Discussion Group: 3

ME3200 Machine Design Project

Semester 5

Final Report

By

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1 Introduction

The module Machine Design Project ME 3200 is tasked to design a gearbox for a suitable application with 3 forward speeds and 1 reverse speed. Identifying the problem, background study, initial design calculations, and detailed design calculations are done in previous stages. A solid model is modelled using Solid Edge.

Mushroom cultivation is mostly done at household level as a small business. It is a profitable business if done correctly. So, in this project aim is to design a gearbox for a mushroom substrate mixer. Because lot of machines made for paddy particles. I am going to design a machine for sawdust.

2 Problem Description

Mushroom cultivation is mostly done at household level as a small business. It is a profitable business if done correctly. So, in this project aim is to design a gearbox for a mushroom substrate mixer. Because lot of machines made for paddy particles. I am going to design a machine for sawdust.

Oyster mushroom can be cultivated in any type of lingo cellulose material like straw, sawdust, rice hull. There are lots of growing mixture can be use. In Sri Lanka For 50 kg mixture use Saw dust (strew paddy) (20kg), Rice bran (3kg), Cao (400g), Soya flour (400g), MgSO4 (40g).[1]

In the industry there are lot of mushroom mixture machine, made for paddy. As my specimen I use Satrise-STM 2 machine. This machine use motor as a prime mover. following are the design parameters.

Container length	2100mm
Container wight	800mm
container thickness	20 gauge
Central shaft	40mm diameter *2100mm length
Blade length	600mm *10

Container hight	1100mm

Table 1- Design parameter

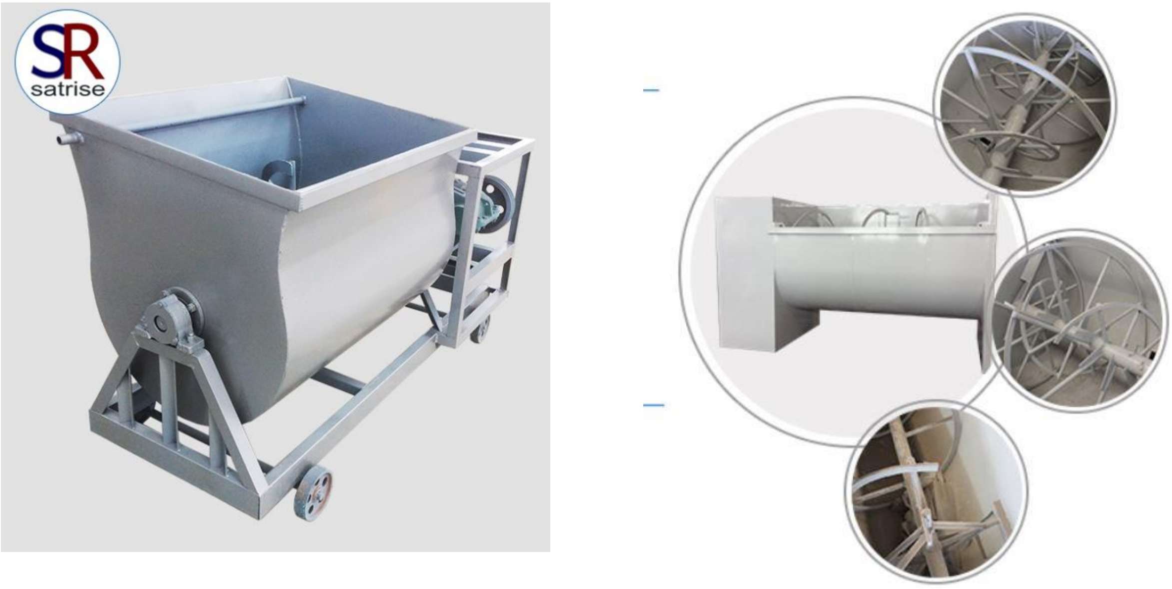


Figure 01-Substrate mixer

Product Parameters

Model	Specification	Power	Capacity	Weight
STM-1	1750*920*1260mm	2.2/3KW	100kg/time	260kg
STM-2	2100*850*1100mm	4kw/3phase	500kg/time	320kg
STM-3	2800*850*1100mm	4kw/3phase	500kg/time	380kg
STM-4	2800*900*1100mm	5.5kw/3phase	300kg/time	460kg
STM-5	2800*1200*1600mm	7.5kw/3phase	500kg/time	500kg
STM-6	4300*1800*2300mm	18kw/3phase	900kg/time	750kg
STM-7	4500*1800*2600mm	18kw/3phase	1300kg/time	750kg
STM-8	1200*700*700mm	4kw/3phase	500kg/time	220kg
STM-9	1200*700*850mm	8 horsepower	900kg/time	220kg

Figure 02-product parameters

My plan to design to use my gear first gear (hight speed low toque) wood sawdust mixture. Because there can contain some moisture content. So first we want to mix that moisture content.

Some country use hardwood pellets. In that situation we can use first gear to chop it into small particles like wood sawdust.

Figure 03 - Hardwood pellets

My second gear I plan to use after including other substances. Because there is a weight change in this mixture.so want more power to mix. 100kg mixture want to add 19kg substances.

My third gear (low speed hight toque) plan to use after including water to the mixture. Water requirements for a 100Kg substrate are 50L.

In my reverse gear is used to cleaning purpose. It should always be washed after working with this device so that all the mixture partial attached to the floor of this device is clean and free of any material. most mixture machines used today. the drum is cleaned manually. Therefore, it is important to have a reverse gear in the mixer. And another problem is how to get rid of this mixed mixture. This is when the need for a reverse gear arises again.

3 Background Study

3.1 What is a gearbox and why do we need a gearbox?

Gear is a wheel with teeth used to transmit power and motion from one shaft to another when constant velocity reduction is desired. Gear drives can multiply the torque while reducing the rotational speeds and changing the direction of shaft rotation. Compared to belt and chain drives gear drives can transmit large powers with higher efficiency and they are compact. also, they can operate at high speeds as well.

A gearbox is designed to control the input and output torque and speeds in the desired manner. According to MechStuff.com, an automobile gearbox is used to [2]

- To carry high loads OR climb steep slopes as well as achieve high speeds on straight roads.
- To keep the engine running even when the vehicle is not moving.
- To be able to drive the car backward by shifting to reverse gear.

The application range of gearbox is widespread. They are used in oil and mining, power generation, automobiles and locomotives, marine and machine tool applications, etc[3]. Torque and speed requirements might change depending on the application.

3.2 Classification of the gearbox

Gearbox can be classified into main types based on their mesh type,

- Sliding mesh gearbox
- Constant mesh gearbox
- Synchromesh gearbox

3.2.1 Sliding mesh gearbox

This type of gear mesh is used when the input can be stopped to change the gear otherwise the speed of the input and output shafts can be matched when changing the gear to avoid crashing of gear teeth[4], [5]. The design of this kind of gearbox is simple thus it is the oldest type of gearbox.

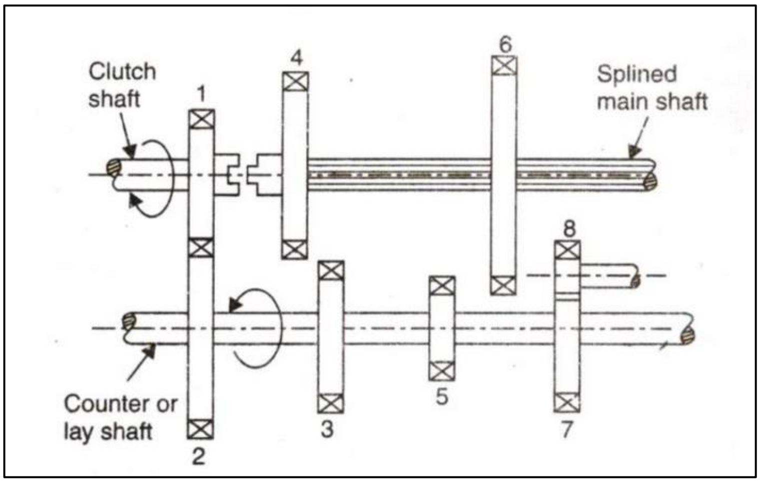


Figure 1:Sliding mesh gearbox[6]

3.2.2 Constant mesh gearbox

The gears of the lay shaft (or countershaft) are constantly in mesh with the gears of the output shaft. This type of gearbox is more advanced than the sliding mesh gearbox and operates more quietly. Dog clutches are used to shift between gears and therefore if there is a bad change in gears, the damage will be limited to the dog clutches[5].

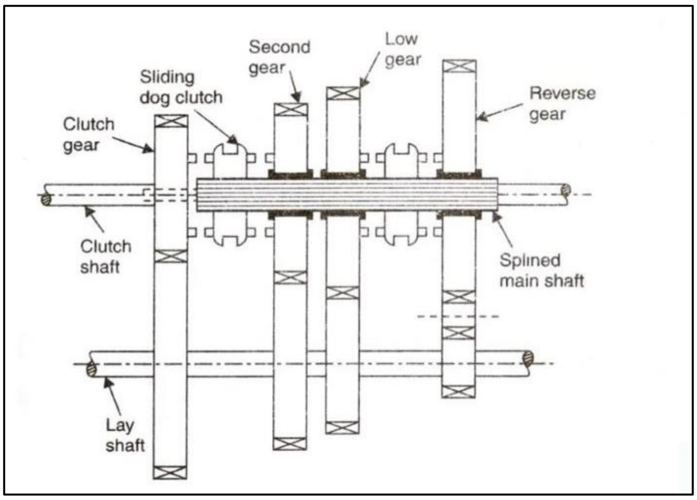


Figure 2:Constant mash gearbox[6]

3.2.3 Synchromesh gearbox

A synchromesh gearbox is like a constant mesh gearbox and the difference is instead of dog clutches, synchronizers are used in gear engagement and disengagement. This is the most advanced type among these. Unlike a constant mesh gearbox, there is no slip-in synchromesh gearbox. Because locking action is fully satisfied in the synchromesh gearbox while it is partially satisfied in the constant mesh gearbox[7].

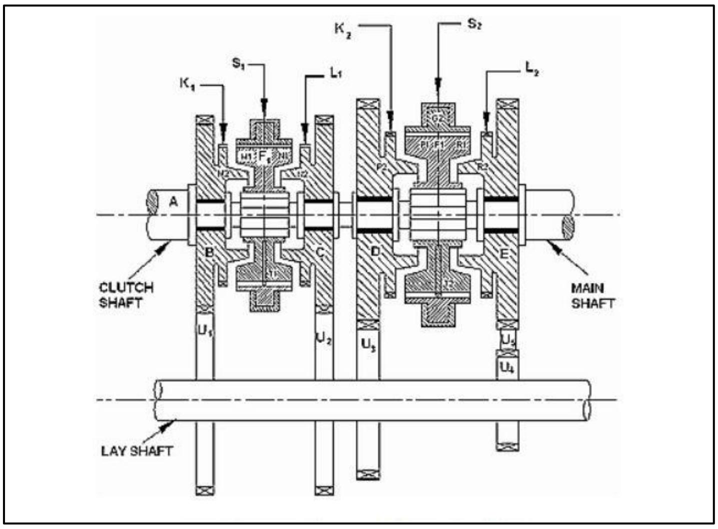


Figure 3:Synchromesh gearbox[8]

3.3 Essential components of a gearbox and their functionality

Mainly there are four components needed to fulfill the required function. These components are

3.3.1 Counter shaft

A countershaft is a shaft that connects with the clutch shaft directly. It contains the gear that connects it to both the clutch and main shafts. It can operate at engine speed or lower than engine speed depending on the gear ratio[9].

3.3.2 Main shaft

The shaft is the one that rotates at the vehicle's speed. It transfers power from the counter shaft via gears and depending on the gear ratio, it operates at a different speed and torque than the countershaft. This shaft connects to the universal shaft at one end[9].

3.3.3 Gears

Power is transferred from one shaft to another via gears. They are the most useful component of a gearbox because the torque variation of the countershaft and main shaft is determined by the gear ratio. The gear ratio is defined as the ratio of driven gear teeth to driving gear teeth[9].

3.3.4 Bearing

Bearings are essential whenever rotational motion interacts to maintain the turning portion and decrease friction. The bearing in the gear box supports both the counter and main shafts. The bearing in the gear box supports both the counter and main shafts[9].

3.4 Working principal of a gear box

A gear box connects the counter shaft to the clutch through a couple of gears. As a result, the counter shaft is always operating. When the counter shaft comes into touch with the main shaft via meshing gears, the main shaft begins to rotate in accordance with the gear ratio. When the driver wants to change the gear ratio, he or she simply presses the clutch pedal, which disconnects the counter shaft from the engine and connects the main shaft to the counter shaft through a different gear ratio via the gearshift lever. The gear teeth and other moving metal in a gear box must not come into contact. A tiny coating of lubricant must keep them apart at all times. This reduces wear and early failure. As a result, a gearbox is only half filled with lubrication oil[9].

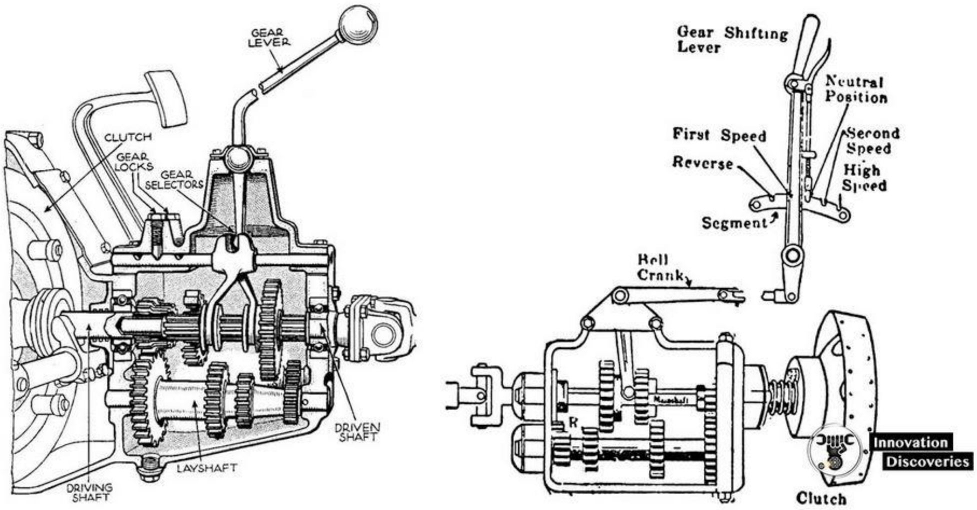


Figure 4 : Working of a principal gear box[10]

4 Design Calculations and Materials Selection

4.1 Prime mover selection

When we going to select prime mover want to consider maximum load. So, in my case study that in use in my third gear as a maximum.

Assume the mixture container in cylindrical one and center mixing blade like metal bar.

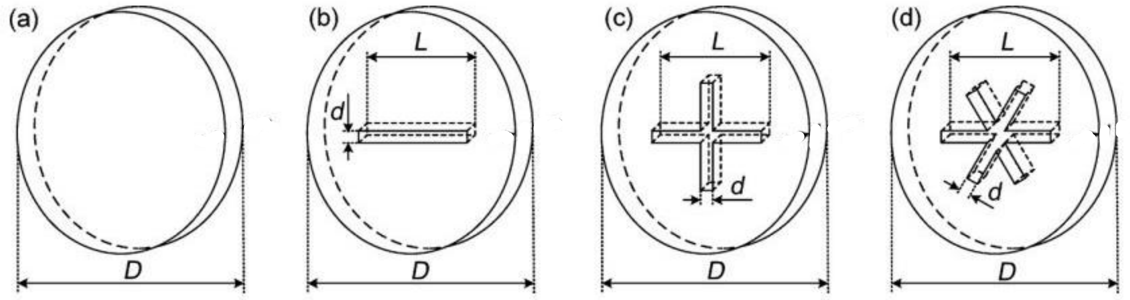


Figure 4

Also, center metal bar is assume like following.

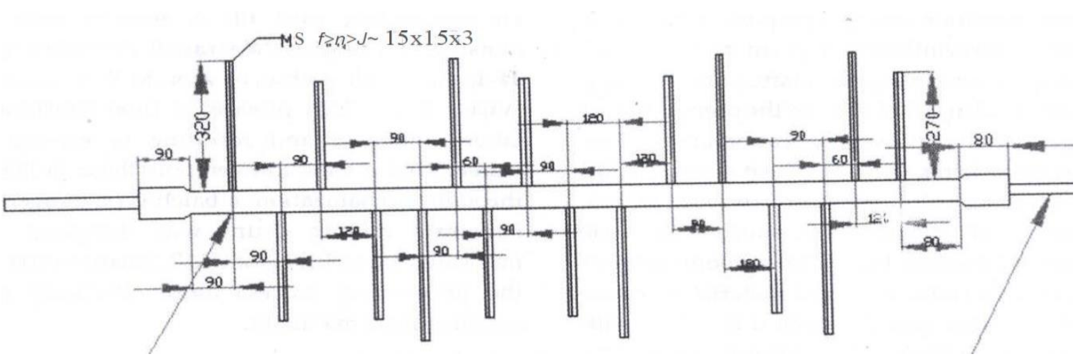


Figure 5

First, we calculate mass of the one bar. For that there is some assumption. I get blades are a constant cross section steel metal bar.

$$mass\ of\ the\ bar = \rho_{matal} A l$$

$$\text{Mean radius of blade} = 0.4$$

The density of stainless steel is approximately 7,500kg/m3 to 8,000kg/m3[11]

$$mass = \frac{8{,}000\text{kg}}{\text{m3}} \times (0.02 \times 0.05) \times 0.4$$

Assume there are 10 blades in the machine.

$$mass = \frac{8{,}000\text{kg}}{\text{m3}} \times (0.02\text{m} \times 0.05\text{m}) \times 0.4\text{m} \times 10 = 32\text{kg}$$

$$Substrate\ mass = saw\ dust + other\ indrediant + water$$

$$Maximum\ substrate\ mass = 100 + 19 \times 1 + 50 \times 1 = 169kg$$

$$Total\ mass = 169 + 32 = 201kg$$

Considering the critical moment of blade in third gear the mass in the mixture is maximum. Using paddy mixture machine date in internet. We can get the requested rpm for the mixture.

