ME 208
Machine Design

Liou

Exam 1

- Mohr's Circle P 83, 3-D, p 87

  $R = \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$ 82

  $\sigma_{avg} = C = \frac{\sigma_x + \sigma_y}{2}$ 82

  $\sigma_1, \sigma_2 = C \pm R$ 81

- Stress / strain p 33
- ductile: $\epsilon_f > 5\%$ limited by shear strength
- brittle: under torsion, breaks @ 45°
  limited by tensile strength

Stress Analysis

Pure torsion: $\sigma_1 = \tau$

$\sigma_{bend} = \frac{My}{I}$ * C goes at Point of interest (usually r) 90

$\sigma_{torsion}\ \tau = \frac{Tc}{J}$ * $J = I_{xx} + I_{yy}$ 101

$\sigma_{shear}\ \tau = \frac{TQ}{Ib}$ * $Q = \int_y^c y\,dA$, b=width, beam

Strain energy p 162 (castigliano's)

STATIC LOADING FAILURES CH 5, 213

- MSS theory: $n = \frac{S_y}{\sigma'}$

  $\sigma' = (\sigma_A^2 - \sigma_A\sigma_B + \sigma_B^2)^{1/2}$ 223

- Brittle & ductile summary - theories 219

  Ductile Summary 231

  Brittle Summary 238

Ch 6 Fatigue by Variable Loading 265

- S_e roadmap endurance limit p 345
- Yield line equations (Gerber, Goodman, etc) 306
- S-N diagram 274
- S_f fatigue strength p 285
  if N cycles > 10^6, $S_f = S_e'$
- $n_f = \frac{S_f}{\sigma_a}$ OR $n_f = \frac{S_e}{\sigma_a}$ (if $\sigma_m = 0$)
- σ_a, σ_m p 301
- n_y, static yield $n_y = \frac{S_y}{|\sigma_{max}|}$ p 306
- K_f, use stress conc. A-15 & q, p 295

  * σ_a & σ_m for fatigue $= \frac{F}{A} K_f$ OR $\frac{My}{I} K_f$

- S_y A-20, OR .9 S_{ut} OR f(BHN) p 41

Ch 4 Deflection & Stiffness 147-

- basic deflection p 149
- curved beam deflection p 170, 172
- strain energy p 163

Ch 8 - Bolts

load factor n_L p 440

S_p, proof strength p 434 Table 810-811

A_t tensile stress Area p 412 Table 8-1, 8-2

F_i preload WRT proof strength P 442

$F_p = A_t S_p$ p 442

Nomenclature 436 P, P_b, C, k_b

K_b formula p 426

E, table A-5 p 1007

* don't forget - choose standard bolt length TABLE A-17

Ch 7 - shafts & tolerances P 359-400

shafts

- fatigue failure criteria 369
- shaft stresses 367
- check for yielding 370
  stress concentrations Table 7-1, 373
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  FUNDAMENTAL DEVIATIONS } 1023

finding fatigue factor of safety for a shaft

a) start pg 369

b) get K_t, K_{ts}

Ch 11 - bearings 570-604

- combined loadings P 579
  F_a, axial F_R radial
  F_e, equivalent radial load
- load, life, reliability p 577, 575

— TABLES —

Equivalent load factors 11-1, 580

Dims. & load ratings 581, 582

load application factors 583

$X_D = \frac{L_D}{L_R}$ p 575

$C_{10} =$ .... p 575, 578 "basic load rating"

$L_R = L_{10} = 10^6$

finding Load Rating C_{10} for combined load

a) estimate C_{10} using only F_R, & eq (11-6)

b) choose 02 series bearing, use C_0 to find e, then X, Y, & F_e

c) find C_{10} using F_e and choose new 02 series bearing

d) find e, X, Y, F_e, & check that C_{10} is still under C_{10} in chart

Ch 14 Gears 733-767

teeth, pitch, etc p676

- Spur gear bending & surface wear 766, 767
  W_t, H, V P707
- design factor = S_H or S_F

note cycles will be accounted for in Y_N or Z_N
only need Y_N, Z_n if cycles other than 10^7

Ch 13 gears

- gear anatomy p 676, 680
- force analysis, bevel gears p 709 (W_t, W_r, W_a)
- example, bearing reactions p711
  1) find R from bearing to G
  2) Σ moments = $\Sigma(R_{B/G} \times F) + T = 0$

Exams

ME 208 F10 Exam #3 70 minutes Name: David Malawey

Open Textbook; Closed Notes, Only one sheet of paper allowed

1. (10 pts)
For a rotating shaft with constant bending and torsion conditions, which of the following is (are) zero?

A) $\frac{16K_{fs}T_a}{\pi d^3}$ 0

B) $\frac{16K_{fs}T_m}{\pi d^3}$ $\neq 0$

C) σ_a $\neq 0$

D) $\frac{32K_f M_a}{\pi d^3}$ $\neq 0$

E) σ_m $= 0$

2. (10 pts)
Which of the following is (are) true about the bearing constant reliability contours? (Circle all that are correct)

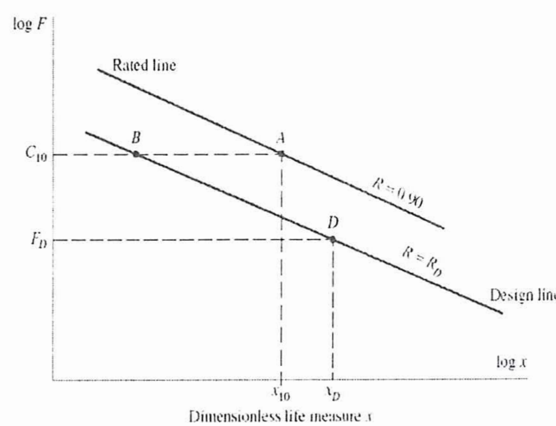
A) $F_B\, x_B^{1/a} = F_D x_D^{1/a}$

B) $F_A = C_{10}$

C) Reliability of A = 90%

D) $X_{10} = 1.0$

E) Reliability of D = Reliability of B



F_R

3. (10 pts)

F_A

An 02-series ball bearing (deep-groove) is to be selected to carry a radial load of 8 kN and a thrust load of $\boxed{4 \text{ kN}}$. The desired life is to be $\boxed{5000 \text{ h}}$ with an inner-ring rotation rate of $\boxed{900}$ rev/min. To find the basic load rating that should be used in selecting a bearing for a reliability goal of $\boxed{0.90}$. Which of the following is (are) true?

R_D

$X_D = \frac{L_D}{L_R} = \frac{2.7\text{E}8}{10^6} = 270$

A) ~~L_{10} = 5000 x 900 x 60~~

B) ~~x_D > 300~~

$C_0 = 40.5$

C) $\theta - X_0 = 4.439$

D) ~~If we select a 02-75mm ball bearing, then $C_0 < 40$ kN.~~

$X_2 =$

E) If we select a 02-75mm ball bearing, then we should select $X_2 = 0.56$

F) If we select a 02-75mm ball bearing, then we should select $Y_2 > 1.45$

$\frac{F_a}{C_0} = \frac{4}{40.5}$ .0988

G) If we select a 02-75mm ball bearing, then we should select $X_1 = 1$

H) If we select a 02-75mm ball bearing, then we should select $Y_1 = 0$

$\frac{F_a}{V F_R} = \frac{4}{(1)(8)} = 0.5$

I) If we select a 02-75mm ball bearing, then e < 0.3

J) ~~V = 1.2~~

$\frac{F_A}{V F_R} > e, \quad X_2 = 0.56 \qquad 1.45 < Y_2 < 1.55 \qquad .28 < e \leq .30$

V = 1 when inner ring rotates

4. (5 pts)

Which of the following is (are) true about bearings (Circle all that are correct)?

A) ~~The deep-groove bearing will take radial load but no thrust load at all.~~

B) ~~The use of a filling notch in the inner and outer rings enables a greater number of balls to be inserted, thus increasing the ~~thrust~~ load capacity.~~

C) Thrust bearings are commonly used in barstools and Lazy Susan turntables.

D) Bearing Life can be defined as the number of revolutions of the inner ring until the first tangible evidence of fatigue.

E) Bearing Life can be defined as the number of hours of use at a standard angular speed until the first tangible evidence of fatigue.

5. (5 pts)

Two mating gears should have the same: (Circle all that are correct)

(A) Module

~~(B) Addendum~~

(C) Diametral pitch

~~(D) Pitch diameter~~

~~(E) Dedendum~~

-2

6. (5 pts)

Which of the following is (are) true about gears (Circle all that are correct)

(A) Spur gears are in general cheaper than helical gears

~~(B) Helical gears' main function is the redirection of power~~

~~(C) In worm gears, gear is input and worm is output~~

(D) Worm gears are used mainly for light load applications

(E) The shafts of two mating helical gears may or may not be parallel to each other

7. (10 pts)

A commercial enclosed gear drive consists of a 20° spur pinion having 16 teeth driving a 48-tooth gear. The pinion speed is 300 rev/min, the face width 2 in, and the diametral pitch 6 teeth/in. The gears are grade 1 steel, through-hardened at 200 Brinell, made to No. 6 quality standards, uncrowned, and are to be accurately and rigidly mounted. Assume a pinion life of 10^8 cycles and a reliability of 0.90. To determine the AGMA bending and contact stresses and the corresponding factors of safety if 5 hp is to be transmitted. Which of the following is (are) true? (Circle all that are correct)

$d = \frac{N}{P}$ $\quad d_{gear} = \frac{48}{6} = 8$

$d_{pin} = \frac{16}{6} = 2.67$

A) Pitch diameter of the gear is greater than 7.5 in

~~B) Pitch diameter of the pinion is less than 2.5 in~~

~~C) Tangential transmitted load is greater than 800 lbf~~

D) $k_v > 1$ $\quad k_v = \left(\frac{A+\sqrt{V}}{A}\right)^B = 1.196$

E) $k_v > 1.1$ $\quad A = 50 + 56(1-B) = 59.77$

F) $F = 2$ $\quad B = .25(12-Q_v)^{2/3} = .8255$

G) In this case, k_s is different for pinion and gear

~~H) $C_{mc} = 0.8$~~

I) $C_{ma} < 0.2$

~~J) $K_o = 2$~~

$W_t = \frac{33000(5hp)}{V} = 785$ lbf

$V = \frac{\pi d n}{12} = \frac{\pi(2.67\text{in})(300\text{rev/min})}{12} = 210$ ft/min

$A + B(2) + C(4)$

comm. enclosed → $.127 + 2(.0158) - .93(10^{-4})(4)$

8. (10 pts)

Continue from problem 7, which of the following is (are) true? (Circle all that are correct)

A) $C_p > 2{,}000$ $\quad C_p = 2300\sqrt{psi}$

~~B) $C_p > 2{,}500$~~

C) $I > 0.1$ $\quad I = \frac{\cos 20 \sin 20}{2 m_N}\left(\frac{m_G}{m_G+1}\right)$

D) $I > 0.11$ $\quad = .1205$

E) $I > 0.12$

$M_G = \frac{48}{16} = 3$

$m_N = 1$ for spur gears

~~F) In this case, S_t is different for pinion and gear~~

~~G) In this case, S_c is different for pinion and gear~~

H) In this case, Y_N is different for pinion and gear

I) In this case, Z_N is different for pinion and gear

J) $C_H = 1$

$S_c = 322(200) + 29\,100$ psi

= Same

S_t, allowable bending stress Same

9. (15 pts)
If the diameter of a shaft is 1.49 inch, the slope at a gear seat on the shaft is found to be - 0.000545 rad, and the design factor is 1.5. Do you need to redesign the shaft? If so, what is the shaft redesigned ?

p 379

$$d_{new} = 1.49 \left| \frac{1.5(-.000545)}{(-.0005)} \right|^{1/4}$$

$d_{new} = 1.685$ inches or 1.75 to be more standard in sizing

Slopes for all shafts Table 7-2

uncrowned spur gear:

allowable slope < .0005

to be safe, redesign the shaft.

10. (20 pts)
An interference fit of a cast-iron hub of a gear on a steel shaft is required. Make the dimensional decisions for a 45-mm basic size locational transition fit.

Tol. grades

→ H7/k6 ← (go with tighter tolerance)

H7/n6

hole IT7 = .025 mm

shaft IT6 = .016 mm

k = +.002 mm

hole $D_{min} = D = \boxed{45.000\text{ mm}}$

$D_{max} = D + \Delta D = \boxed{45.025\text{ mm}}$

shaft $d_{min} = d + \delta_F = 45 + .002 = \boxed{45.002\text{ mm}}$

$d_{max} = d + \delta_F + \Delta d = 45 + .002 + .016 = \boxed{45.018\text{ mm}}$

δ_F = fundamental deviation = .002 mm

= lower deviation (k) =

$\frac{88}{100}$

ME 208B F10 Exam #2 70 minutes Name: David Malawey

Open Textbook; Closed Notes, Only one sheet of paper allowed

1. (5 pts)
Which of the following is (are) true (Circle whichever is correct)
(a) Aluminum alloys have no endurance limit
(b) S_e implies the fatigue strength of the real world specimen
(c) The stress required to cause fatigue is always below the yield strength
(d) S-N diagram is applicable to alternating (completely reverse) loads
(e) Fatigue is always due to time varying loads

2.6 vs

$K_f \to 1 \Leftarrow 1 + q\ (k_t - 1)$

$K_f = 1 + .8\ (2.6 - 1)$

2. (5 pts)
Which of the following is (are) true (Circle whichever is correct)
(a) Fatigue stress concentration factor is in general less than stress concentration factor
(b) When notch sensitive is closer to zero, stress concentration factor is closer to one
(c) When notch sensitive is closer to zero, stress concentration factor is closer to zero $q \to 1$ $k_t \to k_f$
(d) When notch sensitive is closer to one, stress concentration factor is closer to fatigue stress concentration factor
(e) None of the above.

-2

3. (5 pts)
A gear is to be chosen with a given pitch diameter d. The diametral pitch P and the pressure angle ϕ must be chosen. Indicate whether the following parameters will be affected by the choice of diametral pitch or pressure angle.(Circle P or ϕ or both)

(a) (P ϕ) Clearance
(b) (P ϕ) Number of teeth ✓
(c) P (ϕ) Base circle diameter ✓
(d) (P ϕ) Dedendum ✓
(e) P (ϕ) Transmitted force

P = teeth/in

Clearance = - dedendum circ + base circ

-1

4. (5 pts)
The point of contact between two mating gear teeth is (circle each that applies)
(a) always on the center line.
(b) always at the pitch point.
(c) always on the pressure line.
(d) always on the pitch circle.
(e) always on a line tangential to the base circle.

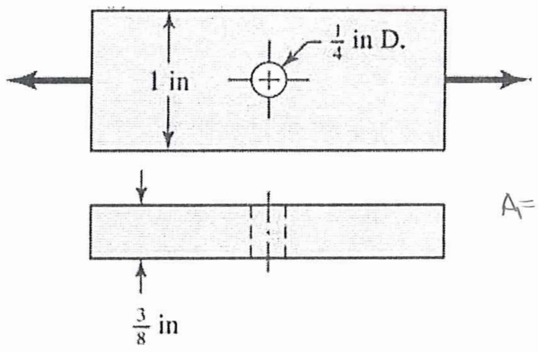
5. (5 pts)

A rotating shaft is simply supported with ball bearings on both ends. The shaft is subjected to a critical bending moment and an axial compression load from both ends. We can conclude that this shaft (circle each that applies)

~~**(a)**~~ **is under completely reverse loading condition.**
(b) is under fluctuating stress situation.
~~**(c)**~~ **has negative amplitude stress.**
(d) has negative midrange stress. (comp.)
~~**(e)**~~ **none of the above.**

6. (10 pts)

The cold-drawn AISI 1040 steel bar shown in the figure is subjected to a completely reversed $\boxed{\text{axial}}$ load fluctuating between 6200 lbs in compression to 6200 lbs in tension. Circle each that applies.



A= .28125

(a) $\sigma_a > 22$ **kpsi**
~~**(b)**~~ ~~$k_a < 0.8$~~
(c) $k_b > 0.98$
(d) $k_c < 0.9$
(e) This bar is under fluctuating stress situation

(-2)

$\sigma_a = K_f \frac{F}{A} =$

$S_{ut} = 85$ kpsi
$S_y = 71$kpsi

$k_a = 2.70(85)^{-.265} = .832$
$k_b = 1$
$k_c = .85$

7. (10 pts)

The same as Problem 6, circle each that applies.

~~**(a)**~~ ~~$S_e' > 45$ **kpsi**~~
(b) $q > 0.8$
(c) $k_t > 2.2$
(d) If n_f is found to be 0.6, this means that the structure will fail in less than **one cycle**
~~**(e)**~~ **None of the above**

(-2)

$k_a k_b k_c \cancel{k_d} \cancel{k_e} k_f$

$r = \frac{1}{8} = .125$

$k_t \Rightarrow \frac{d}{w} = .25$

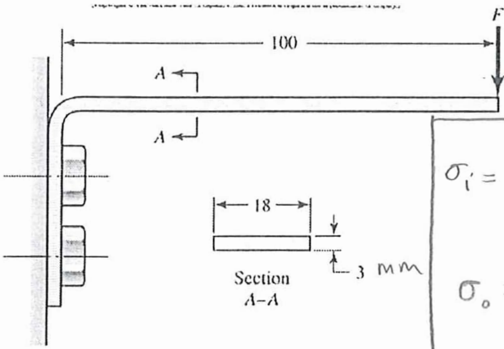
$k_t = 2.42$

$S_e' = \frac{85}{2} = 42.5$kpsi

$q = .82$

8. (10 pts)

The figure is a drawing of a 3- by 18-mm latching spring. A preload is obtained during assembly by shimming under the bolts to obtain an estimated initial deflection of 2 mm. The latching operation itself requires an additional deflection of exactly 4 mm, thus a total deflection of 6 mm. The material is ground high carbon steel, bent then hardened and tempered to a minimum hardness of $\boxed{\text{490 Bhn.}}$ The radius of the bend is 3 mm. Circle each that applies.



$\boxed{\text{P.118-119}}$

$\sigma_i = \dfrac{Mc_i}{Aer_i}$ M is (-) so σ_i is (-)

$\sigma_o = \dfrac{Mc_o}{Aer_o}$ $\sigma_i > \sigma_o$

$S_{ut} = 3.4\,H_B\,(MPa)$
$= 3.4(490)$
$= 1666$ MPa

(a) S_{ut} ~~S_t~~ > 1600 Mpa
(b) d_e > 5.8 mm
~~(c)~~ S_e' > 720 Mpa
(d) **The critical location has the largest compression force**
(e) **The critical location has the largest bending moment**

$d_e = 0.808\sqrt{hb} = .808\sqrt{18 \cdot 3} = 5.94$ mm

$S_{ut} > 1400$ MPa
$S_e' = 700$ MPa

9. (20 pts)

The same as Problem 8. If in the inner area, we calculated $\sigma_a = 244.4$ Mpa, σ_m ~~σ_a~~ = -488.8 Mpa, $S_y = 1504$ Mpa, and S_e = 605 ~~1504~~ Mpa, find safety factors n_y= ? and n_f= ?

$$n_y = \frac{S_y}{\sigma_{max}} = \frac{1504\ MPa}{|-488 - 244|} \quad \boxed{n_y = 2.054}$$

$$\frac{n_f \sigma_a}{S_e} = 1 \qquad n_f = \frac{S_e}{\sigma_a} = \frac{605\ MPa}{244.4\ MPa} \quad \boxed{n_f = 2.475}$$

-5

10. (5 pts)

Estimate S_e' for AISI 4340 steel heat-treated to a tensile strength of 250 kpsi.

$S_{ut} = 250\,kpsi$

$\boxed{S_e' = 100\,kpsi}$

11. (20 Pts)

An AISI 1020 cold-drawn part is loaded with stresses $\sigma_{max} = 60$ kpsi, $\sigma_{min} = 55$ kpsi. The completely adjusted endurance limit is $S_e = 30$ kpsi. On the plot of σ_a vs σ_m, sketch the modified Goodman line and the yield line.

- **a) Mark on the plot the fluctuating stress for the loaded part. (<u>Label the key numbers</u> on the plot.)**
- **b) From the sketch, determine whether the predicted life has an infinite life, finite life, or just fail in one cycle** (No calculation is necessary.)

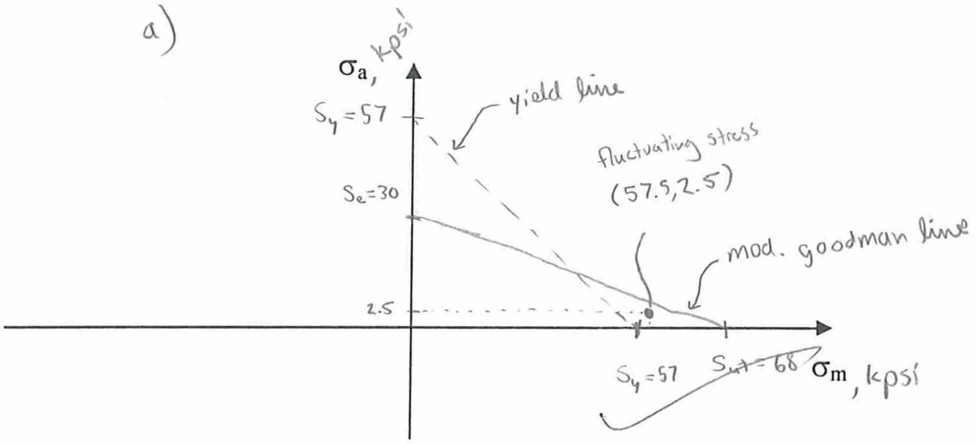
$\sigma_a = \frac{60-55}{2} = 2.5\,kpsi$

$\sigma_m = \frac{60+55}{2} = 57.5\,kpsi$

$S_y = 57\,kpsi$

$S_{ut} = 68\,kpsi$

a)



b) fail in one cycle

69/100

ME 208B F10 Exam #1 70 minutes Name: David Malawey

Open Textbook; Closed Notes, Only single page (one side) crib sheet allowed (you need to turn in the crib sheet with your exam)

1. (5 pts)
The theoretical stress concentration factor K_t, is dependent on which of the following? (Circle all that are correct)

a) Part geometry
b) Load magnitude
c) Stress magnitude
d) Type of load (e.g. Axial, Bending, etc.)
e) Material

2. (12 pts)
Identify a static failure theory for each of the following descriptions: (Write at the end of each description, using notation: DCM, DE, BCM, MM, MSS, or MNS)

(a) Failure envelope on the principal stress axis is a square: MNS

(b) Need to calculate von Mises stress: DE

(c) For materials such as case iron, generally conservative: BCM

(d) For ductile materials, generally more conservative: MSS

-1

(e) For materials such as low (ductile) carbon steel, generally more accurate: DE

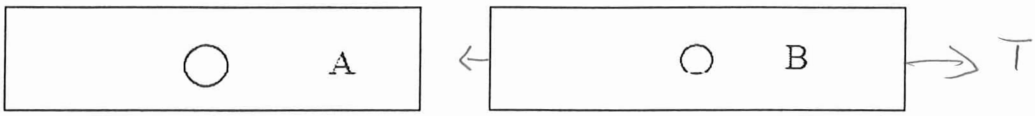
(f) A theory for brittle material, but the same theory can be used for both ductile and brittle materials: ~~BCM~~/DCM

3. (8 pts)
Stress concentration can generally be neglected for: (Circle all that apply)

(a) mild steel
(b) case iron
(c) structures under static loads
(d) structures under dynamic loads
(e) low carbon steel with static loads
(f) ductile materials with dynamic loads
(g) brittle materials with dynamic loads
(h) none of the above

4. (6 pts)

$\frac{d}{w}$ small



Two plates are identical except for the size of the hole. They are subject to the same tension load.

