Design Project

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Project

Helical

Hi (Power): 3 KW $\qquad n_i$: 1500 rpm

i : 3 $\qquad L$: 10 Kh

Assumptions:- $- m_n = 2$ mm $\qquad - B$ = between (10-15) m_n

- Ψ (helix angle) = 30° $\qquad$ = 20 mm

- $\phi_n = 20°$ $\quad \phi_t = 22.95$ $\qquad$ 1st

- N_2 = 20 teeth $\qquad$ Pinion

- $N_3 = N_2 \times i = 60$

angular velocity = $\frac{2\pi n}{60} = \omega$

$= \frac{2 \times \pi \times 1500}{60}$

= 157.08 rad/s

Torque = $\frac{Power}{\omega} = \frac{3000}{157.08} = 19.1$ N.m

Diameter of pinion gear = $m_t N = \frac{m_n N}{\cos \Psi} = \frac{2 \times 20}{\cos(30)}$

D_P = 46.18 mm

Diameter of gear 3 = $D_P \times i = D_G = 138.56$

$\omega_t = \frac{Torque}{radius_2} = \frac{19.1}{0.0239} = 799.163$

$\omega_a = \omega_t \tan \Psi = 461.4$ N

$\omega_r = \omega_t \tan \phi_n = 290.87$ N

$m_t = \frac{m_n}{\cos(30)} = 2.309$

Bending Stress

$$\sigma = \frac{W_t}{B m_t J} K_v K_o K_s K_H K_B$$

$$= \frac{551.227}{20 * 3.414 *}$$

$J = J' *$ modifying factor (from graphs)

for pinion $= 0.5 * 0.99 = 0.495$ for gear $= 0.54 * 0.94$
$= 0.5076$

Calculating :- for both pinion & gear

→ K_v : tangential velocity (v) = ωr =

$= 157.08 * 0.0239$

$= 3.75$ m/s

$B = 0.825$ @ $Q_v = 6$

$A = 59.77$

$K_v = 1.365$

→ assuming uniform power source $K_o = 1$

→ $K_s = \frac{1}{K_b}$ from table using $m_n = 3mm$

$K_b = 0.956$ $\therefore K_s = \frac{1}{0.956} = 1.046$

→ K_H from table @ face width of 35 & accurate mounting $= 1.3$

→ assuming thick Rim $K_B = 1$

$\sigma_{BP} = 64.96$ MPa $\sigma_{BG} = 63.346$ MPa

Bending Strength :-

$\sigma_{FP} = \sigma'_{FP} \left(Y_N / Y_\theta Y_z \right)$

→ assuming Hardened Steel grade 1

$H_B = 200$ MPa

$\sigma'_{FP} = 0.703\ H_B + 113$

$= 253.6$

→ assuming 0.99 reliability $Y_z = 1$

$N_p =$ revolutions per hour * Life cycles

$= n_p * 60 * 10000$

$= 9 \times 10^8$ cycles

$N_G = n_G' * 60 * 10000 = \frac{N_p}{i} = 3 \times 10^8$

$Y_\theta = 1$ (temperature factor)

$Y_{NP} = 1.3558 * N^{-0.0178} = 0.9393$

$Y_{NG} = 0.95786$

$\sigma_{FP,P} = 238.20648$ MPa $\quad \sigma_{FP_g} = 242.913$ MPa

$\eta_P = 3.667 \qquad \eta_G = 3.8347$

Contact Stress :-

$$\sigma_c = C_p \sqrt{\frac{W_t}{b d_p I} K_v K_o K_s K_H C_f}$$

$C_p = 191$, $C_f = 1$ for both p & g
assuming the same material

$d_p = 46.18$ mm
$b = 20$ mm
$W_t = 799.163$ N

$$I = \frac{\cos\phi_t \sin\phi_t}{2 m_N} * \frac{i}{i+1}$$

$$m_N = \frac{P_N}{0.95 Z} = \qquad P_N = P_n \cos\phi_n$$
$$= \pi m_n \cos(20) = 5.9$$

$$Z = \sqrt{(r_p + a_p)^2 - (r_p \cos\phi_t)^2} + \sqrt{(r_g + a_g)^2 - (r_g \cos\phi_t)^2} - C \sin\phi_t = 9.472$$

13.614 ; 31.648 ; 35.787

$a_p = a_g = m_n = 2$ mm
$C = r_p + r_g = \frac{d_p}{2} + \frac{d_g}{2} = \frac{46.18}{2} + \frac{138.56}{2} = 92.37$
$= 23.09 + 69.28 = 92.37$