"A substrate mixer drum was developed and tested for pre-wetting of substrate in batch mode (Fig. 2). The capacity of substrate mixing drum was 30 kg dry weight of wheat straw/paddy straw. The initial moisture contents of air-dried wheat and paddy straws were at 10.7%and 9.6%, respectively. The effect of drum rotation speed and mixing time on substrate moisture content is shown in Figure 3. The highest moisture content (75.1%) was obtained for wheat straw for the 35rpm rotational speed and 15 min mixing time, whereas the minimum moisture content (40.2%)was found in 15 rpm drum speed and 5 inmixing time. This might probably be due to highest rotational speed imparting the mechanical actions like shredding, trampling, rubbing etc."[1]

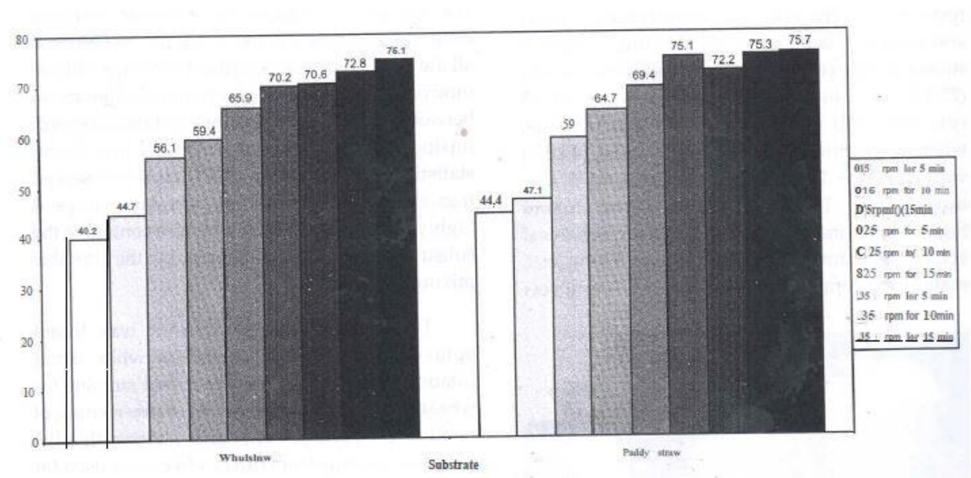


Figure 06

For the batter mixture, want to include water moisture content is 75.1%. There for it want to reach speed 35 rpm.

Also, we can assume haff of the mass acting as a point mass at the diameter of the cylinder.

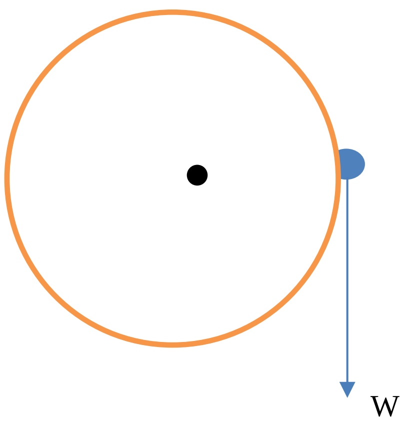


Figure -07

We can calculate requested toque using above assumption like in figure 07.

$$\tau = W \times R$$

$$\tau = \frac{201}{2} \times 9.81 \times 0.4 = 394.4\ Nm$$

There is maximum rpm 35rpm

$$anguler\ velocity = 35\ \times \frac{2\pi}{60} = 3.6652\ rads^{-1}$$

$$Mix\ power\ want\ to\ mix = \ \tau \times \omega = 394.4Nm \times 3.6652rads^{-1}$$

$$= 1445.41222Kw$$

Considering power loss and other things as a service factor (1.4)

$$1445.413 \times 1.35 = kw = 1951.307kw$$

So according to above calculation a 2 kW motor is enough to do this mixing process. There for I chose as prime mover 2kW motor. **TEC Electric 2kW 3ph 4 Pole Premium Efficiency foot mounting 100L frame AC motor.**

Details

TEC Electric - 2.2kW 3ph 4 Pole Premium Efficiency Foot Mounting (B3) 100L Frame AC Motor for 230V or 400V 3 phase supply. Inverter Rated for use with a Variable Frequency Inverter Drive having 1ph or 3ph input and 3ph output, or a fixed frequency mains supply at 50Hz.

Fixed mains supply output: 2.2kW (3HP), 1450RPM at 230V or 400V, 50Hz 3ph.
Inverter Supply Output: 20:1 Speed Control range from 145RPM to 2030RPM for 5Hz to 70Hz control, with suitable de-rating at the lower speeds.

Full Load Current: 7.93A at 230V or 4.58A at 400V.
Power Factor: 0.8 when mains connected at 50Hz.
Efficiency: 86.7%
Full Load Torque: 14.4Nm

198mm Wide x 369mm Long x 252mm high overall in its IP55 enclosure.
Foot mount on 10mm holes at 160mm wide x 140mm centres 63mm back from the shaft shoulder.
Shaft is 28mm dia x 60mm with 8mm wide key.
Weight is 23.9kg.

Rated at 40C Ambient.
Ventilation space required at rear cooling air intake.
Thermistor for over-temperature protection.
Manufacturer Part Number: 2.243TECAB3-IE3. Frame Ref: T3A 100L4A-TECA

Extended Warranty Available: When a TEC AC Motor is used with a TECDrive AC Inverter (see linked products below) the warranty period is 2 years (from dispatch).
Standard warranty is 1 year.
For the extended warranty to be valid the TEC AC Inverter and TEC Motor must be on the same order and used together when on site.

From Jan 1st 2017, three phase electric induction motors with a rated output of 0.75kW to 375kW and efficiency less than IE3 must be equipped with a variable speed drive (Inverter Drive). For exceptions, see EC Commission Regulation 640/2009.

TEC Drawing IE3 100L B3 (226 kB)
TEC Datasheet IE3 TA100 2.2kW 4 Pole (167 kB)

Figure 08 - motor parameters

4.2 Power transmission methods

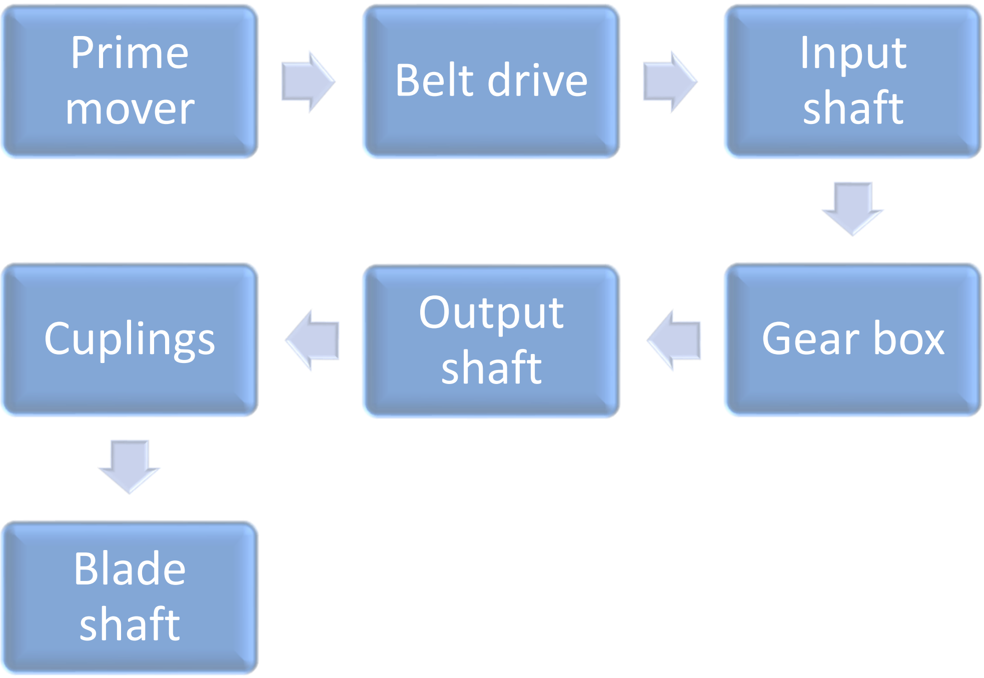


Figure 10- Power Transmission methods in flow chat

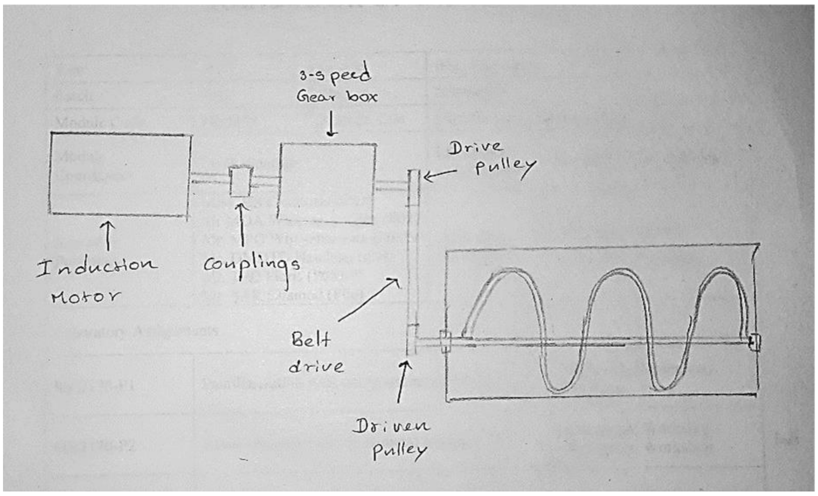


Figure 11-Power Transmission methods in actual chart

4.3 Gearbox type selection

Gearbox types such as sliding mesh, constant mesh and synchro-mesh are the types that are available in the industry. Most of the automotive and locomotive applications use constant mesh and synchromesh type gearboxes because it allows them to shift gears while moving thus achieve higher speeds. In earlier stages of automotive industry, sliding mesh type gearboxes are used.

Sliding mesh gearbox is selected because the mechanism is very simple and easy to design and cheap to manufacture. And, in this type of gear box fluctuating loads are less in a sliding mesh as only one gear is meshed at a time. Although efficiency of constant mesh and synchronous mesh type gear box has higher efficiency than sliding mesh gear box, designing, manufacturing, and assembling is difficult in constant mesh and synchronous mesh type gear boxes.

The major issue of sliding mesh gear box is the fact that speeds of the input and output shafts should matched when changing gears. Otherwise, gearwheels do not align and crash into one another. This will not be a problem as our application does not require to change gears will in the move.**[12]**

4.4 Gear ratio selection

mass	169kg	119kg	100kg
toque	394.4Nm	296.3Nm	259Nm
power	2KW	2KW	2KW
Service factor	1.35	1.35	1.35
Angular velocity	3.6651914	4.878831	5.5810869
Motor output rpm	1440	1440	1440
Gear reduction	2.742857	2.0605544	1.80128

Assume belt reduction =5 and lay shaft reduction =3

Mass (kg)	Blade mass (Kg)	Total mass (kg)	Blade Dimeter (m)	G (ms-2)	Toque (Nm)	Output Speed (rpm)	Out put speed (rads-1)	Power (W)	Motor rpm		Reduction	Gear box input rpm	Gear Ratio
100	32	132	0.4	9.81	259	53.29545455	5.5810869	1445.412222	1440	Reduction in v-belt	5	96	1.8012793
119	32	151	0.4	9.81	296.3	46.58940397	4.878831	1445.412222		Reduction in lay shaft	3	96	2.0605544
169	32	201	0.4	9.81	394.4	35	3.6651914	1445.412222				96	2.7428571
50	32	82	0.4	9.81	160.9	50	5.2359878	842.3866541				96	1.92

Initial calculation

4.5 Gear type selection

Since sliding mesh gear box was selected, spur gear type is most suitable gear type . spur gears are very simple and compact therefore it is make easy to design , manufacturing and assembling . in other hand this type of gears changes the shaft speed with high accurate at a constant velocity. And another advantage of spur gear is reliability. Spur gears have a low risk of premature failure since they are durable and unlikely to slip during operation. the other major advantage of using spur gear in this application is cost effectiveness. The simplicity of design and manufacturing make less expensive to fabricate and purchase. in other hand efficiency of spur gear is very high. In number it is between 95%-99%.

4.6 Tooth profile selection

In gear tooth designing, a fundamental law called ‘law of gearing’ must be satisfied to maintain a constant velocity ratio between gears. There are 2 types of commonly used gear tooth profiles that satisfies the above condition

1. Involute gears
2. Cycloidal gears

4.6.1 Involute gears

An involute of a circle is a plane curve generated by a point on a tangent, which rolls on the circle without slipping or by a point on a taut string which is unwrapped from a reel as shown in Figure.

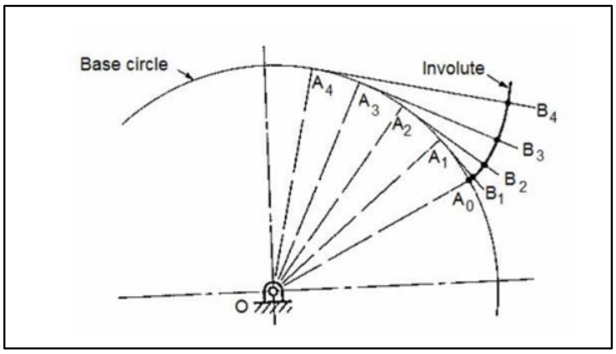
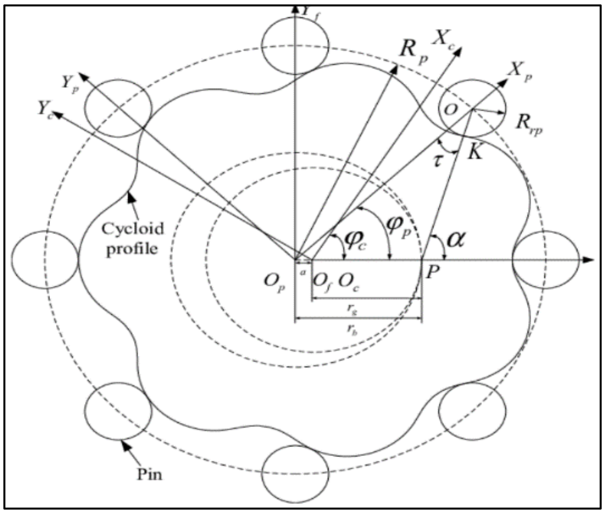


Figure 1.3.1: Involute gears

4.6.2 Cycloidal gears

A ***cycloid*** is the curve traced by a point on the circumference of a circle which rolls without slipping on a fixed straight line. When a circle rolls without slipping on the outside of a fixed circle, the curve traced by a point on the circumference of a circle is known as ***epicycloid*.** On the other hand, if a circle rolls without slipping on the inside of a fixed circle, then the curve traced by a point on the circumference of a circle is called ***hypocycloid***.



