DESIGN OF THE DRIVE MECHANISM FOR A

RECIPROCATING COAL FEEDER

A THESIS SUBMITTED IN PARTIAL FULFILLMENT OF THE REQUIREMENTS FOR THE DEGREE IN

Bachelor of Technology

in

Mechanical Engineering

By

SAMBIT DAS
DEBASHISH DAS

Department of Mechanical Engineering
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NATIONAL INSTITUTE OF TECHNOLOGY
ROURKELA

CERTIFICATE

This is to certify that the thesis entitled , " DESIGN OF THE DRIVE MECHANISM FOR A RECIPROCATING COAL FEEDER" submitted by Sri Debashish Das and Sri Sambit Das in partial fulfillment of the requirements for the award of Bachelor of Technology Degree in Mechanical Engineering at the NATIONAL INSTITUTE OF TECHNOLOGY, ROURKELA (Deemed University) is an authentic work carried out by him under my supervision and guidance .

To the best of my knowledge ,the matter embodied in the thesis has not been submitted to any other University / Institute for the award of any Degree or Diploma .

Date : 02.05.2007

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CONTENTS

Brief Abstract of the Project

Material handling is undoubtedly the most important and in fact an indispensable job in industries for it is encountered at every stage right from the time raw materials enter the factory gate to the point when it leaves in form of finished products. The engineering of material handling falls under two categories depending on form of material: bulk solid handling and unit handling.

In case of handling lumpy materials like coals etc. , feeder plays a vital role as an uninterrupted source of uniform feed provider to the conveyor system. Although several feeders like belt, apron, screw, feeders etc are available, reciprocating feeders are still in use because it ensures a continuous and controlled feed rate, is low in cost, its drive mechanism is simple, it can handle wide range of miscellaneous materials including lumps, easy in assembly and disassembly and maintenance requirement is quite low.

The challenge which we have taken via the project, is to design a drive mechanism for a reciprocating coal feeder. We call it a challenge because we have to design various intricate components like couplings, worm reducers, gearbox etc. We call it complicated since all the components are interdependent on each other to a great extent. So we cannot design anything randomly . We have to take into considerations the smallest of small things like the various forces acting, how each component can fail under various stress conditions. We have to optimize everything right from the motor selection, to speed reduction ratio selection, to the capacity of coal which we can handle.

We are going to follow the above-mentioned strategies so that our project does not remain just a theory but can become a reality for industries.

INTRODUCTION

PARTS TO BE DESIGNED:

The different parts of the drive Mechanism of the reciprocating coal feeder are as follows:

1) Tyre Coupling
2) Worm Reducer
3) Worm Reducer Housing
4) Gear and Pinion
5) Plumber Block Bearing
6) Eccentric Disc
7) Tie Rod
8) Flat Plate

Description and design considerations of the above mentioned parts of the drive Mechanism that we are going to design:

(1) TYRE COUPLING: It is a flexible type Of coupling where the flexible member is in the shape of a tyre. This type of coupling accommodates or infact compensates angular misalignments up to about 4 degrees and parallel misalignments up to about 4mm and also compensates any end float present.

Some other characteristics of tyre coupling are as follows :

1) Torsionally soft: hence they absorb shock forces easily.
2) Free of Back – Lash: so does not create 'snatch' on take up of the drive.
3) Reduces Damping and Torsional oscillations. This coupling is designed on the basis of H.P. (Torque)to be transmitted and shock loading . The major dimension to be designed are tyre thickness, grip plate diameter and thickness, cap screws , key.

2) WORM REDUCER: Worm and worm-wheel drives are normally used for non-parallel, non intersecting, right angled gear drive systems where high velocity reduction ratio is required. Since in our design also, Power is to be transmitted under high velocity reduction (1:25), hence we also consider worm reducers.

The wide applications of the worm drive system stems from the fact the system affords to have a design with higher Transmission ratio with comparatively lower weight, smaller overall dimensions and space requirements. The arrangement is thus compact and also ensures a smooth and noiseless operation. Self locking ability or irreversibility of drive, is another big advantage of this system.

In this worm gear drive, the worm is designed based upon the center distance and velocity ratio. The worm shaft is designed on bending and twisting moment on it. Diameter, pitch and length of worm and shaft are the major dimensions to be designed. Worm gear is designed on the basis of heat dissipation, wear, bending load on the teeth of worm. Gear shaft is designed on the basis of bending load due to tangential, radial forces acting on it. Dimensions to be designed are worm gear tooth proportions, face width, diameter of shaft etc.

(3) WORM REDUCER HOUSING: The housing accommodates the bearings and the worm and worm wheel. It is designed based on the bearing thickness and the thrust and tangential loads on the worm shaft. While designing the housing, we should keep in mind for simplicity that the number of projected parts, ribs etc. should be minimum to ensure stiffness. The bearings are to be designed on the basis of dynamic load rating of bearing which will be evaluated from the thrust and radial loads of the worm shaft.

Foundation bolts are to be designed based on bending stresses induced due to axial and tangential loads of the worm.

(4) GEAR AND PINION: The gear and pinion drive is designed on the basis upon the horsepower to be transmitted, speed of the pinion, velocity ratio and the center distance. The shafts are to be designed on the combined effect of torque transmitted and bending. Generally, worm gear reducers are seldom repaired, they are replaced after their life span is over. It is the pinion gear drive, which is repaired if any problem occurs. Gear pinion drive helps in further speed reduction, thereby decreasing load on a single stage worm reducer.

(5) PLUMMER BLOCK BEARING: This bearing is selected from the manufacturers S.K.F bearing catalogue based upon the bearing life dynamic load rating and speed of the rotation of shaft. The Plummer block cap is to be designed by taking it as a simply supported beam loaded centrally over a certain span. The rest dimensions of the body are to be determined from empirical relations. Rotating shaft is always supported on bearing. If a long shaft is supported only at two ends, it will deflect at its center due to its own weight . So to have a long shaft straight and its running smooth, shaft is always supported at suitable intervals by bearing. A Plummer block or pedestal bearing is very useful when we are considering high speed and large size shafts and is a split type of journal bearing where the bearing pressure is perpendicular to the axis of shaft. Its split construction facilitates: -

(a) Installing and removing the bearing on and from the shaft.

(b) Adjustment for wears in brasses, when they cannot be reconditioned.

(6) Eccentric Disc: The load to be pulled or pushed by the coal tray through the connecting rods is the main criteria basing on which the eccentric disc is designed.

(7), (8) Tie rod, Flat Plate: The function of the eccentric disc is convert the rotary motion of the gear to the reciprocating motion of the coal tray or the flat plate through the tie rod. Thus Tie rod is the main link between eccentric disc and the flate plate on which coal falls through the hooper. The flat plate continuously reciprocates and the coal which falls on it then goes down to the belt drive (thereby to required destination) which is running continuously only in the forward stroke of the flat plate. The reverse stroke or the back ward stroke of the coal tray (flat plate) is an idle stroke.

FAILURE CRITERIA:

Various failure criteria which have to be considered while designing various parts of drive mechanism of Reciprocating coal feeder.

Failure criteria is the most important aspect in design because its analysis enables the designer to know the types of failures and the failure zone and the life of components under applied load and how the components will behave under varied loading condition. By failure analysis we are able to know whether what we have designed is practically feasible or not.

Following are the various failure criteria of the differents parts to be designed .

1) TYRE COUPLING : The member which is most susceptible to failure is the tyre itself which may fail under the torsional load to be transmitted . Hence
reinforcement of the rubber for Tyre imparts strength it to withstand the loading. The key for coupling shaft has a tendency to fail under shear and crushing.

2) WORM REDUCER: The Failure criteria for worm and worm wheel assembly is bending of teeth, wear and heat generation.

3) WORM REDUCER HOUSING : The bearing which are present in the housing may fail due to overheating, vibration, turning on shaft, binding on shaft displacements etc. Housing walls are subjected to compressive loads.

4) GEAR AND PINION : The gear tooth may fail under bending and wear .The arm may fail in bending . Interference and pitting are other predominant failure criteria for the gear design.

5) PLUMMER BLOCK DESIGN: The bearing may fail due to overheating, vibrations,wear, etc.

6) ECCENTRIC DISC: The pin in the eccentric disc has a tendency to fail due to shear and bending. The eccentric disc may fail due to tearing.

List of Figures

Chapter 1

TYRE COUPLING

DESIGN ANALYSIS

TYRE COUPLING:

MATERIAL FOR TYRE COUPLING:

1) **HUB AND PRESSURE DISC:**

For uneven running ,medium mass accelereation and shock loads,the Ref:1
Cast Iron , IS:210, Grade:20
Cast Iron , IS:1030,grade 2
Forged Steel ,C40,IS:1570

2)Screws: Hexagonal Socket Head Cap Screw Ref:1
Mild Steel IS:2269 page:9.64
3)Tyre: Rubber bonded with rayon chord . Material for tyre
Hardness: (70+5)degree to (70-5)degree IRH . coupling

factor of safety chosen for the type of duty = f1 =1.7 page:9.63
Table:9.37

Minimum factor of safety f1 for types of duty

For above 16 hours of daily operation period, the factor of safety Ref:1
Chosen = f2 =1.25 page:9.64
Table 9.38

Minimum factor of safety for daily operation

Taking frequency of start 1 to 20 ,and with uneven running,medium mass Ref:1
acceleration and shock load ,the factor of safety chosen =f3 =1.07 page:9.64
Table:9.39

Factor of safety f3 for frequency of start

Design Power = Nominal Power *f1*f2*f3
Nominal Power =Power of motor = 20 h.p.(given)
Therefore , Design Power = 20 *1.7*1.25*1.07 =45.475 h.p
Power/Speed = (Power in metric horse power(PS))/(speed in R.P.M)
=(45.475)/(1440) = 0.031579 PS/R.P.M .
Therefore, PS/100 R.P.M = 0.031579*100 =3.1579 h.p .
Corresponding to the above designed power (i.e. 3.1579 h.p per 100 R.P.M),
the next higher value is taken from the standard table :

For P.S/100 r.p.m = 6.3 (standard value),we have:
Size = 5
Torque = 450 Nm
Maximum Speed = 2600 r.p.m
Bore (Minimum Rough) = 38 mm
Bore (Maximum) = 90 mm
A = 277 mm ,B = 145 mm, C = 225 mm , D = 205 mm , E =132 mm , F = 76 mm,
G = 53 mm .

CHECKING FOR DESIGN FAILURE:

Fig 1.2 A Tyre Coupling

Design h.p = 45.475

Therefore , Design Torque = (4500 * Design h.p) /(2* 3.14* R.P.M) (kgm)
= (4500 * 45.475) / (2 * 3.14 * 1440)
= 22.6174 kgm
= 22.6174 * 9.81 Nm
= 221.8767 Nm

which is less than 450 Nm (The standard value), corresponding to size number 5 . Also ,the speed of the shaft is 1440 r.p.m which is within the limit of 2600 r.p.m (maximum speed) .Therefore the selection of TYRE COUPLING is satisfactory .

DESIGN OF KEY FOR TYRE COUPLING:

Let us select parallel keys and keyways for our design .
Material of key is selected as C 50 for which the shear strength
= τ (s) =720 kg /cm^2
And the crushing strength = σ (c) =1500 kg/ cm^2
As the shaft diameter = 90 mm (standard maximum bore diameter) , the corresponding standard dimensions of parallel key are : -

For shaft diameter above 85 mm and below 95 mm

Ref –2
Page :6.6
Table 6.2
Dimensions of parallel keys and keyways

KEY CROSS-SECTION: Width = b= 25 mm ,Height = h = 14 mm
Keyway depth in shaft = t1 =9 mm
Keyway depth in Hub = t2 =5.4 mm
Preferred length of key =L = 90 mm

Ref:2
Page 6.7
(Preferred length of parallel keys)

CONSIDERING THE KEY TO FAIL IN SHEARING :
Tmax = L * b *τ (shear) *(dshaft /2)
Where Tmax is the maximum torque which coupling can transmit .
➔ 450 Nm = (90/1000) * (25 /1000) * τ(shear) *(90/1000) *(1/2)
Max shear stress which can develop = τ (shear) = 4444444.44 N/m^2
= 45.30479 kg/cm^2
< 720 kg/cm^2

Since the maximum induced shear stress which can develop in the key considering maximum possible Torque Transmission is less than the allowable shear stress for the key material , hence the key design is safe as per shear consideration .

NOW CONSIDERING THE KEY TO FAIL IN CRUSHING :

$T_{max} = L * (h \div 2) * \sigma(crushing) * (d(shaft) \div 2))$

$450 * 1000$ N mm $= 90$ mm $* (14 \div 2)$ mm$* \sigma(crushing) * (90 \div 2)$ mm

Therefore, $\sigma(crushing) = 15.873$ N/mm^2 = 161.8044 kg/cm^2

< 1500 kg/cm^2 (σ(crushing) allowable)

Since induced crushing stress is very-very less in comparision to allowable crushing stress of the selected key material (C 50 ,$\sigma(c)$ =1500 kg/cm^2) ,

Therefore the key is safe as per crushing consideration .

Therefore the Key Design For The Tyre Coupling is safe from point of consideration of shear and crushing .

