Design and Manufacturing of Multi-Clutch Gearbox

A Final Year Project Report

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ABSTRACT

Our project is centered on manufacturing and designing a multi-clutch gear box. The project has two parts design and manufacturing. The parts of the gear box would be manufactured in the local market and will be assembled to make a single operating unit.

Power transmission is an essential part in every industry and there are different modes of power transmission which are used in the industry. Gear boxes are used where the user required to deliver power at different RPM. The gear boxes also enable us to deliver power to multiple outputs from a single input power source. All the gear boxes provide power in a continuous manner to all outputs and if a user wants to disconnect a single output it won't be possible without shutting down the whole system.

Our design of the gear box is the answer to that problem. The addition of the clutches is a unique feature of our design which enable us to engage or disengage any output without disturbing the other outputs.

PREFACE

The gear box system provides an easy to use and convenient way to transmit power to multiple outputs from a single input power source. The clutches will facilitate the user to use any desired number of outputs without disconcerting the other outputs. The efficiency of this design is about 90-95% due to electromagnetic clutches. In friction clutches the power losses are due to the slippage of the clutch plate while in electromagnetic clutches the phenomenon of slippage does not happen which make these clutches nearly 100% efficient. High efficient clutch will automatically increase the overall efficiency of the system.

ACKNOWLEDGMENTS

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NOMENCLATURE

G	Gear
P	Pinion
N_g	Number of teeth on gear
N_p	Number of teeth on pinion
T	Torque (Nm)
ϕ	Pressure angle (degrees)
ω	Rotational speed (rad/s)
V	Pitch line velocity (m/s)
P	Circular pitch (mm)
m	Module
W	Power (Watt)
e	Velocity ratio
D_o	Outer diameter of gear/pinion (mm)
D_P	Pitch diameter of gear/pinion (mm)
D_b	Base diameter of gear/pinion (mm)
$D_{o/p}$	Outer diameter of pinion (mm)
$D_{o/g}$	Outer diameter of gear (mm)
$D_{b/p}$	Base diameter of pinion (mm)
$D_{b/g}$	Base diameter of gear (mm)
$D_{P/P}$	Pitch diameter of pinion (mm)
$D_{P/g}$	Pitch diameter of gear (mm)

h_a	Addendum (mm)
h_d	Dedendum (mm)
b	Width of gear tooth (mm)
t_o	Top land width (mm)
h	Whole depth (mm)
F_t	Tangential component force acting on gear tooth (N)
F_r	Radial component force acting on gear tooth (N)
σ	Bending stress on teeth (MPa)
K_V	Dynamic factor
K_O	Overload factor
K_h	Load distribution factor
K_s	Size factor
K_B	Rim thickness factor
J	Geometric factor
Y	Levis factor
d	Shaft diameter (mm)
M	Bending moment on shaft (Nm)
T	Twisting moment on the shaft (Nm)
T_E	Equivalent torque on the shaft (Nm)
K_M	Combined shock and fatigue factor for bending
K_T	Combined shock and fatigue factor for twisting
τ_{all}	Maximum allowable shear stress (MPa)
C_{10}	Basic dynamic radial load rating (k N)

L_{10}	Life associated with 90% reliability (revolutions)
F_D	Static radial load (k N)
L_D	Desires life (hours)
R	Reliability
a	Bearing life factor
n_D	Desired speed (rev/min)
t_R	Rim thickness below tooth
h_t	Tooth height
m_B	Back up ratio
σut	Ultimate tensile strength

ABBREVIATIONS

RPM	Revolutions Per Minute
FS	Factor of Safety
PA	Pressure Angle
DCT	Dual Clutch Transmission

CHAPTER 1: INTRODUCTION

1.1 Background

Our project is based upon an industrial process of power transmission. There are different mode of power transmission and gear boxes are widely used in almost every industry. The main purpose of using gear boxes is to provide power at different RPM and torque according to the requirement of the output. The existing gear boxes provide power continuously to all the outputs. In our design a clutch is coupled at each output so that it can be engage or disengage without disturbing the whole system. Our project includes the designing and manufacturing of a multi-clutch gear box. The gear box will deliver power from one input source to three output sources. The input source would be an engine which provide power to one pump (PVP33) and two generators of 10 kW and 12 kW power. Each pump and generator would operate at different RPM. There are different types of gears available like spur gears, helical gears, bevel gears, rack and pinion. Bevel gears are used when we need to supply power to an output shaft which is an angle to the input shaft. In our design all the outputs are inline or parallel to each other so spur gears would be the most suitable option for us. Another reason for using spur gears is that they can operate at higher RPM and their gear ratios are quite high. Also this type of gears is easily available in the market.

1.2 Problem Statement

The project is an industry requirement and we are manufacturing it for a particular client. The client needs a customized gear box for a particular design specifications which is not available in the market. It is more of a customized gear box which has some extra features. The existing gear boxes do not have clutches as in our design clutches make it a user friendly system.

1.3 Project Description

Project consists of two parts, first part is designing of gear box and the second part consists of manufacturing and assembly of the gear box.

1.4 Designing

This part includes mathematical and analytical modeling of the gear box using the knowledge of mechanical engineering which includes the concept of machine design, theory of machines and mechanics of materials. After the mathematical modeling three dimensional model will be manufactured on Solid works. The models will be run through simulations and results would be compared with theoretical values.

1.5 Manufacturing

After mathematical modeling the parts of gear box will be manufactured. To reduce the cost of manufacturing all the parts would be manufactured in the local market. Different parts will be assembled to a single unit and experimental results will be calculated and compared with our theoretical calculations.

1.6 Clutches

Our gear box system is just like automobile transmission system where clutches are used to turn on and off the power supply. We will use four electromagnet clutches. One at input power supply and three on output shafts for pump and generators .The design of the clutch is not included in the project rather standard clutches would be used available in the market.

1.7 Aims and Objectives

The main objective of the project is to design and manufacture a customized multi-clutch gear box according to the given design specifications from the customer. The system will be able to deliver power to three units at different RPM.

To make the system user friendly clutches will be clamped at each coupling of output unit.

To make our system financially economical all the parts will be manufactured in local market which will reduce our cost of shipping.

For accurate results 3D model will be run through simulations. The results of the software would be compared with the theoretical calculations which will help us to reduce the error to a minimum extent.

The design is made in such a way that it can be modify for general purposes rather than for a particular operation.

1.8 Applications

The gear box system is designed for a particular client at specific design specifications but can be used at any industry where power transmission is required.

This system is the most suitable where multiple systems operate at the same time and it would be easy to operate them with a single input rather than having a separate power source for every single system.

The gear boxes can be of great use at the construction site where a number of pumps and motors are operating at the same time. The clutch is a unique addition to the gear box, it is just like deriving of a car where you can change the gear according the requirement. In our case a user can engage and disengage any output without disturbing the other outputs.

Gear boxes can also be used in sugar machinery, automobile industry and cement machinery.

Electric overhead travelling cranes use gear box for their operations. Sometimes crane has to carry the load at higher speed and sometimes speed should be low according to the nature of the load. Gear box can be of great help to provide different speed from the same input speed.

Gear boxes find their applications in machines like lathe machines, milling machines, grinding machines and drilling machines.

CHAPTER 2: LITERATURE REVIEW

The gearboxes are used to transmit power from an input power source to an output member according to variable needs. A Gearbox is a device that is used to deliver speed and torque conversions from a rotating input power source to an output member. The torque and rotational speed usually measured in revolutions per minute (RPM) are inversely proportional to each other. As the speed of the shaft decreases, the torque transmitted increases and vice versa. The use of gear boxes depends on the application. For example Multi-speed gearboxes are used when there is a need of frequent changes in torque and speed at output shaft .The basic principle on which gearboxes operate is of meshing of teeth which enable the transfer of motion and power from input power source to the output member

2.1 Uses of gearbox

Gearboxes are widely used in almost every industry. Some of the main uses [1] of gearboxes are as follows:

- They are used to change the direction of the power through which is transmitted.
- To change the value of the magnitude of force or torque which is transmitted from the prime mover.
- The main purpose is to change the rotating speed of the input relative to the output.

2.2 Type of Gearboxes

Gearboxes are classified on the basis of following categories [1]:

2.2.1 Relative position of output and input shaft:

- Parallel axes gearboxes
- Co- axial gearboxes
- Intersecting axes gearboxes
- Non-intersecting and perpendicular axes gearboxes

2.2.2 Number of Stages

- Single stage gearboxes
- Multi-stage gearboxes

2.2.3 Number of Speed Ratio

- Single speed gearboxes
- Multi speed gearboxes

2.2.4 Type of mesh

- Constant mesh gearboxes
- Sliding mesh gearboxes
- Synchromesh gearboxes.

2.2.5 Number of Clutches

- Single clutch gearboxes
- Dual clutch gearboxes

The details of the above mentioned gearboxes is as follows,

2.2.5.1 <u>Parallel Axes Gearboxes</u>

In this type of gearbox all the shafts input/output are parallel to each other or we would say they are inline. As shown in the figure all the shafts are parallel to each other.

Figure 1: Parallel axis gearbox

2.2.5.2 Coaxial Gearbox

In this type of gearboxes the input shaft and the output shaft are on the same axis of rotation. They are also referred to as an inline gearbox. Coaxial gearboxes are usually constructed by using spur gears. Spur gears come in different versions.

The direction of rotation of the output shaft and the input shaft depends upon the number of gear meshes, and can be in the same direction or in the opposite direction.

If single spur gears are not making coaxial gearboxes in principle, concentricity can be obtained by using two pairs of gear wheels with a total of four gearwheels. Two of the gearwheels are arranged in such a way that they are in parallel on one axis when doing this.

One very special type of coaxial gearboxes is planetary gearbox.

2.2.5.3 Intersecting axes gearboxes

In this type of gearboxes the axes of rotation of input shaft and the output shaft intersect each other at some angle. To achieve this purpose bevel gears are used to manufacture Intersecting axes gearboxes. They are typically used to alter the direction of the power or force of the input shaft relative to output shaft. The intersecting angle changes from 90 degrees to less than 180 degrees. A typical applications of Intersecting axes gearbox is the differential mechanism in the automobile in which straight bevel gears are used to equalize the rotational speed of both the wheels during the turning operation.

2.2.5.4 Non-intersecting and perpendicular axes gearboxes

In this type of gearboxes the two axes of rotation are perpendicular to each other but they don’t intersect each other. This type of mechanism is achieved by using hypoid gear and the worm gear. There are some typical applications of this type of gearboxes are in the passenger lifts used in the buildings in which worm gears are used. Another typical example is in the rear axle of the busses and heavy vehicles in which hypoid gear are used to manufacture non-intersecting and perpendicular axes gearbox.

2.2.5.5 Single stage gearboxes

To describe single stage transmission one has to understand what are stages in power transmission. Stages are the number of times the reduction in the rotational speed need to be done from input shaft to the output shaft. The speed can be increased in stages it does not mean that speed has to be reduced in stages. If the speed changes happens in only one stage then the gear box is called single stage gearbox. This type of gearbox is used where speed changes are low. It also depends on the torque, the single stage gearbox operates the best at low torques.

2.2.5.6 *Multi-stage gearboxes*

In this type of gearboxes the reduction/increment in the rotational speed is done in multi-stages. Counter shafts are used to achieve the desired speed at multiple stages. For each gear stage, the direction of rotation between the drive shaft and the driven shaft is reversed. The overall multiplication factor of multi-stage gearboxes is calculated by multiplying the gear ratio of each stage. The typically used gears are spur gears and they are connected in a parallel manner. The main application is where speed changes are very high.