Cycloidal gears

In involute gears, centre distance of a gear pair can be varied within limits without changing the velocity ratio but for cycloidal gears, the exact centre distance must be maintained. Moreover, pressure angle remains same from the start of the tooth engagement to the end of the tooth engagement for involute profile. For cycloidal profile this is not the same. Therefore, considering above factors, **involute gear** tooth profile is selected

4.6.3 Pressure angle

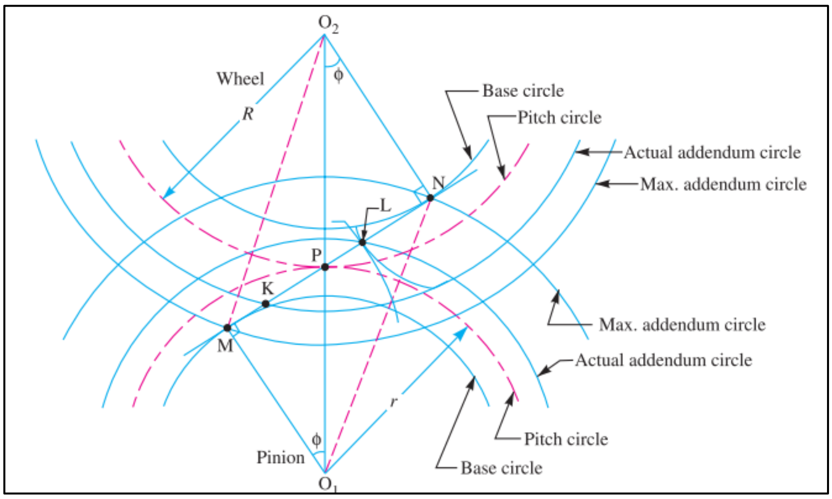
There are 4 commonly used pressure angle and tooth profiles

1. 14 ½ composite system
2. 14 ½full depth involute system
3. 20^0 full depth involute system
4. 20^0 stub involute system

14 ½ composite system is used for general-purpose gears. When the pressure angle increases, tooth strength increases. Thus, 20^0 pressure angle has stronger teeth. Stub involute system is stronger than full depth involute system. Therefore, they are used for heavy loads. According to above factors, **20^0 full depth involute system** is selected. The standard proportion of gear system can be seen in Figure

S. No.	*Particulars*	*$14\frac{1}{2}°$ composite or full depth involute system*	*20° full depth involute system*	*20° stub involute system*
1.	Addendum	1*m*	1*m*	0.8 *m*
2.	Dedendum	1.25 *m*	1.25 *m*	1 *m*
3.	Working depth	2 *m*	2 *m*	1.60 *m*
4.	Minimum total depth	2.25 *m*	2.25 *m*	1.80 *m*
5.	Tooth thickness	1.5708 *m*	1.5708 *m*	1.5708 *m*
6.	Minimum clearance	0.25 *m*	0.25 *m*	0.2 *m*
7.	Fillet radius at root	0.4 *m*	0.4 *m*	0.4 *m*

Standard Proportion of gear



Error! Use the Home tab to apply 0 to the text that you want to appear here..1:Interference in involute gears

4.7 Minimum no. of teeth on the pinion to avoid interference

Interference of the gear teeth will take place if there is a contact between faces that are not in involute shape. The involute shape is only defined outside of the base circle of the gear. Therefore, to avoid interference the contact of the gear teeth should be only take place between involute faces. If interference is present, speed ratio will not remain constant and high tooth wear will occur. The minimum no. of teeth that should be in the pinion (T1) to avoid interference can be calculated using the gear ratio (G) and the pressure angle (ψ) as follows.

$$\boldsymbol{T_1} \geq \frac{\mathbf{2}}{\sqrt{\boldsymbol{G^2+Sin(\psi)^2(1+2G)}}-\boldsymbol{G}} \quad \text{Equation 1.4.1}$$

Based on the gear ratio, the number of teeth in the wheel (T_2) can be calculated.

$$\boldsymbol{T_2 = T_1 \times G} \quad \text{Equation 1.4.2}$$

4.8 Module Selection

According to AGMA (American Gear Manufacturers Association) following is the standard values for the module of a gear teeth and two meshing gears should be equal for a proper mesh between them.

Set I: 1.0, 1.25, 1.5, 2.0, 2.5, 3.0, 4.0, 5.0, 6.0, 8.0, 10.0, 32.0, 40.0

Using iteration As a module, **3mm** is chosen for the application.

4.9 Gear material selection

Gear material selection is an important process of the gear design, because all the following calculations are based on that. The size of the gear and the face width is highly affected by the material properties.

The material used must be strong enough to prevent tooth breakages, must be hard enough to resist contact stresses, and ductile enough to absorb shock loads. Case hardening is done to reduce the tooth wear and usual practice is to use a material of 50 B. H. N. harder than the wheel as the pinion material. There are few methods that are used to increase the hardness of the material.[13]

1. Through hardening
2. Induction hardening
3. Nitriding
4. Case carburized and hardened

In through hardening, the material is heat treated before machining. Hardness cannot be increases to a higher value because it is yet to machine[4]. In induction hardening, gear is hardened after cutting gear teeth and by this method a ductile core can be obtained. Nitriding is a low temperature process and high hardness can be achieved. Disadvantage of this method is it is time consuming. Case carburizing and hardening can obtain higher hardness values and it also strengthen the core of the gear.

According to calculation this **AISI 1040 Q&T 650^{0}C** is selected for wheels of reverse-idler, and lay-output gear pairs. Moreover, **ASTM class 50 Standard gray iron** is chosen for input-lay gear pair.

The allowable static stress (σ_o) for steel gears is approximately one-third of the ultimate tensile strength (σ_u).

$$\sigma_o = \frac{\sigma_u}{3}$$

1 AISI No.	2 Treatment	3 Temperature °C (°F)	4 Tensile Strength MPa (kpsi)	5 Yield Strength, MPa (kpsi)	6 Elongation, %	7 Reduction in Area, %	8 Brinell Hardness
1030	Q&T*	205 (400)	848 (123)	648 (94)	17	47	495
	Q&T*	315 (600)	800 (116)	621 (90)	19	53	401
	Q&T*	425 (800)	731 (106)	579 (84)	23	60	302
	Q&T*	540 (1000)	669 (97)	517 (75)	28	65	255
	Q&T*	650 (1200)	586 (85)	441 (64)	32	70	207
	Normalized	925 (1700)	521 (75)	345 (50)	32	61	149
	Annealed	870 (1600)	430 (62)	317 (46)	35	64	137
1040	Q&T	205 (400)	779 (113)	593 (86)	19	48	262
	Q&T	425 (800)	758 (110)	552 (80)	21	54	241
	Q&T	650 (1200)	634 (92)	434 (63)	29	65	192
	Normalized	900 (1650)	590 (86)	374 (54)	28	55	170
	Annealed	790 (1450)	519 (75)	353 (51)	30	57	149
1050	Q&T*	205 (400)	1120 (163)	807 (117)	9	27	514
	Q&T*	425 (800)	1090 (158)	793 (115)	13	36	444
	Q&T*	650 (1200)	717 (104)	538 (78)	28	65	235
	Normalized	900 (1650)	748 (108)	427 (62)	20	39	217
	Annealed	790 (1450)	636 (92)	365 (53)	24	40	187

Mechanical properties of AISI 1040 steel**[14]**

ASTM class 50 Standard gray iron test bars, as cast

Categories: Metal; Ferrous Metal; Cast Iron; Gray Cast Iron

Material Notes: 1638 kg transverse load on test bar.

Vendors: No vendors are listed for this material. Please click here if you are a supplier and would like

Printer friendly version Download as PDF Download to Excel (requires Excel and Windows)
Export data to your CAD/FEA program

Physical Properties	Metric
Density	7.15 g/cc

Mechanical Properties	Metric
Hardness, Brinell	262
Hardness, Knoop	292
Hardness, Rockwell C	22.6
Hardness, Vickers	277
Tensile Strength, Ultimate	362 MPa
Modulus of Elasticity	130 - 157 GPa
Ultimate Compressive Strength	1130 MPa
Poissons Ratio	0.29
Fatigue Strength	148 MPa
Shear Modulus	50.0 - 55.0 GPa
Shear Strength	503 MPa

Input shaft - lay shaft gear pair material

4.10 Calculation of Gear teeth for input and lay shaft gears pair

Considered as an example calculation for the input shaft and lay shaft gear pair. For that following equation are used.

- $T_p \geq 2 / \{[G^2 + \text{Sin}^2(\psi)\ (1+2G)]^{1/2} - G\}$
- P.C.D. of the gear = $d = m\ T$
- 0.154 – 0.912 / T, for 20^0 full depth system
- Strength of the teeth = ($\boldsymbol{\sigma_o}\ \boldsymbol{y}$) $b\ m\ \pi\ C_v$
- The pitch line velocity ($\boldsymbol{v}$) = $\omega\ d$ / 2
- Velocity factor ($\boldsymbol{C_v}$) = 3 / (3+v)

4.10.1 Determining the no. of Teeth in Pinion

Need to find minimum no. of teeth for interference for the pinion according to the following equation,

$$T_1 \geq 2 / \{[G^2 + \text{Sin}^2(\psi)\ (1+2G)]^{1/2} - G\} = 14.9809 = 15$$

Calculated minimum no. of teeth for interference is 15. Taking the no. of teeth close to the minimum requirement will be beneficial in manufacturing process. But taking 16 as the input pinion no. of teeth clearance for the reverse pinion and gear will be not sufficient. Therefore no. of teeth of the input pinion is selected to be 22. Center distance should be calculated using input and lay shaft gears and center distance calculated can be carried out to the rest of the gears as well.

4.10.2 Determining the Center Distance

Using the Excel spreadsheet, a **3mm module** was chosen after many iterations.

Number of teeth of gear = 22

Pitch circle diameter of pinion = 3 × 22 = 66mm

Number of teeth of gear = 22× 3 = 66

Pitch circle diameter of gear = 3 × 66 = 198mm

- Therefore, center distance between shafts = $\frac{\mathbf{D_1 + D_2}}{\mathbf{2}}$ = **132 mm**

Using gear ratio and center distance, P.C.D. of other gears can be calculated

For pinion and gear wheel of 1st gear,

Let x be the P.C.D. of the pinion,

Then P.C.D. of the gear = G x x = 1.8013x

Since center distance should be the same

$$x + 1.8013x = 132 * 2 \quad x = 94.241$$

Therefore, P.C.D. of the pinion = 94.24mm =96mm

P.C.D. of the gear = 169.75mm =168mm

Teeth in pinion = d / m = 96 / 3 = 32

Teeth in gear = D / m = 168 / 3 = 56

These final diameters should align with the center distance calculated previously.

Likewise calculating for other gears,

Gear	Pinion Teeth	Wheel Teeth	Pinion Diameter	Gear Diameter
Input gear /lay gear	22	66	66	198
1 Gear	32	56	96	168
2 Gear	29	59	87	177
3 Gear	24	64	72	192

4.10.3 Determining the teeth strength using Lewis's equation

From service factor (C_S) and transmission power (P_T), design power (P_D) can be calculated. P_T is selected as 2000W and C_S as 1 (steady and 8-10 hour working time per day).[15]

Type of load	*Type of service*		
	Intermittent or 3 hours per day	*8-10 hours per day*	*Continuous 24 hours per day*
Steady	0.8	1.00	1.25
Light shock	1.00	1.25	1.54
Medium shock	1.25	1.54	1.80
Heavy shock	1.54	1.80	2.00

Values of service factor.

Design Power = $\boldsymbol{P_d} = \boldsymbol{C_s} \times \boldsymbol{P_t}$ = 1 x 2000W = 2000 W

Design Power = $\boldsymbol{P_d} = \boldsymbol{W_t} \times (\boldsymbol{d_1}/2) \times \boldsymbol{\omega_1}$

Therefore, tangential tooth load = $\boldsymbol{W_t}$ = 2.00953kN

Allowable static stress of pinion (Cast Iron) = $(\sigma_o)_p$ = 105 MPa

The face width (b) may be taken as 3 pc to 4 pc (or 9.5 m to 12.5 m) for cut teeth and 2 pc to 3 pc (or 6.5 m to 9.5 m) for cast teeth. Face width is selected as **3pc** because it is suitable for above two gear manufacturing method.

$$Face\ width = 3 \times \pi \times m = 3 \times \pi \times 3 = 28.27mm$$

Lewis Form Factor of the pinion ($\boldsymbol{y_p}$) $= 0.154 - 0.912 / \boldsymbol{T_1}$

$= 0.112545$

The pitch line velocity ($\boldsymbol{v}$) $= \boldsymbol{\omega_1 d_1 / 2}$

$= 2\pi \times 288 \times 33/60$

$= 0.99525 ms^{-1}$

Therefore, velocity factor ($\boldsymbol{C_v}$) $= \boldsymbol{3 / (3+v)}$

$= 0.75089 sm^{-1}$

Using Lewis Equation,[16]

Strength of the pinion teeth $= (\boldsymbol{\sigma_o\, y})_p\, \boldsymbol{b\, m\, \pi\, C_v}$

$= 105 \times 10^6 \times 3\boldsymbol{\pi} \times (3 \times 10^{-3})^2 \times \boldsymbol{\pi} \times 0.75089 \times 0.1125$

$= 2.3645$ kN

Strength of the wheel teeth $= (\boldsymbol{\sigma_o\, y})_p\, \boldsymbol{b\, m\, \pi\, C_v}$

$= 105 \times 10^6 \times 3\boldsymbol{\pi} \times (3 \times 10^{-3})^2 \times \boldsymbol{\pi} \times 0.75089 \times 0.14018$

$= 2.945$ kN

Strength of the pinion teeth is lower than the strength of the wheel teeth. Also consider tangential tooth load and strength of pinion teeth see this is safe. Using the tangential tooth load and pinion strength, conclude that the gear is secure.

4.10.4 Determining the Dynamic Load[15]

$$W_D = W_T + W_I = W_T + \frac{21v(bC+W_T)}{21v+\sqrt{bC+W_T}}$$

- W_D = Total dynamic load
- W_T = Steady load due to transmitted torque
- W_I = Increment load due to dynamic action
- v = Pitch line velocity
- b = Face width of gears
- C = A deformation or dynamic factor

For find Deformation factor first want to get e = Tooth error action in mm. Below table can be use to datamined the tooth error. Module is 3mm and gears are first-class commercial gears, deformation factor is selected as **0.051mm**.

Module (m) in mm	Tooth error in action (e) in mm		
	First class commercial gears	Carefully cut gears	Precision gears
Upto 4	0.051	0.025	0.0125
5	0.055	0.028	0.015
6	0.065	0.032	0.017
7	0.071	0.035	0.0186
8	0.078	0.0386	0.0198
9	0.085	0.042	0.021
10	0.089	0.0445	0.023
12	0.097	0.0487	0.0243
14	0.104	0.052	0.028
16	0.110	0.055	0.030
18	0.114	0.058	0.032
20	0.117	0.059	0.033

Values of tooth error in action (e) verses module

Deformation factor can be determined using the following table,[15]

Material		Involute tooth form	Values of deformation factor (C) in N-mm				
			Tooth error in action (e) in mm				
Pinion	Gear		0.01	0.02	0.04	0.06	0.08
Cast iron	Cast iron	$14\frac{1}{2}°$	55	110	220	330	440
Steel	Cast iron		76	152	304	456	608
Steel	Steel		110	220	440	660	880
Cast iron	Cast iron	20° full depth	57	114	228	342	456
Steel	Cast iron		79	158	316	474	632
Steel	Steel		114	228	456	684	912
Cast iron	Cast iron	20° stub	59	118	236	354	472
Steel	Cast iron		81	162	324	486	648
Steel	Steel		119	238	476	714	952

The deformation factor for a cast iron pinion and gear with 20^0 full depth involute teeth and a module of 3mm and a tooth error of 0.051 is **290.4 Nmm**. (Using above table and interpolation)

$\boldsymbol{W_T}$ = 2.00953kN

v = Pitch line velocity = 0.75089 ms^{-1}

$$W_I = \frac{21\ x\ 0.9952\ x\ (28.2743*290.4/1000\ +2.00953)}{21\ x\ 0.9952 + \sqrt{28.2743*\frac{290.4}{1000}\ + 2.00953}}$$

$$W_D = W_T + W_I = 2.0184\ kN$$

Load on gear teeth = $\boldsymbol{W_t}$ = 2.0184 kN < $\{\boldsymbol{\sigma_o\ y\ b\ m\ \pi\ C_v}\}_{\boldsymbol{p}}$ = 2.365 kN Strength of the gear teeth. Therefore, **the teeth of this spur gear with a module of 3 mm will have sufficient strength to transmit the specified power required**.

4.10.5 Determining the static tooth load

$$W_s = \sigma_e \times b \times p_c \times y = \sigma_e \times b \times \pi \times m \times y$$

σ_e – flexural endurance limit

b – Face width

p_c – $\pi \times m$

y – Lewis form factor

Flexural endurance limit can be get below table.