Chapter 2

WORM REDUCER

DESIGN ANALYSIS
WORM WHEEL
WORM GEAR

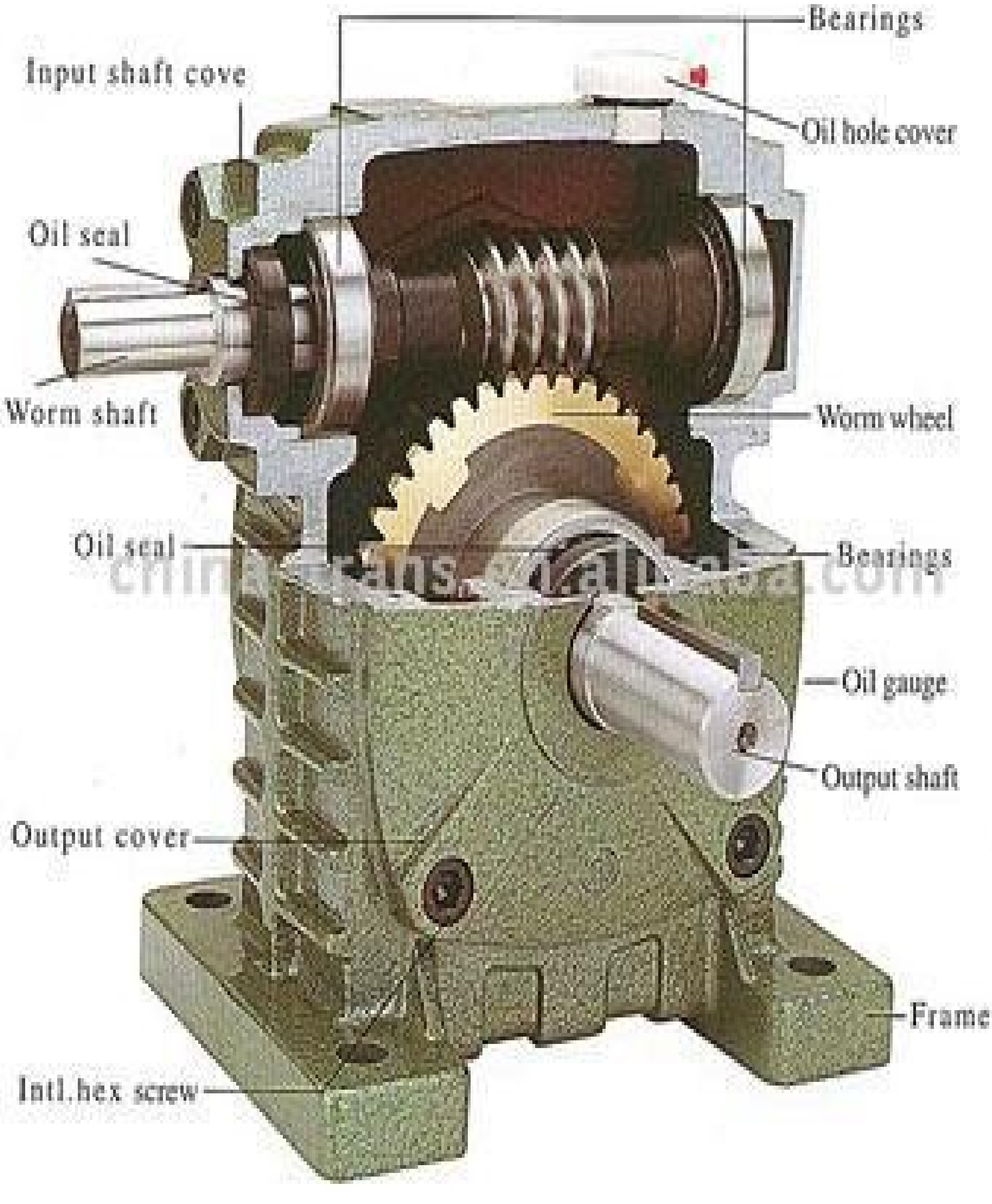


Fig 2.1 Inside view of a Worm Reducer

DESIGN OF WORM REDUCER:

The primary considerations while designing a worm reducer for an industrial purpose like reciprocating coal feeder are as follows :

(1) To transmit power efficiently .
(2) To transmit power at a considerable reduction in velocity .
(Though it can give velocity reduction ratios as high as 300:1 or more in a minimum of space but it has lower efficiency .)
(3) To provide a considerable mechanical advantage so that a given applied force must be able to overcome a comparatively high resisting force .

The meshing action in a worm - drive is a combination of sliding and rolling motion with sliding prevailing at higher reduction ratios . In this respect , this drive is similar to crossed helical gear drive system . But the worm drive has greater load carrying capacity than the crossed helical system because worm drive has a line contact whereas a crossed helical drive has a point contact only . Another important thing which we should note is that since a considerable amount of driving energy is dissipated mainly in the form of heat due to the sliding action of the mating components in a worm -set entailing frictional power loss ,it is therefore logical for us as a designer to minimize the coefficient of friction by selecting dissimilar metals for the worm and gear .

DESIGN CALCULATIONS :

The centre distance (C_w) is defined as the perpendicular distance between the axis of worm shaft and the worm wheel shaft and is given by the relation :

$$(C_w \div Ln) = (1 \div 2\Pi) * ((1 \div \sin\lambda) + (V.R \div \cos\lambda))$$

where Ln = Normal Lead , V.R = Velocity Ratio = 25:1
λ = Lead Angle

LEAD ANGLE IS GIVEN BY THE RELATION :

$(\cot\lambda)^3 = V.R = 25 \Rightarrow (\tan\lambda)^3 = (1 \div 25)$

- ➔ λ = 18.88 degrees
- ➔ This value of λ gives us the minimum centre distance that can be used with a given lead or inversely the maximum lead that can be used with a given centre distance .

Let us take : STANDARD CENTRE DISTANCE = C_w = 250 mm = 25 cm

Ref -3
Page 4.18
Table 4.4

Standard centre distance and other parameters of worm gearing

Therefore we have :

=> $(C_w \div L_n) = (1 \div 2\Pi) * ((1 \div \sin\lambda) + (V.R \div \cos\lambda))$

=> $(25 \div L_n) = (1 \div 2\Pi) * ((1 \div \sin 18.88) + (25 \div \cos 18.88))$

=> $L_n = 5.3225$ cm

Therefore the Axial Lead = $L = (L_n \div \cos\lambda) = (5.3225 \div \cos 18.88)$
=> $L = 5.625$ cm

Number of starts to be used on the worm =2 (double start) (for velocity ratio 25:1)

Ref .4
Table 31.2
Page:993

(Number of starts to be used on the worm for different velocity ratio)

Axial Pitch of the threads = P_a = (Axial Lead ÷ No. of starts)
=> $P_a = L \div 2 = 5.625 \div 2$
=> $P_a = 2.8125$ cm

Now, $P_a = \Pi *$ module $= \Pi * m$
=> $m = P_a \div \Pi = 2.8135 \div \Pi = 0.89556$ cm
=> $m = 8.9556$ mm

Therefore Standard Module Chosen = m(standardized) = 10 mm =1 cm
(next higher value of module is chosen)

Ref .3
page 4.15
Table :
4.3

(Standard Dimensions Of Worm Gearing)

<u>NOW WE CALCULATE THE VALUES AS PER THE STANDARD MODULE CHOSEN :</u>

Axial Pitch = $P_a = \Pi * m = \Pi * 1$cm = 3.14159 cm
Axial Lead = L = No. of start * axial pitch = $2 * P_a = 2 * 3.14159$ cm
=> $L = 6.28318$ cm
Normal Lead = $L_n = L * \cos\lambda = 6.28318 * \cos 18.88$
=> $L_n = 5.945$ cm

NOW WE AGAIN CALCULATE THE CENTRE DISTANCE (Cw) :

$((C_w)\text{calculated} \div L_n) = (1 \div 2\Pi) * ((1 \div \sin\lambda) + (V.R \div \cos\lambda))$

Putting the values : $L_n = 5.945$ cm , $\lambda = 18.88$ degrees , V.R = 25 in the above relation we get :

(C_w) calculated = 27.923 cm

As the calculated value of C_w is greater than the assumed value of C_w ,so we standardize the centre distance with the next higher value of C_w .
Therefore (C_w) standardized = C_w = 31.5 cm

Ref :3
Page:4.18
Table :4.4

(Standard centre distance and other parameters of worm gearing)

Velocity ratio = (No. of teeth on worm wheel ÷ No.of starts on worm)
=> V.R = n(g) ÷ 2
=> n(g) = 2 * V.R = 2 * 25 = 50 Teeth

Now we know that : $\tan\lambda = (L \div \Pi d(p))$ where $\Pi d(p)$ = pitch circumference of worm
=> $d(p) = (L \div \Pi \tan\lambda) = (6.28318 \div (\Pi \tan 18.88))$
=> d(p) = pitch circle diameter of worm = 5.848 cm

PROPORTIONS OF WORM :

The pitch circle diameter of worm d(p)(worm) in terms of centre distance between the worm and the worm wheel shaft is given by an empirical relation :

$d(p)\,(worm) = (C_w)^{(0.875)} \div 1.416$ where Cw is in mm — Ref:4 page:993 Notes :1

$d(p)\ (worm) = (31.5 * 10)^{(.875)} \div 1.416 = 108.382$ mm

- ⇨ d(p) (worm) = 10.838 cm
- ⇨ Since d(p)(worm)(= 10.8382 cm) > 5.848 cm (previously calculated value of d(p)) , hence we take the higher value as the pitch circle diameter of worm .

Therefore , d(p)(worm) = d(p) = 10. 838 cm

VARIOUS PROPORTIONS ARE :

FOR DOUBLE THREADED OR START WORMS , WE HAVE :

1) Normal pressure angle = Φ = 14.5 degrees — Ref .4 page 993 Table 91.3

2) Pitch circle diameter = d(p) = 10.838 cm
3) Maximum bore for shaft = [p(c) mm + 13.5 mm] = [p(a) cm +1.35 cm]
(since axial pitch (p(a)) = circular pitch (p(c)) for a worm gear drive)

VARIOUS PROPORTIONS ARE:

For Double Threaded or start worms, we have :

---

1) Normal Pressure angle = $\phi = 14\ 1/2^0$

2) Pitch Circle diameter = dp = 10.838 cm Ref.4
of worm page 993
3) Maximum bore for shaft Table 91.3
= p_c + 13.5mm = p_a + 13.5mm Proportions for worm
Since axial pitch (pa) = circular pitch (pc)

= 3.14159 cm + 1.35 cm = 4.49159cm
4) Hub diameter = 1.66 pc + 25mm
= 1.66 pa + 2.5 cm
=(1.66 x 3.14159 + 2.5)
= 7.715 cm
5) Face Length or the length of threaded
portion = L =Pa(4.5 + 0.02 x no. of start)
L = 3.14159cm x [4.5 +0.02 x 2]
L = 14.263 cm
The face length of the worm should be increased by 25 mm to 30 mm for the feed marks produced by the vibrating grinding wheel as it leaves the thread root.
∴ L = 14.263cm + 30mm = 17.263cm.

6) Addendum Circle diameter
d(a) = dp + 2x(module) = 10.838cm + (2x1)cm.
d(a) = 12.838cm.

7) Dedendum Circle diameter = d(d)
d(d) = d(p) – 2.4m = 10.838cm – (2.4x1cm)
= 8.438cm.

PROPORTIONS OF WORM WHEEL:
No of Teeth on Worm Wheel = n_g = 50 (already calculated)

Pitch circle diameter of worm wheel
D(p) = n_g x m where m = module
= 50 x 1cm = 50cm

Addendum circle diameter for worm wheel
D(a) = D(p) + 2m = 50 + (2x1) = 52cm.

Dedendum circle diameter for worm wheel.
D(d) = D(p) – 2.4m = 50 – (2.4x1) = 47.6cm.

Outside diameter of wheel = D(o) = Da + 1.5cm.