(a) If stress concentration is considered, which part has the higher stress Concentration factor?

Circle one: A B Can't tell

(b) If stress concentration is ignored, which part has the higher nominal stress?

Circle one: A B Can't tell

(c) If stress concentration is considered, which part has the higher peak stress?

Circle one: A B Can't tell

5. (15 pts)

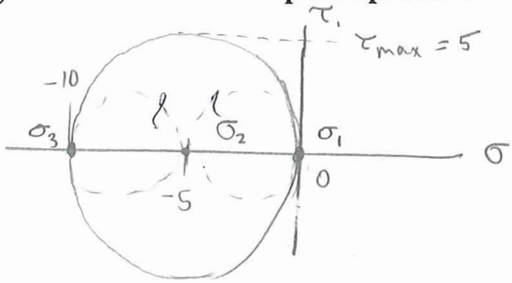
a) Sketch the 3-D Mohr's circles (could have 3 circles) corresponding to the condition: $\sigma_x = -2$, $\sigma_y = -8$, and $\tau_{xy} = 4$ cw.

b) What are the three principal stresses?

$$\sigma_{1,3} = \frac{\sigma_x+\sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x-\sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

$$= \frac{-2-8}{2} \pm \sqrt{\left(\frac{6}{2}\right)^2 + (-4)^2}$$

$$= \underbrace{-5}_{C} \pm \underbrace{5}_{R}$$



-2

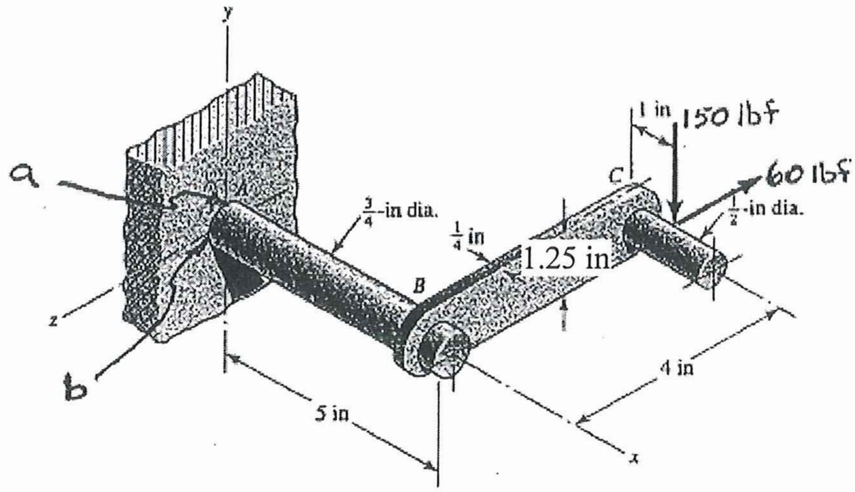
$\sigma_1 = 0$

$\sigma_3 = -10$

$\sigma_2 = -5$

-1

6. (10 pts)
Analyze the stress situation in beam AB by obtaining the following information.



$I = \frac{\pi D^4}{64} = .0155$

$J = 2I = .0311$

$A = \frac{\pi D^2}{4} = .4418$

$Y =$

Compute all stresses acting at points a (top), and b (front) on the surface of the beam at the wall. Transverse shear may be neglected. Circle all that are correct:

a) ~~Shear stress $\tau_a = \tau_b$~~ ×
b) ~~$\tau_a > 7{,}000$ psi~~ ×
c) Bending stress $\sigma_a < 21{,}000$ psi
d) ~~Bending stress $\sigma_b > 9{,}000$ psi~~ ×
e) ~~None of the above~~ ×

-6

$\tau_a = \frac{Tr}{J} = \frac{150\ \text{lbf}\ (.75)}{.0311} = 3617$ psi

$+ \frac{V}{A} = \frac{60\ \text{lbf}}{.4418} = 135.8$

$\sigma_a = \frac{My}{I} = \frac{150(4)(.375)}{.0155}$

$\sigma_B = \frac{My}{I} = \frac{60(5)(.375)}{.0155} = 7258$

-Z, x, A: σ_x Bending (150 lbf) $\frac{My}{I}$

$\tau_{xy} = \frac{V}{A}$ shear 60 lbf

$+ 150\ \frac{Tr}{J}$ 150 lbf

Y, x, B: $\sigma_x =$ Bending (60 lbf) $\frac{My}{I}$

$\tau_{xy} = -\frac{V}{A}$ shear 150 lbf

$+ 150\ \frac{TR}{J}$ 150

14,516

7. (10 pts)
Continue from Problem 6. Determine the location of the critical stress element. Circle all that are correct:

a) Critical stress is NOT located at point "a"
b) Bending stress on critical stress element $\sigma_{critical} < 23{,}000$ psi
c) ~~$\sigma_{critical} < 25{,}000$ psi~~
d) Torque on critical stress element $\tau_{critical} > 8{,}000$ psi
e) ~~None of the above~~

-5

A] $\tau_{xy} = 3753$ psi
$\sigma_x = 14{,}516$ psi

b] $\tau_{xy} = 14{,}177$ psi
$\sigma_x = 7{,}258$ psi

$\tau_{xy}(b) = -\frac{150}{.4418} + \frac{150(4)(.375)}{.0155} = 14177$ psi

$\sigma' = (\sigma_x^2 + 3\tau_{xy}^2)^{1/2} = 15905$ psi (A) OR 25606 psi (B)

8. (10 pts)

A hot-rolled steel bar has a minimum yield strength in tension and compression of 50 kpsi. To use the distortion-energy and maximum-shear-stress theories to determine the factors of safety, n, for the following plane stress states:

$$\sigma_x = -6 \text{ kpsi}, \sigma_y = -10 \text{ kpsi}, \tau_{xy} = -5 \text{ kpsi}$$

Circle all that are correct:

a) **If using maximum-shear-stress theory, n > 4**
b) **If using maximum-shear-stress theory, n > 5**
c) **~~If using distortion-energy theory, n > 5~~**
d) **~~If using distortion-energy theory, n < 4~~**
e) **None of the above**

-5

MSS: $\frac{\sigma_1 - \sigma_3}{2} = \frac{S_y}{n}$ radius $= \frac{S_y}{n}$ $\frac{-6 + -10}{2}$

$R \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2} = \sqrt{\left(\frac{-6+10}{2}\right)^2 + (-5)^2} = \sqrt{4+25} = 5.38$

$5.38 \text{ kpsi} = \frac{50 \text{kpsi}}{n}$ $\boxed{n = 9.294}$

D.E. $\sigma' = \frac{S_y}{n}$

$\sigma' = (\sigma_x^2 - \sigma_x\sigma_y + \sigma_y^2 + 3\tau_{xy}^2)^{1/2}$

$= [36 - 60 + 100 + 3(25)]^{1/2}$

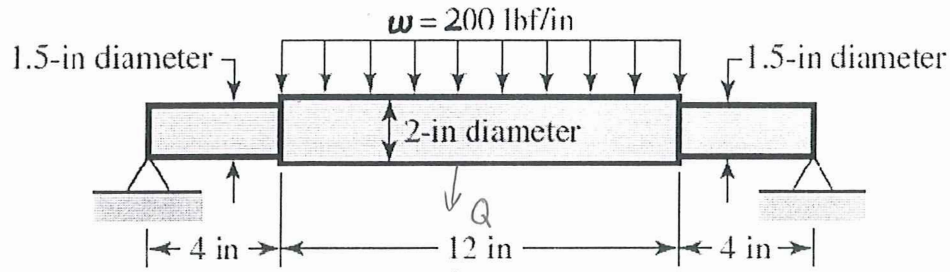
$= 12.29 \text{ kpsi}$

$\boxed{n = \frac{50}{12.29} = 4.068}$

9. (24pts)

Determine the maximum deflection for the steel beam shown **using Castigliano's theorem (Energy method).**

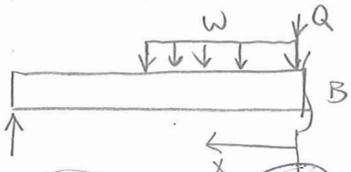
$\frac{\pi D^4}{64}$



$I_{2in} = .7854$

$I_{1.5in} = .~~2485~~$

$E =$



$R = \frac{12w}{2} + \frac{Q}{2}$ -2

$\frac{\partial M}{\partial Q}$

$M_B = \begin{cases} xw\left(\frac{x}{2}\right) + & 0 < x < 6 \Rightarrow 0 \\ ~~6w(3)~~ & 6 < x < 10 \Rightarrow 0 \\ xw\frac{x}{2} - \left(xw + \frac{Q}{2}\right)(10) & x = 6 \Rightarrow 5 \end{cases}$

$0 \le x \le 4$

$M = 1200x - \frac{Q}{2}x$

$4 < x \le 10$

$M = 1200x - \frac{Qx}{2} - \frac{200}{2}(x-4)^2$

-1

not certain how to choose the intervals

$\delta_i = \frac{\partial U}{\partial F_i} = \int \frac{1}{EI}\left(M \frac{\partial M}{\partial F_i}\right) dx$

$= \int_0^6 \frac{1}{EI_{2in}}\left(\frac{x^2 w}{2} \cdot 0\right) + \int_6^{10} \frac{1}{EI_2}\left(\cdot 0\right) + \int_0^{10} \frac{1}{EI_{1in}}\left[\left(\frac{x^2 w}{2} - 5xwQ\right)(5)\right] dx$

-2

$\frac{5}{EI_{1in}}\left[\frac{x^3 w}{6} - \frac{5w\cancel{Q}x^2}{2}\right]_0^6 = \frac{5w}{EI_{1in}}\left[\frac{6^3}{6}\right] = \frac{5(200)(36)}{30mpsi\,(.2458)}$

$\boxed{\delta = .0049 \text{ in}}$

δ will be greater than this

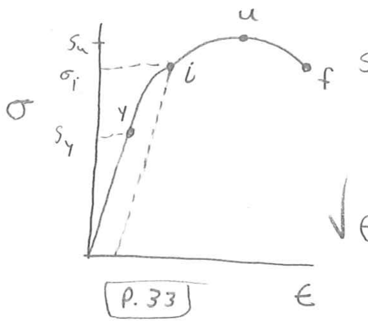
208 Exam 1 Formulas

Mohr's circle

$$R = \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

$$\sigma_{avg} = C = \frac{\sigma_x + \sigma_y}{2}$$

$x: (\sigma_x, \tau_{xy})$ $y: (\sigma_y, -\tau_{xy})$



$$\sigma = \frac{P}{A}$$

$$\epsilon = \frac{\delta}{L}$$

$\epsilon_f > 5\% \Rightarrow$ ductile material

$$\sigma_{bend} = \frac{My}{I}$$

Stress Concentration:

usu. neglect for ductile, static loads

$k_t = f$(geometry, type of load)

$$\sigma' = (\sigma_A^2 - \sigma_A\sigma_B + \sigma_B^2)^{1/2}$$

brittle bar under torsion breaks at 45°

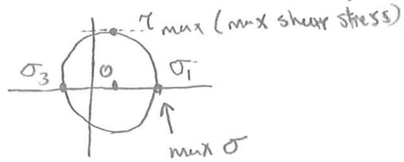
Pure torsion $\Rightarrow$ $\sigma_1 = \tau$

$$\sigma_{1,2} = \frac{\sigma_x + \sigma_y}{2} \pm R$$

tension (+)
compression (−)


τ_{xy} (+)

Max shear stress failure theory:



factor of safety $n = \frac{S_y}{\sigma'}$

Ductile mat'l : limited by shear strength
Brittle mat'l : limited by tensile strength

Principal stresses, σ_1, σ_2

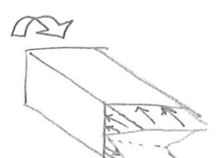
Deflection

$$\delta = \frac{Fl}{AE} \quad \theta = \frac{Tl}{JG} \quad J_z = I_{xx} + I_{yy}$$

Strain energy

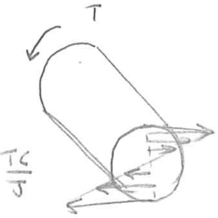
U (tension, torsion, shear, bending, etc) p 162

M



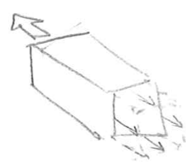
$\sigma = \frac{My}{I}$

T



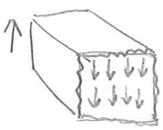
$\tau = \frac{Tc}{J}$

N



$\sigma = \frac{N}{A}$

V



$\tau = \frac{TQ}{Ib}$

* c goes where stress element is (in this case, r)

* $Q = \int_y^c y\,dA$, b = width of beam

* $J = I_{xx} + I_{yy}$

Find n_f, n_y in bending, rotating shaft, cycles

1) find critical location

√ 2) find Se. Se = endurance limit, actual conditions (Kpsi, MPa) $\boxed{S_e < S_e'}$ p 283, Se'

√ 3) find S_f. • if $\boxed{N \text{ cycles} > 10^6,\ S_f = S_e'}$
fatigue strength • if N cycles given, $\boxed{S_f = aN^b}$ p. 285

4) find σ_a (for rotating, $\sigma_m = 0$)

√ 5) plug S_f & σ_a into $\boxed{n_f = \frac{S_f}{\sigma_a}}$ OR $n_f = \frac{S_e}{\sigma_a}$

√ 6) $\boxed{n_y = \frac{S_y}{|\sigma_{max}|}}$

Find n_f, n_y where $\sigma_m \neq 0$ NOTCH

√ 1) find Se based on Se' & mod. factors 287 OR 345

√ 2) find K_f based on stress concentration A15 & q, notch sensitivity 295

√ 3) for fatigue, find σ_a (amplitude) & σ_m (average stress). These give pt. A, 305

√ • find criterion for n_f using σ's, Se, Sut $\boxed{\text{p 306, criterion}}$

√ * σ_a & σ_m for fatigue $= \frac{F}{A} K_f$ or $\frac{My}{I} K_f$

√ 4) for yield, $n_y = \frac{S_y}{|\sigma_{max}|}$

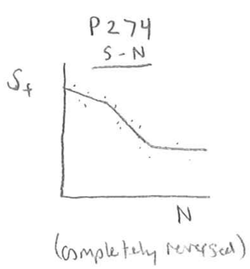
√ * S_y: find in A-20 OR given, 0.9 Sut, OR, f(Sut = f(BHN)) p 41

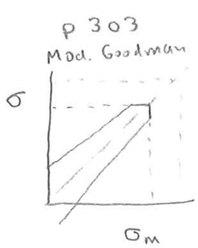
Solve for bearing reactions: Example p. 711

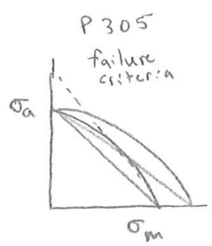
√ 1) find W(i, j, k) p 709
• use γ to get components, use power formula to get force

√ 2) find R from bearing to G

√ 3) Σ moments $= \Sigma (R_{B/G} \times F) + T = 0$



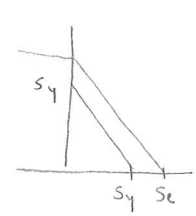




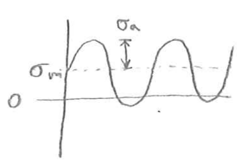
Torque $= \frac{\text{Power}}{\omega \text{ (radians)}}$

log: catalog > log

this can happen



tooth thickness $= \frac{p}{2}$



when $\sigma_m = 0$, $n_f = \frac{S_e}{\sigma_a}$ > 1 infinite life
< 1 finite life

Exam practice

ME 208B SP09 Final Exam 80 minutes Name:____________________

Open Textbook; Closed Notes, Only two sheets of paper allowed (which need to be turned in at the end of the exam).

1. (3 pts)
Which of the following static failure theories is considered more conservative?
- **a) Distortion energy**
- **b) Octahedral shear stress theory**
- **c) Von-Mises theory**
- **d) MSS theory**
- **e) They are all the same**

Solution:

d

2. (5 pts)

The theoretical stress concentration factor k_t is dependent on which of the following? (Circle all that are correct)
- **a) Material**
- **b) Geometry**
- **c) Magnitude of load**
- **d) Type of load**
- **e) Material hardness**

Solution:

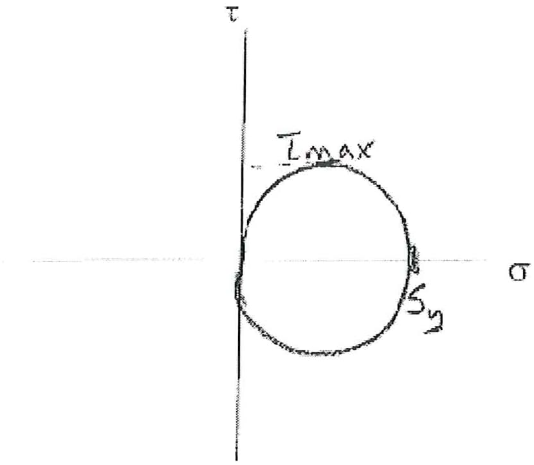
b, d

3. (5 pts)
Circle all that are about Distortion Energy Theory:
(a) For ductile materials
(b) Failure envelope on the principal stress axis is elliptical.
(c) Predicts factor of safety by comparison of a von Mises stress to a strength.
(d) In a pure shear case, it predicts failure exactly the same as MSS theory
(e) $S_{ut} = S_{ue} = 100$ kpsi, $S_y = 70$ kpsi, An analysis is being performed to check why a part failed, so a theory typical of actual failure data is desirable.

Solution:
b, c, e

4. (5 pts)
Sketch a Mohr's circle for a tension test specimen at yielding. Label τ_{max} and S_y.

Solution:



5. (5 pts)
A mechanical component is loaded with alternating and mean stresses; $\sigma_a = 20$ kpsi and $\sigma_m = -10$ kpsi. $S_e = 40$ kpsi (completely adjusted), $S_{yt} = S_{yc} = 45$ kpsi, $S_{ut} = 60$ kpsi. Determine the factor of safety for fatigue (Based on Goodman's). (It is not required to estimate the life.)

Solution:

$n_f = S_e / \sigma_a = 40/20 = 2.0$

5. (15 pts)

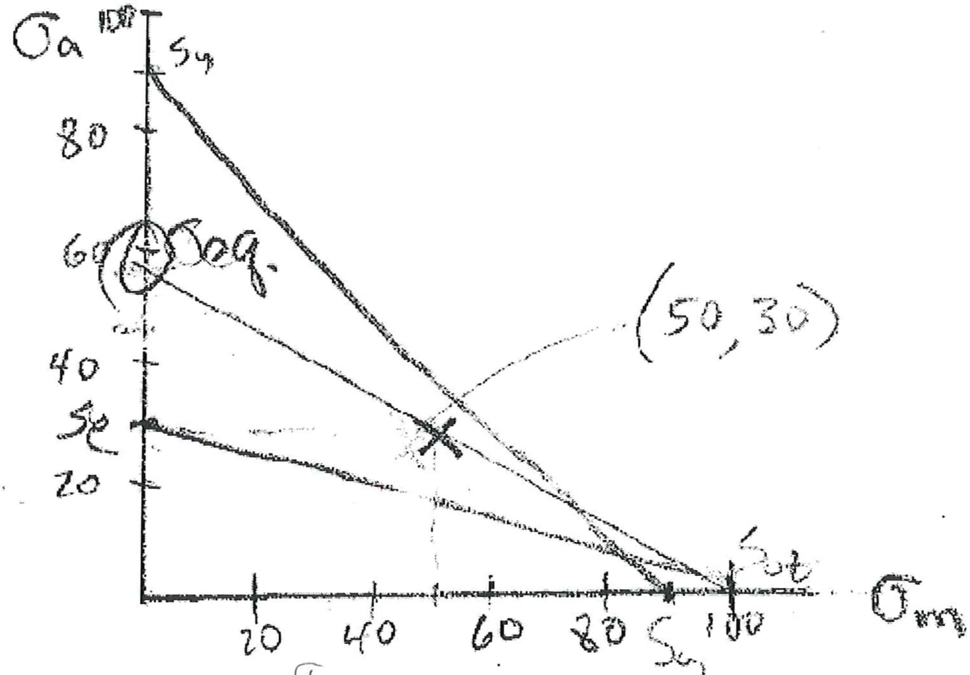
A part is loaded with stresses $\sigma_{max} = 80$ kpsi, $\sigma_{min} = 20$ kpsi. The completely adjusted endurance limit is $S_e = 30$ kpsi. Material properties are $S_y = 90$ kpsi, $S_{ut} = 100$ kpsi.

(a) On the plot of σ_a vs σ_m, sketch the modified Goodman line and the yield line.
(b) Mark on the plot the fluctuating stress for the loaded part.
(c) What is the estimated life (finite, infinite?) of this case (No calculation needed)

Solution:

$$\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2} = \frac{80+20}{2} = 50$$

$$\sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2} = \frac{80-20}{2} = 30$$



6. **(15 pts)**
A gear with a nominal 60 mm bore is to be pressed onto a shaft with a medium drive fit. Specify the maximum and minimum dimensions for both the hole and the shaft.