$m_N = 0.6558$

$I = 0.1815$ $\qquad \sigma_c = 540.17$ MPa

Contact Strength

$$\sigma_{HP} = \sigma_{HP}' \frac{Z_N C_H}{Y_\theta Y_Z}$$

$Y_\theta \to 1$, $Y_Z \to 1$ @ reliability 0.99

$\sigma_{HP}' = 2.22 * 200 + 200 = 644$

$C_H = 1$ assuming same material

$Z_N = 1.4488\ N^{-0.023}$

$Z_{N_P} = 0.9017$

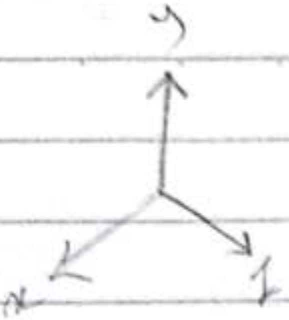
$Z_{NG} = 0.92477$

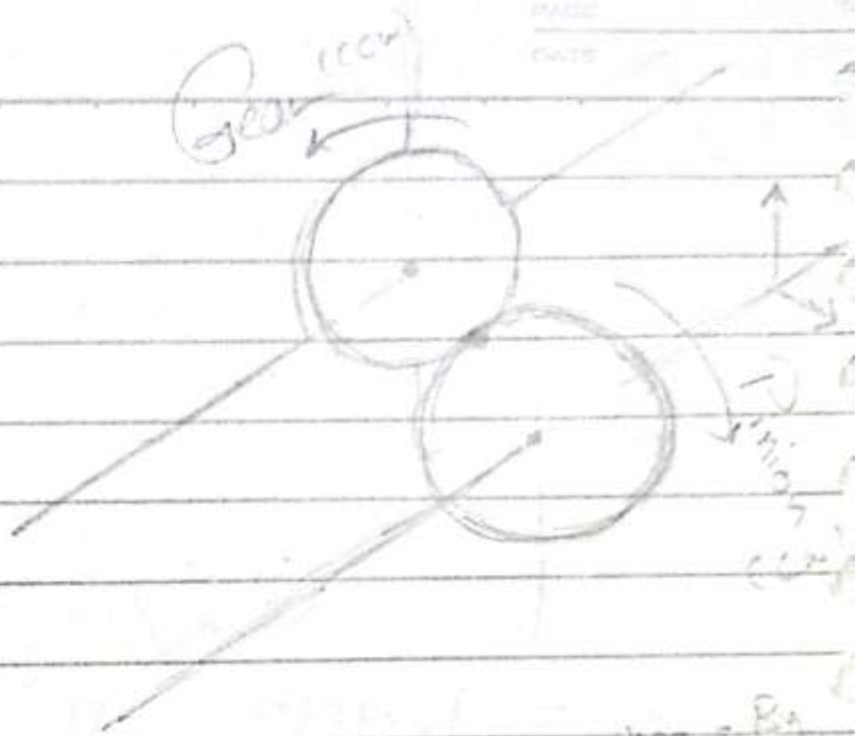
$\sigma_{HP_P} = 580.7$ MPa

$\sigma_{HP_G} = 595.55$

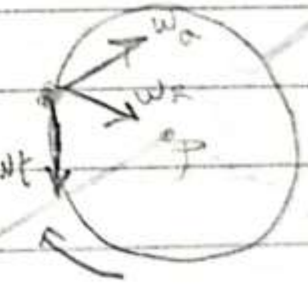
$\eta_P = \frac{\sigma_{HP_P}}{\sigma_c^2} = 1.1557$ $\quad \eta_G = 1.2155$