$D(o) = 52 + 1.5cm = 53.5cm.$

Rim width $Br \leq 0.75 \times d(a)$

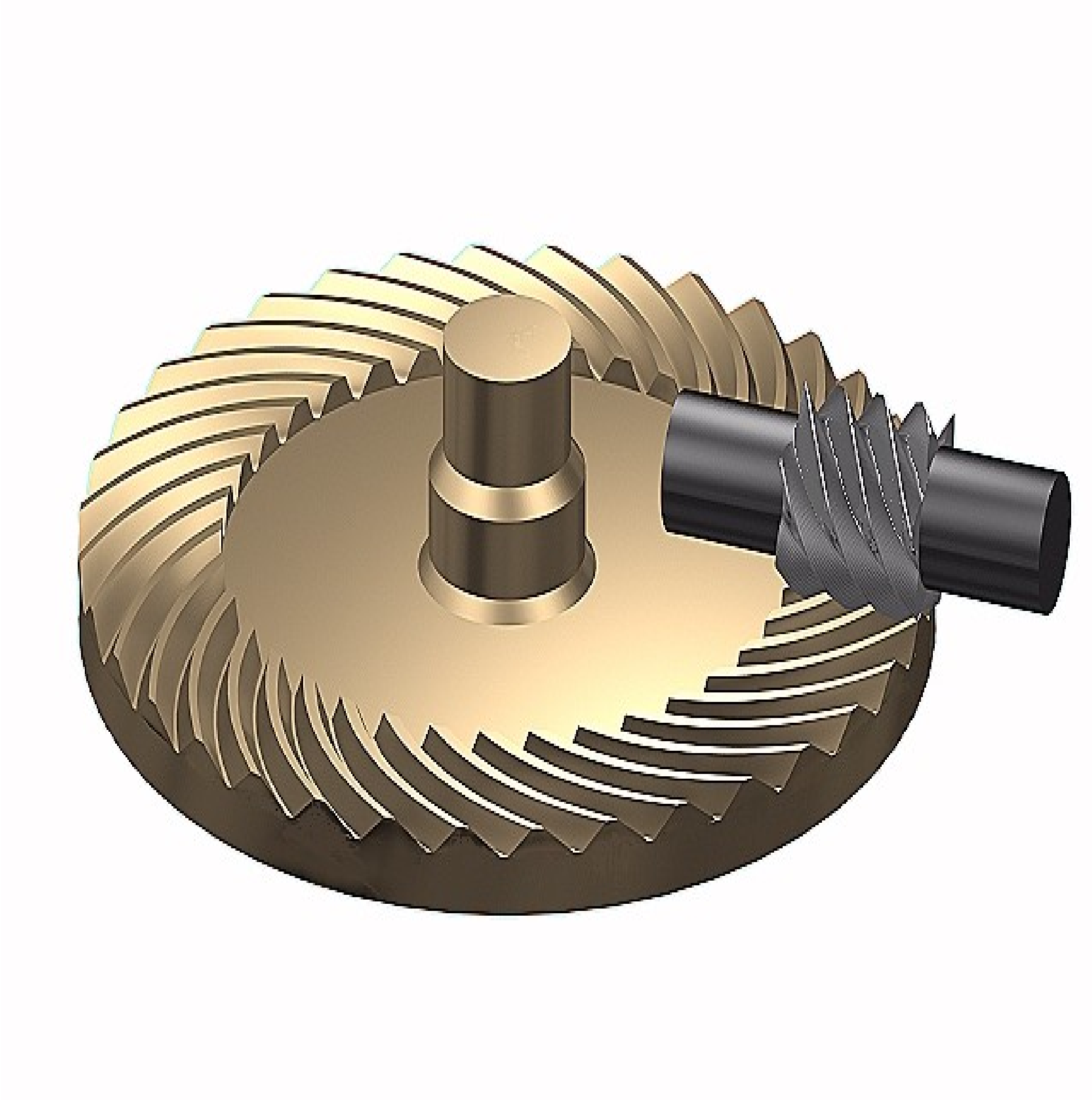


Fig 2.2 Meshing of worm and worm wheel

$Br \le 0.75 \times 12.838cm \le 9.6285$

$Br = 9.6cm.$

Face width(f) :- Let the face angle = $2V=90^0$

Therefore, $V = 45^0$

$$F = (da - 2m) \times \left(\frac{V x \Pi}{180^0} \right)$$

$$F = (12.838 - 2x1) \times \frac{45^0 x \Pi}{180^0} = 8.512cm.$$

The Adopted value of f = 8.6cm.

DESIGN LOAD CONSIDERATIONS AND STRENGTH OF WORM GEAR TEETH:

Pitch line velocity of the worm gear = $Vm = \frac{\Pi\ DpNg}{60}$

$$Vm = \frac{\Pi \times (0.5) \times (1440)}{60 \times (25)} = 1.50796 \text{ m/s}$$

$$= 90.4776 \text{ m/min}$$

Where Dp = pitch circle diameter of worm wheel or gear = 50cm = 0.5m

Useful Transmitted Torque Load = Ft

$$Ft = \frac{\text{(H.P)rating of motor}}{\text{Pitch Line velocity}}$$

Ft = (20 x 4500)÷(90.4776) =994.7213 kgf =9758.2163 N

Velocity factor = Cv = 6÷(6+Vm) where " Vm " is the peripheral velocity of the worm gear in m/s .

Therefore , Cv = 6÷(6+1.50796) = 0.79915 page995

Strength of worm Gear teeth

Let us choose overload factor Co =1.25x1.25 Ref:5

where (1.25) is for 24 hours operation per day for moderate shock condition Table—22.7 Page –824

Co = 1.25x1.25 =1.5625

Reliability factor = Cr =1

Life factor = C(L) =1

Therefore , Design load = [(Co x Cr) ÷(Cv x C(L))] xFt

F(design) = 19079.2879 N

MATERIAL FOR WORM WHEEL :

We choose the material of worm wheel to be Phosphor gear bronze SAE 65 (chilled cast) whose allowable static stress(σo) = 82.4 Mn/m^2

σ_o = 82.4 N/mm^2

Brinell Hardness Number = 100

Ref:2
Page –15.8
Table—15.4
Static allowable stress

We have chosen the above material for worm wheel since Bronzes are very popular in worm gear drives because of their ability to withstand heavy sliding loads and ability to "Wear in" to fit hardened steel worms. They are also very useful for corrosive conditions and can also be easily cast into complex shapes. Another reason for selecting the material is that because the worm wheel teeth are subjected to greater wear because of continuous contact.

STRENGTH OF WORM GEAR TEETH:

In considering the strength of worm gear Teeth, it is always safe to assume that the teeth of worm gear are always weaker than the threads of worm since the worm gear is subjected to greater wear than worm. In worm gearing, two or more teeth are usually in contact, but due to uncertainty of load distribution among themselves it is assumed that the load is transmitted by one tooth only.

(a) We know that according to Lewis Equation :- The beam strength equation can be written as $W_T = (\sigma_o \times C_V) b \times \pi m \times y$.

where W_T = permissible tangential tooth load or beam strength of worm gear teeth.

σo= Allowable static stress = 82.4 N/mm^2

b = face width = f (already calculated) = 8.6cm

m = module = 1cm.

y= Tooth form factor = Lewis factor

$y = 0.124 - \dfrac{0.684}{n_G}$ for 4½°involuteTeeth.

Ref-4
page – 995
art 31.8
Strength of worm Gear teeth

$y = 0.124 - \dfrac{0.684}{50} = 0.11032$

$\therefore \quad W_T = (82.4 \times 0.79915) \times 86 \times (\pi \times 10) \times (0.11032)$

W_T = 19627.18846 N.

Since the beam strength of Work Gear teeth (W_T=19627.18846N) is more than the design load (F(design) =19079.2879 N)
∴ the design is safe.

(b) CHECK FOR DYNAMIC LOAD:

Dynamic tooth Load on worm gear = W_D

$$W_D = \frac{W_T}{C_V} = \frac{19627.18846}{0.79915}$$

W_D = 24560.08066N

Since Dynamic Tooth load on worm gear > useful transmitted torque load (Ft = 9758.2163N) acting on worm gear, therefore the design is safe from stand point of dynamic load.
Even though we have checked for dynamic load case, still it is not so required because it is not so severe due to sliding action between worm and the worm gear.

(c) CHECK FOR STATIC LOAD OR ENDURANCE STRENGTH:

We know that flexural endurancelimit for phosphor bronze =σ_e = 168Mpa or N/mm^2.

Page--995
Art:31.8
Strength of worm gear teeth

The endurance strength of worm gear tooth
W_s =σe x.b(πm)y.
W_s=(168 x 86 x (π x 10) x 0.11032)N
W_s=50073.95086N
Since Ws is much more than permissible Tangential Tooth Load (W_T=19627.18846 N).

∴ The design is safe from standpoint of static load or endurance strength.

(d) CHECK FOR WEAR:

The limiting or maximum Load for wear (Ww)is given by :-
Ww= Dp x f x k
Dp= 50cm = pitch circle diameter of worm gear.
F = face width of worm gear = 8.6cm.
For a suitable combination of worm gear material as chilled phosphor bronze and worm material as Hardened steel, the load stress factor or material combination factor 'K' is given by :

k=.830 /mm^2. Ref:4

Page: 996
Art: 31.9
Wear tooth load for worm gear

Since the lead angle for our worm gear drive = λ = 18.88^0 (which lies between 10^0 and 25^0)

∴ Value of K should be increased by 25% .

∴ $K = \frac{125}{100} \times 0.830 = 1.0375$ N/mm^2

∴ Ww = (500mm) x (86 mm) x (1.0375)N/ mm^2
= 44612.5 N

∴ Since the maximum load for wear > Design Load(F design = 19079.2879 N), hence our design is safe from standpoint of wear .

(e) CHECK FOR HEAT DISSIPATION:

The effect of heat generation is an important design criterion for the worm drive. The relative sliding action between the teeth of worm and worm wheel causes generation of a considerable amount of heat. And heat dissipation is normally achieved by providing fans which are provided inside the gear box housing or the housing may be so cast as to have cooling ribs or both the above measures can be taken. The heat dissipation depends upon the size and surface of housing and on the velocity of air surrounding the housing.

The rubbing velocity = Vr = $\frac{\pi \times dp \times Nw}{100 \times \cos\lambda}$

dp, Nw = Pitch circle diameter and speed of worm respectively .

$$Vr = \frac{\pi \times 10.838 \times 1440}{100 \times \cos(18.88)^\circ}$$

Vr = 518.1784 m/min = 8.6363 m/sec.

Coefficient of friction = $\mu = 0.025 + \frac{Vr}{18000}$

(here 'Vr' is in m/min)

$$\mu = 0.025 + \frac{518.1784}{18000} = 0.053787$$

∴ angle of friction = $\Phi_1 = \tan^{-1}(\mu)$

$\Phi_1 = \tan^{-1}(0.053787) = 3.0788^0$

$\therefore$ efficiency of worm gearing = $\eta = \dfrac{\tan\lambda}{\tan(\lambda + \phi 1)}$

Fig 2.3 Full view of a gear box

$$\eta = \frac{\tan 18.88^\circ}{\tan(18.88^\circ + 3.0788^\circ)} = 0.8482$$

$\therefore$ efficiency of worm gearing = 100 x η
= 84.82 %

Assuming 25% overload, heat generated is given by:

Q = 1.25 x power transmitted x (1-η)
Q =1.25 x 20h.p x (1-0.8482)
Q =1.25 x (20 x 746.5)W x (1-0.8482)
Q = 2832.9675 W

Now, the projected area of worm gear=A_g

A_g = (π/4)x(50^2) = 1963.495 cm^2

Projected area of worm = A_w = (π/4)(10.838^2) = 92.2546 cm^2

Total projected area of worm and worm gear

= Total heat dissipating Area = $A_g + A_w$

$\therefore$ Total heat dissipating Area = $A_g + A_w$

$\therefore$ A= Ag + Aw =(1963.495 + 92.2546) Cm^2

A = 2055.7496 cm^2

Heat dissipating capacity = Qd = $A(t_2 - t_1)$ x K
A= Total heat dissipating Area = Area of housing
$(t_2 - t_1)$ = temperature difference between the housing surface and sorrounding air
K = conductivity of material = 378 $W/m^2/^0C$(average value)
= (378 x 10^{-4}) $W/cm^2/^0C$

Qd = A $(t_2 - t_1)$ x k = 2055.7496 x $(t_2 - t_1)$ x (378 x 10^{-4})

Qd = 77.7073 $(t_2 - t_1)$ W

The heat generated must be dissipated in order to avoid overheating of drive

$\therefore$ Q = Qd
2832.9675 = 77.7073 $(t_2 - t_1)$
$\therefore$ $(t_2 - t_1)$ = 36.45688 ^0C

For worm gear drive, the permissible limit or safe limit for temperature difference is from 27^0 C to 39^0C

Since our calculated temperature difference lies within that range, hence the design is safe.

Force acting on worm and worm gear :

In this analysis, it is considered that the worm is the driving member while the worm wheel is the driven member. The various component of resultant forces acting on the worm and worm wheel are as follows :-

1. (a) Tangential component on worm (F_{tw})
   (b) Tangential component on worm gear (F_{tg})
2. (a) Axial component on worm (F_{aw})
   (b) Axial component on worm gear (F_{ag})
3. (a)Radial component on worm and worm gear which are equal in magnitude and opposite in direction (F_r) .

From the figure it is clear that :

1) Axial component on worm =Tangential component on worm gear(equal in magnitude and opposite in direction)

   Faw = Ftg

2) Tangential component on worm = Axial component on worm gear (equal in magnitude and opposite in direction)

   Ftw = Fag

3) Radial component on worm = radial component on worm gear =Fr

   The radial or separating force tends to force the worm and worm gear out of mesh . This force also bends the worm in a vertical plane .

DESIGN OF WORM SHAFT :

Torque acting on the gear shaft = Tg = (1.25 x h.p. x 4500) ÷(2πNg)

Tg = (1.25x20x4500)÷[2π(1440÷25)]

Tg = 3049.4335 Nm

(assuming 25% extra over load)

Torque acting on worm shaft = $Tw = \dfrac{Tg}{(V.R)x\eta}$

$$Tw = \frac{3049.4335}{25 \times 0.8482} = 143.8072 N - m$$

Tangential Force on the Worm Shaft = Ftw = $\frac{Tw}{dp/2}$

$$Ftw = 2\frac{Tw}{dp} = \frac{2x143.8072 \times 1000 N - mm}{10.838 \times 10 mm}$$

Ftw = 2653.7589 N = (Axial Force on Wormgear Fag)

Axial Force on Worm = Faw = Tangential force on worm gear shaft = Ftg

$$Faw = Ftg = \frac{2Tg}{Dp} = \frac{2 \times 3049.4335 \times 1000 N - mm}{(50 \times 10) mm} = 12197.734 N$$

Radial or separating force on worm

= Radial or separating on force on worm gear

= Fr = Faw tanΦ = 12197.734 x tan(14 ½ ^{0}) = 3154.5485 N

The minimum length between two bearings should be equal to the value of the diameter of worm wheel. But for easy assembling and dismantling purposes, the provided length between two bearings is more than the worm wheel diameter .