Material of pinion and gear	*Brinell hardness number (B.H.N.)*	*Flexural endurance limit (σ_e) in MPa*
Grey cast iron	160	84
Semi-steel	200	126
Phosphor bronze	100	168
Steel	150	252
	200	350
	240	420
	280	490
	300	525
	320	560
	350	595
	360	630
	400 and above	700

Flexural endurance limit

Face width =28.2743mm

y – Lewis form factor = 0.112545

$$W_s = 84 \times 10^6 \times \frac{28.2743}{1000} \times \pi \times 3 \times 0.112545 = 2.5192kN$$

W_D =2.0184kN < Ws = 2.5192kN therefore tooth is safety against static load,

4.10.6 Determining Limiting Load for Wear

$$W_W = D_p \times b \times Q \times K$$

- Ww = Maximum or limiting load for wear
- D_P = Pitch circle diameter of the pinion
- b = Face width of the pinion
- Q = Ratio factor = $\frac{2T_G}{T_G + T_P}$
- $V.R.$ = Velocity ratio = T_G / T_P
- K = Load-stress factor where,

$$K = \frac{(\sigma es)^2 \sin \varphi}{1.4} \left(\frac{1}{E_P} + \frac{1}{E_G}\right)$$

- σ_{es} = Surface endurance limit
- φ = Pressure angle
- EP = Young's modulus for the material of the pinion
- EG = Young's modulus for the material of the gear

$$K = \frac{(630 \text{ x } 10^6)^2 \sin 20}{1.4}\left(\frac{1}{125 \times 10^9} + \frac{1}{125 \times 10^9}\right) = 1.5514$$

Face width =28.2743mm

$$Q = \frac{2T_G}{T_G + T_P} = \frac{2 \times 66}{22 + 66}$$

$$W_w = 66 \times 28.2743 \times 1.5 \times \frac{1.5514}{1000} = 4.34261kN$$

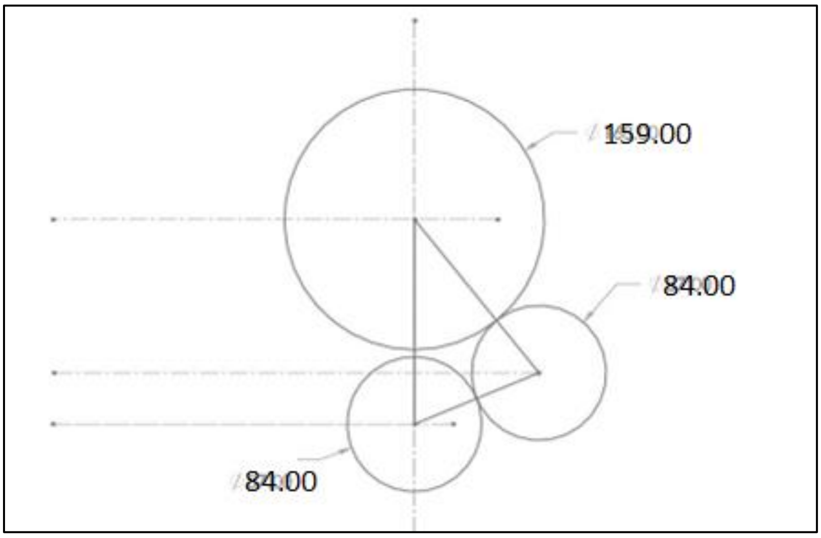
Load on gear teeth (W_t = 2.0184 kN)< Limiting load for wear (W_W = 4.34261 kN) . Therefore, the teeth of this spur gear with a module of 3 mm will have sufficient limiting load for wear to transmit the specified power required in this mixer machine.

Therefore, input pinion design is safe for this application. Continuing the same procedure for other teeth to determine whether teeth size is enough to overcome the dynamic and wear loads.

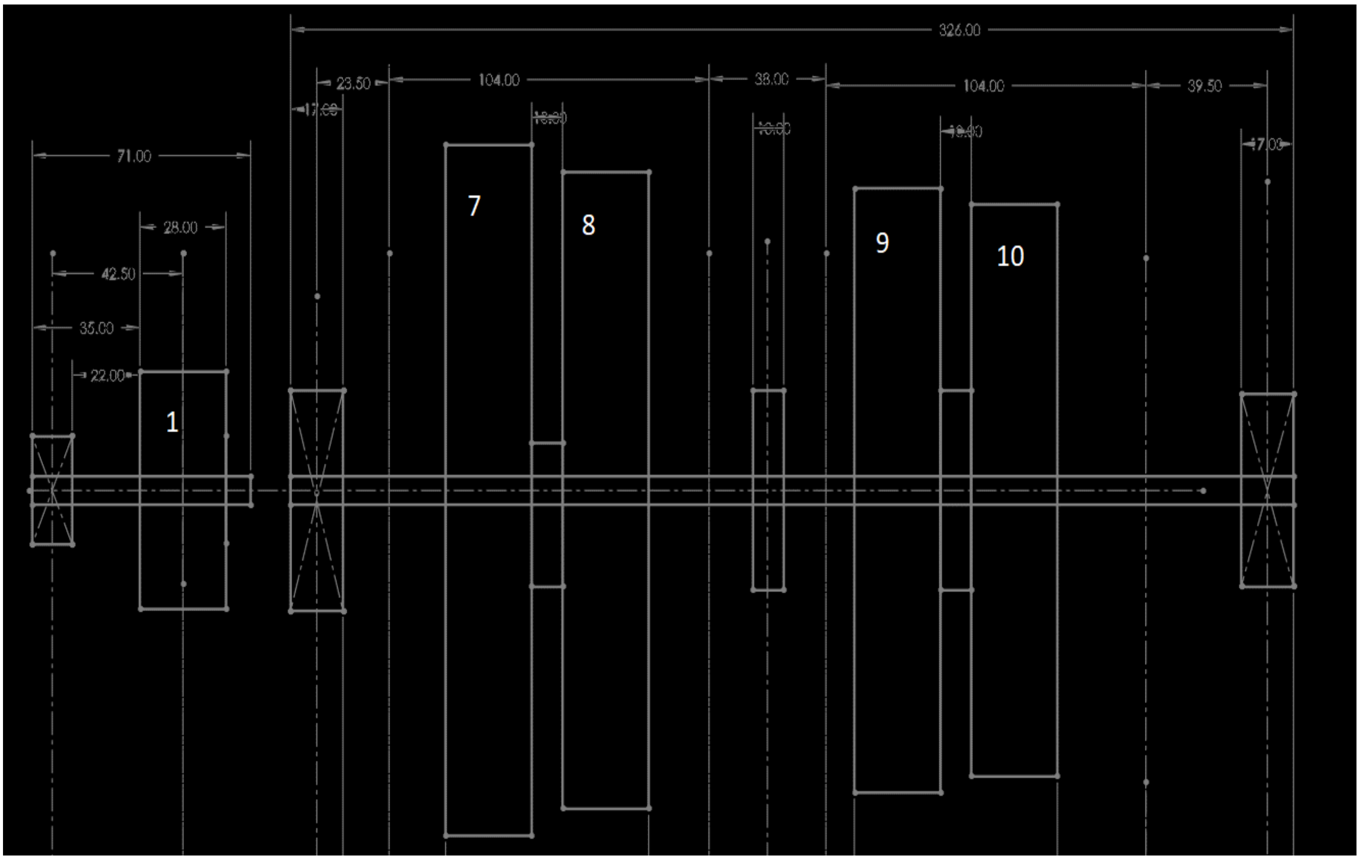
Idler gear is assumed to be of the same size of the pinion gear for reverse since it will face same forces as the pinion and if the pinion doesn't fail then idler gear will not either. Also gear ratio will be equal to 1. Therefore, idler will have same angular velocity which means the same torque will be transmitted. This will make the calculations easier.

4.10.7 Gear ratio calculation for reverse gear

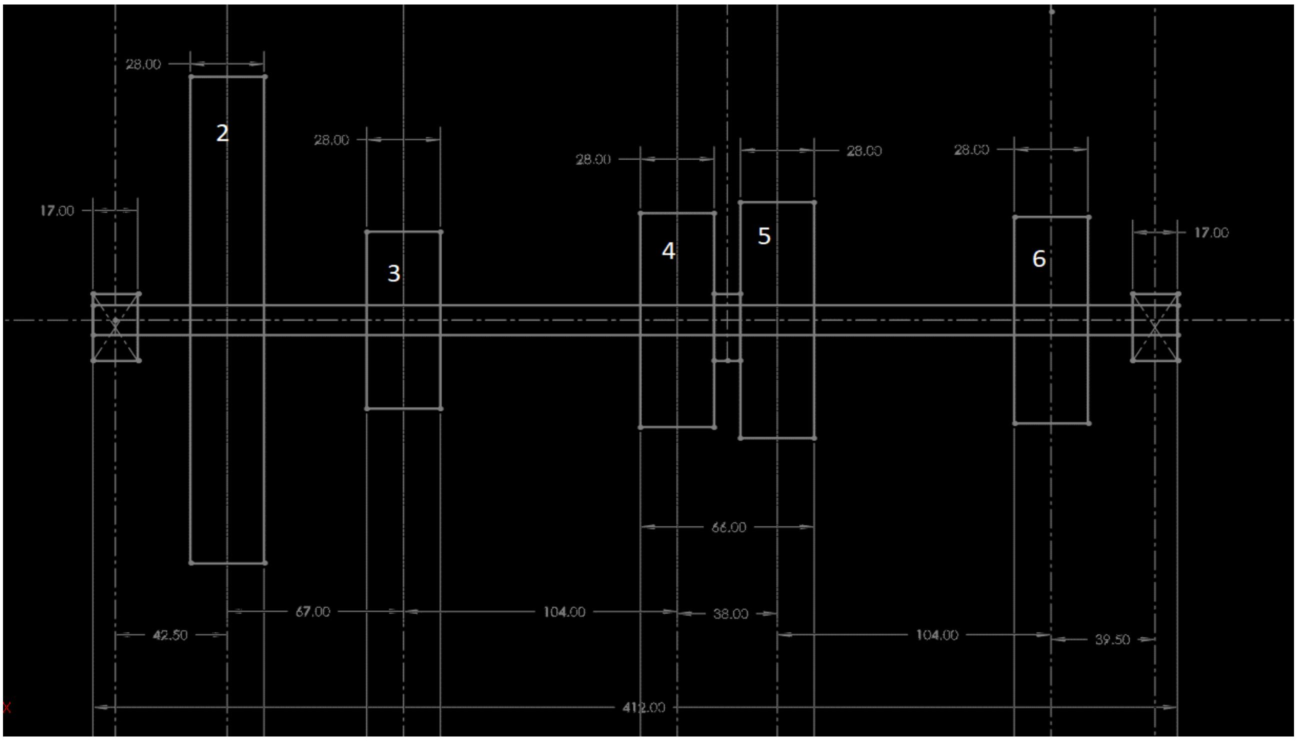
Gear box is designed to use same wheel for first gear and reverse gear. In above calculation first gear wheel tooth is 53(159mm). Also, the gear ratio is selected 1 between input reverse gear and ideal wheel. Therefore, 28(84mm) teeth selected for input reverse gear wheel and ideal wheel. For that use calculate center distance.



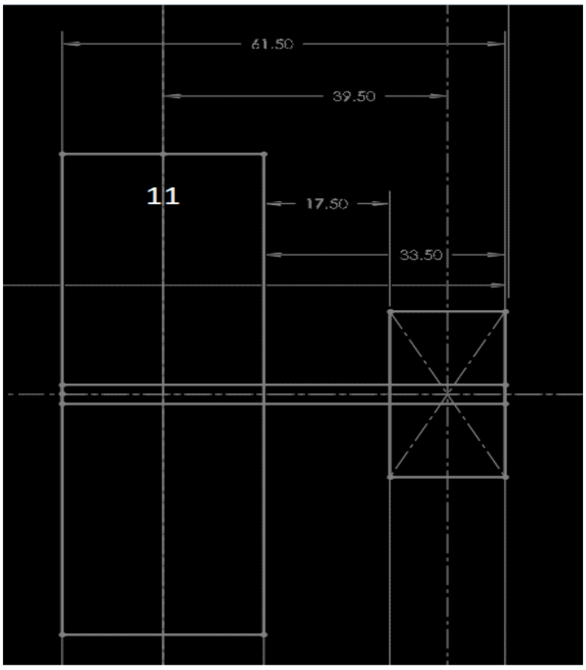
Reverse gear configuration



4.10.7-1 Input and output shaft



4.10.7-2 Lay shaft



4.10.7-3 Idler gear configuration

After the calculation final result for each gear include below table

Gear	Tangential tooth load(kN)	Dynamic load(kN)	Static Tooth Load (kN)	Wear Load (kN)	Suitability
1	2.00953211	2.0184	2.5193	4.34262479	SAFE
2	2.00953211	2.0184	3.13786	13.0278	SAFE
3	5.526213302	5.5397	7.7897	5.9904	SAFE
4	4.573417905	4.5874	8.2296	4.7776	SAFE
5	4.144659976	4.11588	8.4276	4.3872	SAFE
6	4.573417905	4.5874	8.2296	4.9764	SAFE
7	5.68410511	5.6976	9.3846	10.0280	SAFE
8	4.632026012	4.6460	9.3035	8.52234	SAFE
9	4.266109103	4.28026	9.2478	7.6777	SAFE
10	4.547284088	4..5613	9.2478	7.9467	SAFE

4.11 Conclusion

Gear	Number of Teeth (T)	P.C.D. (mm) [d]	Module (mm) [m]	Face Width (mm) [b]	Material
1	22	66	3	28.27	ASTM class 50 Standard gray iron
2	66	198	3	28.27	ASTM class 50 Standard gray iron
3	24	72	3	28.27	AISI 1040 Q&T 6500
4	29	87	3	28.27	AISI 1040 Q&T 6500
5	32	96	3	28.27	AISI 1040 Q&T 6500
6	29	87	3	28.27	AISI 1040 Q&T 6500
7	64	192	3	28.27	AISI 1040 Q&T 6500
8	59	118	3	28.27	AISI 1040 Q&T 6500
9	56	168	3	28.27	AISI 1040 Q&T 6500
10	29	87	3	28.27	AISI 1040 Q&T 6500

Final details about gears

4.12 Shaft Calculations

Before beginning of the shaft design process, the layout of the gearbox needs to be finalized. The distance between gears in the same shaft must be determined. The distance between gears of the same shaft is determined as when gearbox is in the neutral gear. First and second gears in the lay shaft is attached together to reduce the shaft length. Second and third gear in the output shaft is attached together to ease the shifting mechanism. Then the placement of the gears when running in a gear in output shaft is also can be determined. Required total shaft length for lay shaft is calculated as **326 mm**.

To calculate the required diameter of the shaft, following procedure is followed.

$$\tau = \frac{P_T}{\omega}$$

$$F_T = \frac{2\tau}{D}$$

$$F_R = F_T \times \tan(20^0)$$

Using above equation can calculate F_R and F_T. Banding moment is calculated using these values. Input shaft, output shaft, and ideal shaft act like cantilever beam. Therefore, Same procedure is used to calculate bending moment. Max bending moment is acted on the support point. After find the max BM value we can calculate minimum dimeter

4.12.1 Shaft material selection

Shaft material was selected to be **ASTM 1040 Carbon Steel Cold Draw** due to its balance between cost effectiveness and shear strength.

		Yield Strength (MPa)	Shear Strength (MPa)
ASTM 1040 Carbon Steel	Cold Draw	530	390

Material properties

4.12.2 Required minimum diameter of the shaft[15]

Since the shaft is made from ductile materials, to calculate the required minimum diameter of the shaft,

$$\tau_{allowable} \geq \frac{16}{\pi d^3} \times \sqrt{(K_m \times M)^2 + (K_T \times T)^2}$$

$\tau_{allowable}$: Allowable shear stress

d: Shaft diameter

K_M: Shock fatigue factor for bending

K_T: Shock fatigue factor for torsion

M: Bending moment of the shaft

T: Torsion of the shaft (Torque transmitted)

Shafts made of brittle materials are designed using the maximum normal stress criterion of failure.

$$\tau_{allowable} \geq \left(\frac{16}{\pi d^3}\right)\left[(K_m M_b) + \sqrt{(K_m \times M)^2 + (K_T \times T)^2}\right]$$

Nature of load	K_m	K_t
1. Stationary shafts		
(*a*) Gradually applied load	1.0	1.0
(*b*) Suddenly applied load	1.5 to 2.0	1.5 to 2.0
2. Rotating shafts		
(*a*) Gradually applied or steady load	1.5	1.0
(*b*) Suddenly applied load with minor shocks only	1.5 to 2.0	1.5 to 2.0
(*c*) Suddenly applied load with heavy shocks	2.0 to 3.0	1.5 to 3.0

Here Shock fatigue factor for bending and torsion is used to account the dynamic and fatigue effect of the shaft operation. K_M is selected as 1.5 and K_T is selected as 1.0 by considering load as a gradually applied rotating shaft.

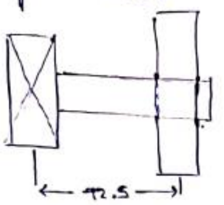
Using above equation can calculate dimeter all calculation includes in annex.

4.12.3 Find maximum bending moment on shaft

Using below method we can analysis maximum bending moment of the shafts. Lay shaft and output shaft is simply supported beam and other shaft are cantilever beam.

Cantilear Beam.

For input shaft.