Scanned with CamScanner





Pinion $w_a = 461.4$ $w_r = 290.87$ $w_t = 799.43$



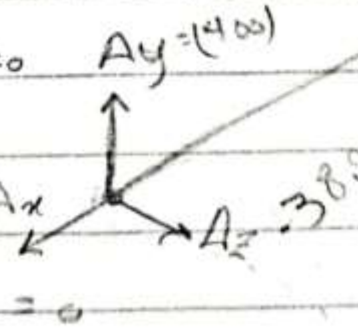
$\sum F_x = 0 \quad A_x - w_a = 0$

$A_x = w_a = 461.4$

$\sum F_y = 0 \quad A_y + B_y - w_t = 0$

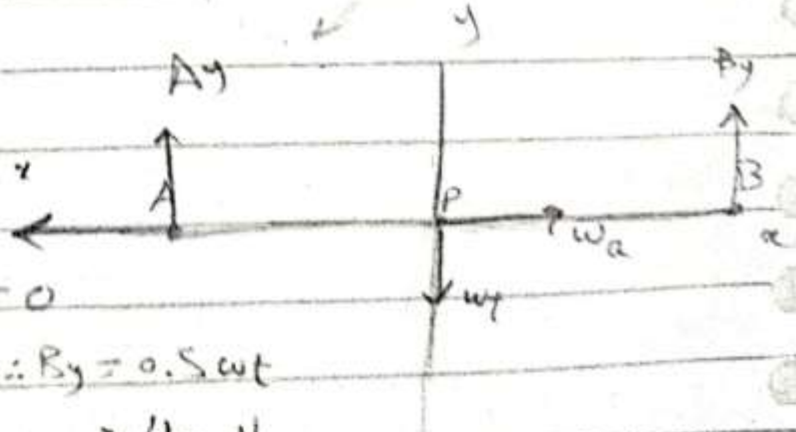
$A_y = w_t - B_y$

$A_y = 400N$



$\sum F_z = A_z + w_R + B_z = 0$

@ x-y plane



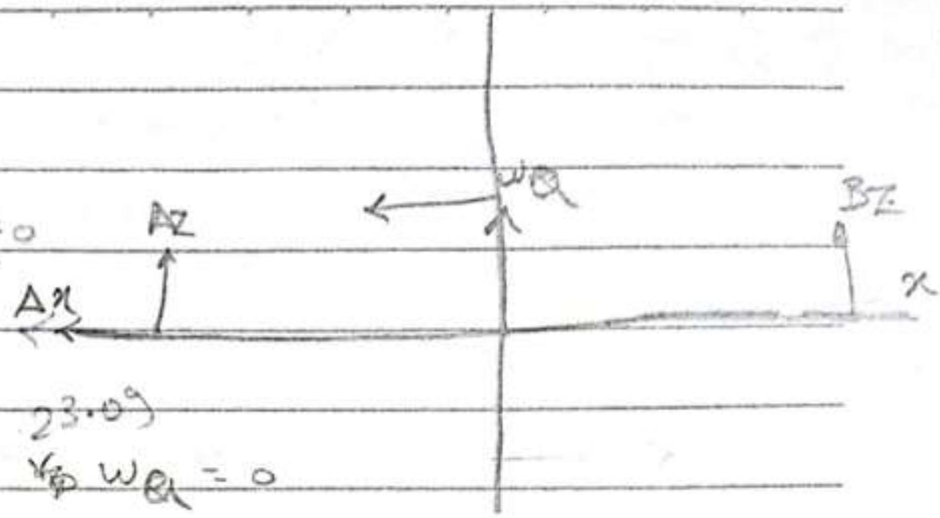
$M_A = -50w_t + 100B_y = 0$

$100B_y = 50w_t \quad \therefore B_y = 0.5w_t$

$= 400\,N$

@ x-z plane

$M_A = 0 \quad 50 W_R + 100 B_Z = 0$



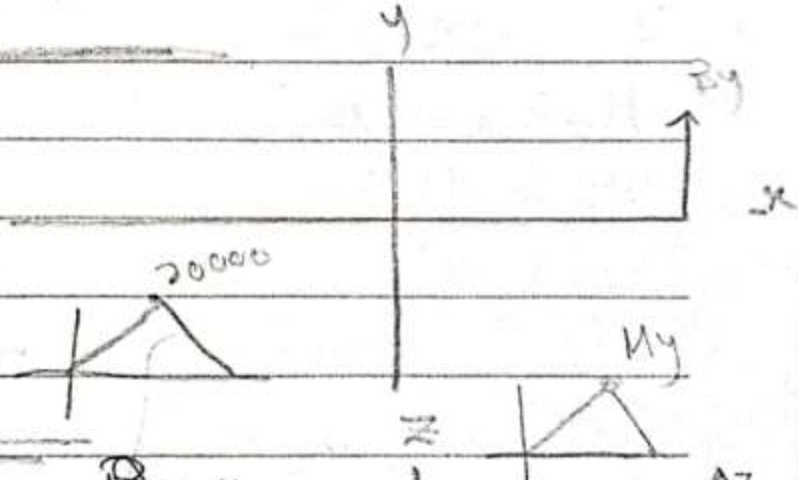
$50 W_R + 100 B_Z + 23.09\, W_Q = 0$

$B_Z = \dfrac{-50 W_R - 23.09 W_a}{100}$

$B_Z = -281.97$ #

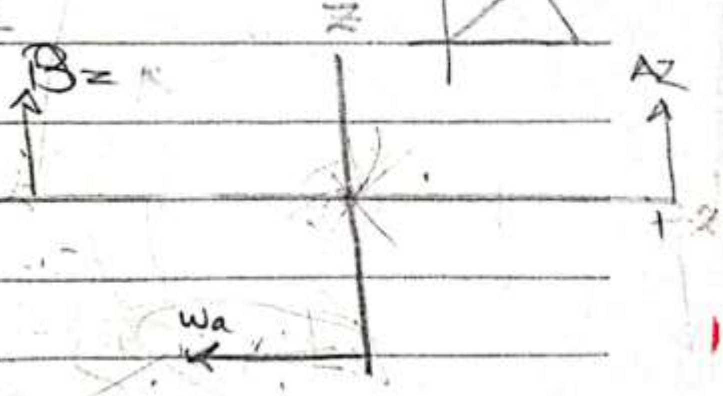
$A_Z = -38.9$ #

x-y plane



$M_{Bz} = 50 B_y$
$= 20000$ N.mm

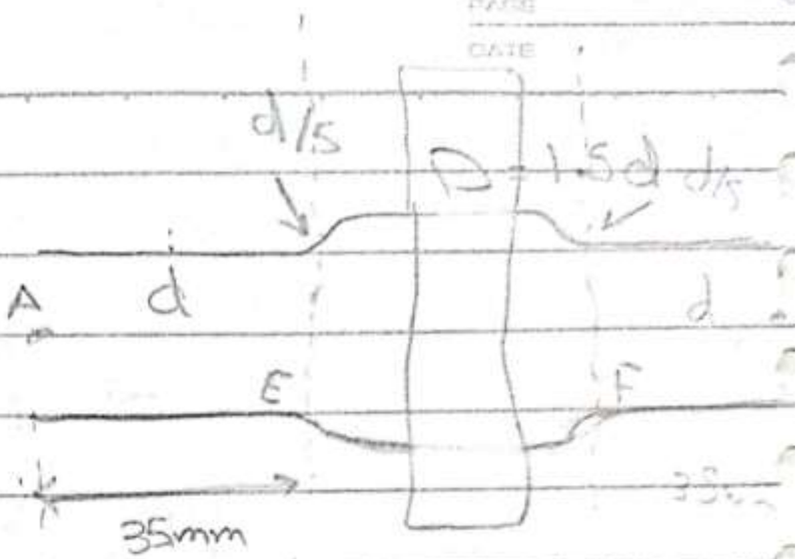
#



$M_y = -50 B_z - 23.09 W_a$
$= 1946.274$ N.mm

Res Moment = 30671.42 N.mm
= 30.671 N.m

Design of Shaft



∵ E & F are possible Critical points ∵ it has change of diameter

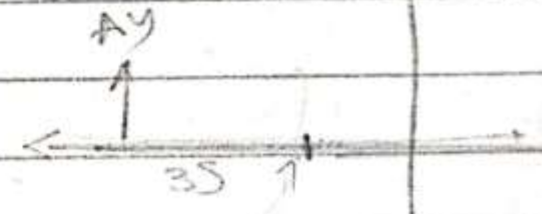
Calc. Moment @ E

in x-y plane

$M_E - 35A_y = 0$

$M_{Z_E} = +35A_y$

$M_E = 14000$



@ F $M_F = 14000$ as well ∵ $A_y = B_y$

∴ we will choose the more critical from (Z, x)

∵ B_Z is more than A_Z ∴ Moment on M_F is more than M_{y_E}. They. M_F is bigger & F is critical point

