TOBB EKONOMİ VE TEKNOLOJİ ÜNİVERSİTESİ

MAK 303 MAKİNA ELEMANLARI I

Dönem Sonu Sınav

07 Nisan 2010

Ad, Soyad ____________________

Dr. M. Ali Güler

Öğrenci No. ____________________

Verilen Zaman: 3 saat (08.30-11.30)

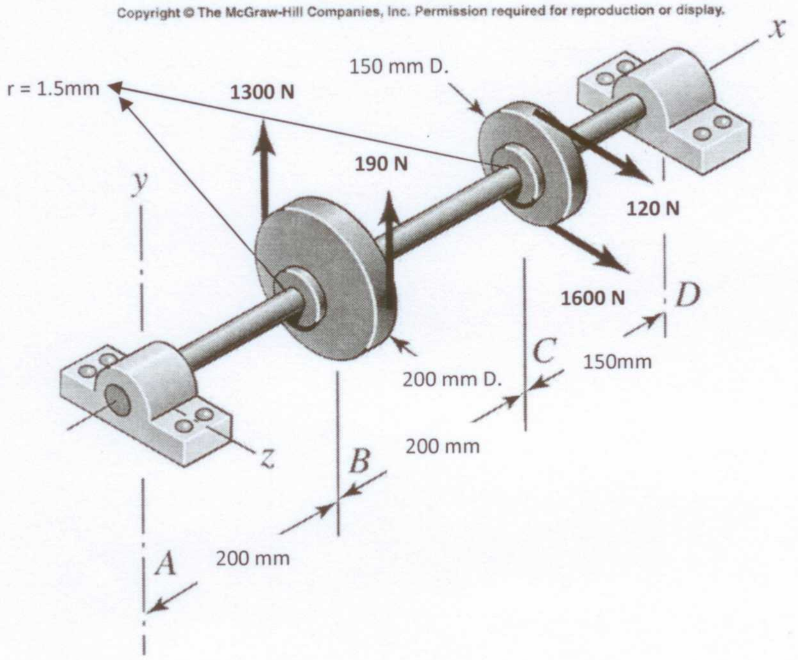
Kitap ve Notlar Kapalı

- *Her soruyu dikkatle okuyunuz.*
- *Yaptığınız işlemleri gösteriniz.*
- *Eksik yada yanlış olduğunu düşündüğünüz durumlarda kendi varsayımınızı kullanınız.*
- *Sınav salonunda cep telefonu kullanmak yasaktır*

Soru No	Maksimum Puan	Puan
1	25	
2	25	
3	25	
4	25	
5	10	
Toplam	110	

Ön sayfa dahil, bu sınav kağıdında toplam *(11)* sayfa vardır.

Soru 1.(25P)



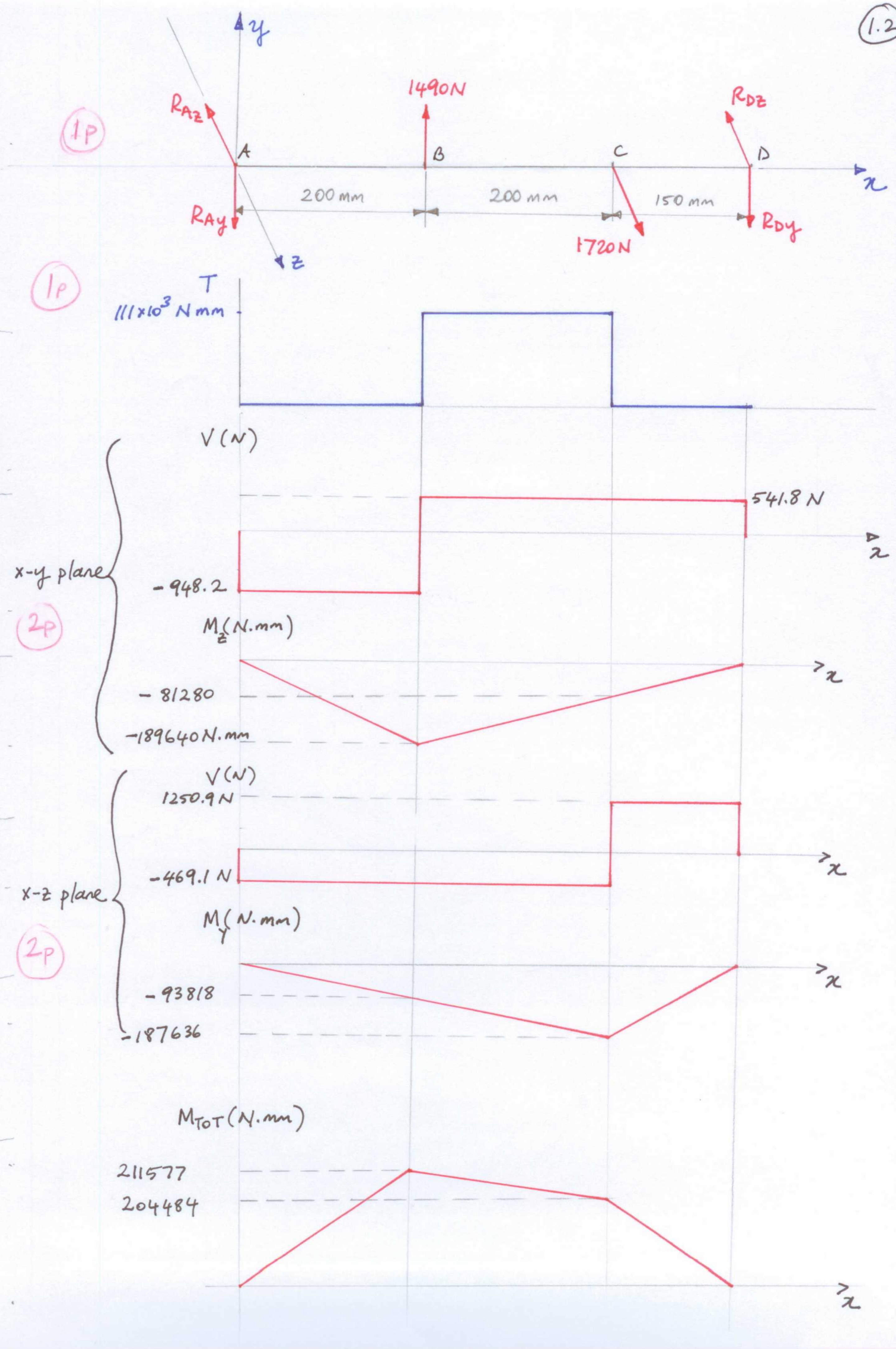
Shigley's Mechanical Engineering Design, R.G. Budyans, J.K. Nisbett, Eight Edition

