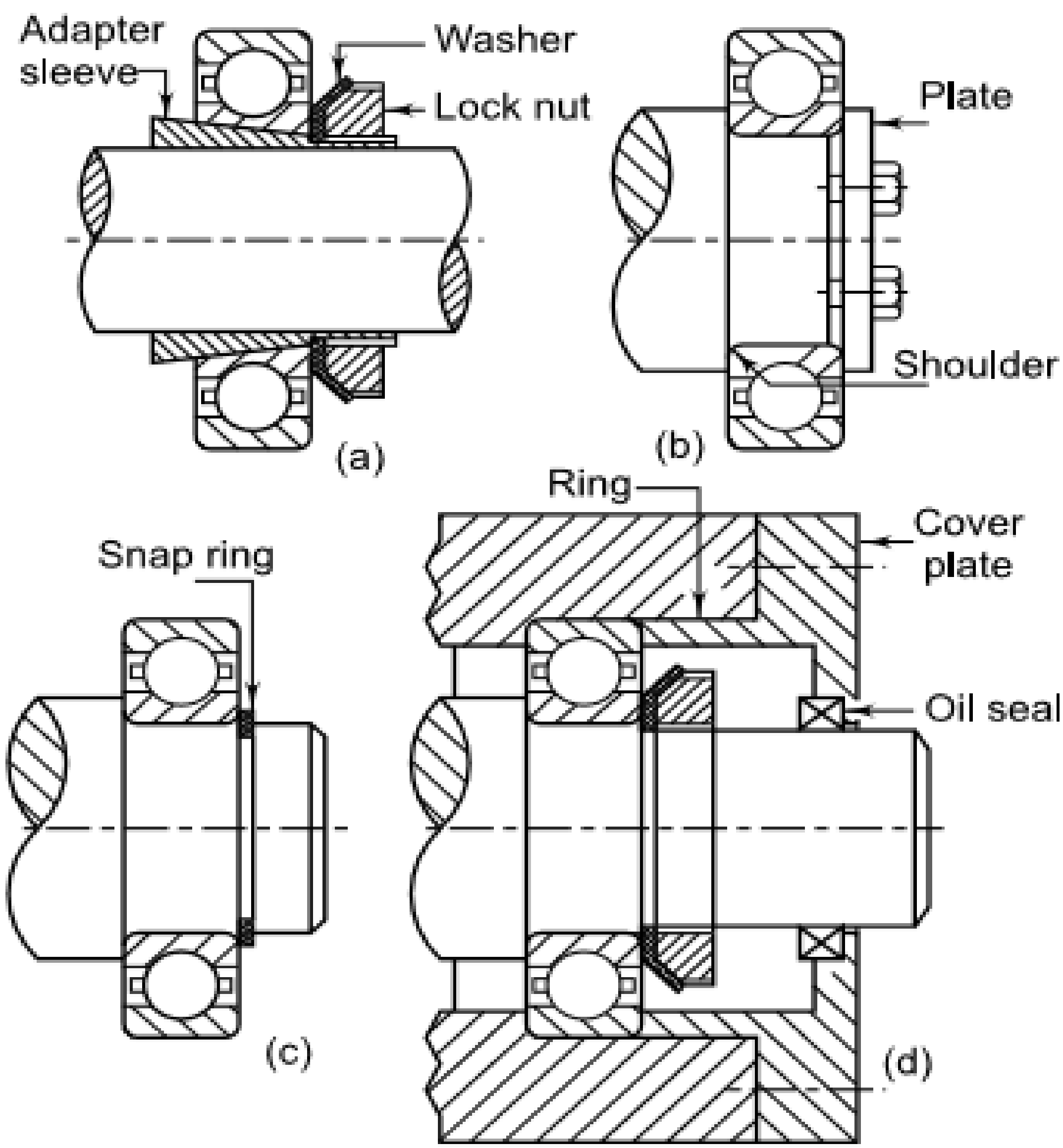
Study of Ball bearing mountings and its selection preloading of bearings.

The inner race of the bearing is fi tted on the shaft by means of an interference fit. It prevents the relative rotation and the corresponding wear between the inner race and the shaft. Tolerances for shaft diameter, corresponding to this type of interference fit, are given in the manufacturer's catalogue. Care should be taken to select the fi t in such a way that it provides sufficient tightness to give a firm mounting and at the same time, it is not too tight a fit to cause deformation of the inner race and destroying clearance between the rolling elements and the races. The outer race is also mounted in the housing with interference fi t, but to a lesser degree of tightness than that of the inner race. Insufficient tightness of the outer race in the housing seat may cause 'creep'. In bearing terminology, creep is slow rotation of the outer race relative to its seating. It is caused when the shaft is subjected to external force that rotates and changes its direction. When two bearings are mounted on the same shaft, the outer race of one of them should be permitted to shift axially to take care of axial deflection of the shaft caused either by thrust load or by the temperature variation. It is necessary to position inner and outer races axially by positive means. There are several methods such as providing shoulders for the shaft or the housing, lock nut, snap ring or cover plates as shown in Fig.. The basic principle is to restrict the displacement of inner as well as outer race in axial direction by positive means. Figure shows the mounting suitable for a long and continuous shaft. It consists of an adapter sleeve, which is provided with a small taper. The bearing is press fi tted on this adapter sleeve. Because of the taper, the displacement of the inner race to the left side is restricted. A washer and lock nut is provided to restrict the displacement of the inner race to the right side. Two methods of restricting the displacement of the inner race are illustrated in Figs. In both the cases, the shaft is provided with a shoulder to restrict the displacement of the inner race to the left side. In Fig. , the displacement of the race to the right side is restricted by a plate, which is bolted to the shaft. In Fig. a snap-ring is used in place of the plate. In Fig. the housing is provided with a shoulder to restrict the displacement of the outer race to the left side. A circular ring of the cover plate restricts the displacement to the right side. The cover plate is bolted to the housing. Commercial oil seal unit is used to prevent the leakage of lubricating oil. The shoulders for the shaft and housing bore have standards dimension, which can be obtained from the manufacturer's catalogue. Shafts and spindles in machine tools and precision equipment should rotate without any play or clearance either in axial or radial direction. This is achieved by preloading the ball bearings. The objective of preloading is to remove the internal clearance usually found in the bearing. Preloading of cylindrical roller bearing is obtained by the following methods:

(i) The roller bearing is mounted on a taper shaft or sleeve, which causes the inner race to expand and remove the radial clearance.

(ii) The outer race is fi tted in the housing bore by an interference fit. It causes the outer race to contract and remove the radial clearance.



Ball bearings, such as angular contact bearing, are preloaded by axial force by tightening the lock nut during the assembly. It is essential to use the correct method of mounting and to observe cleanliness if the bearing is to function with satisfaction and achieve the required life. The precautions to be taken during the mounting operation are as follows:

(i) Mounting should be carried out in a dust free and dry environment. Machines which produce metal particles, chips or sawdust should not be located in the vicinity of the mounting operation.

(ii) Before assembly, the shaft and the housing bore should be inspected. The burrs on the shaft and the shoulders should be removed. The accuracy of the form and dimensions of the shaft and bearing seat in the housing should be inspected.

(iii) The bearing should not be taken out from its package until before it is assembled. The rust-inhibiting compound on the bearing should not be wiped except on the outer diameter and bore surface. These inner and outer surfaces are cleaned with white spirit and wiped with clean cloth.

(iv) Small bearings are mounted on the shaft with the help of a small piece of tube or ring. Blows are applied by means of a hammer on this tube or ring. Direct blows should never be applied to the bearing surface, otherwise the race or the cage may get damaged. The tube or the metallic ring is placed against the inner race and the blows are applied with an ordinary hammer all around the periphery of the ring.

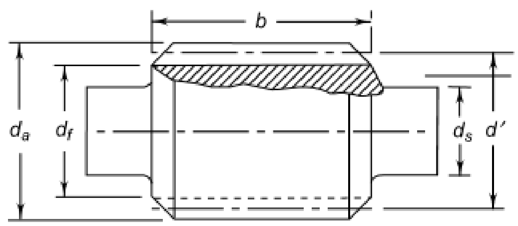
(v) Medium size bearings are mounted on the shaft by pressing the tube or the metallic ring by means of a hydraulic or mechanical press. Large size bearing is mounted by heating it to 80° to 90°C above the ambient temperature by induction heating and then shrinking it on the shaft. The bearing should never be heated by direct fl ame. The interference fi t between the outer race and the housing is obtained by similar methods, viz., by applying hammer blows on a metallic ring or tube which is in contact with the outer race or by using hydraulic or mechanical press or by heating the housing.

Construction of Gear

Depending on the purpose and the size, there are different constructions for gears. These constructions are broadly classifi ed into the following three groups: (i) Small size gears; (ii) Medium size gears; and (iii) gears with large diameter. In this article, we will consider the salient features of these constructions.

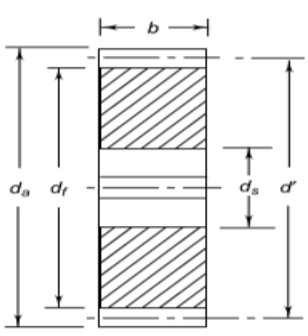
(i) Small Sized Gears :

A pinion with root diameter near to the required diameter of the shaft is made integral with the shaft. This type of construction is shown in Fig.. The rule of thumb for making integral gear is as follows: 'If the diameter of dedendum circle (df) exceeds the diameter of the shaft (ds), at the point where the pinion is fi tted, by less than (ds/2), the pinion is made integral with the shaft

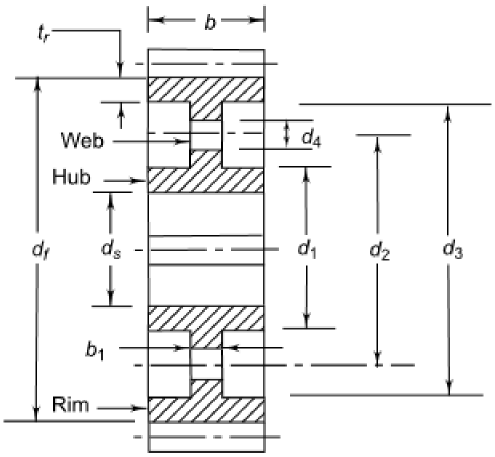


(ii) Medium Sized Gears :

There are two methods to manufacture medium size gears. They include machining from bar stock and forging. Gears of addendum circle diameter up to 150 mm are machined on rolled steel bars. The gear blanks in this case are turned on lathe. Gears of addendum circle diameterfrom 150 mm to 400 mm are mostly forged in open or closed dies. It again depends upon the volume of production. Even small diameter gears are forged, if the volume of production is large. Forged gears offer the following advantages: (a) The factor of material utilisation is equal to (1/3) when the gear is machined from bar stock. In case of forgings, material utilization factor is (2/3), which is twice. This reduces the cost of the material. (b) Forged gear has lightweight construction which reduces inertia and centrifugal forces. (c) The fi bre lines of the forged gears are arranged in a predetermined way to suit the direction of external force. In case of gears, prepared by machining methods, the original fi bre lines of rolled stock are broken. Therefore, the forged gear is inherently strong compared with machined gear. The limiting factor governing the choice of forged gears is their high cost. The equipment and tooling required to make forged gears is costly. Forged gears become economical only when they are manufactured on large scale.



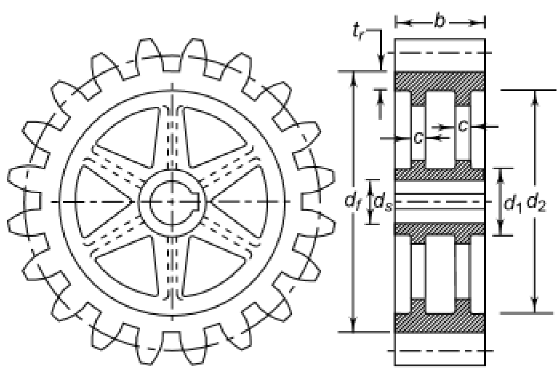
Machined Gear



Forged Gear

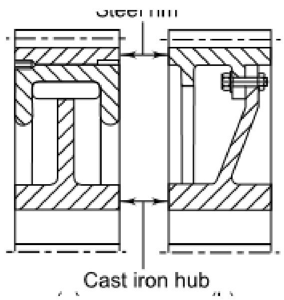
Gears with Large Diameter :

There are two varieties of large size gears—solid cast gears and rimmed gears. When the addendum circle diameter is up to 900 mm, a solid cast iron gear with one web is recommended. When the addendum circle diameter is more than 1000 mm, two webs are provided. A solid cast gear with two webs is shown in Fig. Solid cast iron gears are extensively used due to low cost. Though cast iron gears are cheaper than steel gears, their torque transmitting capacity is low. The dimensions of cast iron gears are determined by thumb rules and principles of casting design



Cast Iron Web Type Gear

A rimmed gear consists of a steel rim fi tted on the central casting with hub, arms or webs. The rim is forged from alloy steel. There are two varieties of rimmed gears, which are illustrated in Figs and (b). In the fi rst case, the rim is press fi tted on the casting and setscrews are used to prevent displacement of the rim with respect to casting. In the second type of construction, the rim is bolted to the central casting. Rimmed gears save costly high strength material, but they are more expensive to manufacture. The thickness of the rim from the inside diameter to the root circle diameter of tooth is usually taken as (7m) to (8m).