2.2.5.7 Single Speed Gearboxes

In this type of gearbox there is a single input speed and only one output speed. The gears are running constantly and provide power to the output at desired speed. The output speed can be greater or lesser relative to the input rotating shaft.

The typical examples of single speed gearbox are electric motor derive system, drive mechanism for electric car, park locking mechanism, sunroof motor with roof lid position detector and easy shift sector for transfer case.

2.2.5.8 Multi Speed Gearboxes

These type of gearboxes are used to transmit power at different speeds. It means that the input speed remains the same but output speed and torque can be varied by using different combinations of gears. The input shaft is connected with the primary mover by a clutch. The clutch is used to engage or disengage the power transmitted. The different rotational speeds are achieved by changing the combination of gears.

The typical application of this type of gearbox is transmission system in vehicles where a combination of gears is attached with the engine through a clutch. The engine which is prime mover in this case rotates at a constant speed. The combination of the gears can be changed to achieve the desired output speed. The clutch is used to engage and disengage the input power.

These type of gear boxes are further classified into three types:

2.2.5.9 Constant Mesh Gearboxes

This is the most famous type of gearboxes widely used in the twentieth century and most of the automobile industries are also using this type. In this type gears always remain in a continuous mesh. It consists of two dog clutches. These clutches are attached on the main shaft, one between clutch gear and the second gear and the other is between the first gear and the reverse gear. When the second dog clutch is engaged with the clutch gear the output shaft runs at full speed. Similarly by changing the position of dog clutches the changes in the torque and speed are achieved.

2.2.5.10 Sliding Mesh Gearboxes

This is the oldest type of gearboxes. The name is derived from the fact that the gears are meshed by sliding to obtain different combination. The gears on the counter shaft remain at the same point or in other words do not change their position. But the gears on the main shaft are moved to mesh them with the gears on the counter shaft to obtain desired speed and torque. There are more chances of wear and tear due to sliding.

2.2.5.11 Synchromesh Gearboxes

Constant mesh gearbox has also some drawbacks. One of which is the wearing of the gears due to a continuous meshing. The problem is solved by using another type of gearbox which is called synchromesh gearbox. This gearbox operates at the same principle as constant mesh gearbox, but the difference between the two is dog clutches are replaced by synchromesh devices. These devices brings the gears to be meshed in frictional contact first, which equalizes their speed and then they are engaged smoothly. The devices look like the cone clutches where the outer surface consists of the frictional surface. These are widely used in automobile industry.

2.2.6 Number of Clutches

The gearboxes are also classified with the number of clutches attached with it. Although the clutch has nothing to do with the magnitude or direction of speed and torque but it is of great help to change the combination of gears inside the box. Clutches are mostly used in multi-speed gearboxes. The transmission system in automobiles is one of the examples.

On the basis of clutches the gearboxes are classified into two categories:

2.2.6.1 Single Clutch Gearboxes

In this type of gearboxes a clutch is attached between prime mover and the gearbox. The basic purpose of the clutch is to disengage the engine power so that the gear mesh can be selected according to the requirement. The transmission system in the automobile is an example of single clutch gear system. The input power is coming from the engine. The clutch is attached with the fly wheel of the engine. When the driver press the clutch paddle the gears on the main shaft become free and dog clutches are moved left or right to select the desired combination of the gears. After selecting the gear the clutch is released and input shaft is engaged again with the fly wheel.

2.2.6.2 Dual-Clutch Transmission

The dual-clutch transmission [2] also called semi-automatic transmission is most widely used in racecars where gear change is required in a continuous manner. In conventional manual transmission where single clutch is used the power supply goes from on to off and on again. During the period when the clutch is engaged the power supply to the wheels is stopped and the combination of the gears is changed by using synchromesh devices. After that clutch is released causing a phenomenon called “shift shock” or “torque interrupt”.

A dual-clutch gearbox [3], by contrast uses two clutches for its operation which helps in a smooth transmission of power without interrupting the torque. This system consists of two input shafts an outer shaft and inner shaft. The two clutches share the gears, the first gear is linked with odd number of gears while the second gear is linked with even number of gears. A six gear dual-clutch transmission is shown in the figure given below.

It is clear from the figure that one clutch is used to engage and disengaged the gear number one, three and five. The second clutch is used to engage and disengage the even gears two, four and six. That's the reason that allows smooth and fast gear changes and keeps power delivery constant. We cannot achieve this changing of gears with standard manual transmission because it uses only one clutch for all odd and even gears.

Dual-clutch transmission is similar to automatic transmission in which a torque converter is used. But in case of dual-clutch transmission there is no need of such thing, instead DCTs uses wet multi-plate clutches. Wet multi-plate clutches use hydraulic pressure to drive the gears just like torque converters in automatic transmission.

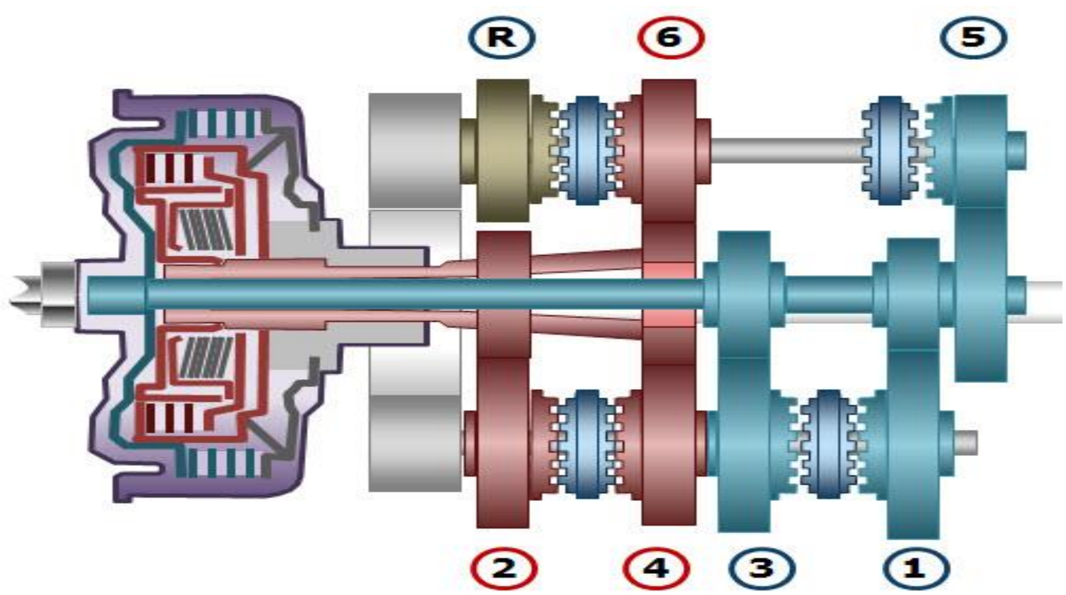

Figure 2: DCT [2]

2.3 What is Multi-clutch Gearbox System?

Our project is to design and manufacture a multi-clutch gear box system. So, what is multi-clutch system? As it is clear from the title there would be more than two clutches. In this system a clutch is attached at every output so that it can be engage or disengage without disturbing the whole system. The clutches are always used to disengage the power input that make it easy to change the gear mesh but in our design the clutches would be used to engage or disengage the output member rather than changing the gear mesh.

The question is why we need such type of gearbox. The answer is simple, the existing gear boxes provide power to the output members in a continuous manner and the user just can't disengage a single output without cutting the power of the whole system. The clutches would help the user to achieve this type of transmission.

Electromagnetic clutches would be most suitable for the system due to high efficiency and ease of use. All the outputs are parallel to each other so spur gears is the most reasonable type of gears that would be used in the manufacturing.

2.4 Components of gearbox

There are some components which are used in every type of gearbox. We can say a gearbox is incomplete without any of the following component:

- Gears
- Bearing
- Shaft

2.4.1 Gears

A gear [4] is a rotating wheel with teeth that mesh together with other gears to transmit torque at different speed keeping the power constant. Gears are used nearly in all mechanical equipment to attain different torque and speed ratio. It is basic component of all mechanical devices. The most common types of gears are spur gears, helical gears, bevel gears, worm gears and rack and pinion.

2.4.1.1 Spur Gears

Spur gears are used to connect parallel and co-planer shafts. The meshing is called spur gearing. These gears have straight teeth and are parallel to the axis of rotation. Spur gears are the most common and widely used gears. The recompenses of spur gears are their simplicity in design, economy of manufacture and maintenance. When the spur gears are connected to the shaft there is only radial loads on the bearing.

2.4.1.2 Helical Gears

The teeth of the helical gears are at an angle to the axis of rotation making a helix. The name of the gears is derived from the fact that its teeth form a type of helix.

They are usually considered high speed gears and can bear much more load as compare to the similarly designed spur gears.

Single helical gears impose radial as well as axial load on the bearings hence thrust bearings are used when helical gears are in operation. The angle of the helix must be the same on both the gears but opposite in direction.

2.4.1.3 Bevel Gears

Bevel gears are used to connect two intersecting but co-planer shafts. This type of arrangement is called bevel gearing. Straight bevel gears can be used at different angles but the right angle is the most commonly used angle.

There are further two types of the bevel gears.

2.4.1.3.1 Spiral Bevel Gears

In this type of bevel gears the teeth are oblique and they are quieter and can take more load as compared to straight bevel gears.

2.4.1.3.2 Zero Bevel Gears

These spiral gears are similar to straight bevel gears but the difference is in the form of teeth. Their teeth are curved lengthwise and meshing is done in such a way that overall spiral angle is zero.

2.4.1.4 Worm Gears

Worm gears are used to transmit power at right angle. They are used where high reduction in speed is required. Their axes do not cross each other but they cross in space. In this type of gears, one gear has screw threads. Due to this, worm gears are vibration free, quiet, and give a smooth output. Worm gears and worm gear shafts are almost at right angles to each other.

A very special feature lies with worm gears which is called self-locking system. It means that they can only rotate in one direction. The most common use is in harbor which is used to lift heavy load.

2.4.1.5 Rack and Pinion

A rack is a toothed rod that can be considered as a part of a gear with infinite radius of curvature. The rack is meshed with a pinion to convert rotational motion of the pinion to straight motion of the rack. This mechanism is used in automobiles the rotational motion of the steering wheel is converted into the left to right motion of the tie.

2.4.2 Shafts

Shafts are rotating members usually of circular cross section used in the gearbox to transmit power or motion from one stage to the other. It provides axis of rotation to rotating elements such as gears, pulleys, flywheels and sprockets. The shafts are clamped between the bearings so that shafts can rotate without much friction. In gearboxes they are subjected to bending as well as twisting moment due to tangential and radial loads on the gears. Two types of shafts are primarily used in the gearboxes, Keyed shafts and Splined shafts.

2.4.2.1 Keyed shaft

In these type of shafts a slot is machined on the shaft and gear. A key is placed between the slots of gear and shaft to mount the gear firmly. This key enable the rotation of the gear without any slippage. It is one of the most effective and economical process to transfer power and motion. They are used to transfer low torque. The slot in the shaft can cause unbalance during the rotation.

2.4.2.2 Splined shaft

In these type of shafts splines are cut to mount the gears which have an opposite mating spline cut into them to transmit rotational motion from one gear to the other without causing slip. Splines are essentially stubby gear teeth formed on the outside of the shaft and inside of the gear in opposite direction. The spline shafts are more expensive to manufacture than keyed shafts and used to transfer high torque.