Şekildeki mil, B ve C noktalarında bulunan kasnakların üzerine bağlı olan kayışlardan iletilen torkun etkisi altında sabit hızla dönmektedir. Milin malzemesi AISI 1018 sıcak haddelenmiş (hot rolled) çeliktir ve mil A'da ve D'de noktasal olarak yataklanmıştır. Çalışma sıcaklığı 20^0 C olup, hesaplamalarınızı %50 güvenirlik için emniyet katsayısını '2' alarak yapınız. Kasnakların mile tutturulduğu noktalardaki 1.5 mm yarıçaplı yuvarlatmalar için teorik gerilme yığılma faktörleri olan "K_t" ve "K_{ts}" değerlerini '2' alınız. (S_{ut} = 400 MPa, S_y = 220 MPa, Bhn<200, f = 0.9)

Mil üzerindeki en kritik noktayı tespit ederek;

a) DE-Goodman yaklaşımıyla **mil çapını(d)** bulunuz ve elde edilen çap değerinin **akmaya karşı kontrolünü** yapınız. 18p

b) DE-Soderberg yaklaşımıyla 16000 çevrime karşılık gelen **yorulma dayanımını 'S_f'** ve **mil çapını(d)** hesaplayınız. 7p


y
1P
RAz
1490N
RDz
A
B
C
D
x
200 mm
200 mm
150 mm
RAy
RDy
1720N
z
1P
T
111x10^3 N mm
V (N)
541.8 N
x
x-y plane
-948.2
2P
Mz (N.mm)
x
-81280
-189640 N.mm
V (N)
1250.9 N
x
-469.1 N
x-z plane
My (N.mm)
2P
x
-93818
-187636
MTOT (N.mm)
211577
204484
x

$\circlearrowleft + \Sigma M_A = 0$ (about z axis)

$-1490(200) + R_{Dy}(550) = 0 \Rightarrow R_{Dy} = 541.8\ N$

$+\uparrow \Sigma F_y = 0$

$1490 - R_{Dy} - R_{Ay} = 0 \Rightarrow R_{Ay} = 948.2\ N$

$\circlearrowleft + \Sigma M_B = 0$ (about x axis)

$(1300)(100) - (190)(100) - T = 0 \Rightarrow T = 111000\ N.mm$

$\circlearrowleft + \Sigma M_A = 0$ (about y axis

$(1720)(400) - R_{Dz}(550) = 0 \Rightarrow R_{Dz} = 1250.9\ N$

$+\uparrow \Sigma F_z = 0$

$1720 - R_{Dz} - R_{Az} = 0 \Rightarrow R_{Az} = 469.1\ N$

Total moment at point B

$$M_B^{Tot} = \sqrt{M_z^2 + M_y^2} = \sqrt{189640^2 + 93818^2} = 211577\ N.mm$$

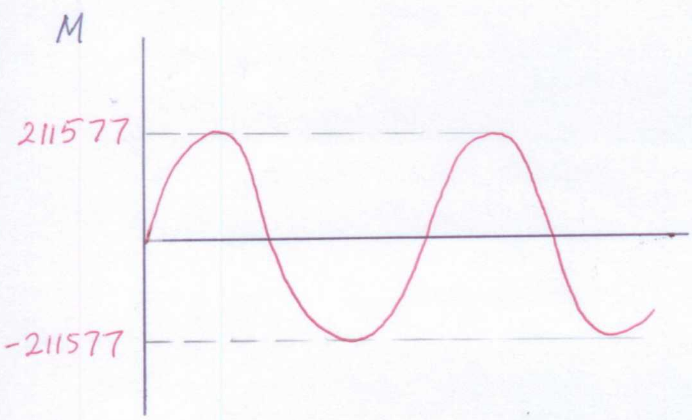
Total moment at point C

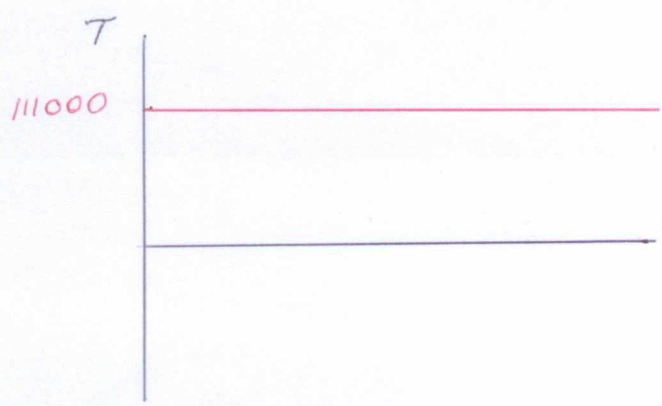
$$M_C^{Tot} = \sqrt{M_z^2 + M_y^2} = \sqrt{81280^2 + 187636^2} = 204484\ N.mm$$

For a rotating shaft constant bending moment will produce a completely reversed bending stress

$M_a = 211577\ N.mm$ $\quad M_m = 0$ (2p)

$T_m = 111000\ N.mm$ $\quad T_a = 0$ (2p)





$S_e = k_a k_b k_c k_d k_e \cdot S_e'$

$S_e' = 0.5\, S_{ut} = (0.5)(400) = 200$ MPa

$k_a = a\, S_{ut}^{\,b} = 57.7\,(400)^{-0.718} = 0.781$

$k_b = \left(\frac{d}{7.62}\right)^{-0.107}$; for an estimated value of 40 mm shaft diameter

$k_b = 0.84$

$k_c = 1$

$k_e = 1$ % 50 reliability

$S_e = (0.781)(0.84)(1)(1)(1)(200) = 131.2$ MPa 2p

$K_f = 1 + q\,(K_t - 1) = 1 + 0.7(2-1) = 1.7$ 1p

$K_{fs} = 1 + q_s\,(K_{ts} - 1) = 1 + 0.9(2-1) = 1.9$ 1p

stresses are combined using distortion energy (DE) theory and Goodman criteria is used for fatigue