Let us assume L1 = 60 cm

assume L1 = 60cm

Bending Moment due to radial Force Fr in the

vertical plane = $(B.M)_r = \frac{Fr \times L1}{4} = \frac{3154.5485 \times 60}{4}$

$\therefore (B.M_w)_r$ = 47318.2275 Ncm

Bending Moment due to axial force Faw in the vertical

Plane $(B.M_w)_a$ = Faw x $\frac{dp}{4} = \frac{12197.734 N \times 10.838 cm}{4}$

$(B.M_w)_a$ =33,049.76027 N-cm

$\therefore$ Total Bending Moment in the vertical plane

$$(B.M_w)_v = (B.M_w)_r + (B.M_w)_a$$

$(B.M_w)_v$ = 47318.2275 + 33,049.76027 = 80,367.98777 N-cm

Bending Moment due to Ftw in the horizontal plane

$(B.M_w)_H$ = Ftw x $\frac{L_1}{4}$ = $2653.7589 \times \frac{60}{4}$ = 39,806.3835 N-cm.

$\therefore$ Resultant bending Moment on the worm

Mw = $\sqrt{(B.Mw)_v^{\,2} + (B.Mw)_H^{\,2}}$

Mw = $\sqrt{(80{,}367.98777)^2 + (39{,}806.3835)^2}$

Mw = 89,685.9053 N-cm

$\therefore$ Equivalent Twisting moment on worm shaft

(Te)worm = $\sqrt{(Tw)^2 + (Mw)^2}$

(Te)worm = $\sqrt{(14380.72)^2 + (89685.9053)^2}$ = 90,831.5294 N-cm

MATERIAL FOR WORM SHAFT:

The material for worm shaft chosen is $\longrightarrow$ Grey Cast Iron
Is Designation $\longrightarrow$ FG 2 00 Tensile strength (MP a or N/mm^2)=300
Brinell Hardness No. $\longrightarrow$ 180 to 230
We choose the above material because of the followings reasons:-

1) A very good property of grey cast iron is that the free graphite (Carbon present in the from of free graphite gives it grey colour) in its structure acts as a lubricant. Hence low lubrication requirement can be achieved.
2) Grey cast iron have good wearing characteristics and excellent mach inability and can be given any complex shape without involving costly machining operations.
3) It has an excellent ability to damp vibrations, (i.e) it has high damping Characteristics.
4) It has more resistance to wear even under the conditions of boundary lubrication.

<u>The design of the worm shaft is based on the shear strength of grey cast iron</u>

The shear strength of grey cost iron chosen is τ_S=225 N/mm^2 (ultimate shear strength)

Ref-2. Page-1.2 Table 1.1 Properties of different varieties of cast iron.

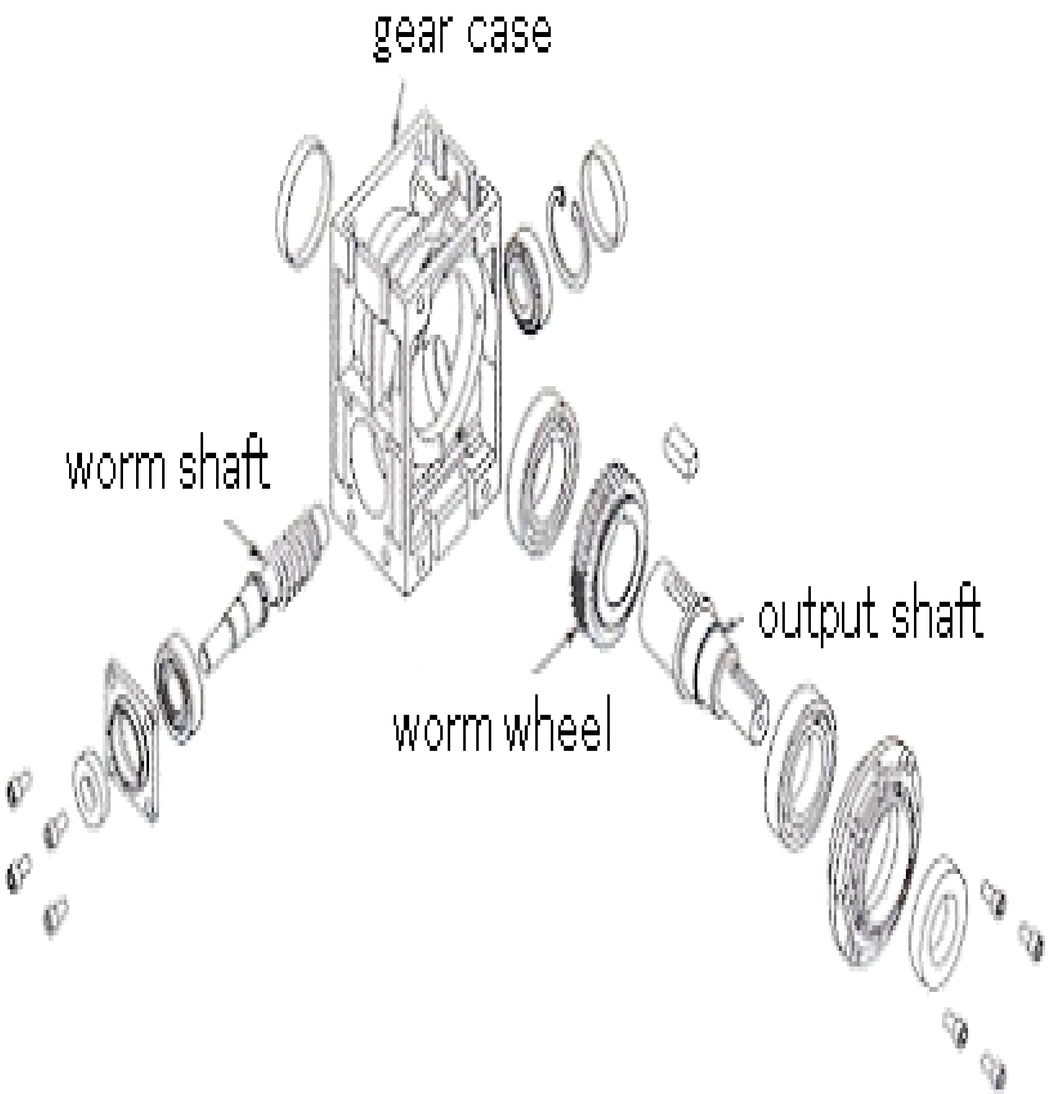


Fig 2.4 Gear Box Assembly

We have the relation:

$\frac{\pi}{16} \times (d_W)^3 \text{ X } \tau_S$ = Equivalent Twisting Moment on worm shaft

$\rightarrow \frac{\pi}{16} \times (d_W)^3 \text{ X } 225_{(N/mm2)} = 90831.5294 \text{ X}10\text{(N-mm)}$

$\rightarrow d_W^3 = 20560.05247 \text{ .mm}^3$

Ref –2 page 7.6 Table 7.5 Standard shaft sizes, mm

$\therefore$ diameter of worm shaft = $d_W = (20560.05247)^{\frac{1}{3}}$

$\rightarrow d_W = 27.3952$ mm

$\therefore$ Standard diameter of worm shaft

= (d_W) standardized = d_W = 30mm =3cm

WORM GEAR SHAFT :

Axial force on the worm wheel (F_{tg}) =Tangential force
On the worm shaft = 2653.7589 N.

Twisting force worm (F_{tg}) =axial force on worm
F_{tg} = 12197.734 N.
Radial Force on worm wheel =Fr = 3154.5485 N.
Bending Moment due to axial force will be in the vertical plane and is given by

$(B.Mg)_{V1}$ = Axial force$\times \frac{Dp}{4}$

$(B.Mg)_{V1} = 2653.7589 \times \frac{50}{4}$ (Cm)
(N)

$(B.Mg)_{V1}$ = 33,171.9863 N-Cm
= 331719.863 N-mm

Bending Moment due to radial force will also be in the vertical plane and is given by
$(B.Mg)_{V2}$ = Radial force$\times \frac{L_2}{4}$

The value of $L_{2 \text{ chosen}}$ = 40 cm. =400mm.

$\therefore (B.Mg)_{V2} = (3154.5485 \times \frac{400}{4})$ N-mm

$= \frac{1261819.4}{4}$ N-mm = 315454.85 N-mm

$\therefore$ Total Bending Moment in the vertical plane

$(B.Mg)_V = (B.Mg)_{V1} + (B.Mg)_{V2}$

$= 331719.863 + \frac{1261819.4}{4}$

Bending Moment due to the Twisting force or Turning force will be in the horizontal
Plane = $(B.Mg)_H$ = Turning force x L_2

$$(B.Mg)_H = \left(12197.734 \times \frac{400}{4}\right) \text{ N-mm}$$

$$= 1219773.4 \text{ N-mm}$$

$\therefore$ Resultant bending Moment on worm wheel

$$Mg = \sqrt{(B.Mg)_H^2 + (B.Mg)_v^2}$$

$$Mg = \sqrt{(1219773.4)^2 + (647174.713)^2}$$

$$Mg = 1,380,826.657 \text{ N-mm}$$

Twisting Moment on wheel shaft.

$$Tg = 3049.4335 \text{ N.m}$$

$$\text{(i.e) } Tg = 3049433.5 \text{ N.mm}$$

$\therefore$ Equivalent Twisting Moment for the worm gear shaft = (Te) gear/$_{wheel}$

$$(Te)\ gear/_{wheel} = \sqrt{(Mg)^2 + (Tg)^2}$$

$$(Te)\ gear = \sqrt{(1,380,826.657)^2 + (3,049,433.5)^2}$$

$\therefore$ (Tg) gear = 3, 347,495.62 N-mm

MATERIAL FOR WORM GEAR SHAFT:

We choose the material for worm gear shaft as 35 Nil Cr 60 whose Composition is as follocos:-

Ref-4 page-29. Table2.8 Composition and uses of alloy steels according to IS:1570-1961 (Reaffirmed 19993)

Carbon $\longrightarrow$ 0.30 – 0.40 %

Silicon $\longrightarrow$ 0.10 – 0.35 %

Manganese $\longrightarrow$ 0.60 – 0.90 %

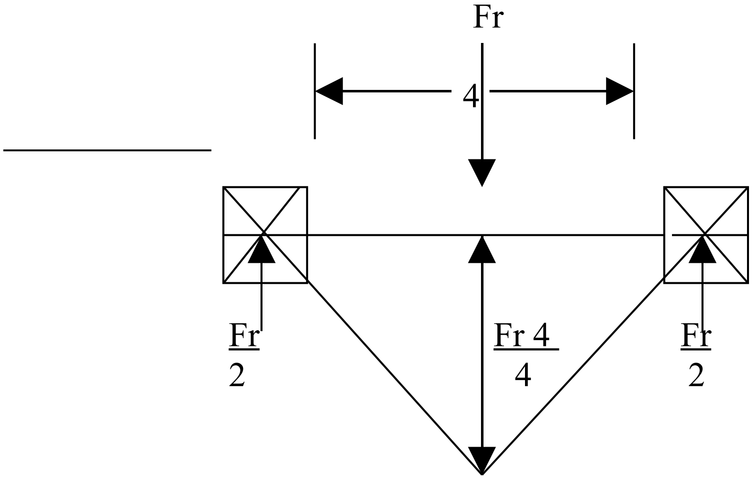
Nickel $\longrightarrow$ 1.00 –1.50 %

Cromium $\longrightarrow$ 0.45 – 0.75 %

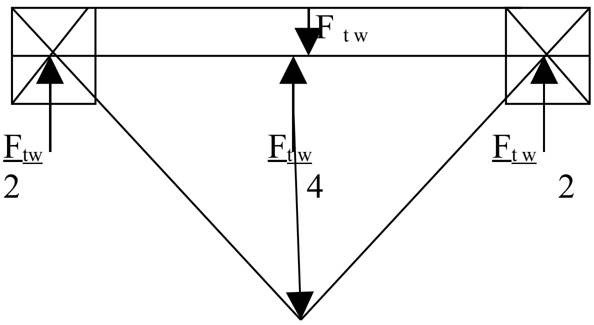
The working stress for this material is τ_S =300 N/mm^2

We have chosen this material (i.e) (Ni-Cr Steel)because of the following reasons :
_Because is this range of Nil-steel which we have chosen nickel contributes great strength and hardness with high elastic limit, good ductility and good resistance to corrosion.

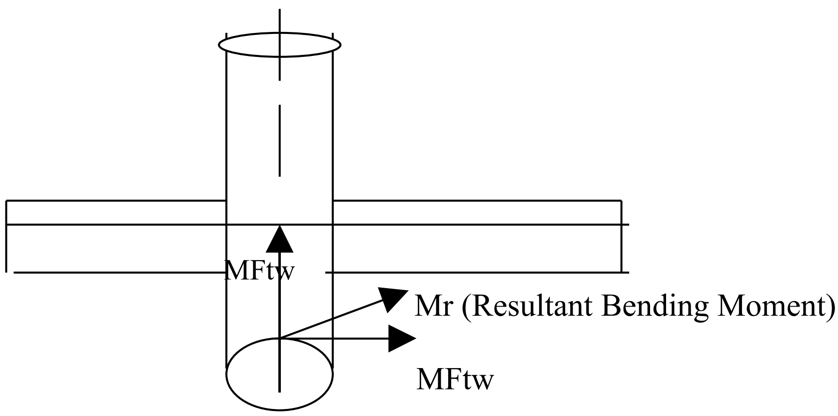
•



Bending Moment due to radial Force Fr in the vertical Plane



Bending Moment in Horizontal plane



Resultant Bending Moment

Fig 2.5 Bending moment diagram of Worm shaft

1) Chromium which is added as an alloying element to combine hardness with high strength and elastic limit.