2.4.3 Bearings

Bearings [5] are used in mechanical components to support motion of machine components, either rotatory or linear with nominal of friction. It is nearly impossible to design and operate a mechanical with no bearings. Almost everything that moves have bearings. Bearings are available in variety and are choose according to applications. On the basis of contacts bearing are divided in to two categories, contact and rolling. Bearings can have either a sliding or a rolling contact between mechanical parts. Sliding contact bearings are also termed plain or slider bearings and rolling contact bearings are also referred as anti-friction bearings in industry. On the basis of load bearings are divide into three categories as radial bearings, thrust bearings and linear bearings. Radial bearings are used to support rotating shafts with no axial loading, thrust bearings are used to support rotating shafts with axial loads and linear bearings are used to guide moving parts in a straight line.

2.4.3.1 Sliding contact bearings

Sliding contact bearings are used in low or modest speed applications. In this type of bearing moving mechanical parts have sliding contact between their surfaces. As they move relative to each other their surfaces have slide on each other. Due to sliding contact friction is present and heat is produced as a result of friction. In order to reduce friction between moving surfaces and heat produced lubrication is used in sliding contact bearings. As contact area between surfaces is greater in sliding contact bearings so higher friction is present and more lubrication is required. These bearings also require very precise alignment. Their operation is not noisy and their cost is also low.

Sliding contact bearings are further divided into three different categories as radial bearings, thrust bearings and linear bearings explained below.

2.4.3.1.1 Radial sliding contact bearings

These bearings are used to support rotating shafts with no axial loading. These bearings are usually have inner ring rotating. Outer ring is fixed in housing and the rotating shaft is inner ring is mounted on shaft. These type of bearings are used for high load and low speed shafts with normally spur gears mounted on the shafts. They require high precision in mounting for better operation.

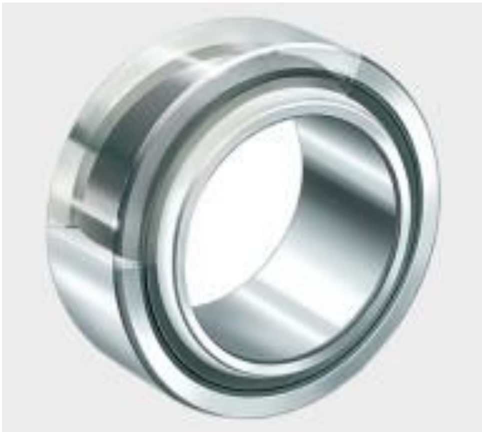

Figure 3: Sliding Contact Bearing

2.4.3.1.2 Thrust sliding contact bearings

These bearings are used to support rotating shafts with axial loading. When rotating shaft is subjected to axial load with high load and low speed thrust bearings are recommended.

2.4.3.1.3 Linear sliding contact bearings

These bearings are used to guide moving parts in a straight line. They are similar to rotary bearings with very large radius of curvature. They come in variety as dovetail slides, compound slides and rack slides.

2.4.3.2 Rolling contact bearings

Rolling contact bearings are used when application require high speed and low friction. In these type of bearings inner and outer surface of bearings are separated by rolling contacts between them. These rolling contacts may be ball, cylinder or needle depending upon the application. They can have either inner or outer ring rotating. As the contact between moving surfaces is low in these bearings so, low friction is present consequently low lubrication is required in these bearings.

On the basis of loading and speed these rolling contact bearings are also divided into following categories.

2.4.3.2.1 Ball bearings

Most commonly used type of rolling bearing in everyday application is ball bearing. These bearings are used where speed requirement is high and load is low. Due to low contact surface friction is minimum. Ball bearings can be either single row or double row depending on the no. of rows between the inner and outer surfaces of bearings. Single row ball bearings are also called deep groove ball bearings. They are used to handle low loads as they made point contact between bearing surfaces. Double row ball bearings are used in high load applications as contact between bearing surfaces is line contact. Due to line contact load is distributed over a larger area. Ball bearings are low cost and are used for continuous service.

Figure 4: Ball bearing

2.4.3.2.2 Roller bearings

Roller bearings are used in heavy load applications. In these bearings rolling contact is cylinder so sliding contact is a line rather a point as in the case of ball bearings. Due to line contact load is distributed over a larger area. Due to rolling contact friction is also low and less lubrication is required. If space for bearings is small then roller bearings with small diameters cylinder are used. These bearings are referred as needle bearings.

Figure 5: Roller Bearing

2.4.3.2.3 <u>Tapered bearings</u>

Tapered bearings are essentially like roller bearings except the fact the roller tilted rather than straight. These bearings are used to support large radial and axial thrust loading. They are often mounted in opposite direction on opposite side of shaft so they can support thrust on both sides. Due to rolling contact friction is also low and less lubrication is required.

Figure 6: Tapered Bearing

CHAPTER 3: METHODOLOGY

This chapter explains the procedure or methodology used in designing the gearbox. The process of gearbox design is divided into four categories.

- Gearbox layout
- Gear design
- Shaft deign
- Bearing design
- Housing
- Clutch selection

These process are discussed in details.

3.1 Gearbox layout

First step towards gearbox design is layout of gear train. Layout is a diagram showing how input and outputs are positioned. When drawing a basic layout following different factors are considered.

- Alignment of output
- Rotational direction of output
- Distance between outputs

These factors are dependent on one another. When drawing a basic layout all the requirements must be fulfilled.

3.1.1 Alignment of output

In a gearbox layout design first factor to be consider is the alignment of output, whether they are parallel or at some angle in relation to each other. The angle between outputs is decided on the basis of application. For our design all the outputs are three outputs are parallel to each other and are on the same side of gearbox.

3.1.2 Rotational direction of output

This is the most important to be consider while drawing basic layout of gearbox. Output will require certain rotational direction which cannot be neglected, otherwise desired results cannot be obtained. In our design both 12 kW generator and PVP 33/23 can rotate in either CW or CCW direction while 10kW generator is specified to rotate in CCW direction viewing from shaft side. These factors are considered in drawing basic layout of gearbox.

3.1.3 Distance between outputs

All the output equipment certainly have their own dimension that must be consider as the outputs are parallel to each other. Therefore, the distance between the outputs must also be consider in basic layout drawing. In this gearbox it is specified that all the outputs must be at least 210mm apart from each other.

Considering all the factors explained above a basic layout of gearbox is designed and is shown below. It meets all the requirements and is shown in results.

3.2 Gear Design

After drawing the basic layout next step is to design gear. Gear designing procedure [5] consist of following steps:

- Gear parameters
- Stresses on gear tooth

When all requirements of speed, torque and power are specified then gear parameters and stresses on gear tooth are calculated using the basic concepts of gear designing procedure.

3.2.1 Gear parameters

Different parameters [4] of gear design are shown in the figure [1] below and are their designing process is explained in the following section.

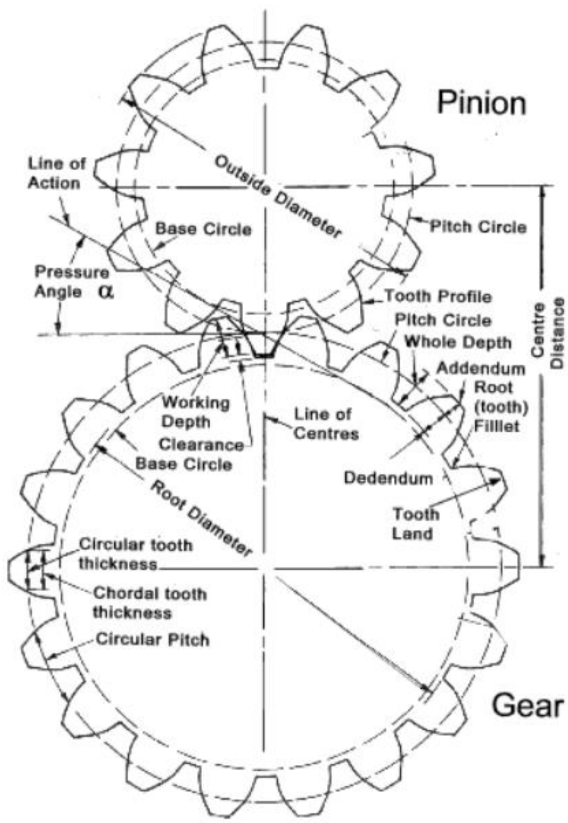

Figure 7: Gear-mesh

3.2.1.1 No. of teeth

The process of calculating no. of teeth is dependent on each gear-mesh RPM ratio e.

$$\mathrm{e} = \mathrm{output\ speed}/\mathrm{input\ speed}$$

After calculating e no. of teeth of pinion can be calculated from the relation,

$$N_p = \frac{2k\left(e + \sqrt{e^2 + (1+2e)sin\phi^2}\right)}{(1+2e)sin\phi^2}$$

After specifying the no. of teeth on pinion, no. of teeth on gear will be calculated using the relation of gear ratio,

$$N_g = \frac{\omega_p}{\omega_g} N_p$$

Pinion is driver while gear is driven.

No. of teeth of each gear and pinion is calculated from the formulas given above and are shown in the Results.

3.2.1.2 Module

Module is the ratio of pitch circle diameter to the no. of teeth. In our case it is selected to be 3mm.

3.2.1.3 Diameter of gears

In gears diameters are divided into further three categories as mentioned earlier.

a) Outside diameter (D_o)

The outer diameter of the gear is called outside diameter. It is calculated as,

$$D_o = (N + 2)m$$

b) Base/Root Circle diameter (D_b)

The diameter on which the involute teeth profile is based is known as Base Circle diameter. It is calculated as,

$$D_b = D_p - 2d$$

c) Pitch circle diameter (D_p)

Pitch circle is a theoretical circle upon which all calculations are based and the diameter of the pitch circle is calculated by using the relation,

$$D_p = m * N$$

Diameter of all the gears are calculated using the relation above and are shown in results.

3.2.1.4 Addendum

Addendum is the radial distance between the top land and the pitch circle. Addendum of 20^o full depth gear is equal to its module. In our case,

$$a = m$$

3.2.1.5 Dedendum

Dedendum is the radial distance between the bottom land and the pitch circle. Dedendum of 20^o full depth gear is calculated from the relation,

$$d = 1.25 * m$$

3.2.1.6 Circular pitch

Along the pitch circle, the distance between the corresponding points on adjacent teeth is termed as circular pitch and is calculated as,

$$p = \frac{\pi D_p}{N} = \pi * m$$

3.2.1.7 Tooth thickness

The thickness of a tooth measured along the pitch circle is called tooth thickness. As its definition indicates it is half of module.

$$t = 0.5 * m$$

3.2.1.8 Width of space

The amount of space between two adjacent gears on pitch circle and is equal to tooth thickness on pitch circle. Sum of tooth thickness and width of space is equal to module.

$$s = t = 0.5 * m$$

3.2.1.9 Tooth width

Width of a gear parallel to its axis of rotation is defined as tooth width. For each mesh both pinion and gear have same tooth width. Criteria for tooth width is that it should be between the range of $9m \leq b \leq 15m$ or up to 60% of pinion diameter. The width of all the gears are 30mm except the gear mesh 2 where it taken 54mm due to high torque requirement.

3.2.1.10 *Center to center distances*

When two parallel gears are in a mesh the distance between their centers is called center distance. It is calculated by averaging the diameter of mating gears.

$$h = \frac{D_p + D_g}{2}$$

3.2.1.11 *Whole depth*

Total depth of the space between teeth of a gear from top to the depth of tooth is named as whole gear and is equal to the sum of addendum and dedendum.

$$h_w = a + d$$

3.2.1.12 *Clearance*

The clearance is the difference between dedendum and addendum of mating gears in a mesh.

$$c = d - a$$

3.2.1.13 *Pressure angle*

The pressure angle is defined as the angle formed between the radial line and the line tangent to involute profile at the pitch point. In our case all the gears have pressure angle of 20^o. Other available PA are 14.5^o and 25^o.