Rimmed Gears

STANDARD SYSTEMS OF GEAR TOOTH

	14.5° full depth system	*20° full depth system*	*20° stub system*
Pressure angle	14.5°	20°	20°
Addendum	m	m	0.8 m
Dedendum	1.157 m	1.25 m	m
Clearance	0.157 m	0.25 m	0.2 m
Working depth	2 m	2 m	1.6 m
Whole depth	2.157 m	2.25 m	1.8 m
Tooth thickness	1.5708 m	1.5708 m	1.5708 m

14.5° Full Depth Involute system :

The basic rack for this system is composed of straight sides except for the fi llet arcs. In this system, interference occurs when the number of teeth on the pinion is less than 23. This system is satisfactory when the number of teeth on the gears is large. If the number of teeth is small and if the gears are made by generating process, undercutting is unavoidable.

20° Full Depth Involute System :

The basic rack for this system is also composed of straight sides except for the fi llet arcs. In this system, interference occurs when the number of teeth on the pinion is less than 17. The 20° pressure angle system with full depth involute teeth is widely used in practice. It is also recommended by the Bureau of Indian Standards and adopted in this chapter. Increasing pressure angle improves the tooth strength but shortens the duration of contact. Decreasing pressure angle requires more number of teeth on the pinion to avoid undercutting. The 20° pressure angle is a good compromise for most of the power transmission as well as precision gearboxes

20° Stub Involute System

The gears in this system have shorter addendum and shorter dedendum. The interfering portion of the tooth, that is, a part of the addendum, is thus removed. Therefore, these teeth have still smaller interference. This also, reduces the undercutting. In this system, the minimum number of teeth on the pinion, to avoid interference, is 14. Since the pinion is small, the drive becomes more compact. Stub teeth are stronger than full depth teeth because of the smaller moment arm of the bending force. Therefore, the stub system transmits very high load. Stub teeth results in lower production cost, as less metal must be cut away. The main drawback of this system is that the contact ratio is reduced due to short addendum. Due to insuffi cient overlap, vibrations are likely to occur

Project I Problem Statement-

It is required to design a gear train for two roller sugarcane crusher. The gear train reduces the speed in two stages. The gear train receives power from a four stroke Engine. The force of failure of sugarcane for rate of 15 Kg/hr is 140 N. Consider factor of safety of 1.75 for failure force. Consider factor of safety 2 for gear design. Choose suitable material for gear and design the components of the unit such as gears, shaft key also select suitable bearings belt and pulleys.

Given:

m = 15 kg/hr, $(F)_{failure}$= 140 N, $(FOS)_{crusher}$ = 1.75,

$(FOS)_{geardesign}$= 2, $(n)_{output}$ =35 rpm, $(D)_{crusher}$ = 130 mm

Belt drive reduction = 4:1

Solution:

I]First we determine torque.

1) Force required for crushing (P_c) = $(F)_{failure}$ * $(FOS)_{crusher}$ = 140*1.75 = 245 N

2) Let us consider 4 sugarcane passes through rollers,

Total crushing force (P_t) = 4*P_c = 980 N

3) Torque (M_t) = P_t * $(R)_{crusher}$= 980*(130/2) = 63700 Nmm

4) Power required for crushing = $(2\pi*n*M_t)/(60*10^6)$ = $(2\pi*35*63700)/(60*10^6)$

= 0.23347 kW

5) Let $(\eta)_{motor}$ = 80%

6) Power of motor required = 0.23347/0.8 = 0.29184 kW

II] Selecting standard motor of 1 HP = 0.7457 kW

1)Input Speed at gear train = 1440/4 = 360 rpm (Due to belt reduction)

Output speed at gear train = 35 pm

2) Transmission ratio (i’) = 360/35 = 10.2857

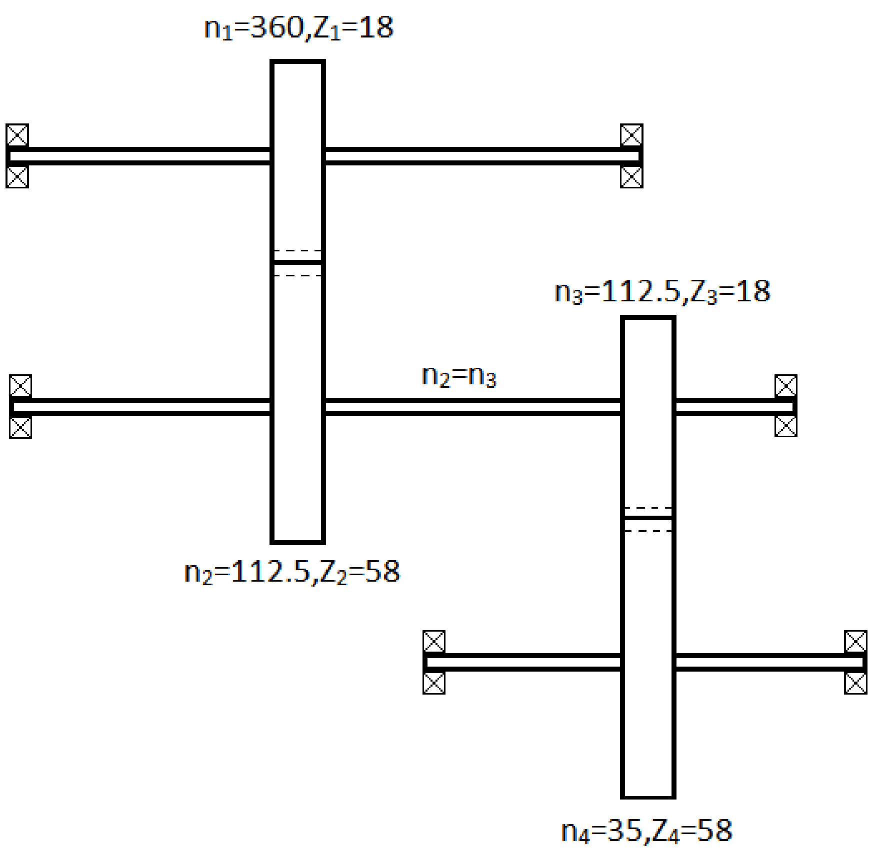
3) Gear ratio = i = √i’ = 3.2017

III]

1)We select first CI material(FG260) with S_{ut}= 260 n/mm^2

2) Designing Gear pair for First stage

n_1= 360 rpm, n_2=n_3= 112.25 rpm, n_4= 35 rpm,Z_1=18, Z_2= 58, Z_3= 18,Z_4= 58



For 20^0 Pressureangle,

Z_p= 18 ,Z_g = i*Z_p = 58

Assume pitch line velocity (v) = 5 m/s

C_v= 3/(3+v) = 3/(3+5) = 0.375

From table 17.4, For heavy shock and uniform, C_s = 1.75

Lewis form factor for 18 teeth (y) = 0.308

For First stage,

$m = \{(60*10^6/\pi)*(P*C_s*FOS)/[Z_1*n_1*C_v*(b/m)*(S_{ut}/3)*y]\}^{1/3}$

$= \{(60*10^6/\pi)*(0.7457*1.75*2)/[18*360*0.375*10*(260/3)*0.308]\}^{1/3}$

m = 5 mm

d_1' = 5*18 = 90 mm

d_2'= 5*58 = 290 mm

b = 10*5 = 50 mm

IV] By using Buckingham's Equation

$S_b = m*b*\sigma_b*y$ = 5*50*(260/30)*0.308 = 6673.33 N

Tangential Force,

Torque = $(60*10^6*P)/(2\pi*n_1) = (60*10^6*0.7457)/(2\pi*360)$ = 19780..3Nmm

P_t= (2*Toque)/d_1' = (2*19780.3)/90 = 439.56 N

Now dynamic load

1) Pinion (grade 8)

e = 16+1.25ϕ,

but, $\phi = m+0.25\sqrt{d_1}$'

e_p= 16+1.25(5+0.25$\sqrt{90}$) = 25.214 µm

2) Gear

Here, $\phi = m+0.25\sqrt{d_2}$'

e_g= 16+1.25(5+0.25$\sqrt{290}$) = 27.57 µm

now,

$e = e_p + e_g = 52.7856\ \mu m = 52.7856*10^{-3} mm$

Deformation factorC for CI material is 5700.

$v = (\pi*d_1'*n_1)/(60*10^3) = 1.6964\ m/s$

$P_d = \{21*v*(C*e*b+P_t)\}/\{21*v*\sqrt{(C*e*b+P_t)}\}$

$= 6919.54\ N$

$P_{eff} = (C_s*P_t)+P_d = (1.75*439.56)+6919.54 = 7688.77\ N$

But $P_{eff} > S_b$, Hence design is not safe.

Hence we need to increase the module.

Let us consider m = 8

$d_1' = 8*18 = 144\ mm$

$d_2' = 8*58 = 464\ mm$

$b = 8*10 = 80\ mm$

By using Buckingham's Equation

$S_b = m*b*\sigma_b*y = 8*80*(260/30)*0.308 = 17083.73\ N$

$Torque = (60*10^6*P)/(2\pi*n_1) = (60*10^6*0.7457)/(2\pi*360) = 19780.3\ Nmm$

$P_t = (2*Toque)/d_1' = (2*19780.3)/144 = 274.72\ N$

Now dynamic load

1) Pinion (grade 8)

$e = 16+1.25\phi$,

but, $\phi = m+0.25\sqrt{d_1'}$

$e_p = 16+1.25(8+0.25\sqrt{144}) = 29.75\ \mu m$

2) Gear

Here, ϕ = m+0.25$\sqrt{d_2'}$

e_g= 16+1.25(8+0.25$\sqrt{464}$) = 32.73 µm

now,

e = e_p+e_g = 62.48 µm = 62.48*10^{-3}mm

Deformation factor C for CI material is 5700.

v = (π*d_1'*n_1)/(60*10^3) = 2.714 m/s

P_d= {21*v*(C*e*b+P_t)}/{21*v*$\sqrt{(C*e*b+P_t)}$}

= 7235.1249 N

P_{eff} = (C_s*P_t)+P_d = (1.75*274.72)+7235.1249 = 7715.8849 N

FOS = S_b / P_{eff} = 2.214 > 2.   Hence design is safe and acceptable.

Also, S_w= FOS*P_{eff} = 7715.88*2.214 = 17083.719 N

But S_w= b*q*d_p' *k = 80*(2*58/58+18)*144*(BHN/100)^2*0.16

(BHN/100)^2 = 6.072

BHN = 246.41

Now mass of gear and pinion,

P = 7000 kg/m^3

M_p=π*(d_1')^2*b*ρ/4 = 9.12 kg = 89.72 N

M_g = π*(d_2')^2*b*ρ/4 = 94.69 kg = 928.92 N

V] Design for second stage

n_3 = 112.25 rpm, n_4 = 35 rpm, y = 0.308

C_s = 1.75, C_v= 3/(3+v) = 3/(3+5) = 0.375

$m = \{(60*10^6/\pi)*(P*C_s*FOS)/[Z_1*n_1*C_v*(b/m)*(S_{ut}/3)*y]\}^{1/3}$

$= \{(60*10^6/\pi)*(0.7457*1.75*2)/[18*112.25*0.375*10*(260/3)*0.308]\}^{1/3}$

$m = 6.2697 \sim 8$ mm

$d_3' = 8*18 = 144$ mm

$d_4' = 8*58 = 464$ mm

$b = 8*10 = 80$ mm

By using Buckingham's Equation

$S_b = m*b*\sigma_b*y = 8*80*(260/30)*0.308 = 17083.73$ N

Torque $= (60*10^6*P)/(2\pi*n_3) = (60*10^6*0.7457)/(2\pi*112.25) = 63437.95$Nmm

$P_t = (2*\text{Toque})/d_3' = (2*63437.95)/144 = 881.08$ N

Now dynamic load

1) Pinion (grade 8)

$e = 16+1.25\phi$,

but, $\phi = m+0.25\sqrt{d_3'}$

$e_p = 16+1.25(8+0.25\sqrt{144}) = 29.75$ µm

2) Gear

Here, $\phi = m+0.25\sqrt{d_4'}$

$e_g = 16+1.25(8+0.25\sqrt{464}) = 32.73$ µm

now,

$e = e_p+e_g = 62.48$ µm $= 62.48*10^{-3}$mm

Deformation factor C for CI material is 5700.

$v = (\pi*d_3'*n_3)/(60*10^3) = 0.84635$ m/s

$P_d = \{21*v*(C*e*b+P_t)\}/\{21*v*\sqrt{(C*e*b+P_t)}\}$

$= 2759.8$ N

$P_{eff} = (C_s*P_t)+P_d = (1.75*881.08)+2759.8 = 4301.69$ N

FOS = $S_b / P_{eff} = 3.9713 > 2$. Hence design is safe and acceptable.

