NEAR EAST UNIVERSITY
37 Years in Education

MOD 308 Makina Tasarımı II
ECC 308 Machine Design II

Spring 2018 Midterm examination

March 31, 2018, 09:00 - 11:00

Name: Mahmoud Shaban Hashkel

ID Number: 20173983

Question No	Max. Point	Point
1	20	20
2	40	38
3	20	20
4	40	40
Total	120	

Instructions

1. Yükseköğretim Kurumları 2015 Öğrenci Disiplin Yönetmeliği Madde 5-d ve 7-e'ye göre "sınavlarda kopyaya teşebbüs veya kopya çekmek yapmak veya yaptırmak veya bunlara teşebbüs etmek" fiilinin suçu YÜKSEKÖĞRETİM KURUMUNDAN BİR VEYA İKİ YARIYIL İÇİN UZAKLAŞTIRMA cezasıdır.

UYARI VE KURALLARI OKUDUM.

Signature:

Good luck!

Question 1 (20 points)

Boşlukları doldurunuz
Fill the blanks

① working depth (hw)
② addendum (a)
③ dedendum (bd)
④ face width (length)
⑤ addendum circle.
⑥ circular pitch (p)
⑦ teeth width.
⑧ ~~width of spacing~~ Clearance (f)
⑨ fillet radius
⑩ dedendum circuit
⑪ width of spacing
⑫ pitch circuit.
⑬ ~~bottom~~ Top land.
⑭ ~~face~~ teeth faces
⑮ bottom land.
⑯ Clearance circuit.
⑰ -

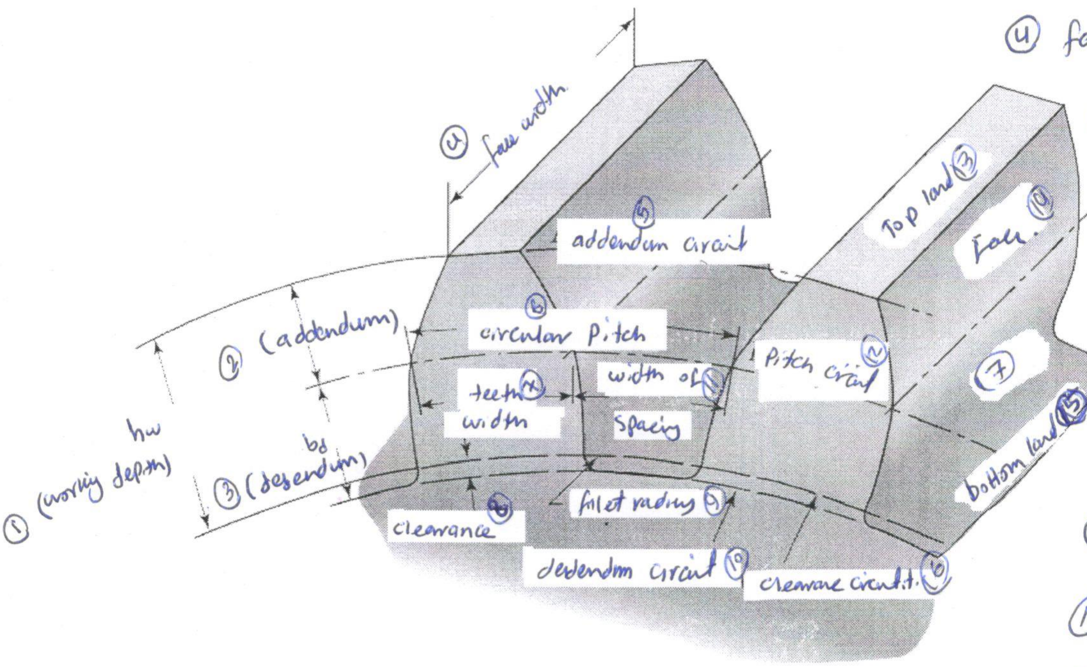


Figure 1: Figure for Question 1

Question 2 (40 points)

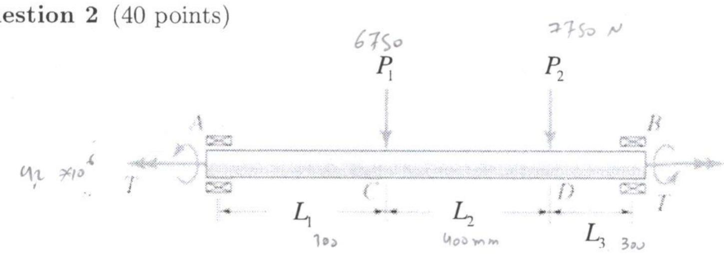


Figure 2: Figure for Question 2

Şekil 2'de gösteriln ve D çapına sahip içi dolu mil dönmekte ve şekilde gösterilen yüklere maruz kalmaktadır.

