>
<th>BHN</th>
<th>K (wear factor)</th>
</tr>
</thead>
<tbody>
<tr>
<td>C1030 Q&amp;T</td>
<td>84 ksi</td>
<td>302</td>
<td>2.756</td>
</tr>
</tbody>
</table>

Assumption:

\[ \varphi = 20^\circ \text{ full depth involute} \]

Module (m) = 2.75 mm

Pinion teeth \( Z_p = 17 \)

Face width \( b = (3-4)\pi m \)

\[ = 3.5 \times \pi \times 2.75 = 30.23 \text{ mm} \ (\text{say 31}) \]

\[ Z_g = Z_p \times i = 17 \times 4 = 68 \]

Diameters:

\[ D_p = m \times Z_p = 2.75 \times 17 = 47 \text{ mm} \]

\[ D_g = m \times Z_g = 2.75 \times 68 = 188 \text{ mm} \]

\[ \omega = \frac{2\pi N}{60} = \frac{2\pi \times 320}{60} = 33.51 \text{ rad/s} \]
First: Static check (Lewis check):

# assume safety factor = 3

\[
F_s = \sigma_d \times m \times b \times y = \frac{\text{yield stress} \times 6.89 \times m \times b \times y}{\text{safety factor}} = \frac{84 \times 6.89 \times 2.75 \times 0.302 \times 31}{3} = 4966.822 \text{ N}
\]

\[
F_t = \frac{\text{Power}}{\text{velocity}}, \quad v = \frac{\omega \times D}{2} = \frac{33.51 \times 47}{2 \times 1000} = 0.787 \text{ m/s}
\]

\[
F_t = \frac{2926.76}{0.787} \times 0.98 = 3644.5 \text{ N} \quad \text{(The efficiency of spur gear = 0.98)}
\]

\[
1.25 < \frac{F_s}{F_t} = \frac{4966.822}{3644.5} = 1.36 < 1.85
\]

\[\therefore \text{ The set is statically safe, thus no bending or shear failure will occur.}\]

Second: Dynamic check (wear check):

As well as failure occurs due to bending stresses in gears, the failure also can occur due to wear on the surface of gear and it should be considered:
\[ Q = \frac{2i}{1 + i} = \frac{2 \times 4}{1 + 4} = 1.6 \]

\[ c_{v} = \frac{6 + v}{6} \]

\[ c_{v} = \frac{6 + 0.787}{6} = 1.131 \]

\[ F_{d} = c_{r} \times F_{t} = 1.131 \times 3644.5 = 4122.53 \text{ N} \]

\[ F_{w} = k \times Q \times D_{p} \times b = 2.756 \times 1.6 \times 47 \times 31 = 6424.787 \text{ N} \]

\[ 1.25 < \frac{F_{w}}{F_{d}} = \frac{6424.787}{4122.53} = 1.56 < 1.85 \]

\[ \therefore \text{The set will resist wear and it would be safe} \]

**Summary:**

Table 6; summary of set 2

<table>
<thead>
<tr>
<th>Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Module</td>
<td>2.75 mm</td>
</tr>
<tr>
<td>Reduction ratio</td>
<td>4</td>
</tr>
<tr>
<td>Pinion teeth</td>
<td>17</td>
</tr>
<tr>
<td>Gear teeth</td>
<td>68</td>
</tr>
<tr>
<td>Pinion diameter</td>
<td>47 mm</td>
</tr>
</tbody>
</table>

19
### Description

<table>
<thead>
<tr>
<th>Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gear diameter</td>
<td>188 mm</td>
</tr>
<tr>
<td>Power input*</td>
<td>2868.225 Watt</td>
</tr>
<tr>
<td>N input</td>
<td>360 r.p.m.</td>
</tr>
<tr>
<td>Pressure angle ($\varphi$)</td>
<td>20º</td>
</tr>
<tr>
<td>Set width</td>
<td>31 mm</td>
</tr>
<tr>
<td>Check 1</td>
<td>1.36</td>
</tr>
<tr>
<td>Check 2</td>
<td>1.56</td>
</tr>
</tbody>
</table>

* Power input differs from the main input power (2926.76 W) due to friction losses which occur throw each set of gears

# The efficiency of spur gear is 0.98.

**Set 3:**

**Assumption:**

$$i = 4$$

$$N = 90 \text{ rpm}$$

From table of materials:

<table>
<thead>
<tr>
<th>Material</th>
<th>Yield strength</th>
<th>BHN</th>
<th>K (wear factor)</th>
</tr>
</thead>
<tbody>
<tr>
<td>C1030</td>
<td>84ksi</td>
<td>302</td>
<td>2.756</td>
</tr>
<tr>
<td>Q&amp;T</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

**Assumption:**

$$\varphi = 20^\circ \text{ Full depth involute}$$
Module (m) = 4 mm

Pinion teeth $Z_p = 19$

Face width (b) = (3-4)$\pi m$

$$= 3.7 \times \pi \times 4 = 46.4955 \text{ mm (say 47)}$$

$$Z_g = Z_p \times i = 19 \times 4 = 76$$

Diameters:

$$D_p = m \times Z_p = 4 \times 19 = 76 \text{ mm}$$

$$D_g = m \times Z_g = 4 \times 76 = 304 \text{ mm}$$

$$\omega = \frac{2\pi N}{60} = \frac{2\pi \times 80}{60} = 8.377 \text{ rad/s}$$

$$T = \frac{Power}{\omega} = \frac{2926.76 \times 0.98 \times 0.98}{8.377} = 335.545 N.m$$

**First: Static check (Lewis check):**

# assume safety factor = 3
\[ F_s = \sigma_d \cdot m \cdot b \cdot y = \frac{\text{yield stress} \cdot 6.89 \cdot m \cdot b \cdot y}{\text{safety factor}} = \frac{84 \cdot 6.89 \cdot 4 \cdot 47 \cdot 0.314}{3} = 11388.45 \, N \]

\[ F_t = \frac{\text{Power}}{\text{velocity}}, \quad v = \frac{\omega \cdot D}{2} = \frac{8.377 \cdot 76}{2 \cdot 1000} = 0.318 \, m/s \]

\[ F_t = \frac{2926.76}{0.318} \cdot 0.98^2 = 8839.18 \, N \]

\[ 1.25 < \frac{F_s}{F_t} = \frac{11388.45}{8839.18} = 1.3 < 1.85 \]

\[ \therefore \text{The set is statically safe, thus no bending or shear failure will occur.} \]

**Second: Dynamic check (wear check):**

\[ Q = \frac{2i}{1+i} = \frac{2 \cdot 4}{1+4} = 1.6 \]

\[ c_v = \frac{6 + \nu}{6} \]

\[ c_v = \frac{6 + 0.318}{6} = 1.053 \]

\[ F_d = c_r \cdot F_t = 1.053 \cdot 8839.18 = 9307.65 \, N \]
\[ F_w = k \cdot Q \cdot D_p \cdot b = 2.756 \cdot 1.6 \cdot 47 \cdot 76 = 15751.09 \, N \]

\[
1.25 < \frac{F_w}{F_d} = \frac{15751.09}{9307.65} = 1.69 < 1.85
\]

\[ \therefore \text{The set will resist wear and it would be safe.} \]

**Summary:**

Table 7; summary of set 3:

<table>
<thead>
<tr>
<th>Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Module</td>
<td>4 mm</td>
</tr>
<tr>
<td>Reduction ratio</td>
<td>4</td>
</tr>
<tr>
<td>Pinion teeth</td>
<td>19</td>
</tr>
<tr>
<td>Gear teeth</td>
<td>76</td>
</tr>
<tr>
<td>Pinion diameter</td>
<td>76 mm</td>
</tr>
<tr>
<td>Gear diameter</td>
<td>304 mm</td>
</tr>
<tr>
<td>Power input*</td>
<td>2810.86 Watt</td>
</tr>
<tr>
<td>( N_{in} )</td>
<td>80 rpm</td>
</tr>
<tr>
<td>( N_{out} )</td>
<td>20 rpm</td>
</tr>
<tr>
<td>Pressure angle (( \phi ))</td>
<td>20°</td>
</tr>
<tr>
<td>Set width</td>
<td>70 mm</td>
</tr>
<tr>
<td>Check 1</td>
<td>1.3</td>
</tr>
<tr>
<td>Check 2</td>
<td>1.69</td>
</tr>
</tbody>
</table>
3.1.2 Shaft design:

Shaft is a rotating machine element which is used to transmit power from one place to another. Shafts are usually cylindrical and it may be hollow.