Solution:

H7/s6

Table A-11, $\Delta D = 0.03$ mm $\quad \Delta d = 0.019$ mm

Table A-12, $\delta_F = 0.053$ mm

$D_{max} = D + \Delta D = 60 + 0.03$

$\boxed{D_{max} = 60.03 \text{ mm} \quad D_{min} = 60.00 \text{ mm}}$

$d_{min} = d + \delta_F = 60 + 0.053$

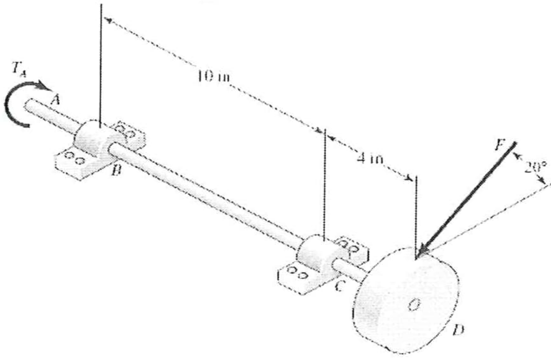
$\boxed{d_{min} = 60.053 \text{ mm}}$

$d_{max} = d + \delta_F + \Delta d = 60 + 0.053 + 0.019$

$\boxed{d_{max} = 60.072 \text{ mm}}$

7. (22 pts)

The rotating solid steel shaft is simply supported by bearings at points B and C and is driven by a gear (not shown) which meshes with the spur gear at D, which has a 6-in pitch diameter. The force F from the drive gear acts at a pressure angle of 20°. The shaft transmits a torque to point A of $T_A = \boxed{3000\ \text{lbf} \cdot \text{in}}$. The shaft is machined from steel with $S_y = 60$ kpsi and $S_{ut} = 80$ kpsi. Using a factor of safety of 2.5, determine the minimum allowable diameter of the 10 in section of the shaft based on an ASME fatigue-failure analysis. Assume sharp fillet radii at the bearing shoulders for estimating stress concentration factors. (Hint: assume d = 2 in to estimate the size factor. Also due to time constraint, no iteration is needed to find the final design)



Solution:

$$F\cos 20°(d/2) = T, \quad F = 2T/(d\cos 20°) = 2(3000)/(6\cos 20°) = 1064\ \text{lbf}$$

$$M_C = 1064(4) = 4257\ \text{lbf}\cdot\text{in}$$

For sharp fillet radii at the shoulders, from Table 7-1, $K_t = 2.7$, and $K_{ts} = 2.2$. Examining Figs. 6-20 and 6-21, with $S_{ut} = 80$ kpsi, conservatively estimate $q = 0.8$ and $q_s = 0.9$. These estimates can be checked once a specific fillet radius is determined.

Eq. (6-32): $\quad K_f = 1 + (0.8)(2.7 - 1) = 2.4$

$$K_{fs} = 1 + (0.9)(2.2 - 1) = 2.1$$

$$k_a = 2.70(80)^{-0.265} = 0.845$$

Assume $d = 2.00$ in to estimate the size factor,

$$k_b = \left(\frac{2}{0.3}\right)^{-0.107} = 0.816$$

$$S_e = 0.845(0.816)(0.5)(80) = 27.6\ \text{kpsi}$$

Selecting the DE-ASME Elliptic criteria, use Eq. (7-12) with $M_m = T_a = 0$.

$$d = \left\{\frac{16(2.5)}{\pi}\left[4\left(\frac{2.4(4257)}{27\,600}\right)^2 + 3\left(\frac{2.1(3000)}{60\,000}\right)^2\right]^{1/2}\right\}^{1/3} = 2.133\ \text{in}$$

8. (25 pts)
A 02-series deep-groove ball bearing is to be selected to carry a radial load of 8 kN and a thrust load of 4 kN. The desired life L_D is to be 5000 h with an inner-ring rotation rate of 900 rev/min. What is the basic load rating that should be used in selecting a bearing for a reliability goal of 0.97? (Hint: if you are running out of time, you only need to do one full iteration so that you can get most of the points)

Solution:

8. $$X_D = \frac{5000 \times 900 \times 60}{10^6} = 270$$

$$C_{10} = 10.6\left[\frac{270}{0.02+4.439[\ln(1/0.97)]^{1/1.483}}\right]^{1/3}$$

$$= 10.6\left[\frac{270}{0.02+\cancel{4.439}\ 0.422}\right]^{1/3}$$

$$= 10.6 \times 8.4847 = 89.94$$

Trial # 1: select 02-90 mm

$C_{10} = 95.6 \quad C_0 = 62$ KN

$$\frac{F_a}{C_0} = \frac{4}{62} = 0.0645$$

$$\frac{Y_2 - 1.71}{1.83-1.71} = \frac{0.0645-0.056}{0.07-0.056} = 0.607$$

$Y_2 = 1.661$, $F_e = 0.56(8) + 1.661(4)$
$= 11.12$ KN.

$$C_{10} = 11.12\left[\frac{270}{0.442}\right]^{1/3}$$

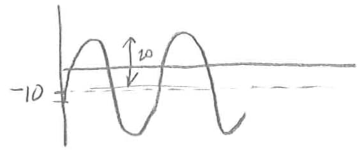
$= 94.33 < 95.6$ KN.

O.K.

4)

$\sigma_a = 20$ kpsi

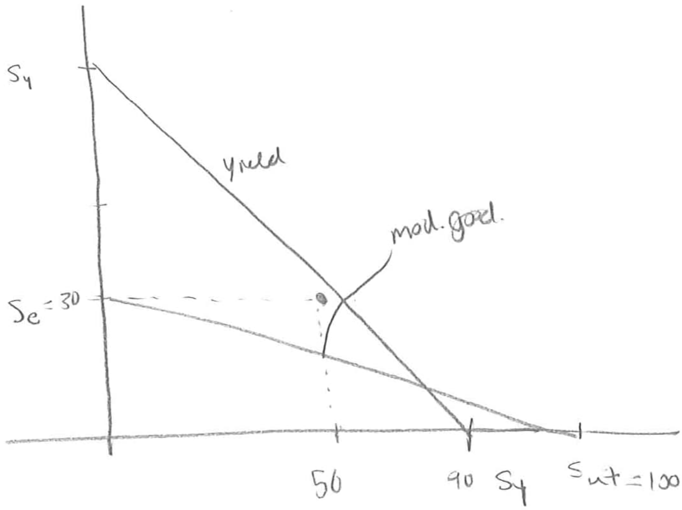
$\sigma_m = -10$ kpsi



$$\frac{\sigma_a}{S_e} + \frac{\sigma_m}{S_{ut}} = \frac{1}{n}$$

$$\frac{20}{40} + \frac{-10\cancel{0}}{\cancel{6}0 \text{ kpsi}} = \frac{1}{n} \qquad \boxed{n=2}$$

5)



$\sigma_m = \frac{80+20}{2} = 50$

$\sigma_a = \frac{80-20}{2} = 30$

$\boxed{\text{finite life}}$

6) 60 mm medium drive fit H7/s6

IT6 = .019 IT7 = .030

$d_{min} = d + \delta_f = 60 + .053$

$d_{max} = d + \delta_f + \Delta d = 60 + .019 + .053$

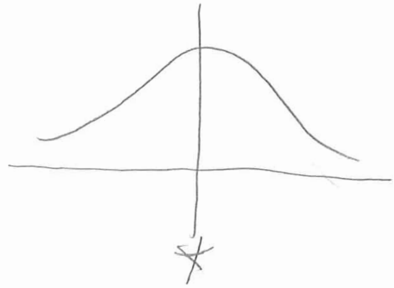
$\boxed{\begin{matrix} 60.053 \\ 60.072 \end{matrix}}$

$D_{min} = D = 60$ mm

$D_{max} = D + \delta D = 60 + .030$

$\boxed{\begin{matrix} 60.000 \\ 60.030 \end{matrix}}$

a. The magnitude of plane stresses
b. The direction of plane stresses
c. The magnitude of shear stress
d. The direction of shear stress



2. This type of material is limited by tensile strength
   a. Ductile
   b. Brittle
   c. Low carbon steels
3. Which is NOT true of the Modified Goodman Diagram?
   a. It is designed for fluctuating loads
   b. When $\sigma_m<0$, sm has no effect in calculating factor of safety
   c. When $\sigma_m= 0$, $n_f =$
   d. It cannot be used when midrange stress is zero
4. Infinite Life for fatigue strength is defined as
   a. 10^5 cycles
   b. 10^6 cycles
   c. 10^7 cycles
   d. 10^8 cycles
5. Which failure theory line passes closest to actual test failure points? for ductile materials
   a. Modified mohr best for brittle
   b. Ductile coulomb-mohr
   c. Distortion energy theory
   d. Britle coulomb mohr
6. Which failure theory is considered conservative? ~~For ductile materials~~
   a. Distortion Energy
   b. Maximum Shear stress
   c. Ductile Coulomb-Mohr
   d. Modified Mohr
7. Which of the following is not a factor of S_e, fatigue endurance limit?
   a. Surface condition factor
   b. Stress concentration factor
   c. Temperature modification factor
   d. Reliability factor
8. Temperature is not an important fatigue factor if it is less than:
   a. 25 C
   b. 250 C
   c. 250 K
   d. 400 C
9. If no reliability factor is used, what is the statistical reliability?
   a. 50%
   b. 90%
   c. 99%
   d. 99.9%
10. Equivalent diameter is used in the size factor k_b for
    a. An axial-loaded noncircular cross-section
    b. A non-rotating round bar
    c. A rotating round bar
    d. A rotating noncircular cross-section

11. Stress concentration factor is NOT a function of which?
    a. Part geometry
    b. Notch radius
    c. Part material
    d. Type of loading

12. A part will fail in less than 1 cycle if: always
    a. σ_{max} exceeds S_y
    b. σ_a is above the modified Goodman Line
    c. σ_m is negative
    d. σ_a is greater than σ_m

13. For a basic hole size of 20mm and sliding fit, the Fundamental deviation will be:
    a. .021
    b. -.013
    c. .013
    d. -.007

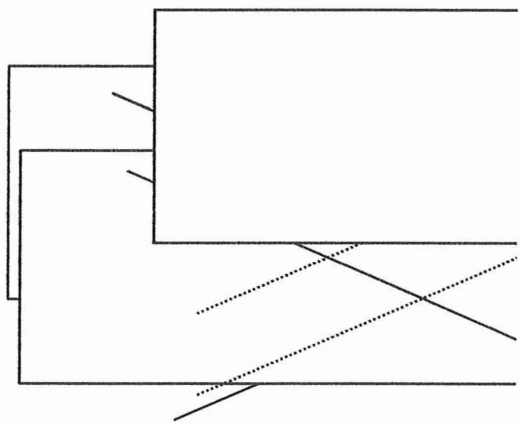
14. Which type of bearing would you choose for a design that requires a large radial load and a moderate thrust load and involves some shaft deflection
    a. Deep groove bearing
    b. Straight roller bearing
    c. Thrust bearing
    d. Double row ball bearing

15. Rating life is NOT
    a. L_{10} life
    b. Minimum life
    c. C^{10}
    d. Given in revolutions

16. Which of the following is always the same for gear and pinion?
    a. Y_N
    b. W^t
    c. C_H
    d. Geometry factor J

17. Stress concentration can be neglected for:
    a. Brittle materials, dynamic loads
    b. Brittle materials, static loads
    c. Ductile materials, dynamic loads
    d. Ductile materials, static loads

18. Which is true of the diagram to the right?
    a. Midrange stress is zero
    b. σ_a is zero
    c. σ_a is 6kpsi
    d. σ_m is 5kpsi

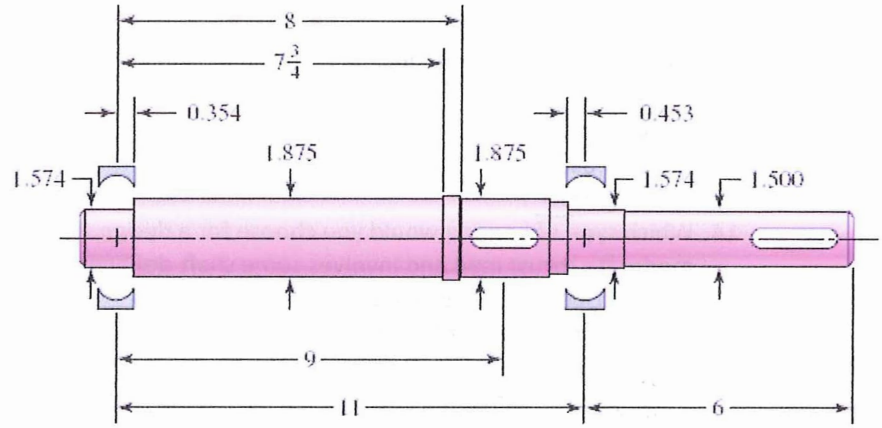


19. Which metals have ~~no~~ an endurance limit?
    a. Aluminum and brass
    b. Steels
    c. Steel and aluminum
    d. Aluminum

20. The typical S-N diagram is applicable for which kind of loading?
    a. Static

2. In the figure is a proposed shaft design to be used for the input shaft a in Prob. 7–7. A ball bearing is planned for the left bearing, and a cylindrical roller bearing for the right. (a) Determine the fatigue factor of safety by evaluating at pinion seat keyway (assuming end mill keyway radius = 0.01"). Use ASME Elliptic fatigue failure criteria. Also ensure that the shaft does not yield in the first load cycle. (b) To avoid complex calculation, approximate the shaft as a uniform shaft with diameter of 1.875 in, and check the design for adequacy with respect to deformation, according to the recommendations in Table 7–2. Use 1030 HR.

Shoulder fillets at bearing seat 0.030-in radius, others $\frac{1}{8}$-in radius, except right-hand bearing seat transition, $\frac{1}{4}$ in. The material is 1030 HR. Keyways $\frac{3}{8}$ in wide by $\frac{3}{16}$ in deep. Dimensions in inches.



Solution

Candidate critical locations for strength:

- Pinion seat keyway
- Right bearing shoulder
- Coupling keyway

Table A-20 for 1030 HR: $S_{ut} = 68$ kpsi, $S_y = 37.5$ kpsi, $H_B = 137$

Eq. (6-8): $S'_e = 0.5(68) = 34.0$ kpsi

Eq. (6-19): $k_a = 2.70(68)^{-0.265} = 0.883$

$$k_c = k_d = k_e = 1$$

b. Fluctuating
c. Completely reversed loading

21. Which surface finish yields the ~~smallest~~ (best) surface factor k_a?
    a. Ground
    b. Machined or cold-drawn
    c. Hot rolled
    d. As forged

closest to 1

22. Which of the following factors can be greater than 1?
    a. k_b
    b. k_c
    c. k_d
    d. k_e
    e. k_f
    f. K_f
23. The point of contact between two gear teeth is always
    a. On the pitch point
    b. On the pressure line
    c. On the pitch circle
    d. On the base circle
24. The equivalent radial load Fe combined loads
    a. Uses V=1.2 when the inner ring rotates
    b. Is necessary when only a thrust load is applied
    c. Is necessary when the shaft is not round
    d. Uses Y=0 for cylindrical roller bearings
25. The external load carried by the bolt if the joint does NOT separate is:
    a. (1-C)P
    b. C
    c. P_b
    d. Increased if the member stiffness decreases

Pinion seat keyway

See Table 7-1 for keyway stress concentration factors

$$\left.\begin{aligned} K_t &= 2.2 \\ K_{ts} &= 3.0 \end{aligned}\right\} \text{Profile keyway}$$

For an end-mill profile keyway cutter of 0.010 in radius.

From Fig. 6-20: $q = 0.50$

From Fig. 6-21: $q_s = 0.65$

Eq. (6-32):

$$K_{fs} = 1 + q_s(K_{ts} - 1)$$
$$= 1 + 0.65(3.0 - 1) = 2.3$$
$$K_f = 1 + 0.50(2.2 - 1) = 1.6$$

Eq. (6-20): $k_b = \left(\dfrac{1.875}{0.30}\right)^{-0.107} = 0.822$

Eq. (6-18): $S_e = 0.883(0.822)(34.0) = 24.7\,\text{kpsi}$

Eq. (7-11):

$$\frac{1}{n} = \frac{16}{\pi(1.875^3)}\left\{4\left[\frac{1.6(2178)}{24\,700}\right]^2 + 3\left[\frac{2.3(2500)}{37\,500}\right]^2\right\}^{1/2}$$

$= 0.3$ and thus $n = 3.33$.

$\sigma'_{max} = 11{,}342$ psi using $d = 1.875$" (see example 7.7 in the class note, depending on the q and q_s you used, σ'_{max} may be different. I used the same numbers as used in example 7.7)

For yield, $n = S_y / \sigma'_{max} = 37500 / 11342 = 3.31$ ok Eq 7-15

ME 208 W10 Exam #2 50 minutes Name:______________________________

Open Textbook; Closed Notes, Only one page crib sheet allowed (you need to turn in the crib sheet with your exam)

1. (10 pts)
For what type of fatigue loading is there no size effect (i.e. no need to correct the size effect)? (Select all that apply)

- (a) Bending
- (b) Torsion
- (c) Axial tension/compression
- (d) Combined loading
- (e) None of the above

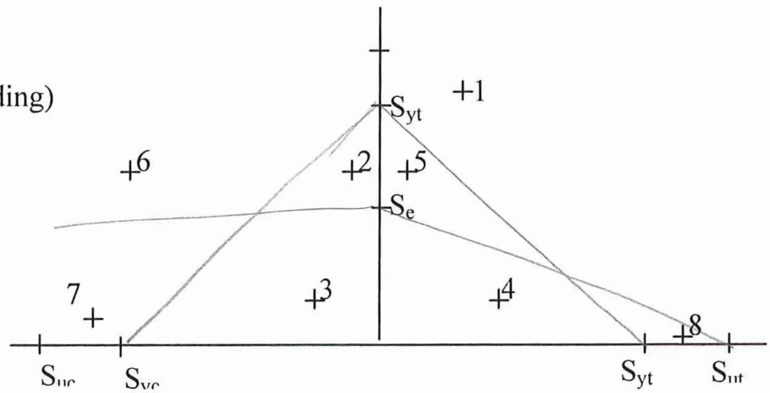
Solution:

(c)

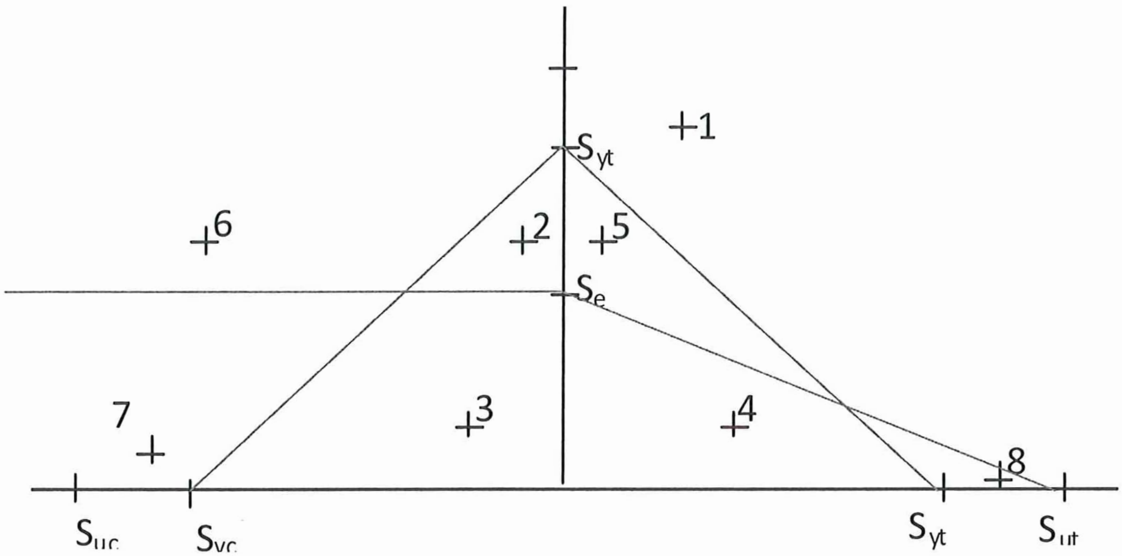
2. (10 pts)
On the σ_a vs. σ_m diagram, sketch the modified Goodman line and the yield line for both positive and negative mean stresses. For each fluctuating stress case plotted, indicate whether the predicted life is:

- (a) finite
- (b) infinite
- (c) less than one cycle (failed by yielding)

Case 1:	a	b	c	(circle one)
Case 2:	a	b	c	(circle one)
Case 3:	a	b	c	(circle one)
Case 4:	a	b	c	(circle one)
Case 5:	a	b	c	(circle one)
Case 6:	a	b	c	(circle one)
Case 7:	a	b	c	(circle one)
Case 8:	a	b	c	(circle one)

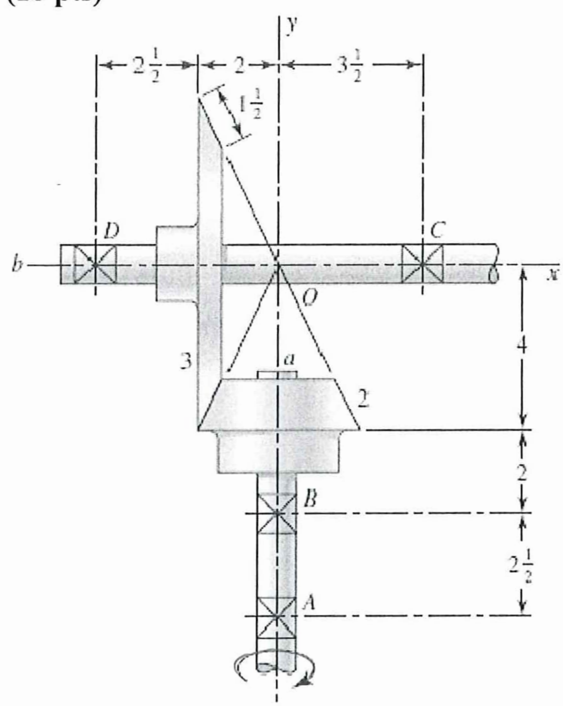


Solution:



Case 1: a b c (circle one)
Case 2: a b c (circle one)
Case 3: a b c (circle one)
Case 4: a b c (circle one)
Case 5: a b c (circle one)
Case 6: a b c (circle one)
Case 7: a b c (circle one)
Case 8: a b c (circle one)

3. (10 pts)



Pinion:

W_T $+z$

W_R $+x$

W $-y$

$\gamma = ? = \tan^{-1}\left(\frac{2}{4}\right) =$

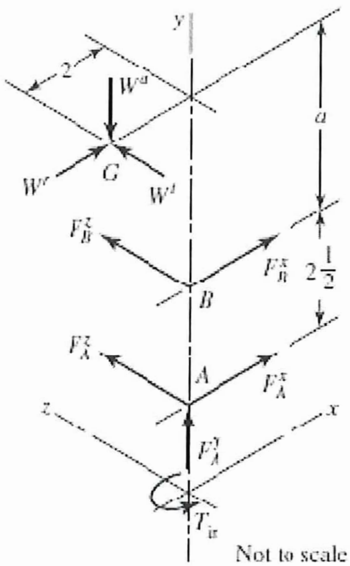
$\gamma = 26.57°$

The figure shows a 16T 20° straight bevel pinion driving a 32T gear, and the location of the bearing centerlines. Pinion shaft "a" receives 2.5 hp at 240 rev/min. To determine the bearing reactions at A and B if A takes both radial and thrust loads, which of the following is (are) true for the pinion? (circle all that apply)

a) γ is 26.6°
b) γ is 20°
c) W^t is in +Z direction
d) W^r is in +x direction
e) W^a is in +y direction

Solution:

$$T_{in} = 63\,025H/n = 63\,025(2.5)/240 = 656.5 \text{ lbf} \cdot \text{in}$$



$$W^t = T/r = 656.5/2 = 328.3 \text{ lbf}$$

$$\gamma = \tan^{-1}(2/4) = 26.565°$$

$$\Gamma = \tan^{-1}(4/2) = 63.435°$$

$$a = 2 + (1.5 \cos 26.565°)/2 = 2.67 \text{ in}$$

$$W^r = 328.3 \tan 20° \cos 26.565° = 106.9 \text{ lbf}$$

$$W^a = 328.3 \tan 20° \sin 26.565° = 53.4 \text{ lbf}$$

$$\mathbf{W} = 106.9\mathbf{i} - 53.4\mathbf{j} + 328.3\mathbf{k} \text{ lbf}$$

$$\mathbf{R}_{AG} = -2\mathbf{i} + 5.17\mathbf{j}, \quad \mathbf{R}_{AB} = 2.5\mathbf{j}$$

$$\sum \mathbf{M}_4 = \mathbf{R}_{AG} \times \mathbf{W} + \mathbf{R}_{AB} \times \mathbf{F}_B + \mathbf{T} = 0$$

a) γ is 26.6°
b) γ is 20°
c) W^t is in +Z direction
d) W^r is in +x direction
e) W^a is in +y direction

4. (10 pts)

To estimate the ideal endurance limit S_e', which of the following is (are) true? (circle all that apply)

34 $S_e' = \frac{S_{ut}}{2}$

(a) For ANSI 1020 CD steel, S_e' is greater than 30 kpsi.
(b) For ANSI 1080 HR steel, S_e' is greater than 50 kpsi. 56
~~(c)~~ For ANSI 1080 HR steel, S_e' is less than 55 kpsi.
~~(d)~~ For 2040 T3 aluminum, S_e' is greater than 10 kpsi. DNE
~~(e)~~ For ANSI 4340 steel heat treated to a tensile strength of 250 kpsi, S_e' is greater than 110 kpsi. ×

Solution:

- **S_{ut} = 68 kpsi, S_e = 0.5(68) = 34 kpsi**
- **S_{ut} = 112 kpsi, S_e = 0.5(112) = 56 kpsi**
- **2024T3 has no endurance limit**
- **S_e = 100 kpsi**

(a) For ANSI 1020 CD steel, S_e' is greater than 30 kpsi.
(b) For ANSI 1080 HR steel, S_e' is greater than 50 kpsi.
(c) For ANSI 1080 HR steel, S_e' is less than 55 kpsi.
(d) For 2040 T3 aluminum, S_e' is greater than 10 kpsi.
(e) For ANSI 4340 steel heat treated to a tensile strength of 250 kpsi, S_e' is greater than 110 kpsi.

5. (10 pts)

A 21-tooth spur pinion mates with a 28-tooth gear. The diametral pitch is 3 teeth/in and the pressure angle is 20°. Which of the followingis (are) true? Circle all that apply:

~~(a)~~ The addendum is greater than 0.4 in
(b) The dedendum is greater than 0.4 in
(c) The circular pitch is greater than 1 in
(d) The tooth thickness is greater than 0.5 in
(e) The base circle circular diameter of the pinion is less than 7 in

Solution:

$a = \frac{1}{P} = \frac{1}{3} = .33$ $\qquad b = \frac{1.25}{P} = \frac{1.25}{3} = .417$

$P = 3$

$p = \frac{\pi}{P} = 1.047$ $\qquad$ Circ. diam $= d = \frac{N}{P} = \frac{21}{3} - .417$

$$a = 1/P = 1/3 = 0.3333 \text{ in} \quad Ans.$$

$$b = 1.25/P = 1.25/3 = 0.4167 \text{ in} \quad Ans.$$

$$c = b - a = 0.0834 \text{ in} \quad Ans.$$

$$p = \pi/P = \pi/3 = 1.047 \text{ in} \quad Ans.$$

$$t = p/2 = 1.047/2 = 0.523 \text{ in} \quad Ans.$$

$$d_1 = N_1/P = 21/3 = 7 \text{ in}$$

$$d_{1b} = 7 \cos 20° = 6.578 \text{ in} \quad Ans.$$

$$d_2 = N_2/P = 28/3 = 9.333 \text{ in}$$

$$d_{2b} = 9.333 \cos 20° = 8.770 \text{ in} \quad Ans.$$

$$p_b = p_c \cos\phi = (\pi/3)\cos 20° = 0.984 \text{ in}$$

(a) The addendum is greater than 0.4 in
(b) The dedendum is greater than 0.4 in
(c) The circular pitch is greater than 1 in
(d) The tooth thickness is greater than 0.5 in
(e) The base circle circular diameter of the pinion is less than 7 in

6. (10 pts)
"A commercial enclosed gear drive consists of a 20° spur pinion having 16 teeth driving a 48-tooth gear. The pinion speed is 300 rev/min, the face width 2 in, and the diametral pitch is 6 teeth/in. The gears are grade 1 steel, through-hardened at 200 Brinell, made to No. 6 quality standards, uncrowned, and are to be accurately and rigidly mounted. Assume a pinion life of 10^8 cycles and a reliability of 0.90." Based on the above statements, which of the following is (are) true in calculating the stress and strength conditions of the gear drive? Circle all that apply:

not on Exam II

(a) $C_H = 1.06$
(b) $(Y_N)_{Pinion} = (Y_N)_{Gear}$
(c) $C_P > 2180\ \sqrt{psi}$
(d) $K_R = 0.85$

(e) $(J)_{Pinion} = (J)_{Gear}$

Solution:

(a) $C_H = 1.06$

(b) $(Y_N)_{Pinion} = (Y_N)_{Gear}$

(c) $C_P > 2180\ \sqrt{psi}$

(d) $K_R = 0.85$

(e) $(J)_{Pinion} = (J)_{Gear}$

7. (10 pts)

Which of the following is (are) true about fatigue? (Circle all that apply)

a) In general, $S_e' > S_e$

b) Fatigue failure is normally below S_{ut}

c) It is possible that $S_y < S_e$

d) Fatigue is initiated from microcracks due to cyclic plastic deformation

~~e)~~ Today predicting fatigue is mostly based on science rather than engineering

~~f)~~ In general, $S_f\ (N>10^5) = S_e$

~~g)~~ Stress life method is used to predict fatigue when $N > 10^6$ cycles

h) In fatigue analysis, size does not need to be corrected for S_e if under only axial loads

i) For non-rotating shaft, the correction factor is taken into account in size factor

j) If no correction is made for S_e in terms of reliability, it is assumed to be 50% reliable.

Solution:

a) In general, $S_e' > S_e$

b) Fatigue failure is normally below S_{ut}

c) It is possible that $S_y < S_e$

d) Fatigue is initiated from microcracks due to cyclic plastic deformation

e) Today predicting fatigue is mostly based on science rather than engineering

f) In general, $S_f\ (N > 10^5) = S_e$

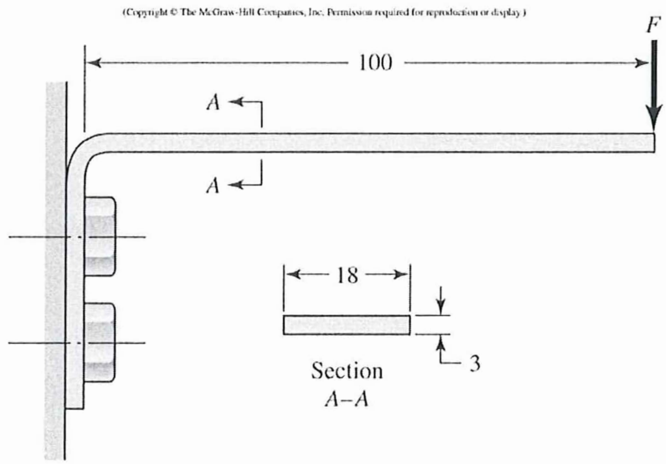
g) Stress life method is used to predict fatigue when $N > 10^6$ cycles

h) In fatigue analysis, size does not need to be corrected for S_e if under only axial loads

i) For non-rotating shaft, the correction factor is taken into account in size factor

j) If no correction is made for S_e in terms of reliability, it is assumed to be 50% reliable.

8. (30 pts)
The figure shown below is a drawing of a 3- by 18-mm latching spring. A preload is obtained during assembly by shimming under the bolts to obtain an estimated initial deflection of 2 mm. The latching operation itself requires an additional deflection of exactly 4 mm, thus a total deflection of 6 mm. The material is ground high-carbon steel, bent then hardened and tempered to a minimum hardness of 490 Bhn (Use equation 2-17 for relationship of hardness to strength). The radius of the bend is 3 mm.



Estimate the yield strength to be 90 percent of the ultimate strength. **<u>As the spring is likely to fail at the inner radius, the outer radius situations can be ignored</u>**. Also the maximum and minimum stresses at the inner radius have been calculated below:

$$(\sigma_i)_{\min} = -4.859(150.9) = -733.2 \text{ MPa}$$

$$(\sigma_i)_{\max} = -4.859(50.3) = -244.4 \text{ MPa}$$

(a) Find the fatigue factor of safety based on infinite life, using the modified Goodman criterion (consider only the inner radius).
(b) Find the static factor of safety (consider only the inner radius).

Solution:

$S_{ut} = 1666$

$n_s = \dfrac{S_{ut}}{\sigma_{max}} = \dfrac{490(3.4)(.9)}{733.2}$ $\quad n_y = \cancel{2.27}$ 2.045 $\quad S_y = 1500$ MPa

$n_f = \dfrac{S_e}{\sigma_a}$ $\quad \sigma_a = \dfrac{-733.2 - (-244.4)}{2} = \boxed{-244.4}$ $\quad \boxed{\sigma_M = -488.8 \text{ MPa}}$

$* S_e = k_{a, b, c, d, e, f}$

$S_e' = 700$ MPa

$k_a = 1.58(1666)^{-.085} = .841$

$k_b = \quad de = \quad 1.027$

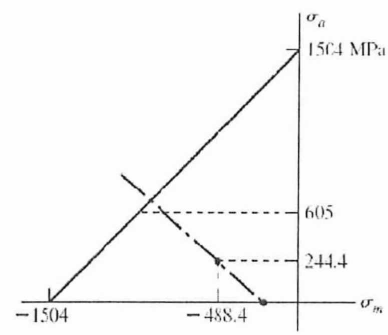
Eq. (2-17) $$S_{ut} = 3.41(490) = 1671 \text{ MPa}$$

Per the problem statement, estimate the yield as $S_y = 0.9S_{ut} = 0.9(1671) = 1504$ MPa. Then from Eq. (6-8), $S'_e = 700$ MPa; Eq. (6-19), $k_a = 1.58(1671)^{-0.085} = 0.841$; Eq. (6-25) $d_e = 0.808[18(3)]^{1/2} = 5.938$ mm; and Eq. (6-20), $k_b = (5.938/7.62)^{-0.107} = 1.027$.

$$S_e = 0.841(1.027)(700) = 605 \text{ MPa}$$

At Inner Radius $$(\sigma_i)_a = \left|\frac{-733.2 + 244.4}{2}\right| = 244.4 \text{ MPa}$$

$$(\sigma_i)_m = \frac{-733.2 - 244.4}{2} = -488.8 \text{ MPa}$$



Load line: $\sigma_m = -244.4 - \sigma_a$

Langer (yield) line: $\sigma_m = \sigma_a - 1504 = -244.4 - \sigma_a$

Intersection: $\sigma_a = 629.8$ MPa, $\sigma_m = -874.2$ MPa

(Note that σ_a is more than 605 MPa)

Yield: $$n_y = \frac{629.8}{244.4} = 2.58$$

Fatigue: $$n_f = \frac{605}{244.4} = 2.48$$ Thus, the spring is likely to fail in fatigue at the inner radius. *Ans.*

SP09 Exam 3

#4)

ϕ pressure angle $= 20°$

γ pitch angle $= 63.44$

$W^r = W_t \tan\phi \cos\gamma = 6.510$

$W^a = W_t \tan\phi \sin\gamma = 13.02$

$W = \sqrt{W_{,,,}^2} =$

$W = W_t \hat{k} + (-)W_r \hat{i} - W_a \hat{j}$

~~#5)~~ $\cancel{\sigma = W^t K_o K_v K_s \frac{P_d}{F} \frac{K_m K_B}{J}}$

~~#~~ HW6 #1)

$V = \frac{\pi d n}{12}$ $d = 4$ $n = 240$ rev/min $V = 251$ ft/min

$W_t = 33{,}000 \frac{H}{V} = \frac{(2.5 \text{ hp})}{}$ $\boxed{W_t = 329 \text{ lbf}}$ z

$r_B(2.5) + W_t(4.5) = 0$ -592

$\times$ -675.5

γ ~~θ~~ $= \tan^{-1}\left(\frac{2}{4}\right)$ $R_{G/B} \left(2 - \frac{1.5}{2} \cos\gamma\right)$

$\gamma = 26.57$

#8 endurance limit:

find Sut $\frac{1}{4}$" diam 450 brinell $S_u = .495(BHN)$ $= 223$ kpsi $S_e' = 100$ kpsi

$S_e = k_a k_b k_c k_d k_e k_f S_e'$

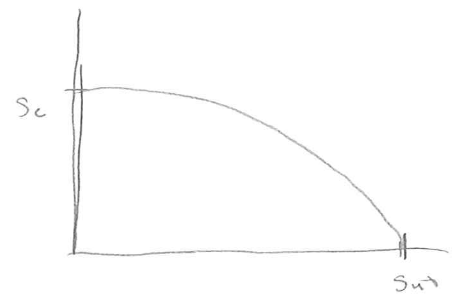
mistake here $k_a = a S_{ut}^{b} = 1.34 S_{ut}^{-.085}$ $k_a = .840$ mistake here

$k_b = .879 d_e^{-.107} =$ $d_e = .370 d$ $k_b = 1.134$ ✓

$k_c = 1$ $k_e = 1$

$k_d = 1$ $k_f = 1$ $S_e' = 100$ kpsi

$S_e = (.84)(1.134)(100\text{kpsi}) = \boxed{95.26 \text{ kpsi}}$



#9 axial fluctuating load $\boxed{\text{find } n_y, n_f}$ use gerber

AISI 1018 $S_{ut} = 64$ kpsi $S_y = 54$ kpsi

Axial 800 & -3000 lbf $m = \left(\frac{1}{A}\right)\frac{800 + (-3000)}{2} = -1100 \text{ lbf}\left(\frac{1}{A}\right) = -3911$ psi σ_m'

$A = \left(\frac{3}{8}\right)(1 - .25) = .2813 \text{ in}^2$ $\sigma_a = \frac{1}{A}\,\frac{800 - (-3000)}{2} = 6756$ psi σ_a'

use Gerber $S_e' = 0.5\, S_{ut} = 32$ kpsi

$k_a = 2.70(S_{ut})^{-.265} = .897$

$k_b = 1$ $k_c = 0.85$ $k_d = 1$ $k_e = 1$ $S_e = k_{a,b,c,d,e,\cancel{f}}\, S_e' = \boxed{24.40}$ not included in answer

$k_f \Rightarrow \frac{d}{w} = .25 \Rightarrow k_t = 2.41$ $[1 + q(k_t - 1)]$ q? $r = .125$ $q = .79$

$k_f = [1 + .79(2.41 - 1)]$ $k_f = 2.11$

$\sigma_m = k_f(\sigma_m') = 2.11(-3911) = -8.45$ kpsi

$\sigma_a = k_f(\sigma_a') = 2.11(6756) = +14.59$ kpsi

↑ * 2.16

n_f, Gerber $\Rightarrow \frac{n\sigma_a}{S_e} + \cancel{\left(\frac{n\sigma_m}{S_{ut}}\right)^2}^{\,0} = 1$ $n_f = \frac{S_e}{\sigma_a} = \frac{24.40}{14.59} = \boxed{1.672}$

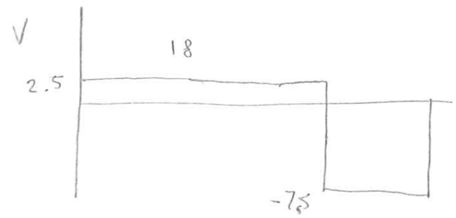
$n_y = \frac{S_y}{\sigma_{max}} = \frac{54}{10.67}$ $\boxed{n_y = 5.06}$

$\sigma_{max} = \frac{-3000}{A} = 10.67$ kpsi

7) exam SP 09

$d = 2.5 \text{ in}$ $R_1 = 2.5 \text{ kip}$ $R_2 = 7.5 \text{ kip}$

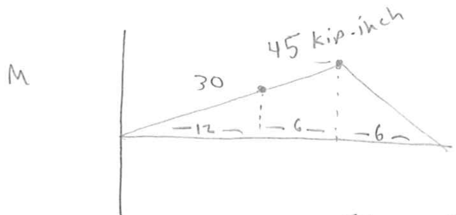
$N = 3450$ cycles



so, critical stress is at 12"

$30 \frac{y}{I}$ vs $45 \frac{y}{I}$ $\Rightarrow$ 30 vs 4.30

$= \frac{d/2}{\pi d^4/64} = \frac{32}{\pi d^3} \Rightarrow \frac{I}{c} = \frac{\pi d^3}{32}$



$S_e' = .5 S_{ut}$ $S_{ut} = 120 \text{ kpsi}$. $S_y = 66 \text{ kpsi}$

$S_e = k_{a,b,c,d,e,f} S_e'$ $k_a = 2.7(S_{ut})^{-.265} = .7592$

$k_b = 0.91 d^{-.157} = .7881$ $k_c = 1$ $k_d = 1$ $k_e = 1$ $K_f = 1.63$

$k_f = 1 + q(k_t - 1)$ where $r = \frac{d}{10} = .25"$ $\Rightarrow q = 0.9$ $\frac{r}{d} = \frac{1}{10}$ $\frac{D}{d} = 1.5$

$k_t = 1.7$

$S_e = 36.3 \text{ kpsi}$ find $S_f = aN^b$ $a = \frac{f(S_{ut})^2}{S_e} = \frac{.9(120)^2}{36.3} = 321.3$

$b = -\frac{1}{3}\ln\left(\frac{f S_{ut}}{S_e}\right) = -.1578$ $S_f = 88.85 \text{ kpsi}$

$\sigma_0 = \frac{My}{I} =$ $y = \frac{d}{2}$ $I = \frac{\pi d^4}{64}$ $\frac{y}{I} = \frac{d\,64}{2\,\pi d^4} = \frac{32}{\pi d^3}$

$\sigma_0 = 19.56 \text{ kpsi}$

$\sigma_a = K_f \sigma_0 = 31.88 \text{ kpsi}$

$n_f = \frac{S_f}{\sigma_a} = \frac{88.85}{31.88} = 2.79$

How to solve n_f & n_y in bending shaft, rotating

1) find critical location
2) find S_e, S_e = endurance limit in actual conditions kPa
3) find S_f. $S_f = aN^b$ fatigue strength based on # cycles pg. 285
4) find σ_a (for rotating, $\sigma_m = 0$)
5) plug S_f & σ_a into $n_f = \frac{S_f}{\sigma_a}$

ME 208B F09 **Exam #2** **65 minutes** **Name:** MASTER

Open Textbook; Closed Notes, Only one sheet of paper allowed

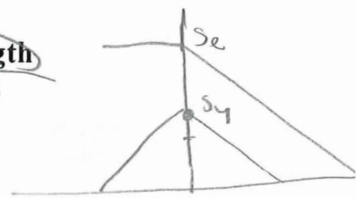
1. (5 pts)
What is the basis for using Se' = 0.5 S_{ut}? (Pick one)
(a) Maximum shear stress theory
(b) Based on the Stress-Life Method Stress life used if N > 1000
(c) Typical line through experimental data
(d) Tradition
(e) None of the above (explain) ______________________________

Solution:

(c)

2. (5 pts)
Which of the following is (are) true (Circle whichever is correct)
(a) Fatigue is always due to time varying loads
(b) The stress required to cause fatigue is always below the yield strength
(c) S-N diagram is applicable to alternating (completely reverse) loads
(d) Copper alloys have no endurance limit
(e) S_e' implies the fatigue strength of the real world specimen

this can happen



depending on material

Solution:

(a) (c) (d)

3. (5 pts)
A gear is to be chosen with a given pitch diameter d. The diametral pitch P and the pressure angle ϕ must be chosen. Indicate whether the following parameters will be affected by the choice of diametral pitch or pressure angle.(Circle P or ϕ or both)

(a) P ϕ) Addendum
(b) P ϕ) Number of teeth
(c) P ϕ) Base circle diameter
(d) P ϕ) Dedendum
(e) P ϕ) Bearing force

P = teeth per inch
ϕ = pressure angle

Solution:

(a) P ϕ) Addendum
(b) P ϕ) Number of teeth
(c) P ϕ) Base circle diameter
(d) P ϕ) Dedendum
(e) P ϕ) Bearing force

Pressure
Pressure angle changes bearing force

4. (5 pts)

The point of contact between two mating gear teeth is (choose one)

(a) always at the pitch point.