2) This types of steel also has good shock resistance and high toughness.

We have: $\frac{\Pi}{16} \times (dg)^3 \times \tau_S$ = equivalent Twisting moment on worm gear shaft.

Ref-2 Page-7.6 Table 7.5 Standard shaft sizes mm

$\longrightarrow \frac{\Pi}{16} \times (dg)^3 \times 300 \ (N/_{mm2}) = 3,347,495.62$

$\therefore$ dg = 38.446 mm

$\therefore$ Standard value for worm gear shaft =(dg) standardized = 40mm = 4cm

KEY FOR GEAR :

Ref-2 page6.6, Table6.2 Dimensions of Parallel keys and key ways.

As the diameter of worm gear shaft is 40mm, the standard dimensions of parallel keys chosen are as follows:-

Key Crass Section : Width =b-12mm
Height = h = 8mm

Key way depth (nominal). : in shaft = t_1 = 5mm
: in hub = t_2 = 3.3mm

As the face width of worm gear (wheel) =8.6cm
= 86 mm, Let us take the length of key = L = 90mm.
The material for key selected is C =0.60 to 0.80 with
τultimate =575 N/mm^2
τyield = 206 N/mm^2

Ref-2, Page1.6 table1.5 Properties Carbon Steel.

Factor of Safety = $\frac{\tau ultimate}{\tau yield} = \frac{575}{206}$ = 2.79 N 3

$\therefore$ Working or allowable shear stress

$(\tau_S)_{allowable} = \frac{\tau ultimate}{\text{Factor of safety}} = \frac{575}{3} = \underline{191.667}$ N/mm^2

Considering the key to be failing in shearing :

Twisting moment of wheel shaft (T) = L x b x τs x $\frac{dg}{2}$

3,049, 433.5 = 90 x 12 x τs x $\frac{40}{2}$

$\therefore$ τs = 141.1774 N/mm^2
Since the induced shear stress (τs = 141.1774 N/mm^2) is less than the allowable shear stress

$\therefore$ (τs) allowable =191.667 N/mm^2), hence the design is safe as per shear consideration.

Now, Let us Consider the key to fail in Crushing:

T = L x $\frac{h}{2}$ x 6c x $\frac{dg}{2}$

(allowable Crushing stress ≥ 2 x allowable shear stress)
(τc) allowable = 2 x (τs) allowable

$$=(2 \times 191.667) = 383.334\ N/mm^2$$

Now we have:

$$T = L \times \frac{h}{2} \times \sigma c \times \frac{dg}{2}$$

$$3{,}049{,}433.5 = 90 \times \frac{8}{2} \times \sigma c \times \frac{40}{2}$$

$\therefore \sigma c = 333.654\ N/mm^2$

Since induced crushing stress (σc) <
Allowable crusting stress ((σc) allowable)

$\therefore$ The design of key is safe from the point of the view of crushing:

Chapter 3

HOUSE BEARING

DESIGN ANALYSIS

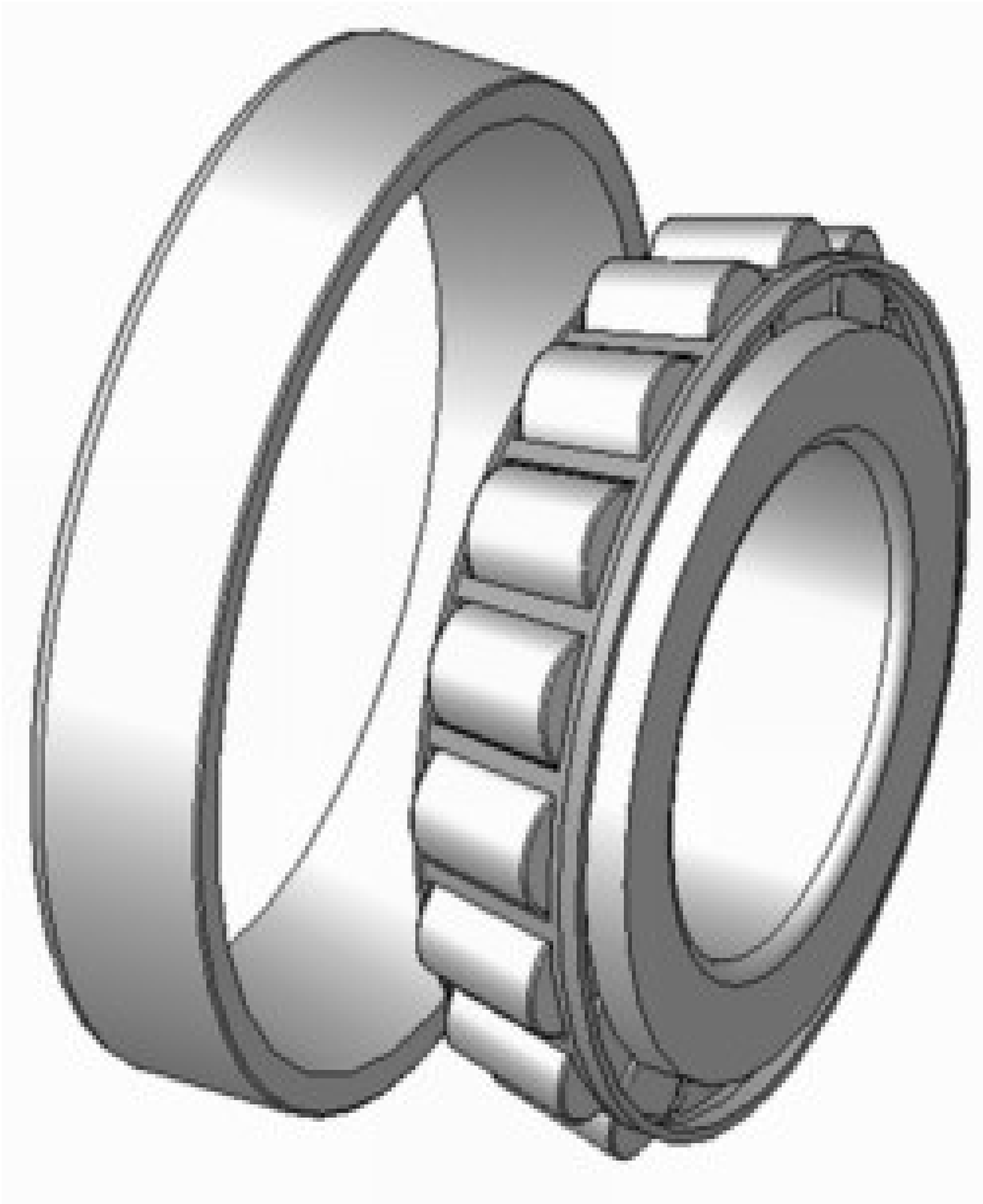


Fig 3.1 Tapered Roller Bearing

HOUSE BEARING :

Worm Bearing:

We will be using "roller bearings" which consists of an inner race which is mounted on the shaft and an outer race which is carries by the housing or casing. In rollers bearings the contact between the bearing surfaces rolling instead of sliding as in sliding contact bearings.

Advantages of Rolling contact bearing Over Sliding contact bearing.

1) Low starting and running friction except at very high speeds. This is an outstanding advantage.
2) Ability of withstand momentary shock loads.
3) Accuracy of shaft alignment.
4) Low cost of maintenance, as no lubrication is required which in service
5) Small overall dimensions.
6) Easy to mount and erect.
7) Cleanliness.

Dynamic Load Rating for Rolling Contact Bearings under variable Loads :

The approximate rating (or service) life of roller bearings is based on the fundamental equation:-

$$L = \left(\frac{C}{W_E}\right)^K \times 10^6 \text{ revolutions.}$$

where L = Rating Life

C = Basic dynamic Load rating.

W_E = Equivalent dynamic Load

K = a constant $= 3$, for ball bearing

$= \frac{10}{3}$ for roller bearing.

The relationship between the life in revolution (L) And the life in working house (L_H) is given by:

$L = 60\ N.\ L_H$ revolutions.

Where $N = 1440$ r.p.m.

And for machines used for continuous operation (24hrs per day), L_H varies from 40,000 to 60,000 hrs

Ref –6 page-425, Recommended bearing life

Let us Consider the value of L_H to be 50,000 (i.e) L_H = 50,000 hr.

So we have:-

$$L = \left(\frac{C}{W_E}\right)^K \times 10^6 \text{ revolutions.}$$

$$\rightarrow \left(\frac{C}{W_E}\right)^K \times 10^6 = 60\,N\,L_H$$

$$L \quad \left(\frac{C}{W_E}\right)^{\frac{10}{3}} = \frac{60 \times 1440 \times 50{,}000}{10^6} = 4320$$

$$\therefore \frac{C}{W_E} = (4320)^{\frac{3}{10}} = 12.32097$$

Equivalent Bearing Load:

In actual application, the force acting on the bearing has 2 components redial and thrust. It is there fore necessary to convert the two components acting on the bearing into a single hypothetical load, fulfilling the conditions applied to the dynamic load carrying capacity. Then the hypothetical load can be compared with the dynamic load capacity. The equivalent dynamic load is defined as the constant radial load in radial bearings (or thrust load in thrust bearings) which if applied to the bearing would give same life as that which the bearing will attain under actual condition of forces.
The value of x and Y can be found out from table 18.5 (Factors x and Y for radial bearings) Choosing Single – row bearings and the contact angle Lo=15°

For Tapered roller bearings:

We have, the constant 'e' which represents a transition point is given by:-
$e = 1.5 \tan \alpha o = 1.5 \tan 15° = 0.401923$
we have the ration = $\frac{W_A}{W_R} = 7.7334$ (already) (calculated)

Ref:7 page 544-545 Table 18.5 Factor X and Y for redial bearing

Since $\frac{W_A}{W_R} > e$

So for single –row Bearings we have X = 0.40, Y = 0.4cot αo = 0.4cot 15°

Y=1.4928

The expression for equivalent dynamic 1oad is:

$W = (X.V.W_R + Y.W_A)\,K_S$

$\rightarrow W = [\,(0.40 \times 1 \times 1577.274) + (1.4928 \times 12197.734)]\,x2.5$

$W = [\,630.9096 + 18{,}208\,.\,77732] \times 2.5$

$\therefore W = 47{,}099.21729\ N = 47.09921729\ KN$

.

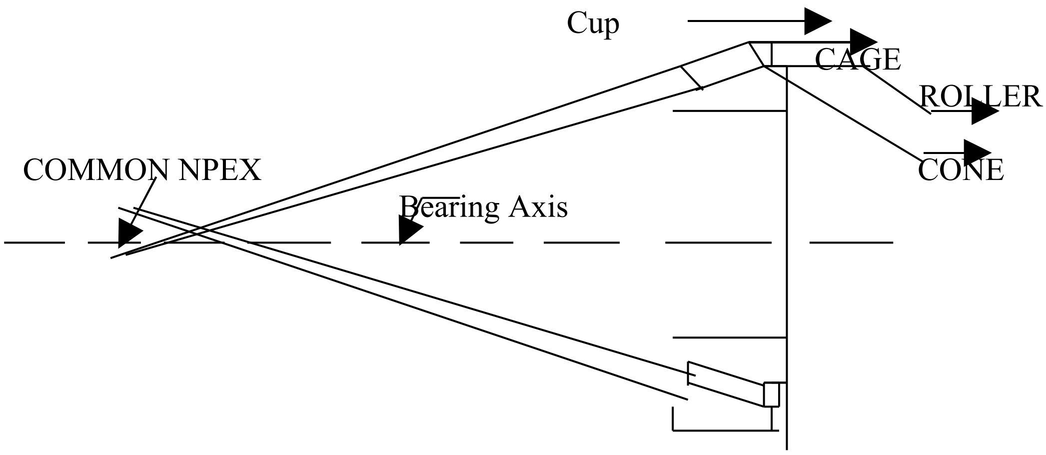


Fig 3.2 Nomenclature of Tapered Roller Bearing

We have already Calculated:

$\frac{C}{W} = 12.32097$

$\therefore C = W \times 12.32097 = (47.09921729 \times 12.32097$

$\longrightarrow C = 580.\ 30804$ KN = 5,80,308.04 N.

The recommended dimensions for Taper roller bearings, single row are (considering next higher value of dynamic load rating).

