Design of Bar Peeling Attachment on Lathe Machine

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DEPARTMENT OF MECHANICAL ENGINEERING

INSTITUTE OF TECHNOLOGY

NIRMA UNIVERSITY

AHMEDABAD-382481

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Design of Bar Peeling Attachment on Lathe Machine

Major Project

Submitted in partial fulfillment of the requirements

for the degree of

Master of Technology in Mechanical Engineering

(Design Engineering)

By

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Guided By

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DEPARTMENT OF MECHANICAL ENGINEERING

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Declaration

This is to certify that

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I, **Ankur N Patel, Roll No. 12MMED08**, give undertaking that the Major Project entitled **"Design of Bar Peeling Attachment on Lathe Machine"** submitted by me, towards the partial fulfillment of the requirements for the degree of **Master of Technology in Mechanical Engineering (Design Engineering) of Nirma University, Ahmedabad**, is the original work carried out by me and I give assurance that no attempt of plagiarism has been made. I understand that in the event of any similarity found subsequently with any published work or any dissertation work elsewhere; it will result in severe disciplinary action.

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Abstract

The purpose of this project is to design bar peeling attachments on lathe machine. Bar peeling is a kind of machining process. In bar peeling operation, cutting tool is rotated and bar is fed longitudinally towards the cutter head. Bar peeling operation is used to remove surface cracks, oxides and chilled skin from forged bar. This semi finished bars are delivered to the industries and are used as an intermediate stage in the production of products. For example: axle components for the automobile industry, extrusion blanks for tube manufacturing and for hydraulic cylinders, etc.

In lathe machine, bar is fixed in the chuck and rotated by the head stock. During the conventional turning operation, the cutting tool is traveling against the side of the bar and it is fed paralleling the longitudinal axis of the bar. The modified lathe machine comprising of a cutter head and bar feeding mechanism. In which cutting tool only rotate in a chuck and bar is fed longitudinally towards the cutter head. There are four cutting toolholders in cutter head. Out of which, two cutting tools are used for the rough turning and remaining are used for the finishing operation. By this modified lathe turning operation, long bars can easily peeled.This is low cost solution provided for bar peeling.

These report discusses the detailed design of Bar peeling attachments which includes Cutter head and bar feeding mechanism.

Keywords :- *Bar peeling, Lathe, cutter head, Bar feeding mechanism.*

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Nomenclature

d	Diameter of workpiece(mm)
a_p	Depth of cut(mm)
n	Spindle speed(rpm)
f	Feed(mm/rev)
V_C	Cutting velocity(m/min)
α	Rake angle
μ	Co-efficient of friction
ϕ	Shear angle
F_S	Shear Force
w	Width of the workpiece under cutting(mm)
t_1	Uncut thickness(mm)
R	Resultant force
F_C	Cutting force
F_T	Thrust force
F	Friction force
N	Normal load
a'	Working centre distance
b	Face width
m	Module
m_n	Normal module
m_t	Transverse module
p_f	Probability of failure
u	Ratio of gearing
x	Profile correction factor
F_t	Tangential load at the reference circle
HB	Brinell hardness of the material
K_A	Application factor
$K_{H\alpha}$	Transverse load distribution factor for contact stress
$K_{H\beta}$	Longitudinal load distribution factor for contact stress
$K_{F\alpha}$	Transverse load distribution factor for bending stress
$K_{F\beta}$	Longitudinal load distribution factor for bending stress
K_R	Reliability factor
K_V	Dynamic load factor
$K_{V\alpha}$	Factor for calculating K_V
$K_{V\beta}$	Factor for calculating K_V
L_A	Auxiliary value
L_H	Life in hours
L_N	Life in cycles
P	Transmitted power
S_H	Factor of safety for contact stress
S_B	Factor of safety for bending stress

Y_n	Notch sensitivity factor
Y_N	Life factor for bending stress
Y_K	Stress concentration factor
Y_S	Size factor for bending stress
Y_R	Factor of relative surface roughness
Z_E	Elasticity factor for contact stress
Z_H	Zone factor for contact stress
Z_R	Roughness factor for contact stress
Z_S	Size factor for contact stress
Z_V	Velocity factor for contact stress
Z_W	Work hardening factor for contact stress
Z_β	Helix angle factor for contact stress
α	Pressure angle
α_a	Tip pressure angle
α_{an}	Normal pressure angle at the tooth tip
α_{pro}	Protuberance angle
α_t	Transverse pressure angle
α_{ta}	Transverse pressure angle at the tooth tip
α_{wt}	Transverse working pressure angle
β	Helix angle
β_a	Helix angle at the tip circle
β_b	Base helix angle
τ	Auxiliary angle
ε_α	Transverse contact ratio
$\varepsilon_{\alpha n}$	Normal contact ratio
ε_β	Overlap ratio
υ_{50}	Viscocity of the lubricant at $50°C$
υ_{40}	Viscocity of the lubricant at $40°C$
ϕ	Auxiliary angle
μ	Poisson's ratio
σ_F	Maximum working tensile stress
σ_{FO}	Maximum nominal static stress calculated at the tooth root
σ_{FP}	Permissible stress for the material at the tooth root
σ_{HO}	Calculated contact stress
σ_H	Working contact stress
σ_{HP}	Permissible contact stress
σ_{Hlim}	Endurance limit for contact stress
σ_{FE}	Nominal endurance limit of an un-notched specimen at full elasticity of the material

Chapter 1

Introduction

This chapter includes objective of project and basic introduction, advantages, application of bar peeling machine.

1.1 Preliminary Remarks

Bar peeling is kind of machining process. In which, surface cracks, chilled skin and oxides are removed from the forged bar. Due to the bar peeling operation, good surface quality, dimensional tolerances and roundness are achieved. Bar peeling is suitable for the machining of different materials. For example: high speed steel, tempered steel, high alloyed steels, stainless steels, and even for titanium. As a results of bar peeling operation, the semi-finished bars are mainly delivered to the industries. Later the bars may be machined as per the different types of application. For example, axles for wind turbines, axle components for the automobile sector, extrusion blanks for tube manufacturing, and for hydraulic cylinders.

1.2 Construction of Bar Peeling Machine

The main Parts of bar peeling machine are :

- Head stock

- Feed Gear box
- Cutter head
- Feed Rollers
- Coolant system

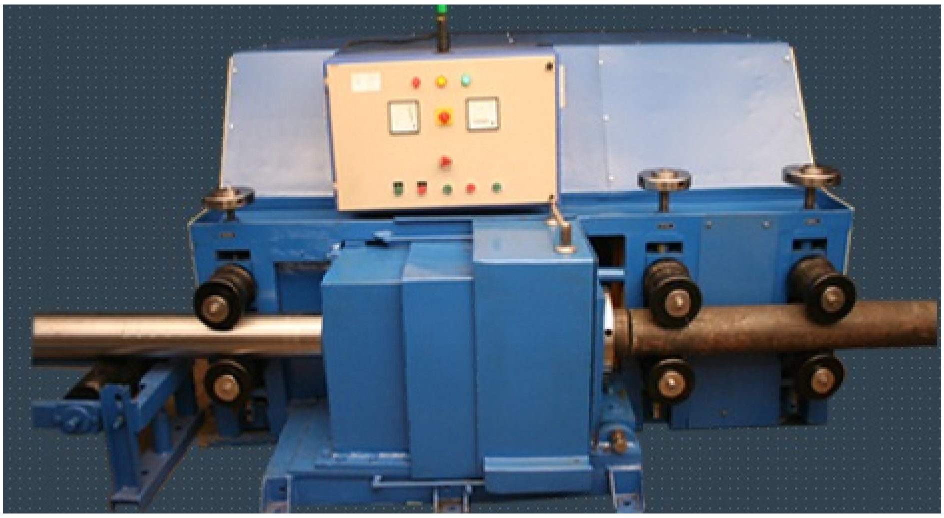

Figure 1.1: Bar Peeling Machine[1]

1. Head Stock :- The Head stock made of rigid casting containing a revolving main spindle driven through the main motor. The cutter head is fixed on the main spindle. It is designed to hold a cutting tools. Number of cutting tools are used in the cutter head as per requirement and space available on it. Different spindle speeds can be achieved through changed gears.

2. Feed Gear Box :- The feed gear box is also made of rigid casting. The rollers are operated by a motor through gearbox and coupling mechanism. The feed rollers feed the bar against the cutter head. In the gearbox, all the shafts are mounted on bearings.

3. Cutter Head :- The cutter head is made of cast steel. The cutter head consists of number of cutting tool inserts. For example, single cartridge, tandem cartridge, triple cartridge. The entire assembly is bolted to a face plate. The bar is fed longitudinally towards the cutter head with the help of bar feeding mechanism.

4. Feed Rollers :- Axial feed of bar in to the cutter head is done with the help of rotation of rollers. The rotation of rollers feed the bar in to cutter head with specified rate. The rotational speed of rollers is variable to suit the size of the bars. The drive is affected from a separate motor.

5. Coolant System :- Different types of cooling oils is used as a coolant system. Generally, mineral based cutting oil is used in cutter head to cool the tool.

1.2.1 Working Principle of Bar Peeling Machine

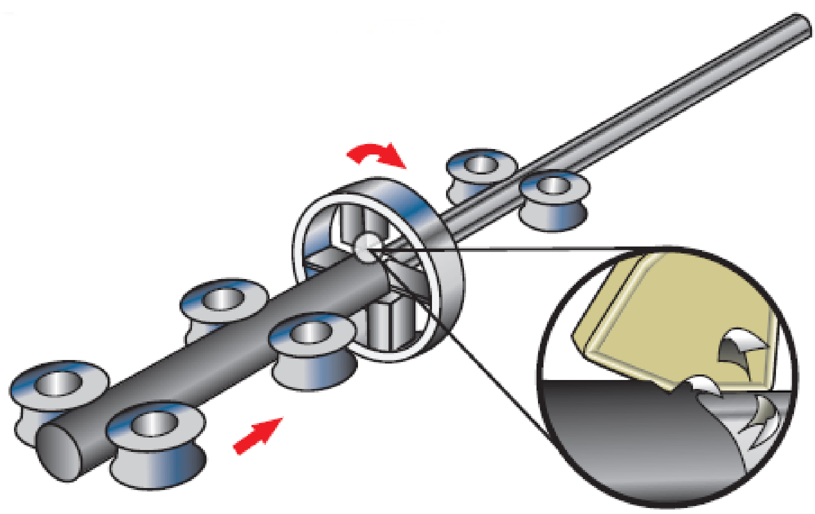

Figure 1.2: Working Principle of BPM[2]

Bar peeling is a machining process. In bar peeling machine, cutter head is mounted on main spindle. The bar is supported between an entry guide and steady rollers.

The tool holders with cutting inserts is mounted on cutter head. The feed rollers are rotate with the help of bar feeding mechanism. Bar feeding mechanism consist of feed gearbox and coupling assembly. The rotational speed of the rollers is variable to suit the size of the bars. In this peeling operation, the bar is fed longitudinally towards the revolving turning head. During the peeling operation, surface cracks, chilled skin and oxides are remove from bars and good surface quality, dimensional accuracy and roundness are achieved.

1.2.2 Advantages of Bar Peeling Machine

- Provides much faster machining:

  The cutter head carries more than one cutting tools holders. This provides a large number of cutting tools that increases the volume of metal that is machined from the bar surface. Therefore, high production is possible compared to conventional lathe machine.

- Hollow bars can be peeled:

  In lathe machine, end centers are required to support the bars. The bar peeling machine does not require bar end centers as is required in lathe machine and hollow bar can be peeled easily.

- Provides higher productivity:

  In lathe machine, bar is not rotating. while in bar peeling machine, bar does not rotated and instead the cutting tools are rotated. Therefore, higher rotational speed is achieved compared to lathe machine, which provides faster surface speed and increases peeling productivity.

- Yields are improved:

  In lathe machine, bars have center holes at bar ends for supporting between head stock and tail stock. This holes must be removed by cutting the ends. Also, long bars is exhibit a mismatch somewhere along the length that is created when bars are reversed end for end to complete the turning process and may require that the mismatch be cut out. This type of conditions never occur when turning on bar peeling machine and all of these conditions allow yields to be improved.

- Produces tighter tolerances and better surface finishes:
  In lathe turning process, bar is rotating and cutting tool is pushed against one side of the bar. In bar peeling machine, multiple cutting inserts on cutter head, which rotates around the bar and the cutting forces against the bar is balanced. In Bar peeling machine, the bar is continuously supported between guides as it enters and leaves the cutting tools. This type of design prevents the bars from deflected away from the cutting tool. Also, lathe turning may require two or more passes to achieve the require diameter tolerances that can be achieved in one pass in bar peeling machine. In bar peeling process, the bar and tool holder stability is excellent. Therefore, the surface finish, tolerances that can be achieved in one turning pass from a forged rolled bar much improved compared to lathe turning process.