F_t - 2009.5321

F_R - 731.4098

Maximum Bending moment (Horizontal) $= F_t \times l$

$= 2009.5321 \times 42.5$

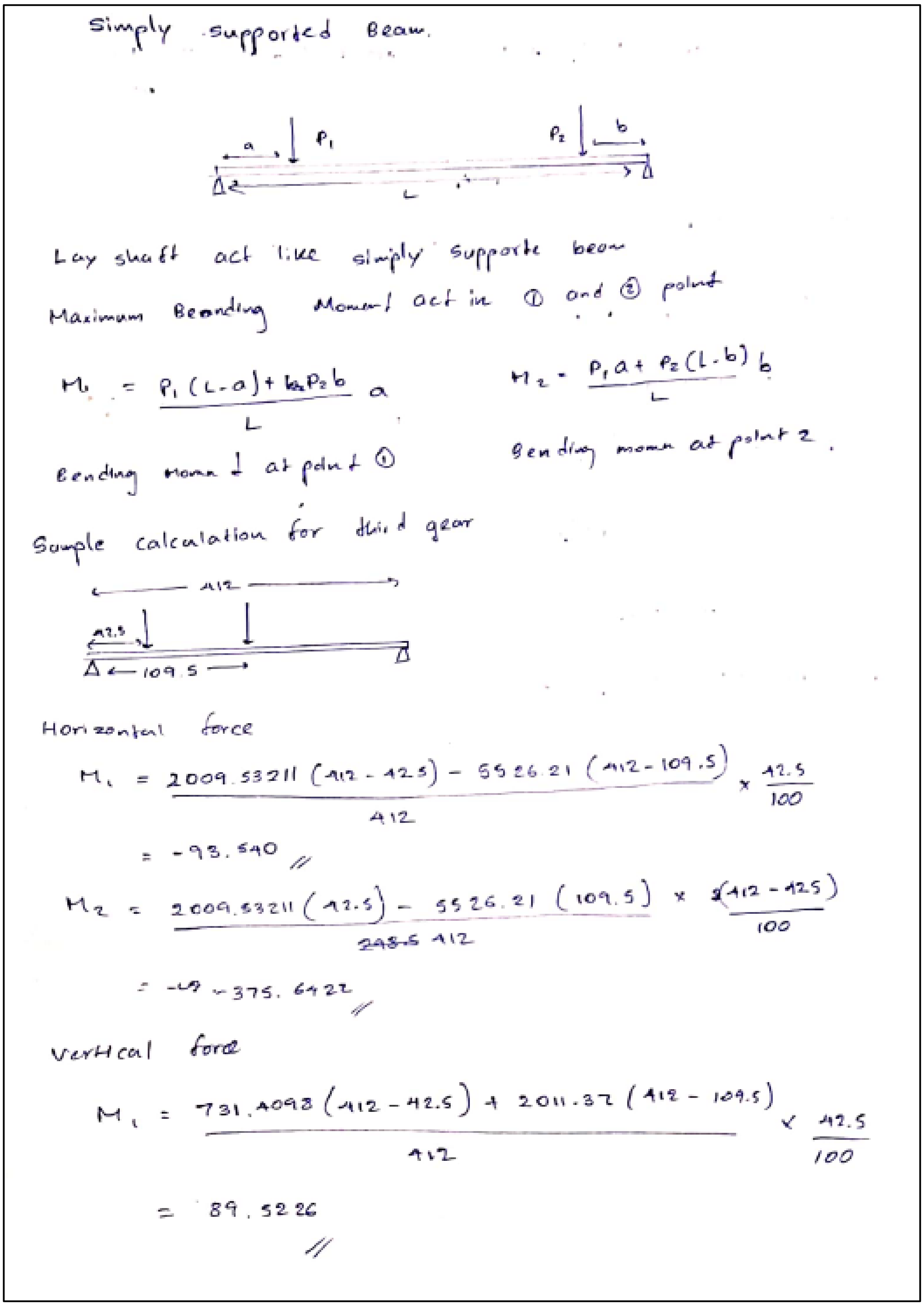
$= 85.405$ KN

Maximum BM (vertical) $= F_t \times l$

$= 731.4098 \times 42.5$

$= 31.071$ KN

Resultam BM $= 90.88622$



Simply supported Beam.

Lay shaft act like simply supporte beam

Maximum Bending Moment act in ① and ② point

$$M_1 = \frac{P_1(L-a) + P_2 b}{L} a \qquad M_2 = \frac{P_1 a + P_2(L-b)}{L} b$$

Bending moment at point ① Bending moment at point 2.

Sample calculation for third gear

Horizontal force

$$M_1 = \frac{2009.53211(412-42.5) - 5526.21(412-109.5)}{412} \times \frac{42.5}{100}$$

$$= -93.540$$

$$M_2 = \frac{2009.53211(42.5) - 5526.21(109.5)}{\cancel{248.5}\ 412} \times \frac{(412-42.5)}{100}$$

$$= \cancel{-49}\ -375.6422$$

Vertical force

$$M_1 = \frac{731.4098(412-42.5) + 2011.37(412-109.5)}{412} \times \frac{42.5}{100}$$

$$= 89.5226$$

Error! Use the Home tab to apply 0 to the text that you want to appear here.sample calculation for simply supported beam

$$M_2 = \frac{731.4098(42.5) + 2011.37(412 - 109.5)}{412} - x'\left(\frac{412-42.5}{100}\right)$$

$$= 181.6580$$

Left gear ① point

$$R = \sqrt{M_{V_1}^2 + M_{H_1}^2}$$

$$= 129.1788$$

Right gear ② point

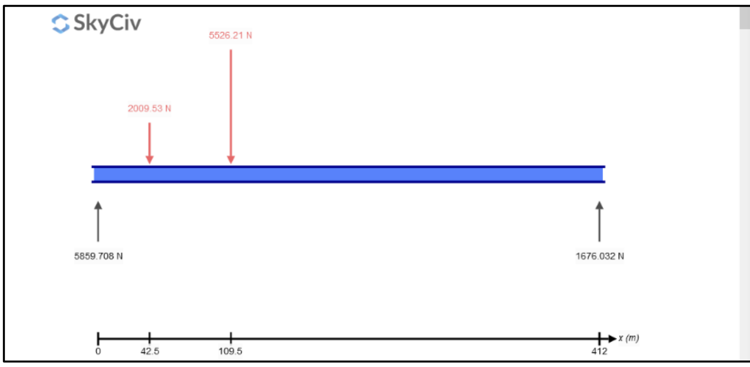
$$R = \sqrt{M_{V_2}^2 + M_{H_2}^2}$$

$$= 417.2610$$

Max BM at ② point.

sample calculation for simply supported beam

Also these same moment value can get using bending moment diagram.[17]



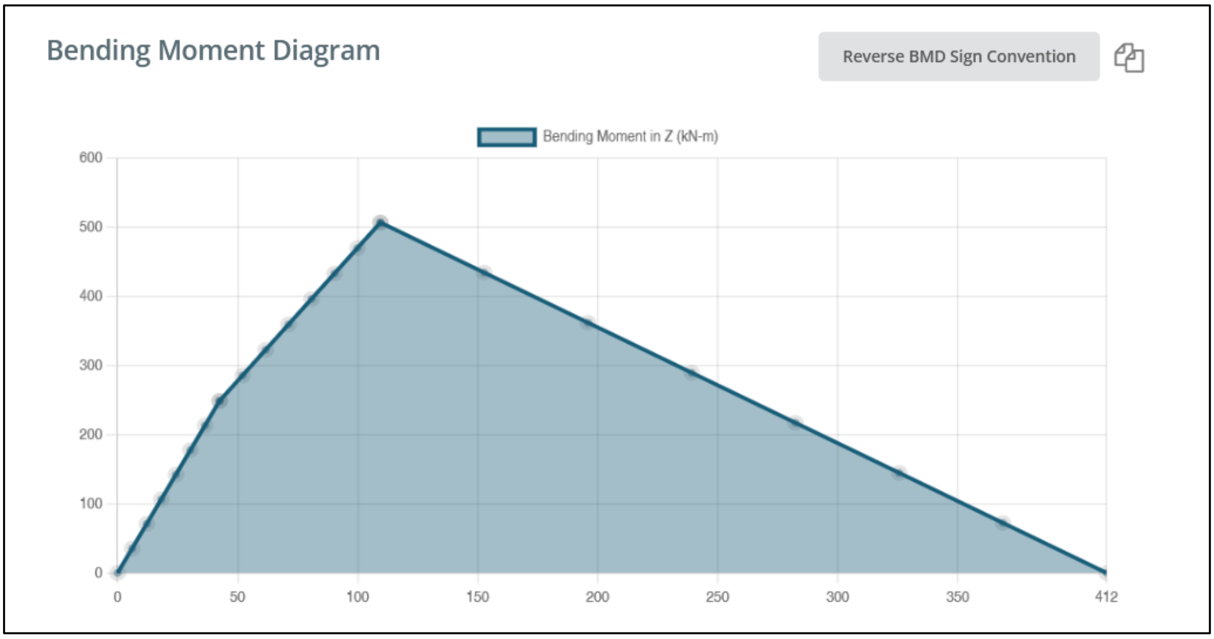
Reactions

Support at	X	Y	Mx
0	0 N	5859.708 N	0 kN-m
412	0 N	1676.032 N	0 kN-m

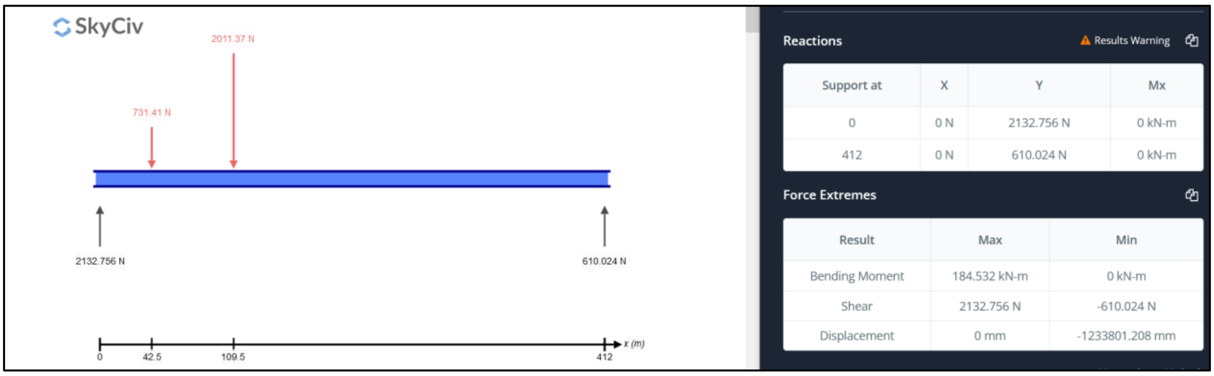
Force Extremes

Result	Max	Min
Bending Moment	507 kN-m	0 kN-m
Shear	5859.708 N	-1676.032 N
Displacement	0 mm	-3389849.092 mm

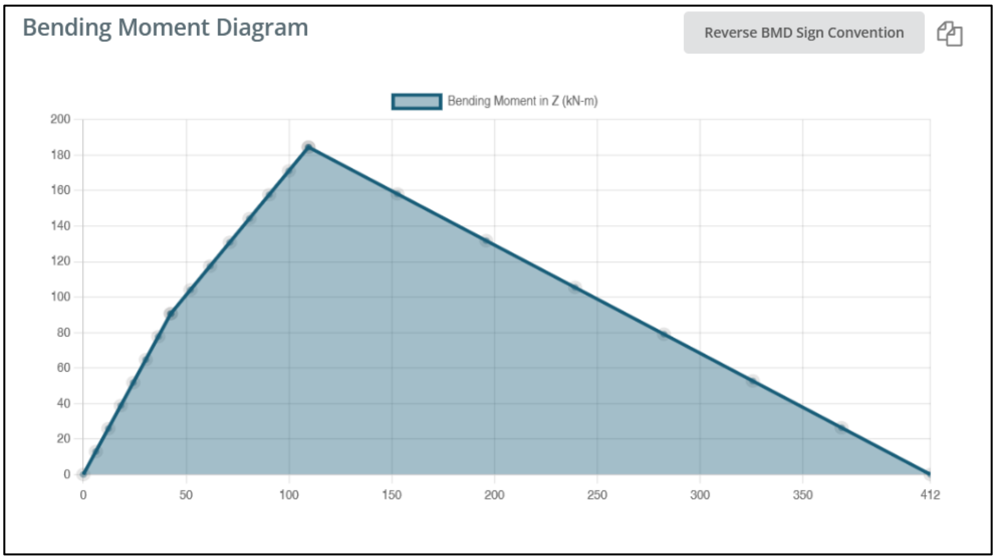
Bending moment diagram



Bending moment diagram



Bending moment diagram



Error! Use the Home tab to apply 0 to the text that you want to appear here.**Bending moment diagram**

4.12.4 Input shaft calculation

Input shaft										
Gear	Distance from left supprot (mm)	Distance from Rigth supprot (mm)	Power need to be transmitted (w)	Angular speed of the shaft (rads-1)	Torque on shaft (Nm)	PCD (mm)	Tangential force on gear (N)	Radial force on gear (N)	Resultant force on gear (N)	Vertical BM (Nm)
-	42.5	-	2000	30.15928947	66.31455962	66	2009.53211	731.40987	2138.499	85.40511466

Horizontal BM (kNm)	Resultant BM (kNm)	σ_max (MPa)	τ_max (MPa)	Safety factor	Km value	Kt value	d_τ (mm)	d_σ (mm)	Diameter	After Bearing selection
31.08491959	90.88622468	441.6666667	325	1.2	1.5	1	13.34333354	14.91838153	15	15

4.12.5 Lay shaft calculation

Lay Shaft										
Gear	Distance from left supprot (mm)	Distance from Rigth supprot (mm)	Power need to be transmitted (w)	Angular speed of the shaft (rads-1)	Torque on shaft (Nm)	PCD (mm)	Tangential force on gear (N)	Radial force on gear (N)	Resultant force on gear (N)	HBM_LEFT
input	42.5	352.5	2000	10.05309649	198.9436789	198	2009.53211	731.40987	2138.499	
3	109.5	285.5	2000	10.05309649	198.9436789	72	5526.2133	2011.3772	5880.873	-93.54022211
2	213.5	181.5	2000	10.05309649	198.9436789	87	4573.4179	1664.588	4866.93	-13.09594796
1	251.5	143.5	2000	10.05309649	198.9436789	96	4144.65998	1508.5329	4410.655	12.22293136
Reverse	355.5	39.5	2000	10.05309649	198.9436789	84	4736.75426	1724.0376	5040.749	56.08475116

LEFT GEAR		RIGHT GEAR								
VBM_LEFT	Resaltant	HBM_RIGHT	VBM_RIGHT	Resultant	BM_Max	σ_{max} (MPa)	τ_{max} (MPa)	Safety factor	Km value	Kt value
						441.6666667	325	1.2	1.5	1
89.52653582	129.4788545	-375.6422824	181.6580249	417.2610243	417.2610243	441.6666667	325	1.2	1.5	1
60.24721452	61.6541216	-409.4178705	177.5825659	446.2718459	446.2718459	441.6666667	325	1.2	1.5	1
51.03189608	52.47527483	-347.6612171	149.1240863	378.2939532	378.2939532	441.6666667	325	1.2	1.5	1
35.06749925	66.14551244	-159.8511024	64.39802712	172.3353732	172.3353732	441.6666667	325	1.2	1.5	1

d_τ (mm)	d_σ (mm)	Diameter	After Bearing selection
21.75180568	24.54671861	25	25
21.75180568	24.54671861	25	
22.20166444	24.07808288	25	
21.12161911	23.79752959	24	
17.22612126	18.89160074	19	

4.12.6 Output shaft

Out put shaft										
Gear	Distance from Rigth supprot (mm)	Power need to be transmitted (w)	Angular speed of the shaft (rads-1)	Torque on shaft (Nm)	PCD (mm)	Tangential force on gear (N)	Radial force on gear (N)	Resultant force on gear (N)	Vertical BM (Nm)	Horizontal BM (Nm)
1	165.5	2000	5.581086473	358.3531647	168	4266.1091	1552.73673	4539.8985	256.9779	706.0410566
2	127.5	2000	4.878830559	409.934302	177	4632.026	1685.91959	4929.2991	214.9547	590.5833165
3	23.5	2000	3.665191429	545.6740906	192	5684.1051	2068.84507	6048.8983	48.61786	133.5764701
Reverse	39.5	2000	5.235987756	381.9718634	159	4804.6775	1748.75961	5113.031	69.076	189.7847623

Resultant BM (Nm)	σ_{max} (MPa)	τ_{max} (MPa)	Safety factor	Km value	Kt value	d_τ (mm)	d_σ (mm)	Diameter	
751.3531989	441.6666667	325	1.2	1.5	1	24.46337217	23.86353072	25	25
628.4856382	441.6666667	325	1.2	1.5	1	23.25572798	22.3248352	24	
142.1491104	441.6666667	325	1.2	1.5	1	18.93913029	14.9646735	19	
201.9647256	441.6666667	325	1.2	1.5	1	17.69522076	14.88913483	18	

4.12.7 Ideal shaft

Idler Shaft										
Gear	Distance from left supprot (mm)	Distance from Rigth supprot (mm)	Power need to be transmitted (W)	Angular speed of the shaft (rads-1)	Torque on shaft (Nm)	PCD (mm)	Tangential force on gear (N)	Radial force on gear (N)	Resultant force on gear (N)	Vertical BM (Nm)
-	-	39.5	2000	10.05309649	198.9436789	84	4736.75426	1724.0376	5040.749	187.1017932

Horizontal BM (kNm)	Resultant BM (kNm)	σ_{max} (MPa)	τ_{max} (MPa)	Safety factor	Km value	Kt value	d_τ (mm)	d_σ (mm)	Diameter	After Bearing selection
68.09948351	199.1095695	441.6666667	325	1.2	1.5	1	17.7829203	19.64544987	20	22

4.12.8 Conclusion

Input shaft diameter	15 mm (Standard diameter for calculated value)
Input shaft material	Cold Drawn 1040 Carbon Steel

Lay shaft diameter	25 mm (Standard diameter for calculated value)
Lay shaft material	Cold Drawn 1040 Carbon Steel

Output shaft diameter	25 mm (Standard diameter for calculated value)
Output shaft material	Cold Drawn 1040 Carbon Steel

Idler shaft diameter	22 mm (Standard diameter for calculated value)
Idler shaft material	Cold Drawn 1040 Carbon Steel

4.13 Bearing selection[15]

Bearing loads are calculated in shaft design process, the bearings are selected to have a lifespan of 3 years. It is assumed that the machine is working 8 hours per day, 8 month per year for 3 years.

$$L_{10} = \frac{RPM \times 60 \times 3 \times 30 \times 8 \times 8}{1000000}$$

Where L_{10} is the no. of revolution in millions that bearing can run with 10% failure rate. To find the appropriate bearing for the task,

$$L_{10} = ({}^{C_r}/_{P})^p$$

C_r: Basic dynamic load rating

P: Equivalent dynamic bearing load

Considering the shaft diameter and bearing life, suitable bearing can be selected with the help of catalogues provided.