Also, $S_w = FOS*P_{eff} = 3.9713*4301.69 = 17083.301$ N

But $S_w = b*q*d_p'*k = 80*(2*58/58+18)*144*(BHN/100)^2*0.16$

$(BHN/100)^2 = 6.072$

BHN = 246.41

VI] Design for third stage

$N_5 = 35$ rpm, $n_6 = 35$ rpm, $y = 0.308$, $Z_5 = Z_6 = 37$,

$C_s = 1.75$, $C_v = 3/(3+v) = 3/(3+5) = 0.375$

$m = \{(60*10^6/\pi)*(P*C_s*FOS)/[Z_5*n_5*C_v*(b/m)*(S_{ut}/3)*y]\}^{1/3}$

$= \{(60*10^6/\pi)*(0.7457*1.75*2)/[37*35*0.375*12*(260/3)*0.308]\}^{1/3}$

$m = 6.3802 \sim 7$ mm

$d_5' = 7*37 = 259$ mm

$d_6' = 7*37 = 259$ mm

$b = 7*12 = 84$ mm

$M = \pi*(d_5')^2*b*\rho/4 = 30.97$ kg $= 303.90$ N

By using Buckingham's Equation

$S_b = m*b*\sigma_b*y = 7*84*(260/30)*0.308 = 19137.33$ N

Torque $= (60*10^6*P)/(2\pi*n_5) = (60*10^6*0.7457)/(2\pi*35) = 203454.58$ Nmm

P_t= (2*Toque)/d_5' = (2*203454.58)/259 = 1571.07 N

e = 2*e_p= 2*[16+1.25(7+0.25√259)] = 59.55 µm = 5955*10^{-3}

v = (π*d_1'*n_5)/(60*10^3) = 4.746*10^{-3} m/s

P_d= {21*v*(C*e*b+P_t)}/{21*v*√(C*e*b+P_t)}

= 1.5817 N

P_{eff} = (C_s*P_t)+P_d = (1.75*1571.07)+1.5817 = 2750.95 N

FOS = S_b / P_{eff} = 6.95> 2. Hence design is safe and acceptable.

Also, S_w= FOS*P_{eff} = 6.95*2750.95 = 19119.1025 N

But S_w= b*q*d_p'*k = 84*[(2*37)/(37+37)]*259*(BHN/100)^2*0.16

(BHN/100)^2 = 5.4927

BHN = 234.36

VII] Design of Pulley

Motor speed = 1440 rpm

ω = 2πN/60 = 150.79 rad/s

Motor torque = Motor Power / ω = (0.7457*10^3)/150.79 = 4.9450 Nm

Motor pulley diameter (D_m) = 58*(Motor torque)$^{1/3}$ = 58*(4.9450)$^{1/3}$ = 98.8136 ~ 100mm

Now,

D_m/D_s = N_2/N_1 ... (where D_s is shaft diameter).

100/D_s = 360/1440

D_s = 400 mm

Let weight of pulley = 26 N

VIII] Belt Design

Let thickness of belt = 11 mm   …(selected B pulley)

Center distance (C) = $0.55*(D_1+D_2)+t_b$ = 0.55*(100+400)+11 = 286 mm

Length of belt (L) = $2*C+[\pi*(D_1+D_2)/2]+[(D_1+D_2)/4*C]^2$

$= 2*286+[\pi*(100+400)/2]+[(100+400)/4*286]^2$

= 1.357 m

From standard catalogue,

L = 1.37 m

Angle of wrap(α):

$\sin(\beta) = (r_2-r_1)/C = 0.5244$

$\beta = 31.63^0$

1)For motor pulley, $\alpha_1 = 180-2\beta = 116.73^0$

2) For shaft pulley, $\alpha_2 = 180+2\beta = 243.262^0$

Belt tension,

$(T_1-T_2)*D_2/2$ = Torque = 19780 Nmm

(T_1-T_2) = 19780*2/400 = 98.9 N

Also,

$T_1/T_2 = e^{\mu\theta}$

Let $\mu = 0.24$ and $\theta = 243.26^0 = 4.246^c$

$T_1/T_2 = e^{0.24*4.246} = 2.7703$

$T_1 = 2.7703*T_2$

Hence,

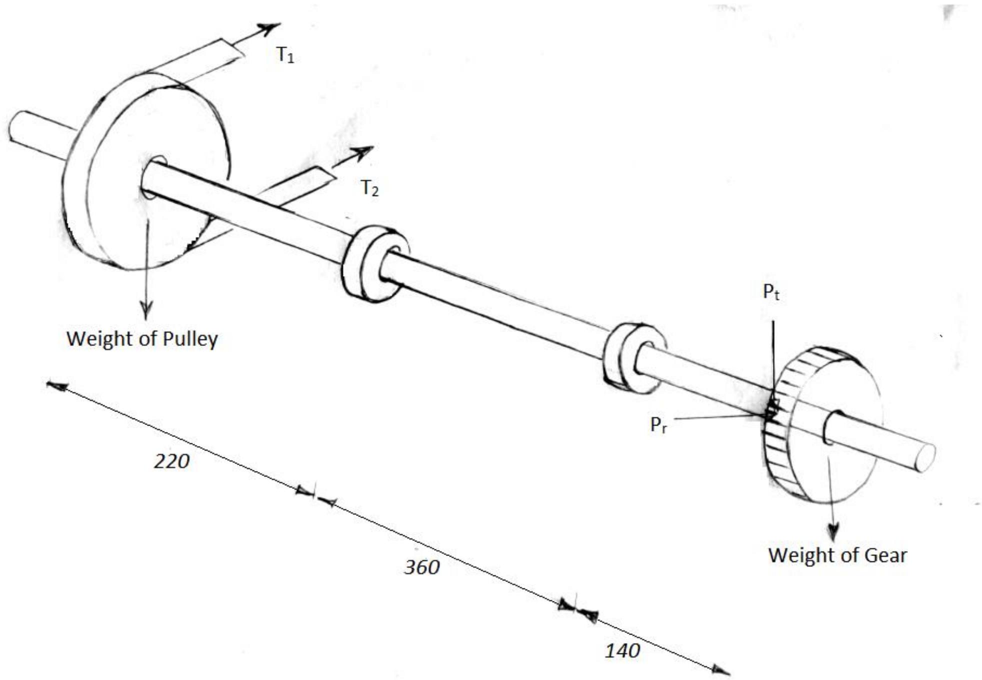
$(2.7703-1)*T_2 = 98.9$

$T_2 = 55.8654$ N

$T_1 = 111.7308$ N

IX] Design of Shaft

1) Shaft 1



Let material of shaft is 50C4

$S_{ut} = 700$ N/mm^2

$S_{yt} = 460$ N/mm^2

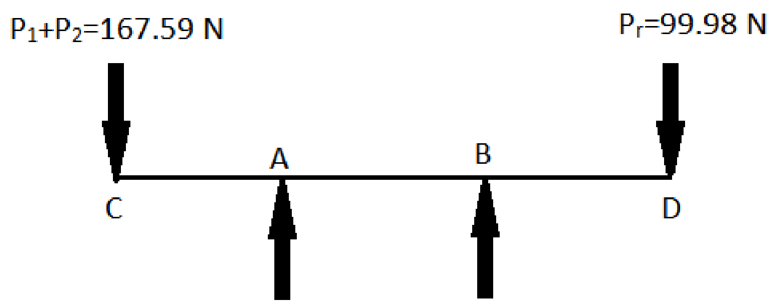
Permissible Shear stess,

$= 0.3*S_{ut}=138$ N/mm^2

$= 0.18*S_{yt} = 126$ N/mm^2

$\tau_{max} = 0.75*126 = 94.5$ N/mm^2

i)Horizontal Plane



Taking Moment about D,

$(-167.59*700)+(500*R_A)+(R_B*140) = 0$

$R_B*140 = 117313-500*R_A$

But,

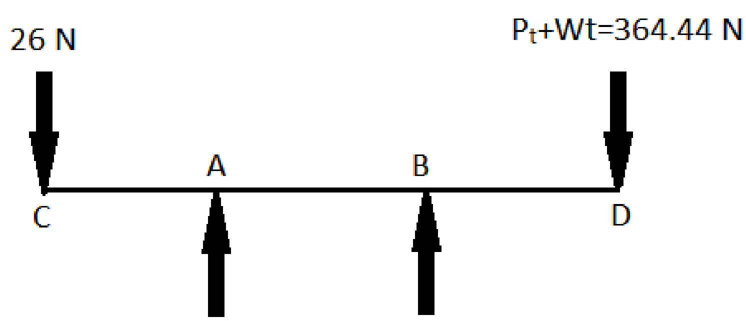
$R_A+R_B = 267.57$ N

$R_B = 45.75$ N and $R_A = 221.81$ N

Bending moment at A = 167.59*200 = 33518 Nmm

Bending moment at B = 99.98*140 = 13997.2 Nmm

ii) Vertical Plane



$R_A+R_B = 390.44$ N

Taking Moment about A,

$(364.44*500)-(R_B*360)-(26*200) = 0$

$R_B = 491.72$ N

Hence, R_A = -101.28 … (It means R_A is in opposite direction).

R_A = 101.28 N

Bending moment at B = 364.44*140 = 51021.6 Nmm

BM_{max} = 51021.6 Nmm

$d^3 = \{(16/\pi*\tau_{max})*\sqrt{[(1.5*BM_{max})^2+(1.5*Torque_{max})^2]}\}$

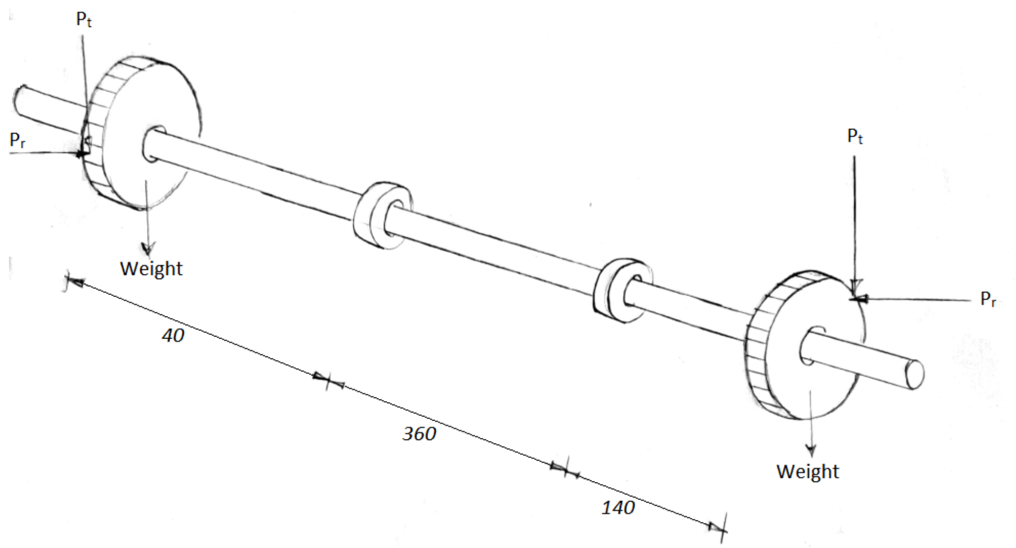
$= \{(16/\pi*94.5)*\sqrt{[(1.5*51021.6)^2+(1.5*19780)^2]}\}$