DE – Goodman

$$d = \left(\frac{16n}{\pi}\left\{\frac{1}{S_e}\left[4\,(K_f M_a)^2\right]^{1/2} + \frac{1}{S_{ut}}\left[3\,(K_{fs} T_m)^2\right]^{1/2}\right\}\right)^{1/3}$$

$$d = \left(\frac{16(2)}{\pi}\left\{\frac{1}{131.2}\left[4\left((1.7)\,211577\right)^2\right]^{1/2} + \frac{1}{400}\left[3\left((1.9)\,111000\right)^2\right]^{1/2}\right\}\right)^{1/3}$$

$$= \left[\frac{32}{\pi}\left(5482.94 + 913.22\right)\right]^{1/3} \Rightarrow$$

$d = 40.24$ mm 2p

All estimates have probably been conservative so select $d = 40$ mm check for yielding

$$\sigma'_{max} = \left[(\sigma_m + \sigma_a)^2 + 3(\tau_m + \tau_a)^2 \right]^{1/2}$$

$$\sigma_m = K_f \cdot \frac{M_m}{\frac{\pi d^3}{32}} = 0$$

$$\sigma_a = K_f \cdot \frac{M_a}{\frac{\pi d^3}{32}} = \frac{(1.7)\,.\,(211577)}{\frac{\pi\,(40)^3}{32}} = 57.245 \text{ MPa}$$

$$\tau_m = K_{fs} \cdot \frac{T_m}{\frac{\pi d^3}{16}} = \frac{(1.9)\,.\,(111000)}{\frac{\pi\,(40)^3}{16}} = 16.783 \text{ MPa}$$

$$\tau_a = K_{fs} \cdot \frac{T_a}{\frac{\pi d^3}{16}} = 0$$

$$\sigma'_{max} = \left[(57.245)^2 + 3(16.783)^2 \right]^{1/2} = 64.2 \text{ MPa}$$

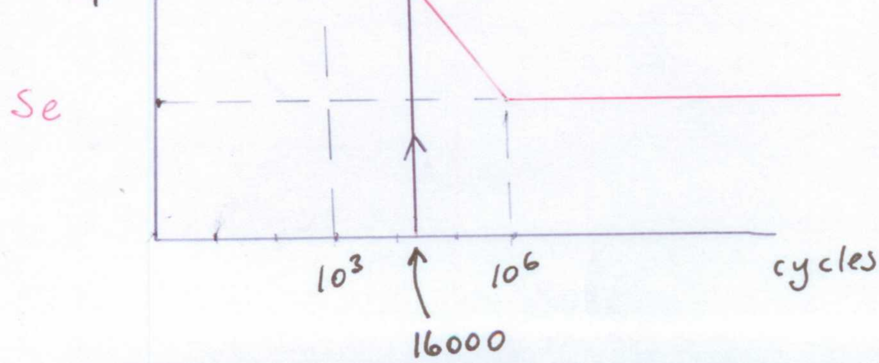
Factor of safety agains yielding

$$n_y = \frac{S_y}{\sigma'_{max}} = \frac{220}{64.2} = 3.43 \quad > 2$$

2p

Therefore, the selected size (40 mm diameter) is safe.

b.)



or $S_f = a N^b$

$$a = \frac{(f S_{ut})^2}{S_e} = \frac{(360)^2}{131.2} = 987.8$$

$$b = -\frac{1}{3}\log\left(\frac{f S_{ut}}{S_e}\right) = -0.146123$$

$$S_f = a N^b = (987.8)(16000)^{-0.14612}$$

$$S_f = 240 \text{ MPa}$$

2p

$$\frac{\log(f \cdot S_{ut}) - \log S_e}{\log(f \cdot S_{ut}) - \log S_f} = \frac{\log 10^6 - \log 10^3}{\log 16000 - \log 10^3}$$

$$\frac{\log 360 - \log 131.2}{\log 360 - \log S_f} = \frac{6 - 3}{4.2 - 3}$$

$$\log 360 - \log S_f = 0.17595$$

$$\log S_f = 2.38$$

$$S_f \cong 240 \text{ MPa}$$ 2p

The shaft diameter corresponding to this stress

$$d = \left(\frac{16n}{\pi} \left\{ \frac{1}{S_f} \left[4 (K_f M_a)^2 \right]^{1/2} + \frac{1}{S_{ut}} \left[3 (K_{fs} T_m)^2 \right]^{1/2} \right\} \right)^{1/3}$$

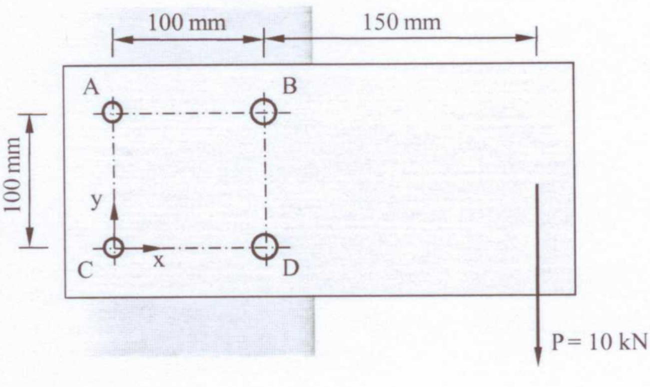
$$= \left(\frac{16(2)}{\pi} \left\{ \frac{1}{240} \left[4 ((1.7)211577)^2 \right]^{1/2} + \frac{1}{400} \left[3 ((1.9)(111000))^2 \right]^{1/2} \right\} \right)^{1/3}$$

$$= \left[\frac{32}{\pi} (2997.34 + 913.22) \right]^{1/3}$$

$d = 34.15$ mm 3p

$d \cong 34$ mm

Soru 2.(25P)

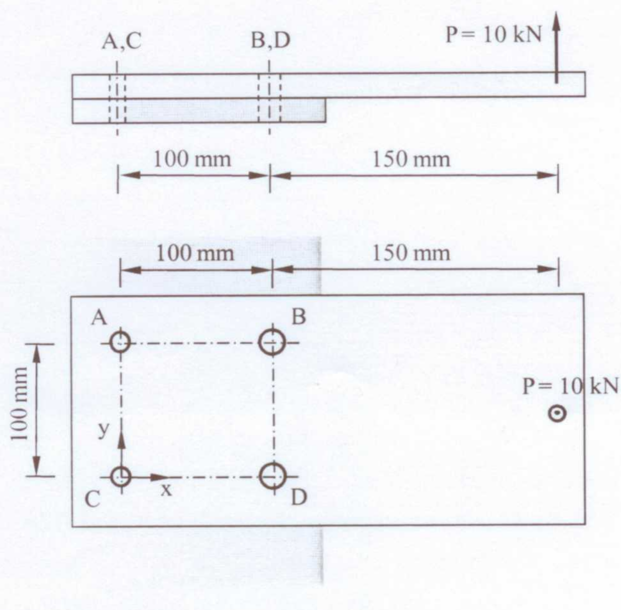


(a)