1.2.3 Applications

Application areas vary, but semi finished bars are delivered to the industry and are used as an intermediate stage in the production of products which are to be processed further. Examples of these are axle components for the automobile industry, extrusion blanks for tube manufacturing, for hydraulic cylinders, etc.

1.3 Objective of Project

To design bar peeling attachment, consist of the cutter head and Bar feeding mechanism which offers high feed rates and small depth of cut. By peeling operations surface layers of oxides, rolled contaminants and cracks appeared at hot forging or rolling are removed.

Other objective of this attachments on lathe is to increase the Productivity. It provides low production costs. The surface quality and dimensional tolerances of the bar are high. Therefore, less machining required at later stages.

1.4 Outline of Thesis

The first chapter describes the objective of project and introduction. The second chapter on the literature survey deals with different patent of bar peeling machine and company catalogue related to bar peeling attachments. Design of cutter head is explained in third chapter. The fourth chapter describes the design of bar feeding mechanism. And finally conclusion and future scope of work has been discussed in seventh chapter.

Chapter 2

Literature Review

In this section, work done by various researchers for the bar peeling machine has been discussed in detail.

2.1 Review of Previous work

Anatol Michelson [3] Work was related to the metal cutting and specifically to an improved bar peeling device. The single metal removing cutter head creates large torsional forces on the bar between the work receiving unit or clamping unit and cutter head. This work was directed to an improved apparatus for turning long bars or workpieces with a minimum number of clamping units. Also, providing a reduction of torsional forces on the bar.

They provided an improved cutter head arrangement which removes a clamping forces required to prevent the workpiece from rotating. At large magnitudes of cutting depth, when single turning head is provided, the torsional forces exerted on the workpiece can be extremely high. Such high torsional forces require high clamping forces by the workpiece receiving unit, and such clamping forces requires heavy and expensive machine construction. Also there is possibility of workpiece deformation by the clamping jaws. In the present invention, two turning heads were provided, which eliminates the requirement of high clamping forces. This two turning heads contain multiple cutting tools and rotating in opposite directions to eliminate or atleast reduce the torsional stresses within the workpiece and reduce the clamping forces on the bar required by the workpiece receiving unit.

The present invention also provides a machine construction to reduce the length to diameter ratio between the cutting tools. The reduction of this ratio is achieved by providing a cutter arrangement where the first cutter head rotates in one direction and a second cutter head rotates in opposite direction with close spacing between the two cutter heads. This close spacing between the turning heads reduces the length to diameter ratio along the axial direction of the workpiece.

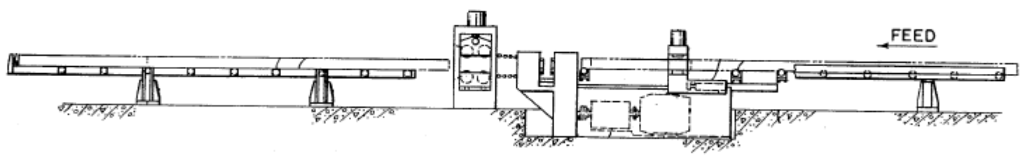

Figure 2.1: Bar Peeling Device[3]

Klaus Scholz[4] Invention was related to cutting inserts in the cutter head for the rough turning of rods and workpieces. The cutter is attached on a base plate has a slider arrangement in the radial direction. When a workpiece having an oval cross section, the cutting blade performs a back and forth movement in radial direction during each revolution of the blade. Therefore, the primary object of the present invention is to provide a cutter head in which the blade holders can easily follow the contour of the process workpiece. Another object of this invention is to eliminate frictional forces between the slider and the blade holder.

In keeping with these objects, to reduce the frictional forces between slider and blade holder, the spring and hinge arrangement is provided. The blade holder hinged at one end for slider. The blade is holding at free end of blade holder. The spring arrangement between the slider and the free end of the blade holder to push the blade against the workpiece. Also the connection line between the hinge joint of the blade holder and the cutting point of the blade forms a tangent line at the cutting point. Therefore, required spring force for compression of the blade holder is reduced and the wear of the tracing member is reduced.

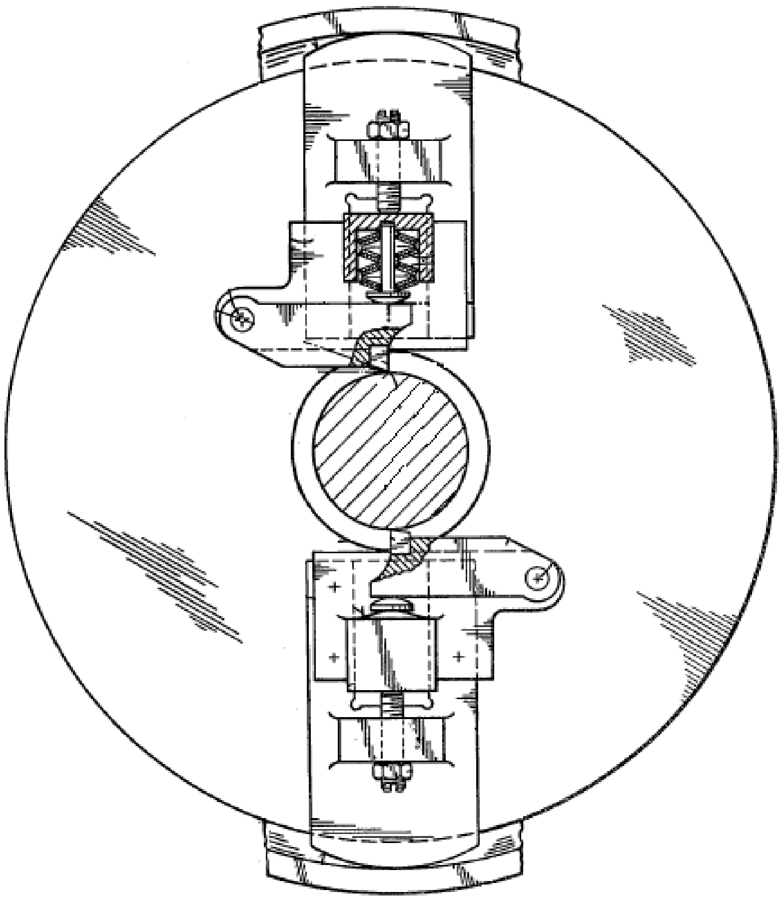

Figure 2.2: Cutter Head[4]

Yngve Dahllof [5] Work was related to a cutting toolholder for a bar peeling operation. In bar peeling machine, tool holders mounted on cutter head. Each tool holder equipped with one or several inserts of cemented carbide or wear resistant material.

Therefore, the purpose of this invention is to provide a toolholder that is equipped with means that will enable radial adjustment without having to provide separate fixture for the inserts and holders during the adjustment. it is a further purpose of this invention is to provide such adjustment, that can accomplish accurate fine adjustments without removing the toolholder from its location in the bar peeling machine.

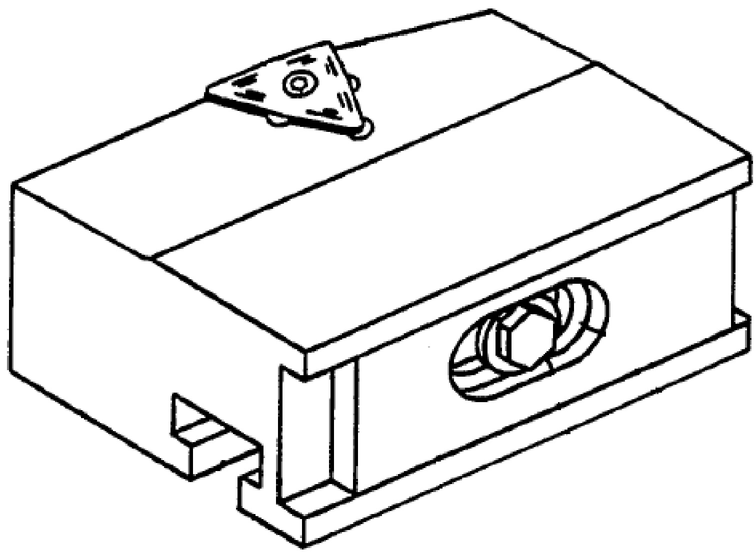

Figure 2.3: Toolholder with cutting insert[5]

Some company catalogue give the information about bar peeling machine.

Table 2.1: Details of Company Catalogue

Company name	catalogue name	Summary
Ceratizit[6]	Cutting Tools	Give the information about designation system for cutting inserts which is used for peeling operation and different parts of the cutter head.
ATI Stellram Allegheny Technologies[7]	Heavy Duty Machining	Give the technical information to calculate spindle speed, metal removal, power required for cutting operation.
Widia[8]	Widia	Bar peeling machine require a high level of Operation and high performance from the cutting inserts. Widia offers specially developed widia tools with indexable inserts for bar peeling, which are enable to meet these demands and reducing the manufacturing cost.
LMT Boehlerit[9]	Bar peeling	Give the information about detail view of cutter head and different types of cartridges.
Korloy[10]	Turning toolholder	It gives the information about different types of toolholders. It also gives the information about coding system of tool holder.
Lamina Technologies[11]	Magia (Turning cutting tools)	It gives the information about cutting insert designation (based on ISO norms). It also gives the details about machining condition for different types of cutting inserts.

Chapter 3

Design of bar Peeling Attachment

3.1 Description of bar peeling attachments on lathe machine

The layout of the bar peeling attachments on lathe machine is shown in fig. 3.1. The main parts of the lathe machine are: Head stock, Tail stock, chuck, etc. In this modified lathe machine, the head stock containing a revolving main spindle which is driven through the main motor. The cutter head is mounted in place of chuck. The cutter head consist of more than one tool holders. Each tool holder equipped with one or several cutting inserts of carbide material. Here, four tool holders are used in cutter head with cutting inserts. In this modified lathe machine the cutting tool are rotated. The bar feeding attachment is placed on lathe bed. In bar feeding mechanism, the rollers are rotated with the help of gearbox arrangement. In the gearbox, all the shafts are mounted on bearings. Here, the universal coupling is used for the connection between gearbox and rollers. The drive is affected from a separate geared motor. In this bar feeding mechanism, upper two rollers are rotate in clockwise and lower two rollers are rotate in anticlockwise direction. The distance between the two rollers is variable to suit the size of the bars. The cutter head will rotate with the main spindle while the bar is fed longitudinally towards the cutter head. Axial feed

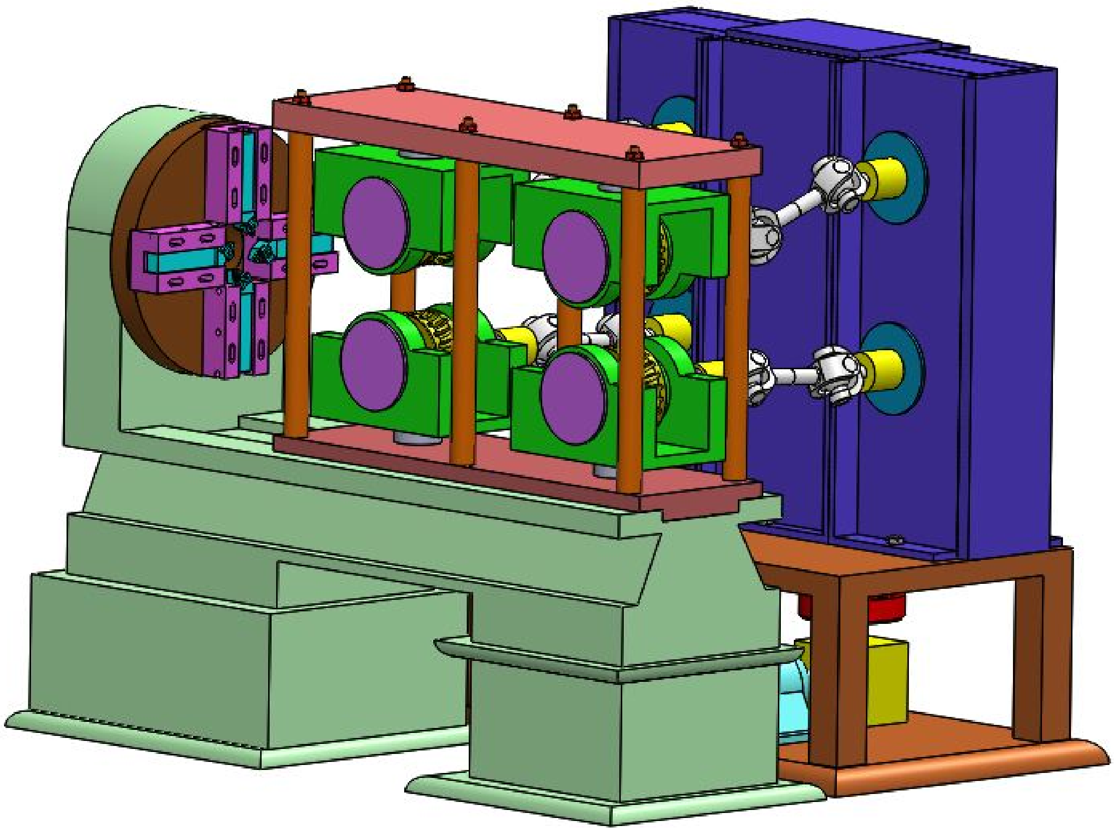

Figure 3.1: Layout of Bar Peeling Attachments on Lathe Machine

of the bar in to the cutter head is done with the help of feed rollers. For the cooling system, mineral based cutting oil is splattered on cutter head to cool the cutting tool. Now, the major components of the bar peeling attachment are:

- Cutter Head and
- Bar Feeding Mechanism

In this section, the design of cutter head has been discussed. The next chapter will discuss about design of bar feeding mechanism.