3.2.1.14 *Working depth*

Difference between whole depth and clearance is called working depth. It is two times the module.

$$h_{wr} = a + d - c = 2 * m$$

3.2.2 Stress calculation

After calculating and establishing all the tooth parameters, next step is to determine the stress on gear tooth. Stress calculation on gear is further divided into two steps.

3.2.2.1 <u>Force on gear tooth</u>

When gears are in mesh force acts on gear tooth as they rotate. This force act on gear tooth as pinion try to drive the gear. The force on pinion and gear is always equal in magnitude and opposite in direction. This acting force on the gear tooth can be divided into tangential and radial components.

3.2.2.1.1 <u>Tangential/Transmitted Force</u>

When gears are in mesh the deriving torque is due to the tangential component of force. This component acts tangent to the pitch circle.

$$F_t = \frac{2 * T}{D_p}$$

For the calculation of the torque following relation is used,

$$T = \frac{W * 60}{n * 2\pi}$$

We require power W and speed in RPM at each shaft to calculate the torque. The torque on the shaft is transmitted through gears. The input and output power at each output is known. Similarly we know the RPM of each shaft. By using the torque relation above mentioned the torque of each shaft or gear is calculated. For the calculation of tangential force the diameter of small gear was used in each mesh. Small diameter gives us the maximum tangential force acting in a mesh or pair of gears.

3.2.2.1.2 Separating Force

Radial component of force acts along the radius towards the center of the gear. It tends to separate the gears from each other. The two forces are related by the following relation.

$$F_s = F_t \tan \phi$$

The force on each gear and pinion is given below in the table.

3.2.2.2 Bending Stress calculation

Stress on gear tooth is only due to tangential component of the force as it tend to bend, break or shear the tooth from root. For the sake of simplification we consider gear tooth a simple cantilever beam with tangential force acting at its free end as shown in figure [4] (a). Gear tooth

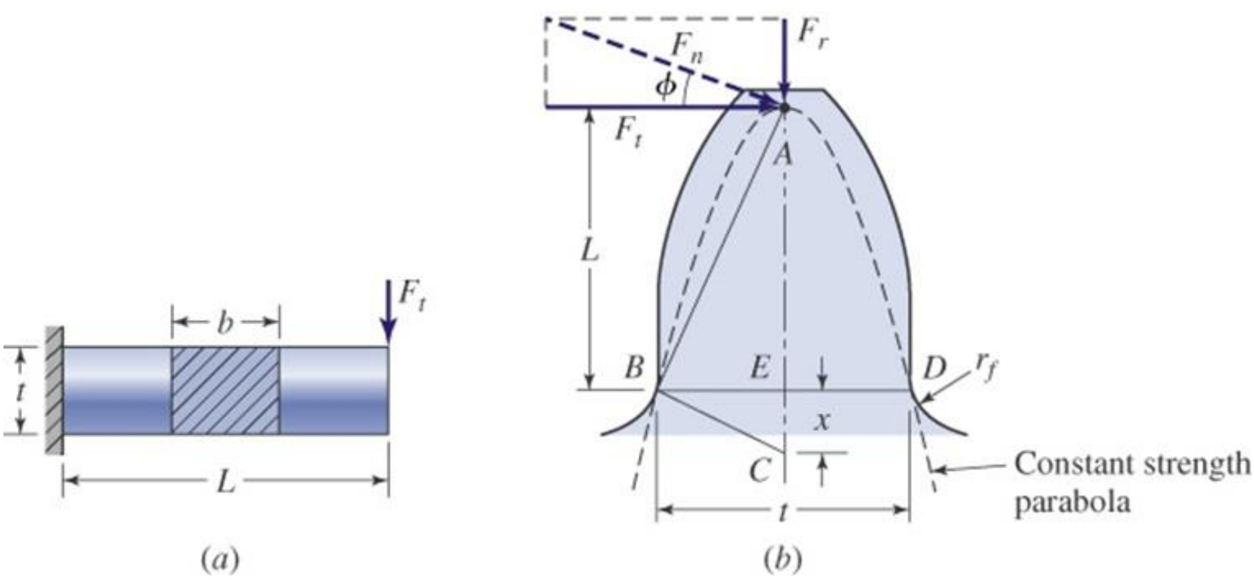

Figure 8: Cantilever beam (a) Gear-tooth (b) [4]

is similar to cantilever beam with variable thickness profile, all the forces and tooth profile is indicated is indicated in figure (b).

Using this assumption the governing equation for calculation of stress on gear tooth is given by Modified Lewis Stress Equation [4] for bending strength,

$$\sigma = \frac{F_t}{bmJ} K_v K_o K_h K_s K_B$$

Here,

σ is stress on gear tooth,

F_t is tangential force,

b is width of gear, m is module,

J is geometry factor for bending strength,

K_v is dynamic factor,

K_o is overload factor,

K_m is load distribution factor,

K_b is rim thickness factor,

K_h is load distribution factor,

K_s is size factor.

Tangential force, width and module for each gear is already mentioned above. All other parameters are explained below.

3.2.2.2.1 <u>Geometry factor (J)</u>

The geometry factor [6] is used to estimate tooth profile and the position at which the most detrimental load is acting. It includes both, tangential and radial components of force. A detailed study is conducted for estimation of geometry factor in the past. Most authentic and most used research in this field is *AGMA 908-B89.* An extracted graph from this paper for geometry factor calculation of 20^{o} full depth spur gear is shown in figure below. Using this graph and gear parameters, the geometry factor for corresponding gears are estimated and shown in results.

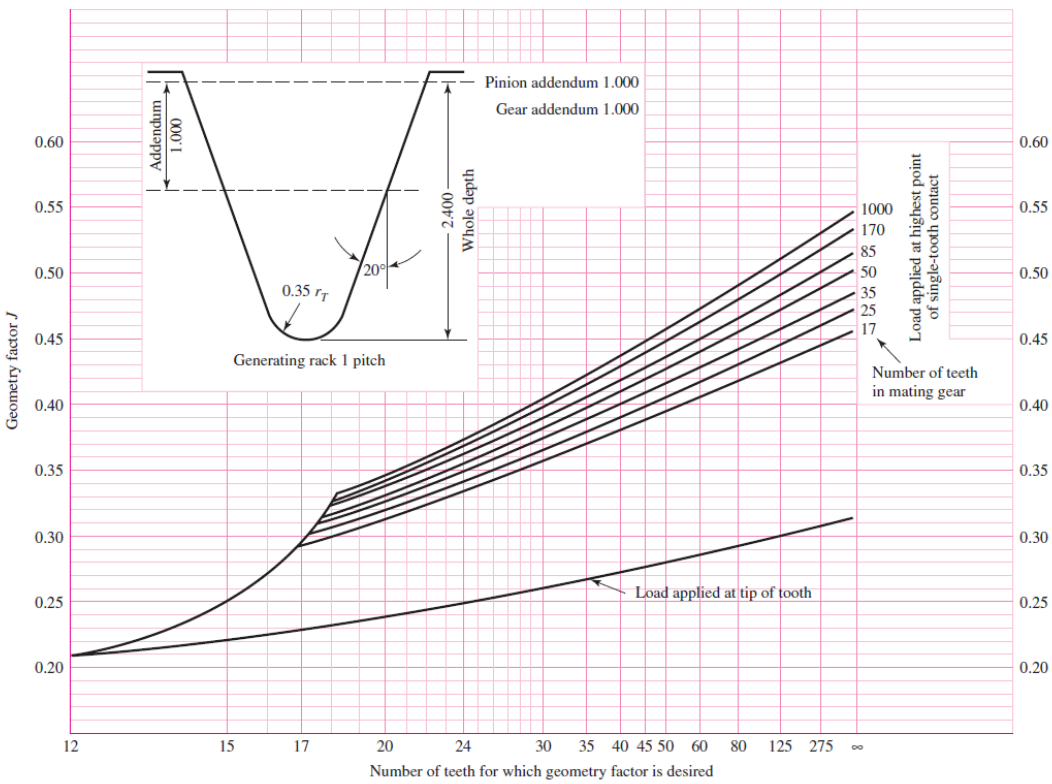

Figure 9: Spur-gear geometry factors J [4]

3.2.2.2.2 Dynamic factor (K_v)

Dynamic factor [7] accounts for the forces due to relative velocity and acceleration during the motion of gears. This relative acceleration or velocity is as the power, torque or speed requirement from input to output vary. Even for constant power, torque and speed considerable vibrations in mating gear exist, therefore we have to account for dynamic force. These dynamic forces are due to relative accelerations between gears due to their vibration. This vibration is due to excitation called *Transmission error*, defined as the deviation in speed from constant relative angular speed of mating gears. Dynamic factor includes both the internal dynamic effects and the tangential force on gear tooth. Dynamic factors for different gear types are listed [4] below.

$$K_v = \frac{3.05 + V}{3.05} \quad \text{(cast iron, cast profile)}$$

$$K_v = \frac{6.1 + V}{6.1} \quad \text{(cut or milled profile)}$$

$$K_v = \frac{3.56 + \sqrt{V}}{3.56} \quad \text{(hobbed or shaped profile)}$$

$$K_v = \sqrt{\frac{5.56 + \sqrt{V}}{5.56}} \quad \text{(shaved or ground profile)}$$

$$K_v = \frac{50 + \sqrt{V}}{50} \quad \text{(hobbed or shaped profile)}$$

$$K_v = \sqrt{\frac{78 + \sqrt{V}}{78}} \quad \text{(shaved or ground profile)}$$

$$K_v = \frac{600 + V}{600} \quad \text{(cast iron, cast profile)}$$

$$K_v = \frac{1200 + V}{1200} \quad \text{(cut or milled profile)}$$

Gears manufactured in our project are machined so we use the relation

$$K_v = \frac{6.1 + V}{V}$$

Here, V is *pitch line velocity*, defined as the linear velocity of a point on the pitch circle. It is always tangent to pitch circle at pitch point. Dynamic factor pitch line velocity and Geometry factor for all the gears are shown in results.

3.2.2.2.3 Overload factor (K_o)

Overload factor [7] is used calculate for all external forces that are not included in calculations for required application. It is obtained from the table given below.

Table of Overload Factors, K_o

	Driven Machine		
Power source	Uniform	Moderate shock	Heavy shock
Uniform	1.00	1.25	1.75
Light shock	1.25	1.50	2.00
Medium shock	1.50	1.75	2.25

Figure 10: Overload Factor [4]

Our application operates moderate shock for driven machine and medium shock for power source so overload factor comes out to be ***1.75***.

3.2.2.2.4 Load distribution factor (K_h)

Load distribution factor [7] is used in the stress equation to indicate the non-uniform load distribution across the line of contact. Load distribution factor is determined from the table given below for specific type of application.

	Face width (mm)			
Characteristics of Support	0 - 50	150	225	400 up
Accurate mountings, small bearing clearances, minimum deflection, precision gears	1.3	1.4	1.5	1.8
Less rigid mountings, less accurate gears, contact across the full face	1.6	1.7	1.8	2.2
Accuracy and mounting such that less than full-face contact exists	Over 2.2	Over 2.2	Over 2.2	Over 2.2

Figure 11: Load Distribution Factor [8] [8]

Gears in this application are designed for accurate mounting, small bearing, minimum deflaction and preccision gears with nearly 0-50mm face width so K_h is choosed to be 1.3.