Stresses in shaft:

1. Shear stress.
2. Bending stress.
3. Stress due to combined torsional and bending loads.

Table 8; shafts length:

<table>
<thead>
<tr>
<th>No.</th>
<th>Shaft</th>
<th>Length</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Input</td>
<td>( \frac{b_1}{2} + \Delta + \frac{\text{bearing width}}{2} = \frac{20}{2} + 2 + \frac{19}{2} = 21.5 \text{ mm} )</td>
</tr>
<tr>
<td>2</td>
<td>1(^{st}) intermediate</td>
<td>( b_1 + b_2 + b_3 + 4 \times \Delta + \text{bearing width} = 19 + 31 + 47 + 4 \times 2 + 20 = 125 \text{ mm} )</td>
</tr>
<tr>
<td>3</td>
<td>2(^{nd}) intermediate</td>
<td>( b_1 + b_2 + b_3 + 4 \times \Delta + \text{bearing width} = 19 + 31 + 47 + 4 \times 1 + 20 = 125 \text{ mm} )</td>
</tr>
<tr>
<td>4</td>
<td>Output</td>
<td>Intermediate shaft length + ( \Delta + \frac{\text{chain sprocket width}}{2} )</td>
</tr>
</tbody>
</table>

1. Input shaft:

Assumptions:

Safety factor = 3

\( k_t = 1.5 \) (profile key way)

\( k_b = 1.5 \) (no sharp ends)
Material: C1030 normalized

\[ \sigma_{yield} = 358 \text{ MPa}. \]

\[ \tau_{max} = \frac{\sigma_{yield}}{2} \times 0.75 \times 6.894 = 59.66 \times 10^6 \text{ MPa} \]

* (multiplied by 0.75 because we are going to use keys)

Horizontal analysis:

\[ F_t = 1213.04 \text{ N} \quad \text{(From set 1)} \]

\[ \Sigma_{\text{horizontal forces}} = 0 \]

\[ RH = F_t = 1213.04 \text{N} \]

\[ M_{v_{\max}} = \frac{F_t \times 21.5}{1} = \frac{1213.04 \times 21.5}{1} = 26080.36 \text{ N.mm} \]

\[ M_{\text{torsional}} = 19.4 \text{ N.m} \]

Vertical analysis:

\[ F_r = F_t \times \tan\phi = 1213.04 \times \tan20 = 441.51 \text{ N} \]

\[ \Sigma_{\text{vertical forces}} = 0 \]

\[ RV = F_r = 441.51 \text{ N} \]

\[ M_{v_{\max}} = 21.5 \times 441.5 = 9492.25 \text{ N.mm} \]

Resultant Bending Moment = \[\sqrt{(M_{v_{\max}})^2 + (M_{h\max})^2} = \sqrt{26080.36^2 + 9492.25^2} \]

\[ = 27754.1 \text{ N.mm} \]
\[ M_t = 19.4 \, N.m \]

\[ d^3 = \frac{16}{\pi \cdot \tau_{max}} \sqrt{(k_b M_b)^2 + (k_t M_t)^2} \]

\[ = \frac{16}{\pi \cdot 59.66 \cdot 10^6} \sqrt{(1.5 \cdot 27.75)^2 + (1.5 \cdot 19.4)^2} \]

\[ \therefore d = 16.3\,mm = 17mm \ * \]

**Summary:**

Table 9; summary of input shaft:

<table>
<thead>
<tr>
<th>Shaft</th>
<th>Length (mm)</th>
<th>Diameter (mm)</th>
<th>Material</th>
<th>BHN</th>
</tr>
</thead>
<tbody>
<tr>
<td>Input</td>
<td>21.5</td>
<td>17</td>
<td>C1030</td>
<td>179</td>
</tr>
</tbody>
</table>

**2. First intermediate shaft:**

**Assumptions:**

\textit{Safety factor} = 3

\[ k_t = 1.5 \, (\text{profile key way}) \]

\[ k_b = 1.5 \, (\text{no sharp ends}) \]

**Material:** C1050

\[ \sigma_{yield} = 358 \, Mpa \]
\[
\tau_{\text{max}} = \frac{\sigma_{\text{yield}}}{2} \cdot \frac{0.75}{3} \cdot 6.894 = 59.66 \cdot 10^6 \text{MPa}
\]

Horizontal analysis:

\[
\Sigma_{\text{forces}} = 0
\]

\[
RH_1 + RH_2 = 3227.504 + 1617.576 = 4845.08 \text{ N}
\]

\[
\Sigma M_{RH_1} = 0
\]

\[
RH_2 \cdot 125 = 1213.04 \cdot 21.5 \cdot 3642.2 \cdot 48.5
\]

\[\Rightarrow RH_2 = 1621.8 \text{ N}\]

\[\therefore RH_1 = 4855.24 - 1621.8 = 3233.4 \text{ N}\]

\[
BM_{H_{@Ft1}} = RH_2 \cdot 112.5 = 4305.41 \cdot 112.5 = 484358.86 \text{ N.m}
\]

\[
BM_{H_{@Ft2}} = RH_1 \cdot 32 = 8261.3 \cdot 32 = 264361.53 \text{ N.m}
\]

Vertical analysis:
\[ F_{r_2} = F_{t_2} \times \tan \phi = 9481.82 \times \tan 20 = 3451.1 \, N \]

\[ F_{r_1} = 3084.89 \times \tan 20 = 1122.8 \, N \]

\[
\Sigma \text{forces} = 0
\]

\[ RV_2 - RV_1 = 884.2 \, N \]

\[
\Sigma RV_1 = 0
\]

\[ RV_2 \times 125 + 441.5 \times 21.5 = 1325.7 \times 48.5 \]

\[ \Rightarrow RV_2 = 438.4 \, N \]

\[ \therefore RV_1 = -445.76N \]

\[ M_{v_{\text{max}}} = 124068.9 \, N.mm \]

\[ M_{h_{\text{max}}} = 33541.8 \, N.mm \]

\[
\text{Resultant Bending Moment} = \sqrt{(M_{v_{\text{max}}})^2 + (M_{h_{\text{max}}})^2} = \sqrt{124068.9^2 + 33541.8^2}
\]

\[ = 128522.9 \, N.mm = 128.5 \, N.m \]

\[ M_t = 87.34 \, N.m \]
\[ d^3 = \frac{16}{\pi \times 59.66 \times 10^6} \sqrt{(1.5 \times 128.5)^2 + (1.5 \times 87.34)^2} \]

\[ d = 27.1 \text{ mm} \]

Summary:

Table 10; summary of 1\textsuperscript{st} intermediate shaft:

<table>
<thead>
<tr>
<th>Shaft</th>
<th>Length (mm)</th>
<th>Diameter (mm)</th>
<th>Material</th>
<th>BHN</th>
</tr>
</thead>
<tbody>
<tr>
<td>1\textsuperscript{st} intermediate</td>
<td>125</td>
<td>27.1</td>
<td>C1030</td>
<td>179</td>
</tr>
</tbody>
</table>

3. Second intermediate shaft:

Assumptions:

Safety factor = 3

\[ k_t = 1.5 \text{ (profile key way)} \]

\[ k_b = 1.5 \text{ (no sharp ends)} \]

Material: C1030

\[ \sigma_{yield} = 358 \text{ MPa} \]

\[ \tau_{max} = \frac{\sigma_{yield}}{2} \times \frac{0.75}{3} \times 6.894 = 59.66 \times 10^6 \text{ MPa} \]

Horizontal analysis:

\[ \Sigma_{forces} = 0 \]

\[ RH_1 + RH_2 = 12481.4 \text{ N} \]
\[ \sum M_{RH_1} = 0 \]

\[ RH_2 \times 125 = 8839.18 \times 48.5 + 3642.2 \times 21.5 \]

\[ \Rightarrow RH_2 = 4056.1 N \]

\[ RH_1 = 8425.3 N \]

\[ BM_{H_{max}} = 31.287.7 N.m \]