(a) Mil için serbest cisim diyagramını çiziniz

(b) A ve B rulmanlarındaki reaksiyon kuvvetlerini bulunuz

(c) Kesme $(V - x)$, moment $(M - x)$ ve tork $(T - x)$ diyagramlarını çiziniz

(d) Mil üzerindeki kritik noktayı tesbit ederek bu noktadaki moment ve torku bulunuz

(e) Maksimum şekil değiştirme enerjisi teoremini Goodman teoremi ile birleştirerek emniyet faktörünü $n^{Goodman}$ bulunuz ve şekil üzerinde gösteriniz

(f) Maksimum kayma gerilmesi teoremini Soderberg teoremi ile birleştirerek emniyet faktörünü $n^{Soderberg}$ bulunuz ve şekil üzerinde gösteriniz

Verilenler: $D = 40$ mm, $S_y = 850$ MPa, $S_{ut} = 1500$ MPa, $T = 4.2 \times 10^6$ N.mm,
$P_1 = 6750$ N, $P_1 = 7750$ N, $L_1 = 700$ mm, $L_2 = 400$ mm, $L_3 = 300$ mm

Yüksek şoklu (heavy shocks) tork % 20 oranında sabit değerden dalgalanmaktadır ve güvenirlilik oranı % 99.9'dır. Maksimum çalışma sıcaklığı 950 °F'dır
Tasarım Kararı: Mil sıcak haddelenmiş çentiksiz çelikten (hot-rolled from an unnothced steel) imal edilmiştir. $K_f = 1$

$$\sigma_a = \frac{32M_a}{\pi D^3}, \quad \sigma_m = \frac{32M_m}{\pi D^3}, \quad \tau_a = \frac{16T_a}{\pi D^3}, \quad \tau_m = \frac{16T_m}{\pi D^3}$$

Maksimum şekil değiştirme enerjisi teoremi Goodman teoremi ile birleştirerek emniyet faktörü $n^{Goodman}$ aşağıdaki formüllerden bulunabilir

$$\sigma_{ea} = \sqrt{\sigma_a^2 + 3\tau_a^2}, \quad \sigma_{em} = \sqrt{\sigma_m^2 + 3\tau_m^2}, \quad \frac{1}{n^{Goodman}} = \frac{\sigma_{em}}{S_{ut}} + \frac{\sigma_{ea}}{S_e}$$

Maksimum kayma gerilmesi teoremi Soderberg teoremi ile birleştirerek emniyet faktörü aşağıdaki formüllerden bulunabilir

$$\sigma_{ea} = \sqrt{\sigma_a^2 + 4\tau_a^2}, \quad \sigma_{em} = \sqrt{\sigma_m^2 + 4\tau_m^2}, \quad \frac{1}{n^{Soderberg}} = \frac{\sigma_{em}}{S_y} + \frac{\sigma_{ea}}{S_e}$$

A solid shaft of diameter D rotates and supports the loading shown in Figure 2.

(a) Draw the free body diagram of the shaft

(b) Find the reactions at A and B

(c) Draw the shear $(V - x)$, moment $(M - x)$ and torque $(T - x)$ diagrams

(d) Find the critical point on the shaft and determine the moment and torque at that point

(e) Calculate the factor of safety $n^{Goodman}$ for the shaft using the maximum energy of distortion theory of failure combined with the Goodman criterion and show it in a diagram

(f) Calculate the factor of safety n for the shaft using the maximum shear stress theory of failure combined with the Soderberg criterion and show it in a diagram