3.2.2.2.5 Size Factor (K_s)

Size factor [7] is used to indicate non-uniformities of properties in gear material present due to variation in size. It is affected by various factors such as heat treatment of gear material, gear diameter, its overall size and their ratio. Standard size factor for gear tooth is not yet been estimsated for circumstances when detrimental size affects. So, for these scenario AGMA incourages to use $K_s > 1$. If detrimental size effect is zero, as in our case, use $K_s = 1$.

3.2.2.2.6 Rim thickness factor (K_B)

When the rim is not thick enough to fully support for the tooth base, the gear may fail through the rim of gear instead of tooth base. For these scenario, Rim thickness factor [7] K_h is used and its value is greater than unity otherwise not. It is basically used for thin rim gears. Its value is estimated from backup ratio m_B,

$$m_B = \frac{t_R}{h_t}$$

Here, t_R is rim thickness and h_t is tooth height. Knowing the value of m_B the value of rim thickness factor can be established from graph [7].

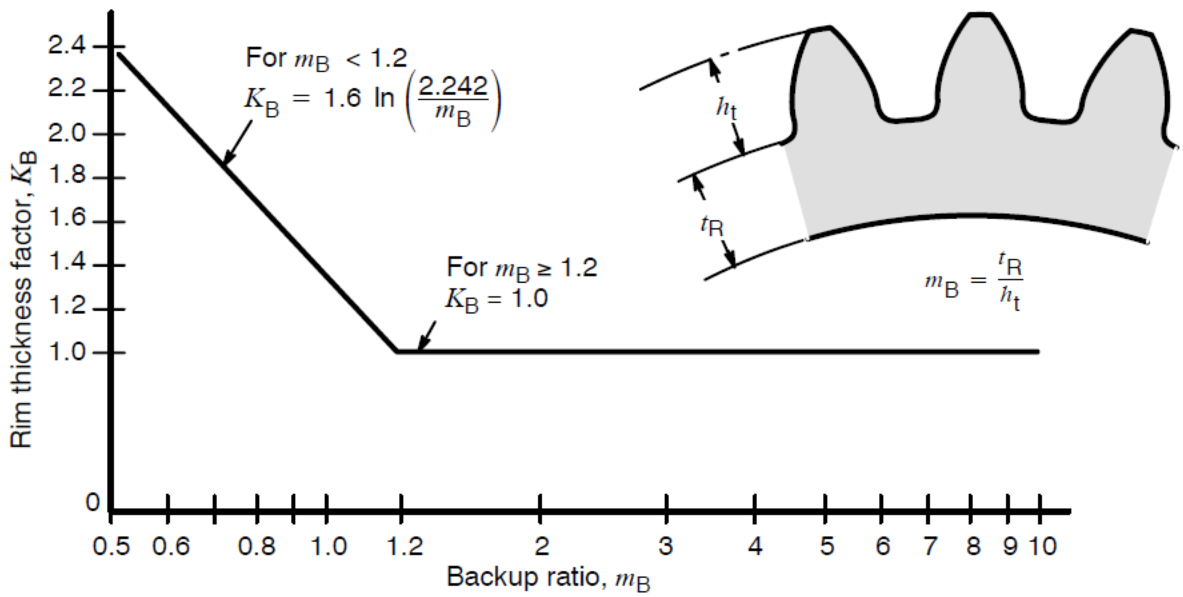

Figure 12: Rim thickness factor [7]

In our case backup ratio is more than 1.2 for every gear so rim thickness factor is unity.

3.2.2.2.7 Factor of safety (FS)

Factor of safety indicates by how much value the allowable stress exceeds the applied stress. It is calculated as,

$$FS = \frac{\sigma_{ut}}{\sigma}$$

Bending stress on each gear and FS are shown in results.

Bending stress calculations are considered to be complete if FS meet the design requirements. After completing the bending stresses calculations for gears next step is to design shafts.

3.3 Shaft design

Shafts are used to mount gears on them. First of all we require a layout to know the position of gears.

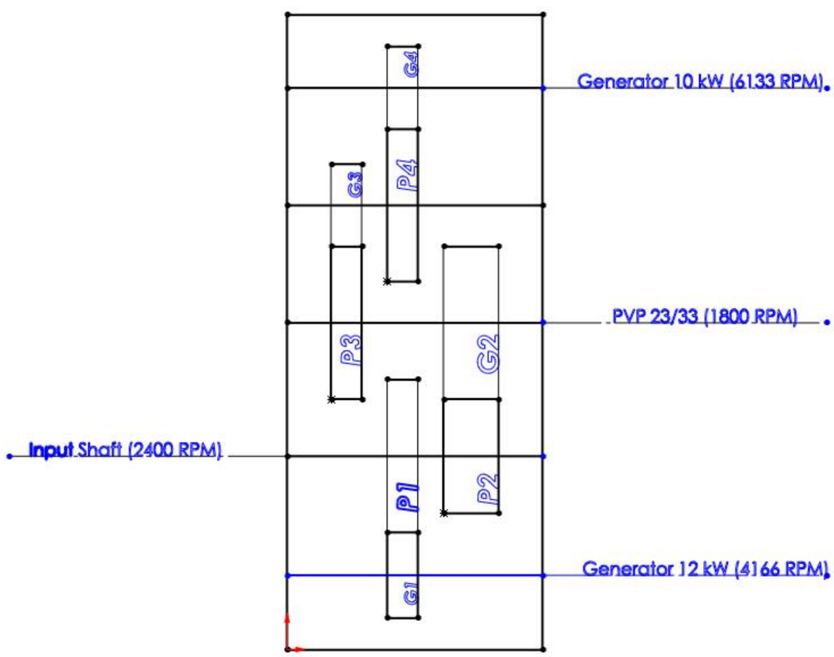

Figure 13: Basic layout of Gearbox

Once the basic layout is established length can be evaluated as shown in figure. After evaluating the length of shafts and distances of the points where force is applied (gear mounting points) from end of shaft are known, so we can now perform shear stress analysis. Shaft designing process [9] is basically carried out by determining shear stress, bending moment and twisting moment or torque calculations on shaft during its rotation. Shear stress is calculated by drawing all the forces on the shear force diagram. These forces include force on gear and the reaction forces due to bearing. As the force on gear is resolved into its radial and tangential components, the reaction force responds in the very same way. Reaction forces act in the same plan as acting forces. As the values for these forces and their reaction are evaluated we draw shear force diagram for both radial and tangential components in the same plane. Values of these forces are calculated using simple concepts of mechanics and are shown in results for each shaft.

With the help of shear force diagrams then the bending moment diagram for both tangential and radial component of bending moments are drawn and its magnitude is evaluated. Resultant of maximum bending moment is then calculated for further calculations. Moment values for each shaft are shown in results.

Torque or twisting moment is dependent on power requirement at each output. If a shaft is undergoing different torque at different gear mesh, maximum value of torque is used for calculations. Value of torque or twisting moment is calculated from the relation,

$$T = \frac{W * 60}{RPM * 2\pi}$$

Value of torque for each shaft is calculated and shown in results.

Resultant of bending moment and torque is called *equivalent twisting moment (T_e)* [4] and it can be defined as the twisting moment, which can produce same shear stress as actual twisting and bending moment.

$$T_e = \sqrt{(K_m M)^2 + (K_t T^2)}$$

Here, K_m is combined shock and fatigue factor for bending and K_t is combined shock and fatigue factor for torsion. Values of Kt and Km are estimated from the table below.

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

Figure 14: Combined shock and fatigue factor for bending and torsion [8]

Our application is rotating shaft with suddenly applied load and moderate shock so, Km and Kt are choose to be 2. T_e value for each shaft is shown in results.

Equivalent twisting moment (T_e) is related to shear stress and diameter of the shaft by following formula,

$$T_e = \frac{\pi}{16} * \tau_{all} * d^3$$

Here, τ_{all} is allowable shear stress and d is shaft diameter to be calculated. Rearranging the terms in above equation diameter of shaft can be determined.

$$d = \left[\frac{16 * T_e}{\pi * \tau_{all}}\right]^{1/3}$$

Length of shafts depends upon the distance between width of bearings and gears, distance of output equipment from gearbox and clearance from the box.

As all the shafts are designed with FS value of 2 so we can say it's a safe design, yet it is analyzed on SolidWorks for maximum deflection. Results are favorable and are discussed in next results.

3.4 Bearing design

Bearing are used for constrained rotational motion between two or more parts. In gearbox application they are used to support the shaft and help it in rotation. It is placed between the shaft and the housing. It can be inner ring rotating or outer ring rotation. In our case it is inner ring rotating. Bearing are of standard sizes to reduce manufacturing cost. The bearing are used for this gearbox are from SKF. Different models are used for different shaft diameters. Deep groove ball bearings are used for this gearbox as no axial load is applied and high speed is required. Also it has low noise and little maintenance. A detailed diagram of deep groove ball bearing is shown in figure.

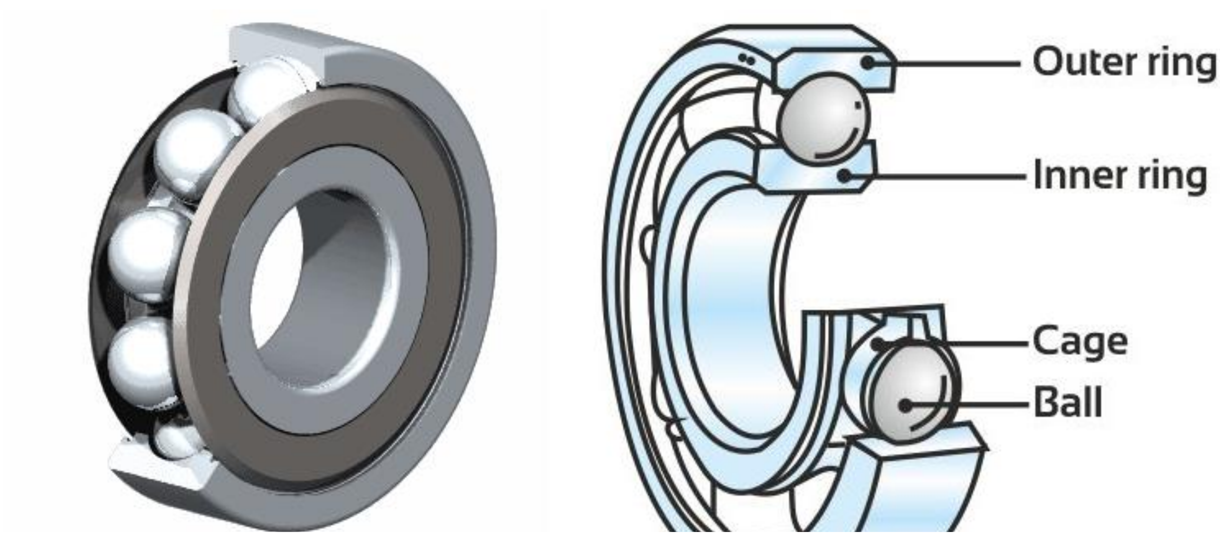

Figure 15: Ball Bearing

Design process is carried out first estimating the *design life* or *rating life* of the bearing which can be defined as, for required speed 90% of the bearings from the same group will complete or exceed the number of revolutions or hours of operation before failure. SKF urges to use L_{10} life to be one million revolution.