$M_{y_F} = 35 B_z = 35 * -252 = -8820$

M_F resultant $= \sqrt{8820^2 + 14000^2} = 16546$ N.mm

$= 16.546$ N.m

∴ it's Completely Reversed Cycle

$\sigma_m = 0$ & $\sigma_a = \frac{MC}{I} = K_f \frac{32M}{\pi d^3}$

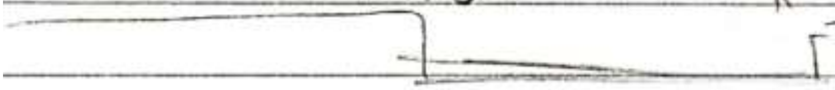
$\sigma_{a_{max}} = K_f \frac{168536.17}{d^3}$

Torque for whole shaft is the same

Torque $= w_t * r_p = 800 * 23.09$

Steady Torque $\tau_a = 0$ $= 18472$ N.mm

$\tau_m = \frac{16T}{\pi d^3} = \frac{16 * 18472}{\pi d^3} = K_{fs} \frac{94077.17}{d^3}$



K_f to get we need $\frac{D}{d} = 1.5$

$K_t =$ $\frac{r}{d} = \frac{1}{5}$

$K_t = 1.4$

for all diameters

$K_{ts} = 1.15$

assuming $q = q_s = 0.9$

$K_{fs} = 1.135$

$K_f = 1.36$ for all diameters iterations

$\sigma_{eff} = \sqrt{\sigma_a^2 + 3\tau^2} = \frac{S_y}{n}$

$S_y = 250$ MPa for steel, structural ASTM A36 Steel

$d = 11.0476$

$\simeq 11$

Scanned with CamScanner

Fatigue

$S_{ut} = 450$ MPa

$S_e' = 225$ MPa

Machined

$K_a = 0.8935$ Same daymen

@ $a = 4.51$ $b = -0.265$ $a(S_{ut})^b$

K_b for $d = 11.0476$

$K_b = 0.96$

$K_c = 1$ ∵ there is bending

$K_{d,e,f} = 1$

$K_e = 0.897$ @ 90%

$S_e = 0.8935 * 1.24 * S_e' * K_e \, d^{-0.107}$

$= 223.61 \, d^{-0.107}$ @ $d = 11$

$= 173$ MPa

ASME eliptic Line Method

new $d =$

$$\left\{ \frac{32n}{\pi} \sqrt{\left(\frac{(K_f M_a)^2}{S_e^2}\right) + \frac{\frac{3}{4}(K_{fs} T_m)^2}{S_y^2}} \right\}^{1/3}$$