Given: $D = 40$ mm, $S_y = 850$ MPa, $S_{ut} = 1500$ MPa, $T = 4.2 \times 10^6$ N.mm,
$P_1 = 6750$ N, $P_2 = 7750$ N, $L_1 = 700$ mm, $L_2 = 400$ mm, $L_3 = 300$ mm
Torque invloves heavy shocks and fluctuates % 20 each way from the mean value and the survival rate is 99.9%. The maximum operating temperature is 950 °F
Design Decision: The shaft is to be hot-rolled from an unnothced steel. $K_f = 1$
The maximum energy of distortion theory of failure combined with the Goodman criterion

$$\sigma_{ea} = \sqrt{\sigma_a^2 + 3\tau_a^2}, \quad \sigma_{em} = \sqrt{\sigma_m^2 + 3\tau_m^2}, \quad \frac{1}{n^{Goodman}} = \frac{\sigma_{em}}{S_{ut}} + \frac{\sigma_{ea}}{S_e}$$

The maximum shear stress theory of failure combined with the Soderberg criterion

$$\sigma_{ea} = \sqrt{\sigma_a^2 + 4\tau_a^2}, \quad \sigma_{em} = \sqrt{\sigma_m^2 + 4\tau_m^2}, \quad \frac{1}{n^{Soderberg}} = \frac{\sigma_{em}}{S_y} + \frac{\sigma_{ea}}{S_e}$$

Question 3 (20 points)

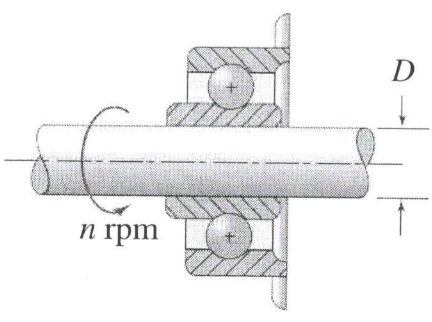


Figure 3: Figure for Question 3

2. sorudaki milin 2500 dev/dak (rpm) hızla dönmekte olduğuna göre A ve B mafsallarındaki (02-seirisi) derin yivli bilyalı rulmanların (deep groove ball bearing) ömürlerini saat cinsinden hesaplayınız. Dış bilezik dönmekte (outer ring rotates)
Verilenler: $D = 40$ mm, $n = 2500$ rpm.

For the (02-series) deep groove ball bearings A and B used in Question 2, Calculate the rating life in hours. The outer ring rotates and the loads applied are steady
Given: $D = 40$ mm, $n = 2500$ rpm.

Question 4 (40 points)

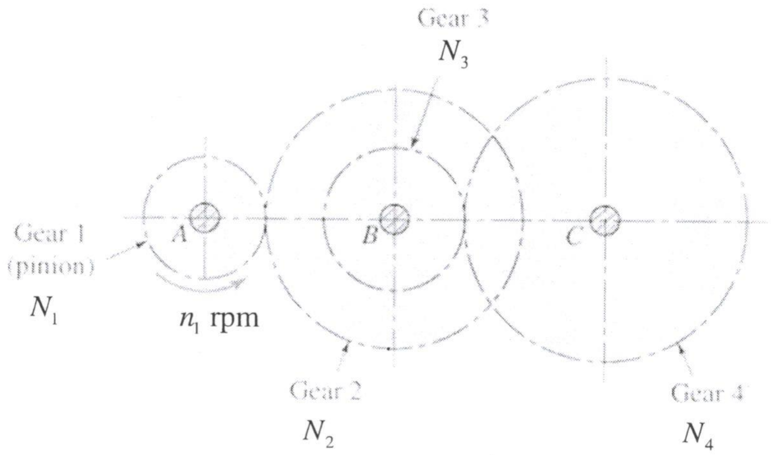


Figure 4: Figure for Question 4

Şekil 4'te gösterilen dişliler (P = 6 teeth/inch) çapsal adıma ve 25 °'lik basınç açısına sahiptir.

(a) Her bir dişli için serbest cisim diyagramını çiziniz

(b) Her bir dişlinin çapını bulunuz.

(c) Radyal ve teğetsel kuvvetleri bulunuz Determine the tangential and radial forces for each gear

(d) 1. dişli ve 3. dişli'deki torkları bulunuz $T_1 = ?$, ve $T_4 = ?$

(e) A, B ve C millerindeki reaksiyon kuvvetlerini bulunuz.

Verilenler: $n_1 = 2500$ rpm, $N_1 = 24$, $N_2 = 42$, $N_3 = 30$, $N_4 = 60$.