= 4423.72

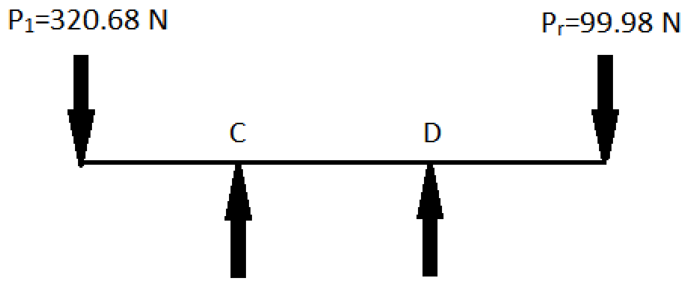
Hence, d = 16.41 mm

From standard catalogue d = 20 mm

2) Shaft 2



i)Horizontal Plane



Taking Moment about C,

$(99.98*500)-(360*R_D)-(320.6*40) = 0$

$R_D = 103.23$ N

But,

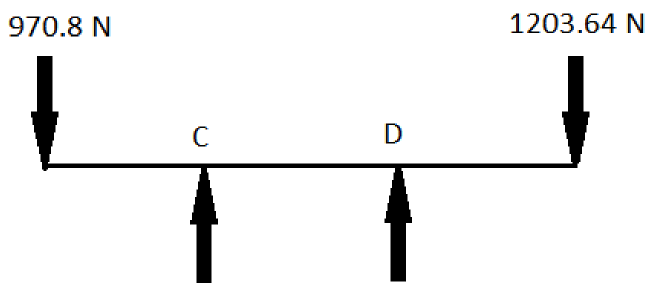
$R_C+R_D = 420.66$ N

$R_C = 317.4211$ N

Bending moment at C = 12827.2Nmm

Bending moment at D = 99.98*140 = 13997.2 Nmm

ii) Vertical Plane



$R_C+R_D = 2174.44$ N

Taking Moment about C,

$(1203.64*500)-(R_D*360)-(970.8*40) = 0$

$R_D = 1563.85$ N

Hence, $R_C = 610.58$ N

Bending moment at C = 970.8*40 = 38832 Nmm

Bending moment at D = 1203.64*140 = 168509.6 Nmm

BM_{max} = 168509.6Nmm

$d^3 = \{(16/\pi*\tau_{max})*\sqrt{[(1.5*BM_{max})^2+(1.5*Torque_{max})^2]}\}$

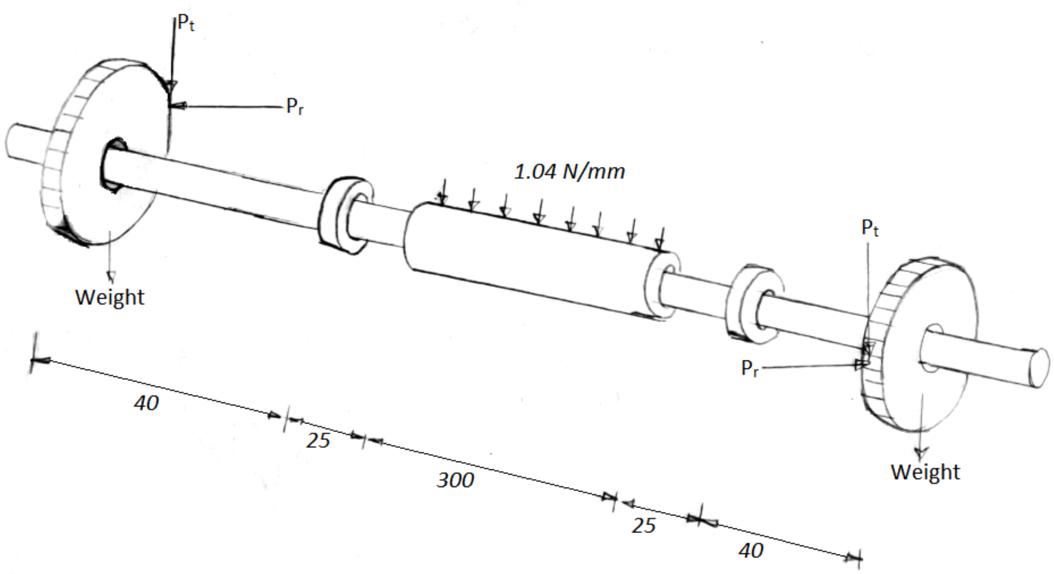
$= \{(16/\pi*94.5)*\sqrt{[(1.5*168509.6)^2+(1.5*63437.95)^2]}\}$

= 4423.72

Hence, d = 24.41 mm

From standard catalogue d = 25 mm

3) Shaft 3



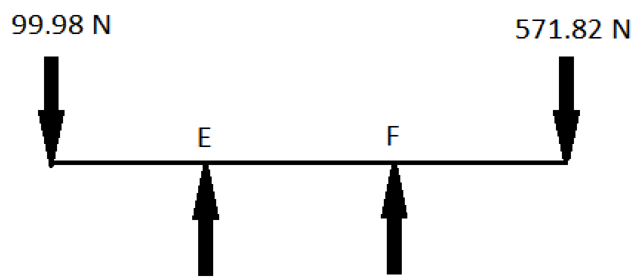
Crusher roller material is stainless steel.

P = 8000 kg/m^3

Roller diameter = 130 mm

Mass of roller = $(\pi/4)*(0.130)^2*0.3*8000$ = 31.85 kg = 312.5 N

i)Horizontal Plane



R_E+R_F = 671.8 N

Taking Moment about E,

$(571.82*390)-(R_F*350)-(99.98*40) = 0$

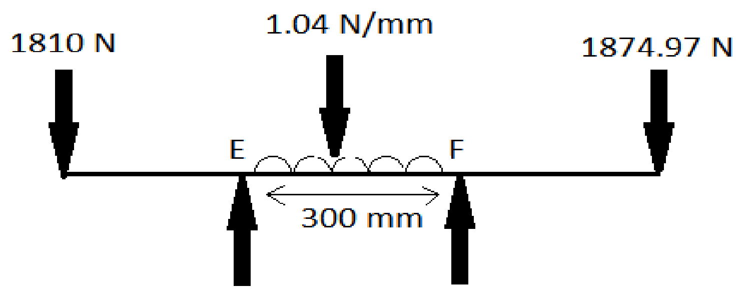
R_F = 625.74 N

R_E = 46.055 N

Bending moment at E =99.98*40 = 3999.2Nmm

Bending moment at F = 571.82*40 = 22872.4Nmm

ii) Vertical Plane



R_E+R_F = 3684.97+(1.04*300) = 3996.97 N

Taking Moment about E,

$(1874.97*300)-(R_F*350)+(1.04*300*175)-(1810*40) = 0$

R_F = 2038.39 N

Hence, R_E = 1958.57 N

Bending moment at E = 1812*40 = 72400Nmm

Bending moment at F = 1874.97*40 = 74998.8 Nmm

BM_{max} = 74998.8Nmm

Torque = 203454.58 Nmm

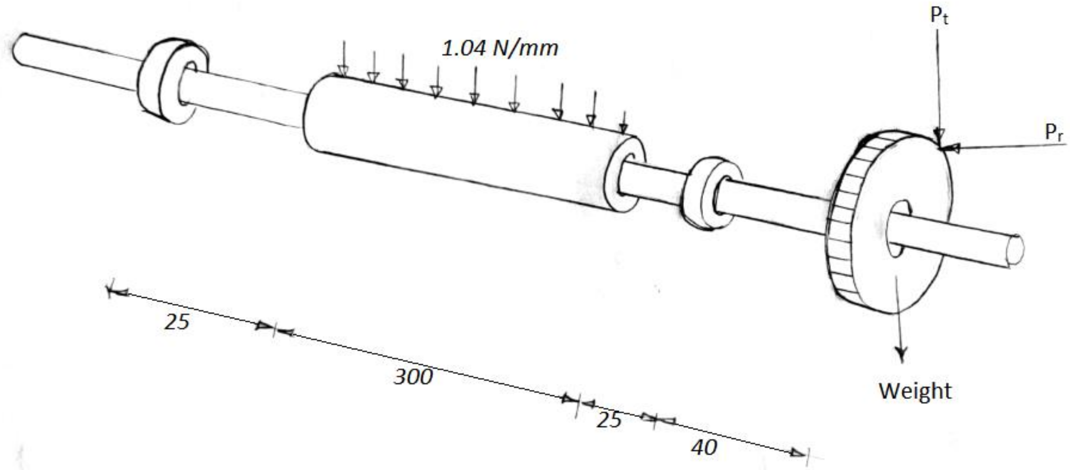
$d^3 = \{(16/\pi*\tau_{max})*\sqrt{[(1.5*BM_{max})^2+(1.5*Torque_{max})^2]}\}$

$= \{(16/\pi*94.5)*\sqrt{[(1.5*74998.8)^2+(1.5*203454.58)^2]}\}$

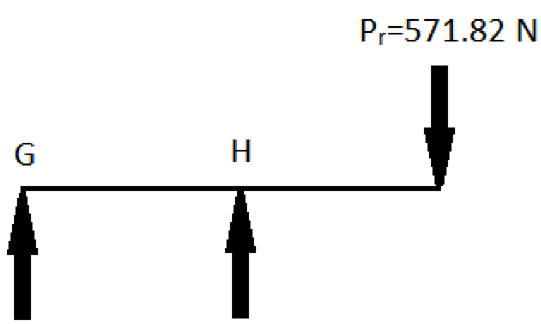
Hence, d = 25.97 mm

From standard catalogue d = 30 mm

4) Shaft 4



i) Horizontal Plane



$R_G+R_H = 571.82$ N

Taking Moment about G,

$(571.82*390)-(R_H*350) = 0$

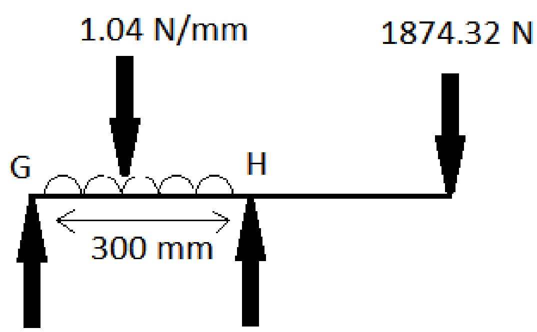
$R_H = 67.17$ N

Hence, $R_G = -65.35$ N … (It means R_G is in opposite direction).

$R_G = 65.35$ N

Bending moment at H = 571.82*40 = 22872.8 Nmm

ii) Vertical Plane



$R_G+R_H = 1874.32+1.04*300 = 2186.32$ N

Taking Moment about G,

$(1874.32*390)-(R_H*350)+(1.04*300*175) = 0$

$R_H = 2244.528$ N

Hence, $R_G = -58.208$ N … (It means R_G is in opposite direction).

$R_G = 58.208$ N

Bending moment at H = 1874.32*40 = 74972.8 Nmm

$BM_{max} = 74972.8$ Nmm

Torque = 203454.58 Nmm

$d^3 = \{(16/\pi*\tau_{max})*\sqrt{[(1.5*BM_{max})^2 + (1.5*Torque_{max})^2]}\}$

$= \{(16/\pi*94.5)*\sqrt{[(1.5*74972.8)^2 + (1.5*203454.58)^2]}\}$

$= 17528.56$

Hence, d = 25.97 mm

From standard catalogue d = 30 mm

X] Bearing Selection [We use single row deep groove ball bearing]

1) Shaft 1

n = 360 rpm, d = 20 mm, L_{10h} = 20000 hrs, Load factor = 2.5

From shaft design,

P_{HA} = 221.81 N , P_{HB} = 45.75 N

P_{VA} = 101.25 N , P_{VB} = 491.72 N

$P_A = \sqrt{[(P_{HA})^2 + (P_{VA})^2]} = \sqrt{[(221.81)^2 + (101.25)^2]} = 243.83$ N

$P_B = \sqrt{[(P_{HB})^2 + (P_{VB})^2]} = \sqrt{[(45.75)^2 + (491.72)^2]} = 493.84$ N

This Bearing reaction are in radial direction, Axial reactions = 0

$L_{10} = (60*n*L_{10H})/10^6 = (60*360*20000)/10^6 = 432$ million rev.

Dynamic load carrying capacity,

$C_A = P_A$*(Load factor)*$(L_{10})^{1/3}$ = 4608.09 N

$C_B = P_B$*(Load factor)*$(L_{10})^{1/3}$ = 9332.99 N

At A, bearing no.16404 with C = 7020 N and

At B, bearing no.6204 with C = 12700 N are selected.

2) Shaft 2

n = 112 rpm, d = 25 mm, L_{10h} = 20000 hrs, Load factor = 1.4

From shaft design,

P_{HC} = 317.42N , P_{HD}= 103.23 N

P_{VC}= 610.58N , P_{VD} = 1563.85 N

$P_C = \sqrt{[(P_{HC})^2 + (P_{VC})^2]} = \sqrt{[(317.42)^2 + (610.58)^2]} = 688.15$ N

$P_D = \sqrt{[(P_{HD})^2 + (P_{VD})^2]} = \sqrt{[(103.23)^2 + (1563.85)^2]} = 1567.253$ N

This Bearing reaction are in radial direction, Axial reactions = 0

$L_{10} = (60*n*L_{10H})/10^6 = (60*112*20000)/10^6 = 134.4$ million rev.

Dynamic load carrying capacity,

$C_C = P_C*(\text{Load factor})*(L_{10})^{1/3} = 4934.89$ N

$C_D = P_D*(\text{Load factor})*(L_{10})^{1/3} = 11239.15$ N

At C, bearing no.16005 with C = 7610 N and

At D, bearing no.6205 with C = 14000 N are selected.

3) Shaft 3

n = 35 rpm, d = 30 mm, L_{10h} = 20000 hrs, Load factor = 1.4

From shaft design,

P_{HE} = 46.05 N , P_{HF} = 625.74 N

P_{VE} = 1958.57 N , P_{VF} = 2038.39 N

$P_E = \sqrt{[(P_{HE})^2 + (P_{VE})^2]} = \sqrt{[(46.05)^2 + (1958.57)^2]} = 1959.11$ N

$P_F = \sqrt{[(P_{HF})^2 + (P_{VF})^2]} = \sqrt{[(625.74)^2 + (2038.39)^2]} = 2132.27$ N

This Bearing reaction are in radial direction, Axial reactions = 0

$L_{10} = (60*n*L_{10H})/10^6 = (60*35*20000)/10^6 = 42$ million rev.

Dynamic load carrying capacity,

$C_E = P_E*(\text{Load factor})*(L_{10})^{1/3} = 9533.88$ N

$C_F = P_F*(\text{Load factor})*(L_{10})^{1/3} = 10376.55$ N

At E, bearing no.16006 with C = 11200 N and

At F, bearing no.16006 with C = 11200 N are selected.

4) Shaft 4

$n = 35$ rpm, $d = 30$ mm, $L_{10h} = 20000$ hrs, Load factor = 1.4

From shaft design,

$P_{HG} = 65.35$N , $P_{HH} = 637.17$ N

$P_{VG} = 58.208$N , $P_{VH} = 2244.528$ N

$P_G = \sqrt{[(P_{HG})^2 + (P_{VG})^2]} = \sqrt{[(65.35)^2 + (58.208)^2]} = 87.51$ N

$P_H = \sqrt{[(P_{HH})^2 + (P_{VH})^2]} = \sqrt{[(637.17)^2 + (2244.528)^2]} = 2333.21$ N

This Bearing reaction are in radial direction, Axial reactions = 0

$L_{10} = (60*n*L_{10H})/10^6 = (60*35*20000)/10^6 = 42$ million rev.