(b) always on the pressure line.

(c) always on the pitch circle.

(d) always on the base circle.

(e) will fail if the number of cycles > 10^8.

Solution:

(b)

5. (5 pts)

A rotating shaft is simply supported with ball bearings on both ends. The shaft is subjected to a critical bending moment and an axial compression load from both ends. We can conclude that this shaft (circle each that applies)

(a) × is under completely reverse loading condition.

(b) is under fluctuating stress situation.

(c) × has negative amplitude stress.

(d) has negative midrange stress. compression = negative

(e) none of the above.

Solution: ✓

(b), (d)

6. (5 pts)

Estimate S_e' for 2024 T3 aluminum. S_e' = endurance limit, rotating beam specimen

Solution:

Aluminum has no endurance limit

$S_{midrange} = \frac{50+(-10)}{2} = 20kpsi$

7. (20 Pts)

An AISI 1020 cold-drawn part is loaded with stresses σ_{max} = 50 kpsi, σ_{min} = -10 kpsi. The completely adjusted endurance limit is $\boxed{S_e = 30 \text{ kpsi.}}$ On the plot of σ_a vs σ_m, sketch the modified Goodman line and the yield line.

a) **Mark on the plot the fluctuating stress for the loaded part. (Label the key numbers on the plot.)**

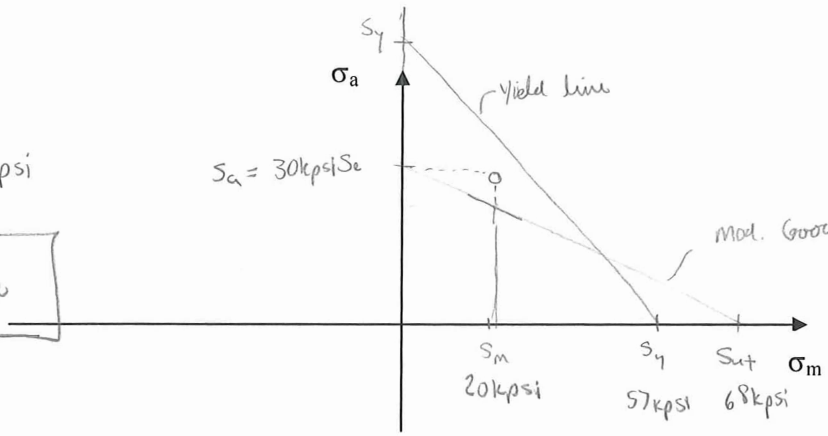
b) **From the sketch, determine whether the predicted life has an infinite life, finite life, or just fail in one cycle** (No calculation is necessary.)

$S_t = 68$ kpsi

$S_y = 57$ kpsi

$\sigma_a = \frac{50-(-10)}{2} = \frac{60}{2} = 30$ kpsi

b) $\boxed{\text{finite life}}$



Solution:

(a) Table A-20 for **AISI 1020 cold-drawn**

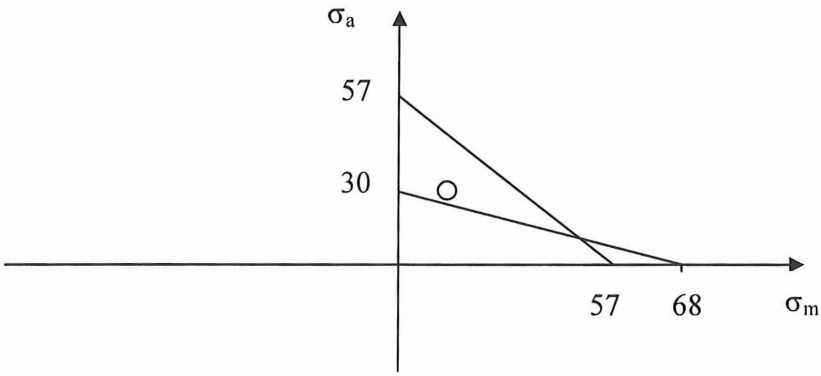
S_{ut} = 68 kpsi

S_y = 57 kpsi

σ_a = (50-(-10))/2= 30 kpsi

σ_m = (50-10)/2 = 20 kpsi

(b) Finite life



8. (20 pts)

A 1/4 -in round rod was heat-treated and ground. The measured hardness was found to be 450 Brinell. Estimate the endurance strength if the rod is used in <u>non-rotating</u> bending.

Solution:

$$S_{ut} = 0.495\,(450) = 223 > 200$$

$$S'_e = 100 \text{ kpsi}$$

$$a = 1.34, \quad b = -0.085$$

$$k_a = 1.34(242.6)^{-0.085} = 0.840$$

$d_e = 0.37\,d = 0.0925$

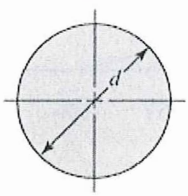
$k_b = (0.0925/0.3)^{-0.107} = 1.134$

$S_e = k_a\, k_b\, S_e' = (0.84)\,(1.134)\,(100) = 95.3$ kpsi

$$S'_e = \begin{cases} 0.5 S_{ut} & S_{ut} \le 200 \text{ kpsi (1400 MPa)} \\ 100 \text{ kpsi} & S_{ut} > 200 \text{ kpsi} \\ 700 \text{ MPa} & S_{ut} > 1400 \text{ MPa} \end{cases} \qquad (6\text{–}8)$$

Surface Finish	Factor a: S_{ut}, kpsi	Factor a: S_{ut}, MPa	Exponent b
Ground	1.34	1.58	−0.085

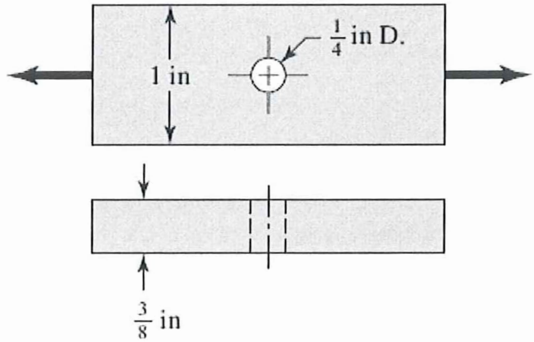
$$k_b = \begin{cases} (d/0.3)^{-0.107} = 0.879d^{-0.107} & 0.11 \le d \le 2 \text{ in} \\ 0.91d^{-0.157} & 2 < d \le 10 \text{ in} \end{cases} \qquad (6\text{–}20)$$



$$A_{0.95\sigma} = 0.01046d^2$$
$$d_e = 0.370d$$

9. (30 pts)

The cold-drawn AISI 1018 steel bar shown in the figure is subjected to an axial load fluctuating between 800 and -3000 lbf. Estimate the factors of safety n_y and n_f using Gerber fatigue failure criterion as part of the designer's fatigue diagram.



Solution:

Table A-20: $S_{ut} = 64$ kpsi, $S_y = 54$ kpsi

$$A = 0.375(1 - 0.25) = 0.2813 \text{ in}^2$$

$$\sigma_{\max} = \frac{F_{\max}}{A} = \frac{3000}{0.2813}(10^{-3}) = 10.67 \text{ kpsi}$$

$$n_y = \frac{54}{10.67} = 5.06 \quad \textit{Ans.}$$

$$S'_e = 0.5(64) = 32 \text{ kpsi}$$

$$k_a = 2.70(64)^{-0.265} = 0.897$$

$$k_b = 1, \qquad k_c = 0.85$$

$$S_e = 0.897(1)(0.85)(32) = \boxed{24.4 \text{ kpsi}}$$

Table A-15-1: $w = 1$ in, $d = 1/4$ in, $d/w = 0.25 \ \therefore K_t = 2.45$.

Fig. 6-20, with $r = 0.125$ in, $q \doteq 0.8$

Eq. (6-32): $K_f = 1 + 0.8(2.45 - 1) = 2.16$

$$\sigma_a = 2.16\left|\frac{0.800 - (-3.000)}{2(0.2813)}\right| = 14.59 \text{ kpsi}$$

$$\sigma_m = 2.16\left[\frac{0.800 + (-3.000)}{2(0.2813)}\right] = -8.45 \text{ kpsi}$$

→ We have a compressive midrange stress for which the failure locus is horizontal at the S_e level.

$$n_f = \frac{S_e}{\sigma_a} = \frac{24.4}{14.59} = 1.67 \quad \textit{Ans.}$$

ME 208B SP09 Exam #2 45 minutes Name: MASTER

Open Textbook; Closed Notes, Only one sheet of paper allowed

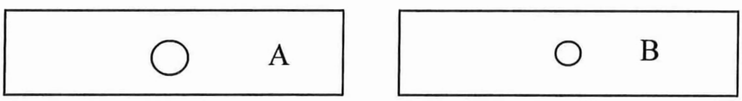
1. (5 pts)

Stress concentration can generally be neglected for: (pick one)
(a) all brittle materials
(b) all ductile materials
(c) all static loads
(d) all dynamic loads
(e) ductile materials with static loads
(f) ductile materials with dynamic loads
(g) brittle materials with static loads
(h) brittle materials with dynamic loads

Solution:

(e)

2. (15 pts)



Two plates are identical except for the size of the hole.
(a) If stress concentration is ignored, which part has the higher nominal stress?
Circle one: A B Can't tell

(b) If stress concentration is considered, which part has the higher stress concentration?
Circle one: A B Can't tell

(c) If stress concentration is considered, which part has the higher peak stress?
Circle one: A B Can't tell

Solution:
(a) A
(b) B
(c) C

3. (5 pts)
What is the basis for using Se' = 0.5 Sut? (Pick one)
(a) Maximum shear stress theory
(b) Typical line through experimental data
(c) Based on the Stress-Life Method
(d) Tradition
(e) None of the above (explain) ________________________________

Solution:

(b)

4. (10 pts)
A rotating round shaft with a 2 inch diameter is loaded in completely reversed bending. Determinate the value for the size factor, k_b.

Solution:

de = 0.37 d = 0.74

$$k_b = (0.74/0.3)^{-0.1133} = 0.903$$

OR

$$k_b = (2/0.3)^{-0.1133} = 0.807$$

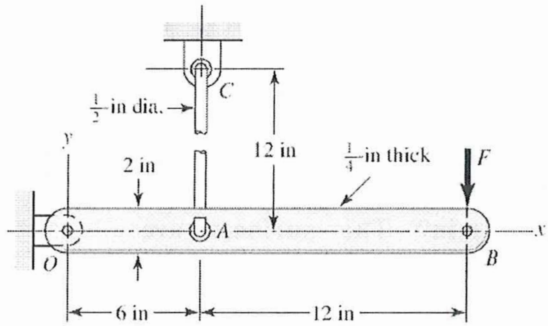
5. (5 pts)
Which of the following is (are) true (Circle whichever is correct)
(a) Fatigue is always due to time varying loads
(b) The stress required to cause fatigue is normally below the ultimate strength
(c) The method we discussed in class in mostly based on the linear-elastic fracture mechanics method
(d) S-N diagram is applicable only to alternating (completely reverse) loads.
(e) Aluminum and copper alloys have no endurance limit

Solution:

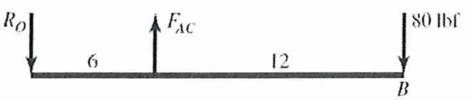
(a) (b) (d) (e)

6. (30 pts)

The rectangular member OAB, shown in the figure, is held horizontal by the round hooked bar AC. The modulus of elasticity of both parts is 10 Mpsi. Use Castigliano's theorem (Energy Method) only to find the deflection at B due to a force F = 100 lbf.



Solution:



$$6F_{AC} = 18(100)$$

$$F_{AC} = 300 \text{ lbf}$$

$$R_O = 200 \text{ lbf}$$

$$I = \frac{1}{12}(0.25)(2^3) = 0.1667 \text{ in}^4$$

- **Note: $F_{AC} = 3F$; $R_O = 2F$**

AB:

$$\delta_{AB} = \frac{\partial U}{\partial F} = \frac{1}{EI}\int_0^{12} M\left(\frac{\partial M}{\partial F}\right)dx = \frac{1}{EI}\int_0^{12}(Fx)(x)dx = \frac{F(12^3)}{3EI}$$

AO:

$$\delta_{AO} = \frac{\partial U}{\partial F} = \frac{1}{EI}\int_0^{6} M\left(\frac{\partial M}{\partial F}\right)dx = \frac{1}{EI}\int_0^{6}(2Fx)(2x)dx = \frac{4F(6^3)}{3EI}$$

AC:

$$\delta_{AC} = \frac{\partial U}{\partial F} = \frac{1}{EA}\int_0^{12} F_{AC}\left(\frac{\partial F_{AC}}{\partial F}\right)dx = \frac{1}{EA}\int_0^{12}(3F)(3)dx = \frac{9F(12)}{EA}$$

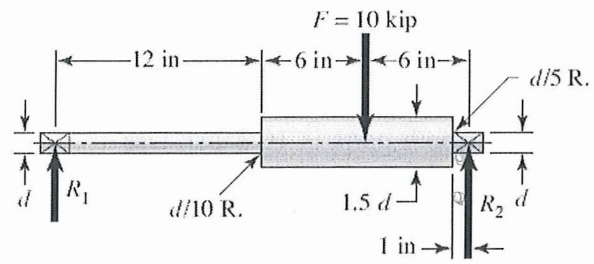
Deflection at B

$$\delta = \frac{F(12^3)}{3EI} + \frac{4F(6^3)}{3EI} + \frac{9F(12)}{EA} = \frac{F(1728+864)}{3EI} + \frac{F(108)}{EA}$$

$$= \frac{(100)(2592)}{3(10^7)(0.1667)} + \frac{(100)(108)}{(10^7)(\pi(0.5)^2/4)} = 0.0518 + 0.0055 = 0.0573\ in$$

7. (30 pts)

Bearing reactions R_1 and R_2 are exerted on the shaft shown in the figure, which rotates at 1150 rev/min and supports a 10-kip bending force. Use a 1095 HR steel. With a diameter d = 2.5 in, what is the design factor for a life of 3 min? The surfaces are machined.



Solution:

A priori design decisions:

The design decision will be: Design factor $n_f = ?$

Material and condition: 1095 HR and from Table A-20 S_{ut} = 120

Life: (1150)(3) = 3450 cycles

Function: carry 10 000 lbf load

Preliminaries to iterative solution:

$$S'_e = 0.5(120) = 60 \text{ kpsi}$$

$$k_a = 2.70(120)^{-0.265} = 0.759$$

$\frac{Y}{I}$

$$\frac{I}{c} = \frac{\pi d^3}{32} = 0.09817d^3$$

$$M(\text{crit.}) = \left(\frac{6}{24}\right)(10\,000)(12) = 30\,000 \text{ lbf} \cdot \text{in}$$

I/C = 1.5339

The critical location is in the middle of the shaft at the shoulder. From Fig. A-15-9: D/d = 1.5, r/d =0.10, and K_t = 1.68. With no direct information concerning f, use f = 0.9.

k_b = $(2.50/0.30)^{-0.107} = 0.797$

S_e = 0.759(0.797)(60) = 36.3 kpsi

a = $[0.9(120)]^2/36.3$ = 321.3

b = − 1/3 log [0.9(120)/36.3] = -0.15784

S_f = $321.3(3450)^{-0.15784}$ = 88.8 kpsi

σ_0 = Mc/I = 30 /(1.5339) = 19.6

$r = d/10 = 2.5$

Fig. 6-20: $q = 0.88$

Eq. (6-32): $K_f \doteq 1 + 0.88(1.68 - 1) = 1.59$

$\sigma_a = K_f \sigma_0 = 1.59(19.6) = 31.16$ kpsi

$n_f = S_f / \sigma_a = 88.8/31.16 = 2.85$

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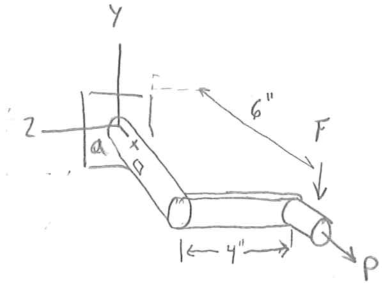
#6) a) Location of critical stress on AB?



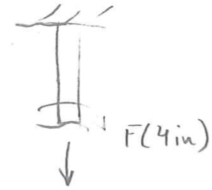
$\Sigma M_{az} = -F(6in) = -900$ lb·in

$\Sigma M_{ay} = -P(4in) = -800$ lb·in

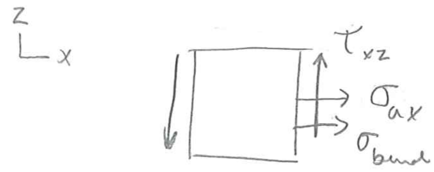
$\Sigma F_x = P = 200$

$\Sigma F_y = 0$

$\Sigma F_z = F = 150$

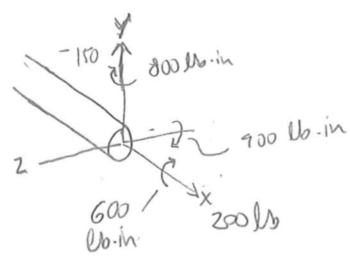


Total moment $= \sqrt{900^2 + 800^2} = 1204.2$ in·lb

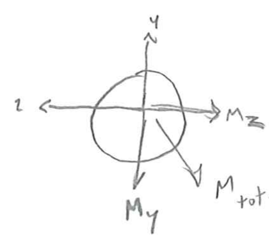


$$\sigma_{ax} = \frac{F_x}{A} = \frac{-200 \text{ lb}}{\frac{\pi}{4}(.75in)^2} = 452.7 psi$$

$$\sigma_{bend} = \frac{M_{total}\, r}{I} = \frac{1204.2(\frac{3}{8}")}{\pi(.75)^4/64} = 29074$$



$$\sigma_x = \sigma_{ax} + \sigma_{bend} = 452.7 + 29074 = 29526 \text{ psi}$$



- How do I know which quadrant? (plug in moments to figure at right)

$$\tau_{xz} = \frac{Tr}{J} = \frac{150(4)(.75/2)}{\pi(.75)^4/32} = 7243 \text{ psi}$$

$$\sigma' = (\sigma_x^2 + 3\tau_{xy}^2)^{1/2} = [29526 + 3(7243)^2]^{1/2} = 32081$$

$S_y = 32$ kpsi

$$n = \frac{S_y}{\sigma'} = \frac{32}{32.08} = \boxed{.998}$$

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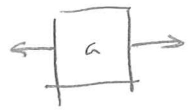
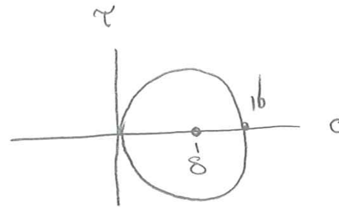
#3) a)

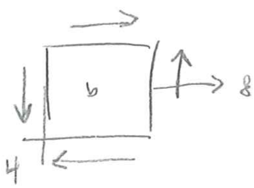




$R = \sqrt{\left(\frac{0+16}{2}\right)^2 + 0^2}$

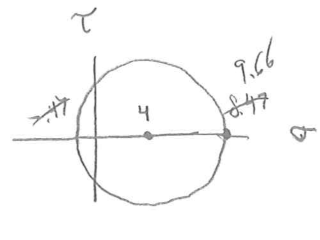
$= 8$

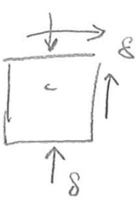
$C = \frac{\sigma_x + \sigma_y}{2} = 8$



$R = \sqrt{16 + 16} = 5.66$: 4.17

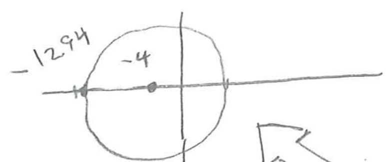
$C = 4$



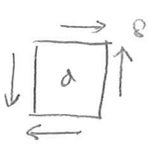


$R = \sqrt{16 + 64} = 8.94$

$C = \frac{(-8+0)}{2} = -4$

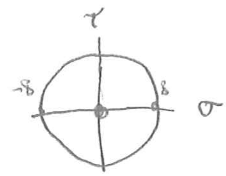


most critical element, greatest R



$R = \sqrt{64} = 8$

$C = \frac{0+0}{2}$



#4) Brinell Hardness = 270 kpsi
estimate S_u?

$S_u = .5 H_B$ kpsi $=$ $\boxed{135 \text{ kpsi}}$

#5) if $S_f > 5\%$, material is ductile

#2) a) ~~Dist. Energy~~ D.E b) MSS c) DE

d) MSS e) DE f) DE why?