The gears shown in Figure 4 have a diametral pitch (P = 6 teeth/inch) and 25 ° pressure angle. Determine

(a) Draw free body diagrams for each gear

(b) Determine the diameters for each gear

(c) Determine the tangential and radial forces for each gear

(d) Determine the torques on gear 1 $T_1 = ?$, and gear 4 $T_4 = ?$

(e) Determine the reactions at the shafts A, B and C

Given: $n_1 = 2500$ rpm, $N_1 = 24$, $N_2 = 42$, $N_3 = 30$, $N_4 = 60$. , hp = 40 hp

Name: Mahmoud Shaban Ramadan Hashkel

Student no: 20173983 Subject: Machine design II

(Q2)

Given: $D = 40\,mm$

$S_y = 850\ MPa$

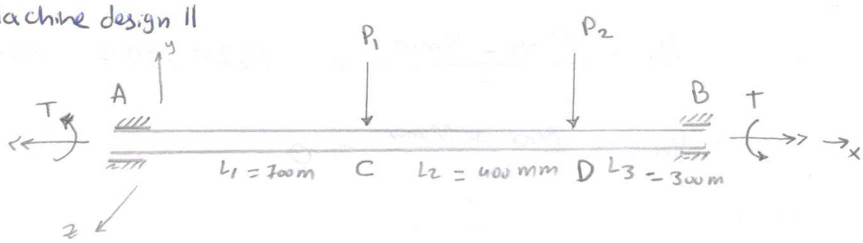
$S_{ut} = 1500\ MPa$

$T = 4.2 * 10^6\ N.mm$, $K_f = 1$

$P_1 = 6750\ N$, $P_2 = 7750\ N$

fluctuation of $T = 20\%$, survival rate $= 99.9\%$

hot rolled - unnotched steel.



· in plane of (x-y) :-

$\Sigma M_A = 0$ ↻+

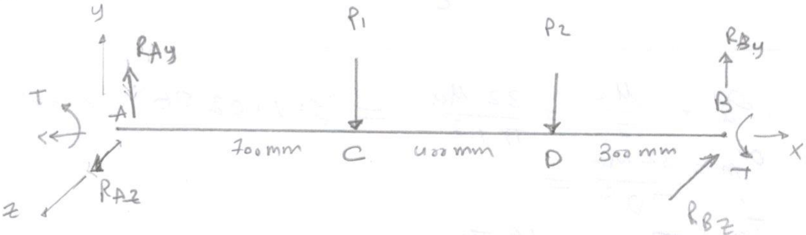
∴ $6750(700) + 7750(1100) = R_{By}(1400)$