Specification for Bar Peeling Attachment:

The selection of cutting speed and feed is based on the different parameters are:

- Material of workpiece
- Material of tool
- Tool geometry
- Size of chip
- Types of finish desired
- Rigidity of the machine
- Types of coolant used

Depth of Cut = 2 mm

Diameter of Workpiece = 50 mm

From above two parameters,

The selected cutting speed and feed from design data book, [11]

Feed = 0.4 mm/rev and V_c = 250 m/min

$$\therefore \; V_c = \frac{\pi \times d \times n}{1000}$$

$$\therefore \; \text{n} = 1600 \text{ rpm}$$

Now, Feed per Minute $= f \times n$

$$= 640 \; \frac{mm}{min}$$

3.2 Design of Cutter Head

The function of cutter head is to hold the cutting tools. It is made of cast steel. It consists of number of tool holders as per requirement and space available on cutter head. Each tool holder equipped with one or several cutting inserts of carbide material. Here, four tool holders are used in cutter head with cutting inserts. In lathe machine, the chuck is removed and the cutter head is mounted in place of chuck with the help of faceplate. The block is used to guide the tool holder in cutter head. The block is mounted on cutter head with the help of bolts.

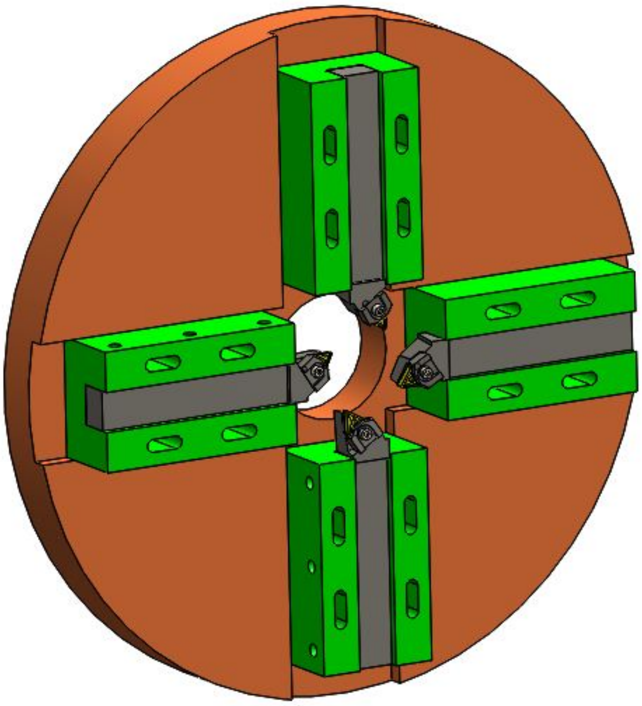

Figure 3.2: Cutter Head

3.2.1 Force Analysis

The measurement of cutting force is easily done with the help of a tool dynamometer. But here, the various components of forces have been obtained with the help of merchant circle diagram. The relationship among the different components and the resulting cutting force can be understood with the help of a diagram as shown in fig. 3.3. In this fig. the resultant cutting force R is resolved in to the normal force N and friction force F. The angle between N and F is thus the friction angle. The force R is also resolved in to normal force F_N and shear force F_S. The force R is also resolved in to thrust force F_T and cutting force F_C. In figure the resultants of F_N, F_S and F_C,

F_T are the same and those of N and F are the same in magnitude.

Now, Rake angle, $\alpha = 5°$

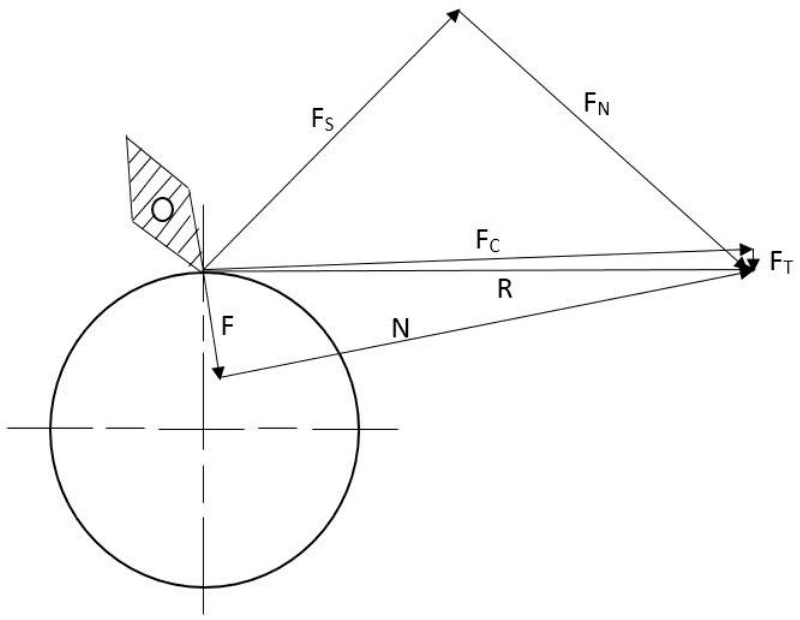

Figure 3.3: Force Diagram[12]

Co-efficient of friction, $\mu = 0.2$ [13]

where μ is the average coefficient of friction between tool and chip. The coefficient of friction can be expressed as,

$$(1)\ \mu = \tan\lambda$$

$$\therefore\ \lambda = 11.30°$$

The determination of the cutting force is depend on the calculation of shearing force F_S. To calculate this force, Ernst and Merchant suggested the following equation.

$$(2)\ 2\phi + \lambda - \alpha = 90°$$

$$\therefore\ \phi = 41.85°$$

If τ_S is the ultimate shear stress of the work material, then the shear force F_S along the shear plane can be written as,

$$\text{(3) Shear Force, } F_S = \frac{w \times t_1 \times \tau_S}{\sin\phi}$$
$$= 3884.99 \text{ N}$$

From fig.3.3, the resultant force can be calculated as,

$$\text{(4) Resultant Force, R} = \frac{F_s}{\cos(\phi + \lambda - \alpha)}$$
$$= 5822.97 \text{ N}$$

Thus, the other component can also be calculated using relation from fig.3.3,

(5) Cutting Force, $F_c = R \times \cos(\lambda - \alpha)$

$= 5787.80$ N

(6) Thrust Force, $F_T = R \times \sin(\lambda - \alpha)$

$= 638.97$ N

(7) Normal Force, $F_N = F_c \sin\phi + F_T \cos\phi$

$= 4338.15$ N

(8) Friction Force, F $= F_c \sin\alpha + F_T \cos\alpha$

$= 1141.06$ N

(9) Normal Load, N $= F_c \cos\alpha$ - $F_T \sin\alpha$

$= 5711.08$ N

3.2.2 Design of Bolts

A block is prepared to hold the toolholder. The arrangement of block on cutterhead is as shown in fig.,3.4.

As shown in fig.3.5, the eccentricity of the external force R is e from the center of gravity. This eccentric force can be considered as corresponding to an imaginary force R at the centre of gravity and a moment $(R \times e)$ about the same point.

From Fig.3.4,

r $= \sqrt{25^2 + 26.3^2} = 36.28$ mm

$\therefore\ r_1 = r_2 = r_3 = r_4 =$ r $= 36.28$ mm

The imaginary force R at the centre of gravity results in primary shear forces

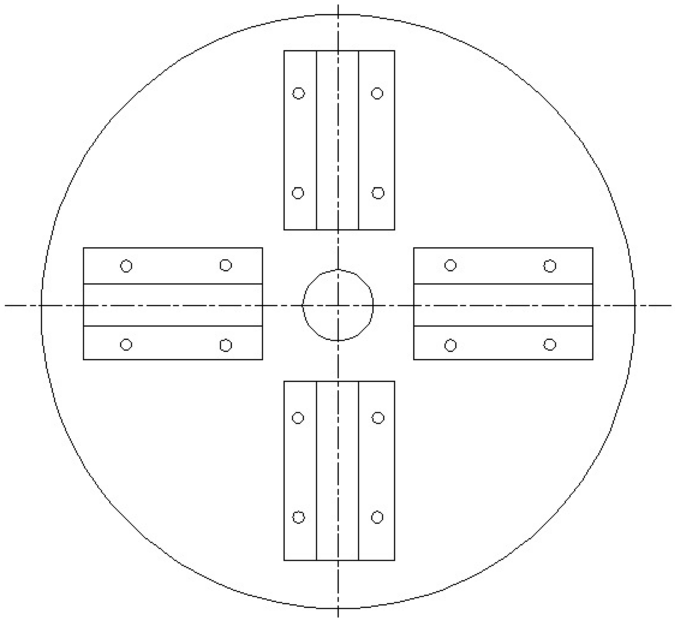

Figure 3.4: Arrangement of block on cutterhead

P'_1,P'_2,....,etc., is given by the following equation.

$$P'_1 = P'_2 = P'_3 = P'_4 = \frac{R}{Number of Bolts}$$

$$= \frac{5.822}{4}$$

$$= 1.455 \text{ kN}$$

The moment $(R \times e)$ about the centre of gravity results in secondary shear forces P''_1,P''_2,....,etc. If r_1, r_2,...,etc., are the radial distances of the bolt centres from the centre of gravity, then the equation written as,

$$(R \times e) = P''_1 \times r_1 + P''_2 \times r_2 + P''_3 \times r_3 + P''_4 \times r_4$$

It is assumed that the secondary shear force at any bolt is proportional to its distance from the centre of gravity. Therefore,

$$P''_1 = c \times r_1$$

$$P''_2 = c \times r_2$$

$$P''_3 = c \times r_3$$

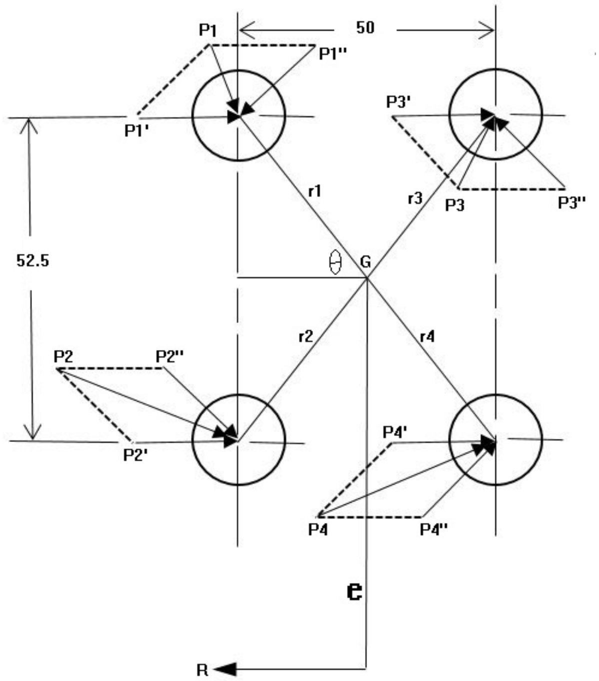

Figure 3.5: Free Body Diagram[14]

$$P_4'' = c \times r_4$$

$\therefore$ From above two equations,

$$\text{c} = \frac{R \times e}{r_1^2 + r_2^2 + r_3^2 + r_4^2}$$

The calculated value of c is 0.1227.

Therefore, $P_1'' = c \times r_1$

$$= 0.1227 \times 36.28$$

$$= 4.4538 \text{ kN}$$

From fig.3.4,

$$\tan\theta = \frac{25}{26.3}$$

$$\theta = 46.39$$

Now, Resultant shear force

$$P_1^2 = (P_1'' cos\theta - P_1')^2 + (P_1'' sin\theta)^2$$

$P_1 = 3.9257$ kN

$P_2^2 = (P_2''cos\theta)^2 + (P_2''sin\theta + P_2')^2$

$P_2 = 4.8546$ kN

$P_3^2 = (P_3''cos\theta - P_3')^2 + (P_3''sin\theta)^2$

$P_3 = 3.9257$ kN

$P_4^2 = (P_4''cos\theta)^2 + (P_4''sin\theta + P_4')^2$

$P_4 = 4.8546$ kN

From fig.3.4, Bolts 2 and 4 are subjected to maximum shear force,

$$\tau = \frac{P_2}{A}$$

$$\frac{S_sy}{f_s} = \frac{P_2}{A}$$

Therefore,

$$\frac{0.5 \times S_{yt}}{1.5} = \frac{4854.6}{\frac{\pi}{4} \times d_c^2}$$

$$\therefore\ d_c = 6.80 \text{ mm}$$

Now,

$d_c = 0.8 \times d$

d = 8.50 mm

From IS standard M10 is selected.