**Vertical analysis:**

\[ F_{r_2} = 8839.18 \times tan \ 20 = 3217.2 \ N \]

\[ F_{r_3} = 22359.31 \times tan \ 20 = 8138.12 \ N \]

\[ \Sigma forces = 0 \]

\[ RV_2 + RV_1 = 3217.2 - 1325.7 = 1891.5 \ N \]

\[ \Sigma RV_1 = 0 \]

\[ RV_2 \times 125 + 1325.7 \times 21.5 = 3217.2 \times 48.5 \]

\[ \Rightarrow RV_2 = 1020.3 \ N \]

\[ \therefore RV_1 = 871.2 \ N \]
\[ M_{v_{\text{max}}} = 78047.1 \text{ N.mm} \]

\[
\text{Resultant Bending Moment} = \sqrt{(M_{v_{\text{max}}})^2 + (M_{h_{\text{max}}})^2} = \sqrt{31.287.7^2 + 78047.1^2} = 319952.8 \text{ N.mm} = 319.95 \text{ N.m}
\]

\[ M_t = 335.4 \text{ N.m} \]

\[
d^3 = \frac{16}{\pi \times 59.66 \times 10^6} \sqrt{(1.5 \times 319.95)^2 + (1.5 \times 335.4)^2}
\]

\[ d = 39 \text{ mm} \]

**Summary:**

Table 11; summary of 2nd intermediate:

<table>
<thead>
<tr>
<th>Shaft</th>
<th>Length (mm)</th>
<th>Diameter (mm)</th>
<th>Material</th>
<th>BHN</th>
</tr>
</thead>
<tbody>
<tr>
<td>2(^{nd}) intermediate</td>
<td>125</td>
<td>39</td>
<td>C1030</td>
<td>179</td>
</tr>
</tbody>
</table>

4. Output shaft:
speed = 20 rpm

\[ F_t = 8839.18N \]

\[ Shaft\ length = 10 + 2 + \frac{47}{2} = 35.5\ mm \]

Horizontal analysis:

\[ RH = 8839.18N \]

\[ M_{H_{\text{max}}} = 8839.18 \times 35.5 = 313790.89\ N.mm \]

Vertical analysis:

\[ F_{r_3} = F_{r_3} \times \tan \phi = 8839.18 \times \tan 20 = 3217.2N \]

\[ RV = 3217.2\ N \]

\[ M_{v_{\text{max}}} = 3217.2 \times 35.5 = 114210.6\ N.mm \]

\[ \text{Resultant Bending Moment} = \sqrt{(M_{v_{\text{max}}})^2 + (M_{h_{\text{max}}})^2} \]

\[ = \sqrt{114210.6^2 + 313790.98^2} = 333929.3\ N.mm \]
\[ M_t = 1315.22 \text{ N.m} \]

\[ n \]

\[ d^3 = \frac{16}{\pi \times 59.66 \times 10^6 \sqrt{(1.5 \times 333.929)^2 + (1.5 \times 1315.22)^2}} \]

\[ d = 55.8 \text{ mm} \]

Summary:

Table 12; summary of output shaft:

<table>
<thead>
<tr>
<th>Shaft</th>
<th>Length (mm)</th>
<th>Diameter (mm)</th>
<th>Material</th>
<th>BHN</th>
</tr>
</thead>
<tbody>
<tr>
<td>Output</td>
<td>35.5</td>
<td>55.8</td>
<td>C1030</td>
<td>179</td>
</tr>
</tbody>
</table>

3.1.3 Bearing Selection:

Bearing is used to refer the assembly formed by two surfaces that have the capacity for relative motion.
1. **Bearing selection @input shaft:**

   **Assumption:**

   **Life time (L):** 3 year, 12 hours/day

   \[ N = 1440 \text{ rpm} \]

   \[ L_{\text{per million rev.}} = \frac{3 \text{ year} \times 365 \text{ days} \times 12 \times 1440 \times 60}{10^6} = 1135.3 \text{ hours} \]

   **Bearing type:**

   Single-row deep groove ball bearing open type manufactured by Koyo.

   \[ P_{\text{LOAD}} = X \times F_A + Y \times F_R \]

   Where:

   \[ F_A \equiv \text{Axial force} \]

   \[ F_R \equiv \text{Radial force} \]

   *Note:*

   Since we have used spur gear, thus there is no axial force & if it exist, it will be too small with no effect so can be neglected.

   \[ P = \sqrt{441.51^2 + 1213.04^2} = 1290.9 \text{ N} \]
\[ c = P(L)^{\frac{1}{K}} \]

For deep groove:

\[ K = 3 \]

\[ C = 1290.9(1135.3)^{\frac{1}{3}} = 13466.7 \text{ N} \quad \text{say 13.5 kN} \]

\[ C = 13.5 \text{ kN} \]

→From bearings table:

Bearing no. = 6203

d = 17

B = 12mm

\[ D = 40 \text{ mm} \]

\[ C_r = 9.55 \text{ kN} \]

We choose double bearing

2. Bearing selection @ 1st intermediate shaft:

\[ P_1 = \sqrt{RH_1^2 + RV_1^2} = \sqrt{445.8^2 + 3233.4^2} = 3264 \text{ N} \]

\[ P_2 = \sqrt{RH_2^2 + RV_2^2} = \sqrt{1621.8^2 + 438.4^2} = 1680 \text{ N} \]
\[ c_{R_1} = 3264(1135.3)^{\frac{1}{3}} = 34050.25 N \]

\[ c_{R_2} = 1680(1135.3)^{\frac{1}{3}} = 17525.86 N \]

→From bearings table:

for left bearing:

Bearing No: 6406

D=90mm  d=30mm  b=23mm  \( c_{R_1} = 43.4 KN \)

For right bearing:

\( c_{R_2} = 17.5 \)

Bearing No: 62/28

D=58mm  d=28mm  b=16mm  \( c_{R_2} = 17.9 kN \)

3. Bearing selection @ 2\textsuperscript{nd} intermediate shaft:

# Step down from \( d_{\text{shaft}} = 72 \text{ mm} \) to \( d_{\text{shaft}} = 70 \text{ mm} \) @ Bearing cause there are no bending moment at the end of the shaft

\[ P_1 = \sqrt{8425.3^2 + 871.2^2} = 8470.22 N \]

\[ P_2 = \sqrt{4056.1^2 + 1020.3^2} = 4182.45 N \]

\[ c_{R_4} = 8470.22(1135.3)^{\frac{1}{3}} = 88.4 kN \]
\[ c_{R_2} = 4182.45(1135.3)^{\frac{1}{3}} = 43.6 \, kN \]

→From bearings table:
for left bearing:
\[ c_{R_1} = 88.4 \, KN \]
We used two bearing
Bearing No: 6408
D=110mm   \, d=40mm   \, b=27mm   \, c_{R_1} = 63.7 \, KN

For right bearing:
\[ c_{R_2} = 43.6 \, kN \]
Bearing No: 6408
D=110mm   \, d=40mm   \, b=27mm   \, c_{R_2} = 63.7 \, kN

Output shaft bearing:
d=58mm \, say \, 60mm
\[
P = \sqrt{8839.18^2 + 3217.19^2} = 9406.46 \, N
\]
\[ c = P(L)^\frac{1}{K} \]

For deep groove:

\[ K = 3 \]

\[ C = 9406.46(1135.3)^{\frac{1}{3}} = 98.13 \text{ kN} \]

\[ C = 98.13 \text{ kN} \]

→ **From bearings table:**

**Bearing no. = 6412.**

B = 35mm

\[ D = 150mm \]

d = 60

\[ C_r = 110 \text{ kN} \]

**Summary:**

**Table 13; summary of bearing design:**

<table>
<thead>
<tr>
<th>Bearing no.</th>
<th>Bearing width</th>
<th>D</th>
<th>d</th>
</tr>
</thead>
<tbody>
<tr>
<td>6203</td>
<td>12</td>
<td>40</td>
<td>17</td>
</tr>
<tr>
<td>6406</td>
<td>23</td>
<td>90</td>
<td>30</td>
</tr>
<tr>
<td>62/28</td>
<td>16</td>
<td>58</td>
<td>28</td>
</tr>
<tr>
<td>6408 two</td>
<td>27</td>
<td>110</td>
<td>40</td>
</tr>
<tr>
<td>6414</td>
<td>27</td>
<td>110</td>
<td>40</td>
</tr>
<tr>
<td>6412</td>
<td>35</td>
<td>150</td>
<td>60</td>
</tr>
</tbody>
</table>
3.1.4 Keys Selection:

A key is a machine element used to connect the driver member such as belt pulley, chain sprocket or gear to the shaft that carries it.