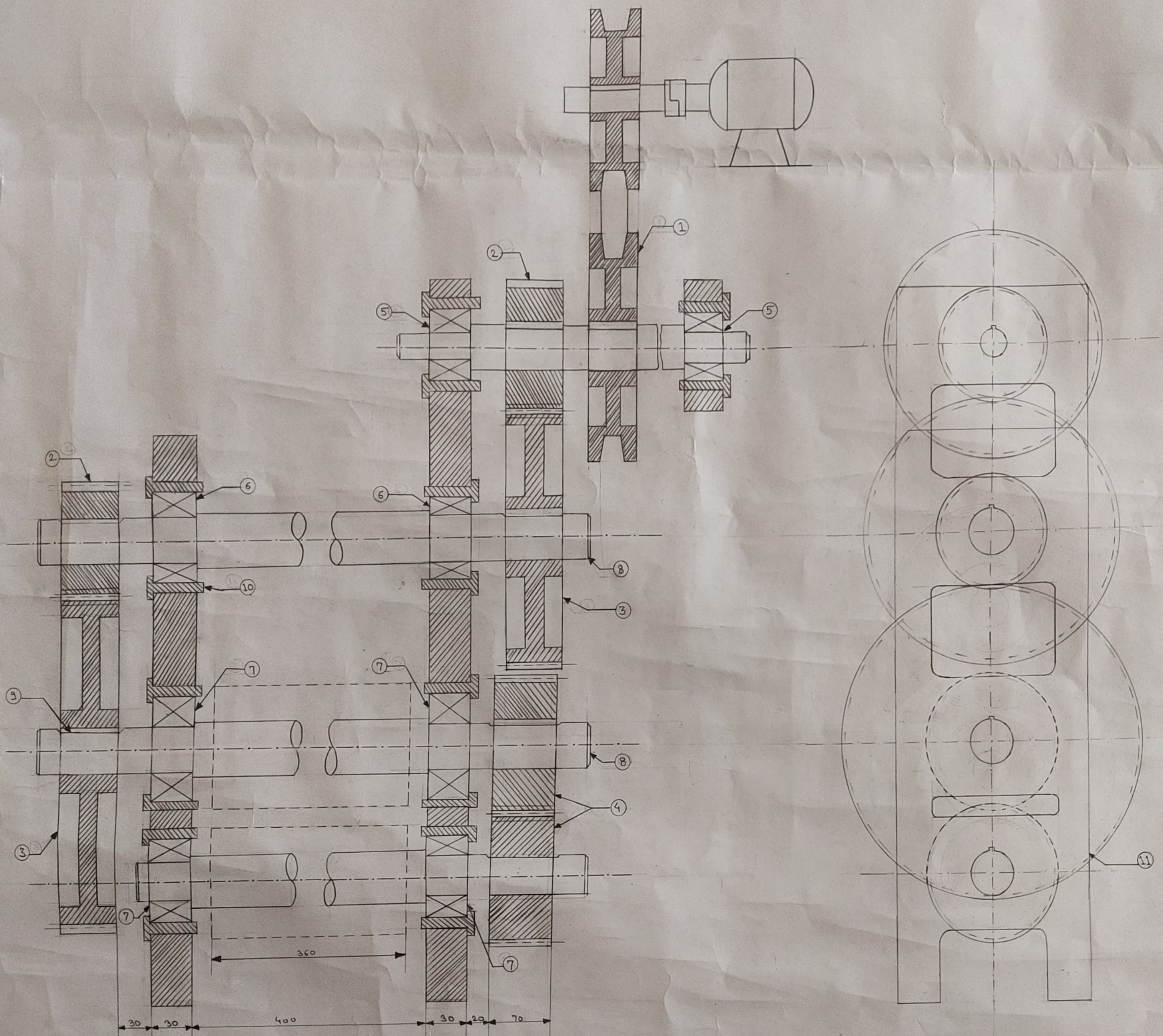
Dynamic load carrying capacity,

$C_G = P_G*(\text{Load factor})*(L_{10})^{1/3} = 425.8619$ N

$C_H = P_H*(\text{Load factor})*(L_{10})^{1/3} = 11354.42$ N

At G, bearing no.61806 with C = 3120 N and

At H, bearing no.6006 with C = 13300 N are selected.



NO.	COMPONENT	MATERIAL	QTY
1	PULLEY ϕ240mm	FG260	1
2	SPUR PINION N=18	FG300	2
3	SPUR GEAR N=57	FG300	2
4	SPUR GEAR N=20	FG300	2
5	BALL BEARING	620522	1
6	BALL BEARING	630822	2
7	BALL BEARING	640922	2
8	SHAFTS	C45 STEEL	4
9	KEYS	C45 STEEL	8
10	COVER PLATE	FG260	5
11	SUPPORTS	FG260	1

KIT'S COLLEGE OF ENGINEERING
KOLHAPUR

NAME :-

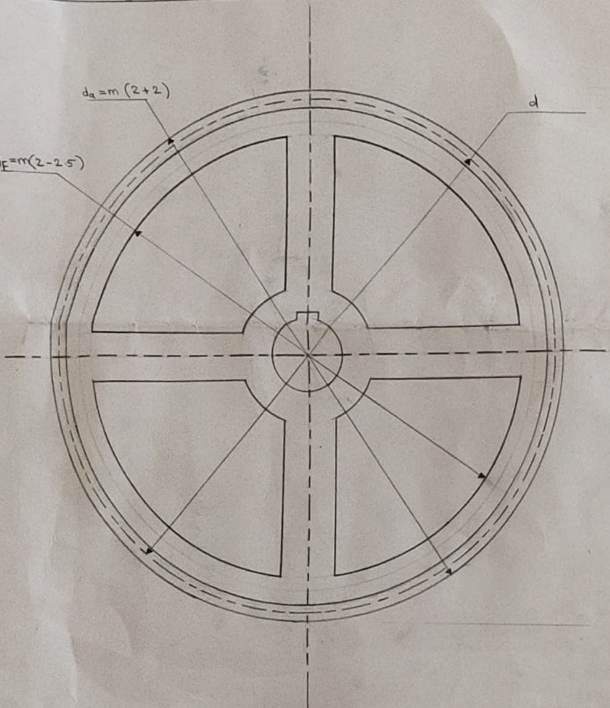
DIV :-

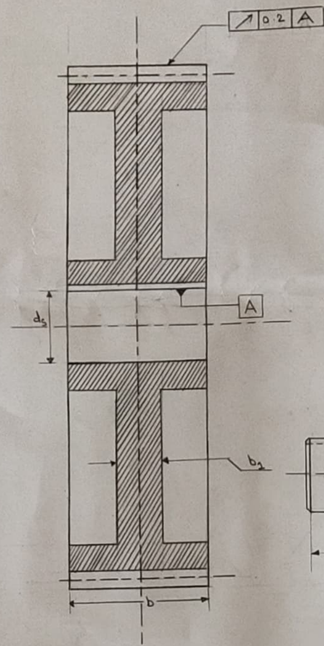
CLASS :-

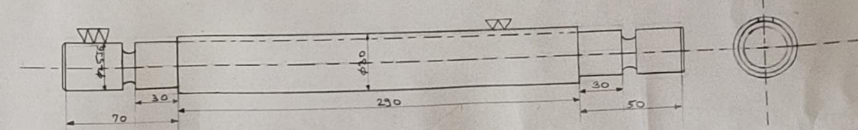
ROLL NO :-

SPUR GEAR DESIGN FOR SUGARCANE CRUSHER

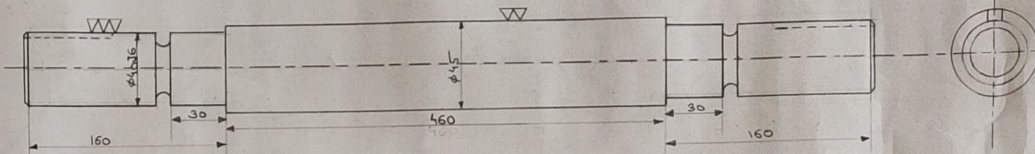
SYMBOL :-



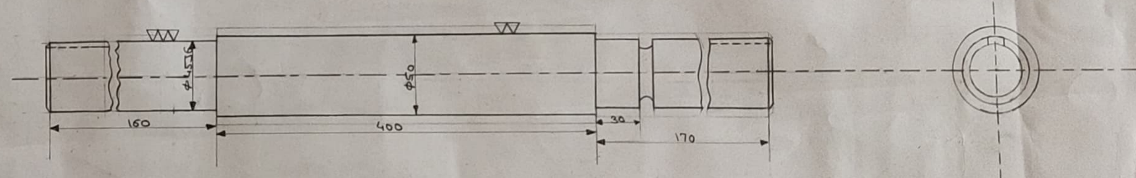




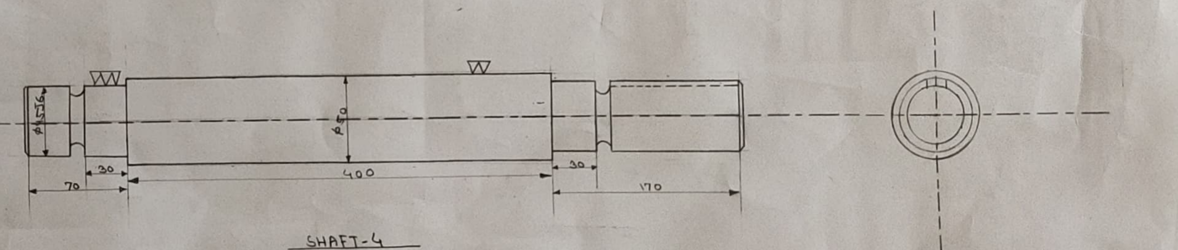
SHAFT - 1



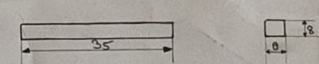
SHAFT - 2



SHAFT - 3



SHAFT - 4



KEY

GEARS	PITCH CIRCLE DIAMETER (d)	FACE WIDTH (b)	MODULE (m)
G_1	d_1 = 144mm	b_1 = 80mm	m_1 = 8
G_2	d_2 = 464mm	b_2 = 80mm	m_2 = 8
G_3	d_3 = 144mm	b_3 = 80mm	m_3 = 8
G_4	d_4 = 464mm	b_4 = 80mm	m_4 = 8
G_5	d_5 = 259mm	b_5 = 84mm	m_5 = 7
G_6	d_6 = 259mm	b_6 = 84mm	m_6 = 7

GEARS	NO. OF TEETH z	ADDENDUM CIRCLE DIAMETER (d_a)	DEDENDUM CIRCLE DIAMETER (d_f)	SHAFT DIAMETER (d_s)	WIDTH OF WEB
G_1	18	160mm	124mm	20mm	24mm
G_2	58	480mm	444mm	25mm	24mm
G_3	18	160mm	124mm	25mm	24mm
G_4	58	480mm	444mm	30mm	24mm
G_5	37	273mm	241.5mm	30mm	25.2mm
G_6	37	273mm	241.5mm	30mm	25.2mm

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★ Problem statement -

* Design a bevel gear pair unit for trasmitting torque from motor of groundnut oil Expeller machine (Power Ghani). The capacity of the is 15kg/hr consider the,

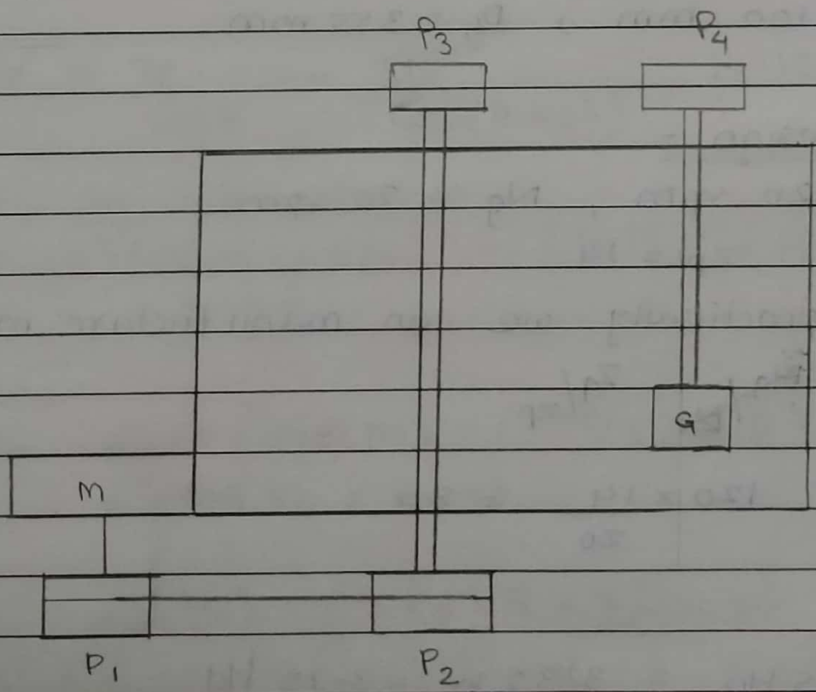
required output of expeller drum as 18 rpm. The power required for crushing groudnut is 1.6 kW & power required to rotate trasmission elements & expeller is 0.6 kW. The transmission efficiency can be considered as 80%. Select the suitable prime mover.

select the standard pulleys & determine reduction.