3.3 Power calculation required for cutting head

The following formula is used to calculate power for bar peeling operation. [7]
To calculate power, required for the peeling operation is:

$$P_W = \frac{V_c \times a_p \times f_r \times k_c}{33000} \times \left[\frac{0.016}{f_r}\right]^{0.29}$$

Where,

P_W = power required at the cutting edge

V_C = cutting speed in ft/min

a_p = max (total) radial cutting depth in inch

f_r = feed in inch/ revolution

k_c = K factor for work piece material (given in table[8])

H_p = Resultant Power (Horsepower)

KW = Kilo watts

So we get the power, $P_W = 6$ kw

3.4 Selection of Cutting Tool Insert

Individual tooling systems are in use on bar peeling machine. To avoid continuous adjustment and reduce cost, special inserts are available.
The require condition for bar peeling operation is : depth of cut = 2.0 mm, feed = 0.4 mm/rev and cutting speed = 200 m/min.
For this cutting condition the selected insert is CNMG 120408.
Where,

C = Insert shape

N = Clearance angle

M = Tolerance

G = Fixing and chip breaker types

12 = Cutting edge length

04 = Insert thickness

08 = Insert corner radius.

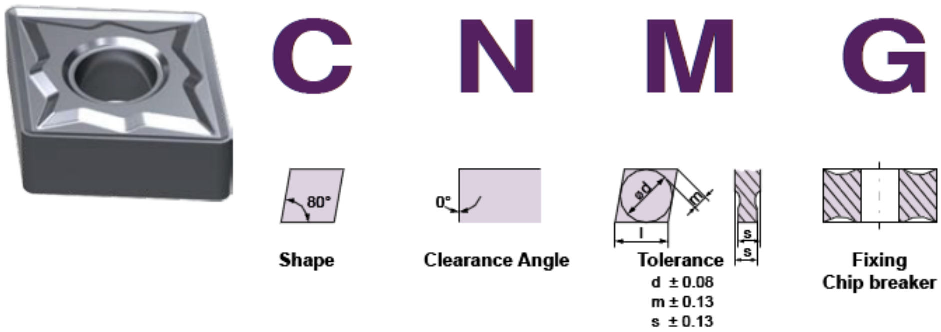

Figure 3.6: Detail of cutting insert[10]

3.5 Selection of Tool holder

There are different types of toolholder available for the mentioned inserts. Some of them are lever lock system, Wedge clamp system, Multi lock system, Screw on system, clamp on system.

For the above cutting insert we select a PCLNR/L 2525M12 tool holder.

Where, P = Clamping method of insert

C = Insert shape

L = Holder style

N = Clearance angle of insert

R/L = Hand of tool

25 = Height of shank

25 = Width of shank

M = Length of holder

12 = Length of cutting edge

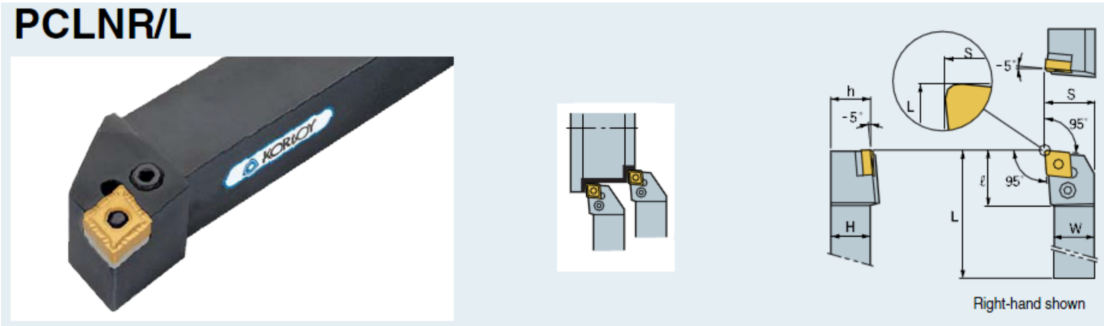

Figure 3.7: Detail of tool holder[10]

Chapter 4

Design of Bar Feeding Mechanism

4.1 Description of Bar Feeding Mechanism

In this section, the selection of roller, design of Gearbox and selection of coupling has been discussed. The layout of the bar feeding is shown in figure 4.1.

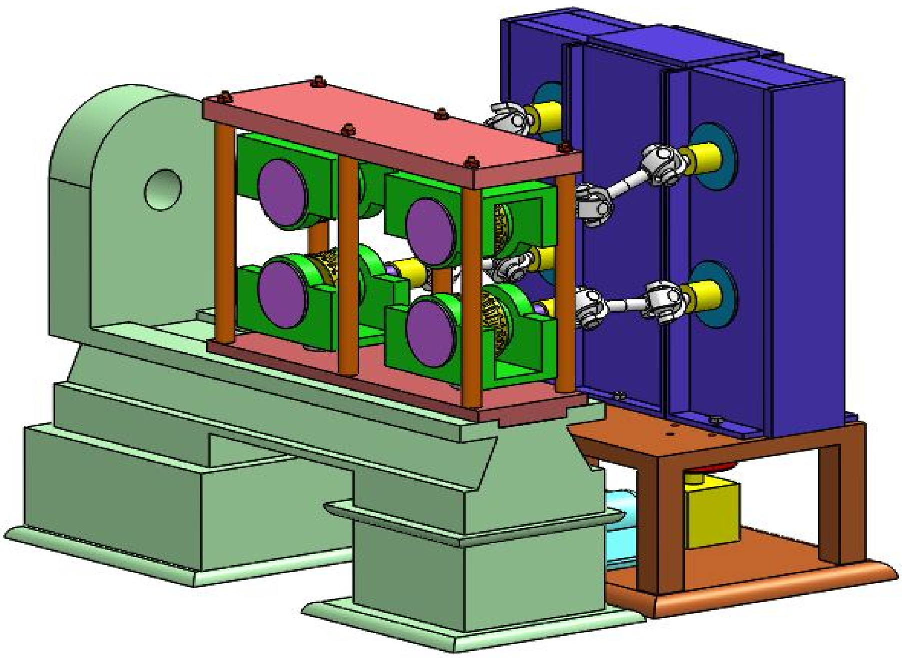

Figure 4.1: Layout of Bar Feeding Mechanism

In lathe machine, the distance between lathe bed and center of chuck is 227 mm. Therefore, for the bar feeding, the selected roller size is 60 mm in radius.

Now, selected feed is 0.4 mm/rev, cutting velocity is 250 m/min and spindle speed is 1600 rpm.

$$\therefore \text{ feed per minute} = 640 \text{ mm/min}$$

Now, $\omega = \frac{v}{r}$

where, $\omega = \frac{2\pi N}{60}$

r = radius of the roller

So. the rotation of the roller is 2 rpm.

4.2 Power requirement for Bar feeding mechanism

The power requirement for a bar feeding can be obtained from the feed force (Q), rate of feed (Vs) and friction force. Now, feed force can be determined by the following equation[15],

$$Q = kP_x + f'(P_z + G) \tag{4.1}$$

where, Q = Feed force (N),

k = Factor taking in to account the influence of the overturning moment,

P_x and P_z = component of the cutting force,

f' = coefficient of friction,

G = Weight of the parts,

Now, component of the cutting force (P_z) is given by,

$$P_z = \frac{6120N}{v}$$

where, N = power at spindle,

v = cutting speed,

So, we get $P_z = 298.577$ kg.f

Similarly, Component of the cutting force P_x is calculated as,

$$\frac{P_x}{P_z} = 0.7$$
$$\therefore \quad P_x = 209.004 \text{ kg.f}$$

Now, coefficient of friction between workpiece material and roller material is 0.8 and weight of the bar is 31 kg. Therefore from equation (4.1) calculated feed force Q is 4944.39 N. So the power requirement for bar feeding mechanism is calculated as,

$$power = (Q + f.f.)v = 55.34 \quad watt$$

Now, different types of geared motors are available. For example: Helical geared motors, Bevel helical geared motors, Parallel shaft geared motors, Worm geared motors, Helical worm geared motors, etc.

Here, selection of the geared motor is based on the power requirement for bar feeding. For that considering inertia of gears, rollers and other components of bar feeding mechanism, helical geared motor has been designed for 1.1 kW. From the catalogue of PBL motors, the motor selected is helical geared motor with output speed of 147 rpm.

Power build geared motors are designated as follows:

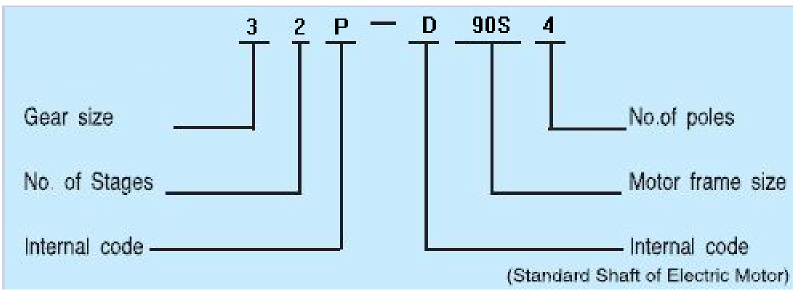

Figure 4.2: Designation of geared motor[16]

4.3 Design of Gearbox

The Cross section of the gearbox is shown in the figure 4.3.

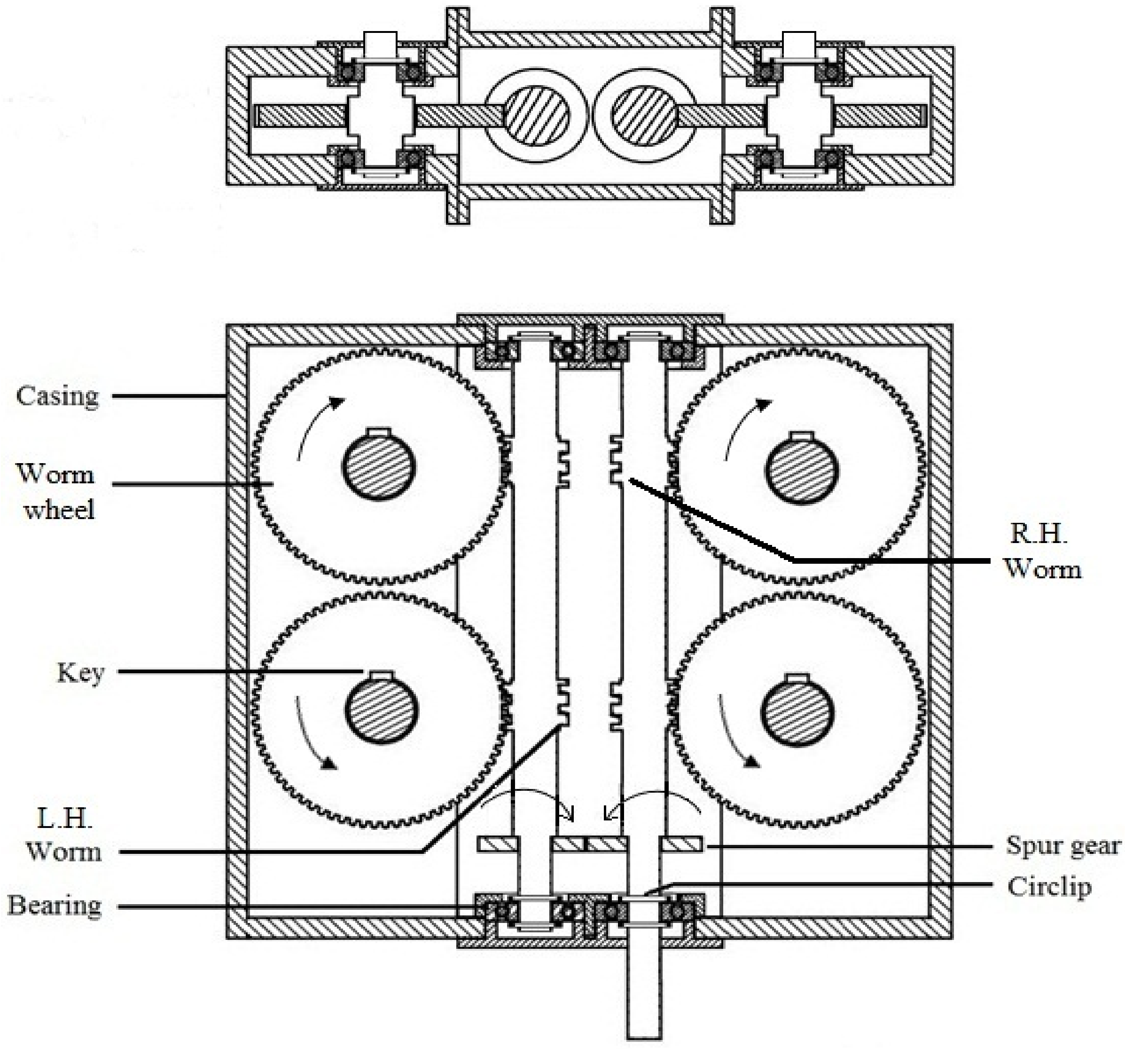

Figure 4.3: Cross Section of Gearbox

The output speed of the selected motor is 147 rpm. For the bar feeding mechanism, the required speed is 2 rpm. Therefore, high speed reduction is required. For that worm gear is selected. For the bar feeding mechanism, it is necessary to rotate rollers in opposite direction. Therefore, in the gearbox, spur gear is used to change the direction of worm wheel. In gearbox, all the shafts are supported on bearings.