By using shear force:

\[ \tau_{\text{max}} = \frac{F_T}{A_{\text{Shear}}} \]

\[ A_{\text{Shear}} = w \times l \]

\[ \sigma_s = \frac{F_T}{\frac{H}{2} \times L} = \frac{F_T}{A_C} \]

Assumptions:

Key material is C1020 annealed

\[ \sigma_{\text{yield}} = 42 \text{ ksi} \]

Safety factor = 3

\[ \therefore \sigma_s = \frac{\sigma_{\text{yield}}}{s.f.} \times 6.894 = \frac{42}{3} \times 6.894 = 96.5 \text{ MPa} \]

\[ \tau_s = \frac{\sigma_s}{2} = \frac{96.5}{2} = 48.3 \text{ MPa} \]
For pinion:

Key length \( b_p \)

For gear:

Key length = \( b_g + \Delta \)

* \( \Delta = 1 \text{ mm} \)

1. Key @ 1st pinion:

\[
A_{\text{Shear}} = w \times l = 19 \times 4.76 = 90.44 \text{ mm}^2
\]

\[
F_T = F_{T_{\text{SET}_1}} = 1213.04 \text{ N}
\]

\[
A_c = \frac{H}{2} \times L = \frac{4.76}{2} \times 19 = 45.22 \text{ mm}^2
\]

\[
\tau_s = \frac{F_T}{A_s} = \frac{1213.04}{90.44} = 12.19 \frac{N}{\text{mm}^2} = 12.19 \text{ MPa} < 48.3 \text{ MPa} \Rightarrow \text{Safe}
\]

\[
\sigma_c = \frac{F_T}{A_c} = \frac{1213.04}{45.22} = 26.83 \frac{N}{\text{mm}^2} = 26.83 \text{ MPa} < 96.5 \text{ MPa} \Rightarrow \text{Safe}.
\]

2. Key @ 1st gear:

Key length = 19 + 1 = 20 mm

\[
F_T = F_{T_{\text{SET}_1}} = 1213.04 \text{ N}
\]

\[
A_{\text{Shear}} = w \times l = 20 \times 6.35 = 127 \text{ mm}^2
\]
\[ A_c = \frac{6.35}{2} \times 20 = 63.52 \text{ mm}^2 \]

\[ \tau_s = \frac{F_T}{A_s} = \frac{1213.04}{127} = 9.55 \frac{N}{\text{mm}^2} = 9.55 \text{ MPa} < 48.3 \text{ MPa} \Rightarrow \text{Safe} \]

\[ \sigma_c = \frac{F_T}{A_c} = \frac{1213.04}{63.5} = 19.1 \frac{N}{\text{mm}^2} = 19.1 \text{ MPa} < 96.5 \text{ MPa} \Rightarrow \text{Safe} \]

3. Key @ 2\text{nd} pinion:

\[ F_T = F_{T@set_2} = 3644.5 \text{ N} \]

\[ A_s = 6.35 \times 31 = 196.85 \text{ mm}^2 \]

\[ A_c = \frac{6.35}{2} \times 31 = 98.43 \text{ mm}^2 \]

\[ \tau_s = \frac{F_T}{A_s} = \frac{3644.5}{196.85} = 18.51 \frac{N}{\text{mm}^2} = 18.51 \text{ MPa} < 48.3 \text{ MPa} \Rightarrow \text{Safe} \]

\[ \sigma_c = \frac{F_T}{A_c} = \frac{3644.5}{98.43} = 37.03 \frac{N}{\text{mm}^2} = 37.03 \text{ MPa} < 96.5 \text{ MPa} \Rightarrow \text{Safe} \]

4. Key @ 2\text{nd} gear:

\[ F_T = F_{T@set_2} = 3644.5 \text{ N} \]

\[ A_{\text{Shear}} = w \times l = 9.53 \times 32 = 304.96 \text{ mm}^2 \]

\[ A_c = \frac{9.53}{2} \times 32 = 152.48 \text{ mm}^2 \]
\[ \tau_s = \frac{F_T}{A_s} = \frac{3644.5}{304.96} = 11.95 \text{ kN/m}^2 = 11.95 \text{ MPa} < 48.3 \text{ MPa} \Rightarrow Safe \]

\[ \sigma_c = \frac{F_T}{A_c} = \frac{3644.5}{152.48} = 23.9 \text{ kN/m}^2 = 23.9 \text{ MPa} < 96.5 \text{ MPa} \Rightarrow Safe \]

5. Key @ 3\textsuperscript{rd} pinion:

\[ A_s = w \times l = 9.53 \times 47 = 447.91 \text{ mm}^2 \]

\[ F_T = F_{T_{@set_3}} = 8839.18 \text{ kN} \]

\[ A_c = \frac{9.53}{2} \times 47 = 223.96 \text{ mm}^2 \]

\[ \tau_s = \frac{F_T}{A_s} = \frac{8839.18}{447.91} = 19.73 \frac{N}{\text{mm}^2} = 19.73 \text{ MPa} < 48.3 \text{ MPa} \Rightarrow Safe \]

\[ \sigma_c = \frac{F_T}{A_c} = \frac{8839.18}{223.96} = 39.46 \frac{N}{\text{mm}^2} = 39.46 \text{ MPa} < 96.5 \text{ MPa} \Rightarrow Safe \]

6. Key @ 3\textsuperscript{rd} gear:

\[ F_T = F_{T_{@set_3}} = 8839.18 \text{ kN} \]

\[ A_s = w \times l = 15.88 \times 48 = 762.24 \text{ mm}^2 \]

\[ A_c = \frac{15.88}{2} \times 48 = 381.12 \text{ mm}^2 \]
\[
\tau_s = \frac{F_T}{A_s} = \frac{8839.18}{762.24} = 11.59 \text{ N/mm}^2 = 11.59 \text{ MPa} < 48.3 \text{ MPa} \Rightarrow \text{Safe}
\]

\[
\sigma_c = \frac{F_T}{A_c} = \frac{8839.18}{381.12} = 23.19 \text{ N/mm}^2 = 13.19 \text{ MPa} < 96.5 \text{ MPa} \Rightarrow \text{Safe}
\]

**Summary:**

Table 14: summary of keys design:

<table>
<thead>
<tr>
<th>Spur gear</th>
<th>Width (mm)</th>
<th>Height (mm)</th>
<th>Length (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1st pinion</td>
<td>4.76</td>
<td>4.76</td>
<td>19</td>
</tr>
<tr>
<td>2nd pinion</td>
<td>6.35</td>
<td>6.35</td>
<td>20</td>
</tr>
<tr>
<td>3rd pinion</td>
<td>9.53</td>
<td>9.53</td>
<td>31</td>
</tr>
<tr>
<td>1st gear</td>
<td>6.35</td>
<td>6.35</td>
<td>32</td>
</tr>
<tr>
<td>2nd gear</td>
<td>9.53</td>
<td>9.53</td>
<td>47</td>
</tr>
<tr>
<td>3rd gear</td>
<td>15.88</td>
<td>15.88</td>
<td>48</td>
</tr>
</tbody>
</table>

### 3.2 Brake design:

To calculate the braking torque needed to accelerate and decelerate we need to calculate the inertia of parts and polar moment of inertia.

See Appendix C Fig 9.

\[ J = m r^2 \]

\[ M = \text{mass (kg)} \]

\[ R = \text{radius (m)} \]

Polar moment of inertia = \[ j \omega \]

We need to calculate it for the all moving parts
**Static Torque** Static torque is that torque required to hold the load stationary at a nominated depth, ignoring frictional resistances.

**For vertical shafts**

\[ T = \text{Mass (Kg)} \times 9.81 \times \text{Drum Radius (M)} \]

\[ T = 40 \times 9.81 \times 0.525 = 206 \text{ NM} \]

**Accelerating or Decelerating Torque** The torque required at the drum shaft to accelerate or decelerate the winder system will be the summation of the various torques created by inertias, frictional resistances, and static torques. With deceleration, frictional resistances are often ignored because frictional resistances vary, and so cannot be relied upon when considering the braking requirements of winders. (8)

Total Torque = Static Torque + Torque to Inertia + Torque from friction.