Şekildeki perçinli bağlantıda A ve C noktalarında 14 mm çaplı, B ve D noktalarında ise 16 mm çaplı perçinler kullanılmıştır.

a) Şekil (a) da gösterilen yükleme durumu için her bir perçine gelen kuvvetleri bulunuz ve şekil üzerinde göstererek bileşkeleri hesaplayınız. Kritik olan perçini belirleyerek, kayma gerilmesi için **emniyet katsayısını** hesaplayınız. (Perçin malzemesi için τ_{all} = 150 MPa)

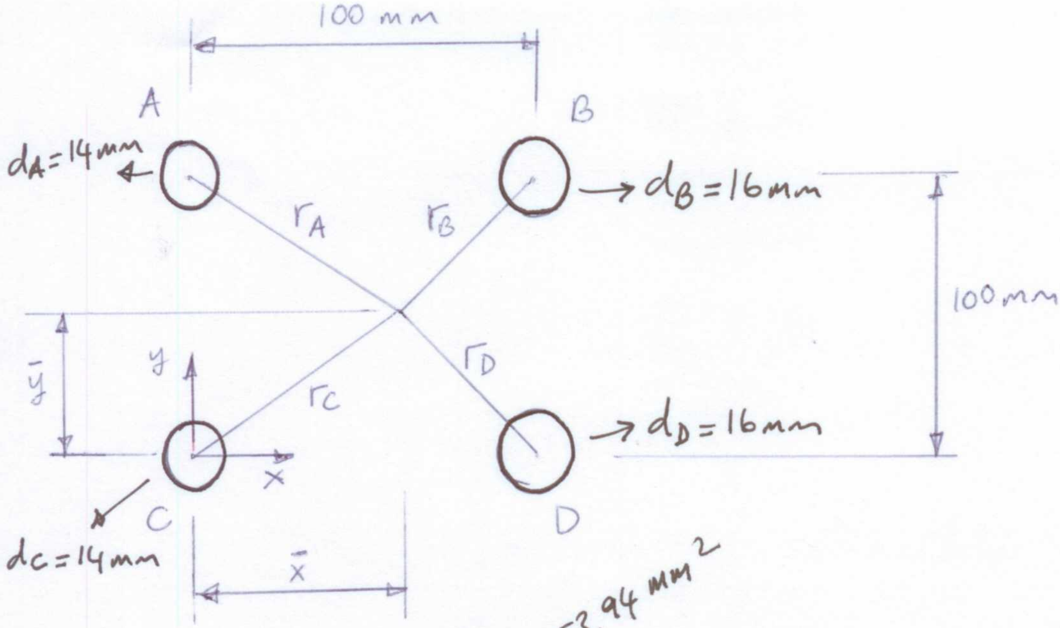
b) Şekil (b) de gösterilen yükleme durumu için hangi perçinlerin kritik olduğuna karar vererek normal gerilme için **emniyet katsayısını** hesaplayınız. (σ_{all} = 200 MPa kabul ediniz.)



(b)

Fundamentals of Machine Elements, B.J. Hamrock, S.R. Schmid, Bo O. Jacobson, Second Edition

(2.2)



$$A_C = A_A = \frac{\pi}{4}14^2 = 49\pi = 153.94\ mm^2$$

$$A_D = A_B = \frac{\pi}{4}16^2 = 64\pi = 201.1\ mm^2$$

$$\Sigma A = A_A + A_B + A_C + A_D = 226\pi$$

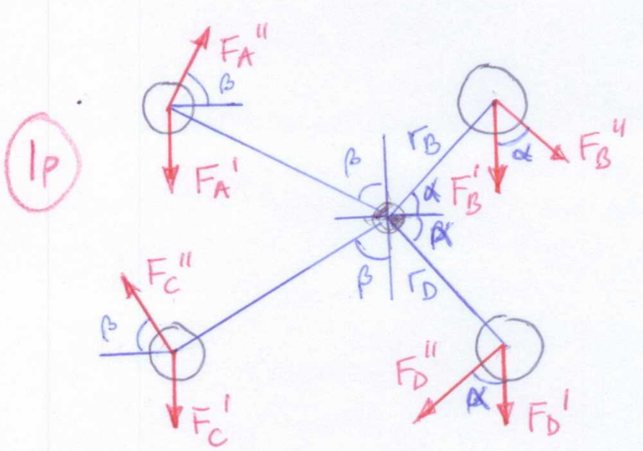
$$\bar{x} = \frac{A_A\cdot(0) + A_C\cdot(0) + A_D\cdot(100) + A_B\cdot(100)}{A_A + A_C + A_D + A_B} = \frac{12800\pi}{226\pi} = 56.64\ mm$$ 1p

$$\bar{y} = \frac{A_A\cdot(100) + A_B\cdot(100) + A_C(0) + A_D(0)}{A_A + A_B + A_C + A_D} = \frac{11300\pi}{226\pi} = 50\ mm$$ 1p

$$r_A = r_C = \sqrt{(56.64)^2 + 50^2} = 75.55\ mm$$

$$r_B = r_D = \sqrt{(43.36)^2 + 50^2} = 66.18\ mm$$

$$F_A' = F_B' = F_C' = F_D' = \frac{10\,kN}{4} = 2.5\,kN$$ 1p



Since the rivet diameters are not equal and rivet farthest from the centroid takes the greatest load.

$$\frac{\tau_A''}{r_A} = \frac{\tau_B''}{r_B} = \frac{\tau_C''}{r_C} = \frac{\tau_D''}{r_D} \quad \text{———} (1)$$

or

$$\frac{\frac{4F_A''}{\pi d_A^2}}{r_A} = \frac{\frac{4F_B''}{\pi d_B^2}}{r_B} \quad \text{———} (2)$$

also

$$F_A'' = F_C'' \quad \text{———} (3)$$

$$F_B'' = F_D'' \quad \text{———} (4)$$

$$M = r_A F_A'' + r_B F_B'' + r_C F_C'' + r_D F_D'' \quad \text{———} \quad (5)$$

using (3), (4) & (5)

$$M = 2 r_A F_A'' + 2 r_B F_B'' \quad \text{———} \quad (6)$$

Using (2)

$$\frac{F_A''}{r_A} = \frac{d_A^2}{d_B^2} \cdot \frac{F_B''}{r_B} \Rightarrow \quad F_A'' = 0.874 F_B'' \quad 2p \quad \text{———} \quad (7)$$

Moment also equals to $10\,kN \cdot (150 + 43.36)\,mm$

$$M = 10000\,N \cdot 193.36\,mm = 1.9336 \times 10^6 \; N.mm \quad 2p \quad \text{———} \quad (8)$$