After establishing the rating life next step is to determine the basic dynamic load rating C_{10} given by *Weibull Distribution* [4],

$$C_{10} = F_D \left[\frac{\left(L_D * N_D * 60 / L_{10} \right)}{0.02 + 4.439[\ln(\frac{1}{R})]^{1/1.483}} \right]^{\frac{1}{a}}$$

Here, F_D is desired radial load and it is resultant of radial and tangential reaction forces calculated in shaft designing process during shear force calculations r, L_D is desired life in hours for which bearing is designed, R is reliability and L_{10} is rating life. In our case desired life is 10,000 hours and reliability is 99.9% for all bearing except two bearings. These two bearing have reliability of 99% due to design constraints and *a* is exponent for the life equation its value is 3 for ball bearing.

Select each shaft rotating at certain speed for which bearing is to be designed for desired life in hours. Reliability is 99.9% for all bearing except two bearings. These two bearing have reliability of 99% due to design constraints. Putting in all the values of variables a basic dynamic load rating value of bearing is calculated which is then compared to the catalog basic dynamic load rating C of the selected bearing. Bearing is selected if $C \geq C_{10}$ for desired diameter of shaft and bearing.

Shafts may have different radial force on its end so bearings are designed accordingly. All the bearings are selected from SKF and are shown in results.

3.5 Housing

When all gears, shafts and bearing are selected, all we need is a housing to put all these components in it. Housing is to be made by casting process. Housing along with all the components assembled in it is shown in result section.

CHAPTER 4: RESULTS

All the results obtained in methodology are shown here.

4.1 Gearbox layout

Basic layout of gearbox is shown here. It fulfill all the factors explained in methodology.

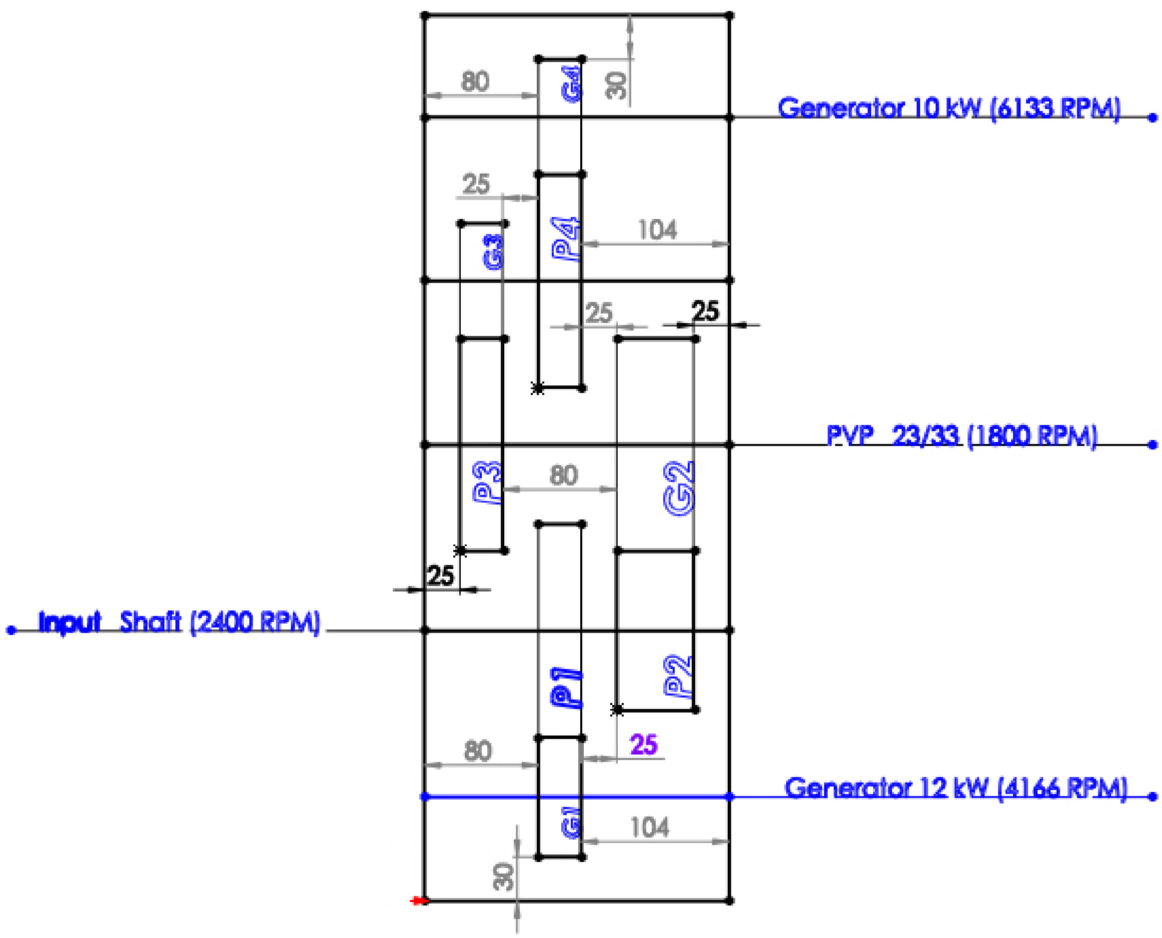

Figure 16: Basic layout diagram of gearbox

Diagram is explained in the next table.

Table 1: Layout explanation

Shaft	Description	Shaft Rotation Direction viewing up to down	Mesh	Gear	
1	Input Shaft	CW		Pinion/ driver	Gear/ Driven
2	Output Shaft 12kW Gen	CCW	1	P1	G1
3	Output Shaft PVP23/33	CCW	2	P2	G2
4	Counter Shaft	CW	3	P3	G3
5	Output Shaft 10kW Gen	CCW	4	P4	G4

4.2 ***Gear parameters***

4.2.1 No. of teeth

No. of teeth for each gear calculated as per requirements are listed below. Gears are named in layout. The no. of teeth were calculated using the relations explained in methodology

Table 2: No. of teeth on gears

Mesh No.	No. of teeth on pinion (Np)	No. of teeth on gear (Ng)
1	48	27
2	36	48
3	48	26
4	48	26

4.2.2 Various other parameters

Module	3mm	Circular pitch	9.425mm
Addendum	3mm	Dedendum	3.75mm
Whole depth	6.75mm	Working depth	6mm
Clearance	0.25mm	Pressure Angle	20^o
Tooth thickness	1.5mm	Width of space	1.5mm

4.2.3 Diameters of gears

Outer diameter, base diameter and pitch diameter are listed below for each gear. These diameters were calculated using the relations explained in methodology.

Table 3: Gear Diameters

Gear	D_o (mm)	D_b (mm)	D_p (mm)
P1	150	136.5	144
G1	87	73.5	81
P2	114	100.5	108
G2	150	136.5	144
P3	150	136.5	144
G3	84	70.5	78
P4	150	136.5	144
G4	84	70.5	78

4.2.4 Center distance and gear width for each mesh

The center distance and gear width for each mesh are listed below. Each mesh is consist of a pinion and gear and are numbered accordingly. Gear width for both pinion and mesh is same in each mesh. Center distance and gear width for each mesh is calculated using the relations explained in methodology.

Table 4: Centre Distance and Width of gears

Mesh	Center distances (mm)	Gear width (mm)
1	112.5	30
2	126	54
3	111	30
4	111	30

4.2.5 Force on each tooth

Magnitude of tangential and radial components of force on gear tooth are listed below. As in a mesh force magnitude both gear tooth is equal and is controlled by smaller gear so, forces for gear are listed below. This is explained in methodology.

Table 5: Force components on Gear tooth

Mesh	Tangential Force F_t (Newton)	Radial Force F_s (Newton)
1	666.6	242.62
2	2944.4	1071.7
3	384.6	140
4	384.6	140

4.2.6 Geometry factor, Pitch line velocity and Dynamic factor

Geometry factor, Pitch line velocity and Dynamic factor for each gear are listed below. Geometry factor and Dynamic factor for calculation of Bending Stress calculation on gear tooth and Pitch line velocity is used in Dynamic Factor Calculation.

Table 6: Geometry factor, Pitch line velocity and Dynamic factor for each gear

Gear	Geometry factor (J)	Pitch line velocity (V) m/s	Dynamic factor (K_v)
P1	0.421	18.1	3.97
G1	0.391	18.1	3.97
P2	0.418	13.6	3.22
G2	0.431	13.6	3.22
P3	0.419	13.6	3.22
G3	0.387	13.6	3.22
P4	0.419	25.1	5.1
G4	0.387	25.1	5.1

4.2.7 Other factors

Overload Factor	1.75	Load distribution factor	1.3
Size factor	1	Rim thickness factor	1

4.2.8 Bending stress and Factor of safety

Bending stress and Factor of safety for each gear is calculated using the method explained in methodology and is listed below. FS is more than required value (1.25) for each gear.

Table 7: Bending stress and Factor of safety for each gear

Gear	Bending stress (MPa)	Factor of safety
P1	158.9	2.61
G1	171.1	2.43
P2	318.5	1.30
G2	308.9	1.34
P3	74.7	5.55
G3	80.9	5.13
P4	118.3	3.50
G4	128	3.24

4.3 <u>Shaft Parameters</u>

4.3.1 Diameter, Direction of rotation Gears mounted and Use

Diameter, Direction of rotation, gears mounted on each shaft and its role in gearbox is tabulated below. Shafts are numbered in layout. The process of designing shafts is explained in methodology.

Table 8: Diameter, Direction of rotation Gears mounted and Role of each shaft

Shaft	Diameter (mm)	Description	Shaft Rotation Direction viewing up to down	Gears mounted on shaft	
				Input Side	Output side
1	40	Input Shaft	CW	P1	P2
2	25	Output Shaft 12kW Gen	CCW	G1	
3	40	Output Shaft PVP23/33	CCW	P3	G2
4	20	Counter Shaft	CW	G3	P4
5	20	Output Shaft 10kW Gen	CCW	G4	

4.3.2 Tangential force and Radial force

Tangential force and Radial force on each shaft are listed below. Shear and moment diagrams corresponding to these values are shown in APPENDIX I.

Table 9: Tangential Plane Forces on each shaft

Shaft	Tangential plane forces (N)		Reaction forces on tangential plane (N)	
	Input Side	Output side	Input Side	Output side
1	666.6	2944.4	406.4	1871.4
2	666.6		367.83	298.2
3	384.6	2944.4	470.8	2089
4	384.6	384.6	91.18	384.6
5	386.6		77.24	62.75

Table 10: Radial Plane Forces on each shaft

Shaft	Tangential plane forces (N)		Reaction forces on tangential plane (N)	
	Input Side	Output side	Input Side	Output side
1	242.6	1071.7	147.83	680.87
2	242.6		133.84	108.75
3	140	1071.7	171.35	760.29
4	140	140	33.19	33.19
5	140		212.19	172.39

4.3.3 Bending moment, Twisting moment and Equivalent twisting moment

Bending moment, twisting moment and equivalent twisting moment for each shaft are listed below. Bending moment diagrams are shown in APPENDIX.

Table 11: Bending moment, Twisting moment and Equivalent twisting moment on shafts

Shaft	Bending moment (Nm)	Twisting moment (Nm)	Equivalent twisting moment (Nm)
1	121.5	186	444.3
2	40.7	27	97.7
3	135.6	144	395.6
4	12.5	15	38.9
5	23.5	15	55.8

4.4 Bearings Parameters

Radial force, Basic Dynamic Load Rating, Catalog Basic Dynamic Load Rating, Bore and model for each bearing on both side of gearbox are listed below. Design procedure for these parameters is explained in methodology.