\( \mu \) = coefficient if friction for shafts
\( \mu = 0.18 \) for vertical winding with fishplate guide rails

**Table 15; inertia refer to drum shaft:**

<table>
<thead>
<tr>
<th>component</th>
<th>Component inertia (Kg.M^2)</th>
<th>Component mass (kg)</th>
<th>Inertia refer to drum shaft (Kg.M^2)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
Angular acceleration at drum
= linear acceleration * 2 Drum diameter = 0.75 * 2 * 1.05 = 1.575 Radians/second^2

Additional torque to accelerate
= J \alpha = 1609.1 * 1.575 = 2534.34 NM = 2.534 kN.m

Torque to overcome friction
= static torque * friction coeff. = 206 * .18 = 37.1 NM

∴ Total torque to accelerate
= 2534.34 + 206 + 37.1 = 2.777 kNM

We choose type ERS FENIX 10
Electrically Released Brake.

Have braking torque up to 4600 Nm.
Suitable for new build gear motors to meet EN 81-1+A3 conformity

ERS FENIX 10
• Compact design
• Low noise operation through the life of the brake
• Standard torque capacity: 200 to 4600 Nm
• Single magnet and two friction disc
• EC 95/16 certified
• No air gap adjustment required
• Very easy installation
• Micro switch equipped
• Dust cover and hand release on request
• Nearly maintenance free
  (Further information in our service manual)
• Over excitation (dual voltage) or single voltage.

3.3 **Winding drum design:**

See Appendix C Fig 10.

**Differences between counterweight and winding drum:**

**The counterweight:**

The Counter Weight Chain Drive system for residential elevators uses weights that offset the need for more powerful motors to lift the cab up. These type of elevators use a chain drive to lift the cab as the weights are lowered creating an offset.

The counter weight system does not require a machine room so valuable living space is preserved without sacrificing functionality.

You can click the image to the left to get a larger version of the picture so you can read the text.

The counter weight chain drive home elevator is very efficient and is low maintenance.
**Winding drum:**

The winding drum type system elevators use 3/8 inch aircraft cable and a motor that can be mounted at the top of the rail system or behind the rail (at the top or bottom of the shaft). Depending on your maximum height in the shaft.

The winding drum uses a variable speed motor that winds the cable onto a drum and then on descent allows the car to use its weight to help with lowering. The variable speed also gives you a soft start and soft stop.

All motors vibrate so you have motor noise associated with this drive system however if this home elevator drive type is installed properly, you are not affected by an uncomfortable amount of noise. (9).

**Rope design:**

- Maximum weight on rope= 1500 kg.
- Assume safety factor for steel rope=10
- Assume rope mass per kg= 1kg/m.
- Assume rope length 10 m.
- Then total mass of rope= 10kg.
- Minimum rope breaking load= 1510*9.81= 14.82 kN.

*Table 16 Minimum breaking load*
<table>
<thead>
<tr>
<th>Nominal Diameter [mm]</th>
<th>Weight [kg/m]</th>
<th>Minimum Breaking Loads</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>1770 Mpa [kN]</td>
</tr>
<tr>
<td></td>
<td></td>
<td>[Kg]</td>
</tr>
<tr>
<td>7</td>
<td>0.23</td>
<td>36</td>
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<tr>
<td></td>
<td></td>
<td>3,671</td>
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<tr>
<td>8</td>
<td>0.30</td>
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<tr>
<td></td>
<td></td>
<td>4,997</td>
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<tr>
<td>9</td>
<td>0.38</td>
<td>60</td>
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<td>6,118</td>
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<td>74</td>
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<td>7,546</td>
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<td>90</td>
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<tr>
<td></td>
<td></td>
<td>9,177</td>
</tr>
<tr>
<td>12</td>
<td>0.67</td>
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<tr>
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<td>1.64</td>
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<td>20</td>
<td>1.82</td>
<td>298</td>
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<tr>
<td>24</td>
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<tr>
<td></td>
<td></td>
<td>43,746</td>
</tr>
</tbody>
</table>
Typical 6*19 Round Strand Ropes

Typical Flattened Strand Ropes

Typical Locked Coil Ropes

Typical Half Locked Coil Rope

Construction of Typical Winding Ropes
Fig 2.3
Figure 1 construction of typical winding Rope

Using typical flattened steel rope

From table above we use the rope with:

\[ \begin{align*}
D &= 15\text{mm} \\
\text{weight} &= 1.04\text{kg/m} \\
\text{breaking load} &= 168\text{KN}.
\end{align*} \]

Safety factor = 168/14.82 = 11.3 safe.

Winding drum:-

Assume the rope angle from drum to sheave (fleet angle) = 1.5 degree.

Assume the drum will have parallel rope grooves.

Distance from drum to sheave = 2 m.

Drum width = 2*(distance to sheave* tan fleet angle)

\[ = 2*2*\tan 1.5 = 105\text{mm} \]

Drum speed up to 6m/s, \( D/d \) drum to rope ratio = 70

Drum to rope diameter = 70

Drum diameter = drum to rope ratio* rope diameter.

Then drum diameter = 70*15 = 1050 mm.

Pitch of rope groove = 15 + (15*0.04) = 15.6 mm.

Number of grooves = 105/15.6 = 6.37 almost 7 grooves.

Then actual drum diameter = 7*15.6 = 109.2 mm.

Rope groove depth = 0.1*15 = 1.5 mm.

Groove nominal radius = 0.05*15 + 15 = 15.75 mm.
Working rope length:-

Allow three dead coils on drum at all times.

Working rope diameter at first layer = 1050 + 15 = 1065mm.

Working rope diameter at second layer = 1065 + (2 * 12.81) = 1090.625mm.

Working rope diameter at third layer = 1090.625 + (2 * 12.81) = 1116.25mm.

Working rope length at first layer = (10 - 3) * π * 1.07 = 23.65m

Working rope length at second layer = 9 * π * 1.1 = 31.14m

Working rope length at third layer = 10 * π * 1.126 = 35.40m

Total drum capacity with 3 layer dead rope = 35.4 + 31.14 + 23.65 = 90.2m

Capacity required = 3 layer of dead rope + length required

Capacity required = (2 * π * 1.05 * 3) + 10 = 29.8m = 30m

Capacity required < total capacity of drum safe.

3.3.1 Winding drum shaft design:

\[ T/j = \tau/r \]

Where \( T \) = Twisting moment (or torque) acting upon the shaft,
\( J \) = Polar moment of inertia of the shaft about the axis of rotation,
\( \tau \) = Torsional shear stress, and

\[ r = \text{Distance from neutral axis to the outer most fibre} \]
\[ = \frac{d}{2}; \text{where } d \text{ is the diameter of the shaft.} \]
\[ J = \pi/32 \cdot d^4 \]

Choosing c1045 steel \( \sigma = 310 \text{ MPa} \) \( \tau = 310/3 = 103.34 \text{ MPa} \)

\[ T = 1315.25 \text{ N.m} \]

\[ T = \pi/16 \cdot \tau \cdot d^3 \]

\[ D = 38.62 \text{ mm} \]

\[ D = 40 \text{ mm} \]

3:4 design of space trust:

The space trust have a weight room

3.4 Design of Turnbuckle:

Diameter of the rods

\( P_d = \pi/4 \cdot d^2 \cdot \sigma_t \)

\[ P = 1500 \times 9.81 = 14,715 \text{ N} \]

\[ P_d = 1.3 \times P = 19.2 \text{ KN} \]

\( \sigma_t = 75 \text{ Mpa} \) \( \tau = 37.5 \text{ Mpa} \)

\( d = 18 \text{ mm} \)

M18 bolt is chosen

Length of coupler nut:-

\( P_d = \pi \cdot d \cdot L \cdot \tau \)

\[ L = 9.06 \text{ mm} \]

We know minimum \( L = 1.5d \)

Then \( L = 18 \text{ mm} \)

From tables the pitch of M18 = 2.5 and working diameter is \( d_c = 16.376 \text{ mm} \)

Number of threads per length \( (n) = 1/2.5 = 0.4 \)
Pd = \{d^2 - dc^2\} * n*L * \sigma_c

\sigma_c = 60.81 \text{ MPa} \text{ less than } \sigma_t

Then the design is satisfactory

**Outside diameter of the coupler nut:**

Let \( D = \text{Outside diameter of the coupler nut} \)

\[ P = \{D^2 - d^2\} \times \sigma_t \]

\( D = 23.95 \text{ mm} = 24 \text{ mm}. \)

**Outside diameter of the coupler:**

The outside diameter of the coupler \( (D_2) \) may be obtained by considering the tearing of the coupler.