Using (6), (7) and (8)

$$1.9336 \times 10^6 = (2 r_A (0.874) + 2 r_B) F_B''$$

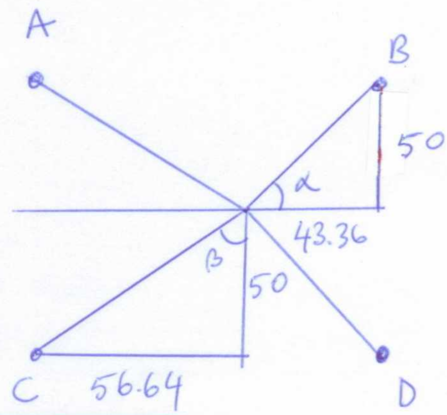
$$F_B'' = 7312.57\,N \quad \Rightarrow \quad F_B'' = 7.313\; kN \quad 1p \quad \text{———} \quad (9)$$

Using (7) and (9)

$$F_A'' = 6.392\; kN \quad 1p$$

$$\alpha = \tan^{-1} \frac{50}{43.36} = 49.07^\circ$$

$$\beta = \tan^{-1} \frac{56.64}{50} = 48.56^\circ$$



Resultant force at A

$$F_A = \sqrt{(F_A' - F_A'' \sin\beta)^2 + (F_A'' \cos\beta)^2}$$

$$F_A = \sqrt{(2.5 - 6.392 \sin 48.56)^2 + (6.392 \cos 48.56)^2}$$

$$F_A = 4.81\; kN = F_C \quad 1p$$

$$F_B = \sqrt{(F_B' + F_B'' \cos\alpha)^2 + (F_B'' \sin\alpha)^2}$$

$$F_B = \sqrt{(2.5 + 7.313 \cos 49.07)^2 + (7.313 \sin 49.07)^2}$$

$$F_B = 9.15\; kN = F_D \quad 1p$$

shear stresses at rivet A and C

$$\tau_A = \tau_C = \frac{F_A}{A_A} = \frac{4.81(10^3)\,N}{49\pi} = 31.25\ MPa$$

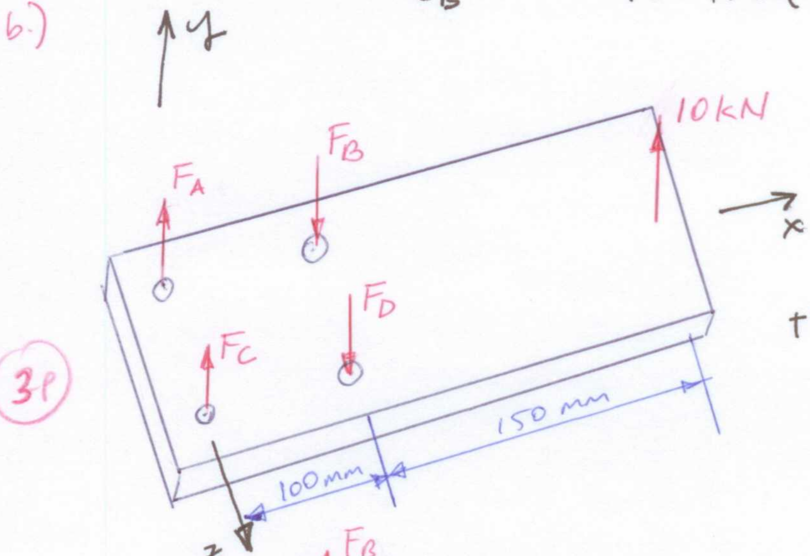
$$\tau_B = \tau_D = \frac{F_B}{A_B} = \frac{9.15(10^3)\,N}{64\pi} = 45.51\ MPa$$ 2p

Rivets B and D are critical.

Factor of safety

$$n = \frac{\tau_{all}}{\tau_B} = \frac{150\ MPa}{45.51\ MPa} = 3.3$$ 2p

b.)



3p

$F_A = F_C$

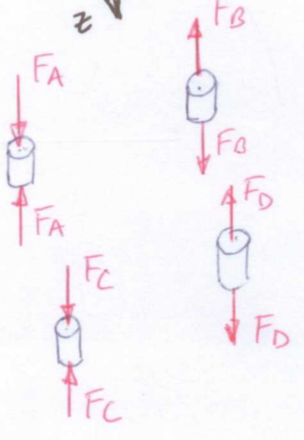
$F_B = F_D$

$+\uparrow \sum F_y = 0$

$10 - F_B - F_D + F_A + F_C = 0$

$2F_A - 2F_B = -10$

$F_A - F_B = -5$



Taking a moment about z axis

$\circlearrowleft + \sum M_z = 0$

$2F_B(100) - 10(250) = 0$

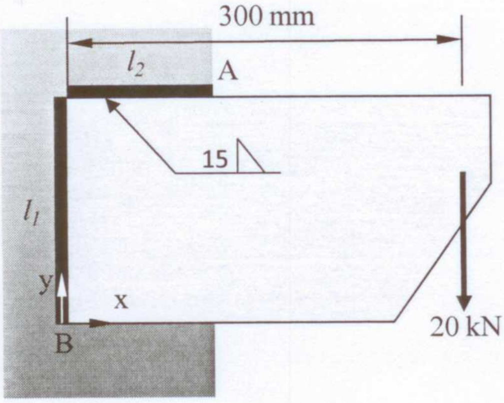
3p $F_B = 12.5\ kN$ (tension on the rivet B)

$F_A = 7.5\ kN$ (compression on the rivet A)

$$\sigma_B = \frac{F_B}{A_B} = \frac{12500}{64\pi} = 62.17\ MPa$$ 2p

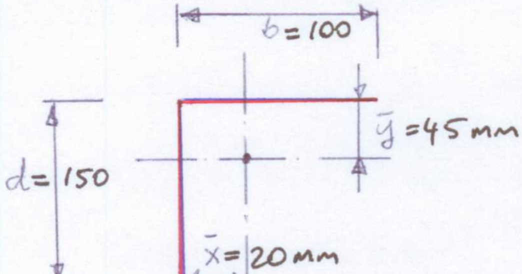
$$n = \frac{\sigma_{all}}{\sigma_B} = \frac{200}{62.17} = 3.22$$ 2p

Soru 3. (25P)



Fundamentals of Machine Elements, B.J. Hamrock, S.R. Schmid, Bo O. Jacobson, Second Edition