Principal dimensions:

D = 180 mm , D=320mm, T=57mm

Ref: SKF bearing Catalogue (From interment) (www. Skf.com)

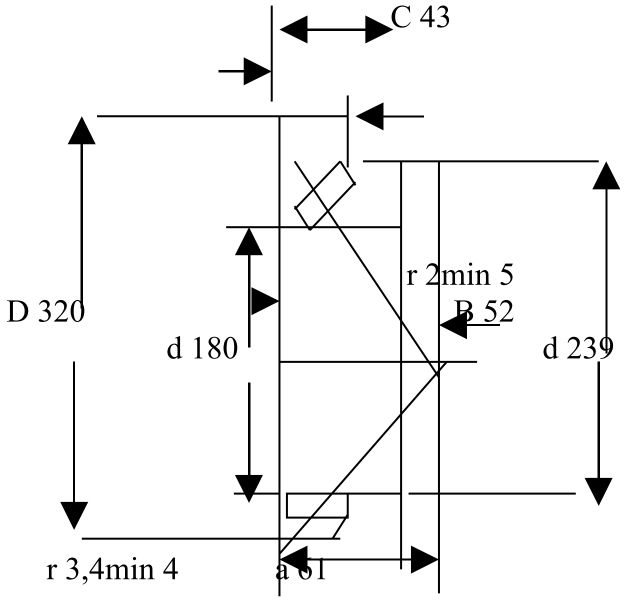
Basic load ratings :

Dynamic load = C = 583 KN
Static load = C° = 815 KN
Fatigue load limit =P_u = 80 KN

Speed Ratings:

-
Reference speed = 1500r.p.m.
Limiting speed = 2000 r.p.m.
Mass = 20 kg
Designation $\longrightarrow$ 30236 J2

T 57



all dimensions are in millimeters.

Taper Roller Bearing :

d = inner diameter of bearing
D = Outer diameter of bearing.
B = Total axial width of bearing.

Fig 3.3 Dimensions of Tapered Roller Bearing

WORM GEAR BEARING:

The approximate rating life of a roller bearing is given by :

$$L = \left(\frac{C}{W_e}\right)^K \times 10^6 \text{ revolutions.}$$

where $K = \frac{10}{3}$ for roller bearing

But we have already seen that

$L = 60\ N.\ L_H$ revolutions

Where $L_H = 50{,}000$ hrs., $N = \frac{1440}{25}$ r.p.m. $= 57.6$ r.p.m. (Reduction ratio)

$$L = \left(\frac{C}{W_e}\right)^K \times 10^6 \text{ revolutions.}$$

$$\left(\frac{C}{W_e}\right)^K \times 10^6 = L$$

$$\left(\frac{C}{W_e}\right)^{\frac{10}{3}} = \frac{60 \times 57.6 \times 50{,}000}{10^6} = 172.8$$

$$\frac{C}{W_e} = (172.8)^{\frac{3}{10}} = 4.690975$$

The equivalent dynamic load is given by:

$W = (X.V.W_R + Y.W_A)\ K_S$

Radial load on worm gear Bearing $= W_R = \frac{Fr}{2} = \frac{3154.5485}{2}$

$W_R = 1577.275$ N.

Axial load on worm gear Bearing $= W_R = 2653.7589$ N

As in previous case, here also V=1, and Service factor $= K_S = 2.5$
For heavy shock load.

Considering αo = Contact angle =15°

We have : $e = 1.5 \tan \alpha o = 1.5.\tan 15°$

$e = 0.401923$

Now: $\frac{W_A}{W_R} = \frac{2653.7589}{1577.274} = 1.68249 > e$

$$\boxed{\text{Since } \frac{W_A}{W_R} > e}$$

hence for tapered roller single row bearing

We have :
$X=0.40$, $Y = 0.4 \cot \alpha o = 0.4 \cot 15°$
$Y = 1.4928$

Ref: 7. Page.54. Factor X and Y for radial bearing

Equivalent Dynamic load is given by :

$W = (X .V. W_R +Y.W_A) K_S$
$W = [(0.40 \times 1 \ \ 1577.274) + (1.4928 \times 2653.7589)]\times 2.5$
$W = 11,481.10221$ N

We have already got:

$\frac{C}{W} = 4.690975$

$\therefore$ Basic Dynamic load Rating = C= w x 4.690975

$\rightarrow$ C = 11, 481.10221 x 4.690975
$\rightarrow$ C = 53,857.56346 N
$\rightarrow$ C = 53.85756346 KN

The recommended dimensions for Taper roller bearings (single row are):

Ref: 2 page 14.30 Tble14.20 Taper Roller Bearings.

d = 55mm, D=100mm, B=25mm, T=26.75mm
C = 21mm, r=2.5mm, r=0.8mm, a=22mm
Basic Capacity (Kg)
Static = C_o= 6300kg.
Dynamic = C = 6700 kg
Max permissible speed (r.p.m.) = 4000r.p.m

GEAR, HOUSING:

Gear Housing are essential for the reduction gear as it serves as a base for mounting the gears and ensures proper relative alignment of their axes, support for the bearing and bearing seals, acts as dust proof providing cogent and as an oil both for the gears.

Advantages of gear housing over open drives:

1) _Abundant lubrication minimizes wear and tear and contributes to high efficiency of the reduction unit.
2) The gear housing promote and provides operational safety.

Housing material:

The material Chosen for housing is cast iron

Design Procedure:

As the bearing thickness is found to be 26.75mm, hence the housing wall thickness can be safely taken as 30mm

$\therefore$ housing wall thickness = t_h =30mm

Bearing Cover Bolt:

The bolt material Chosen is C-45

For which %C= 0.40 –0.05, % Mm=0.60-0.90

Ref: 2 . page 1.6 &1.7 Properties and uses of Carbonated

For C-45, the with mate Tensile strength =583 N/mm^2

Factor of safety =10 for shock reputed in one direction

Ref: 5 page31 Table2.8

$\therefore$ Allowable Tensile stress = f_t

$$\Rightarrow F_t = \frac{\text{ultimate tensile strength of C-45}}{\text{Factor of safety}}$$

$$\Rightarrow F_t = \frac{583\ N/mm^2}{10} = 58.3\ N/mm^2$$

The bearing cover Bolt may fail in tension due to the axial force on the worm shaft

The axial fore on the worm = Faw = 12197.734 N

$$\therefore \frac{\pi}{4} \times d_b^2 \times f_t \times n = Faw$$

where n = no. of bolts = 6

$$\Rightarrow \frac{\pi}{4} \times d_b^2 \times 58.3 \times 6 = 12197.734$$

$$\Rightarrow d_b = 6.6632mm \text{ (calculated corediameter)}$$

$\therefore$ Standard dimension of bolt chosen:-

Coarse series $\longrightarrow$ M10

Pitch = 1.5mm

Major or Nominal diameter of bolt =10.000mm.

Effective or pitch diameter of bolt = d_b =9.026 mm

Ref:4 page 344 table 11.1 Design Dimensions of screw thread, boats and nuts.

Minor or Core diameter = d_c = 8.160mm
Depth of Thread (bolt) = 0.920mm
Stress Area = 58.3mm^2
Material of bearing cover is taken as gray cast iron

Foundation Bolt:

These are used to fix heavy machinery to concrete foundation. The common method of fixing a foundation bolt is to suspend it in the hole kept in the foundation, which is quite large as compared to the hold, and then filling the hole with a fine grout consisting of equal parts of sand and cement. The grout is allowed to set, keeping the bolt vertical. Sometimes matter lead of sulphur is also used which sets in the foundation within a few minutes variously types of foundation bolt used are → Rag bolt, curved Bolt, Lewis Bolt, Cotter Bolt, Hoop Bolt, Squares headed Bolt.

The foundation belt may fail in bending due to axial and tangential force of worm.

We have : F_{aw} = 12197.734 N
F_{tw} = 2653.7589 N

∴ Resultant force = $F = \sqrt{F_{aw}^2 + F_{tw}^2}$

$F = \sqrt{(12197.734)^2 + (2653.7589)^2}$

→ F = 12,483.07458 N

Material for foundation bolt is 40Ni 2 Cr1 Mo28 due to its toughness, hardness, high strength, Ref.2 co car resistance, better machining quality

Tensile strength = F_t = 1320 kg f / cm^2
Bending stress = f_b = 920 Kg f/cm^2
Shear stress = f_S = 550 kg f/cm^2

Ref: 2 Page1.13 Table 1-10 mechanical properties of Alloy steel forgings for general industrial use.

Chapter 5

GEAR AND PINION

DESIGN ANALYSIS

<u>Gear and Pinion</u>

The centre distance between gear and pinion is chosen as 50cm.

The shaft which is integral with the pinion revolves at a speed $\frac{1440}{25}$ r.p.m = 57.6 r.p.m.

<u>We choose 20° full depth involute spur gears</u>:

The 20^0 pressure angle system with full depth in volute teeth is widely used in practice. It is also recommended by Bureau of Indian Standards.

<u>The 20^0 pressure angle system has the following advantages</u>:

1) It reduces the risk of undercutting (i.e.) the danger of interference is less.
2) The tooth is stronger with a higher load carrying capacity.
3) It has greater length of contact.
4) It gives larger radius of curvature.

Since in a multistage gear box consisting of two or more stages, the velocity ratio at each stage should not exceed 6:1, Therefore for the spur gear drive, gear reduction ratio is adopted as 6:1 Centre distance = $\frac{D_p + D_G}{2} = 50_{cm}$

and $\frac{D_G}{D_p} = \frac{N_p}{N_G} = 6$

<u>Solving we get :</u>

Diameter of pinion = Dp = 14.285cm
Diameter of gear = D_G = 6 x D_p = 85.714_{cm}
Pitch line velocity of pinion = V_m
Where $V_m = \pi D_p N_p = \pi \times \frac{(0.14285_m) \times 57.6}{60}$
$V_m = 0.430825_{\ m} = 25.849_{\ m/min}$.

<u>Design Load :</u>

Design load for gear tooth = F_d where
$F_d = \frac{C_o \times C_R}{C_v \times C_L} \times F_L$
Let us choose overload factor = C_o = 1.25 x 1.25
where (1.25) is for 24 hours per day operation of reciprocating feeder under moderate shock condition. Ref:5 Table-22.7 Page 824..

$\therefore C_o$ = 1.25 x 1.25 = 1.5625
Reliability factor = C_R = 1.25
Life factor = C_L = 1
For ordinary cut commercial gears, velocity factor (C_v) is given by:

$$C_v = \frac{183}{183 + Vm} = \frac{183}{183 + (0.430825 \times 60)} = 0.87623$$

(Vm in m/min)

This formula is valid for gears
(Operating with pitch line velocity up to 460 m/min)

Transmitted load F_t which is assumed to act at the pitch radius of the gear is given by :

$$F_t = \frac{4500 \times H.P}{V_{m.}}$$

Power Transmitted to the pinion

$$= (2\pi N_p) \times T_g$$

where T_g = Torque Transmitted by the worm gear

$$= \left(\frac{2\pi \times 57.6}{60}\right) \times 3049.4335$$

$$= 18393.749 \text{ w.}$$

$\therefore$ HP transmitted to the pinion

$$= \frac{18393.749}{745.8} = 24.6631 \text{ h.p.}$$

Now the load Transmitted by the feeder can be calculated alternatively as follows:

Tonnes/ hour of the feeder = 500

(Capacity of the coal feeder)

$$\text{R.P.M. of the gear} = \frac{\text{R.P.M. of pinion}}{6}$$

$$= \frac{57.6}{6} = 9.6 \text{ r.p.m}$$

Hence strokes /minute = 9.6

$$\text{Tonnes per minute} = \frac{500}{60} = 8.333 \text{ tonnes/min.}$$

$$\therefore \text{Tonnes per stroke} = \frac{8.333}{9.6} = 0.86805 \text{ tonne}$$

$$= 868.055 \text{ kg.}$$

$$\therefore \text{Transmitted load} = F_t = \frac{H.P. \times 4500}{V_m}$$

$$F_t = \frac{24.6631 \times 4500}{(0.430825 \times 60)} = 4293.466 \text{ kg.}$$

So to be on the safer side, we consider a higher value for Transmitted load

So we take $F_t = 4295$ kg

$$\therefore \text{design load} = F_d = \frac{Co \times C_R}{Cv \times C_L} \times F_L$$

$$F_d = \frac{1.505 \times 1.25}{0.87633 \times 1} \times 4295$$

$$F_d = 9573.595 \text{ kg}$$

BeamStrength :

The analysis of binding stresses in gear tooth was done by Mr. Wilfred Lewis in his paper, " The investigation of the strength of gear tooth" submitted Engineers Club of Philadelphia is

1892. In the Lewis analysing the gear tooth is treated as a cantilever bean. On which a normal force Fn is acting between the tooth surfaces which is can be resolved into 2 components.The radial component (Fr) and the tangential component (F_L) which causes the bending moment about the base of tooth.

The Lewis equation is based on the following assumptions :
1) The effect of radial component 'Fr' which induces compressive stress is neglected.
2) It is assumed that the tangential component 'F_L' is uniformly distributed over the face width of gear. This is possible when the gears are rigid and accurately machined.
3) The effect of stress concentration is neglected .
4) It is assumed that at any time only one pair of teeth is in contact and takes the total load.

Material for pinion :

The pinion material is selected as the Nickel Chromium Molybdenum steel 40 Ni2Cr 1 M028 due to its high strength, resistance to wear, hardness and good machining quality.

Ref:4
Page 981
Table-28.3
Properties of commonly used gear materials.

Minimum ultimate Tensile strength = 900 N/mn^2

$$n_p = \frac{D_p}{m}$$

$$18 = \frac{14.285}{m} \text{ cm}$$

$$\therefore m = \frac{14.285}{18} = 0.793611 \text{ cm} = 7.936\text{mm}$$

So the standard module chosen = m = 8mm

Ref:2
Page 15.8
Table-15.3
Recommended modules by many European Countries.