4.4 Design of spur gear

4.4.1 Based on Pitting:

The load factors are obtained by calculating the factor of safety and contact stress. Module is assumed for the spur gear. Supplementary data is obtained from the tables and figures[17] which is used for calculation of load factors.

The selected material for gear is case hardened steel and hardness after heat treatment shall be 627 HB and shall have an accuracy grade of 5. The lubricant used has ISO viscosity grade of ISO VG 32 having a kinematic viscosity of 20 mm^2/s.

Basic Data:

Module = 5 (assumed)

Number of teeth = 18

Normal pressure angle = 20°

Transmitted power = 1.1 kW

Speed of gear = 147 rpm

Pitch circle diameter = 90 mm

Outside diameter = 100 mm

Working centre distance = 90 mm

Face width = 40 mm

Supplementary Data:

Ratio of gearing = 1

Modulus of elasticity = 206000 Mpa

Poisson's ratio = 0.3

Viscosity of the lubricant at 50° = 20.0 mm^2/s

Endurance limit for contact stress = 1450 Mpa

Formula[17]:

Linear speed at reference circle, $v = \dfrac{\pi d_1 n_1}{60000}$

$$= 0.6923 \quad m/s$$

Tangential load at the reference circle, $F_t = \dfrac{P}{v} = 1588.74 \quad mm$

Transverse pressure angle, $\tan\alpha_t = \dfrac{\tan\alpha_n}{\cos\beta}$

$$\therefore \quad \alpha_t = 20^\circ$$

Transverse working pressure angle, $\cos\alpha_{\omega t} = \dfrac{m_t(z_1+z_2)\cos\alpha_t}{2a}$

$$\therefore \quad \alpha_{\omega t} = 20^\circ$$

Base circle diameter, $d_b = z_1 m_n \dfrac{\cos\alpha_n}{\cos\beta_b} = 84.57 \quad mm$

Tip pressure angle, $\cos\alpha_a = \dfrac{d_b}{d_a}$

$$\therefore \quad \alpha_a = 32.25^\circ$$

Transverse contact ratio, $\varepsilon_\alpha = \dfrac{z_1(\tan\alpha_a - \tan\alpha_{\omega t})}{2\pi} = 0.7652$

$$\therefore \quad \varepsilon_\alpha = \varepsilon_{\alpha 1} + \varepsilon_{\alpha 2} = 1.53$$

Application factor, K_A : The application factor, K_A considered for a dynamic overloads. The over loads depend on characteristics of driving and driven machines,the couplings and operating conditions. If possible, the application factor should be obtained by accurate measurement or by an analysis. From the table (1), the selected value for the application factor is 1.25.

Auxiliary constant, $A = \dfrac{z_1 v}{100}\sqrt{\dfrac{u^2}{1+u^2}} = 0.088$

Factor $K_{V\alpha}$ and $K_{V\beta}$: From figure (1) and (2), the selected values are $K_{V\alpha} = 1.0$ and $K_{V\beta} = 1.0$

Dynamic load factor, K_V : The dynamic load factor, K_V is defined as the ratio between the maximum force which occurs at the mesh of gear pair and the equivalent load due to the externally applied load. The dynamic load factor is mainly influenced by: Transmission errors, mesh stiffness, Polar moments of inertia of gear, Lubrication, Transmitted load including application factor and stiffness of shaft and bearing.

For spur gears, $K_V = K_{V\alpha} = 1.0$

Longitudinal load distribution factor, $K_{H\beta}$: The longitudinal load distribution factor, $K_{H\beta}$ is to considered for the non-uniform distribution of load around the face width. This depends on the mesh alignment error of the gear pair. From the table (4), the selected value for the $K_{H\beta}$ is 1.27.

Transverse load distribution factor $K_{H\alpha}$: The distribution of the tangential load on several pairs of teeth in contact depends on the dimensions of gears, the accuracy and the value of the tangential force. Transverse load distribution factor considered for the distribution of actual load during the gear mesh. From the table (5), the selected value for the $K_{H\alpha}$ is 1.0.

Zone factor for contact stress, Z_H: The zone factor, Z_H considered for the effect on the Hertzian pressure, of tooth flank curvature at pitch point and converts the tangential force in to the normal force. Z_H can be calculated from the following formula:

$$Z_H = \frac{\cos\beta_b \cos\alpha_{\omega t}}{\cos^2\alpha_t \sin\alpha_{\omega t}} = 2.49$$

Elasticity factor for contact stress, Z_E: The elasticity factor considered for the effect of the modulus of elasticity, material properties, and poisson's ratio on the Hertzian pressure. Z_E can be calculated from the following formula:

$$Z_E = \frac{1}{\pi\left[\frac{1-\mu^2{}_1}{E_1} - \frac{1-\mu^2{}_2}{E_2}\right]} = 189.85$$

Contact ratio factor for contact stress, Z_ε: The contact ratio factor, Z_ε considered for the effect of the overlap ratio and the transverse contact ratio on the specific surface load of gears. Z_ε can be calculated from the following formula:

$$Z_\varepsilon = \sqrt{\frac{4-\varepsilon_\alpha}{3}} = 0.9073$$

$$\text{Factor, } C_{ZI} = \frac{\sigma_{HIM} - 850}{350} * 0.08 + 0.83 = 0.91$$

Lubricant Factor for Contact Stress,Z_I: The lubricant factor, Z_I considered for the effect of the different types of lubricant and its viscosity on the load bearing capacity of the surface. Z_I can be calculated by using the following formulae:

$$Z_I = C_{ZI} + \frac{4(1 - C_{ZI})}{\left(1.2 + \frac{80}{V_{50}}\right)^2} = 0.85$$

Life Factor for Contact Stress, Z_L: The life factor, Z_L considered a higher permissible Hertzian stress if only limited life is required. The main influencing factors are: Material and hardness and number of cycles. From table (7), the calculated value for the Z_L is 1.0

Factor $C_{ZV} = 0.85 + \dfrac{\sigma_{HIM} - 850}{350} * 0.08 = 0.93$

Velocity Factor for Contact Stress, Z_V: The velocity factor, Z_V considered for the effect of the pitch line velocity on the surface load capacity. Z_V can be calculated by using the following formula:

$$Z_V = C_{ZV} + \frac{2\,(1 - C_{ZV})}{\sqrt{0.8 + \frac{32}{v}}} = 0.89$$

Work Hardening Factor for Contact Stress, Z_W: Work hardening factor considered for the increase of surface durability due to meshing of gear The work hardening effect should be calculated from results of tests or field experiences with gears. In the absence of such data, the following formula can be used for the calculation of the approximate value of Z_W:

$$Z_W = 1.2 - \frac{HB - 130}{1700} = 0.91$$

Size Factor for Contact Stress, Z_S: The size factor, Z_S considered for the effect of tooth dimensions on allowable contact stress. Normally, the value of Z_S is unity. The values of size factor depending on module and material hardness condition. From table (9), the selected value for $Z_S = 1$.

Roughness Factor for Contact Stress, Z_R: The roughness factor, Z_R considered for the effect of surface texture of tooth flanks on surface load capacity. Depending on the surface condition of the gears, from table (10) the selected value of the $Z_R = 1$.

Reliability Factor, K_R: The reliability factor is introduced to permit the design of gears with a higher reliability. $K_R = 1.0$, corresponds to a probability of failure of one in 100 at the rated load and required life ($p_f = 0.01$). For some applications, when the requirement of reliability factor is greater or smaller, than K_R can be calculated using the following formula:

$$K_R = 0.79 - 0.105 \log (p_f) = 1.0$$
$$p_f = \text{probability of failure.}$$

Now, Contact stress, $\sigma_{HO} = Z_H Z_E Z_\varepsilon Z_\beta \sqrt{\dfrac{F_t (u+1)}{b d_1 u}} = 401.53 \quad MPa$

Working contact stress, $\sigma_H = \sigma_{HO} \sqrt{K_A K_V K_{H\beta} K_{H\alpha}} = 637.43 \quad MPa$

Permissible contact stress, $\sigma_{HP} = \dfrac{\sigma_{H\lim}}{\sqrt{K_R}} Z_L Z_I Z_R Z_S Z_V Z_W = 1045.30 \quad MPa$

$\therefore$ Factor of safety for contact stress, $S_H = \dfrac{\sigma_{HP}}{\sigma_H} = 1.64$

4.4.2 Based on Bending Strength

The selected material of gear case hardened steel and hardness after heat treatment shall be 627 HB and shall have an accuracy grade of 5. The lubricant used has ISO viscosity grade of ISO VG 32 having a kinematic viscosity of 20 mm^2/s.

Basic Data:

Module = 5 (assumed)

Number of teeth = 18

Normal pressure angle = 20°

Transmitted power = 1.1 kW

Speed of gear = 147 rpm

Pitch circle diameter = 90 mm

Outside diameter = 100 mm

Working centre distance = 90 mm

Face width = 40 mm

Probability of failure = 0.01

Profile correction factor = 0.5 mm

Mean roughness = 3.6 μm

Supplementary data:

Nominal endurance limit of an unnotched specimen at full elasticity of the material = 820 Mpa

Basic rack addendum of gear = 3.75 mm

Protuberance angle = 20°

Tip radius of the basic rack = 0.65 mm

Formula:

Linear speed at reference circle, $v = \dfrac{\pi d_1 n_1}{60000} = 0.6923 \;\; m/s$

Tangential load at the reference circle, $F_t = \dfrac{P}{v} = 1588.74 \;\; mm$

Velocity factor for contact stress, $Z_V = \dfrac{Z_1}{\cos^3 \beta} = 18$

$$inv \;\; \alpha_t = \tan \alpha_t - \alpha_t \frac{\pi}{180} = 0.014$$

Base circle diameter, $d_b = z_1 m_n \dfrac{\cos \alpha_n}{\cos \beta_b} = 84.57 \;\; mm$

Tip pressure angle, $\cos \alpha_a = \dfrac{d_b}{d_a}$

$$\therefore \quad \alpha_a = 36.86^\circ$$

$$inv \;\; \alpha_a = \tan \alpha_a - \alpha_a \frac{\pi}{180} = 0.108$$

Auxiliary angle, $\tau = \left[\dfrac{1}{Z_1} \left(\dfrac{\pi}{2} + 2x \tan \alpha_n \right) + inv \; \alpha_t - inv \; \alpha_a \right] \dfrac{180}{\pi} = 0.77$

Pressure angle at the tooth tip, $\alpha_{ta} = \alpha_a - \tau = 36.23^\circ$

Normal pressure angle at the tooth tip, $\tan\alpha_{an} = \tan\alpha_{ta}\cos\beta_a$

$$\therefore \quad \alpha_{an} = 36.23^\circ$$

Working pressure angle, $\cos\alpha_{\omega t} = \dfrac{m_t(z_1+z_2)\cos\alpha}{2a}$

$$\therefore \quad \alpha_{\omega t} = 20^\circ$$

Contact ratio, $\quad \varepsilon_\alpha = \dfrac{z_1(\tan\alpha_a - \tan\alpha_{\omega t})}{2\pi} = 1.11$

$$\therefore \quad \varepsilon_\alpha = \varepsilon_{\alpha 1} + \varepsilon_{\alpha 2} = 2.23$$

Overlap ratio, $\quad \varepsilon_\beta = \dfrac{b\sin\beta}{\pi m_n} = 0$

Application factor, K_A : The application factor, K_A considered for a dynamic overloads. The over loads depend on characteristics of driving and driven machines, the couplings and operating conditions. If possible, the application factor should be obtained by accurate measurement or by an analysis. From the table (1), the selected value for the application factor is 1.25.

Auxiliary constant, $A = \dfrac{z_1 v}{100}\sqrt{\dfrac{u^2}{1+u^2}} = 0.088$

Factor $K_{V\alpha}$ and $K_{V\beta}$: From figure (1) and (2), the selected values are $K_{V\alpha} = 1.0$ and $K_{V\beta} = 1.0$

Dynamic load factor, K_V : The dynamic load factor, K_V is defined as the ratio between the maximum force which occurs at the mesh of gear pair and the equivalent load due to the externally applied load. The dynamic load factor is mainly influenced by: Transmission errors, mesh stiffness, Polar moments of inertia of gear, Lubrication, Transmitted load including application factor and stiffness of shaft and bearing.
For spur gears, $K_V = K_{V\alpha} = 1.0$

Longitudinal load distribution factor for bending stress, $K_{F\beta}$: Longitudinal load distribution factor for bending stress, $K_{F\beta}$ is considered for the effects of uneven distribution of load along the length of the tooth. For gears in general mechanical use, the value of $K_{F\beta}$ is very close to $K_{H\beta}$. But, in case of surface hardened teeth, $K_{F\beta}$ is slightly less than $K_{H\beta}$. In general, same value can be taken for $K_{H\beta}$ and $K_{F\beta}$.