Şekildeki 20 kN'luk yük altındaki plaka, destek sacına 2 kaynak dikişi ile bağlanmıştır. Köşe kaynak dikişlerinin uzunlukları sırasıyla l_1=150 mm ve l_2=100 mm' dir. Kaynak kalınlığı h=15 mm' dir. A ve B noktalarındaki gerilmeleri ayrı ayrı hesaplayarak ve kaynak dikişi için müsaade edilen kayma gerilmesini τ_{all}=138 MPa alarak, mevcut yükleme durumu için **emniyet katsayısını** hesaplayınız.



$$\bar{x} = \frac{b^2}{2(b+d)} = \frac{100^2}{2(100+150)} = 20 \text{ mm}$$ 1p

$$\bar{y} = \frac{d^2}{2(b+d)} = \frac{150^2}{2(100+150)} = 45 \text{ mm}$$ 1p

$$A = 0.707\, h\,(b+d) = 0.707(15)(100+150)$$

$$A = 2651.25 \text{ mm}^2$$

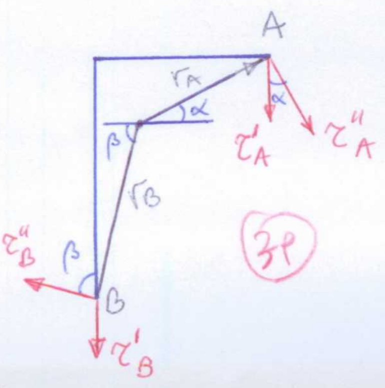
$$J_u = \frac{(b+d)^4 - 6b^2d^2}{12(b+d)} = \frac{250^4 - 6(100)^2(150)^2}{12\cdot(250)} = 852.08\times10^3 \text{ mm}^3$$

$$J = 0.707\, h\, J_u = 0.707(15)(852.08\times10^3) = 9.036\times10^6 \text{ mm}^4$$ 1p

Primary shear

$$\tau'_A = \tau'_B = \frac{V}{A} = \frac{20000}{2651.25} = 7.54 \text{ MPa}$$ 2p

$$\text{Torque} = T = 20000\cdot(300-20) = 5.6\times10^6 \text{ N.mm}$$ 2p



$$r_A = \sqrt{80^2 + 45^2} = 91.788 \text{ mm}$$

$$r_B = \sqrt{20^2 + 105^2} = 106.888 \text{ mm}$$

$$\tau_A'' = \frac{T \cdot r_A}{J} = \frac{5.6\times10^6 \cdot 91.788}{9.036\times10^6} = 56.885 \text{ MPa}$$ 2p

$$\tau_B'' = \frac{T \cdot r_B}{J} = \frac{5.6\times10^6 \cdot 106.888}{9.036\times10^6} = 66.243 \text{ MPa}$$ 2p

$$\alpha = \tan^{-1}\frac{45}{80} = 29.358^\circ$$

$$\beta = \tan^{-1}\frac{105}{20} = 79.216$$

$$(\tau_A)_x = \tau_A'' \sin\alpha = 56.885 \sin 29.358 = 27.889 \text{ MPa}$$

$$(\tau_A)_y = \tau_A' + \tau_A'' \cos\alpha = 7.54 + 56.885 \cos 29.358 = 57.119 \text{ MPa}$$

$$\tau_A = \sqrt{(\tau_A)_x^2 + (\tau_A)_y^2} = 63.56 \text{ MPa}$$ 4p

$$(\tau_B)_x = \tau_B'' \sin\beta = 66.243 \sin 79.216 = 65.073 \text{ MPa}$$

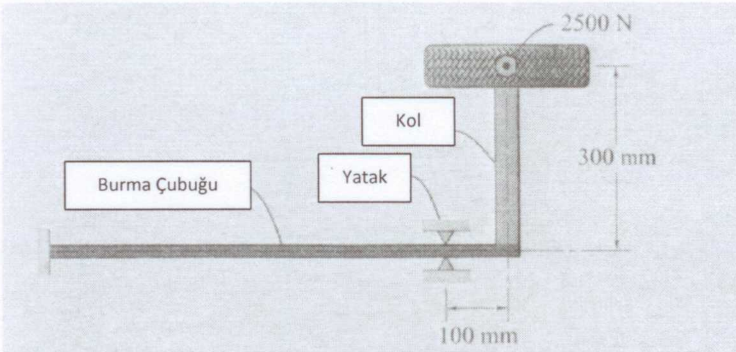
$$(\tau_B)_y = \tau_B'' \cos\beta - \tau_B' = 66.243 \cos 79.216 - 7.54 = 4.85 \text{ MPa}$$

$$\tau_B = \sqrt{(\tau_B)_x^2 + (\tau_B)_y^2} = 65.25 \text{ MPa}$$ 4p

safety factor according to point B

$$n = \frac{\tau_{all}}{\tau_B} = \frac{138}{65.25} = 2.11$$ 3p

Soru 4.(25P)

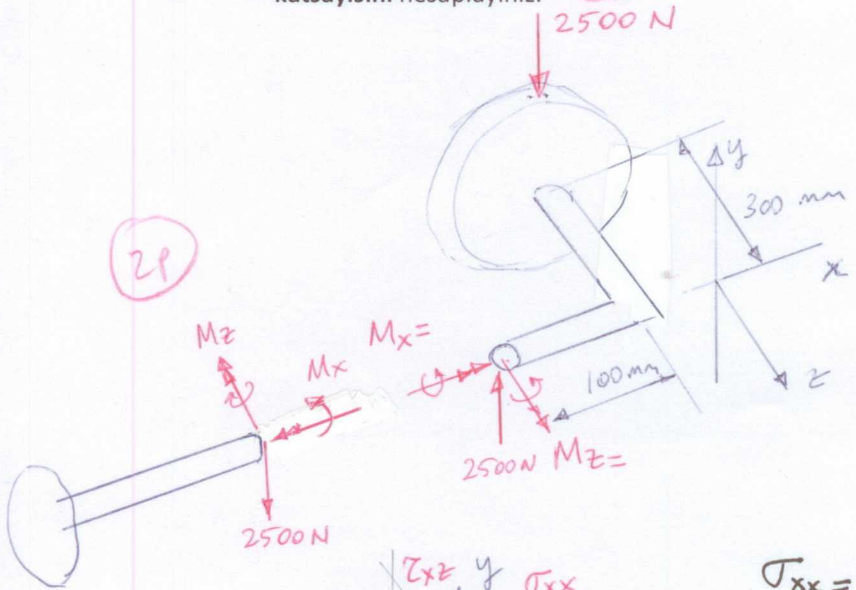


Fundamentals of Machine Elements, B.J. Hamrock, S.R. Schmid, Bo O. Jacobson, Second Edition