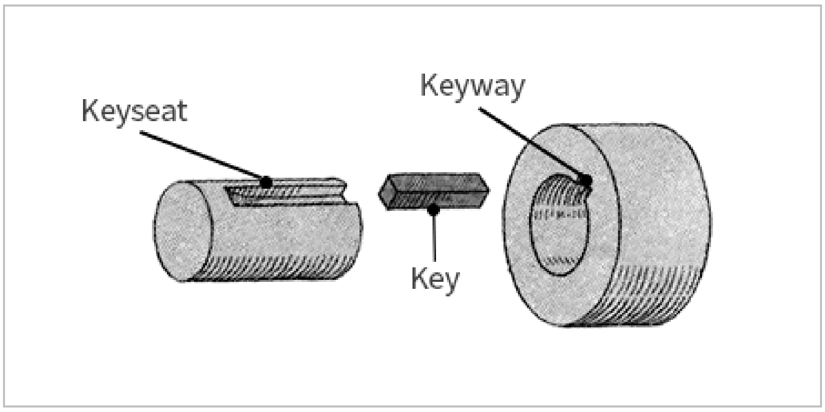
Shaft	Left End Bearing Number	Right End Bearing Number	Expected Lifetime (Millions)
Input	6302	-	124.416
Lay	6305	6305	67.653
Output	6305	6305	67.653
Idler	-	63/22	41.472

Dimensions of the selected bearings according to the catalogue is shown below,

Bearing Number	Nominal bore diameter (d) [mm]	Nominal outside diameter (D) [mm]	Nominal width (B) [mm]	Ring chamfer dimension (r) [mm]
6305	25	62	17	2
6302	15	42	13	1.5
63/22	22	56	16	2

4.14 Key Design

To mount the gear on the shafts, keys are used. A simple **rectangular sun key** with taper ratio on 100:1 is selected. Width (w), thickness (t) and depth of keyway (h) is selected according to catalogue. Length of the key (l) needs to be calculated.



$$T = l \times w \times \tau \times {}^{d}/_{2}$$

$$T = l \times {}^{t}\!/_{2} \times \sigma \times {}^{d}\!/_{2}$$

$$e = 1 - 0.2\left(\frac{w}{d}\right) - 1.1\left(\frac{h}{d}\right)$$

T: Torque need to be transmitted

τ: Allowable shear stress

σ: Allowable crushing stress

d: Diameter of the shaft

e: Strength factor

Since key length is assumed to be equal to face width need to determine whether keys will be safe or not for the shear stress and crushing stress. Since shaft diameter is known key width and thickness can be taken from the standard tables.

Shaft diameter (mm) upto and including	Key cross-section		Shaft diameter (mm) upto and including	Key cross-section	
	Width (mm)	Thickness (mm)		Width (mm)	Thickness (mm)
6	2	2	85	25	14
8	3	3	95	28	16
10	4	4	110	32	18
12	5	5	130	36	20
17	6	6	150	40	22
22	8	7	170	45	25
30	10	8	200	50	28
38	12	8	230	56	32
44	14	9	260	63	32
50	16	10	290	70	36
58	18	11	330	80	40
65	20	12	380	90	45
75	22	14	440	100	50

Error! Use the Home tab to apply 0 to the text that you want to appear here..2:Proportions of standard parallel, tapered and Gib head keys

Sample calculate for one shaft

Shaft diameter d = 15mm

Therefore, w = 6 mm and t = 6 mm

Calculating key length for shearing strength,

$$\boldsymbol{T = l \times w \times \tau_{max} \times \frac{d}{2}}$$

$$49.7359 = 6 * 10^{-3} * l * \frac{184.64}{1.8} * 10^{6} * 0.0075$$

$$l = 10.7746mm$$

Calculating key length for crushing strength,

$$\boldsymbol{T = l \times \frac{t}{2} \times \sigma_c \times \frac{d}{2}}$$

$$49.7359 = 3 * 10^{-3} * l * \frac{250}{1.8} * 10^{6} * 0.0075$$

$$l = 15.9154mm$$

Since 10.7746 mm < 15.9154 mm < 28.2743 mm (Face width) key is safe for shearing stress and crushing stress.

Effect of the keyway,

$$\boldsymbol{e = 1 - 0.2(w/d) - 1.1(h/d)}$$

e = 1 – 0.2 (6 / 15) – 1.1 (3 / 15)

e = 0.7

$$\text{Effect of the keyway (e)} = \frac{\text{strength of the shaft with keyway}}{\text{strength of the shaft without keyway}}$$

Using above equation we can measure the new diameter of shaft. Same procedure was used to ensure lay shaft keys are safe against shearing and crushing and key dimensions for each gear is as follows.

Gear	Width (mm)	Thickness (mm)	Length for shear stress (mm)	Length for crushing stress (mm)	Effect of keyway (e)
Gear 1	6	6	14.36624	21.2206591	0.7
Gear 2	8	7	22.03912	27.2050516	0.752273
Gear 3	8	7	22.03912	27.2050516	0.752273
Gear 4	8	7	22.03912	27.2050516	0.752273
Gear 5	8	7	22.03912	27.2050516	0.752273
Gear 6	8	7	22.03912	27.2050516	0.752273
Gear 10	8	7	22.03912	27.2050516	0.752273

Key Dimensions

4.15 Spline Design

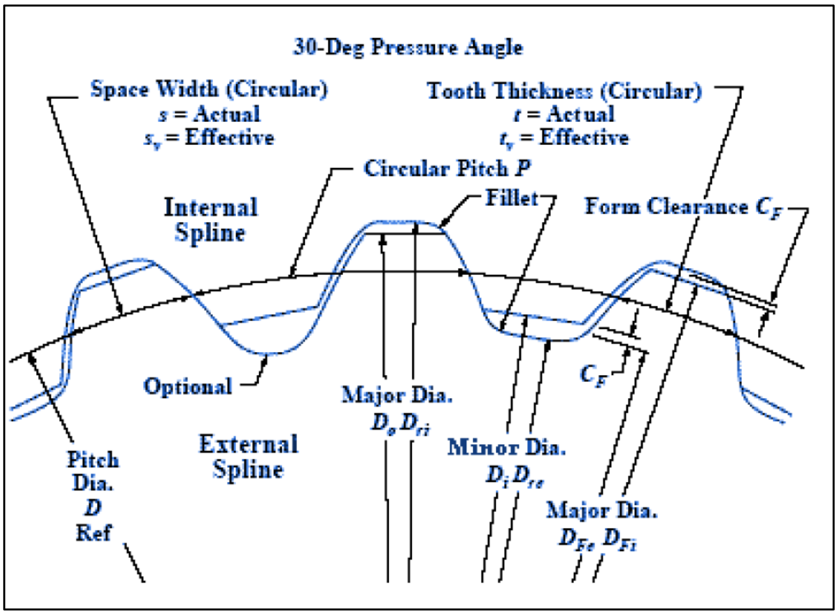
Assumptions

- The shaft and the splines are made out of the same material.
- Number of teeth of the spline is 16.
- The pressure angle of spline is 37.5o, considering the torque transmitting and by literature review Shaft diameter is considered as the minor diameter of the pitch.
- Shaft diameter is considered as the minor diameter of the pitch.

Formulas

$$\text{Torque allowed for the spline} = \frac{\pi d_p d_r L \tau_d}{16}$$

$$\text{Allowable shaft torque} = \frac{\pi d^3 \tau_d}{16}$$



Sample calculation for gear 8

$$\text{Minor diameter external } = \text{ Shaft diameter } = 25 = \frac{16 - 1.3}{p}$$

$$p = 0.588$$

$$\text{Pitch diameter } = \frac{16}{0.588} = 27.21mm$$

$$\text{Major external diameter } = \frac{16 + 1}{0.588} = 28.91mm$$

The torque allowed for the spline should be more than or equal to allowable shaft torque,

$$\frac{\pi \times 27.21 \times 25 \times L_\tau}{16} = \frac{\pi \times 25^3 \times C_d}{16}$$

$$L = 22.97mm$$

$$Minor\ diameter\ intrnal = \frac{16 - 1.3}{0.588} = 25mm$$

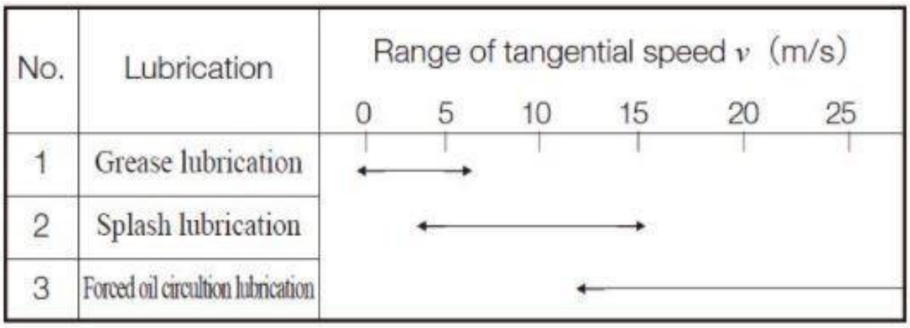
$$\text{Major diameter internal} = \frac{16 + 1.6}{0.588} = 29.93mm$$

Conclusion

Gear	Torque Transmitted (Nm)	No. of Splines	Allowable Pressure for Splines (Mpa)	Minor Diameter (mm) [d]	Pitch [P]	Pitch Diameter (mm) [Dp]	Minimum Length of the spline (mm)	Minor diameter internal (mm)	Major diameter internal (mm)	Major diameter internal rounded (mm)
Gear 7	545.6741	16	20	21	0.7	22.85714	19.29375	21	25.14286	26
Gear 8	409.9343	16	20	25	0.588	27.21088	22.96875	25	29.93197	30
Gear 9	358.3532	16	20	24	0.6125	26.12245	22.05	24	28.73469	29
Gear 10	381.9719	16	20	20	0.735	21.76871	18.375	20	23.94558	24

4.16 Lubrication

Tangential velocity in this case does not exceed 5 m/s. With low tangential speed splash lubrication will not be able to splash the lubricants properly to other gears. Therefore, grease lubrication was selected.



Gear	Pitch line velocity (m/s)[V]
Gear 1	0.995256553
Gear 2	0.995256553
Gear 3	0.482548632
Gear 4	0.468811264
Gear 5	0.437309697
Gear 6	0.431776504
Gear 7	0.361911474
Gear 8	0.351858377
Gear 9	0.422230053
Gear 10	0.422230053
Gear 11	0.416261027

4.17 Belt drive

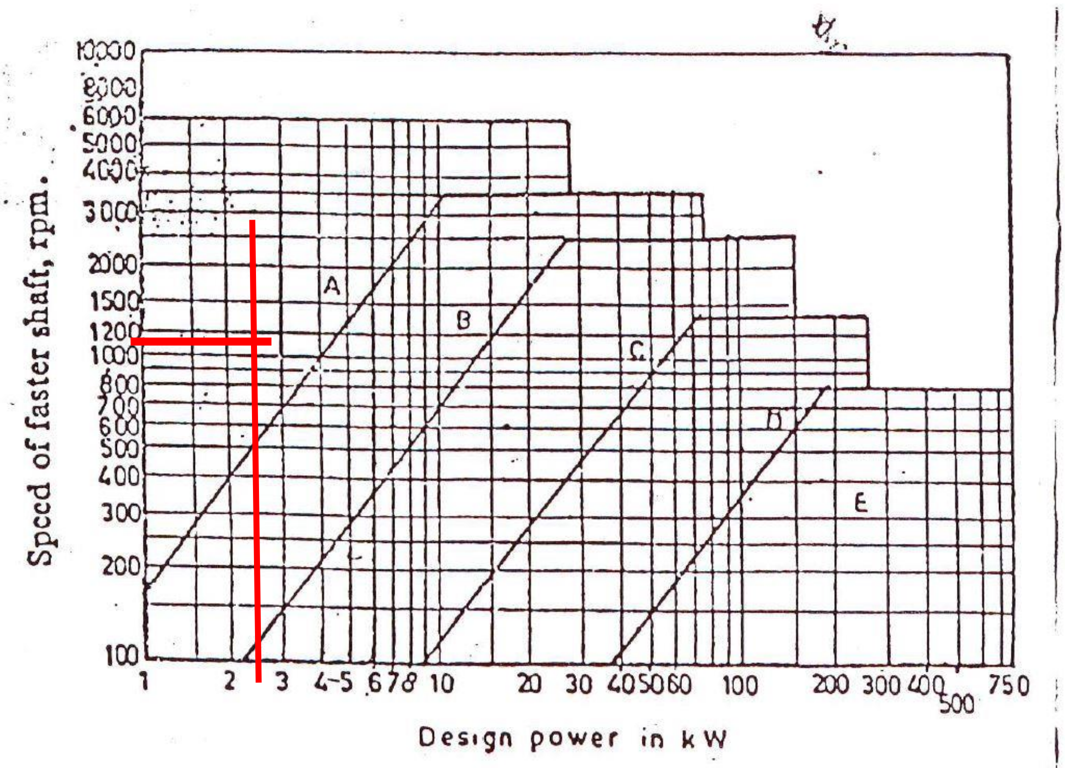
A belt drive is essential to utilize the space available in the machine structure since motor cannot be directly mounted to the input shaft of the gear box. A V-belt type drive is selected considering the centre distances of the shafts and the efficiency.

Based on the prime mover type (Normal torque AC motor), Driven machine (Heavy duty), No. of hours working in a day (~10 hours) and the environmental conditions, a service factor (Ks) of 1.2 is selected.

Rated power	2000W
Rated Torque	13.17Nm
Rated speed	1450rpm

Table 4.12.8-1:Details about prime mover

From the prime mover, the transmission power (NT) is selected as 2000W, and design power (ND) of the belt can be calculated using the below equation.



$$N_D = N_T \times K_s$$

Design power is calculated as 2400W, and the smaller pully RPM is taken as the prime mover output RPM (1450). Even though Khurmi and Gupta [2] suggests using belt cross-section **type A.** According to Khurmi and Gupta [2], the minimum diameter for type A cross-section is declared as **75 mm** and it is taken as the smaller pully diameter (d). Larger pully diameter (D) is calculated based on the speed ratio (5) that is selected in previous initial design report [1].

$$D = Speed\,ratio\ \times d$$

Table 20.1. Dimensions of standard V-belts according to IS: 2494 – 1974.

Type of belt	*Power ranges in kW*	*Minimum pitch diameter of pulley (D) mm*	*Top width (b) mm*	*Thickness (t) mm*	*Weight per metre length in newton*
A	0.7 – 3.5	75	13	8	1.06
B	2 – 15	125	17	11	1.89
C	7.5 – 75	200	22	14	3.43
D	20 – 150	355	32	19	5.96
E	30 – 350	500	38	23	–

From equation, larger pully diameter is calculated and selected from standard size tables as **400 mm**.

A - SECTION		B - SECTION		C - SECTION		D - SECTION	
75	180	125	400	200	600	355	1060
80	190	132	450	212	630	375	1120
85	200	140	500	224	710	400	1250
90	224	150	530	236	750	425	1400
95	250	160	560	250	800	450	1500
100	280	170	600	265	900	475	1600
106	300	180	630	280	1000	500	1800
112	315	190	710	300	1112	530	2000
118	355	200	750	315	1250	560	
125	400	224	800	355	1400	600	
132	450	250	900	375	1600	630	
140	500	280	1000	400		710	
150	560	300	1120	450		750	
160	630	315		500		800	
170	710	355		530		900	
	800	375		560		1000	

4.17.1 Calculation for Center distance

Using the below table, the approximate center distance is set to 337.5mm, and belt drive's speed reduction is used to obtain this**.** Centre distance is selected as 350mm consider my application and there available space.