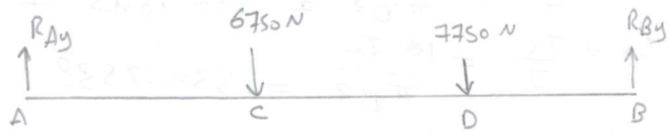
∴ $R_{By} = (9464.285\,N)$ #

$\Sigma F_y = 0$ ↓+

∴ $R_{Ay} + R_{By} = 6750 + 7750$

$R_{Ay} = (5035.714\ N)$ #





(X-y) plane

· in plane (x-z) :-

$\Sigma M_A = 0$ ↻+

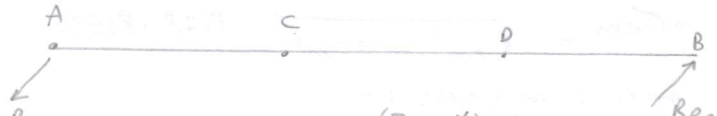
∴ $R_{Bz}(1400) = 0$

∴ $(R_{Bz} = 0)$

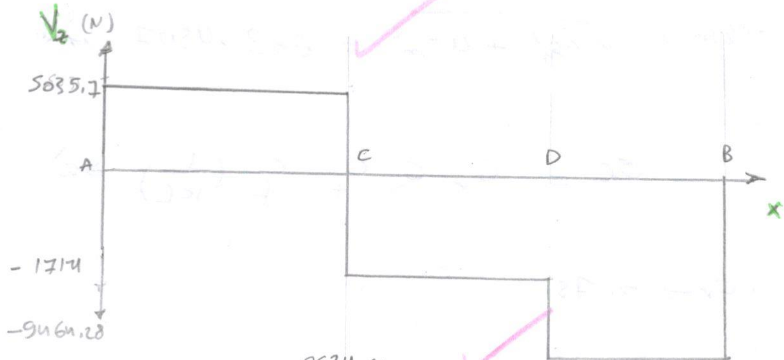
∵ $\Sigma F_z = 0$

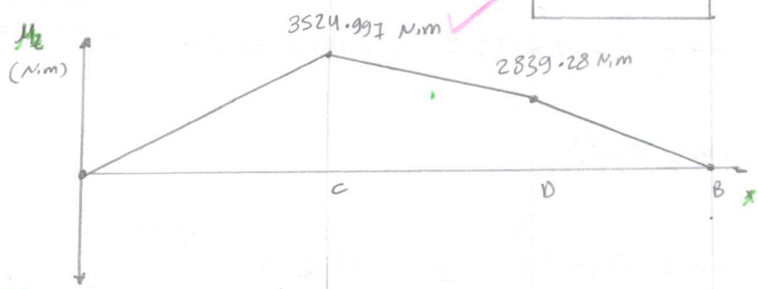
∴ $R_{Bz} - R_{Az} = 0$

∴ $(R_{Az} = 0)$



(Z - X) plane

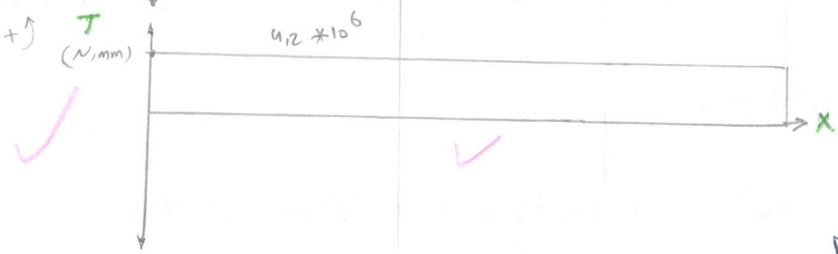




· Critical Point :-

$M_{max} = M_z = [3524.997\ N.m]$ #

$T_{max} = T = [4.2 * 10^6\ N.mm]$ #



∴ Point (C) is the critical point.

Q(2) contin...

- $M_a = \frac{M_{max} - M_{min}}{2} = 3524,997 \; N.m = (3524997 \; N.mm)$

$: M_{max} = 3524,997 \; N.mm$

$: M_{min} = -3524,997 \; N.mm$

- $M_m = \frac{M_{ax} + M_{min}}{2} = 0$

$T_{max} = T + \frac{20}{100} T = (5040000 \; N.mm)$

$T_{min} = T - \frac{20}{100} T = (3360000 \; N.mm)$

- $T_a = \frac{T_{max} - T_{min}}{2} = (8.4 \times 10^5 \; N.mm)$
- $T_m = \frac{T_{max} + T_{min}}{2} = (4.2 \times 10^6 \; N.mm)$

- $\sigma_a = \frac{Mc}{I} = \frac{32 M_a}{\pi D^3} = (561.0206) \; N/mm^2$
- $\sigma_m = \frac{32 M_m}{\pi D^3} = 0$
- $\tau_a = \frac{Tc}{J} = \frac{16 T_a}{\pi D^3} = (66.84507) \; N/mm^2$
- $\tau_m = \frac{Tc}{J} = \frac{16 T_m}{\pi D^3} = (334.22538) \; N/mm^2$

---

max energy distorsion :-

- $\sigma_{ea} = \sqrt{\sigma_a^2 + 3\tau_a^2} = (572.841) \; N/mm^2$
- $\sigma_{em} = \sqrt{\cancel{\sigma_m^2} + 3\tau_m^2} = (578.89533) \; N/mm^2$

max. shear stress :-

- $\sigma_{ea} = \sqrt{\sigma_a^2 + 4\tau_a^2} = (576.729) \; N/mm^2$
- $\sigma_{em} = \sqrt{\cancel{\sigma_m^2} + 4\tau_m^2} = (668.4507) \; N/mm^2$

$$(S_e = C_r \, C_s \, C_t \, C_f \left(\frac{1}{k_f}\right) S_e')$$

- $C_r = (0.75)$
- $C_t = 1 - 0.0032(T - 840) = 1 - 0.0032(950 - 840) = (0.648)$
- $C_s = (0.85)$
- $C_f = A(S_{ut})^b = 14.4(1500)^{-0.718} = (0.07549)$
- $k_f = (1)$
- $S_e' = 0.5(S_{ut}) = 0.5(1500) = (750)$

(Q2). continu...