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#7)

$M_{max} = 1971$ in lb $\quad n = 3 \quad S_{ut} = 20$ kpsi
$S_{uc} = 100$ kpsi

$d = ?$

cast iron

a) brittle coulomb-mohr theory

$\sigma_A = \frac{S_{ut}}{n} = = \frac{20}{3}$ kpsi

$\sigma_B = -\frac{S_{uc}}{n} = -\frac{100}{3}$ kpsi

$\sigma' =$

$\sigma_{Bend} = \frac{My}{I} = \frac{M d}{I\,2} = \frac{M d\, 64}{\pi d^4\, 2} =$ $\quad I = \frac{\pi d^4}{64}$

$= \frac{M\,32}{\pi d^3}$

$\sigma_{max} = \frac{20(10^3)\text{ psi}}{d^3}$

$\tau = \frac{Tc}{J} \quad c = \frac{d}{2} \quad J = \frac{\pi d^4}{32} \quad \tau = \frac{T d\, 32}{2\pi d^4} = \frac{16T}{\pi d^3}$

$\tau = \frac{5.1(10^3)}{d^3}$

$\sigma_{max} = \sigma_x + \sigma_y = 20 \cdot 10^3$ psi

$C = \frac{20(10^3)}{2d^3}$ psi $\quad C = \frac{10(10^3)}{d^3}$ psi

$R = \sqrt{\left(\frac{10(10^3)}{d^3}\right)^2 + \left(\frac{5.1(10^3)}{d^3}\right)^2}$

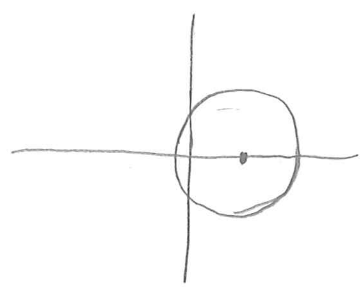
$R = \frac{11.2(10^3)}{d^3}$ $\qquad \sigma_1 = C + R \quad \sigma_3 = C - R$

$T =$

$250(4'')$
$- 333(3)$ $= 0$

$= 1000$ lb·in

$R = \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$

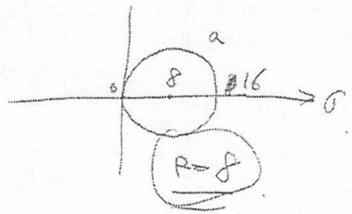


$$\sigma_{avg} = \frac{\sigma_x + \sigma_y}{2}$$

$$R = \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

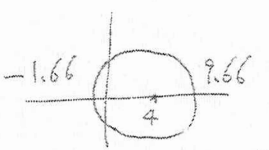
(a) $\sigma_x = 16 \qquad \tau_{xy} = 0$

$\sigma_{av} = 8 \qquad R = 8$



(b) $\sigma_x = 8 \qquad \tau_{xy} = 4$

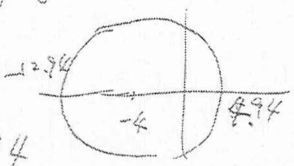
$\sigma_{av} = 4 \qquad R = \sqrt{4^2 + 4^2} = 4\sqrt{2} = 5.66$



(c) $\sigma_x = 0 \quad \sigma_y = -8 \quad \tau_{xy} = 8$

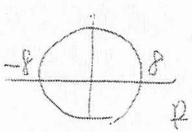
$\sigma_{av} = -\frac{8}{2} = -4$

$R = \sqrt{4^2 + 8^2} = 8.94$



(d) $\tau_{xy} = 8$

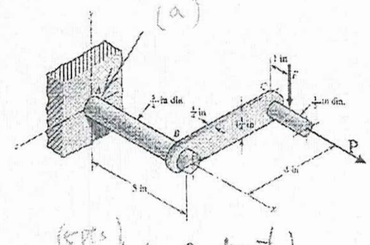
$R = \sqrt{8^2} = 8$



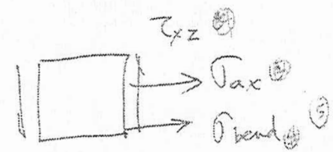
(B) C. Since it has the largest circle (or τ_{max})

6. 5. (30 pts)

The figure shows a crank loaded by a force $F = 150$ lbf and P = 200lbf. The material of the shaft AB is hot-rolled AISI 1018 steel.



(a) Identify the approximate location of the critical stress element on the main rod AB. If it is not on the top, bottom, or side of the rod, specify which quadrant (Use an arrow to show it in the figure). (5pts) 1st quadrant)
(b) Draw the critical stress element, including the stresses acting on it. (15 pts)
(c) Determine the factor of safety of this part (rod AB), based on the maximum-shear-stress theory. (10pts)



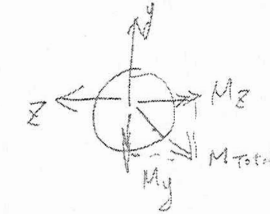
$$\sigma_{ax} = \frac{200}{\pi d^2/4} = \frac{200}{\pi(0.75^2)/4}$$

$$= 452.7 \text{ psi}$$

(b)

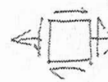
$M_z = 150(6) = -900$ in-lb

$M_y = 200(4) = -800$ in-lb



$M_{Total} = \sqrt{M_y^2 + M_z^2}$

$= \sqrt{900^2 + 800^2} = 1204.2$ in-lb

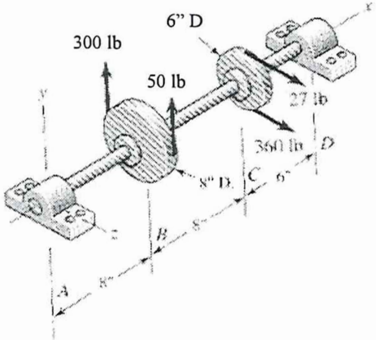


$$\sigma_{bend} = \frac{M_{total}\, r}{I} = \frac{1204.2\,(0.75/2)}{\pi(0.75^4)/64} = \frac{1204.2}{\pi(0.75^3)/32}$$

$= 29073.6$ psi

$\sigma_x = \sigma_{ax} + \sigma_{bend} = 452.7 + 29073.6 = 29526.3$

7. (24pts)
Given: Shaft of a certain cast iron subject to loading shown. We also found S_{ut} = 20 kpsi, S_{uc} = 100 kpsi. We also found that the maximum bending moment 1,971 in-lb occurs at Point B. For a factor of safety of n = 3, what should the diameter of the shaft (d) be (recommend the next largest standard dimension (8th's))?
(a) Use the proper Coulomb-Mohr Theory
(b) Modified Mohr Theory



Solution:

Torque on shaft at B
$T_B = (300-50)(4) = 1000$ in·lb
Torque on shaft at C
$T_C = (360-27)(3) = 1000$ in·lb

$$\boxed{\sigma = \frac{My}{I};\quad y = \frac{d}{2};\quad I = \frac{\pi d^4}{64}}$$

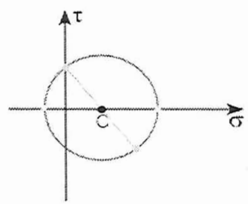
$\sigma_{max} = (20x10^3)/d^3$

σ_{max}

$$\boxed{\tau = \frac{Tc}{J};\quad c = \frac{d}{2};\quad J = \frac{\pi d^4}{32}}$$

$\tau = (5.1x10^3)/d^3$

Center $C = (\sigma_x + \sigma_y)/2$



Now construct Mohr's circle
C at $(10 \times 10^3)/d^3$
$R = (11.2 \times 10^3)/d^3$
$\sigma_1 = (21.2 \times 10^3)/d^3$
$\sigma_3 = (-1.2 \times 10^3)/d^3$

Use Coulomb-Mohr theory for brittle failure

$$\frac{\sigma_1}{S_{ut}} - \frac{\sigma_3}{S_{uc}} = \frac{1}{\eta}$$

$21.2/20d^3 + 1.2/100d^3 = 1/3$

d = 1.476" → 1.5"

MM Theory:

$|\sigma_B/\sigma_A| = 1.2/21.2 < 1$

$n = S_{ut}/\sigma_A = 20/(21.2)/d^3 = 3$

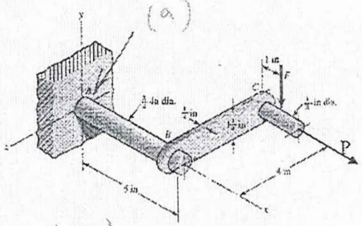
d = 1.47 → 1.5"

$$\boxed{\begin{array}{ll} \sigma_A = \dfrac{S_{ut}}{n} & \sigma_A \ge \sigma_B \ge 0 \\ & \sigma_A \ge 0 \ge \sigma_B \quad \text{and} \quad \left|\dfrac{\sigma_B}{\sigma_A}\right| \le 1 \\ \dfrac{(S_{uc} - S_{ut})\sigma_A}{S_{uc} S_{ut}} - \dfrac{\sigma_B}{S_{uc}} = \dfrac{1}{n} & \sigma_A \ge 0 \ge \sigma_B \quad \text{and} \quad \left|\dfrac{\sigma_B}{\sigma_A}\right| > 1 \\ \sigma_B = -\dfrac{S_{uc}}{n} & 0 \ge \sigma_A \ge \sigma_B \end{array}}$$

6. 5. (30 pts)

The figure shows a crank loaded by a force $F = 150$ lbf and $P = 200$lbf. The material of the shaft AB is hot-rolled AISI 1018 steel.

(a)

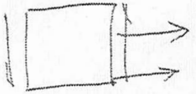


(a) Identify the approximate location of the critical stress element on the main rod AB. If it is not on the top, bottom, or side of the rod, specify which quadrant (Use an arrow to show it in the figure). (5 pts) 1st quadrant)

(b) Draw the critical stress element, including the stresses acting on it. (15 pts)

(c) Determine the factor of safety of this part (rod AB), based on the maximum-shear-stress theory. (10 pts)

τ_{xz}



σ_{ax}

σ_{bend} (5)

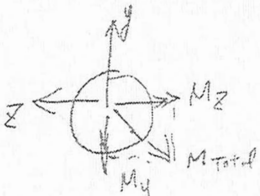
$$\sigma_{ax} = \frac{200}{\pi d^2/4} = \frac{200}{\pi(0.75^2)/4}$$

$$= 452.7 \text{ psi.}$$

(b)

$M_z = 150(6) = -900$ in-lb.

$M_y = 200(4) = -800$ in-lb



$M_{Total} = \sqrt{M_y^2 + M_z^2}$

7243.3

29526.3

$$= \sqrt{900^2 + 800^2} = 1204.2 \text{ in-lb}$$

$$\sigma_{bend} = \frac{M_{total}\, r}{I} = \frac{1204.2\,(0.75/2)}{\pi(0.75^4)/64} = \frac{1204.2}{\pi(0.75^3)/32}$$

$$= 29073.6 \text{ psi}$$

$\sigma_x = \sigma_{ax} + \sigma_{bend} = 452.7 + 29073.6 = 29526.3$

6. Continue.

$$\tau_{xz} = \frac{Tr}{J} = \frac{150(4)(0.75/2)}{\pi(0.75)^4/32} = 7243.3 \text{ psi.}$$

(c)

$$\sigma' = (\sigma_x^2 + 3\tau_{xy}^2)^{1/2} = [29526.3^2 + 3(7243.3)^2]^{1/2}$$

$$= 32081$$

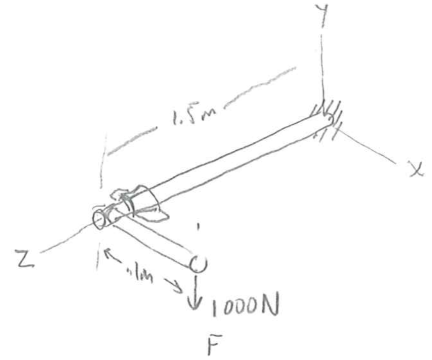
$$n = \frac{S_y}{\sigma'} = \frac{32}{32.08} = 0.998$$

F

Exam 1 F509

6) $d = 12\,mm$

find deflection @ pt. b



Castigliano's thm $\quad \delta_i = \frac{\partial U}{\partial F_i} \quad \theta_i = \frac{\partial U}{\partial M_i}$

$$\theta = \frac{Tl}{GJ} = \frac{\overbrace{1000(.1)\,N\cdot m}^{T}\ \overbrace{(1.5\,m)}^{l}}{\underbrace{207.0\,GPa}_{G}\ \underbrace{\underbrace{\left(\frac{\pi\,(.012)^4}{64}\right)}_{I}(2)}_{J}} = .35596$$

$3.560\ E6\ ($

$$\frac{Fl}{AE} = \frac{F\,(0.1)}{\pi\left(\frac{.012}{2}\right)^2\ 207\,GPa} = 4.271\ E\text{-}6$$

$\delta = \delta_{i\ bending} + \theta_i$

$\delta_B = \quad F$ ———

$T = 0.1\,m\,(F) \quad \frac{\partial T}{\partial F} = 0.1 \qquad$ Torsion $\ U = \frac{T^2 l}{2GJ}$

$M = -F\bar{x} \quad \frac{\partial M}{\partial F} = -\bar{x} \qquad$ Bending: $\ U = \frac{M^2 l}{2EI}$

$$U = \int \frac{M^2}{2EI}\,dx + \int \frac{T^2 L}{2GJ}\,dx = \frac{1}{2EI}\int M^2 dx + \frac{T^2 L}{2GJ}$$

$$\delta_B = \frac{\partial U}{\partial F} = \frac{1}{2EI}\int 2M\cdot\frac{\partial M}{\partial F}\,dx + \frac{2\,T(\partial T/\partial F)L}{2\,GJ}$$

$$\frac{\partial U}{\partial F} = \frac{\partial}{\partial F}M^2 = \frac{\partial}{\partial F}(M\cdot M) = \frac{\partial M}{\partial F}\cdot M + \frac{\partial M}{\partial F}\cdot M = 2M\frac{\partial M}{\partial F}$$

$$\delta_B = \frac{1}{EI}\int_0^{.1}(-F\bar{x})(-\bar{x})\,dx + \frac{(0.1)F(1.5)(0.1)}{GJ} = \frac{F}{EI}\int_0^{.1} x^2\,dx + \frac{.015\,F}{GJ}$$

$$\delta_B = \frac{x^3}{3}\Big]_0^{.1}\frac{F}{EI} + \frac{.015\,F}{GJ} = \frac{(0.1)^3 F}{3EI} + \frac{.015\,F}{GJ}$$

$$\delta_B = 9.45(10^{-5})\,F = \boxed{.0945\,m}$$

Exam 1 FS09

1) 3D Mohr's circles

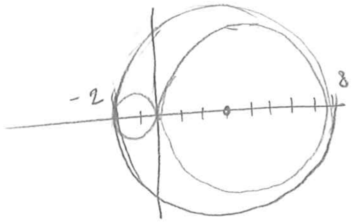
$\sigma_x = 6 \quad \tau_{xy} = -4$

$C = \frac{\sigma_x + \sigma_y}{2} = 3$

$R = \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$

$= \sqrt{3^2 + (-4)^2}$

$= \sqrt{25} = 5$



2) a) DE
b) MSS
c) BCM
d) DE ? (for Ductile mat'ls generally more accurate.)
e) DE
f) ~~MSS ?~~ (same theory, can be used for ductile & brittle mat'l)

$\boxed{\text{DCM \& BCM}}$ ✓

3) Stress concentration can generally be neglected for : ductile, static loads ✓
~~e) brittle mat'l w/ static loads~~ ✗
h) none of the above ✓

4) k_t depends on geometry, type of load ✓
~~Material~~

5) a) A has higher nominal stress
b) B has greater S.C.
c) Cannot tell greatest overall stress
peak

6) ON BACK

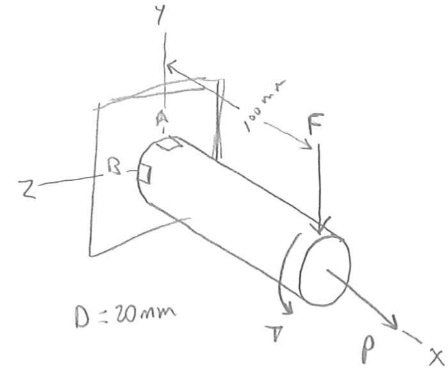
Exam 1 FS09

#7

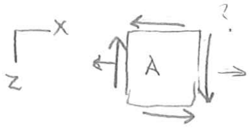
D.E. theory, find n @ elements A & B

F = .55 kN   P = 8 kN   T = 30 N·m

consider shear V



A) $\sigma_x =$



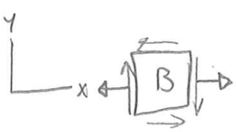
$$\sigma_x = \frac{P}{A} + \frac{My}{I} = \frac{8kN(4)}{\pi(.02\,m)^2} + \frac{.55kN(.1m)\left(\frac{.02}{2}m\right)}{\frac{\pi\,(.02)^4}{64}}$$

$$= 95.49 \times 10^6\ Pa = \boxed{95.49\ MPa}$$

$$\tau_{xy} = \frac{Tc}{J} = T\,\frac{.02\,m}{2}\left(\frac{32}{\pi(.02)^4}\right) = \frac{T\,16}{\pi(.02)^3} = 1.910\text{E}7 = \boxed{19.10\ MPa}$$

$$\sigma' = (\sigma_x^2 + 3\tau_{xy}^2)^{1/2} = 101.1\ MPa$$

$$n = \frac{S_y}{\sigma'} = \frac{280\,MPa}{101.1\,MPa} \quad \boxed{n_A = 2.770}$$



$$\sigma_x = \frac{P}{A} = 25.47\ MPa$$

V_y goes into $\tau_{xy} = \frac{VQ}{Ib}$ $b = d$

$$Q = \int_y^c y\,dA = \frac{\pi r^2}{2} = \frac{\pi d^2}{8}$$

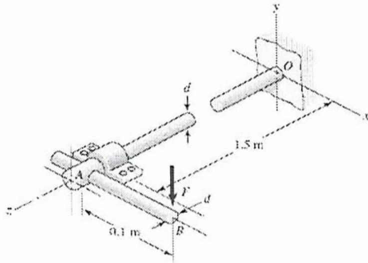
$$I = \frac{\pi d^4}{64}$$

$$= V\,\frac{\pi d^2}{\cancel{8}}\;\frac{\cancel{64}^{\,8}}{\pi d^4}\;\frac{1}{d}$$

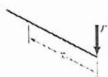
$$= V\,\frac{\pi 8}{d^3} = \frac{.55(10^3)\,\pi 8}{(.02)^3} =$$

$\frac{Tc}{J}$ goes into τ_{xy}, $= 19.10\ MPa$

6. (24pts)
The right figure illustrates a torsion-bar spring OA having a diameter d = 12 mm. The actuating cantilever AB also has d = 12 mm. Both parts are of carbon steel. Use Castigliano's theorem to find the deflection at point B corresponding to a force F = 1000N acting at B.



Solution:



$$T = 0.1F \qquad \frac{\partial T}{\partial F} = 0.1$$

$$M = -F\bar{x} \qquad \frac{\partial M}{\partial F} = -\bar{x}$$

$$\boxed{U = \int \frac{M^2}{2EI}dx}$$

$$U = \frac{1}{2EI}\int M^2\,dx + \frac{T^2 L}{2JG}$$

$$\boxed{U = \frac{T^2 l}{2GJ}}$$

$$\delta_B = \frac{\partial U}{\partial F} = \frac{1}{EI}\int M\frac{\partial M}{\partial F}\,dx + \frac{T(\partial T/\partial F)L}{JG}$$

$$= \frac{1}{EI}\int_0^{0.1} -F\bar{x}(-\bar{x})\,d\bar{x} + \frac{0.1F(0.1)(1.5)}{JG}$$

$$= \frac{F}{3EI}(0.1^3) + \frac{0.015F}{JG}$$

$\bar{x} = .1$

Where

$$I = \frac{\pi}{64}(0.012)^4 = 1.0179(10^{-9})\ \text{m}^4$$

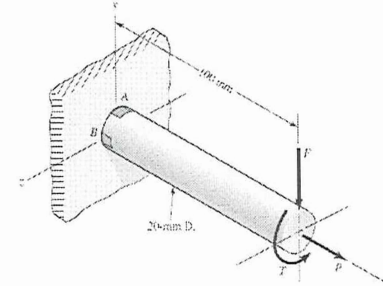
$$J = 2I = 2.0358(10^{-9})\ \text{m}^4$$

$$\delta_B = F\left[\frac{0.001}{3(207)(10^9)(1.0179)(10^{-9})} + \frac{0.015}{2.0358(10^{-9})(79.3)(10^9)}\right]$$

$$= 9.45(10^{-5})F$$

= 0.0945m

7. (27 pts)
Compute factors of safety, based upon the distortion-energy theory, for stress elements at *A* and *B* of the member shown in the figure. This bar is made of AISI 1006 cold-drawn steel and is loaded by the forces *F* = 0.55 kN, *P* = 8.0 kN, and *T* = 30 N · m. Need to consider shear V.



Solution:

A:

$$\sigma_x = \frac{32Fl}{\pi d^3} + \frac{4P}{\pi d^2} = \frac{32(0.55)(10^3)(0.1)}{\pi(0.020^3)} + \frac{4(8)(10^3)}{\pi(0.020^2)}$$

$$= 95.49(10^6)\ \text{Pa} = 95.49\ \text{MPa}$$

$$\tau_{xy} = \frac{16T}{\pi d^3} = \frac{16(30)}{\pi(0.020^3)} = 19.10(10^6)\ \text{Pa} = 19.10\ \text{MPa}$$

$$\sigma' = (\sigma_x^2 + 3\tau_{xy}^2)^{1/2} = [95.49^2 + 3(19.1)^2]^{1/2} = 101.1\ \text{MPa}$$

$$n = \frac{S_y}{\sigma'} = \frac{280}{101.1} = 2.77 \quad \textit{Ans.}$$

B:

$$\sigma_x = \frac{4P}{\pi d^3} = \frac{4(8)(10^3)}{\pi(0.020^2)} = 25.47(10^6)\ \text{Pa} = 25.47\ \text{MPa}$$

wrong, 2

$$\tau_{xy} = \frac{16T}{\pi d^3} + \frac{4}{3}\frac{V}{A} = \frac{16(30)}{\pi(0.020^3)} + \frac{4}{3}\left[\frac{0.55(10^3)}{(\pi/4)(0.020^2)}\right]$$

$$= 21.43(10^6)\ \text{Pa} = 21.43\ \text{MPa}$$

$$\sigma' = [25.47^2 + 3(21.43^2)]^{1/2} = 45.02\ \text{MPa}$$

$$n = \frac{280}{45.02} = 6.22 \quad \textit{Ans.}$$

ME 208 W10 Exam #1 50 minutes Name:____________________

Open Textbook; Closed Notes, Only one page crib sheet allowed (you need to turn in the crib sheet with your exam)

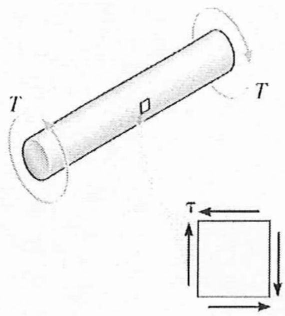
1. (10 pts)
For each of the following failure theories, indicate which would be best described as more "accurate" as discussed in the class? (circle all that apply)

(a) Max. Normal Stress
(b) Max. Shear Stress
(c) Distortion Energy
(d) Coulomb Mohr
(e) Modified Mohr

Solution:

(c), (e)

2. (10 pts)
When the torsional loading T is applied to a bar, it produces a state of pure shear stress in the material. If sufficient T is applied, to find at what angle the bar would break, circle all that apply in the following statements.