Design the gear drive.

Select suitable bearings.

Design the shafts on which the gear pair is mounted.



The pulley drive is as shown in above figure.

Given input speed is 1440 rpm for 5 HP motor.

P_1, P_2, P_3, P_4 are the pulleys & D_1, D_2, D_3, D_4 are the respective dimeters.

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→ Given input speed is 1440 rpm for 5 Hp motor.
P_1, P_2, P_3, P_4 are the pulleys & D_1, D_2, D_3, D_4 are the respective diameters.

Now, $N_1 = 1440$ rpm, $N_4 = 115$ rpm.

$i = N1/N4 = \frac{1440}{115} = 12.52$

$i' = N_1/N_2 = \sqrt{i} = 3.54$

$N_2 = 1440/3.54 = 406.78 rpm = N_3$.

Now,
Assume $D_1 = 100$ mm.
$D_2 = D_1 \times i' = 3.54 \times 100 = 354$ mm
from manufacturing catalogue $D_2 = 355$ mm.
Now similarly,
$D_3 = 100$ mm, $D_4 = 355$ mm.

Gear Design -
$N_p = 120$ rpm, $N_g = 20$ rpm.
Assume $Z_p = 14$.
since practically we can manufacture minimum.

$\frac{N_p}{N_g} = \cancel{Z_g/N_p}\ Z_g/Z_p$

$Z_g = \frac{120 \times 14}{20} = 84$.

Now.
$P = 5Hp = 3730 W = 3.73 kW$
$P = 2 \times \pi \times N \times T / 60$.

$T = 60 \times P / 2 \times \pi \times N$.
$= 60 \times 3.73 \times 10^6 / 2 \times \pi \times 120$.

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$= 296823.97$ Nmm

Also,

$T = P_T \times dp/2$

$$P_T = \frac{2 \times T}{dp} = \frac{2 \times 296823.97}{m \times 14}$$

$$= 42403.42/m N \quad \text{—①}$$

$S_b = m \times b \times \sigma_b \times y\left[1-(b/A_0)\right]$

Assume $b/A_0 = 1/3$ & $b = 12 m$

Now,

$\tan(\gamma) = z_p/z_g = 14/84$

$\gamma = \tan^{-1}(14/84) = 9.46°$

$$z_p' = \frac{z_p}{\cos\gamma} = \frac{14}{\cos(9.46)} = 14.19$$

Now by interpotation,

$(15-14)/(0.289-276) = (14.19-14)/(y'-0.246)$

$y' = 0.2785$

Now,

$S_b = m \times (12 \times m) \times (700/3) \times 0.2785 \times (2/3)$

$= 519.86 \times m^2$ N

Now,

$S_w = (3/4) \times (b \times d \times Q \times x/\cos\gamma)$

But $Q = 2 \times z_g/(z_g + z_p \times \tan\gamma) = 1.94$

$x = 0.16 \times (400/100)^2 = 2.56$

Hence

$S_w = (3/4) \times (12 \times m \times m \times 14 \times 1.94 \times 2.56)/\cos(9.46)$

$= 634.39 \times m^2$ N

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Now Assume $V = 7.5$ m/s

$C_v = 5.6/(5.6 + \sqrt{V}) = 0.671$

$C_s = 1.5$

$P_{eff} = (C_s/C_v) \times P_T = (1.5/0.671) \times (42403.42/m)$

$= 94791.55/m.$

Now assume FOS = 2.

$S_b/FOS = P_{eff}$

$(519.86 \times m^2)/2 = 94791.55/m.$

$m = 7.11 \approx 8$ mm

$d_p = m \times Z_p = 8 \times 14 = 112$ mm.

$d_g = m \times Z_g = 8 \times 84 = 672$ mm.

Now,

$V = (\pi \times d_p \times N_p)/(60 \times 10^6)$

$= (\pi \times 112 \times 120)/(60 \times 10^6)$

$= 0.7$ m/s

For class 3 grade, $e = 0.0125$ mm.

Now from ①.

$P_T = 42403.42/m = 42403.42/8$

$= 5300.43$ N.

$b = 12 \times m = 96$ mm.

Now,

$P_d = \{21 \times v \times (C \times e \times b + P_t)\} / \{21 \times v \times \sqrt{(C \times e \times b + P_t)}\}$

$= \{21 \times 0.7 \times (11400 \times 0.0125 \times 96 + 5300.43)\} / \{21 \times 0.7 \times \sqrt{11400 \times 0.0125 \times 96 + 5300.43)}\}$

$= 1384.26$ N

Now,

$P_{eff}' = (C_s * P_t) + P_d = (1.5 \times 5300.43) + 1384.26$

$= 9334.91$ N

Now,

$S_b / FOS = (519.86 \times m^2)/2 = (519.86 \times 8^2)/2$

$= 16640$ N

$P'_{eff} < S_b / FOS$ Hence design is safe.

Force analysis of gears -

For pinion,

$r_{mp} = (dp/2) + (b \times \sin\gamma / 2) = 40.82$ mm.

Now, as we know,

$P_T = 42403.42 / m = 42403.42 / 8$

$= 5299.74 \sim 5300$ N.

Now,

$P_S = P_T \times \tan(\phi) = 5300 \times \tan(20) = 1930$ N.

Now,

$P_r = P_T \times \tan(\phi) \times \cos(\gamma)$

$= 5300 \times \tan(20) \times \cos(78.43)$

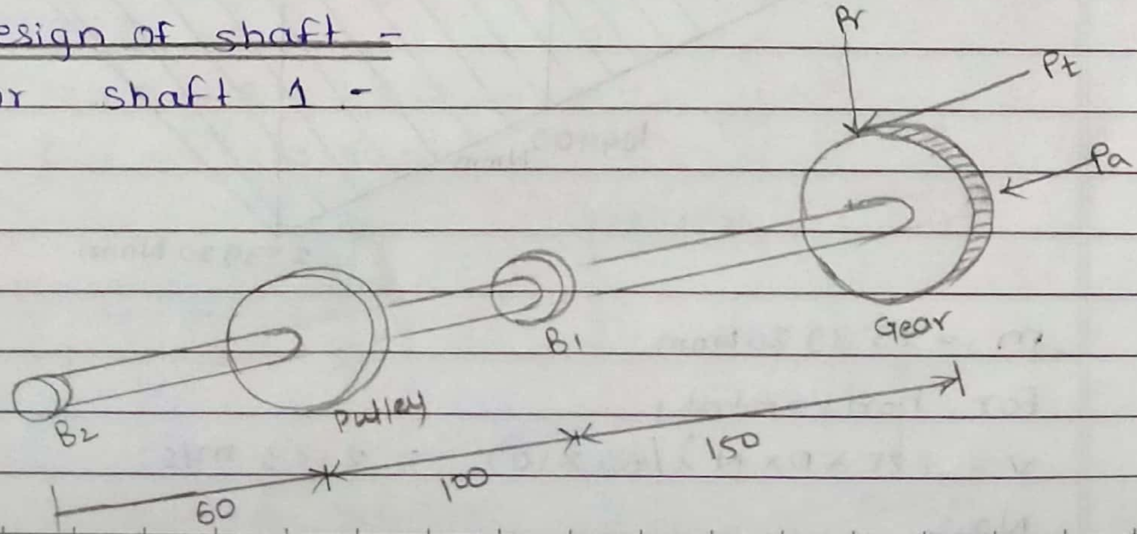
$= 1830.10$ N.

Now,

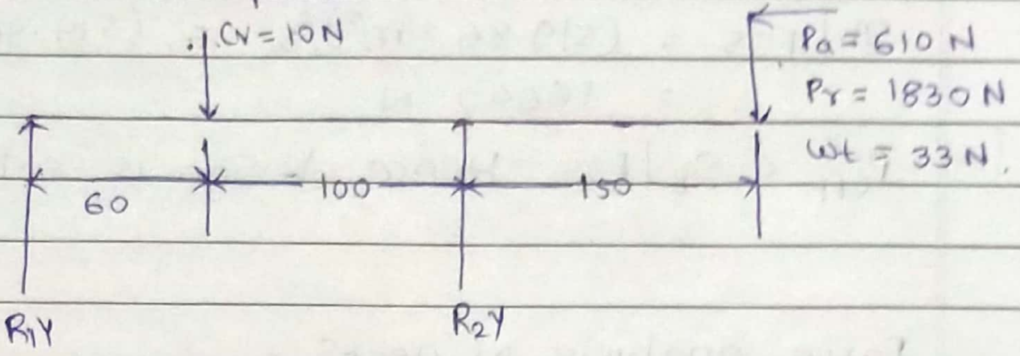
$P_a = P_s \times \sin(\gamma) = 1930 \times \sin(78.53) = 610$ N.

Design of shaft -

For shaft 1 -



FBD of vertical plane -



Taking moment about R_1,

$(60 \times 10) - (R_2V \times 160) + (7863 \times 310) = 0$

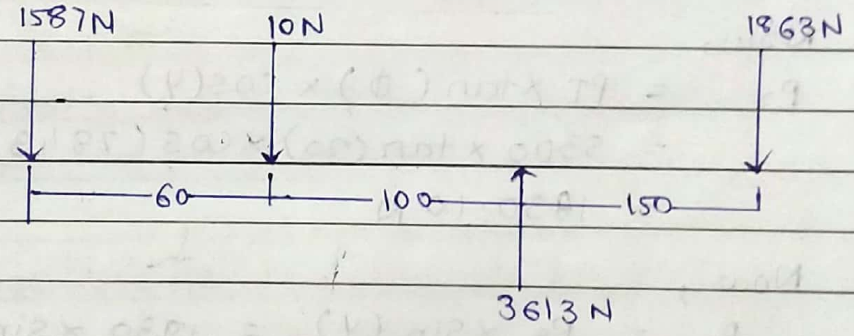
$R_2V = 3460{\cdot}8\ N$

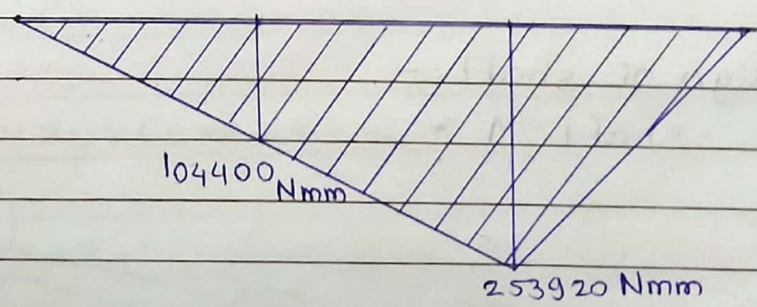
Now,

$\Sigma F_y = 0$

$R_1V + R_2V - 10 - 1830 - 33 = 0$

$R_1V = -1587\ N$





$m = 253920\ Nmm$

for horizontal,

$v = (\pi \times D \times N)/(60 \times 10^3) = 2.23\ m/s$

Now,

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Now,

$P = (T_1 - T_2) \times V$

$3.73 \times 10^3 = (T_1 - T_2) \times 2.23$

$T_1 - T_2 = 1672.64\ N$ —— (2)

Now,

$(T_1 / T_2) = e^{(\mu / \sin(\theta/2))\theta}$

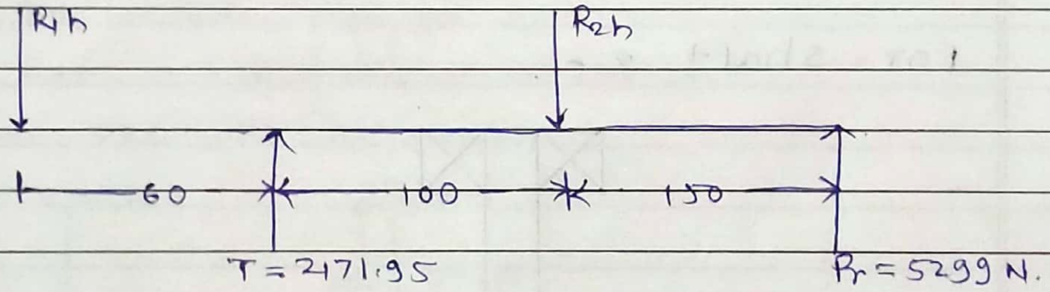
$T_1 = 7.7 \times T_2$

from (2)

$7.7\,T_2 - T_2 = 1672.64\ N$

$T_2 = 249.65\ N$ & $T_1 = 1922.30\ N$

$T = T_1 + T_2 = 2171.95\ N$



Taking moment about R_1H,

$-(217195 \times 60) + R_{2H} \times 160 - (5299 \times 310) = 0$

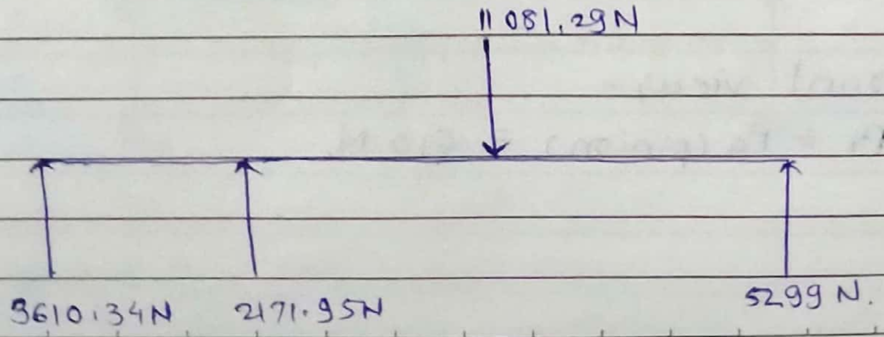
$R_{2H} = 11081.29\ N$

Now

$\Sigma F_y = 0$

$T + P_T - R_2H - R_1H = 0$

$R_1H = -3610.34\ N$



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$M_b = 794849.4$ Nmm

For keyways,

$0.18 \times S_{ut} = 0.18 \times 620 = 111.6 \ N/mm^2$

Now

$M_t = (60 \times 10^6 \times P) / (2 \times \pi \times N)$

$= (60 \times 10^6 \times 3.73) / (2\pi \times 120) = 296823.96$ Nmm.