Recommended Centre Distances:

SPEED RATIO - i	1	2	3	4	5	≥6
CENTRE DISTANCE - C	1.5D	1.2D	D	0.95D	4.5d or 0.9D	5d or 0.85D

Centre distance (C)	350mm
Standard belt length	1430mm

4.17.2 Calculation for Number of belts

To calculate the No. of belts required, basic power rating (P_S) and additional power rating (P_A) are selected from given tables. Arc contact correction factor (K_θ) and belt length correction factor (K_L) are selected by following the given instructions and correction power rating (P_C) is calculated using equation **Error! Reference source not found.**. Finally, No. of belts required c an be calculated using equation **Error! Reference source not found.**. **No. of belts** required to transmit power is **3**.

$$n = {}^{N_D}/_{P_C} \qquad\qquad P_c = (P_s + P_A) \times K_\theta \times K_L$$

(D-d)/C	0.00	0.05	0.10	0.15	0.20	0.25	0.30	0.35	0.40	0.45
θ^0	180	177	174	171	169	166	163	160	157	154
K_θ	1.00	0.99	0.99	0.98	0.97	0.97	0.96	0.95	0.94	0.93

(D-d)/C	0.50	0.55	0.60	0.65	0.70	0.75	0.80	0.90	0.95	1.00
θ^0	151	148	145	142	139	136	133	127	$\boxed{123}$	120
K_θ	0.93	0.92	0.91	0.90	0.89	0.88	0.87	0.85	0.83	0.83

Power Correction Factor K_θ for Arc of Contact

Table 3

B - Section

SPEED OF SMALLER PULLEY (rpm)	SMALLER PULLEY DIAMETER (mm)								
	125	132	140*	150*	160*	170*	180*	190	200*
720	1.61	1.79	1.99	2.24	2.48	2.73	2.97	3.21	3.45
960	2.02	2.24	2.50	2.82	3.13	3.44	3.75	4.05	4.35
1440	$\boxed{2.72}$	3.03	3.39	3.82	4.26	4.68	5.09	5.50	5.90
2880	3.96	4.44	4.95	5.55	6.11	6.62	7.08	7.48	-

Additional power per belt for speed ratio "i"

SPEED OF SMALLER PULLEY (rpm)	1.02 to 1.04	1.05 to 1.08	1.09 to 1.12	1.13 to 1.18	1.19 to 1.24	1.25 to 1.34	1.35 to 1.51	1.52 to 1.99	>2
720	0.03	0.05	0.08	0.10	0.13	0.15	0.18	0.20	0.21
960	0.03	0.07	0.10	0.14	0.17	0.20	0.24	0.27	0.30
1440	0.05	0.10	0.15	0.20	0.25	0.30	0.36	0.41	$\boxed{0.46}$
2880	0.10	0.20	0.30	0.41	0.50	0.61	0.71	0.81	0.91

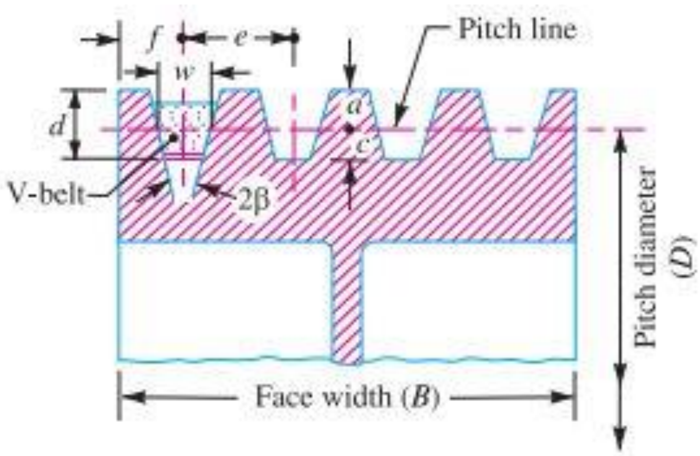
Note: * - preferred pulley diameter

V-Belt Standard Lengths and Correction Factors K_L:

A		B		C		D	
L(mm)	K_L	L(mm)	K_L	L(mm)	K_L	L(mm)	K_L
630	0.80	930	0.81	1560	0.82	2740	0.82
700	0.82	1000	0.83	1760	0.84	3130	0.86
790	0.84	1100	0.85	1950	0.87	3330	0.87
890	0.86	1210	0.87	2090	0.88	3730	0.90
990	0.88	1370	0.90	2190	0.90	4080	0.92
1100	0.90	1440	0.90	2340	0.91	4620	0.94
1250	0.93	1560	0.92	2490	0.92	5400	0.97
$\boxed{1430}$	$\boxed{0.96}$	1690	0.94	2720	0.94	6100	1.00
1550	0.98	1760	0.95	2800	0.95	6840	1.03
1640	0.99	1950	0.97	3080	0.96	7620	1.05
1750	1.00	2070	0.98	3310	0.98	8410	1.07
1940	1.02	2180	0.99	3520	0.99	9140	1.09
2050	1.04	2300	1.00	3710	1.00	10700	1.12
2200	1.05	2500	1.02	4060	1.02	12200	1.16
2300	1.06	2700	1.04	4450	1.04	13700	1.18
2480	1.08	2870	1.05	4600	1.05	16200	1.20
2570	1.09	3090	1.07	5010	1.07		
2700	1.10	3200	1.08	5380	1.08		

4.17.3 Selection the v-belt

Using Khurmi and J.K Gupta we can also find cross section of the v-belt pulley. So belove table use selection pulley detentions.



Belt material	Pulley material						
	Cast iron, steel			Wood	Compressed paper	Leather face	Rubber face
	Dry	Wet	Greasy				
1. Leather oak tanned	0.25	0.2	0.15	0.3	0.33	0.38	0.40
2. Leather chrome tanned	0.35	0.32	0.22	0.4	0.45	0.48	0.50
3. Convass-stitched	0.20	0.15	0.12	0.23	0.25	0.27	0.30
4. Cotton woven	0.22	0.15	0.12	0.25	0.28	0.27	0.30
5. Rubber	0.30	0.18	—	0.32	0.35	0.40	0.42
6. Balata	0.32	0.20	—	0.35	0.38	0.40	0.42

4.17.3-1 Coefficient of Friction for Different Materials

Type of belt	w	d	a	c	f	e	No. of sheave grooves (n)	Groove angle (2β) in degrees
A	11	12	3.3	8.7	10	15	6	32, 34, 38
B	14	15	4.2	10.8	12.5	19	9	32, 34, 38
C	19	20	5.7	14.3	17	25.5	14	34, 36, 38
D	27	28	8.1	19.9	24	37	14	34, 36, 38
E	32	33	9.6	23.4	29	44.5	20	–

	Materials
Belt	Rubber
Pulley	Cast Iron
Friction Coefficient	0.3

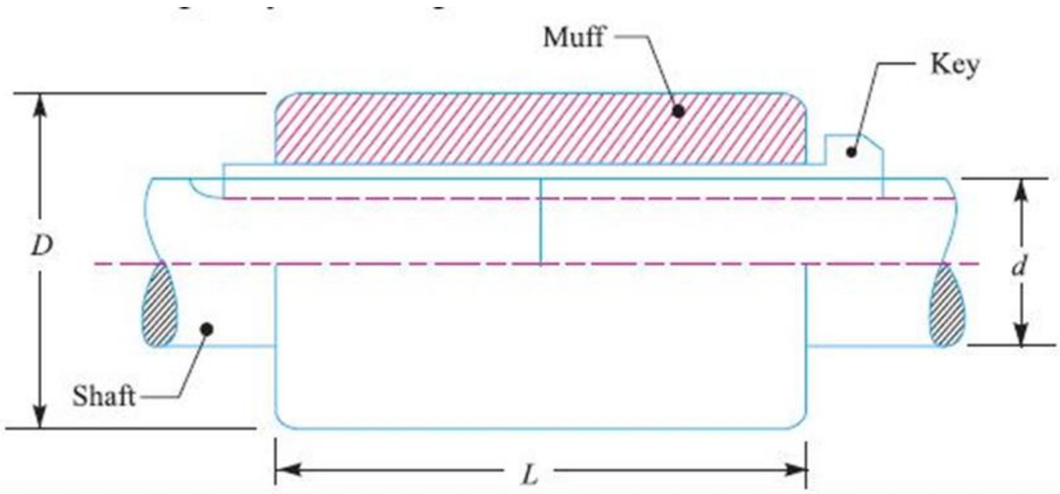
4.17.4 Calculation

service factor	1.2				
transmission power	2000				
Design power	2400				
In put RPM	1450				
Design power			Larger pully diameter		
service factor	1.2		smaller pully diameter	75	mm
transmission power	2000	kW	speed ratio	5	
Design power	2400	kW		375	mm
			larger pully diameter	400	mm
Belt length					
cente distance	350	mm			
Belt length-Ld	1445.75	mm	Correction power rating		
standent length	1430	mm	ps	0.91	kW
			pa	0.17	kW
center distance			K_θ	0.83	
b value	1399.24349		kl	0.96	
center distance-C	556.93				
				860.544	
			numbeer of belt	2.788933512	
(D-d)/C	0.928571429			3	
K_θ	0.83				

4.18 Coupling

Coupling is needed in-between gear box and machine input pully because motor and gearbox are designed as two different parts and need a method to connect the two shafts.

Flange coupling is suggested to be used to couple the prime mover output shaft and gearbox input shaft, but it was changed later to sleeve coupling due to less torque input shaft has to transmit and it is easier to design.



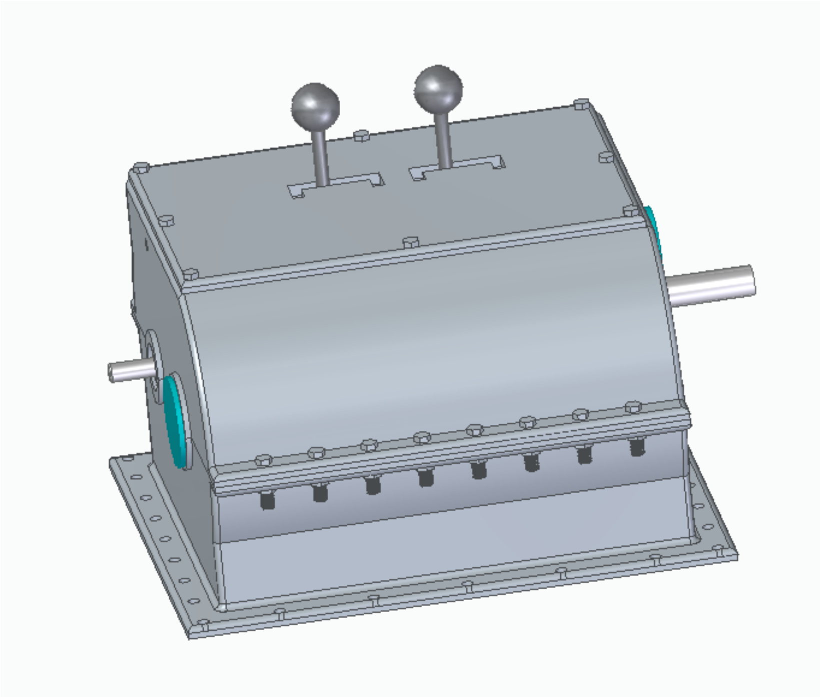
- Diameter of the shaft = **d**
- Outer diameter of the sleeve, **D = 2d + 13mm**
- Length of the sleeve, **L = 3.5d**
- $\boldsymbol{T = l \times w \times \tau_{max} \times \frac{d}{2}}$ Key length for shear stress
- $\boldsymbol{T = l \times \frac{t}{2} \times \sigma_c \times \frac{d}{2}}$ Key length for crushing stress
- $\boldsymbol{T = \frac{\pi}{16}\ \mathrm{x}\ \tau_C \left(\frac{D^4 - d^4}{D}\right)}$ For Sleeve Design

Sleeve						Key					
T (N m)	τ_c (MPa)	d (mm)	D (mm)	d shear stres	Suitability	w (mm)	t (mm)	l (mm)	τ (MPa)	σ_c (MPa)	Suitability
268.7649	14	35	83	2.47207966	SAFE	12	8	61.25	20.8952283	62.6856848	SAFE
307.4507	14	35	83	2.82790931	SAFE	12	8	61.25	23.9028748	71.7086243	SAFE
409.2556	14	35	83	3.7643028	SAFE	12	8	61.25	31.8177312	95.4531937	SAFE
286.4789	14	35	83	2.63501196	SAFE	12	8	61.25	22.2724119	66.8172356	SAFE

Allowable compressive stress for key (MPa)	270
Allowable shear stress for key (MPa)	200
Shear stress for cast iron (MPa)	14

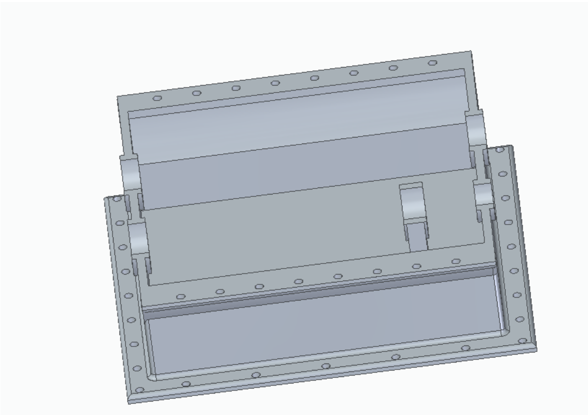
Shaft Diameter (mm) [d]	Outside diameter of the sleeve (mm) [D]	Length of the sleeve (mm) [L]	Width of the key (mm)	Thickness of the key (mm)	Actual Length of the key (mm)
35	83	122.5	12	8	122.5

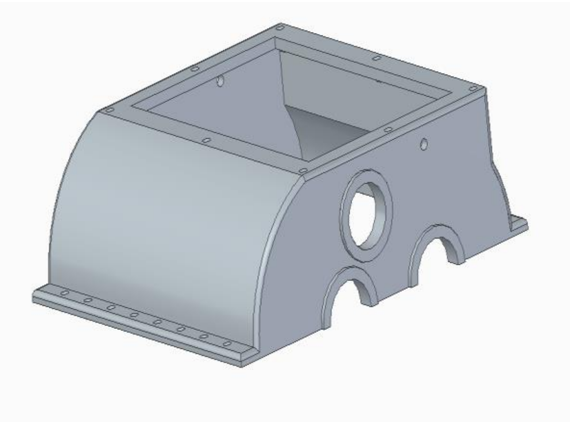
5 Final Design

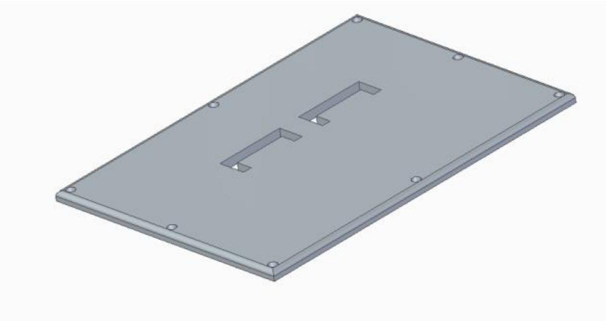


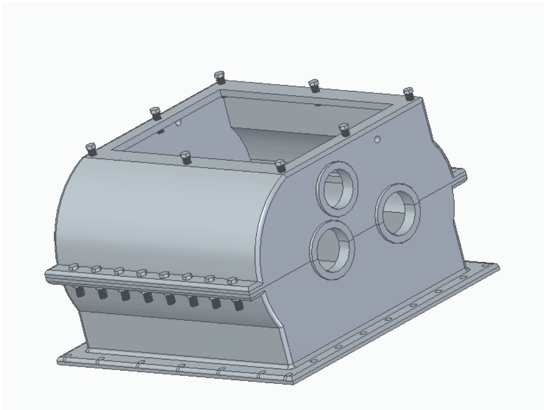
There are so many components in the gear box. some of components and their functions are as follows,