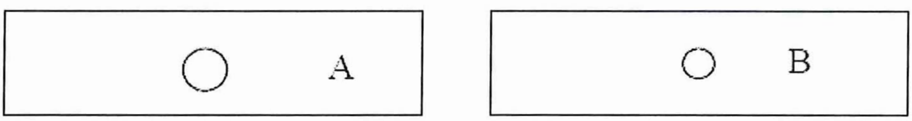


(a) The bar will surely break at the far end
(b) $\sigma_1 = \tau$
(c) If the material is mild steel, it will break at 45°
(d) If the material is cast iron, it will break at 45°
(e) If the material is AISI 1080 steel, it will break at 45°

Solution:

(b), (d)

3. (10 pts)



Two plates are identical except for the size of the hole. They are subject to the same tension load.

(a) (3 pts) If stress concentration is considered, which part has the higher stress concentration?
Circle one: A B C Can't tell

(b) (3 pts) If stress concentration is ignored, which part has the higher nominal stress?
Circle one: A B C Can't tell

(c) (4 pts) If stress concentration is considered, which part has the higher peak stress?
Circle one: A B C Can't tell

Solution:

(a) B
(b) A
(c) C

4. (10 pts)
Identify in which of the following situations the effects of stress concentration would generally be neglected: (circle all that apply)

(a) a shaft made of 1020 CD steel, rotating at constant speed, with steady loads being transmitted through gears supported against shoulders on the shaft.
(b) a bracket made of low carbon angle-iron, mounted to a wall with bolts through bolt-holes, supporting a display case.
(c) a bracket made of 1040 CD steel, bolted to the base of a rocker/recliner chair, supporting the spring.
(d) a high-strength alloy steel part, that has been severely quenched. The part contains a small hole, and is loaded statically.
(e) None of the above

Solution:

(b)

5. (10 pts)
An ASTM cast iron, grade 30, carries static loading resulting in the stress state listed below at the critical locations. To use Modified Mohr theory to estimate the factors of safety n when $\tau_{xy} = 14$ kpsi, which of the following is (are) true?

Circle <u>all</u> that apply (find n and make sure that you check against every answer):

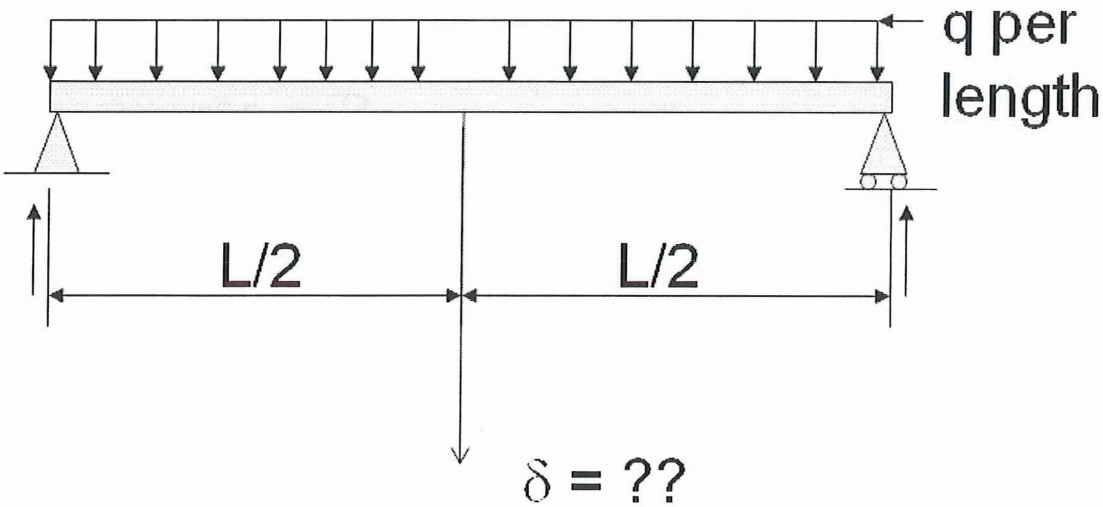
(a) $n < 1$
(b) $n > 1$
(c) $n > 1.5$
(d) $n > 2$

(e) $n > 2.5$

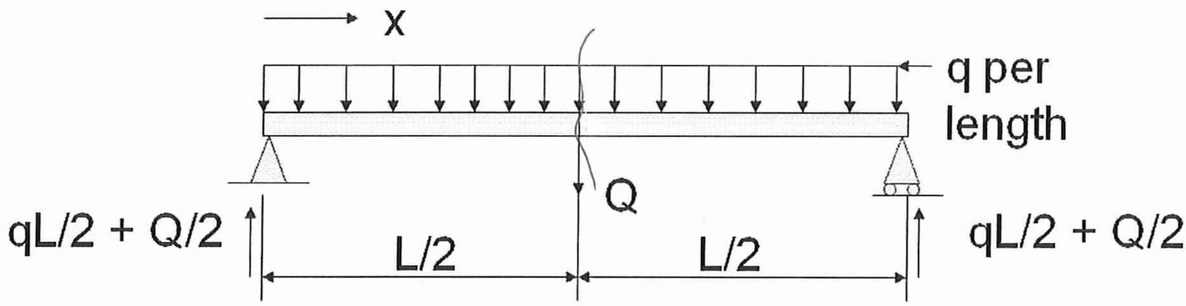
Solution:

(b), (c), (d)

6. (25 pts)
Solve the problem as shown in the figure using Castigliano's theorem. Young's Modulus of the beam is E and moment of inertia of the beam cross section is I. Find the central deflection of the member in terms of q, L, E, I.



Solution:



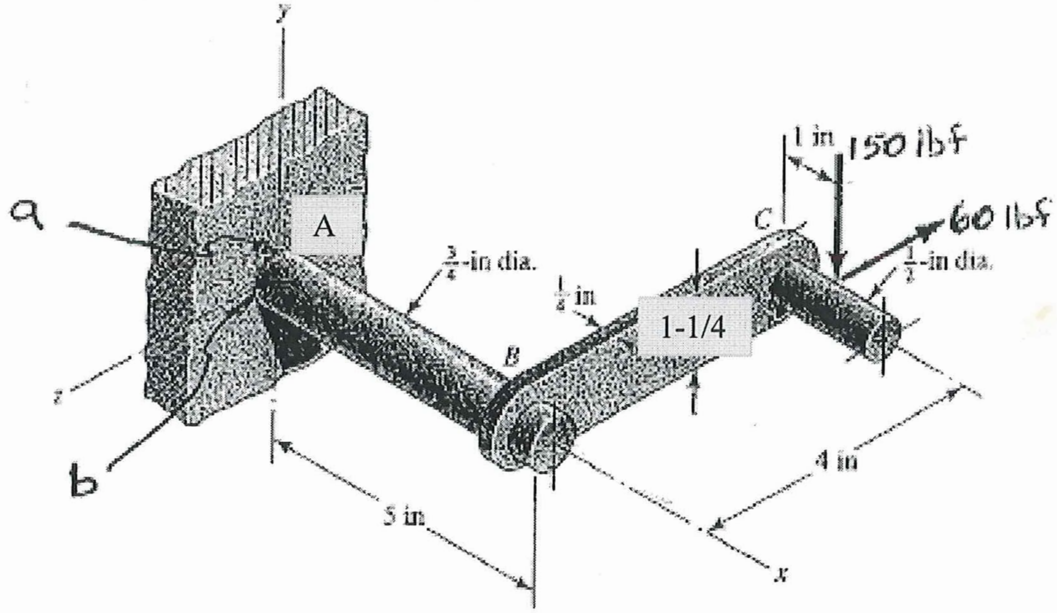
For the left half $M_b = - q\,x^2/2 + qLx/2 + Qx/2$ $\quad \partial M_b/\partial Q = x/2$

The total energy is twice that of the left half.

$$\delta = \frac{\partial U}{\partial Q} = \frac{2}{EI}\int_0^{L/2} M_b\left(\frac{\partial M_b}{\partial Q}\right)dx = \frac{2}{EI}\int_0^{L/2}\left(qLx^2/4 + Qx^2/4 - qx^3/4\right)dx$$

With $Q = 0$ evaluating the integral $\Delta = (5/384)\, qL^4/(EI)$

7. (25 pts)
For the cantilevered beam AB as shown, use the Maximum-Shear-Stress Theory to determine the yielding factor of safety at the critical stress element. Assume the material is 1030 cold drawn steel. (Note: Assuming that you have found the critical stress element to be 21.8° CCW from point *a*, with a normal stress from bending of 23400 psi and a torsional shear stress of 7200 psi.)



Solution:

MSS: $\sigma_1 - \sigma_3 = S_y/n \quad \Rightarrow \quad n = \dfrac{S_y}{\sigma_1 - \sigma_3}$

From A-20, S_y = 64 kpsi for 1030 CD Steel

$$\sigma_{1,2} = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

σ_x = 23.4 kpsi; σ_y = 0; τ_{xy} = 7.2 kpsi

σ_1 = 25.44 kpsi; σ_3 = -2.04 kpsi; σ_2 = 0 kpsi

n = 64/27.48 = 2.33

Homework

$$n_L = \frac{S_p A_t - F_i}{CP}$$

ME 208
Homework 8

1.
We wish to alter the figure for Prob. 8–22 by decreasing the inside diameter of the seal to the diameter A = 100 mm. This makes an effective sealing diameter of 120 mm. Then, by using cap screws instead of bolts, the bolt circle diameter B can be reduced as well as the outside diameter C. If the same bolt spacing and the same edge distance are used, then eight 12-mm cap screws can be used on a bolt circle with B = 160 mm and an outside diameter of 260 mm, a substantial savings. With these dimensions and all other data the same as in Prob. 8–22, find the load factor.

$F_i = .75\,(A_t \cdot S_p) = 37{,}935$

↑ 600 MPa

$(84.3\ \text{mm}^2)$

Solution

$$P = \frac{1}{8}\left(\frac{\pi}{4}\right)(120^2)(6)(10^{-3}) = 8.48\ \text{kN}$$

From Fig. 8-21, $t_1 = h = 20$ mm and $t_2 = 25$ mm

$S_p =$

$$l = 20 + 12/2 = 26\ \text{mm}$$

$$t = 0 \quad \text{(no washer)}, \qquad L_T = 2(12) + 6 = 30\ \text{mm}$$

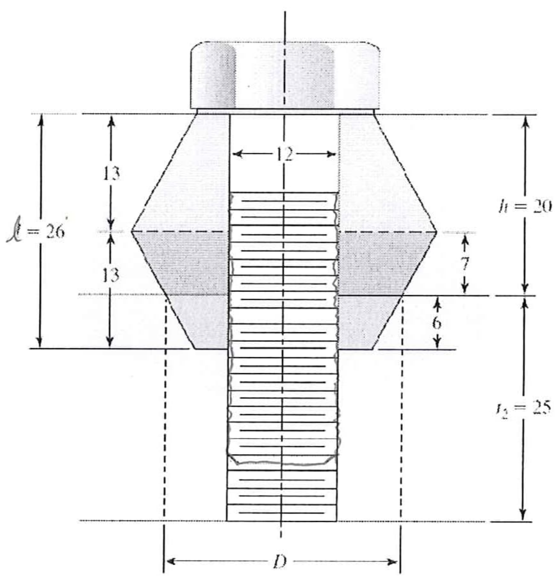
$$L > h + 1.5d = 20 + 1.5(12) = 38\ \text{mm}$$

Use 40 mm cap screws.

$$l_d = 40 - 30 = 10\ \text{mm}$$

$$l_t = l - l_d = 26 - 10 = 16\ \text{mm}$$

$$A_d = 113\ \text{mm}^2, \quad A_t = 84.3\ \text{mm}^2$$



Eq. (8-17):

$$k_b = \frac{113(84.3)(207)}{113(16) + 84.3(10)}$$

$$= 744\ \text{MN/m} \quad \textit{Ans.}$$

$$d_w = 1.5(12) = 18\ \text{mm}$$

$$D = 18 + 2(6)(\tan 30) = 24.9\ \text{mm}$$

$A_d =$

$$k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d} = \frac{113(84.3)(207)}{113(16) + 84.3(10)} = 744\ \text{MN/m}$$

$L = 20 + 1.5(12)$ $\quad h = 20 \quad t_2 = 25$

$d = 12$

$L = 38$ $\quad l_t$

$L_T = 2d + 6\ \text{mm} = 2(12) + 6 = 30\ \text{mm}$

$l_d = L - L_T = 38 - 30 = 8\ \text{mm}$

$l = 20 + \frac{12}{2} = 26\ \text{mm}$ $\quad l_t = l - l_d = 16\ \text{mm}$

From Eq. (8-20): $d = 12$

Top frustum: $D = 18, t = 13, E = 207$ GPa $\Rightarrow$ $k_1 = 5316$ MN/m

Mid-frustum: $t = 7, E = 207$ GPa, $D = 24.9$ mm $\Rightarrow$ $k_2 = 15\,620$ MN/m

Bottom frustum: $D = 18, t = 6, E = 100$ GPa $\Rightarrow$ $\boxed{k_3 = 3887 \text{ MN/m}}$

$$k_m = \frac{1}{(1/5316) + (1/55\,620) + (1/3887)} = 2158 \text{ MN/m} \quad \textit{Ans.}$$

$$C = \frac{744}{744 + 2158} = 0.256 \quad \textit{Ans.}$$

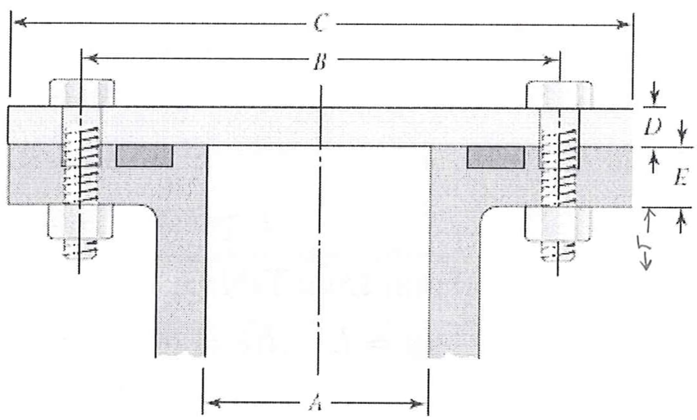
From Prob. 8-22, $F_i = 37.9$ kN

$$n = \frac{S_p A_t - F_i}{CP} = \frac{600(0.0843) - 37.9}{0.256(8.48)} = 5.84 \quad \textit{Ans.}$$

$S_p A_t - .75 (S_p A_t)$

2.
In the figure for Prob. 8–20, let A = 0.9 m, B = 1 m, C = 1.10 m, D = 20 mm, and E = 25 mm. The cylinder is made of ASTM No. 35 cast iron (E = 96 GPa), and the head, of low-carbon steel. There are thirty-six M10 × 1.5 ISO 10.9 bolts tightened to 75 percent of proof load. During use, the cylinder pressure fluctuates between 0 and 550 kPa. Find the factor of safety guarding against a fatigue failure of a bolt using the:

(a) Goodman criterion.
(b) Gerber criterion.
(c) ASME-elliptic criterion.



Problem 8–20
Cylinder head is steel; cylinder is grade 30 cast iron.

$$n = \frac{S_p A_t - F_i}{CP}$$

$$C = \frac{k_b}{(k_m + k_b)} \qquad k_b$$

$$P = \frac{\pi}{4}(.15)^2 \left(6 \times 10^6 \, N/m^2\right)\left(\frac{1}{10 \text{ bolts}}\right) = 10603 \text{ N} = 10.6 \text{ kN}$$

d = 12mm Table A-31 ⇒ H = 10.8mm

$l = D + E = 40$ mm

$L > 40 + 10.8$ So > 50.8 → choose 60mm from A-17

$$P = \frac{pA}{N} = \frac{\pi D^2 p}{4N} = \frac{\pi(0.9^2)(550)}{4(36)} = 9.72 \text{ kN/bolt}$$

Table 8-11: $S_p = 830$ MPa, $S_{ut} = 1040$ MPa, $S_y = 940$ MPa

Table 8-1: $A_t = 58 \text{ mm}^2$

$$A_d = \pi(10^2)/4 = 78.5 \text{ mm}^2$$

$$l = D + E = 20 + 25 = 45 \text{ mm}$$

$$L_T = 2(10) + 6 = 26 \text{ mm}$$

Table A-31: $H = 8.4$ mm

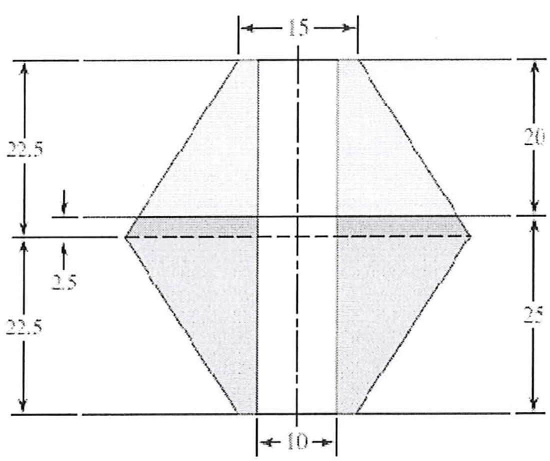
$$L \geq l + H = 45 + 8.4 = 53.4 \text{ mm}$$

Choose $L = 60$ mm from Table A-17

$$l_d = L - L_T = 60 - 26 = 34 \text{ mm}$$

$$l_t = l - l_d = 45 - 34 = 11 \text{ mm}$$

$$k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d} = \frac{78.5(58)(207)}{78.5(11) + 58(34)} = 332.4 \text{ MN/m}$$



Frustum 1: Top, $E = 207$, $t = 20$ mm, $d = 10$ mm, $D = 15$ mm

$$k_1 = \frac{0.5774\pi(207)(10)}{\ln\left\{\left[\dfrac{1.155(20) + 15 - 10}{1.155(20) + 15 + 10}\right]\left(\dfrac{15+10}{15-10}\right)\right\}}$$
$$= 3503 \text{ MN/m}$$

Frustum 2: Middle, $E = 96$ GPa, $D = 38.09$ mm, $t = 2.5$ mm, $d = 10$ mm

$$k_2 = \frac{0.5774\pi(96)(10)}{\ln\left\{\left[\dfrac{1.155(2.5) + 38.09 - 10}{1.155(2.5) + 38.09 + 10}\right]\left(\dfrac{38.09+10}{38.09-10}\right)\right\}}$$
$$= 44\,044 \text{ MN/m}$$

could be neglected due to its small influence on k_m.

Frustum 3: Bottom, $E = 96$ GPa, $t = 22.5$ mm, $d = 10$ mm, $D = 15$ mm

$$k_3 = \frac{0.5774\pi(96)(10)}{\ln\left\{\left[\dfrac{1.155(22.5) + 15 - 10}{1.155(22.5) + 15 + 10}\right]\left(\dfrac{15+10}{15-10}\right)\right\}}$$
$$= 1567 \text{ MN/m}$$

$$k_m = \frac{1}{(1/3503) + (1/44\,044) + (1/1567)} = 1057 \text{ MN/m}$$

$$C = \frac{332.4}{332.4 + 1057} = 0.239$$

$$F_i = 0.75A_t S_p = 0.75(58)(830)(10^{-3}) = 36.1 \text{ kN}$$

Table 8-17: $S_e = 162$ MPa

$$\sigma_i = \frac{F_i}{A_t} = \frac{36.1(10^3)}{58} = 622 \text{ MPa}$$

(a) Goodman Eq. (8-40)

$$S_a = \frac{S_e(S_{ut} - \sigma_i)}{S_{ut} + S_e} = \frac{162(1040 - 622)}{1040 + 162} = 56.34 \text{ MPa}$$

$$n_f = \frac{56.34}{20} = 2.82 \quad \textit{Ans.}$$

(b) Gerber Eq. (8-42)

$$S_a = \frac{1}{2S_e}\left[S_{ut}\sqrt{S_{ut}^2 + 4S_e(S_e + \sigma_i)} - S_{ut}^2 - 2\sigma_i S_e\right]$$

$$= \frac{1}{2(162)}\left[1040\sqrt{1040^2 + 4(162)(162 + 622)} - 1040^2 - 2(622)(162)\right]$$

$$= 86.8 \text{ MPa}$$

$$\sigma_a = \frac{CP}{2A_t} = \frac{0.239(9.72)(10^3)}{2(58)} = 20 \text{ MPa}$$

$$n_f = \frac{S_a}{\sigma_a} = \frac{86.8}{20} = 4.34 \quad \textit{Ans.}$$

(c) ASME elliptic

$$S_a = \frac{S_e}{S_p^2 + S_e^2}\left(S_p\sqrt{S_p^2 + S_e^2 - \sigma_i^2} - \sigma_i S_e\right)$$

$$= \frac{162}{830^2 + 162^2}\left[830\sqrt{830^2 + 162^2 - 622^2} - 622(162)\right] = 84.90 \text{ MPa}$$

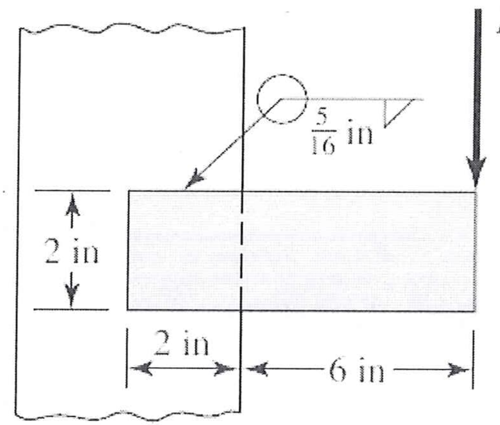
$$n_f = \frac{84.90}{20} = 4.24 \quad \textit{Ans.}$$

3.
The figure shows a weldment just like that of Prob. 9–5 except that there are four welds instead of two. Show that the weldment is twice as strong as that of Prob. 9–5.