Şekilde, bir otomobilin arka aksının görünümü mevcuttur. Otomobilin arka tekeri bir kol aracılığıyla burma çubuğuna bağlanmıştır. Burma çubuğu, teker merkez ekseninden 100 mm içeride kalacak şekilde yataklanmıştır. Burma çubuğunun malzemesi AISI 1030 HR olup, akma gerilmesi(S_y) 260 Mpa dır. Zeminden tekere 2500 N büyüklüğünde kuvvet etki ettiğine göre,

a) Maksimum şekil değiştirme enerjisi teoremine(von Mises) göre, burma çubuğunun **çapını(d)** emniyet katsayısını '2' alarak belirleyiniz. 15p

b) Elde edilen çap değeri için maksimum kayma gerilmesi teoremini kullanarak **emniyet katsayısını** hesaplayınız. 10p



$M_x = T = 2500\,N\,(300\,mm)$

$T = 750 \times 10^3\ N\,mm$ 2p

$M_z = M = 2500\,N\,(100\,mm)$

$= 250 \times 10^3\ N\,mm$ 2p

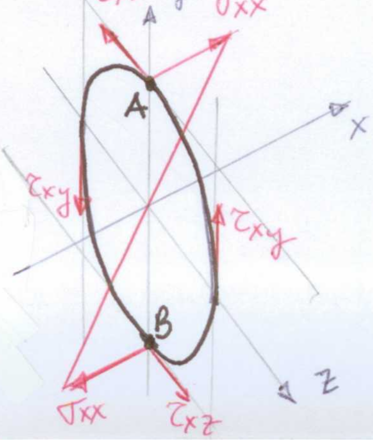
$I = \dfrac{\pi d^4}{64}$

$J = \dfrac{\pi d^4}{32}$

$\sigma_{xx} = \dfrac{M_z \cdot y}{I_z} = \dfrac{250 \times 10^3 \cdot d/2}{\dfrac{\pi d^4}{64}}$

$\sigma_{xx} = \dfrac{(32)\,250 \cdot 10^3}{\pi d^3} = \dfrac{8 \times 10^6}{\pi d^3}$

$\sigma_{xx} = \dfrac{2.5465 \times 10^6}{d^3}$



(4.2)

$$\tau_{xz} = \frac{T \cdot c}{J} = \frac{750 * 10^3 \, d/2}{\frac{\pi d^4}{32}} = \frac{(16)(750)10^3}{\pi d^3} = \frac{12 * 10^6}{\pi d^3}$$

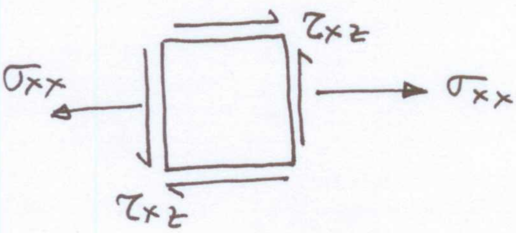
2p

$$= \frac{3.8197 * 10^6}{d^3}$$

Von Misses stress

$$\sigma' = \frac{1}{\sqrt{2}}\left[(\sigma_x - \sigma_y)^2 + (\sigma_y - \sigma_z)^2 + (\sigma_z - \sigma_x)^2 + 6(\tau_{xy}^2 + \tau_{yz}^2 + \tau_{zx}^2)\right]^{1/2}$$

At location A $\quad \sigma_{yy} = \sigma_{zz} = \tau_{xy} = \tau_{yz} = 0$



$$\sigma' = \frac{1}{\sqrt{2}}\left[\sigma_{xx}^2 + \sigma_{xx}^2 + 6\,\tau_{xz}^2\right]^{1/2}$$

$$\sigma' = \sqrt{\sigma_{xx}^2 + 3\,\tau_{xz}^2}$$

For distortion energy theory

$$\sigma' = \sqrt{\sigma_{xx}^2 + 3\,\tau_{xz}^2} = \frac{S_y}{n}$$

2p

$$\sqrt{\left[\frac{32}{\pi d^3} 250.(10^3)\right]^2 + 3\left[\frac{16}{\pi d^3} 750.(10)^3\right]^2} = \frac{260}{2}$$