$\therefore \quad S_e = (0.75)(0.85)(0.648)(0.075\ ug) \frac{1}{(1)} (750)$

$= (23.38868)\ \text{MPa} \quad = (23.388 \times 10^6)\ \text{N/m}$

• for maximum energy of distorstion theory combie with Goodman criterion:-

• $\frac{1}{n_{Goodman}} = \frac{\sigma_{em}}{S_{ut}} + \frac{\sigma_{ea}}{S_e} \rightarrow \frac{1}{n} = \frac{578.8953 \times 10^6}{1500 \times 10^6} + \frac{572.841 \times 10^6}{23.388 \times 10^6}$

$\therefore n_{Goodman} = (0.0401947)$ #

• for maximum shear stress theory of failure with soderberg criterion:-

• $\frac{1}{n_{Soderberg}} = \frac{\sigma_{em}}{S_y} + \frac{\sigma_{ea}}{S_e}$

$\therefore \frac{1}{n} = \frac{668.4507 \times 10^6}{850 \times 10^6} + \frac{576.729 \times 10^6}{23.38 \times 10^6}$

$\therefore n_{Soderberg} = (0.03928650)$ #

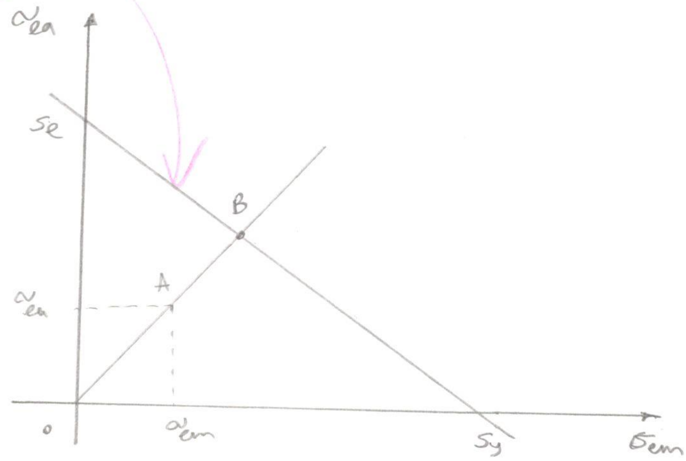


Figure for Goodman (-2)

Name : Mahmoud Shaban Ramadan Hashkel.

Student no: 20173983

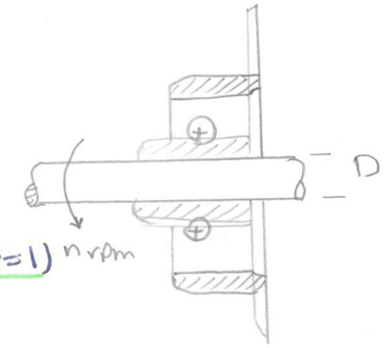
Question (3) :

Given: (02-Series) – Deep Groove-ball bearing

$D = 40\,mm$, $n = 2500$ rpm

$a = 3$ (ball) , outer ring rotating ($V = 1.2$) , steady ($K_s = 1$)

require: L_{10} . (in hours)



bearing (a) :-

$F_{a_A} = $ zero

$F_{r_A} = (5035.714\ N)$

we inter table (10.3) :

at $D = 40\,mm$

- $C = (30.7)\ kN$
- $C_s = (16.6)\ kN$

from table (10-5) :

$F_a / C_s = 0$

$e = 0.19$

$\frac{F_a}{V F_r} = \frac{0}{(1.2) F_r} = 0 < e = 0.19$

bearing (b) :-

$F_{a_B} = 0$

$F_{r_B} = (9464.285\ N)$

at $D = 40\,mm$

- $C = (30.7\ kN)$
- $C_s = (16.6\ kN)$

from table (10-5) :-

$\frac{F_a}{C_s} = 0$

$e = 0.19$

$\frac{F_a}{V F_r} = 0 < e = 0.19$

$\therefore (X = 1)$, $(Y = 0)$

$P = (\underbrace{F_a}_{0} \underbrace{Y}_{0} + F_r\, X\, V)\, K_s \rightarrow P = (F_r\, X\, V)\, K_s$

- $P_A = 5035.714 * 1 * 1.2 * 1$

$= (6042.8568)\ N$

equivalent radial load for (A)