Transverse load distribution factor for bending stress, $K_{F\alpha}$: The transverse load distribution factor for bending stress, considered for the effect of transverse distribution of loads during the tooth meshing. For the gears of general mechanical use, the value of the factor $K_{F\alpha}$ can be taken as equal to $K_{H\alpha}$.

Life Factor for Bending Stress, Y_N: The life factor for bending stress, Y_N considered for the case of limited life $\left(3 \times 10^6 cycles\right)$ where a higher root stress can be allowed. The effect of main influencing factors on life factor are: Material and hardness, number of cycles (required life), and influence factors Y_n , Y_R , Y_S. The life factor Y_N can be taken as unity for a required life of more than 3×10^6 cycles.

Now, Auxiliary value,

$$EA = \frac{\pi}{4} m_n - h_{ao} \tan \alpha_n + h_k \left(\tan \alpha_n - \tan \alpha_{pro}\right) - \left(1 - \sin \alpha_{pro}\right) \frac{\rho_{ao}}{\cos \alpha_{pro}} = 0.9$$

$$G = \frac{\rho_{ao}}{m_n} - \frac{h_{ao}}{m_n} + x = -0.55$$

$$H = \frac{2}{Z_V} \left(\frac{\pi}{2} - \frac{EA}{m_n}\right) - \frac{\pi}{3} = -0.89$$

Auxiliary angle, ϕ: The auxiliary angle can be obtained by substituting an initial value of $\phi = \frac{\pi}{6}$. In case of two solutions, the smaller value of ϕ shall be taken. Auxiliary angle can be calculated from the following equation:

$$\phi = \frac{2G}{Z_V}\tan\phi - H = 0.8547\ rad$$

Now, $\dfrac{S_{Fn}}{m_n} = Z_V \sin\left(\dfrac{\pi}{3} - \phi\right) + \sqrt{3}\left(\dfrac{G}{\cos\phi} - \dfrac{\rho_{ao}}{m_n}\right) = 1.63$

$$\frac{h_{Fa}}{m_n} = \frac{1}{2}\left[\frac{Z_1}{\cos\beta}\left(\frac{\cos\alpha_t}{\cos\alpha_{ta}} - 1\right) + Z_V\left\{1 - \cos\left(\frac{\pi}{3} - \phi\right)\right\} - \frac{G}{\cos\phi} + \frac{\rho_{ao}}{m_n}\right] = 2.27$$

Form Factor for Bending Stress, Y_{Fa}: The tooth form factor, Y_{Fa} considered for the effect of the tooth form on the nominal bending stress. The calculation method is depend on the distance between the contact points of 30° tangents at the root fillet of the tooth profile. The value of the form factor, Y_{Fa} is calculated from the following equation:

$$Y_{Fa} = \frac{6\left(\dfrac{h_{Fa}}{m_n}\right)\cos\alpha_{an}}{\left(\dfrac{S_{Fn}}{m_n}\right)\cos\alpha_n} = 7.19$$

Now, $L_a = \dfrac{S_{Fn}}{h_{Fa}} = 1.1897$

$$\frac{\rho_F}{m_n} = \frac{\rho_{ao}}{m_n} + \frac{2G^2}{\cos\phi\,(Z_V\cos^2\phi - 2G)} = 0.30$$

$$q_n = \frac{S_{Fn}}{2\rho_F} = 2.68$$

Stress Concentration Factor for Bending Stress, Y_K: The stress concentration factor, Y_K is considered for the conversion of the nominal bending stress to the local tooth root stress. Thereby Y_K covers the bending stresses arising at the tooth root. The part of the local stress is independent of the bending moment arm. When the local stress is increases, the point of application of load approaches the critical tooth root section.

Therefore, in addition to its requirement on the notch radius, the stress correction is also dependent on the position on the load application, that is, the size of the bending moment arm. Y_K can be calculated by the following equation:

$$Y_K = (1.2 + 0.13L_a)\, q_n^{\left[\frac{1}{1.21 + \frac{2.3}{L_a}}\right]} = 1.85$$

Contact Ratio Factor for Bending Stress, Y_ε: The contact ratio factor Y_ε, is essential to calculate the tooth root stress σ_{FO}. With Y_ε the change of applied load at the tip, to the critical point of load application is carried out for estimated calculation of maximum tooth root stress arising during the meshing of the teeth. Y_ε can be calculated as,

$$Y_\varepsilon = 0.25 + \frac{0.75}{\varepsilon_\alpha} = 1.0$$

Factor of Notch Sensitivity for Bending Stress, Y_n: Due to notch sensitivity, actual reduction of resistance to fatigue is always less than the value of the stress concentration factor Y_K. The main effective parameters of notch sensitivity factor are the material, heat treatment and the rate of loading. The notch sensitivity factor Y_n can be determined from the following equation:

$$Y_n = 0.04Y_K + 0.93 = 1.01$$

Factor of Relative Surface Roughness for Bending Stress, Y_R: The surface roughness factor considered for the effect of profile roughness on the fatigue life of the gear. The relative surface roughness factor is defined as the ratio between the surface roughness factor of the gear to be determined and a test gear. From table (13), the calculated value for the $Y_R = 1.06$.

Size Factor for Bending Stress, Y_S: Y_S is a function of material and normal module. The size factor, Y_S considered for the reduction in the strength with increasing size. The main influencing factors are: Size of the gear, material and hardness. From table

(14), the calculated value for the $Y_S = 1$.

Reliability factor, $K_R = 0.79 - 0.105\log_{10}(p_f) = 1$

Maximum nominal static stress calculated at the tooth root,

$$\sigma_{FO} = \frac{F_t}{bm_n} Y_{Fa} Y_R Y_\varepsilon Y_\beta = 60.59 \ \ MPa$$

Maximum working tensile stress, $\sigma_F = \sigma_{FO} K_A K_V K_{F\beta} K_{F\alpha} = 96.83 \ \ MPa$

Permissible stress for the material at the tooth root,

$$\sigma_{FP} = \frac{\sigma_{FE}}{K_R} Y_N Y_n Y_R Y_S = 820 \ \ MPa$$

$\therefore$ Factor of Safety for Bending Stress, $S_B = \dfrac{\sigma_{FP}}{\sigma_F} = 8.45$

4.4.3 Dimensions of Spur Gear

General formula for calculation of dimensions of standard spur gear are shown in table:

4.4.4 Force Analysis of Spur Gear

In spur gears, the resultant force P_N can be resolved in to two components - tangential component P_t and radial component P_r at the pitch point as shown in figure 4.2. The tangential component P_t is important to calculate the magnitude of the torque and subsequently the power, which is transmitted. The radial component P_r is a separating force, which is always directed to the centre of gear. The torque transmitted

Table 4.1: Dimension of Spur Gear

Sr. No.	Description	Symbol	Equation	Value
1	Number of teeth	Z_1	Selected as per requirement	18
2	Module	m		5
3	Pressure angle	α	α	20°
4	Addendum	h_a	$h_a = 1m$	5.000
5	Dedendum	h_f	$h_f = 1.25m$	6.250
6	Tip clearance	c	$c = 0.25m$	1.250
7	Total tooth height	h	$h = 2.25m$	11.25
8	Pitch circle diameter	d	$d = mz$	90.00
9	Base circle diameter	d_b	$d_b = d\cos\alpha$	84.57
10	Outside diameter	d_a	$d_a = m\,(z+2)$	100
11	Root diameter	d_f	$d_f = m\,(z-2.5)$	77.50
12	Centre distance	a	$a = m(z_1+z_2)/2$	90

by the gears is given by,

$$M_t = \frac{60 \times 10^6\,(kW)}{2\pi n} = 71493.56 \ \ Nmm$$

Now, Pitch circle diameter, $d_p{}' = mZ_p = 90 \ \ mm$

The magnitudes of the tangential and radial component are calculated by using the following equations:

Tangential component, $P_t = \dfrac{2M_t}{d_p{}'} = 1588.74 \ \ N$

Radial component, $P_r = P_t \tan\alpha = 578.25 \ \ N$

4.5 Design of Worm gear

A pair of worm gears shall be designated by the hand of the thread of the worm, number of starts of the worm, number of teeth of the worm wheel, diametral quotient of the worm, module and centre distance of the gear pair.

$$(R/L)\ Z_1/Z_2/q/m - a$$

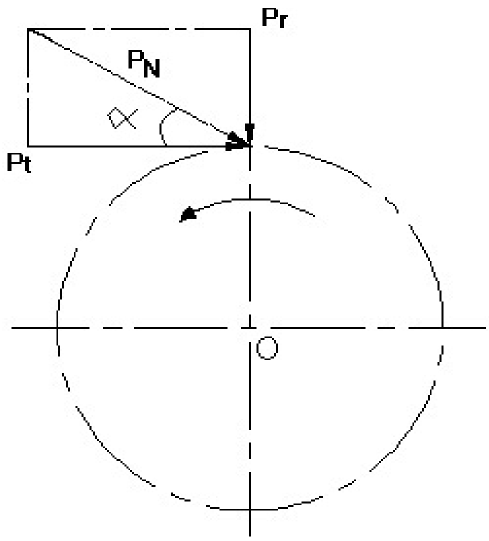

Figure 4.4: Components of Tooth Force

Where, (R/L) = Right or left hand thread

Z_1 = Number of starts on the worm

Z_2 = Number of teeth on the worm wheel

q = Diametral quotient

m = Module (mm)

a = Centre distance (mm)

The selection of materials for the worm gear is more limited than it is for other types of gears. The threads of the worm are subjected to fluctuating stresses and the number of stress cycles is large. Therefore, In the selection of the worm material, surface endurance strength is an important criterion. Therefore, the worm are made of case hardened steel with a surface hardness of HRC and case depth of 0.75 to 4.5 mm. Different types of steel are used for the worm are:

Normalized carbon steels - 40C8, 55C8

Case hardened carbon steels - 10C4, 14C6

Case hardened alloy steels - 16Ni80Cr60, 20Ni2Mo25

Nickel chromium steels - 13Ni3Cr80, 15Ni4Cr1

The worm wheel can not be perfectly generated in the hobbing process. The final shape and finish of the worm wheel teeth is the result of plastic deformation during the preliminary stages of service. Therefore, the soft and conformable material should be used for the worm wheel. For the worm wheel, Phosphor bronze is widely used with a surface hardness of 90-120 BHN. Phosphor bronze is costly and in case of large dimensions of the worm wheels, only the outer rim is made of phosphor bronze.

4.5.1 Specification for Worm Gears

The suggested transmission ratios, centre distances and equivalent values for number of starts on the worm, number of teeth of the worm wheel, diametral quotient of the worm and module are selected from the standard series. Now, the output speed of the selected geared motor is 147 rpm. For the required rotation of the rollers, the selected values for worm gears is 1/71/12/3 - 125[18].

Pressure angle, α: A pressure angle of 20° has been standardised for the general purpose application. From table, selected pressure angle is 20°.