new diameter $= 14.5$ mm $\underline{d_2}$

new $S_e = 167.97$ MPa , atleast $d = 14.588$

new diameter $= 14.588$ d_3

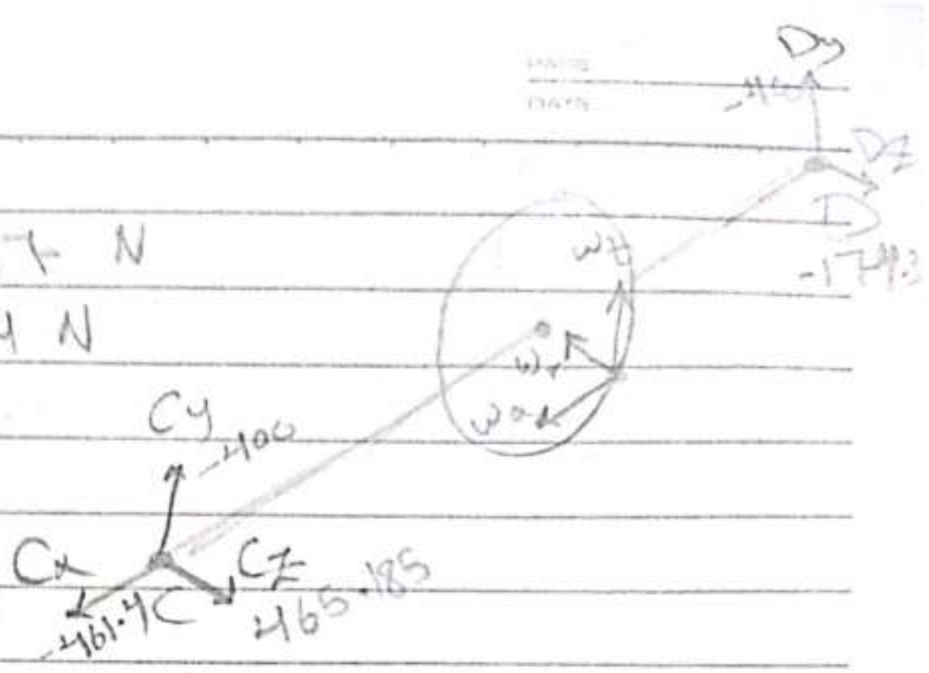
diff between d_2 & d_3 is < 0.2

Scanned with CamScanner

$w_t = 800N$

$w_r = 290.87$ N

$w_a = 461.4$ N



$\sum F_x = 0 \quad C_x + w_a = 0$

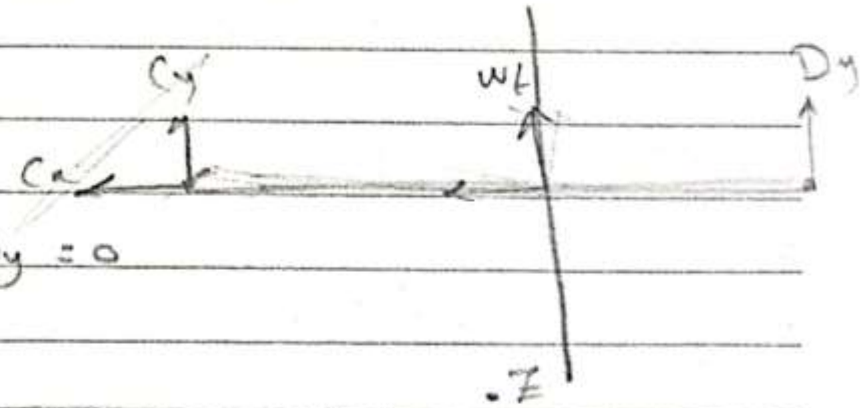
$C_x = -w_a$

$C_x = -461.4N$

$\sum F_y = 0 \quad C_y + w_t + D_y = 0$ ①

$\sum F_z = 0 \quad C_z + D_z - w_r = 0$

in (x-y) planes



$M_C = 50 w_t + 100 D_y = 0$

$D_y = -400$

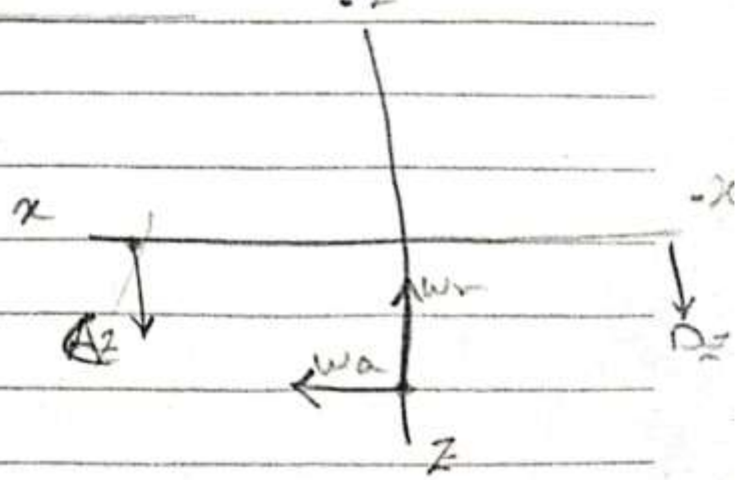
in (x-2) plane

$M_A = 50 w_r - \overset{69.3}{\cancel{}} w_a$

$- 100 D_z = 0$

$D_z = \cancel{}$

-174.3152 N



from 1 $C_y = -400$

$C_z = 465.185$ N

Moment diagram in (x-y) plane

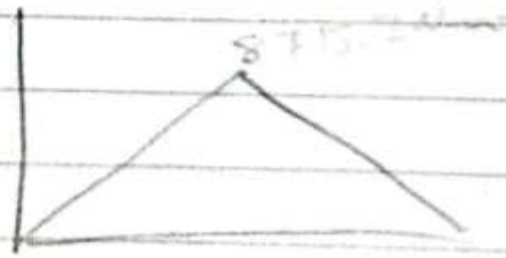
M @ gear center = -400 * 50 = -20000

in (x-z plane)

M @ gear center

$M_G + C_Z * 50 - W_a * 69.3$

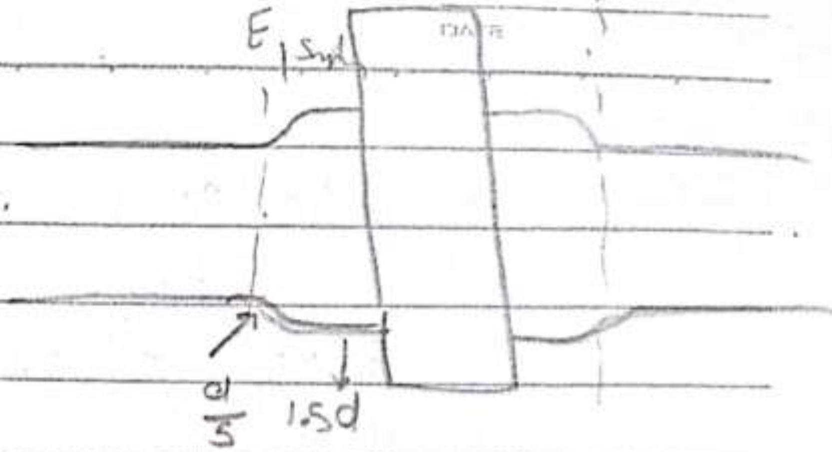
$M_G = 875.27$



Torque = $W_t * r$ = 800 * 69.3 = 55440

N.mm

Scanned with CamScanner



@ E Moment and @ F moment in x-y is same

$= -400 * 35 = -14000$

in $(x-z)$ C_z is $> D_z$

Thus E is Critical

Moment @ E in $(x-z)$

$M_z = 35 * 465.185 = 16281.475$

Moment Resultant = 21472.9 M_a

$K_f = 1.36$ $K_{fs} = 1.135$

$S_y = 250$ MPa $S_{ut} = 450$ MPa

$$d = \left(\frac{32n}{\pi S_y} \left(M_{max}^2 + \frac{3}{4} T_{max}^2 \right)^{\frac{1}{2}} \right)^{\frac{1}{3}}$$

$= 16.2435 \simeq 16$ mm

Scanned with CamScanner

fatigue

$S_e' = 225$ MPa

K_a is same $= 0.8935$
$K_b = 1.24\ d^{-0.107}$
reliability $= 90\%$ $K_c = 0.897$

$S_e = 223.61\ d^{-0.107}$

for $d = 16$

$S_{e_1} = 166.207$ MPa

new d_2 from ASME

$d_2 = 17.867$

$S_{e_2} = 164.256$

$d_3 = 17.895$ $|d_2 - d_3| < 0.2$

d_3 is @ least $17.895 \simeq 18$ mm
for gear's shaft

Coupling Diameter 15 mm

Using a Key :- for pinion's shaft

from shear stress for $n = 2$ S_y for steel Key $= 230$ MPa

$2 = \frac{S_{sy}}{\tau_s}$ $S_{sy} = 0.5 S_y = 115$ MPa

$2 = \frac{115}{\tau_s}$ $\tau_s = 57.5$

$\tau_s = \frac{F}{Area}$ $F \rightarrow$ Torque/radius shaft, Area $\rightarrow L * W$ of Key (L: face width, W: required)