- $L_{10} = \left(\frac{C}{P}\right)^a \frac{10^6}{60\,n}$

$= \left(\frac{30.7 * 10^3}{6042.85}\right)^3 \cdot \frac{10^6}{60\,(2500)} = (874.172)$ hours #

- $P_B = (9464.285)(1)(1.2)(1)$

$= (11357.142)\ N$

equivalent radial load for (B)

$L_{10} = \left(\frac{C}{P}\right)^a \frac{10^6}{60\,n}$

$= \left(\frac{30.7 * 10^3}{11357.142}\right)^3 \cdot \frac{10^6}{60\,(2500)} = (131.6790)$ hours #

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Name: Mahmoud Shaban Hashkel.

St no: 20173983.

• Question (4)

Given: $n_1 = 2500\,rpm$

$N_1 = 24$

$N_2 = 42$

$N_3 = 30$

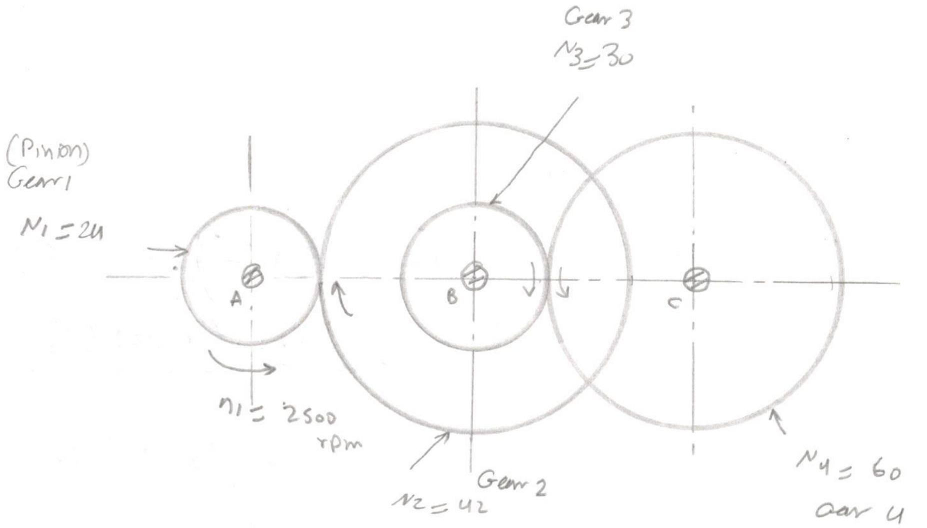
$N_4 = 60$

$P = 6$ teeth/inch

$\phi = 25^\circ$, 40 hp

req: $d_{1,2,3,4}$, f_r, f_t (1,2,3,4)

T_1 , T_4 , $R_{A,B,C}$.



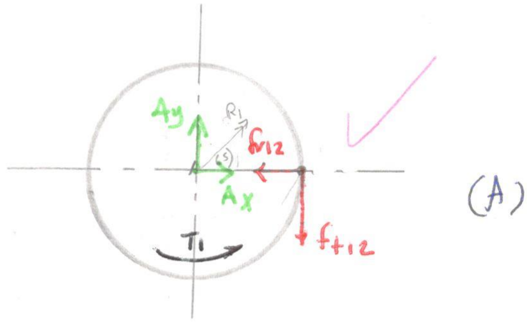
• $\because P = \frac{N}{d}$ $\therefore d = \frac{N}{P}$

$\therefore d_1 = \frac{N_1}{P} = \frac{24}{6} = (4\,in)$ Gear 1

$d_2 = \frac{N_2}{P} = \frac{42}{6} = (7\,in)$ Gear 2

$d_3 = \frac{N_3}{P} = \frac{30}{6} = (5\,in)$ Gear 3

$d_4 = \frac{N_4}{P} = \frac{60}{6} = (10\,in)$ Gear 4. #

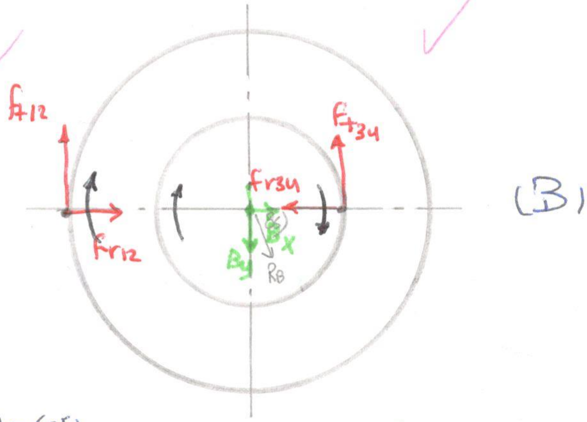


for Gear (1) :-

• $\because hp = \frac{Tn}{63000}$ $\therefore T = \frac{hp(63000)}{n}$

$\therefore T_1 = \frac{40\,(63000)}{2500} = (1008\ lb.in)$ #

Torque in Gear (1)