Table 4.2: Pressure angle

Application	No. of starts	Pressure angle
Indexing mechanism	1 or 2	14.5°
Power transmission	1 or 2	20°
	3 or more	25°
Fine pitch (instrument)	1 to 10	20°

Lead angle, γ: Selection of lead angle depends on a number of factors. Efficiency of power transmission is obtained by lead angle γ and coefficient of friction. The efficiency increases with increasing values of γ and is maximum around $\gamma = 45°$. Thus, higher lead angles are preferred for power transmitting worm gearing. However, high lead angles produce manufacturing problems and therefore require modification in tooth profile parameters like increase in pressure angle, reduction in dedendum and working depth. Therefore it is recommended not to use a lead angle of more than

6° per start. Maximum lead angles are suitable for different pressure angles are as indicated below:

Table 4.3: Lead angle

Pressure angle	Max. lead angle
14.5°	15° approximately max.
20°	25° approximately max.
25°	35° approximately max.
30°	45° max

4.5.2 Power Transmitting Capacity of Worm Gear

A selected pair of worm and worm wheel is designated as, 1/71/12/3. The input speed of the worm is 147 rpm. For centrifugally cast phosphor bronze worm wheel and a nickel molybdenum case hardening steel worm[19]:

Surface stress factor, $S_{c1} = 53.1$

$S_{c2} = 10.5$

Bending stress factor, $S_{b1} = 325$

$S_{b2} = 69$

Zone factor, $Y_z = 1.202$

The maximum permissible torque that the worm wheel can withstand without failure, is given by the lower of the following four values:

$$(M_t)_1 = 17.65 \times X_{b1} \times S_{b1} \times m \times l_r \times d_2 \times cos\gamma$$

$$(M_t)_2 = 17.65 \times X_{b2} \times S_{b2} \times m \times l_r \times d_2 \times cos\gamma$$

$$(M_t)_3 = 18.64 \times X_{c1} \times S_{c1} \times Y_Z \times (d_2)^{1.8} \times m$$

$$(M_t)_4 = 18.64 \times X_{c2} \times S_{c2} \times Y_Z \times (d_2)^{1.8} \times m$$

Where,

$(M_t)_1$, $(M_t)_2$, $(M_t)_3$, $(M_t)_4$ = Permissible torque on the worm wheel (N-mm)

$$X_{b1},\, X_{b2} = \text{Speed factors for strength of worm and worm wheel}$$
$$S_{b1},\, S_{b2} = \text{Bending stress factors of worm and worm wheel}$$
$$X_{c1},\, X_{c2} = \text{Speed factors for the wear of worm and worm wheel}$$
$$S_{c1},\, S_{c2} = \text{Surface stress factors of the worm and worm wheel}$$
$$d_2 = \text{Pitch circle diameter of worm wheel (mm)}$$
$$l_r = \text{Length of the root of worm wheel teeth (mm)}$$
$$Y_z = \text{Zone factor}$$

Now, Face width of worm wheel, $F = 2m + \sqrt{(q+1)} = 21.63\ mm$

Clearance, $c = 0.2m\cos\gamma = 0.5979\ mm$

Outside diameter of the worm, $d_{a1} = m\,(q+2) = 42\ mm$

Length of the root of worm wheel teeth, $l_r = (d_{a1} + 2c)\sin^{-1}\left[\dfrac{F}{(d_{a1}+2c)}\right] = 22.66\ mm$

Rubbing speed, $v_s = \dfrac{\pi d_1 n_1}{60000\cos\gamma} = 0.2782\ m/s$

From figure, For $v_s = 0.2782\ m/s$ and $n_1 = 147\ rpm$, $X_{c1} = 0.32$

and $v_s = 0.2782\ m/s$ and $n_2 = 2\ rpm$, $X_{c2} = 0.57$

For $n_1 = 147\ rpm$, $X_{b1} = 0.38$

and $n_2 = 2\ rpm$, $X_{b2} = 0.64$

So, we get $(M_t)_1 = 38072261.29\ N.mm$

$(M_t)_2 = 11245960.26\ N.mm$

$(M_t)_3 = 17733702.40\ N.mm$

$(M_t)_4 = 6246245.813\ N.mm$

The lower value of the torque on the worm wheel is 6246245.813 N.mm.

$$\therefore\quad \text{Power transmitting capacity, } kW = \frac{2\pi n_2\,(M_t)}{60\times 10^6} = 1.37$$

4.5.3 Dimensions of Worm and Worm Wheel

General formula to determine the dimensions of standard worm gear are shown in table[20]:

Table 4.4: Dimensions of Worm and Worm Wheel

Sr. No	Description	Equation	Value
1	Transmission ratio	$i = (Z_2/Z_1)$	70
2	Output speed	$n_2 = n_1/i$	2.1
3	Lead angle	$\gamma = \tan^{-1}(Z_1/q)$	4.76°
4	Helix angle	$\Psi = (90 - \gamma)$	89.91°
5	Centre distance	$a = 0.5m\,(q + Z_2)$	125
6	Axial pitch	$p_x = \pi m$	9.24
7	Lead	$l = \pi m Z_1$	9.24
8	Addendum	$h_{a1} = m$	3
9	Dedendum	$h_{f1} = (2.2\cos\gamma - 1)\,m$	3.57
10	clearance	$c = 0.2m\cos\gamma$	0.59
11	Pitch circle diameter of worm	$d_1 = mq$	36
12	Outside diameter of worm	$d_{a1} = m\,(q + 2)$	42
13	Root diameter of worm	$d_{f1} = m\,(q + 2 - 4.4\cos\gamma)$	28.84
14	Length of worm	$l = \sqrt{{d_{a2}}^2 - {d_2}^2}$	50.73
15	Pitch circle diameter of worm wheel	$d_2 = mZ_2$	213
16	Throat diameter of worm wheel	$d_{a2} = m\,(Z_2 - 2 + 4\cos\gamma)$	218.95
17	Root diameter of worm wheel	$d_{f2} = m\,(Z_2 - 2 - 0.4\cos\gamma)$	205.80
18	Width of worm wheel	$b = 2m\sqrt{q + 1}$	21.63

4.5.4 Force Analysis of Worm Gear

The analysis of three components of the worm and worm wheel is based on the following assumptions:

(1) The worm is the driving element and the worm wheel is the driven element.

(2) The worm has right handed threads.

(3) The worm rotates in anticlockwise directions.

The components of the resultant force acting on the worm are:

$(P_1)_t$ = tangential component on the worm (N)

$(P_1)_a$ = axial component on the worm (N)

$(P_1)_r$ = radial component on the worm (N)

The component $(P_2)_t$, $(P_2)_a$, $(P_2)_r$ acting on the worm wheel are obtained in a similar way. The force acting on the worm wheel is equal and opposite reaction of the force acting on the worm.

Therefore, $(P_2)_t = (P_1)_a$

$(P_2)_a = (P_1)_t$

$(P_2)_r = (P_1)_r$

Now, 1.1 kW of power at 147 rpm is supplied to the worm shaft and Pressure angle is 20°. Lead angle is 4.76°.

Components of Tooth Force Acting on Worm:

$$\text{Torque, } M_t = \frac{60 \times 10^6\,(kW)}{2\pi n_1} = 71493.56 \;\; Nmm$$

$$\text{Tangential component on the worm, } (P_1)_t = \frac{2M_t}{d_1} = 3971.86 \;\; N$$

Now, rubbing velocity, $V_s = 0.2782$ m/s. Therefore from figure(20.9)[15], the selected value of coefficient of friction $\mu = 0.055$.

$$\text{Axial component on the worm, } (P_1)_a = (P_1)_t \times \frac{(\cos\alpha\cos\gamma - \mu\sin\gamma)}{(\cos\alpha\sin\gamma + \mu\cos\gamma)} = 23826.42 \;\; N$$

$$\text{Radial component on the worm, } (P_1)_r = (P_1)_t \times \frac{\sin\alpha}{(\cos\alpha\sin\gamma + \mu\cos\gamma)} = 7942.52 \;\; N$$

Components of Tooth Force Acting on Worm wheel:

The force components acting on the worm wheel are as follows:

$(P_2)_t = (P_1)_a = 23826.42$ N

$(P_2)_a = (P_1)_t = 3971.86$ N

$(P_2)_r = (P_1)_r = 7942.52$ N

4.5.5 Design of Shaft for Worm and Spur Gear

The layout of the shaft for worm and spur gear is shown in figure 4.5. Design of shaft has been carried out considering different forces of spur gear and worm that are acting on shaft. The pitch circle diameters of the spur gear and worm are 90 and 36 mm respectively. The selected material of the shaft is steel FeE 580 ($S_{ut} = 770 \ N/mm^2 and S_{yt} = 580 \ N/mm^2$). The values of shock and fatigue factors, k_b and k_t for rotating shafts are 1.5 and 1.0 respectively. Here, the gear is connected to the shaft by means of keys.

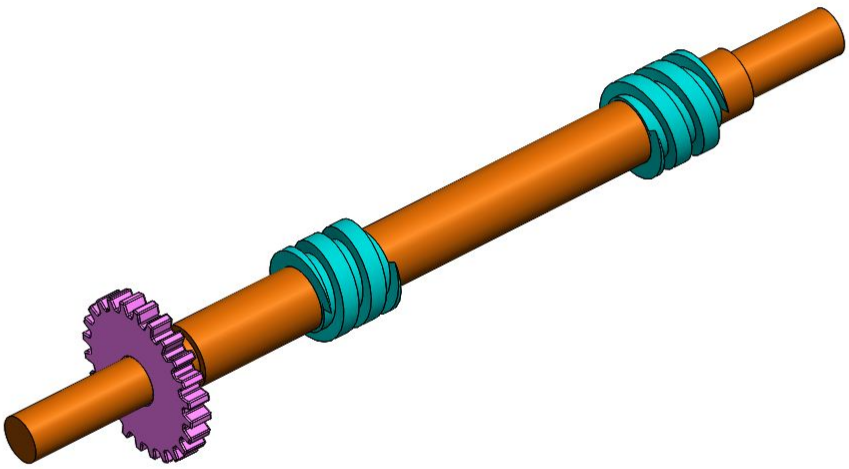

Figure 4.5: Shaft for Worm and Spur Gear

Step :1 Permissible Shear Stress

$0.30S_{yt} = 0.30 * (580) = 174$ N/mm^2

$0.18S_{ut} = 0.18 * (770) = 138.6$ N/mm^2

The lower of the two values is 138.6 N/mm^2 and the shaft with keyways.

Therefore $\tau_{max} = 0.75 * (138.6) = 103.95$ N/mm^2

Step :2 Torsional Moment

The torque transmitted by shaft is given by,

$$M_t = \frac{60 \times 10^6\,(kW)}{2\pi n} = 71493.56\ \mathrm{N}/mm^2$$

Step :3 Bending Moment

The bending moment diagram in the vertical plane is shown in fig 4.6. At point D, the bending moment is maximum, is given by

$$M_b = 1238786.98\ \text{N-mm}$$

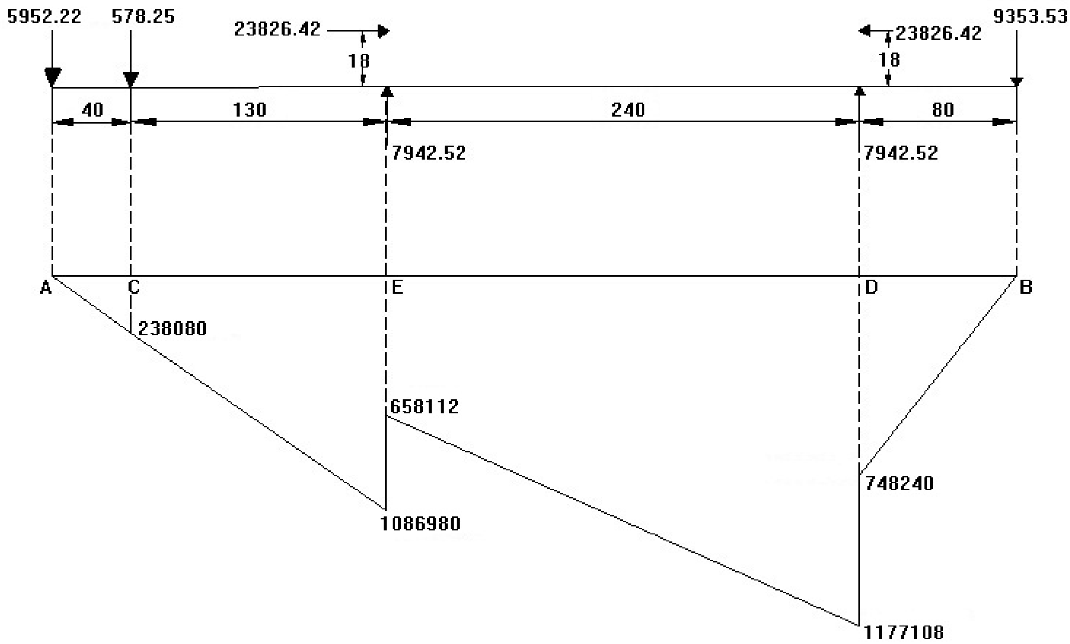

Figure 4.6: Forces and Bending Moment Diagram

Step :4 Shaft Diameter

$$d^3 = \frac{16}{\pi\tau_{max}} \times \sqrt{(k_b M_b)^2 + (k_t M_t)^2} = 60443.97mm$$

d = 39.245 mm

∴ the selected diameter of the shaft is 40 mm[21].

4.5.6 Selection of Bearing for Worm and Spur Gear Shaft

The deep grove ball bearing is selected for a shaft, that is 30 mm in diameter and which rotates at 147 rpm. The bearing is subjected to a radial load of 10538.42 N and axial load of 57.68 N. The expected life of the bearing is 10000 hours.

Now, Bearing life, $L_{10} = \dfrac{60nL_{10h}}{10^6} = 88.2 \quad million\ rev$

Now, Assume shaft diameter is 30 mm and static capacity of the bearing is 24000 N.

Therefore,

$$\left(\frac{F_a}{F_r}\right) = 0.0054 \quad and \quad \left(\frac{F_a}{C_0}\right) = 0.0024$$

From table, $e = 0.22 \quad and \quad \left(\dfrac{F_a}{F_r}\right) < e$

$\therefore$ the values of the factor X = 1 and Y = 0.