$$\frac{1}{d^3}\sqrt{6.4845 * 10^{12} + 4.3770 * 10^{13}} = 130$$

$$d^3 = 54531.5$$

$$d = 37.9 \text{ mm}$$

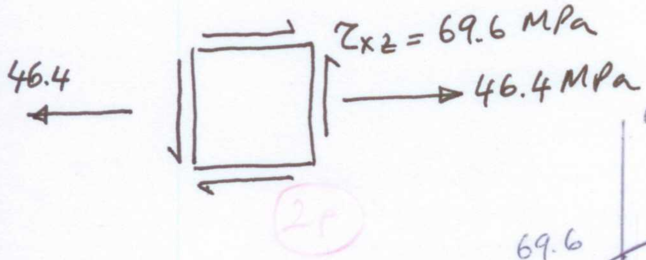
$$d \cong 38 \text{ mm}$$

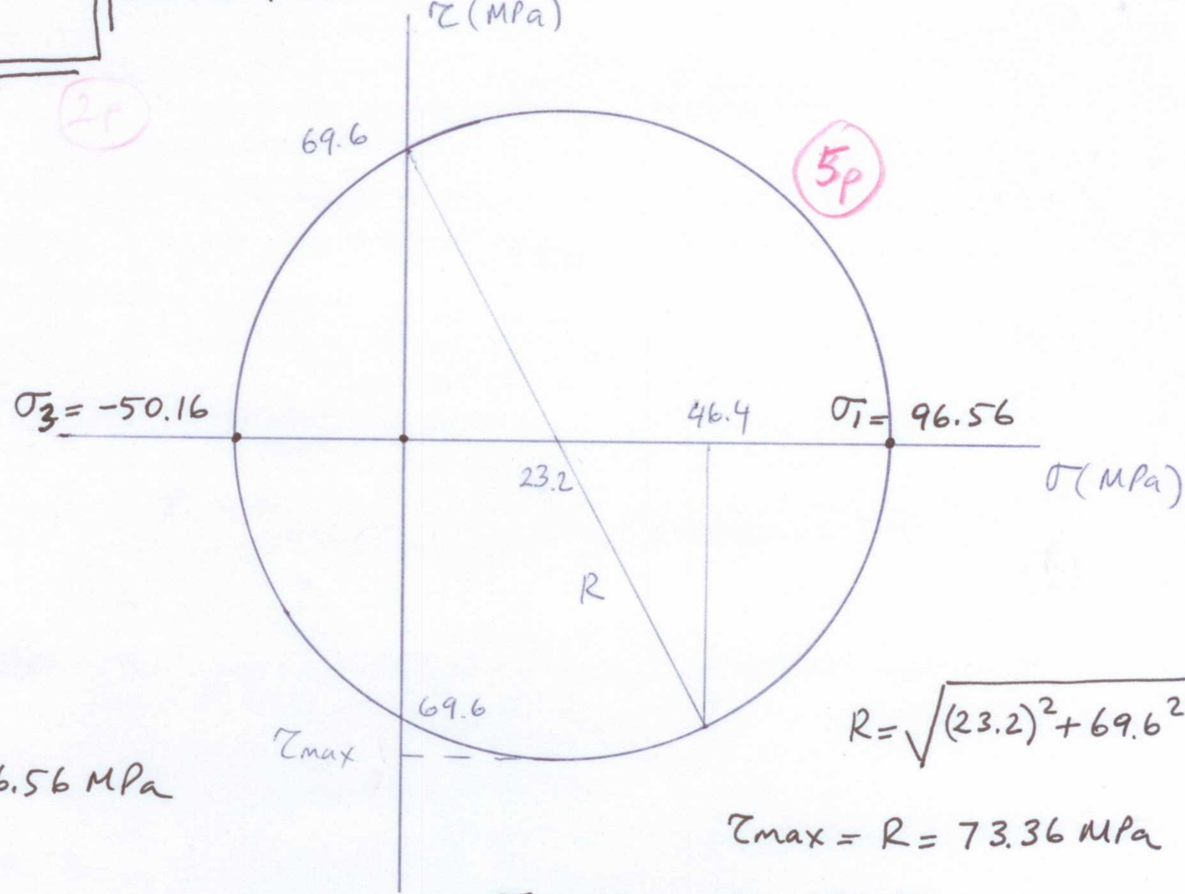
3p

b.) For $d = 38$ mm

$$\sigma_{xx} = \frac{32}{\pi d^3} \cdot 250(10^3) = 46.4 \text{ MPa}$$ 1p

$$\tau_{xz} = \frac{16}{\pi d^3} \cdot 750(10^3) = 69.6 \text{ MPa}$$ 1p





$$R = \sqrt{(23.2)^2 + 69.6^2}$$

$\sigma_1 = 96.56$ MPa

$\sigma_2 = 0$

$\sigma_3 = -50.16$ MPa

$$\tau_{max} = R = 73.36 \text{ MPa}$$

$$\sigma_1 = 23.2 + R = 96.56$$

$$\sigma_2 = 23.2 - R =$$

Multi axial case

$$\tau_{max} = \frac{\sigma_1 - \sigma_3}{2} = 73.36$$

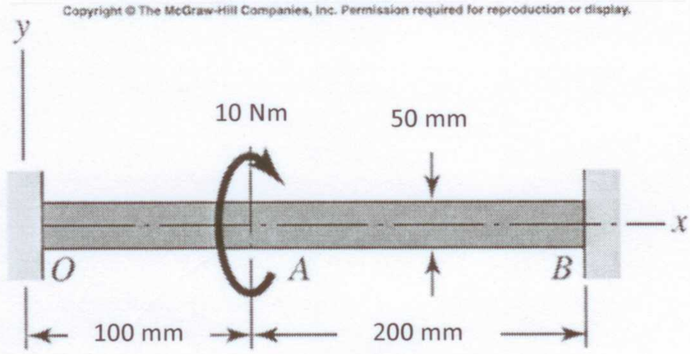
1-D axial test

$$\tau_{max} = \frac{S_y}{2n}$$

$$73.36 = \frac{S_y}{2n} \Rightarrow n = \frac{260}{2 \cdot (73.36)}$$

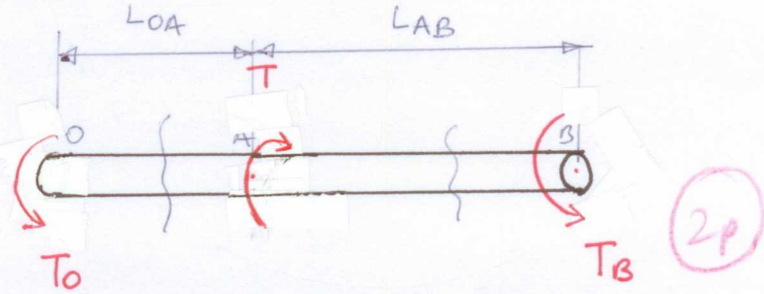
$$n = 1.77$$ 3p

Soru 5.(10P)

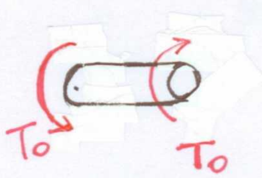


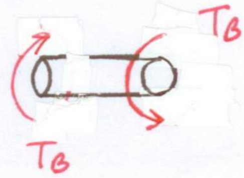
Shigley's Mechanical Engineering Design, R.G. Budyans, J.K. Nisbett, Eight Edition

Şekildeki iki ucu ankastre bağlı kirişe A noktasından 10 Nm büyüklüğünde tork etki etmektedir. O ve B noktalarındaki **tepki torklarını** Castigliano teoreminden yararlanarak bulunuz.



2p





Total strain energy due to Torques T_O and T_B

$$U = \frac{T_O^2 L_{OA}}{2GJ} + \frac{T_B^2 L_{AB}}{2JG}$$ 2p

At point A

$$\theta_{OA} = \frac{\partial U}{\partial T_O} = \frac{T_O L_{OA}}{GJ}$$ 1p

At point A

$$\theta_{AB} = \frac{\partial U}{\partial T_B} = \frac{T_B L_{AB}}{JG}$$ 1p

Since both ends fixed. Using the sign convention (positive away from the bar)

$$\Sigma\theta = -\theta_{OA} + \theta_{AB} = 0$$

$$-\frac{T_O \cdot 100}{JG} + \frac{T_B \cdot 200}{JG} = 0$$

$$T_O = 2T_B \quad (2p)$$

Also from Torque equilibrium

$$T_O - 10 + T_B = 0$$

$$3T_B - 10 = 0$$

$$T_B = \frac{10}{3}\ N.m = 3.33\ N.m \quad (1p)$$

$$T_O = 10 - T_B = 10 - \frac{10}{3} = \frac{20}{3}\ N.m = 6.67\ N.m \quad (1p)$$