Torque on pinion shaft $= 18472$ N.mm

radius of shaft $= 7.3$ mm

$L = 20$ mm

$W = \frac{18472}{20 * 8.5 * 57.5} = \cancel{2.2}$ mm $= 1.9$ mm $\simeq 2$ mm

from Bearing Stress

$n = \frac{\sigma_{cr}}{\sigma_b}$ ($\sigma_{cr} = S_y$) $2 = \frac{230}{\sigma_B}$ $\sigma_B = 115$ MPa

$\sigma_B = \frac{Force}{L * \frac{1}{2} height}$ $\therefore$ height $= \frac{18472}{0.5 \times 20 \times 7.3 \times 115}$

height $= 2.2$ mm

2 mm

for gear using a Key :

Shear Stress $\qquad d = 20\ mm$

$$\tau_s = 57.5 = \frac{\frac{Torque}{radius}}{L \times W}$$

$$w = \frac{Torque}{L \times radius \times 57.5}$$

width = 5.3565 mm

4.8 mm $\approx$ 5 mm

bearing Stress :-

$$\sigma_B = 115\ MPa = \frac{Torque}{L \times 0.5 \times h \times r}$$

height = 5.3565 mm

Bearings : $\qquad$ assume iso standards ballbearing

Pinion:

$$C_R = \overset{1.2}{K_A}\ \overset{F_r}{F_{eq}} \left(\frac{\overset{10000}{L_D}}{\underset{500}{L_R}} * \frac{\overset{1500}{n_D}}{\underset{\frac{1000}{3}}{D_R}} * \frac{1}{0.84} \right)^{\frac{1}{a}} \quad a \to 3$$