$\therefore$ ng= Number of teeth on gear

ng= n_p x reduction ratio

$\Rightarrow$ ng = 18 x 6 = 108 teeth

The exact value of the Lewis form factors 'Y' is determined by the relation :-

$$Y = \pi\left(0.154 - \frac{0.912}{n}\right)$$

$\Rightarrow$ for 20^0 in volute full depth system.

For pinion $\rightarrow$ n = n_p

$$\therefore Y = \pi\left(0.154 - \frac{0.912}{18}\right) = 0.3246$$

For gear $\rightarrow$ n = ng

$$\therefore Y = \pi\left(0.154 - \frac{0.912}{108}\right) = 0.4572$$

Now we apply the bean strength equation for pinion and gear to find the tooth face width.

$$F_d = \frac{6_w \times b \times m \times y}{k}$$

$$93{,}916.96695 = \frac{375 \times b \times 8 \times 0.29}{1.5}$$

$\rightarrow$ b = face width = 161.9258 mm = 16.19258 cm
b $\simeq$ 16.2 cm

Gear Tooth Proportions

Ref:2
Page 15.12
Standard Tooth proportions.
Page 15.12

Addendum = a = m = 8 mm.
Dedendum = d = 1.25m = 10 mm
Clearance = 0.25m = 0.25 x 8 mm = 2 mm.
Total depth = a + d = 8 mm + 10 mm =18 mm
Working depth = (a + d) – Clearance
= (m + 1.25 m) – 0.25 m
= 2 m = 2 x 8 = 16 mm.
Circular pitch = P_c = II m = 25.12mm

Ref:5
Table 22.4
page 810

For m = 8 mm and pitch line velocity Vm <8mps
Back lasb is 0.22mm

Tooth thickness = 1.5708 m = 1.5708 x 8 mm = 12.57 mm.

Tooth space = 1.5708 m = 12.57 mm.
Diameter of Addendum Circle for pinion
= $D_p + 2_a = 142.85_{mm} + 16_{mm}$
= $158.85_{mm} \simeq 159_{mm}$
Diameter of Addendum Circle for gear
= Dg + 2a = 857.14mm + 16mm
= 873.14mm

Diameter of dedendum circle for pinion
= Dp --- 2d =142.85 ----- (2 x 10)
= 122.85mm

Diameter of Dedendum Circle for gear
= Dp – 2d = 857.14 –(2 x10)
= 837.14mm

Check for wear,
Material for Gear:

Ref:8
Table 5
Page 8.5

55cr 70 chosen as the gear material due to following characteristics.

1. It increases the elastic limit.
2. It increases hardness, corrosion resistance and also chilling effect.

Minimum ultimate tensile strength = 981 n/mm²

The ratio of ultimate tensile strength to the yield strength is taken as 1.25.

$$\text{Yield stress} = \frac{981}{1.25} = 784.8 \text{ mm}^2 \simeq 785 \text{ N/ mm}^2$$

Taking factor of pinion = 1.5 (for bending)

$$\text{Allowable or design bending stress} = \frac{785}{1.5} = 523.323 \text{ N/ mm}^2$$

Strength factor of gear

$= (\sigma_w)$ pinion x y = 375 x 0.3246

= 121.725

Strength factor of gear=

$=(\sigma_w)$ gear x y

= 523.333 x 0.4572

239.266

As the strength factor pinion is smaller than that of gear, hence the pinion is weaker .
The maximum or the limiting load for satisfactory wear of gear teeth, is obtained by using the
Buckingham equation :

Ww = Dp. b.Q.k

Where Ww = Maximum or limiting load for wear in New tons

Dp= pitch circle diameter in mm

b= face width of the pinion

$$Q = \text{Ratio factor} = \frac{2 \text{ x} V.R}{V.R+1} \text{ for gear external}$$

$$= \frac{2\text{x}6}{6+1} = 1.714$$

Where V.R = velocity ratio of spur gear drive=6

$$K = \text{load stress factor} = \frac{(\sigma_{es})^2 \sin\text{Ø}}{1.4}\left(\frac{1}{Ep} + \frac{1}{Eg}\right)$$

σ_{es} surface endurance limit in N/mm²

=(2.75 BHN -70)

= (2.75 x444-70) = 1150.91 N/mm²

Ref:2
Page 15.3

Where BHN = 444 for pinion material 40Ni2Crl MO 28 steel

Ref:5
Table 22.6
Page 818

Eg =Ep = 2.1 x 105 N/mm²

SinØ = Sin 20º= 0.342

$$K = \frac{(1150.91) \text{ x } 0.342}{1.4}\left(\frac{1}{2.1\text{x}105} + \frac{1}{2.1 \text{ x } 105}\right)$$

= 3.08188

Ww = Dp x b x Q x k

=142.8 x162 x1.714 x3.08188

=1,22,242.762 N.

Pd =design load = 9573.595kg = 93,916. 96 N

Since where load (W_w) > Design loan (F_d), hence th pinion is safe and will not fail due to wear

Chapter 6

GEAR DRIVE

DESIGN ANALYSIS

DESIGN OF GEAR DRIVE :

PINION: Since the diameter of the pinion is between the range Dp<=14.75 m +6 ,we have Dp=14.3cm and m=0.8cm ,satisfy the above condition .Hence we have to make the pinion solid with uniform thickness equal to the face width .The diameter of the pinion shaft is same as worm wheel shaft.

<u>GEAR</u> : The diameter of the gear is 85.8cm .Hence 6 arms are provided . The arms of gear are designed by assuming them as cantilevers fixed at the hub end and loaded at the pitch line , with the load equally distributed to all arms .

Bending moment on each arm will be

$$M = \frac{F_o \times R_g}{n_a}$$

where $F_g = \dfrac{F_t \times C_o}{C_v}$

$F_t = 26090$ N

Assuming $C_o = 1.56$ and $C_v = 0.88$ for medium shock loads for more than 16 hours operation .

$$\therefore F_o = \frac{2608 \times 1.56}{0.88}$$

$$= 46250 \text{ N}$$

$$M = \frac{F_o \times R_g}{n_a}$$

$$= \frac{46250 \times 85.8}{6 \times 2}$$

$$= 33069 \text{ N.}$$

Elliptical section for arm is taken because of its lightness , material saving and reduces the windage losses and more strength , stiffness .

Since the stresses in the arms are of alternating type , therefore for medium shocks conditions , factor of safety may be taken as 9. Therefore , design stresses in bending f_b = 1000

□□ = 111.12 N / mm^2

9

Now the section modulus , $Z = \dfrac{\pi \times b^3}{64}$

b = major axis of the hub = 2h

h = minor axis of the hub .

Hence , $M = Z \times f_b$

$$333069 = \frac{\pi \times b^3 \times 111.12}{64}$$

$b = 8.46$ cm

$h = 4.23$ cm

Now , the major axis of the rim = b - taper

Usually the arms are tapered towards the rim about 6.5 cm / meter length of the arm .

$\therefore$ major axis at the rim = b - taper

$$= 8.46 - \frac{6.5 \times 42.9}{100}$$

$$= 5.7 \text{ cm}$$

Minor axis at the rim end = 2.85 cm

RIM :

Rim thickness is given as

$$t_r = m \times \sqrt[3]{\frac{n}{2\ n_g}}$$

Now n_g = 108 and n_a = no . of arms = 6.

m = 0.8 cm

$$t_r = \frac{0.8 \times \sqrt[3]{108}}{2 \times 6} = 1.66 \text{ cm} = 16.6 \text{ mm}$$

t_r = 17 cm

depth of circumferential rib , h_r = t_r = 17 mm .

thickness of rib , w= thickness of arm at this end .

SHAFT :

The diameter of the shaft can be found out by designing it under the combined effect of torque transmitted by it and its bending .the bending moment acting on the shaft will be due to the weight of the gear or pinion and due to normal force acting between the tooth surfaces .

Torque on the shaft $T = F_o \times R_g$

$$= 46250 \times 85.2/2 = 1984125 \text{ N}$$

Now normal force $F_n = \frac{F_o}{\cos \alpha} = \frac{46250}{\cos 20^0}$

$$= 49218.3 \text{ N}$$

The weight of the gear W is given by

$W = K . N_g . f_g . m^2$

K = 0.118 for spur gear

$W = 0.118 \times 108 \times 5.8 \times 0.8^2$

= 47.3 kg

W = 50 kg

Resultant loading on the gear will be

$R_1 = \sqrt{W^2 + F_n^2 + 2 . W . F_n \cos \alpha}$

$= \sqrt{50^2 + (49218.3)^2 + 2. 50 .49218.3 \cos 20^0}$

= 49688.4 N.

Taking the distance between right side bearing and gear and left side bearing and gear are 75 cm and 25 cm respectively .

$$\text{Reaction } R_a = \frac{49688.4 \times 75}{100} = 37266.3 \text{ N}$$

Reaction R_b = 49688.4 - 37266.3
= 12422.1 N

Maximum bending moment = 37266.3 × 25 = 93165.75 N mm

$$\text{Torque transmitted by gear shaft } T = F_o \times R_g = 46250 \times \frac{85.8}{2}$$

= 198412.5 N mm

Equivalent torque $T_e = ε\ 198412.5^2 + 93165.75^2$
= 219197.12 N mm

Material for gear shaft :

C-35 is choosen as the shaft material because of sufficient high sytrenght b, ability to withstand heat ,case hardening treatment and it lessen the effect of stress concentration . It also increases the wear resistance .

The ultimate strength in tension for C-35 is 500 - 600 N mm^2 . As we know shear strength should not exceed 18% of the ultimate stenght of the material , 15% of the ultimate strength ,

f_s = 0.15 × 500 = 75 N mm^2

equating the strength of the shaft with equivalent torque

$T_e = \pi/16 \times d^3 \times f_s$

=> 219197.12 = $\pi/16 \times d^3 \times 75$

∴ d = 11.41 cm ψ 11.5 cm

Hub diameter d_h = 1.8 d
= 1.8 × 11.5
= 20.7 cm

Length of hub = 1.25 d
L_h = 1.25 × 11.5
= 14.375 cm

Design of key & key ways (for gear shaft)

The design of key is based on the shaft diameter .

Shaft diameter cross-section of the key is

Width = b_1 = 32 mm

Height = h_1 = 18 mm

Keyway depth in shaft = 11 mm = t_1

Keyway depth in hub = 7.4 mm = t_2

Tolerance on key way depth for t_1 = + 0.3 mm

Tolerance on keyway depth for t_2 = + 0.2 mm

Chamfer radius = r_1 = 0.8 mm (maximum)
= 0.6 mm (minimum)

Keyway radius = R_2 = 0.6 mm (maximum)

Material for key is same as material of shaft and i.e. C-35 .The shear stress for C-35 is 15% of the ultimate strength = 0.15 × 500 =75 N/mm^2

$\therefore$ $T = L_1 \times b_1 \times f_s \times d/2.$

$198412.5 = L_1 \times 3.2 \times 750 \times 11.5/2$

$\therefore$ L_1 = 14.37 cm ψ 14.5 cm

Check for crushing strength :

$$\text{Crushing strength} = \frac{4 \times T}{d \times L_1 \times h_1} = 2644 \text{ N/mm}^2$$

For rectangular shank key ,

$$\frac{\text{crushing strength}}{\text{Shear strength}} = 3$$

$\therefore$ crushing strength = 3 × 750 = 2250 N/mm^2

Since calculated crushing strength is greater than 225 N/mm^2 ,we have to increase the length of the key .

$\therefore$ In practice , $L_1 = 1.5\, d = 1.5 \times d = 1.5 \times 11.5$ =17.25 cm

$\therefore$ L_1 = 17.5 cm

$$\text{Crushing strength} = \frac{4 \times T}{d \times L_1 \times h_1} = 219.08 \text{ N / mm}^2$$

Whichever is less than 2250 N/ mm^2 . Hence the design is safe .

$\therefore$ dimension of parallel key is 32 × 18 × 17.5 mm

Design of key & key ways (for pinion shaft)

Design of key is based on the pinion shaft diameter
Shaft diameter = 60 mm
For the standard cross-section of the lkey is
Width b_1 = 18 mm
Height h_1 = 11 mm
Keyway depth in shaft t_1 = 7mm
Keyway depth in hub t_2 = 4.4 mm
Tolerance for t_1 = +0.2mm
Tolerance for t_2 = + 0.2 mm
Chamfer radius = r_1 = 0.55 mm(max)
= 0.40 mm(min)
Keyway radius r_2 (max) = 0.40 mm
Taking the same material as that of shaft

f_s = 42 N/mm₂
T_{max} = $l_1 \times b_1 \times f_s \times d/2$.
$14788.85 = l_1 \times 1.8 \times 42 \times 6/2.$

l_1 = 6.52 cm

Check for crushing strength :

$$f_c = \text{crushing stress} = \frac{4 \times T}{d \cdot l \cdot h}$$

$$= \frac{4 \cdot 14788.85}{6.0 \times 6.32 \times 1.1} = 137.468 \text{ N/cm}^2$$

$f_c = 3 \times f_s = 3 \times 42 = 126$ N/mm^2

The calculated f_c is greater than 126 N/mm^2

Hence design is safe .