We know that

Tearing resistance of the coupler

\[ = \frac{\pi}{4} \times (D_2^2 - D_1^2) \times \sigma_t \]

Then

\( D_2 = 28.74 \text{ mm} \)

We know that the minimum outside diameter = 1.5d

Then \( D_2 = 60 \text{ mm} \)

Thickness of the coupler

\( T_1 = 0.75d = 13.5 \text{ mm} \)

And thickness of the coupler nut

\( T = 0.5d = 9 \text{ mm}. \)
3.5 Selection of Control:

We choose two sensors:

3.5.1 Ultra Sonic Sensor:

Ultrasonic sensors emit short, high-frequency sound pulses at regular intervals. These propagate in the air at the velocity of sound. If they strike an object, then they are reflected back as echo signals to the sensor, which itself computes the distance to the target based on the time-span between emitting the signal and receiving the echo.

As the distance to an object is determined by measuring the time of flight and not by the intensity of the sound, ultrasonic sensors are excellent at suppressing background interference.
3.5.2 Weight Sensor:

Can be used in atrocious environment and hazardous areas. It is suitable for crane scale, and elevators mechanical conversion scale, hopper scale and other electronic weighing devices beam design and/or compression loading possible, easy installation.

Sense the weight and send signal to the break. If the load is more than 520kg.
3.5.1 Micro controller:

![Diagram of Micro controller](image)

Figure 5 Micro controller

3.6 **Design of space truss:**
A space truss or space frame structure utilizes a three-dimensional truss to resist lateral forces. Unlike a normal truss, in which horizontal, vertical, and diagonal members work together on a single plane, a space truss uses diagonal connections which branch outside of the plane. A space truss usually looks like several interlocking pyramidal outlines.

- See Appendix A.

3.7 **Design of Carriage:**

Is manufactured out of steel which weights 200 kg.

And can carry 520 kg.

See Appendix B.

3.8 **Design of Guide Rails:**

See Appendix C Fig 11 &12.

Guide Rails Required

Elevator cars shall be provided with guide rails.
Material
Guide rails, guide-rail brackets, rail clips, fishplates, and their fastenings shall be either

- Steel
- wood or other suitable nonmetallic materials,

Requirements for Steel, Where Used

(a) Rails, brackets, fishplates, and rail clips shall be made of open-hearth steel, or its equivalent, having a tensile strength of not less than 380 MPa (55,000 psi) and having an elongation of not less than 22% in a length of 50 mm.

Using Trail guide

For selecting the proper guide rails we must know the total load of the car and the spacing between the guide rails.

Total load on car is 1500 kg and maximum spacing distance is 1.5 m.
From the results we use the:

<table>
<thead>
<tr>
<th>Nominal Mass, kg/m</th>
<th>A</th>
<th>B</th>
<th>C</th>
<th>D</th>
<th>E</th>
</tr>
</thead>
<tbody>
<tr>
<td>8.5</td>
<td>68.3</td>
<td>82.6</td>
<td>9.1</td>
<td>25.4</td>
<td>6.0</td>
</tr>
<tr>
<td>9.5</td>
<td>49.2</td>
<td>69.9</td>
<td>15.9</td>
<td>25.4</td>
<td>7.9</td>
</tr>
<tr>
<td>12.0</td>
<td>61.9</td>
<td>88.9</td>
<td>15.9</td>
<td>31.8</td>
<td>7.9</td>
</tr>
<tr>
<td>16.5</td>
<td>88.9</td>
<td>114.3</td>
<td>15.9</td>
<td>38.1</td>
<td>7.9</td>
</tr>
<tr>
<td>18.0</td>
<td>88.9</td>
<td>127.0</td>
<td>15.9</td>
<td>44.5</td>
<td>7.9</td>
</tr>
<tr>
<td>22.5</td>
<td>88.9</td>
<td>127.0</td>
<td>15.9</td>
<td>50.0</td>
<td>12.7</td>
</tr>
<tr>
<td>27.5</td>
<td>108.0</td>
<td>139.7</td>
<td>19.1</td>
<td>50.0</td>
<td>12.7</td>
</tr>
<tr>
<td>33.5</td>
<td>101.6</td>
<td>139.7</td>
<td>28.6</td>
<td>50.8</td>
<td>14.3</td>
</tr>
<tr>
<td>44.5</td>
<td>127.0</td>
<td>139.7</td>
<td>31.8</td>
<td>57.2</td>
<td>17.5</td>
</tr>
</tbody>
</table>

To select the fishplates and fasteners bolts needed to support the rails we selected it from the following table. (9)
For nominal mass of 16.5 kg/m we use

Table 17: Diameter of bolts:

<table>
<thead>
<tr>
<th>Nominal mass of guide rail, Kg/m</th>
<th>Minimum thickness of fish plates, mm</th>
<th>Minimum diameter of bolts, mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>16.5</td>
<td>17</td>
<td>M16</td>
</tr>
</tbody>
</table>

Forces on guide rails

\[ F_Y = 5 \times Q \times g_n \times b / 16h \]
\[ F_X = 5 \times Q \times g_n \times c / 128h \]

Q= rated load (kg)

\( g_n \) = standard acceleration (m/s\(^2\))

B= width of the car (mm)

h = vertical distance between the centerlines of the guide rails shoe (mm)

c= depth of the car

L= spacing between brackets

L=1500mm

H= 2000mm

B=1500mm

C=1000mm.
Table 18: Designation of Guide Rails:

<table>
<thead>
<tr>
<th>Designation</th>
<th>b1</th>
<th>h1</th>
<th>k</th>
<th>n</th>
<th>c</th>
<th>f</th>
<th>g</th>
<th>rs</th>
<th>m1</th>
<th>m2</th>
</tr>
</thead>
<tbody>
<tr>
<td>T127-2c14/BE</td>
<td>127</td>
<td>88,9</td>
<td>15,88</td>
<td>50,8</td>
<td>14</td>
<td>15,9</td>
<td>12,7</td>
<td>5</td>
<td>6,4</td>
<td>6,37</td>
</tr>
</tbody>
</table>

The specification of T127-2c14/BE is

\[ E = 2.1 \times 10^5 \text{ mm} \]

\[ S = 29, 53 \text{ cm}^2 \]

\[ J_x = 198, 8 \text{ cm}^4 \]

\[ J_y = 229, 6 \text{ cm}^4 \]

\[ W_x = 31, 10 \text{ cm}^3 \]

\[ W_y = 36, 20 \text{ cm}^3 \]

\[ I_x = 2, 59 \text{ cm} \]

\[ I_y = 2, 78 \text{ cm} \]

\[ F_y = 5 \times 1000 \times 9.81 \times 1500 / 16 \times 2000 = 460 \text{ N} \]

\[ F_x = 5 \times 1000 \times 9.81 \times 1000 / 128 \times 2000 = 191.6 \text{ N} \]

Bending moments

\[ M_{0x} = 0.22 F_x \cdot L = 63, 228. \text{ N.mm} \]

\[ M_{0y} = 0.2 F_y \cdot L = 138, 000. \text{ N.mm} \]

Stress in bending

\[ \sigma_x = M_{0x} / W_y = 17.5 \text{ N/mm}^2 \]

\[ \sigma_y = M_{0y} / W_x = 44.4 \text{ N/mm}^2 \]

Stresses in bending during the normal operation is very low

Deflection in individual planes

\[ \delta_x = c_3 \cdot F_x \cdot L^3 / E \cdot J_x \]

\[ c_3 = 0.1455 \]

\[ \delta_x = 0.226 \text{ mm} \]

\[ \delta_y = c_3 \cdot F_y \cdot L^3 / E \cdot J_y \]

\[ \delta_y = 0.5 \text{ mm} \]

The deflections are less than 3\text{ mm}

The designed guide rails is satisfactory from all aspects. (10)
CHAPTER FOUR
MANUFACTURING
4. CHAPTER FOUR
MANUFACTURING

4.1 Winder Drum Design:

The purpose of the winder drum is to accommodate the winding rope, together with any excess or testing lengths. It also provides a secure anchorage for the rope and allows the rope to scroll correctly on the drum.