Now, Dynamic load capacity, $P = XF_r + YF_a = 10538.42\ N$

$$C = P(L_{10})^{\frac{1}{3}} = 40401.61\ N$$

For Deep Groove Ball Bearing, **SKF6406** is selected[22].

Designation : 6406

Dynamic Capacity C = 43600 N

Static Capacity C_0 = 24000 N

Inner Diameter = 30 mm

Outer Diameter = 90 mm

Calculated Dynamic Capacity = 40401.61 N.

Hence Bearing selected is safe.

4.5.7 Design of Key for Worm and Spur Gear Shaft

The selected material for the key is 50C4 ($S_{yt} = 460\ N/mm^2$) and the factor of safety is 3. From table, selected key size is $10 \times 8\ mm$.

Now, Permissible compressive and shear stresses,

$$\sigma_c = \frac{S_{yt}}{(fs)} = 153.33\ \ N/mm^2 \qquad \text{and} \qquad \tau = \frac{0.5S_{yt}}{(fs)} = 76.67\ \ N/mm^2$$

Now, $\sigma_c = \dfrac{4M_t}{dhl} = 79.43\ \ N/mm^2 \qquad \text{and} \qquad \tau = \dfrac{2M_t}{dbl} = 31.77\ \ N/mm^2$

Therefore, the selected key size is safe.

4.5.8 Design of Shaft for Worm Wheel

The pitch circle diameter of the worm wheel is 213 mm. The selected material of the shaft is steel FeE 580 ($S_{ut} = 770\ \ N/mm^2 and S_{yt} = 580\ \ N/mm^2$). The values of shock and fatigue factors, k_b and k_t for rotating shafts are 1.5 and 1.0 respectively. Here, the gear is connected to the shaft by means of keys.

Step :1 Permissible Shear Stress

$0.30S_{yt} = 0.30 * (580) = 174$ N/mm^2

$0.18S_{ut} = 0.18 * (770) = 138.6$ N/mm^2

The lower of the two values is 138.6 N/mm^2 and the shaft with keyways.

Therefore $\tau_{max} = 0.75 * (138.6) = 103.95$ N/mm^2

Step :2 Torsional Moment

The torque transmitted by shaft is given by,

$$M_t = \frac{60 \times 10^6\,(kW)}{2\pi n} = 5254777.07 \text{ N}/mm^2$$

Step :3 Bending Moment

The bending moment diagram in the vertical plane is shown in fig 4.7. At point C, the bending moment is maximum, is given by

$$M_b = 738058.36 \text{ N-mm}$$

Step :4 Shaft Diameter

$$d^3 = \frac{16}{\pi\tau_{max}} \times \sqrt{(k_bM_b)^2 + (k_tM_t)^2} = 263239.36mm$$

d = 64.08 mm

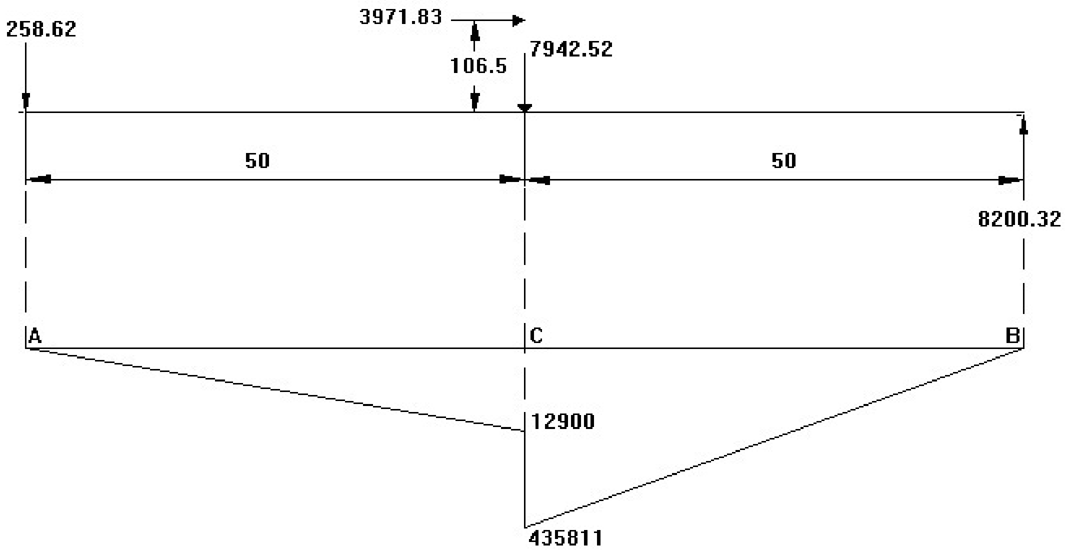

Figure 4.7: Forces and Bending Moment Diagram

$\therefore$ the selected diameter of the shaft is 65 mm[21].

4.5.9 Selection of Bearing for Worm Wheel Shaft

The deep grove ball bearing is selected for a shaft, that is 40 mm in diameter and which rotates at 2 rpm. The bearing is subjected to a radial load of 11913.42 N and axial load of 3971.83 N. The expected life of the bearing is 10000 hours.

Now, Bearing life, $L_{10} = \dfrac{60nL_{10h}}{10^6} = 1.2 \quad million\, rev$

Now, Assume shaft diameter is 40 mm and static capacity of the bearing is 9300 N. Therefore,

$$\left(\frac{F_a}{F_r}\right) = 0.3334 \quad and \quad \left(\frac{F_a}{C_0}\right) = 0.4269$$

From table, $e = 0.34 \quad and \quad \left(\dfrac{F_a}{F_r}\right) > e$

$\therefore$ the values of the factor X = 0.56 and Y = 1.05.

Now, Dynamic load capacity, $P = XF_r + YF_a = 10840.83\ N$

$$C = P(L_{10})^{\frac{1}{3}} = 11450.30\ N$$

For Deep Groove Ball Bearing, **SKF6008** is selected[22].

Designation : 6008

Dynamic Capacity C = 16800 N

Static Capacity C_0 = 9300 N

Inner Diameter = 40 mm

Outer Diameter = 68 mm

Calculated Dynamic Capacity = 11450.30 N.

Hence Bearing selected is safe.

4.5.10 Design of Shaft for Roller

The diameter of the roller is 120 mm. The selected material of the shaft is steel FeE 580 ($S_{ut} = 770\ N/mm^2 and S_{yt} = 580\ N/mm^2$). The values of shock and fatigue factors, k_b and k_t for rotating shafts are 1.5 and 1.0 respectively. Here, the gear is connected to the shaft by means of keys.

Step :1 Permissible Shear Stress

$0.30S_{yt}$ = 0.30 * (580) = 174 N/mm^2

$0.18S_{ut}$ = 0.18 * (770) = 138.6 N/mm^2

The lower of the two values is 138.6 N/mm^2 and the shaft with keyways.

Therefore τ_{max} = 0.75 * (138.6) = 103.95 N/mm^2

Step :2 Torsional Moment

The torque transmitted by shaft is given by,

$$M_t = \frac{60 \times 10^6\,(kW)}{2\pi n} = 5254777.07 \text{ N}/mm^2$$

Step :3 Bending Moment

The bending moment diagram in the vertical plane is shown in fig 4.8. At point C, the bending moment is maximum is given by

$$M_b = 440280 \text{ N-mm}$$

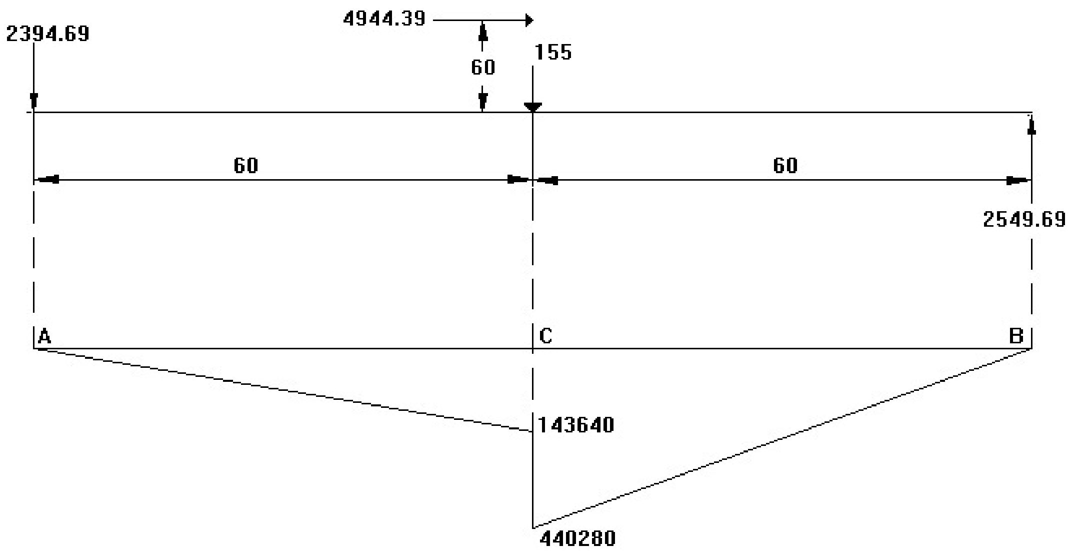

Figure 4.8: Forces And Bending Moment Diagram

Step :4 Shaft Diameter

$$d^3 = \frac{16}{\pi\tau_{max}} \times \sqrt{(k_b M_b)^2 + (k_t M_t)^2} = 259611.10mm$$

$$\text{d} = 63.79 \text{ mm}$$

$\therefore$ the selected diameter of the shaft is 65 mm[21].

4.5.11 Selection of Bearing for Roller Shaft

The deep grove ball bearing is selected, that is 40 mm in diameter and which rotates at 2 rpm. The bearing is subjected to a radial load of 2549.69 N and axial load of 4944.39 N. The expected life of the bearing is 10000 hours.

Now, Bearing life, $L_{10} = \dfrac{60nL_{10h}}{10^6} = 1.2 \;\; million\, rev$

Now, Assume shaft diameter is 40 mm and static capacity of the bearing is 9300 N.

Therefore,

$$\left(\frac{F_a}{F_r}\right) = 1.9392 \quad and \quad \left(\frac{F_a}{C_0}\right) = 0.6338$$

From table, $e = 0.44 \quad and \quad \left(\frac{F_a}{F_r}\right) > e$

$\therefore$ the values of the factor X = 0.56 and Y = 1.0.

Now, Dynamic load capacity, $P = XF_r + YF_a = 6371.44\ N$

$$C = P(L_{10})^{\frac{1}{3}} = 6729.64\ N$$

For Deep Groove Ball Bearing, **SKF16008** is selected[22].

Designation : 16008

Dynamic Capacity C = 13300 N

Static Capacity C_0 = 7800 N

Inner Diameter = 40 mm

Outer Diameter = 68 mm

Calculated Dynamic Capacity = 6729.64 N.

Hence Bearing selected is safe.

4.5.12 Selection of Bush Type Flexible Coupling

Step:1 Nominal torque $= \dfrac{60 \times 10^6 \times (kW)}{2\pi n} = 71.493\ N.m$

Step:2 From table, selected Application Service Factor is 1.25.

$\therefore$ Design torque $= Nominal\, torque \times Service\, factor = 89.366\ N.m$

Step:3 From table, 8S type coupling is selected[23].

Bush type flexible Coupling Specification :

Coupling Type : S type - 41566

Coupling Size : 8S

Calculated Torque : 89.36 N-m

Maximum Torque : 128.24 N-m

Min. Bore : 19 mm

Max. Bore : 50 mm

Outer Diameter : 138 mm

Overall Length : 112 mm

4.5.13 Selection of Universal Coupling

Different types of universal joints are available: D type, HD type, LOJ type, Needle bearing type, DD and DDX type, etc.

Step:1 $RPM \times Workingangle = 6$

Step:2 Nominal torque $= \dfrac{60 \times 10^6 \times (kW)}{2\pi n} = 5254.77\ N.m$

Step:3 From table, selected Application Service Factor is 1.25.

$\therefore$ Design torque $= Nominal\ torque \times Service\ factor = 6568.08\ N.m$

Step:4 From table, DD-14 type universal coupling is selected[23].

Universal Coupling Specification :

Coupling Type : DD type

Coupling Size : DD-14

Calculated Torque : 6568.08 N-m

Maximum Torque : 7389 N-m

Outer Diameter : 76 mm

Overall Length : 460 mm

Chapter 5

Conclusions and Future Scope

5.1 Conclusions

A bar peeling attachments has been designed that can be used for bar peeling operation on lathe machine.

Design of cutter head has been carried out which is attached on lathe spindle. Cutter head is having four toolholders with cutting inserts. Cutting insert and toolholder have been selected based on cutting parameters.

Design of bar feeding mechanism has been carried out which includes design of gearbox and different components such as shaft for mounting gears, selection of bearing for shafts, selection of coupling and motor.

5.2 Future Scope

As detailed design of bar peeling attachments has been carried out which mainly include Cutter head and bar feeding mechanism. These can be manufactured and tested for its purpose.

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