5.1 Housing



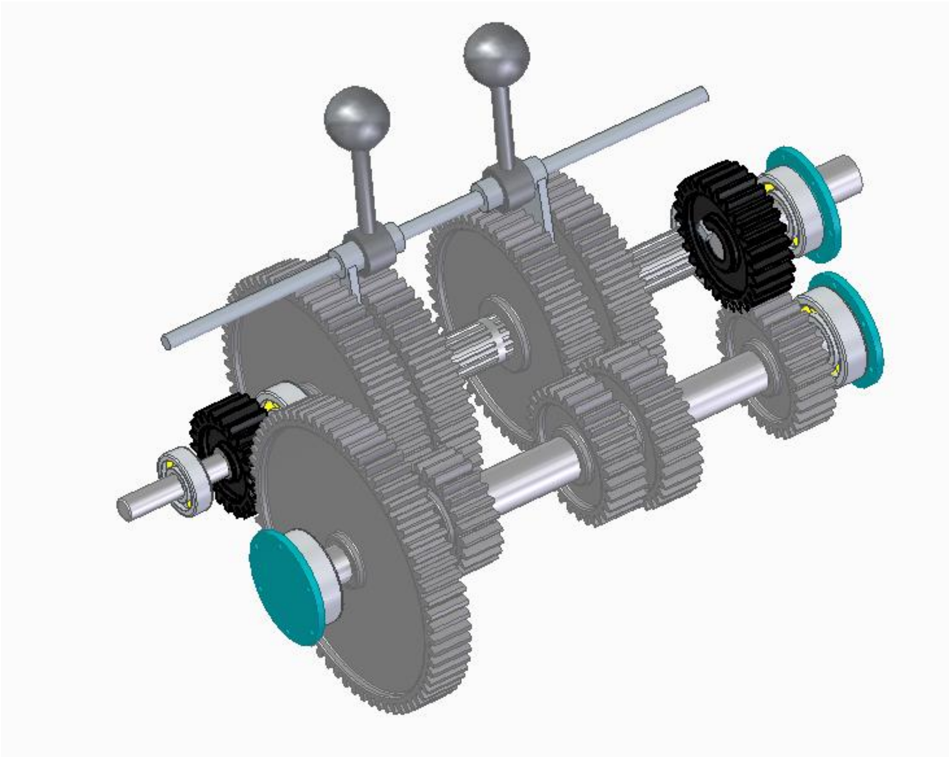




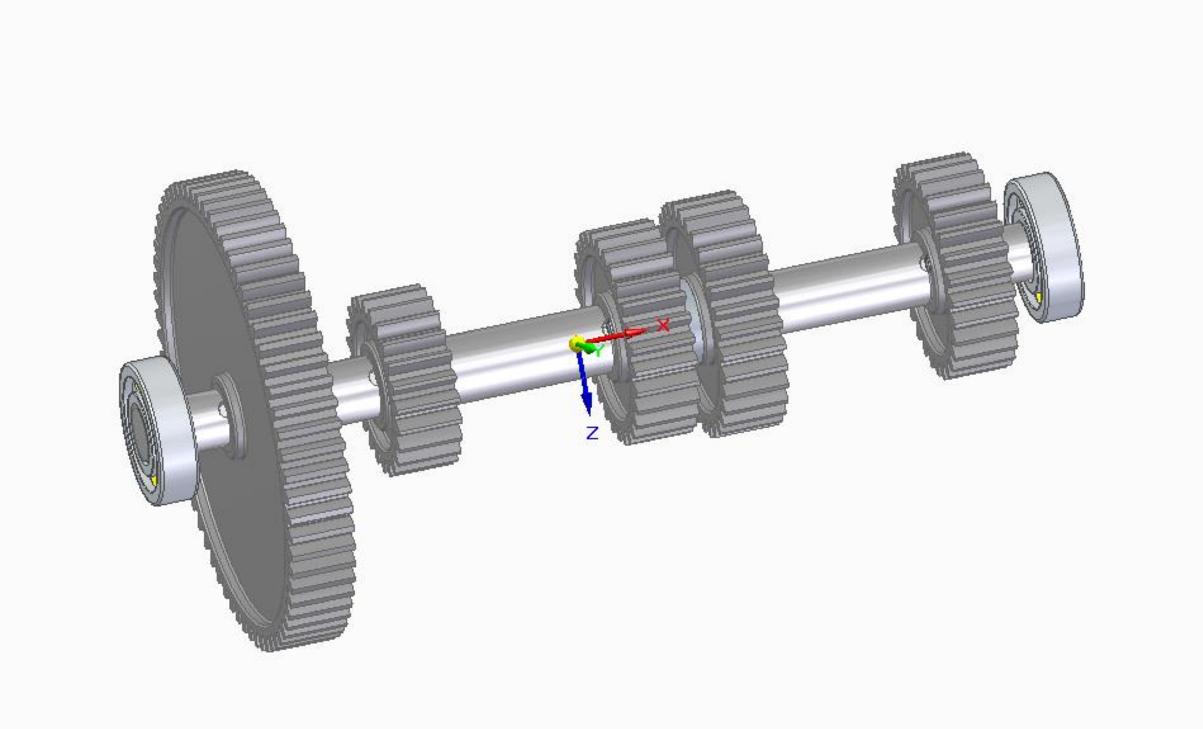


This is the housing for my gear box. There are three parts: the bottom, the top, and the lid. I designed the lid part to allow me to apply grease to the gears without having to open the entire gear box. It is also beneficial to observe inspect. If we want to change the part, we can use the middle open. Furthermore I decide to mount bots to mount the gear box for machine.

5.2 Without housing

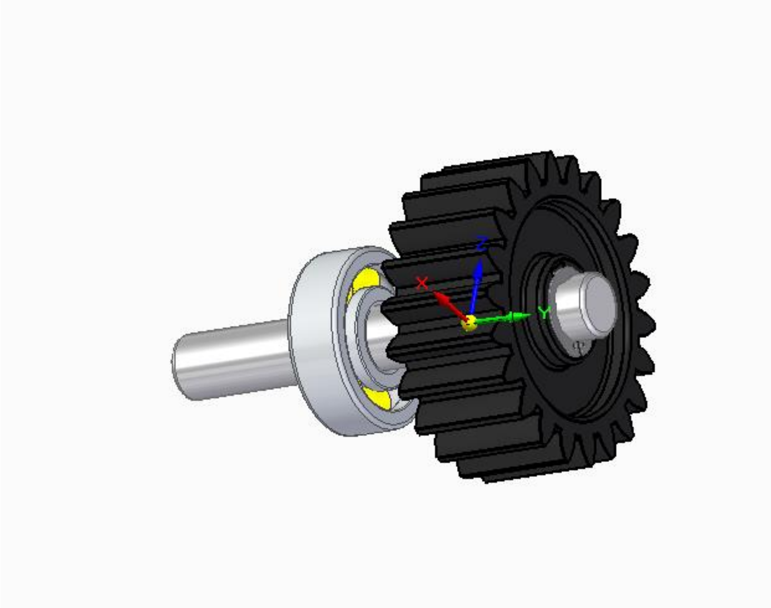


5.3 Lay gear assembly



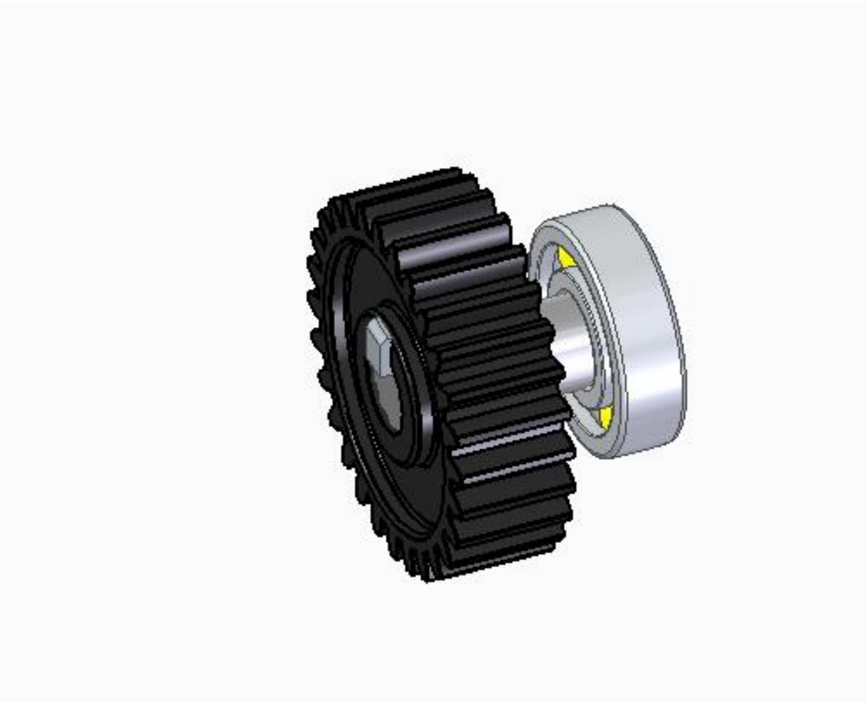
This is the lay sub assembly. There are use gears keys and retaining ring and bearing.in this deign use two bearing and supported to lay shaft.

5.4 Input gear assembly

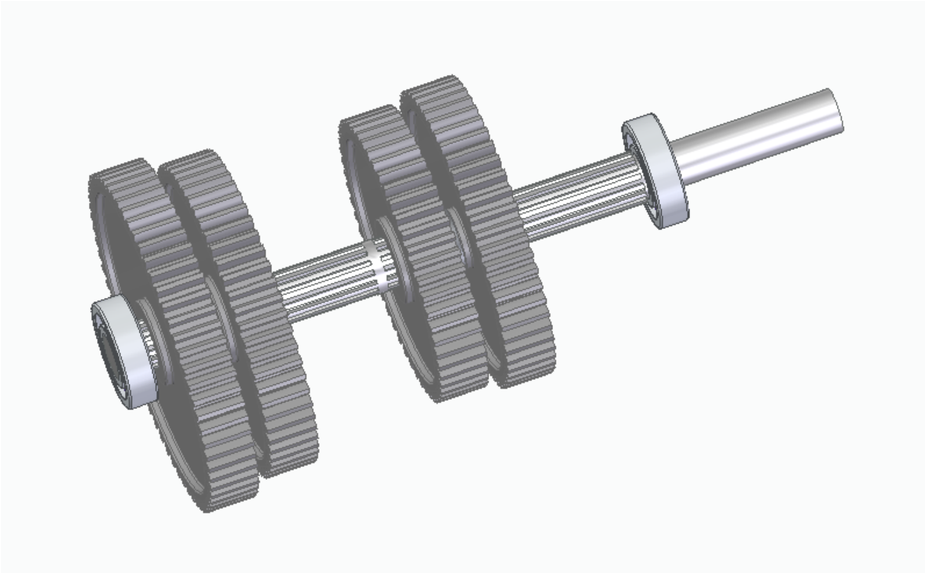


This is the input subassembly. Motor is connected using this side.

5.5 Ideal gar assembly



5.6 Output gear assembly



This is output sub assembly. In this gear move for that I decide to use spline for the shaft. The output shaft supported using two bearings. Belt drive conned to this side.

5.7 Selecting forks and gear shifting method

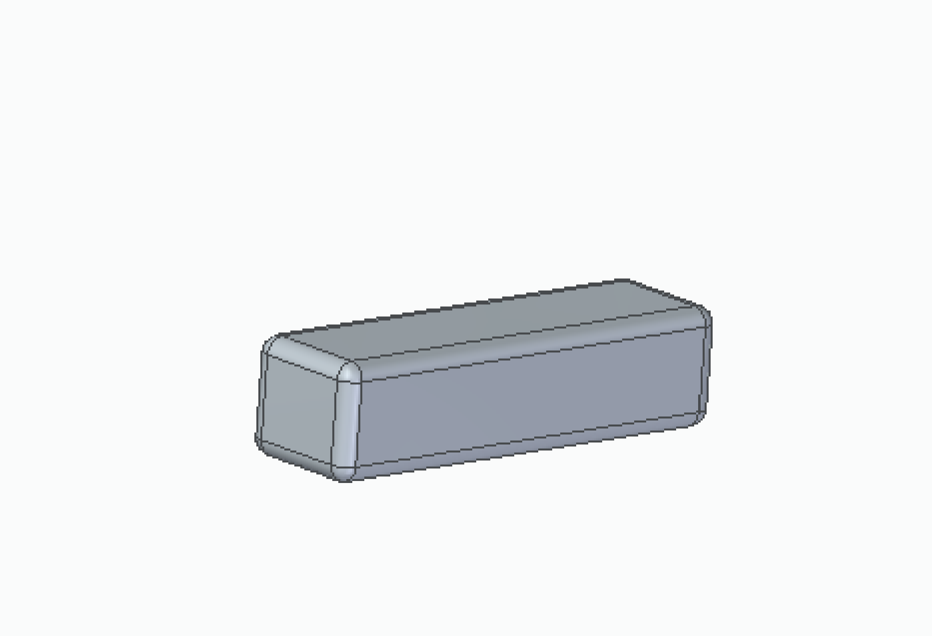
The gear box gear shifting methods are as follows. This fork moves along the long shaft. To make this move smoothly, lubrication should be applied.

5.8 Gear shifting method and output assembly

5.9 Lay shaft

There are keyholes and a groove for a rattan ring in the lay shaft. Also, to avoid stress concentration points, round all edges.

5.10 keys

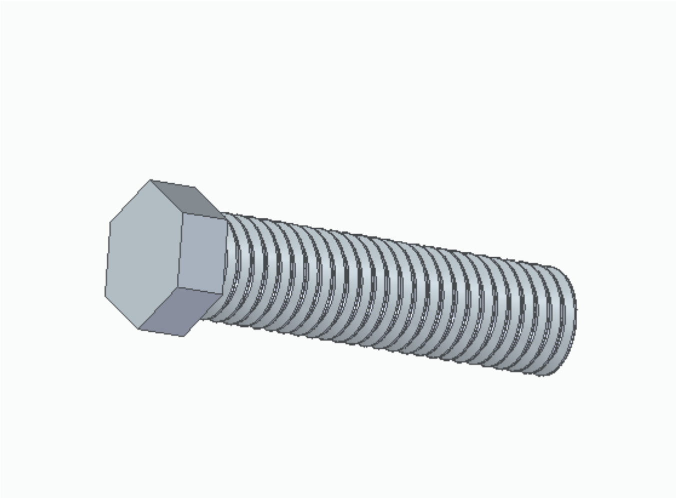


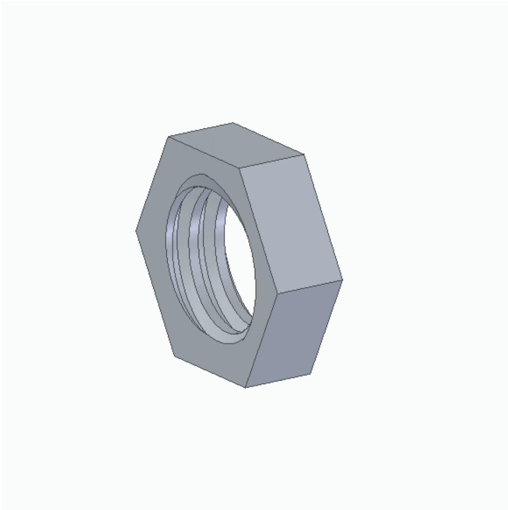
A key is a piece of metal used to connect a rotating machine element to the shaft. Prevent the shaft from rotating in the bore.

5.11 Retaining Ring(circlips)

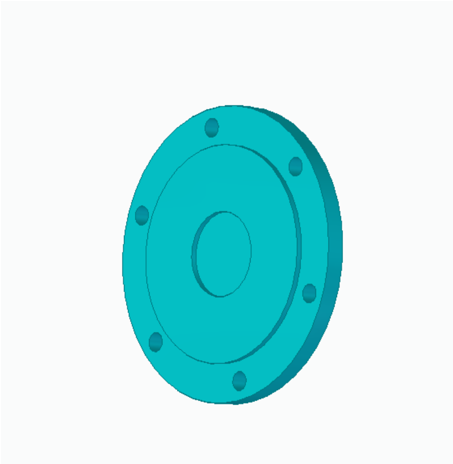
Circlips are used to prevent the axial movement of gears and bearings with respective to the shaft. They are used as retaining rings and are manufactured using semi-flexible metal rings.

5.12 Nuts and bolts





5.13 End caps



5.14 Manufacturing Process

In this case sliding mesh gearbox with spur gears has been designed. It includes spur gears, cylindrical shafts with keyways, cylindrical shafts with splines, keys, ball bearings, couplings, belt drive and selector fork. There are few techniques used in manufacturing process,

- Machining (Milling, Turning, Gear Cutting, Thread Cutting)
- Heat Treatment (Hardening)
- Grinding (Finishing)
- Assembling

5.14.1 Manufacturing of Spur Gears

ASTM class 50 Standard gray iron and Medium Carbon Steel is used to manufacture the gear wheels. Gears can be machined due to good machinability of the grey cast iron. Forging process is used to form raw material needed. Boring process is used to make the internal hole to fit with the shaft. Broaching is used to make the internal splines in the gears to mesh with the splined shafts. Then external teeth of gears are manufactured by using a CNC milling machine. Here appropriate cutting speed and part rotation speed should be maintained in appropriate range to get a good surface finish. Finally, edges are chamfered by chamfer rolling and finishing touches are added.

Then heat treatment is carried out for these gears by carburizing in order to get higher hardness. Here part is heated to the temperatures of 1050C in the carburizing oven. Then quenching process and finally tempering process is done.

5.14.2 Manufacturing of Shaft

ASTM 1040 Carbon Steel is used to manufacture all the shafts.

Cylindrical steel bars are used to manufacture shafts. Shafts are generally manufactured by hot rolling and finally turning to the desired size. CNC lathe machining can be used for turning purposes. Straddle milling can be used to manufacture splined shaft. Milling is used for making the keyways in shafts as well. Finally, heat treatment can be used for carburizing process.

5.14.3 Manufacturing of Keys

40C8 Steel (AISI 1040) is used to manufacture all the keys. First of all, produce the key body by material shaping mould. Then thickness, length and side edge cutting is done by milling process. Finally, heat treatment can be used to harden the material.

5.14.4 Manufacturing of Casing

Casing mainly compose of three parts which are lower, lid and top casings. Each part is machined separately and mating surfaces are defined. Then the casing parts are assembled. Then bore lines are machined to make sure of the accuracy of the bores. Various tapped holes are cut for the attachment of the different components (Nuts & Bolts).

6 Future work

- Casing is designed without doing an Ansys simulations to determine the stress concentrating points. By doing a FEA simulation shape of the casing can be optimised for better strength and compact size. this will help to select the best nut and bolt sizes as well.
- Gear teeth doesn't mesh with each other perfectly. If the first gear mesh perfectly then rest of the gears will have interference. This is due to gear meshing was not taken into consideration when key and keyway was designed. Need to carefully design the keyway position on the lay shaft such that gear teeth are meshed perfectly.
- Tolerances of the gearbox components are to be considered.
- Lubricant circulating system is to be designed.

7 Annexes

Detailed Calculation Excel File

https://docs.google.com/spreadsheets/d/1NVyX350-WVuz5h4ewr0Typ4IblEPMxzX/edit?usp=share_link&ouid=101472894036904749284&rtpof=true&sd=true

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