• $T_1 = f_{t12}\left(\frac{d_1}{2}\right)$ $\therefore f_{t12} = \frac{T_1}{r_1} = \frac{1008}{2}$

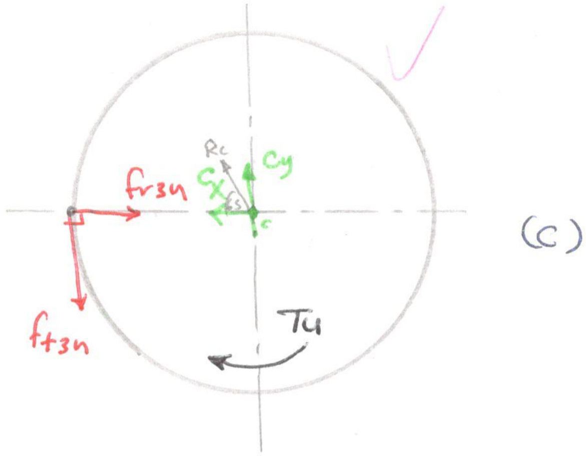
$= (504\ lb)$ #

$f_{r12} = f_{t12} \tan\phi = 504 \tan(25)$

$= (235.01905\ lb)$ #

$\because \Sigma f_y = 0$ $\uparrow^+$

$\therefore A_y = f_{t12} = (504\ lb)$



$\Sigma f_x = 0$ $\rightarrow^+$

$\therefore A_x = f_{r12} = (235.019\ lb)$

$\therefore R_A = \sqrt{R_{Ax}^2 + R_{Ay}^2} = [556.10244\ lb]$

reaction for (A)

• $\left[\theta_{R_1} = \tan^{-1}\left(\frac{A_y}{A_x}\right) = (65^\circ)\right]$

Question (4) conti...

for Gen(2, 3) :-

- $\sum M_B = 0$ ↻+

$$f_{t12}\left(\frac{d_2}{2}\right) - f_{t34}\left(\frac{d_3}{2}\right) = 0$$

$$\therefore f_{t34} = \frac{f_t(12)\left(\frac{d_2}{2}\right)}{d_3/2} = \frac{504 * 3.5}{2.5}$$

$$= (705.6)\ lb \ \#$$

$$\therefore f_{r34} = f_{t34}\tan\phi = 705.6\tan(25^\circ)$$

$$= (329.0266)\ lb \ \#$$

- $\sum f_x = 0$ $\xrightarrow{+}$
  - $B_x = f_{r34} - f_{r12}$

$$= 329.0266 - 235.01905 = (94.0075)\ lb \ \#$$

- $\sum f_y = 0$ ↓+
  - $By = f_{t34} + f_{t12}$

$$= 705.6 + 504 = (1209.6)\ lb \ \#$$

- $R_B = \sqrt{By^2 + Bx^2} = [1213.24\ lb]$ #

reaction on (B)

$\left[\theta_B = \tan^{-1}\frac{By}{(Bx)} = 85^\circ\right]$

for Gen (4) :-

- $f_{r34} = (329.0266)\ lb$
- $f_{t34} = (705.6\ lb)\ lb$

- $\sum f_x = 0$ ← + $\therefore Cx = f_{r34} = (329.0266)\ lb$
- $\sum f_y = 0$ ↑+ $\therefore Cy = f_{t34} = (705.6\ lb)$
  - $R_c = \sqrt{Cx^2 + Cy^2} = [778.5434]\ lb$ #

reaction on (C)

$\left[\theta_c = \tan^{-1}\left(\frac{Cy}{Cx}\right) = 65.00^\circ\right]$

- $T_4 = f_{t34}\left(\frac{d_4}{2}\right) = 705.6 * 5_{in} = [3528\ lb.in]$ #

Torque in gear(4)