For Bearing B, $F_r\ \sqrt{B_y^2 + B_z^2} = 472.8\ N$

$C_R = 2000.84\ N \approx 2\ kN \qquad C_0 = 3.25$

Bearing type 6003

assuming 90% reliability

for Bearing $\underline{A}$ $\quad F_r = \sqrt{A_y^2 + A_z^2} = 401.9\,N$

$F_a = 461.24\,N$

$C_R = 1700.8 = 1.7\,KN \quad C = 6.05$

Type 6003 & $C_o = 3.25$

$\frac{F_a}{C_o} = \frac{0.461.4}{3.25} = 0.142 \qquad e = 0.316$

$X = 0.56$

$\frac{F_a}{F_r} > e \qquad Y = 1.38$

$\therefore F_{eq} = xF_r + yF_a = 834.112$

$C_R = 3529.9\ N$

$\because CR < C \quad \therefore$ bearing is Correct $\underline{6003}$

Gear for Bearing $\underline{D}$

$F_r = \sqrt{D_y^2 + D_z^2} = 436.2$

diameter 18 mm

$C_R = 2662.3\,N = 2.66\,KN \qquad C = 9.36$

Bearing Type 6004 $\qquad C_o = 5$

For Bearing $\underline{C}$ $F_r = \sqrt{C_y^2 + C_z^2} = 613.37N$

$F_a = 2461.4N$

$CR = 2595.7\ N = 2.595\ KN$ $C = 9.36$

$C_0 = 5.00$

$\frac{F_a}{C_0}$ 0.0923 $e = 0.2849$

$x = 0.56$

$y = 1.5257$

$\frac{F_a}{F_r} = 1.33 > e$

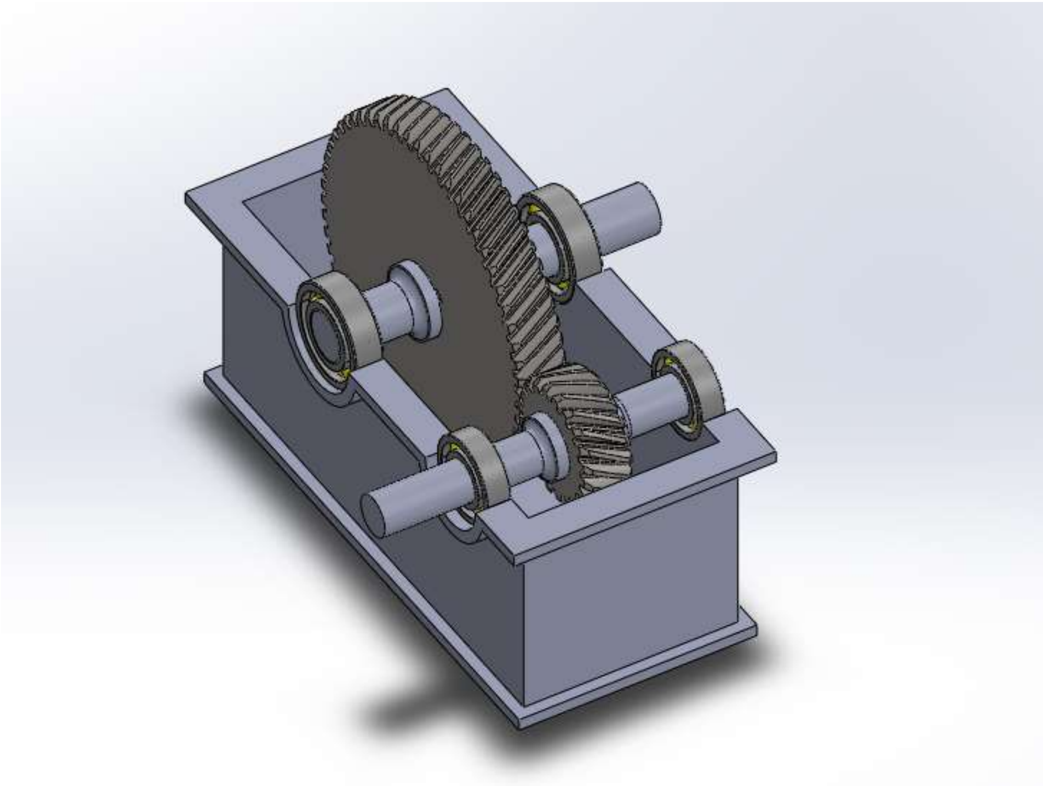
$F_{eq} = xF_r + yF_a$

$= 1047.45\ N$

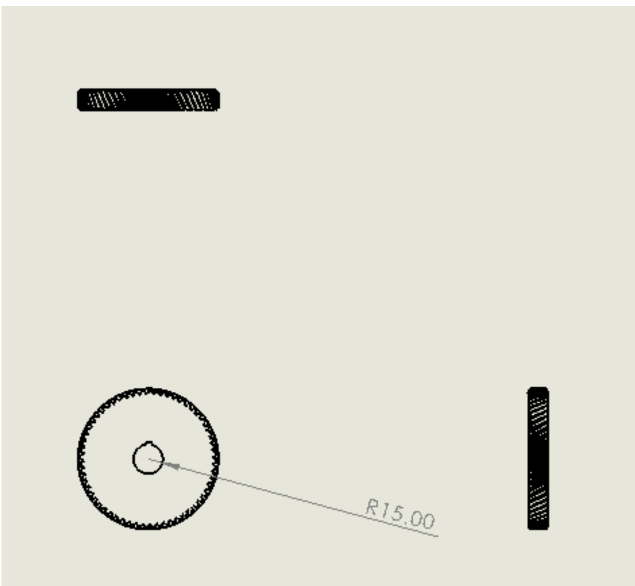
$CR = 4432.72\ N < \underline{C}$

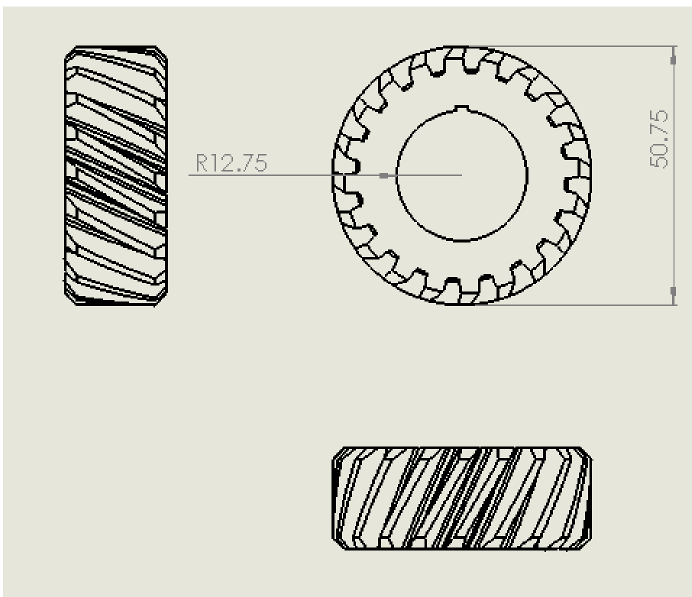
$\therefore$ bearing Types 6004 is suitable