4.1.1 General Construction of Winder Drums:

Modern practice is to fabricate the winder drum using rolled steel plates for the shell. Such drums have flexible end connections in comparison with rigid end connections (with much stiffening) used in older drum construction. (See Section for a guide to sizing the drum for the selected rope). Fabricated drums are normally in mild steel plate. Plates shall be certified free from laminations and inclusions. Any inclusions present at the time of rolling are likely to become laminations during rolling, and the plate could be rejected after much of the work has been done. Before any machining commences the fabricated drum should be stress relieved and all major welds ultrasonically proved.

The brake disc path may be welded or bolted to the drum. Both methods have been successfully used. Currently drum design favors the bolted-on approach.

Give special attention to the shell-to-endplate connection and the method used for welding. The connection must be flexible enough to avoid weld cracking.

For Grade 250 steel a maximum shell compressive stress of 150 MPa should not be exceeded.

For Grade 250 steel bending stresses in the shell should not exceed 40 MPa, and bending stresses in the end plates 60 MPa. (11)
4.2 **Guide Rope Lubrication:**

Rope wear can be reduced to a minimum by the use of anti-wear materials for guide slippers, and well lubricated rope.

Corrosion can be kept to a minimum by efficient lubrication, the most susceptible being the fixing points at top and bottom. The use of galvanized wires generally assists to overcome corrosion problems. Where corrosion in a shaft is considered to be a problem, the use of half-locked coil ropes have advantages in that the resistance to corrosion is greater since the clearances through which moisture can penetrate are smaller. (11)

4.2.1 **Hawse Hole or Rope Entry Position:**

The rope is passed from the rope anchorage position, usually inside the drum endplate, to the first coil through a hole formed in the drum shell and known as the hawse hole. It is important that the correct position and side of the drum be determined for the hawse hole.

Where the center of the sheave falls to one side of the drum rather than on the centerline of it, the hawse hole on that side should be used, irrespective of what hand of lay the rope is. The arrangement should also be such that the number of unused turns of rope on the drum is sufficient to cause the live turns of rope to always be on the side of the drum beyond the sheave centerline with respect to the hawse hole in use.

Always design hawse holes so that the rope enters the drum without sharp turns. All corners and sharp edges should be removed to avoid damage to the rope by nicking or crushing.

Wedges and Risers to avoid abrasion of the rope on its first turn, fit a steel rope wedge against the flange in front of the hawse hole.

When the rope fills the first layer and starts to return on the second layer, the rope will be lifted. At this point severe crushing can occur. To prevent this a steel riser is fitted to the flange and drum shell to lift the rope. (11).
4.3 **Gearbox manufacturing**:

Any gearbox contains number of parts as following:

1. Casing or housing.
2. Gears.
3. Shafts.
5. Bearing.

---

4.3.1 **Casing manufacturing**:  
The manufacturing of the casing is done by casting and CNC milling machine  
Casting is a manufacturing process by which a liquid material is usually poured into a mold, which contains a hollow cavity of the desired shape, and then allowed to solidify. The solidified part is also known as a casting, which is ejected or broken out of the mold to complete the process. Casting materials are usually metals or various cold setting materials that cure after mixing two or more components together; examples are epoxy, concrete, plaster and clay. Casting is most often used for making complex shapes that would be otherwise difficult or uneconomical to make by other methods.

4.3.1.1 **CNC MILLING**:  
Is used to because the final product is not exactly the same as the final shape because there are several limitations for casting. We use the raw part from casting process because the shape of casing is too difficult to be machined and it will have high cost. (12)

4.3.2 **Gears manufacturing process**:  
Gear manufacturing basically is categorized based on the type of gear to be manufactured. The types of gears can be classified as follows:

1. Spur.
2. Helical.
3. Worm Gears.

The manufacturing methodology involved for manufacturing of the above types of gears defines the types of machines to be used. The various production methods for producing the above mentioned gears by machining are as follows.

a) Gear Milling: This is one of the initial and best known metal removal process for making gears. This method requires the usage of a milling machine. It is also to be noted that this method can produce nearly all types of gears. The method involves the use of a form cutter, which is passed through the gear blank to create the tooth gap. This method is right now used only for the manufacture of gears requiring very less dimensional accuracy. To put it correctly this method is currently outdated.

b) Gear Hobbing: Gear Hobbing is a continuous generating process in which the tooth flanks of the constantly moving work piece are formed by equally spaced cutting edges of the hob. The main advantage of this process is its versatility to produce a variety of gears including Spur, Helical, Worm Wheels, Serrations, Splines, etc. The main advantage of the method is the higher productivity rate of the gears.

c) Gear Shaping: Gear shaping is a generating process. The cutter used is virtually a gear provided with cutting edges. The tool is rotated at the required velocity ratio relative to the gear to be manufactured and any one manufactured gear tooth space is formed by one complete cutter tooth. This method can be used to produce cluster gears, internal gears, racks, etc. with ease, which may not have the possibility to be manufactured in gear hobbing.

d) Bevel Gear Cutting: Bevel gear cutting is a very specialized area in the field of gear cutting. This involves a special type for each variety of the bevel gear to be manufactured. Some of the bevel gear types along with the type of machines required are Gleason, Oerlikon, Hypoid, Zerol, etc. Each type of bevel gear is always manufactured only on the corresponding type of machine against its name. The tooling required also tends to vary based on the type of gear. (13)
4.4 Manufacturing of Shafts

Shafts are generally manufactured by hot rolling and finished to size by cold drawing or turning and grinding. The cold rolled shafts are stronger than hot rolled shafts but with higher residual stresses.

The residual stresses may cause distortion of the shaft when it is machined, especially when slots or keyways are cut. Shafts of larger diameter are usually forged and turned to size in a lathe.

4.5 Keys:

A key is a piece of mild steel inserted between the shaft and hub or boss of the pulley to connect these together in order to prevent relative motion between them. It is always inserted parallel to the axis of the shaft. Keys are used as temporary fastenings and are subjected to considerable crushing and shearing stresses. A keyway is a slot or recess in a shaft and hub of the pulley to accommodate a key.

4.5.1 Types of Keys

1. Sunk keys.
2. Saddle keys.
3. Tangent keys.
4. Round keys, and
5. Splines.

4.6 Truss:

There are three types of joining of trusses:-

- Riveted joints
- Welded joints
- Screw joints
4.6.1 **Riveted joints:**
A rivet is a short cylindrical bar with a head integral to it. The cylindrical portion of the rivet is called *shank* or *body* and lower portion of shank is known as *tail*.

The rivets are used to make permanent fastening between the plates such as in structural work, ship building, bridges, tanks and boiler shells. The riveted joints are widely used for joining light metals.

4.6.1.1 **Methods of Riveting:**
The function of rivets in a joint is to make a connection that has strength and tightness. The strength is necessary to prevent failure of the joint. The tightness is necessary in order to contribute to strength and to prevent leakage as in a boiler or in a ship hull.

When two plates are to be fastened together by a rivet, the holes in the plates are punched and reamed or drilled. Punching is the cheapest method and is used for relatively thin plates and in structural work. Since punching injures the material around the hole, therefore drilling is used in most pressure-vessel work. In structural and pressure vessel riveting, the diameter of the rivet hole is usually 1.5 mm larger than the nominal diameter of the rivet.

4.6.2 **Welded joints:**
A welded joint is a permanent joint which is obtained by the fusion of the edges of the two parts to be joined together, with or without the application of pressure and a filler material. The heat required for the fusion of the material may be obtained by burning of gas (in case of gas welding) or by an electric arc (in case of electric arc welding). The latter method is extensively used because of greater speed of welding. Welding is extensively used in fabrication as an alternative method for casting or forging and as a replacement for bolted and riveted joints. It is also used as a repair medium *e.g.* to reunite metal at a crack, to build up a small part that has broken off such as gear tooth or to repair a worn surface such as a bearing surface.