Table 12: Input side Bearings parameters

Shaft	Input side bearing				
	Fr (kN)	C_{10} (kN)	C (kN)	Bore (mm)	Model (SKF)
1	0.43	18.38	22.1	40	62208-2RS1
2	0.39	20.17	23.4	25	6305
3	0.5	19.4	22.1	40	62208-2RS1
4	0.097	4.61	6.37	20	61904
5	0.225	13.13	13.5	20	6204

Table 13: Input side Bearings parameters

Shaft	Output side bearing				
	Fr (kN)	C_{10} (kN)	C (kN)	Bore (mm)	Model (SKF)
1	1.99	56.84	63.7	40	6408
2	0.317	16.03	23.4	25	6305
3	2.22	56.6	63.7	40	6408
4	0.097	4.61	6.37	20	61904
5	0.183	10.68	13.5	20	6204

4.5 <u>Results achieved</u>

Results achieved for different outputs are listed below.

Table 14: Operating and Required speed of each shaft

Output	Detail	Required RPM	Operating RPM
1	Output Shaft 12kW Gen	4000-4500	4166
2	Output Shaft PVP23/33	1800	1800
3	Output Shaft 10kW Gen	5500-6500	6133

Input speed = 2400 RPM

4.6 <u>Gearbox 3D diagram</u>

A 3D diagram of gearbox is shown in following figure.

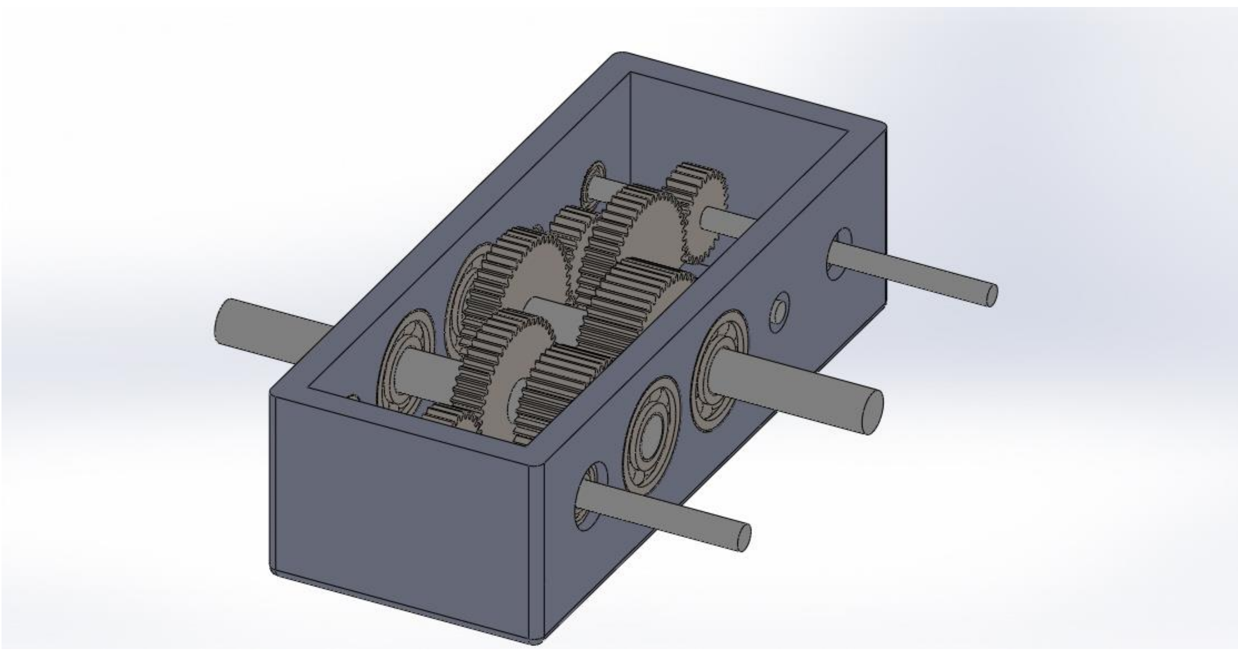

Figure 17: Trimetric View of Gearbox

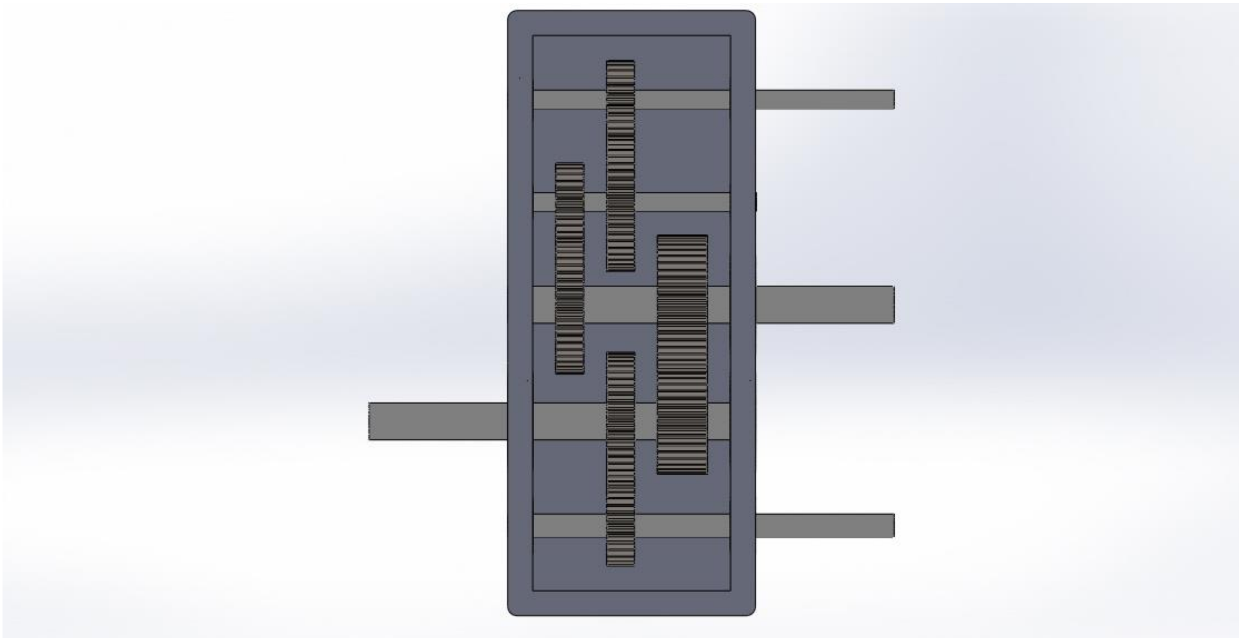

Figure 18: Top View of Gearbox

CHAPTER 5: CONCLUSION

A gear box to be designed that will distribute the power from a single input to multiple outputs to move a mechanical generator and two pumps simultaneously that would run at different RPM and torques. Normally the gear boxes used of such a type provide power in a continuous manner to all outputs and if a user wants to manage a single output it won't be possible without shutting down the whole system.

Our project includes the designing and development of a multi-clutch gear box. The gear box will deliver power from one input source to three output sources. The input source would be an engine which provide power to one pump (PVP33) and two generators of 10 kW and 12 kW power.

In our design all the outputs are inline to each other so spur gears would be the most suitable option for us. Another reason for using spur gears is that they can operate at higher RPM and their gear ratios are quite high. Also this type of gears is easily available in the market.

The project uses Multi-stage gearbox reduction with the help of counter-shaft. For the ease of use electrically actuated clutches to be attached with inputs and outputs which are engaged and disengaged whenever required. This multi-faceted design is very versatile which can be extended in future.

Using modified Lewis Stress equation, transmitted force and taking into account other design factors and dynamic factors gear has been designed. Shear force and bending moment diagram has been made for each shaft and result has been verified according to the safety factors. Great care has to be taken while assembling and mounting the gear box assembly.

Our gear box system is just like automobile transmission system where clutches are used to turn on and off the power supply. We use four electromagnet clutches. One at input power supply and three on output shafts for pump and generators .The design of the clutch is not included in the project rather standard clutches would be used available in the market.

The gear box system is designed for a particular client at specific design specifications but it very adaptable project that can be modified and revised at any stage can be used in any industry where such power transmission is required. This system is the most suitable where multiple systems operate at the same time and it would be easy to operate them with a single input rather than having a separate power source for every single system.

This is a very comprehensive product that revolutionize work in very efficient way, encourage feasible and sustainable industrialization.

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APPENDIX I: SHEAR FORCE AND BENDING MOMENT DIAGRAMS

Shear force and bending moment diagrams of all the shafts are shown here.

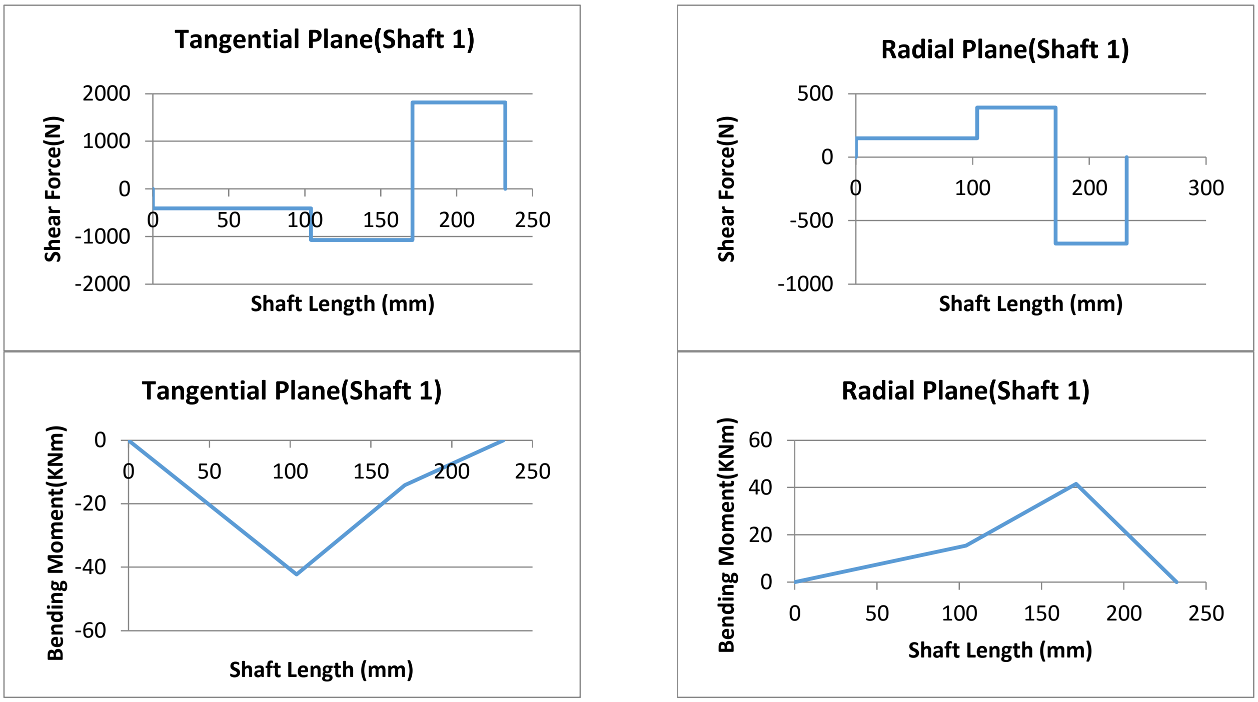

Figure 19: Shear Force and Moment Diagrams of Tangential and Radial Planes for shaft 1