So length of key should be increased .
Taking $l_1 = 1.5 \ d = 1.5 \times 6.0 = 9$cm

$$f_c = \frac{4 \times 14788.85}{6.0 \times 9 \times 1.1} = 995.88 \text{ N/mm}^2$$

which is less than 126 N/ mm^2
Hence the design is safe .
The dimension of parallel key is 18 × 11 × 90.0 mm

Chapter 7

PLUMMER BLOCK
AND
BEARING

DESIGN ANALYSIS

PLUMMER BLOCK & BEARING :

It consists of bearing housing and bearing . Since the roller bearings have higher load capacity than ball bearing for a given overall size ,also roller bearings due to their greater area of contact , have greater capacity for radial loads than ball bearing . We select double row roller bearing.

Types of roller : spherical roller is choosen because of the following reasons .

(a) high load capacity .
(b) high tolerance to shock loads
(c) self aligning capacity which is achieved by grinding bthe outer or inner race way .

In the reciprocating feeder ,the bearing is supposed to operate 24 hrs a day . We have bearing life is 50,000 hrs .

The life rating of roller bearing is given by

$L = (C/W_e)^k \times 10^6$ revolutions

For roller bearing K = 10/3.

Also $L = 60\ N.\ L_h$

$N = 9.69$ rpm

$L_h = 50{,}000$ hrs.

$(C/W_e)^k \times 10^6 = 60 \times N \times L_h$

$$(C/W_e)^{10/3} = \frac{60 \times 9.69 \times 50000}{10^6}$$

$\therefore \quad C/W_e = 2.74$

$\therefore \quad L = (2.74)^{10/3} \times 10^6$ revolutions
$= 28.785 \times 10^6$ revolutions

$W_e = (X_r \times W_r) K_s$

$W_r = R_a = 3926.63$

$W_e = (1 \times 3926.63) \times 2$
$W_e = 7853.26$ kg
$C = 2.74 \times 7853.26 = 21{,}517.93$ kg

From SKF catalogue , the recommended dimensions for spherical roller bearing are
Designation - 22224C

d = 115 mm , D = 215 mm , B = 58 mm
E= 143 mm , r = 3.5 mm , C_o = 40,000 N
C = 40,000 N

Since the calculated value of dynamic load C is less than the recommended value . The selected bearing is safe .

Roller dimensions :

Radial space available for the rollers

$$= \frac{D - E}{2} - \frac{E - d}{2} = \frac{215 - 143}{2} - \frac{143 - 120}{2} = 25 \text{ mm}$$

$$\text{No. of rollers} = \frac{25 \times (E + 50/2)}{25} = 24.253$$

Since there are 24.253 rollers without any spacing between them , so we take 22 rollers with spacing i.e. $\frac{(24.25 - 22)\ 25}{22} = 2.56$ mm

No. of rollers = 22 .

PLUMMER BLOCK BEARING FOR GEAR SHAFT :

Cap design : The cap can be looked upon as a beam simply supported at the center line of the bolts and subjected cto uniformly distributed load on the central 215 mm length .

For the inner race diameter of 115 mm ,the width of the cap is 120mm = 12cm

Length of the cap between the bolt center

$= D_o + 2 \times R_b + 2 \times$ body thickness of the block

$= 205 + 2 \times 0.8 + 2 \times 1.0$

$= 241$ mm where D_o = outer race dia of bearing

Length of uniformly distributed load = 215 mm

$\therefore$ Maximum bending moment at the center

$= R_a/2 \times (24.1)/2 - R_a/2 \times 21.5/2$

$= 1963.18 \times 24.1/2 - 1963.18 \times 21.5/2$

$= 13104.22$ N mm where R_a =load transmitted by the gear shaft .

Now , section modulus

$Z = 1/6 \times 12 \times t^2$ where t = thickness of bolt cap.

Allowable bending stress = $\dfrac{\text{ultimate stress}}{\text{Factor of safety}}$

Material : Taking malleable cast iron , the ultimate tensile strength = 250 N / mm^2

Factor of safety = 6

Allowable stress $f_b = \dfrac{250}{6} = 41.66$ N /mm^2

Bending moment $= f_b \times Z = 41.66 \times 1/6 \times 12 \quad t^2$

$\therefore$ t = 3.95 cm

Adopted t = 4 cm

Length of the body in the top portion

$Y = 24.1 + 2 \times 1.5 \times$ bolt diameter

Bolt material : taking carbon steel (C-40) as the bolt material , allowable tensile strength = 135 N/mm^2

Taking the load on each bolt as 1.3 timea its usual share

$\Rightarrow 1.3 \times 3926.36/2 = 4 \times d_c^2 \times 135$

$d_c = 1.55$ cm

$\therefore$ adopt $d_c = 1.6$ cm =16 mm

Hence , $y = 24.1 + 2 \times 1.5 \times 1.6$

$= 28.9$ cm ψ 29 cm

Distance between the oblong hole centres is kept as L = 28.9 + 2 × 4 =36.9 ψ 37cm

Thickness of the base = 3.5 cm =thickness of the cap

Overall length of the body $L_o = 37 + 2 \times 1.6 + 2 \times 1.5 \times 1.6$

$= 45$ cm

Chapter 8

ECCENTRIC DISC

DESIGN ANALYSIS

ECCENTRIC DISC DESIGN :

The function of the eccentric disc is to convert the rotary motion of the gear to reciprocating of the coal tray ., through the tie rod . the important components where the failure is likely to occur are eccentric disc pin and the key .

Considering the pin to fail in double shear , we have :

$F = \pi \times d_3^2 \times f_s \times 2$

The material for the pin is choosen as C-40 .So the allowable shear stress

$= \dfrac{\text{elastic limit stress}}{\text{Factor of safety}}$

$= 1400 /9 \text{ N/mm}^2 = 15.56 \text{ N/mm}^2$

The capacity of coal tray is 4 tons . Hence the force transmitted to the pin through the tie rod = μ w

Where μ = co-efficient of friction between the wheels of the tray and the rails =0.3

$\therefore$ F = 0.3 × 4000

= 1200 kg

$\therefore$ $F = \pi \times d_3^2 \times f_s \times 2$

$1200 = \pi \times d_3^2 \times 155.56 \times 2$

$d = 2.221$ cm

adopted $d_3 = 25$ mm

Material For Eccentric Disc :

Cast iron is choosen as the material for eccentric disc .

The ultimate tensile stress is 140 N/mm^2.

Disc is subjected to completely reversed stress cycles ,fatigue load comes into picture .

$f_e = 0.4 \times f_{ut}$ for each cast iron .

$= 0.4 \times 140$

$= 56$ N/mm^2.

Working endurance limit ,

$f_e = C_f \times C_s \times C_l \times f_e$

assuming $C_f = 1$

$C_s = 0.85$

$C_l = 0.80$

$f_e = 1 \times 0.85 \times 0.80 \times 56 = 380.8$ N/mm^2

for completely stress cycle

$K_f \times f_V$ [design stress

$$f_v [\frac{f_e}{Kf\ (F.S.)}$$

$d_3 = 25$ mm , $b_3 = 55$mm

$d_3 / b_3 = 25 / 55 = 0.454$

$K_t = 2.22$ for $d/b = 0.454$

$\therefore$ $K_f = 1 + q_f (K_t - 1)$

$= 1 + 0.05 (2.22 - 1)$

$= 1.061$

$$f_v = \frac{380.8}{1.61 \times 9} = 3.98 \text{ N/mm}^2 \ \psi \ 4 \text{ N/mm}^2$$

$F = 2 \times t \times d \times f_v$;

$4000 \times 0.3 = 2 \times t \times 2.5 \times 40$

$t = 60$ mm , thickness of the eccentric disc

DESIGN OF PIN :

Design of the pin is based on the bending and shearing .

The material for the pin is choosen as C-40 .

Bending stress $f_b = 120$ N/mm^2

The force transmitted to the pin is W =120 N/mm^2

$\therefore$ Bending moment $M = W \times L$.

$= 120 \times 5 = 600$ N/mm^2

Assuming effective length of pin to be 5 cm

Section modulus $Z = \pi / 32 \times d^3$

and $f_b = \frac{M}{Z}$

$$120 = \frac{600}{\pi / 32 \times d^3}$$

$d = 3.75$ cm

$d = 38$ mm

The ratio d/b =0.45

$b = 8.5$ cm

on comparing shearing and bending , the required diameter d =38mm and b=85mm.

NOMENCLATURE USED IN THE DESIGN :

SL NO.	DESIGN PARAMETER	SYMBOL	VALUES(in mm)
1.	Tyre coupling		
1.	Outer diameter of tyre	A	277
2.	Hub diameter	B	145
3.	Outer diameter of flange	C	225
4.	Hub length	D	205
5.	Pressure plate thickness	H	8
6.	Thickness of tyre	I	10
7.	Distance between pressure plates	G	53
8.	Thickness of flange	T_f	18.5
2.	Worm reducer		
1.	Centre distance	C_w	315
2.	Lead angle	λ	18.88
3.	Normal lead angle	L_n	59.45
4.	Axial lead	L	62.83
5.	Axial pitch of worm thread	P_a	31.415
6.	Module	m	10
7.	Number of worm starts	Z_1	2
8.	Number of teeth on worm gear	n_g	50
9.	Pitch circle diameter of worm	d_p	108.38

10.	Addendum circle diameter of worm	d_a	128.38
11.	Dedendum circle diameter of worm	d_d	84.38
12.	Length of threaded portion of worm	l	172.63
13.	Pitch circle diameter of worm wheel	D_p	500
14.	Addendum circle diameter of worm wheel	D_a	520
15.	Dedendum circle diameter of worm wheel	D_d	476
16.	Outside diameter of worm wheel	D_o	535
17.	Rim width of worm wheel	B_r	96
18.	Face width of worm wheel	f	86
19.	Diameter of worm shaft	d_w	30
20.	Diameter of worm gear shaft	d_g	40
3.	Worm reducer housing & bearing		
1.	Bore diameter of bearing	d	180
2.	Outside diameter	D	320
3.	Pearing width	T	57
4.	Cup length	C_1	24
5.	Cone length	B	52
6.	Housing wall thickness	t_h	30
7.	Bearing cover bolt diameter	d_b	10
8.	Diameter of foundation bolt	d_f	28
9.	Thickness of washer	t_w	3.5
10.	Outside diameter of washer	d_w	59
11.	Diameter of housing cover bolt	d_h	8
4.	Gear and pinion		
1.	Pitch circle diameter of gear	D_g	857.14
2.	Pitch circle diameter of pinion	D_p	142.85
3.	Face width of tooth of gear	f_g	60
4.	Number of teeth on gear	n_g	108
5.	Number of teeth on pinion	n_p	18
6.	Module for gear and pinion	m	8
7.	Rim thickness	t_r	17
8.	Diameter of gear shaft	d	115
9.	Hub diameter	d_h	207
10.	Length of hub	l_n	143.75
11.	Width of key	b_1	32
12.	Height of key	h_1	18
13.	Keyway depth in shaft	t_1	11
14.	Keyway depth in hub	t_2	7.4
15.	Length of key	L1	175
16.	Diameter of pinion	d_p	60
17.	Width of key	b_1	18
18.	Height of key	h_1	11
19.	Length of key	l_1	65.2
5.	Plummer block and bearing		
1.	Inner diameter of bearing	d	115
2.	Inner race diameter of bearing	E	143
3.	Outer race diameter of bearing	D	215
4.	Width of bearing	B	58
6.	Cap		

1.	Thickness of bolt cap	t	40
2.	Diameter of bolt	d_c	16
3.	Length of body in top portion	y	290
4.	Distance between oblong hole centres	L	370
5.	Overall length of the body	L_o	450
7.	Eccentric disc		
1.	Thickness of eccentric disc	t	60
2.	Diameter of pin	d_s	38

SCOPE FOR IMPROVEMENT

We have taken great pains and made sincere efforts in making our project as realistic and practical as possible for the present day industries. Having said that , we still believe that there is ample scope for improvement for bettering the operation of these reciprocating coal feeders thereby improving the material handling systems in major industries .

First of all, these reciprocating type of feeders are not self-cleaning due to which whenever we are handling a different material and when the contamination is not desirable then the last plateful have to be removed manually .So a suitable arrangement has to be designed and adopted to clean the feeder in order to reduce labour and time of cleaning . Secondly, the feeder is not recommended for highly abrasive material, but its range of use can be effectively increased by improving the surface hardness of plate material .

For reducing the frictional losses, the worm and the worm wheel reduction units may be replaced by chain drive using PIV gearbox using hydraulic link mechanism . Finally for optimization of the entire unit and checking the feasibility of the design with respect to stress consideration ,"Finite Element Analysis " can be adopted .

CONCLUSION

Reciprocating coal feeder is an indispensable part of the material handling industry . It finds its use mainly in the coal mining industries like "MAHANADI COAL FIELDS". Reciprocating coal feeders are used in conjunction with conveyer belts since they guarantee a continuous and controlled feed rate , these are low in cost , its drive mechanism is simple , can handle a wide range of miscellaneous materials including lumps and provides easy assembly and disassembly of various feeder parts .

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Although the reciprocating feeder has many advantages, it has got some limitations too .First of all this type of feeder can't be used for abrasive materials since there is a sliding motion.Secondly the worm reducer assembly is replaced by other reduction units owing to its wear and heat dissipation losses .

So by proper operation of the units, following the standard maintenance and safety instructions one can get a good service for a long time with least repairs without compromising operator's safety.

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