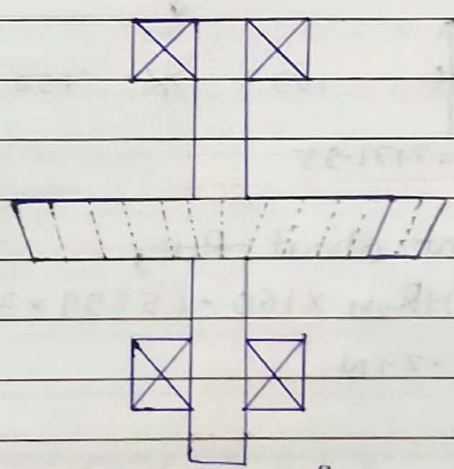
Assume, $K_b = 1.5$, $K_t = 1$.

$d^3 = [16 / (\pi \times \tau_{max})] \times \sqrt{[(K_b \times M_b)^2 + (K_t \times M_t)^2]}$

$d = 42.13$ mm ~ 45 mm

$d = 45$ mm

For shaft 2 -



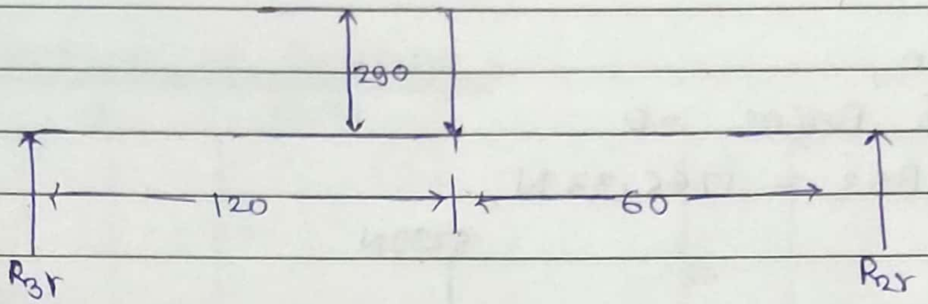
$\gamma_g = 90 - 9.46 = 80.54°$

$r_m = (d_g/2) + [b \times \sin(\gamma_g)]/2$

$= (84 \times 8/2) - [96 \times \sin(80.54)]/2 = 288.65$ mm.

Front view, -

$P_r = P_a$ (pinion) $= 610$ N.



Taking moment about R_{3f}

$(610 \times 120) - (R_{2f} \times 180) + (290 \times 1830) = 0$

$R_{2f} = 73200 / 180 = 3355$ N.

Now

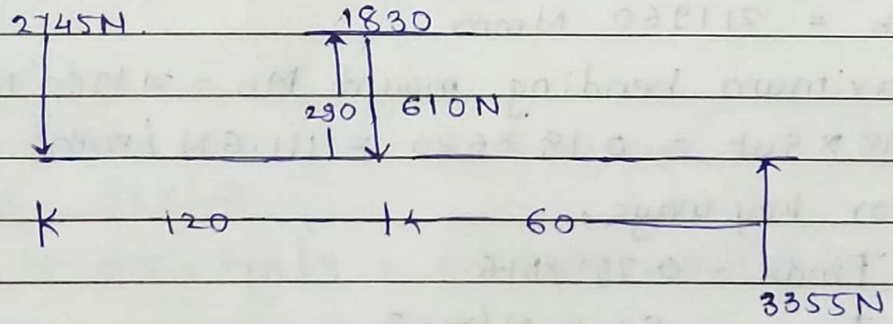
Σ forces $= 0$.

$610 = R_{3f} + R_{2f}$.

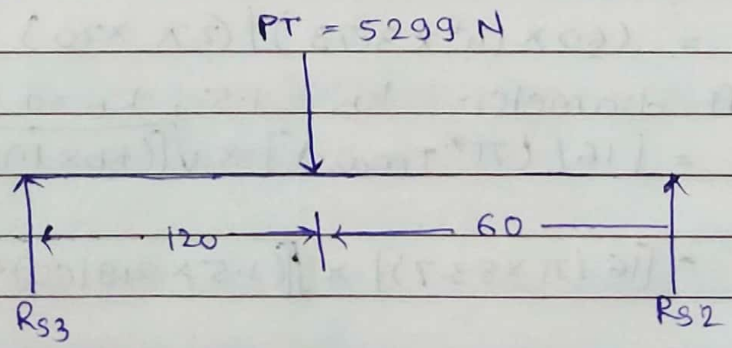
$R_{3f} = 610 - R_{2f}$

$R_{3f} = 610 - 3355 = -2745$ N.

Now, FBD will becomes,



Side view



Moment about R_{s3}.

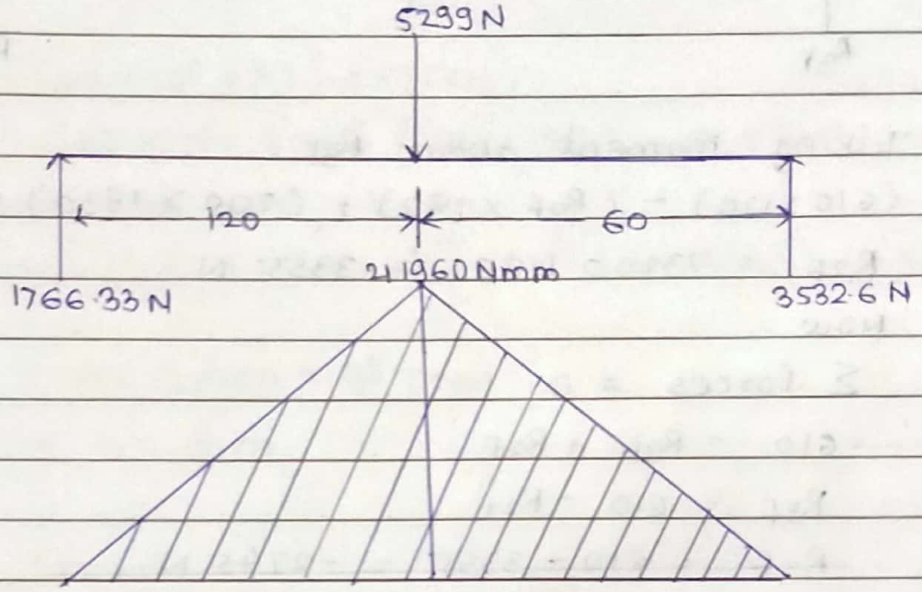
$(5299 \times 120) - (R_{s2} \times 180) = 0$.

$R_{s2} = 3532.6$ N.

Now,

Σ Forces $= 0$.

$R_{S3} = 1766.33$ N.



$M_b = 21960$ Nmm.

Maximum bending mount $M_b = 21960$ Nmm.

$0.18 \times S_{ut} = 0.18 \times 620 = 111.6 \, N/mm^2$.

For key ways,

$\tau_{max} = 0.75 \times 111.6$.

$\tau_{max} = 83.7 \, N/mm^2$

Torsional moment.

$M_t = (60 \times 10^6 \times 3.73) / (2\pi \times 20) = 1780943.8$ Nmm.

shaft diameter $K_b = 1.5$, $K_t = 1$.

$d^3 = [16 / (\pi * \tau_{max})] \times \sqrt{[(K_b \times M_b) + (K_t \times M_t)^2]}$

$= [16 / \pi \times 83.7)] \times \sqrt{[(1.5 \times 219160)^2 + (1 \times 1780943.8)^2]}$

$= 49.9 \sim 50$ mm.

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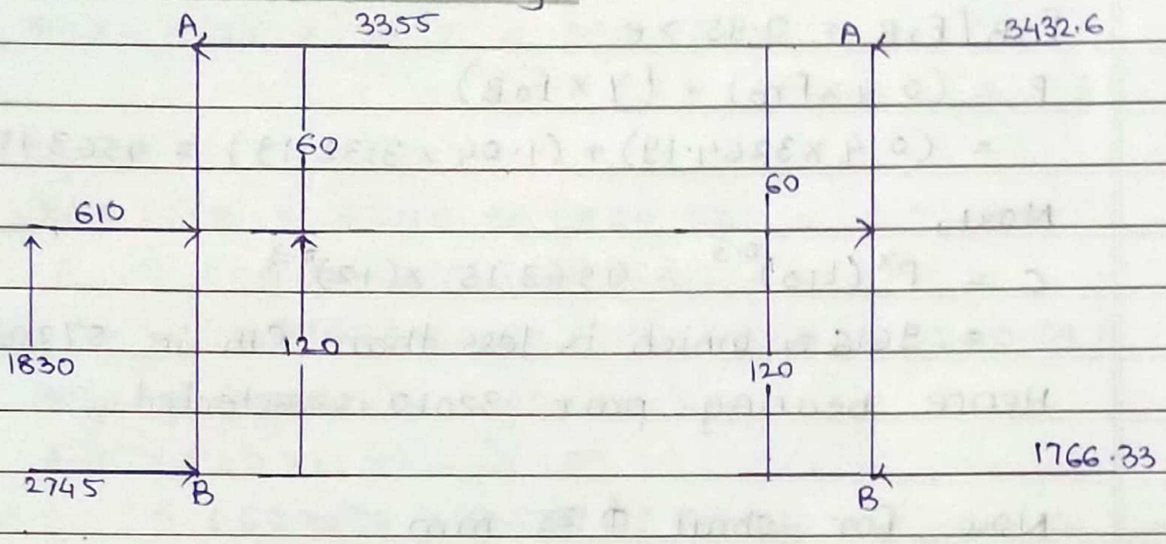
Selection of Bearing -



Front view   Side view.

$Fr_A = \sqrt{[(3555)^2 + (3432.6)^2]} = 4799.87\ N.$

$Fr_B = \sqrt{[(1766.33)^2 + (2745)^2]} = 3264.19\ N$

$Fr_A > Fr_B$

$K_A > 0$

Now

$Fa_B = 0.5 \times Fr_A / Y = 0.5 \times 4799.87 / 1.04 = 2307.63\ N.$

Now,

$Fa_B = 3132.19\ N.$

For A,

$e = Fa_A / Fr_A = 2307.63 / 4795.87 = 0.48\ N.$

$P = Fr = 4799.87\ N.$

Now,

$L_{10} = (60 \times n \times L_{10H}) / 10^6 = 60 \times 20 \times 10000 / 10^6.$

$= 12$ million rev.

Now,

$C = P \times (L_{10})^{0.3} = 4799.87 \times (12)^{0.3}$

$= 10115.42\ N$ which is less than C_{th} i.e. 57200 N.

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For B,

$F_{aB} / F_{rB} = 0.95 > e$

$P = (0.4 \times F_{rB}) + (Y \times F_{aB})$

$= (0.4 \times 3264.19) + (1.04 \times 3132.19) = 4563.15\ N.$

Now,

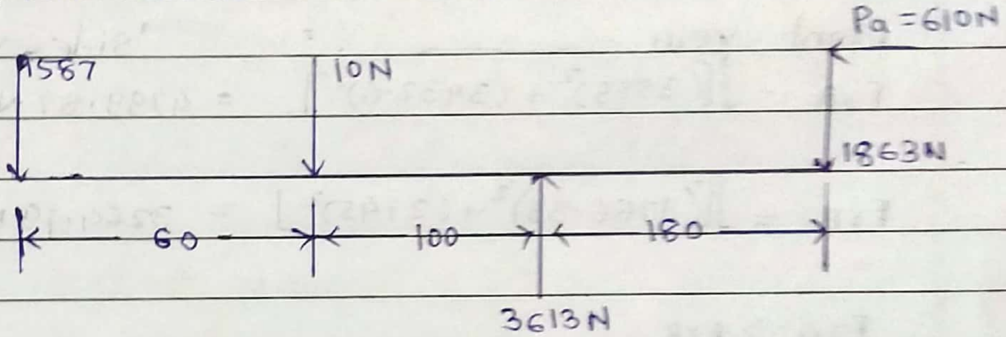
$C = P^{x} (L_{10})^{0.3} = 4563.15 \times (12)^{0.3}$

$= 9616\ N$ which is less than Cth ie 57200 N.