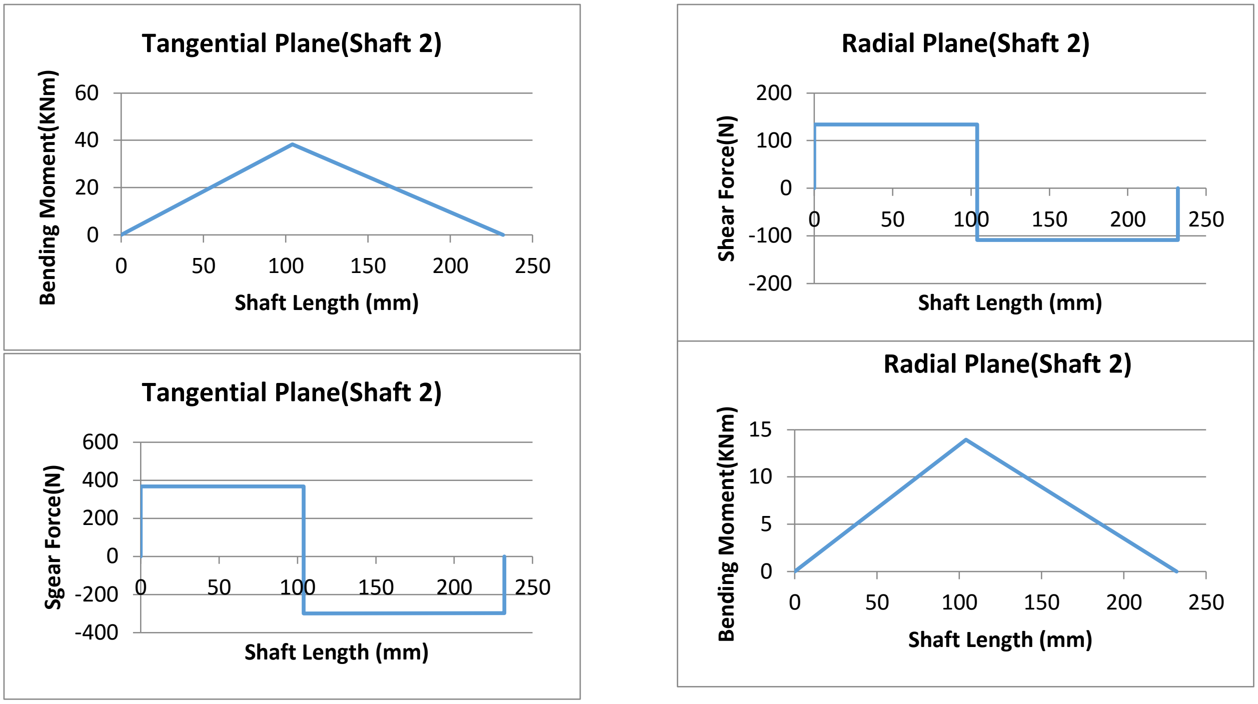

Figure 20: Shear Force and Moment Diagrams of Tangential and Radial Planes for shaft 2

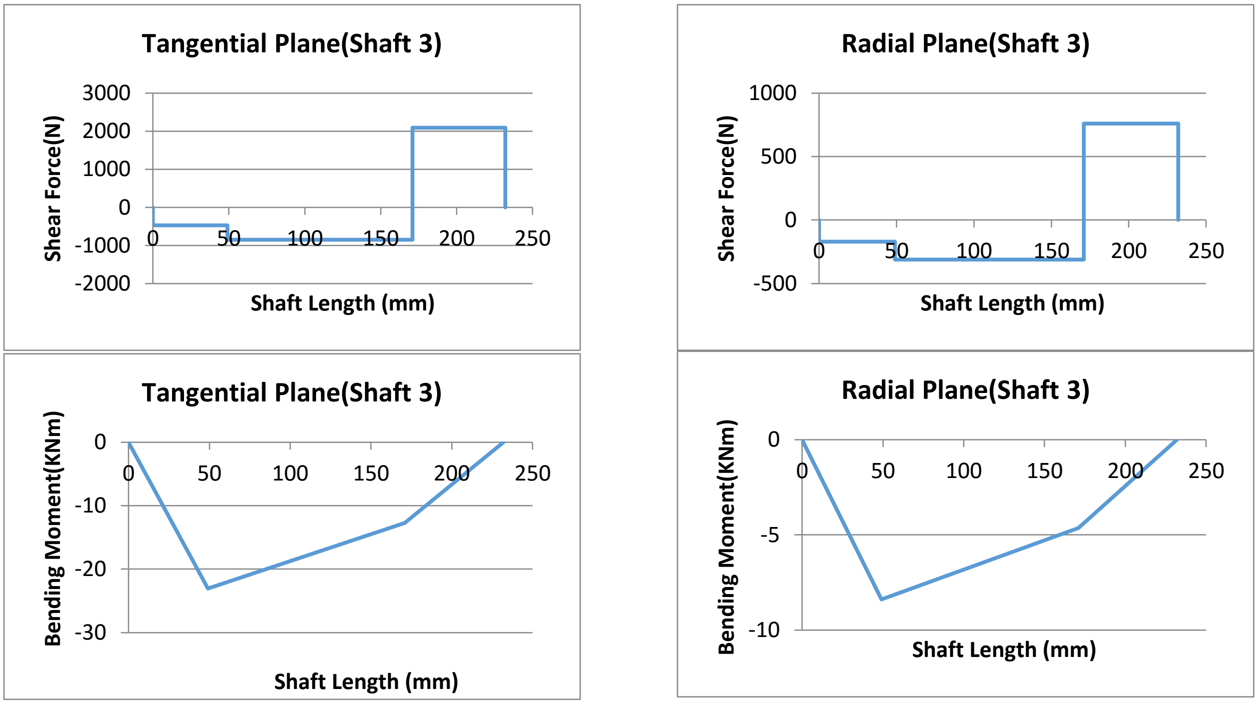

Figure 21: Shear Force and Moment Diagrams of Tangential and Radial Planes for shaft 3

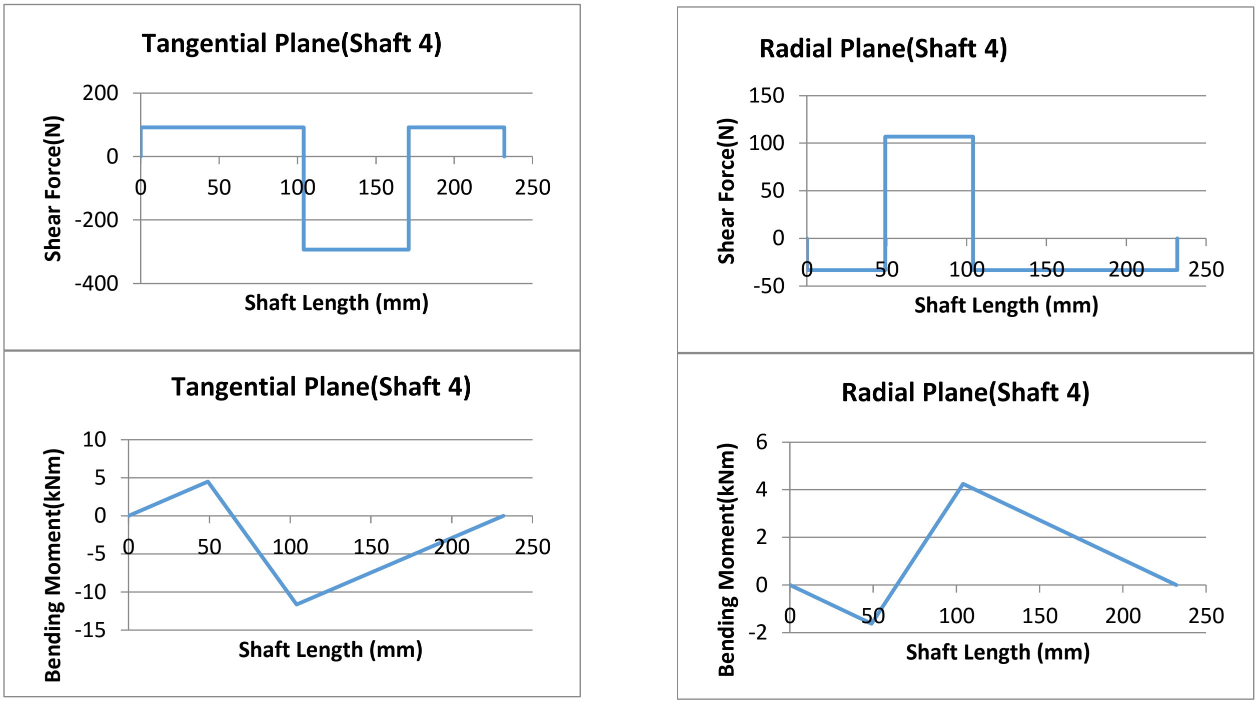

Figure 22: Shear Force and Moment Diagrams of Tangential and Radial Planes for shaft 4

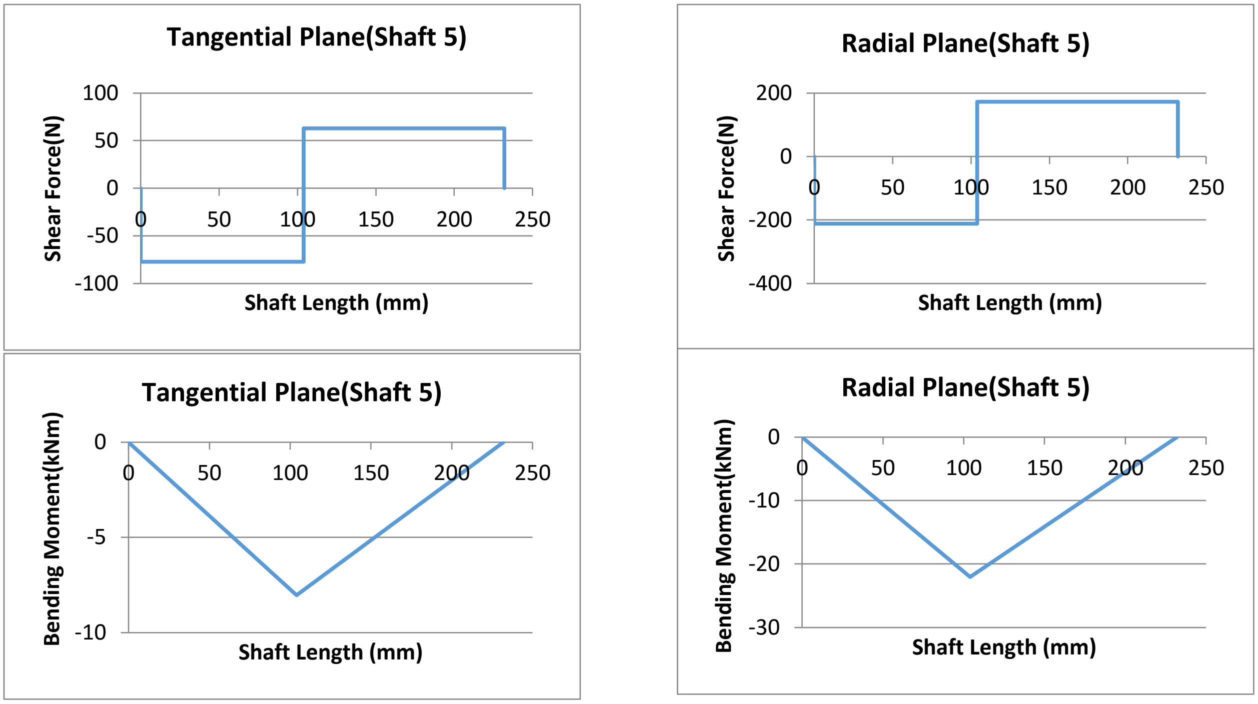

Figure 23: Shear Force and Moment Diagrams of Tangential and Radial Planes for shaft 5