3D Assembly

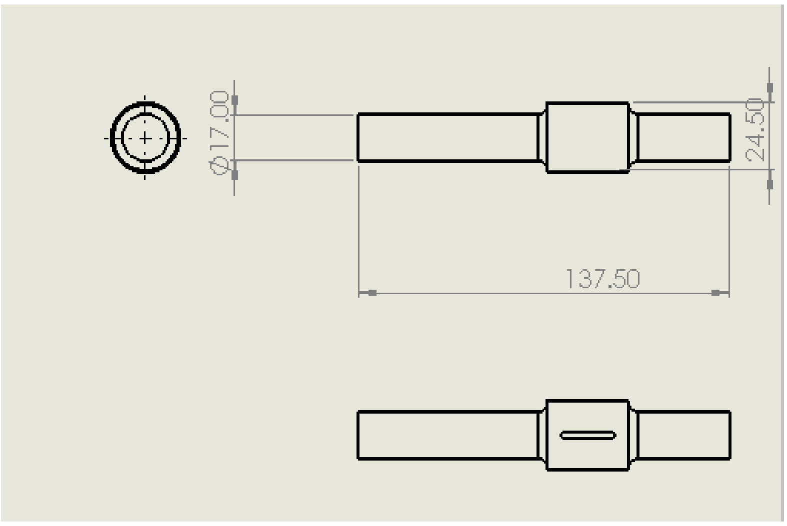


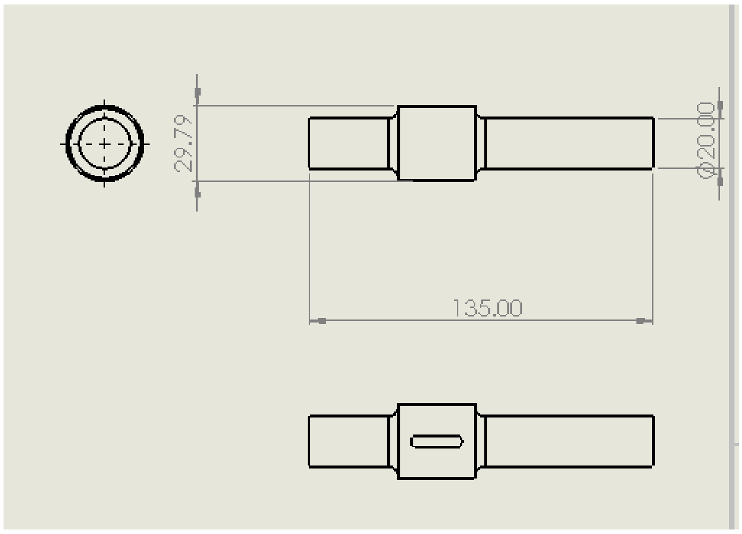
Gear and Pinion



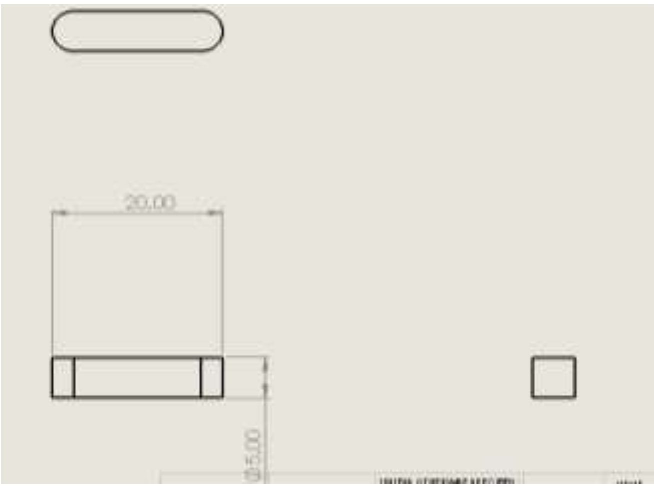


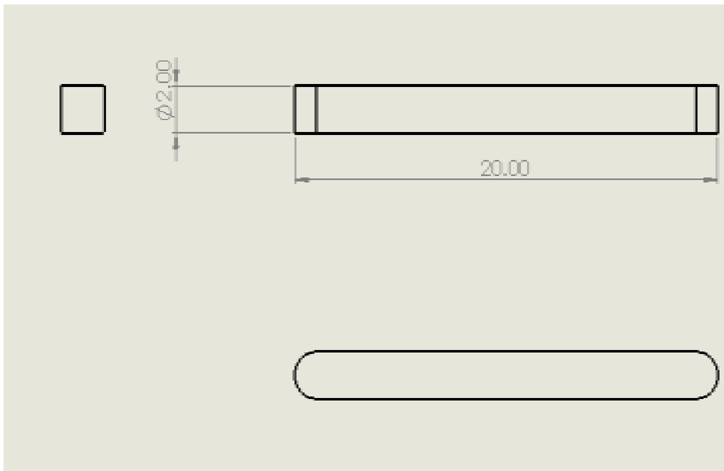
Shafts



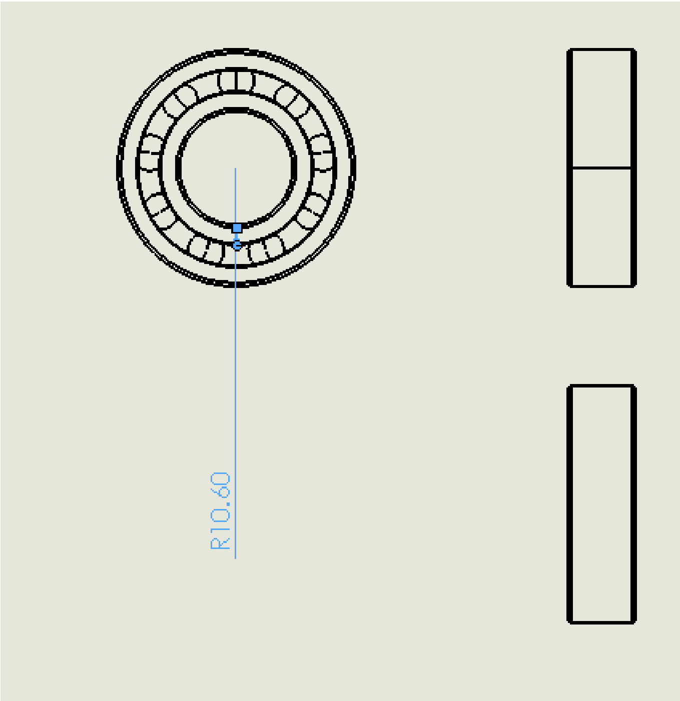


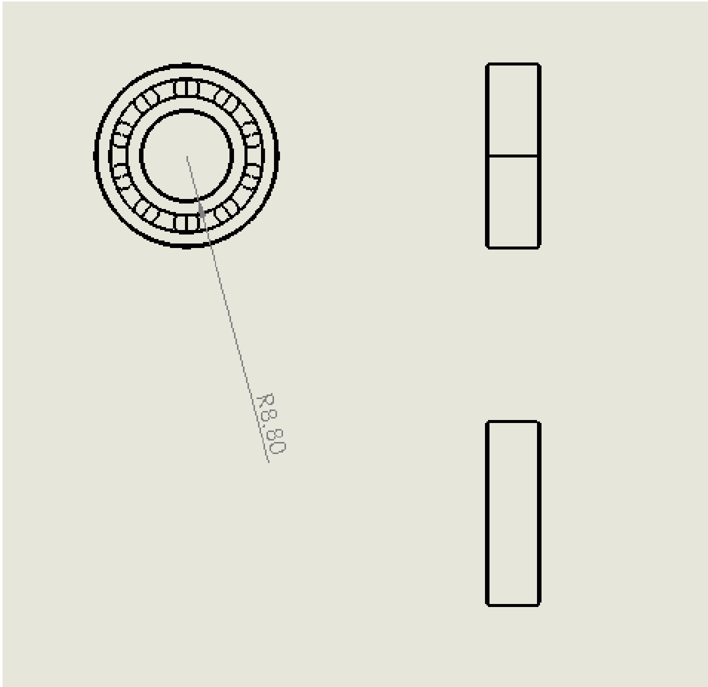
Keys





Bearings





Casing