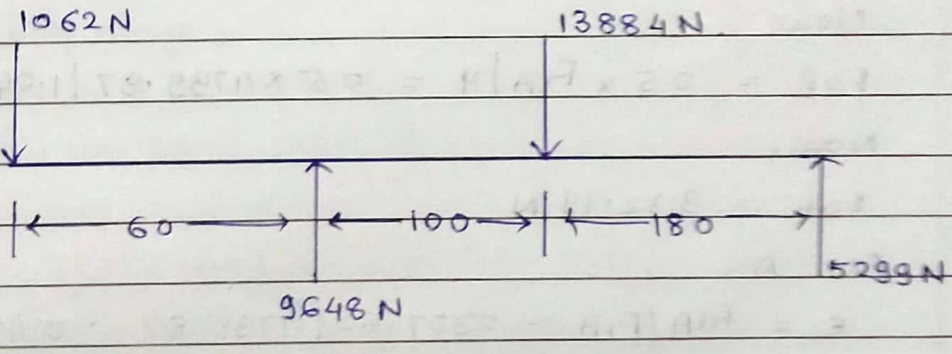
Hence bearing pair 32010 is selected.

Now, for shaft ϕ 25 mm

vertical Shaft



Horizontal shaft



$F_{rA} = \sqrt{[(1587)^2 + (1062)^2]} = 1909.56\ N$

$F_{rB} = \sqrt{[(1062)^2 + (13884)^2]} = 13924.56\ N.$

$K_a = 610\ N.$

$0.5 \times (F_{rB} - F_{rA}) = 6007.5\ N.$

Now,

$F_{aA} = F_{aB} - K_a = 5240.66\ N.$

Now,

$F_{aB} = 0.5 \times F_{rB} / Y = 5850.66\ N$

For A.

$F_{aA} / F_{rA} = 5240.66 / 1909.56 = 2.747 e$

$P = (0.4 \times F_{rA}) + (Y^{*} F_{aA}) =$

$= 0.4 \times 1909.66 + 1.19 \times 5240.66 = 7000.09\ N.$

assuming $(L_{10})_h = 10000$ hrs.

$L_{10} = (60 \times n \times L_{10H}) / 10^6$

$= (60 \times 120 \times 10{,}000) / 10^6 = 72$ million rev.

Now,

$C = P \times (L_{10})^{0.3} = 7000.09 \times (72)^{0.3}$

$= 25252.44\ N$

which is less than Cth ie. 55000 N.

For B,

$F_{aB} / F_{rB} = 5850.66 / 13924.56 = 0.42 = e.$

$P = F_{rB} = 13924.56\ N.$

Now,

$L_{10} = 72$ million rev.

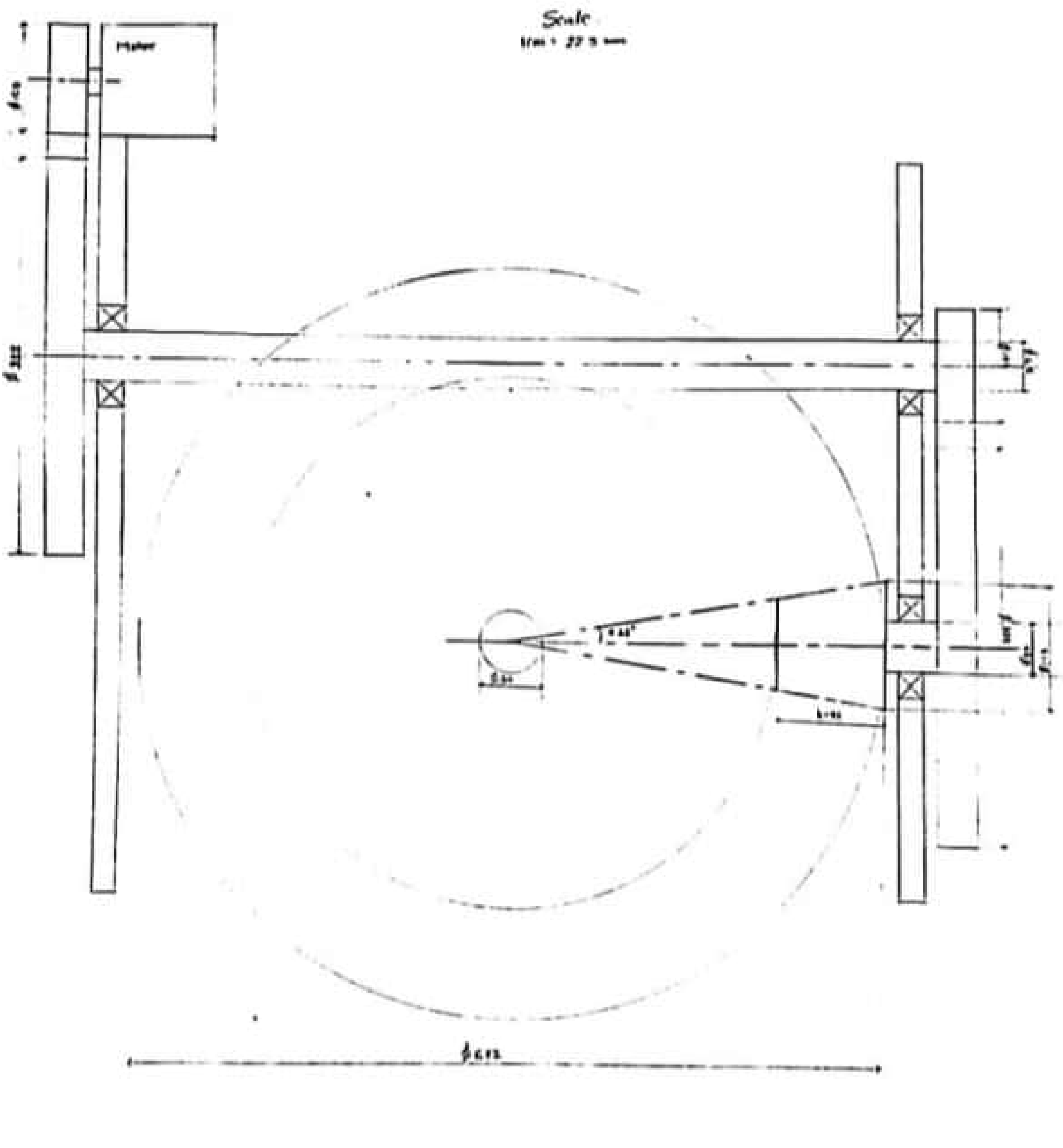
Hence,

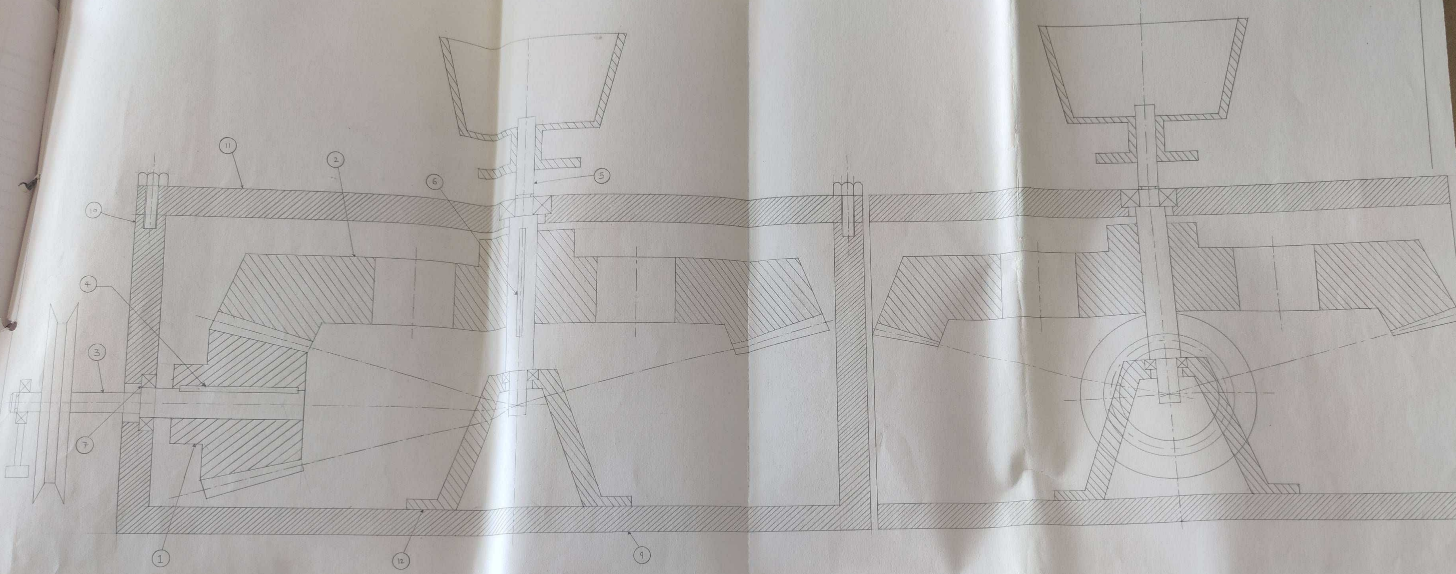
$C = P \times (L_{10})^{0.3} = 13924.56 \times (72)^{0.3}$

$= 50323.09\ N.$

which is less than Cth ie. 55000 N.

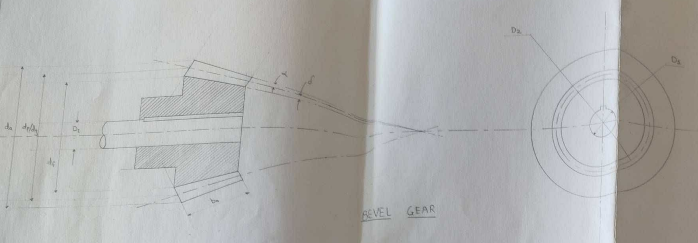
Hence bearing pair 32009 X is selected for both A & B.


Scale
Motor



SR. NO	PART NAME	MATL.	QTY.
12	SUPPORT BLOCK	FG 260	1
11	COVER PLATE	FG 260	1
10	CASTING	C.I.	1
09	BASE PLATE	C.I.	1
08	BALL BEARING - 2	STD	1
07	BALL BEARING - 1	STD	1
06	KEY - 2	C 30	1
05	SHAFT - 2	C 30	1
04	KEY - 1	C 30	1
03	SHAFT - 1	C 30	1
02	BEVEL GEAR	FE 620	1
01	BEVEL PINION	FE 620	1

KIT'S COLLEGE OF ENGINEERING	
NAME :- ABHISHEK RAVINDRA KULLUR	
T.E. MECH.	BEVEL GEAR ASSEMBLY
ROLL NO. :- 109	AND DETIALS
DIV :- B	



BEVEL GEAR

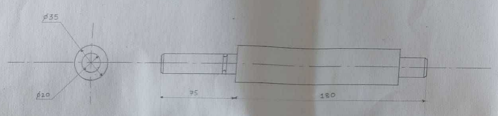
SR. NO	GEAR	TYPE	PCD (mm)	ADD. CONE DIA.	DED. CONE DIA.	FACE (b) WIDTH	MODULE (m)	PITCH ANGLE	ADD. (α) ANGLE	DED. (δ) ANGLE	SHAFT DIAMETER
01	BEVEL PINION	SOLID	$d_p = 135$	$d_{a_1} = 114.04$	$d_{f_1} = 152.76$	90	9 mm	$\gamma = 14.04°$	1.85°	2.22°	$d_1 = 35$ mm
02	BEVEL GEAR	SOLID	$d_g = 540$	$d_{a_2} = 544.36$	$d_{f_2} = 534.76$	90	9 mm	$\gamma = 75.96°$	1.85°	2.22°	$d_2 = 25$ mm

Addendum angle $(\alpha) = \tan^{-1}\left(\frac{m}{A_0}\right)$

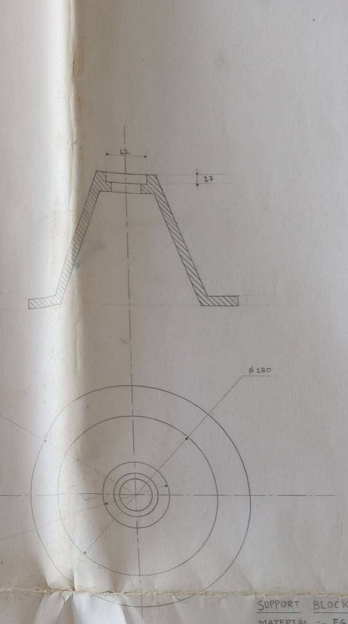
Dedendum angle $(\delta) = \tan^{-1}\left(\frac{1.2m}{A_0}\right)$

Addendum cone dia. $= d_a = d_p + 2m\cos\gamma$

Dedendum cone dia. $= d_f = d_p - 2(1.2m)\cos\gamma$.



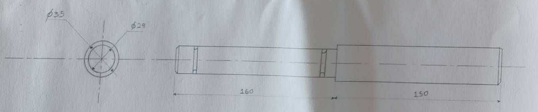
SHAFT - 2



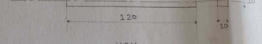
SUPPORT BLOCK

MATERIAL :- FG 260

SCALE :- 1:2.



SHAFT - 1 C-30



KEY

MATERIAL :- FG260

QTY :- 2