4.6.3 **Screw joints:**
A screw thread is formed by cutting a continuous helical groove on a cylindrical surface. A screw made by cutting a single helical groove on the cylinder is known as *single threaded* (or single-start) screw and if a second thread is cut in the space between the grooves of the first, a *double*
threaded (or double-start) screw is formed. Similarly, triple and quadruple (multiple-start) threads may be formed.

The helical grooves may be cut either right hand or left hand. A screwed joint is mainly composed of two elements. A bolt and nut. The screwed joints are widely used where the machine parts are required to be readily connected or disconnected without damage to the machine or the fastening. This may be for the purpose of holding or adjustment in assembly or service inspection, repair, or replacement or it may be for the manufacturing or assembly reasons. (14)
CHAPTER FIVE

FEASIBILITY STUDY
5. CHAPTER FIVE: *FEASIBILITY STUDY*

5.1 **Prices table:**

IPE = I Section beam.

UPE=U section beam.

∟ = angle.

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The Total cost of the project is 141,862.286 SDG.

To minimize material waste the length of the multiply steel element must be 6 m.
CHAPTER SIX

CONCLUSION AND RECOMMENDATIONS
6. Chapter six: Conclusion and recommendations

6.1 Conclusion:

- The space truss has been designed with diminution (1.5m×1m×8m).
- The Gear Box has been designed.
- Although the brake, guide rails, rope, control sensors and electric motor have been selected.
- The winding drum has been designed.
- Load carriage has been designed.

6.2 Recommendations:

1- Further analysis of soil for Blue Nile land, to estimate the exact depth length of pails.
2- The use of gearless motor to reduce the total cost.
3- More up to date market and industrial survey because the prices vary with time.
4- Use anti corrosion coating for the hole space truss to avoid chemical corrosion.
5- Predictive Maintenance for parts of the lift according to the maintenance policy.
7. References


8. Appendix

8.1 Appendix A:
8.2 Appendix B:

Description:

Summarize the FEM analysis on Assem1

Assumptions

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#### Zero strain temperature | Kelvin |

Include fluid pressure effects from SolidWorks Flow Simulation | Off |

Friction: | Off |

Ignore clearance for surface contact | Off |

Use Adaptive Method: | Off |

### Units

| Unit system: | SI |

| Length/Displacement | mm |

| Temperature | Kelvin |

| Angular velocity | rad/s |

| Stress/Pressure | N/m^2 |

### Material Properties

| No. | Body Name | Material | Mass | Volume |

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**Description:**

**Material Source:**

**Material Model Type:** Linear Elastic Isotropic

**Default Failure Criterion:** Max von Mises Stress

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<td>2e+011</td>
<td>N/m²</td>
<td>Constant</td>
</tr>
<tr>
<td>Poisson's ratio</td>
<td>0.26</td>
<td>NA</td>
<td>Constant</td>
</tr>
<tr>
<td>Property</td>
<td>Value</td>
<td>Unit</td>
<td>Constant</td>
</tr>
<tr>
<td>------------------------</td>
<td>----------------</td>
<td>----------</td>
<td>----------</td>
</tr>
<tr>
<td>Shear modulus</td>
<td>7.93e+010</td>
<td>N/m^2</td>
<td>Constant</td>
</tr>
<tr>
<td>Mass density</td>
<td>7850</td>
<td>kg/m^3</td>
<td>Constant</td>
</tr>
<tr>
<td>Tensile strength</td>
<td>4e+008</td>
<td>N/m^2</td>
<td>Constant</td>
</tr>
<tr>
<td>Yield strength</td>
<td>2.5e+008</td>
<td>N/m^2</td>
<td>Constant</td>
</tr>
</tbody>
</table>

**Loads and Restraints**

**Fixture**

<table>
<thead>
<tr>
<th>Restraint name</th>
<th>Selection set</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fixed-1 &lt;Part6-1, Part5-1, Part7-1&gt;</td>
<td>on 2 Edge(s), 10 Face(s) fixed.</td>
<td></td>
</tr>
</tbody>
</table>

**Load**

<table>
<thead>
<tr>
<th>Load name</th>
<th>Selection set</th>
<th>Loading type</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Force-1 &lt;plat-1&gt;</td>
<td>on 1 Face(s) apply normal force 10000 N using uniform distribution</td>
<td>Sequential Loading</td>
<td></td>
</tr>
<tr>
<td>Gravity-1</td>
<td>Gravity with respect to Top Plane with gravity acceleration</td>
<td>Sequential Loading</td>
<td></td>
</tr>
</tbody>
</table>
Connector Definitions

No Connectors were defined

Contact

Contact state: Touching faces - Free

<table>
<thead>
<tr>
<th>Global Contact</th>
<th>Contact component: Bonded on Assem1</th>
</tr>
</thead>
<tbody>
<tr>
<td>Description:</td>
<td></td>
</tr>
</tbody>
</table>

Mesh Information

<table>
<thead>
<tr>
<th>Mesh Type:</th>
<th>Solid Mesh</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mesher Used:</td>
<td>Standard mesh</td>
</tr>
<tr>
<td>Automatic Transition:</td>
<td>Off</td>
</tr>
<tr>
<td>Smooth Surface:</td>
<td>On</td>
</tr>
<tr>
<td>Jacobian Check:</td>
<td>4 Points</td>
</tr>
</tbody>
</table>
Element Size: 54.393 mm
Tolerance: 2.7197 mm
Quality: High
Number of elements: 10093
Number of nodes: 21032
Time to complete mesh(hh:mm:ss): 00:00:04
Computer name: HP-PC

Sensor Results
No data available.

Reaction Forces

<table>
<thead>
<tr>
<th>Selection set</th>
<th>Units</th>
<th>Sum X</th>
<th>Sum Y</th>
<th>Sum Z</th>
<th>Resultant</th>
</tr>
</thead>
<tbody>
<tr>
<td>Entire Body</td>
<td>N</td>
<td>-6.04323</td>
<td>15465.7</td>
<td>-2.5408</td>
<td>15465.7</td>
</tr>
</tbody>
</table>

Free-Body Forces

<table>
<thead>
<tr>
<th>Selection set</th>
<th>Units</th>
<th>Sum X</th>
<th>Sum Y</th>
<th>Sum Z</th>
<th>Resultant</th>
</tr>
</thead>
</table>

<table>
<thead>
<tr>
<th>Selection set</th>
<th>Units</th>
<th>Sum X</th>
<th>Sum Y</th>
<th>Sum Z</th>
<th>Resultant</th>
</tr>
</thead>
<tbody>
<tr>
<td>Entire Body</td>
<td>N-m</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>1e-033</td>
</tr>
</tbody>
</table>

**Free-body Moments**

**Bolt Forces**

No data available.

**Pin Forces**

No data available.

**Beams**

No data available.

**Study Results**
### Default Results

<table>
<thead>
<tr>
<th>Name</th>
<th>Type</th>
<th>Min</th>
<th>Location</th>
<th>Max</th>
<th>Location</th>
</tr>
</thead>
<tbody>
<tr>
<td>Stress1</td>
<td>VON: von Mises Stress</td>
<td>8.88844 N/m²</td>
<td>(-91.218 mm, 1540.46 mm, 691.995 mm)</td>
<td>8.93408e+007 N/m²</td>
<td>(-36.2762 mm, -1104.58 mm, 1400.13 mm)</td>
</tr>
<tr>
<td>Displacement1</td>
<td>URES: Resultant Displacement</td>
<td>0 mm</td>
<td>(-106.449 mm, -1009.54 mm, 1422.02 mm)</td>
<td>0.988242 mm</td>
<td>(662.49 mm, -1065.53 mm, 672.022 mm)</td>
</tr>
<tr>
<td>Strain1</td>
<td>ESTRN: Equivalent Strain</td>
<td>8.89842e-010 Element: 4555</td>
<td>(-163.873 mm, 963.446 mm, 1417.97 mm)</td>
<td>0.00029981 Element: 5978</td>
<td>(-43.5932 mm, -1099.4 mm, 1405.69 mm)</td>
</tr>
</tbody>
</table>
Assem1-Study 6-Stress-Stress1

Assem1-Study 6-Displacement-Displacement1
Assem1-Study 6-Strain-Strain1
8.3 Appendix C:

![Figure 8 Gear Box](image)

![Figure 9 Electrically Released Brake 10](image)
Figure 10 Winding Drum

Figure 11 T Guide Rail
Figure 12 Guia Rail Shoe