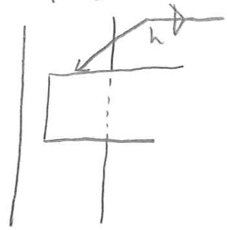
τ_{allow} 140 MPa

E7010

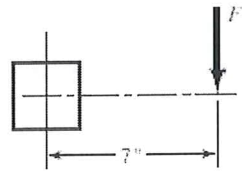
S_{ut} =



9-5



$b = d = 2$ in



Primary shear

$$\tau_y' = \frac{V}{A} = \frac{F}{1.414(5/16)(2+2)} = 0.566F$$

Secondary shear

Table 9-1: $$J_u = \frac{(b+d)^3}{6} = \frac{(2+2)^3}{6} = 10.67 \text{ in}^3$$

$$J = 0.707hJ_u = 0.707(5/16)(10.67) = 2.36 \text{ in}^4$$

$$\tau_x'' = \tau_y'' = \frac{Mr_y}{J} = \frac{(7F)(1)}{2.36} = 2.97F$$

Maximum shear

$$\tau_{\max} = \sqrt{\tau_x''^2 + (\tau_y' + \tau_y'')^2} = F\sqrt{2.97^2 + (0.556 + 2.97)^2} = 4.61F \text{ kpsi}$$

$$F = \frac{\tau_{\text{all}}}{4.61} \quad \textit{Ans.}$$

which is twice $\tau_{\max}/9.22$ of Prob. 9-5.

ME 208
Homework 7

1. An 02-series ball bearing is to be selected to carry a radial load of 8 kN and a thrust load of 4 kN. The desired life LD is to be 5000 h with an inner-ring rotation rate of 900 rev/min. What is the basic load rating that should be used in selecting a bearing for a reliability goal of **0.96**?

From Prob. 11-6, $x_D = 270$ and the final value of F_e is 10.60 kN.

$$C_{10} = 10.6\left\{\frac{270}{0.02 + 4.439[\ln(1/0.96)]^{1/1.483}}\right\}^{1/3} = 84.47 \text{ kN}$$

Table 11-2: Choose a deep-groove ball bearing, based upon C_{10} load ratings.

Trial #1:

Tentatively select a 02-90 mm.

$$C_{10} = 95.6, \quad C_0 = 62 \text{ kN}$$

$$\frac{F_a}{C_0} = \frac{4}{62} = 0.0645$$

From Table 11-1, interpolate for Y_2.

F_a/C_0	Y_2
0.056	1.71
0.0645	Y_2
0.070	1.63

$$\frac{Y_2 - 1.71}{1.63 - 1.71} = \frac{0.0645 - 0.056}{0.070 - 0.056} = 0.607$$

$$Y_2 = 1.71 + 0.607(1.63 - 1.71) = 1.661$$

$$F_e = 0.56(8) + 1.661(4) = 11.12 \text{ kN}$$

$$C_{10} = 11.12\left\{\frac{270}{0.02 + 4.439[\ln(1/0.96)]^{1/1.483}}\right\}^{1/3}$$

$$= 88.61 \text{ kN} < 95.6 \text{ kN}$$

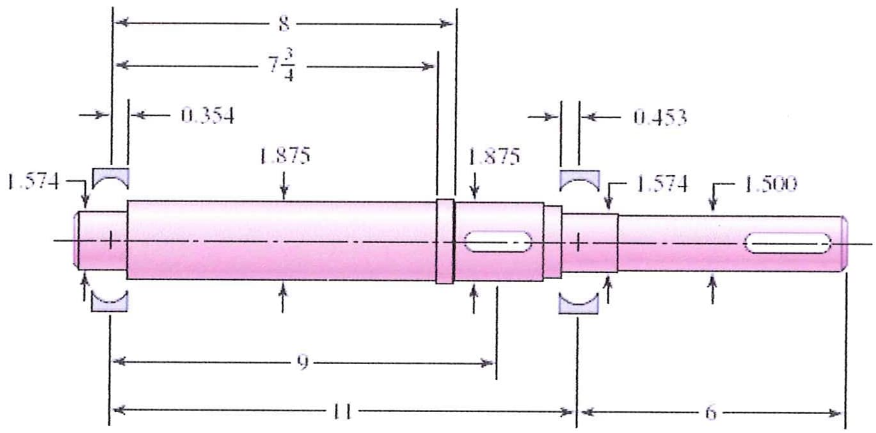
Bearing is OK.

Decision: Specify a deep-groove 02-90 mm ball bearing. *Ans.*

p 402
p 368

2. In the figure is a proposed shaft design to be used for the input shaft a in Prob. 7–7. A ball bearing is planned for the left bearing, and a cylindrical roller bearing for the right. (a) Determine the fatigue factor of safety by evaluating at pinion seat keyway (assuming end mill keyway radius = 0.01”). Use ASME Elliptic fatigue failure criteria. Also ensure that the shaft does not yield in the first load cycle. (b) To avoid complex calculation, approximate the shaft as a uniform shaft with diameter of 1.875 in, and check the design for adequacy with respect to deformation, according to the recommendations in Table 7–2. Use 1030 HR.

Shoulder fillets at bearing seat 0.030-in radius, others $\frac{1}{8}$-in radius, except right-hand bearing seat transition, $\frac{1}{4}$ in. The material is 1030 HR. Keyways $\frac{3}{8}$ in wide by $\frac{3}{16}$ in deep. Dimensions in inches.



Solution

Candidate critical locations for strength:

- Pinion seat keyway
- Right bearing shoulder
- Coupling keyway

Table A-20 for 1030 HR: $S_{ut} = 68$ kpsi, $S_y = 37.5$ kpsi, $H_B = 137$

Eq. (6-8): $S'_e = 0.5(68) = 34.0$ kpsi

Eq. (6-19): $k_a = 2.70(68)^{-0.265} = 0.883$

$k_c = k_d = k_e = 1$

Pinion seat keyway

See Table 7-1 for keyway stress concentration factors

$$\left.\begin{aligned} K_t &= 2.2 \\ K_{ts} &= 3.0 \end{aligned}\right\} \text{Profile keyway}$$

For an end-mill profile keyway cutter of 0.010 in radius.

From Fig. 6-20: $q = 0.50$

From Fig. 6-21: $q_s = 0.65$

Eq. (6-32):

$$\begin{aligned} K_{fs} &= 1 + q_s(K_{ts} - 1) \\ &= 1 + 0.65(3.0 - 1) = 2.3 \\ K_f &= 1 + 0.50(2.2 - 1) = 1.6 \end{aligned}$$

Eq. (6-20): $$k_b = \left(\frac{1.875}{0.30}\right)^{-0.107} = 0.822$$

Eq. (6-18): $$S_e = 0.883(0.822)(34.0) = 24.7\,\text{kpsi}$$

Eq. (7-11):

$$\frac{1}{n} = \frac{16}{\pi(1.875^3)}\left\{4\left[\frac{1.6(2178)}{24\,700}\right]^2 + 3\left[\frac{2.3(2500)}{37\,500}\right]^2\right\}^{1/2}$$

$= 0.3$ and thus $n = 3.33$.

$\sigma'_{max} = 11{,}342$ psi using d = 1.875" (see example 7.7 in the class note, depending on the q and q_s you used, σ'_{max} may be different. I used the same numbers as used in example 7.7)

For yield, $n = S_y/\sigma'_{max} = 37500/11342 = 3.31$ ok

(b) One could take pains to model this shaft exactly, using say finite element software. However, for the bearings and the gear, the shaft is basically of uniform diameter, 1.875 in. The reductions in diameter at the bearings will change the results insignificantly. Use $E = 30(10^6)$ psi.

To the left of the load:

$$\theta_{AB} = \frac{Fb}{6EIl}(3x^2 + b^2 - l^2)$$

$$= \frac{1330(2)(3x^2 + 2^2 - 11^2)}{6(30)(10^6)(\pi/64)(1.825^4)(11)}$$

$$= 2.2143(10^{-6})(3x^2 - 117)$$

At $x = 0$: $\theta = -2.59(10^{-4})$ rad

At $x = 9$ in: $\theta = 2.79(10^{-4})$ rad

At $x = 11$ in:

$$\theta = \frac{1330(9)(11^2 - 9^2)}{6(30)(10^6)(\pi/64)(1.875^4)(11)} = 3.98(10^{-4}) \text{ rad}$$

Obtain allowable slopes from Table 7-2.

Left bearing:

$$n_{fs} = \frac{\text{Allowable slope}}{\text{Actual slope}} = \frac{0.001}{0.000259} = 3.86$$

Right bearing:

$$n_{fs} = \frac{0.0008}{0.000398} = 2.01$$

Gear mesh slope:

Table 7-2 recommends a minimum relative slope of 0.0005 rad. While we don't know the slope on the next shaft, we know that it will need to have a larger diameter and be stiffer. At the moment we can say

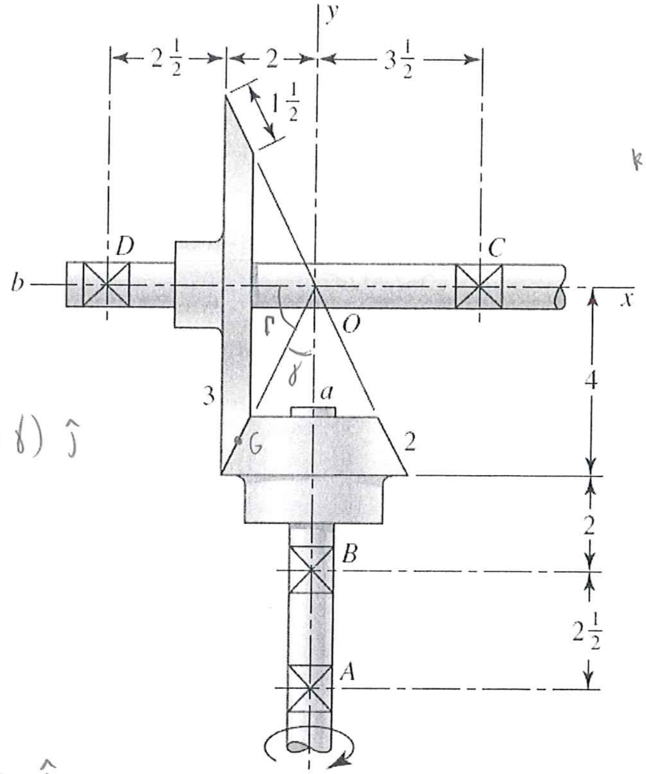
$$n_{fs} < \frac{0.0005}{0.000279} = 1.79$$

ME 208
Homework #6

$\gamma = 26.57$

$\Gamma = 63.43$

1. The figure shows a 16T 20∘ straight bevel pinion driving a 32T gear, and the location of the bearing centerlines. Pinion shaft a receives 2.5 hp at 240 rev/min. Determine the bearing reactions at A and B if A is to take both radial and thrust loads.



$R_{G/A} = (-2 + 1.5\sin\gamma)\,\hat{\imath} + (4.5 + 1.5\cos\gamma)\,\hat{\jmath}$

$= -1.664\,\hat{\imath} + 5.171\,\hat{\jmath}$

$\Sigma M_A = R_{G/A} \times F_G + R_{B/A} \times F_B + T = 0$

$T = \frac{H}{\omega} \quad \frac{2.5\,hp\,(6600\ in\cdot lb/s)}{240(2\pi)\left(\frac{1\,min}{60\,sec}\right)} = 656.5\ in\cdot lb\ \hat{\jmath}$

$W^t = 328.3\ \hat{k}$

$W_r = 106.9\ \hat{\imath}$

$W_a = 53.4\ -\hat{\jmath}$

$(-1698,\ 546.3,\ -463.9) - (565.5\,\hat{\jmath}) = -R_{B/A} \times F_B$

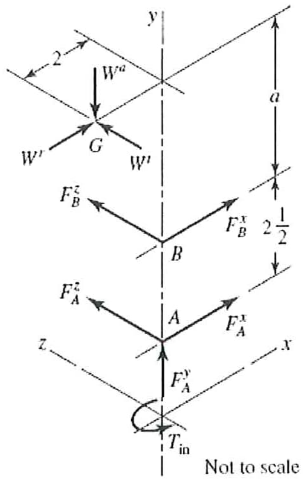
$(-1698,\ -19.2,\ -464) = (0,\ 2.5,\ 0) \times (F_{Bx},\ F_{By},\ F_{Bz})$

$-2.5\,F_{Bx}\,\hat{k} + 0\,F_{By} + 2.5\,F_B\,\hat{\imath}$

$F_B = -679\,\hat{k} - 186\,\hat{\imath}$

$F_{Az} = 679 - 328 = 351\,\hat{k}$

$T_{\text{in}} = 63\,025H/n = 63\,025(2.5)/240 = 656.5\ \text{lbf}\cdot\text{in}$



$$W^t = T/r = 656.5/2 = 328.3\ \text{lbf}$$

$$\gamma = \tan^{-1}(2/4) = 26.565^\circ$$

$$\Gamma = \tan^{-1}(4/2) = 63.435^\circ$$

$$a = 2 + (1.5\cos 26.565^\circ)/2 = 2.67\ \text{in}$$

$$W^r = 328.3\tan 20^\circ \cos 26.565^\circ = 106.9\ \text{lbf}$$

$$W^a = 328.3\tan 20^\circ \sin 26.565^\circ = 53.4\ \text{lbf}$$

$$\mathbf{W} = 106.9\mathbf{i} - 53.4\mathbf{j} + 328.3\mathbf{k}\ \text{lbf}$$

$$\mathbf{R}_{AG} = -2\mathbf{i} + 5.17\mathbf{j}, \quad \mathbf{R}_{AB} = 2.5\mathbf{j}$$

$$\sum \mathbf{M}_4 = \mathbf{R}_{AG} \times \mathbf{W} + \mathbf{R}_{AB} \times \mathbf{F}_B + \mathbf{T} = \mathbf{0}$$

Solving gives

$$\mathbf{R}_{AB} \times \mathbf{F}_B = 2.5F_B^z\mathbf{i} - 2.5F_B^x\mathbf{k}$$

$$\mathbf{R}_{AG} \times \mathbf{W} = 1697\mathbf{i} + 656.6\mathbf{j} - 445.9\mathbf{k}$$

So

$$(1697\mathbf{i} + 656.6\mathbf{j} - 445.9\mathbf{k}) + \left(2.5F_B^z\mathbf{i} - 2.5F_B^x\mathbf{k} + T\mathbf{j}\right) = \mathbf{0}$$

$$F_B^z = -1697/2.5 = -678.8\ \text{lbf}$$

$$T = -656.6\ \text{lbf}\cdot\text{in}$$

$$F_B^x = -445.9/2.5 = -178.4\ \text{lbf}$$

So

$$F_B = [(-678.8)^2 + (-178.4)^2]^{1/2} = 702\ \text{lbf} \quad \textit{Ans.}$$

$$\mathbf{F}_A = -(\mathbf{F}_B + \mathbf{W})$$

$$= -(-178.4\mathbf{i} - 678.8\mathbf{k} + 106.9\mathbf{i} - 53.4\mathbf{j} + 328.3\mathbf{k})$$

$$= 71.5\mathbf{i} + 53.4\mathbf{j} + 350.5\mathbf{k}$$

$$F_A\ (\text{radial}) = (71.5^2 + 350.5^2)^{1/2} = 358\ \text{lbf} \quad \textit{Ans.}$$

$$F_A\ (\text{thrust}) = 53.4\ \text{lbf} \quad \textit{Ans.}$$

HW #6

$\# \text{teeth} (N_p) / P_d$

2. A speed-reducer has 20° full-depth teeth, and the single-reduction spur-gear gearset has 22 and 60 teeth. The diametral pitch is 4 teeth/in and the face width is 3.25 in. The pinion shaft speed is 1145 rev/min. The life goal of 5-year 24-hour-per-day service is about 3(109) pinion revolutions. The absolute value of the pitch variation is such that the transmission accuracy level number is 6. The materials are 4340 through-hardened grade 1 steels, heat-treated to 250 Brinell, core and case, both gears. The load is moderate shock and the power is smooth. For a reliability of 0.99, rate the speed reducer for power (i.e, what is the minimum horsepower this speed reducer can sustain?). (Hint: you need to calculate W_t for both bending and wear and for both gears → 4 W_t s, and find the minimum).

commercial, enclosed units

Do Not need:

Teeth	Y
22	.331
60	.422

$P_d = 4$ teeth/inch $N_p = 22$ $N_G = 60$ life $= 3\times10^9$ revolutions

$F = 3.25$ in $d_p = 5.5$ in

$k_o = 1.25$

Pinion Bending / $\sigma = W^t k_o k_v k_s \frac{P_d}{F} \frac{K_m K_B}{J}$

$W^t = \frac{33{,}000\ H}{V}$ $V = \frac{\pi d n}{12} = \frac{1145 \text{ rev}}{\text{min}} \frac{d_p(\pi)}{12}$ $d_p = \frac{N_p}{P_d} = \frac{22\ t}{4\ t/\text{in}} \rightarrow 5.5$ in

$V = \frac{\pi (5.5 \text{ in})(1145 \text{ rev/min})}{12 \text{ in/ft}} = 1649$ ft/min

$H = ?$ (power)

$k_o = 1.25$

$k_v = \left(\frac{A+\sqrt{V}}{A}\right)^B$ $A = 50 + 56(1-B)$ $B = .25(12-Q_v)^{2/3}$ $B = .25(12-6)^{2/3} = .8255$ $A = 59.77$

$k_v = 1.534$ $\frac{P_d}{F} = \frac{4 \text{ teeth/in}}{3.25 \text{ in}}$ $C_{mc} = 1$

$k_s = 1$ $C_{pf} = \frac{F}{10d} - .0375 + .0125F = .0622$

$C_{pm} = 1$ $C_e = 1$ $C_{ma} = .127 + .0158(3.25) - .930(10^{-4})(3.25)^2 = .1774$

$K_m = 1 + 1[(.0622)(1) + .1774(1)] = 1.240$ $K_B = 1$ $J = .34$

$\sigma = 20.01H(1.25)(1.534)(1)\left(\frac{4}{3.25}\right)\frac{1.240(1)}{.34}$ $N_c = 112$

$\sigma = 172.2\ H$

$\sigma_{all} =$

$Y_P = 0.331$, $Y_G = 0.422$, $J_P = 0.345$, $J_G = 0.410$, $K_o = 1.25$. The service conditions are adequately described by K_o. Set $S_F = S_H = 1$.

$$d_P = 22/4 = 5.500 \text{ in}$$

$$d_G = 60/4 = 15.000 \text{ in}$$

$$V = \frac{\pi(5.5)(1145)}{12} = 1649 \text{ ft/min}$$

Pinion bending

$$_{0.99}(S_t)_{10^7} = 77.3H_B + 12\,800 = 77.3(250) + 12\,800 = 32\,125 \text{ psi}$$

$$Y_N = 1.6831[3(10^9)]^{-0.0323} = 0.832$$

Eq. (14-17):
$$(\sigma_{\text{all}})_P = \frac{32\,125(0.832)}{1(1)(1)} = 26\,728 \text{ psi}$$

$$B = 0.25(12 - 6)^{2/3} = 0.8255$$

$$A = 50 + 56(1 - 0.8255) = 59.77$$

$$K_v = \left(\frac{59.77 + \sqrt{1649}}{59.77}\right)^{0.8255} = 1.534$$

$$K_s = 1, \quad C_m = 1$$

$$C_{mc} = \frac{F}{10d} - 0.0375 + 0.0125F$$

$$= \frac{3.25}{10(5.5)} - 0.0375 + 0.0125(3.25) = 0.0622$$

$$C_{ma} = 0.127 + 0.0158(3.25) - 0.093(10^{-4})(3.25^2) = 0.178$$

$$C_e = 1$$

$$K_m = C_{mf} = 1 + (1)[0.0622(1) + 0.178(1)] = 1.240$$

$$K_B = 1, \quad K_T = 1$$

Eq. (14-15):
$$W_1^t = \frac{26\,728(3.25)(0.345)}{1.25(1.534)(1)(4)(1.240)} = 3151 \text{ lbf}$$

$$H_1 = \frac{3151(1649)}{33\,000} = 157.5 \text{ hp}$$

Gear bending By similar reasoning, $W_2^t = 3861$ lbf and $H_2 = 192.9$ hp

Pinion wear

$$m_G = 60/22 = 2.727$$

$$I = \frac{\cos 20^\circ \sin 20^\circ}{2}\left(\frac{2.727}{1+2.727}\right) = 0.1176$$

$$_{0.99}(S_c)_{10^7} = 322(250) + 29\,100 = 109\,600 \text{ psi}$$

$$(Z_N)_P = 2.466[3(10^9)]^{-0.056} = 0.727$$

$$(Z_N)_G = 2.466[3(10^9)/2.727]^{-0.056} = 0.769$$

$$(\sigma_{c,\text{all}})_P = \frac{109\,600(0.727)}{1(1)(1)} = 79\,679 \text{ psi}$$

$$W_3^t = \left(\frac{\sigma_{c,\text{all}}}{C_p}\right)^2 \frac{F d_P I}{K_o K_v K_s K_m C_f}$$

$$= \left(\frac{79\,679}{2300}\right)^2 \left[\frac{3.25(5.5)(0.1176)}{1.25(1.534)(1)(1.24)(1)}\right] = 1061 \text{ lbf}$$

$$H_3 = \frac{1061(1649)}{33\,000} = 53.0 \text{ hp}$$

Gear wear

Similarly, $\quad W_4^t = 1182 \text{ lbf}, \quad H_4 = 59.0 \text{ hp}$

Rating

$$H_{\text{rated}} = \min(H_1, H_2, H_3, H_4)$$

$$= \min(157.5, 192.9, 53, 59) = 53 \text{ hp} \quad \textit{Ans.}$$

ME 208
Homework 7

1. An 02-series ball bearing is to be selected to carry a radial load of 8 kN and a thrust load of 4 kN. The desired life LD is to be 5000 h with an inner-ring rotation rate of 900 rev/min. What is the basic load rating that should be used in selecting a bearing for a reliability goal of **0.96**?

From Prob. 11-6, $x_D = 270$ and the final value of F_e is 10.60 kN.

$$C_{10} = 10.6\left\{\frac{270}{0.02 + 4.439[\ln(1/0.96)]^{1/1.483}}\right\}^{1/3} = 84.47 \text{ kN}$$

Table 11-2: Choose a deep-groove ball bearing, based upon C_{10} load ratings.

Trial #1:

Tentatively select a 02-90 mm.

$$C_{10} = 95.6, \quad C_0 = 62 \text{ kN}$$

$$\frac{F_a}{C_0} = \frac{4}{62} = 0.0645$$

From Table 11-1, interpolate for Y_2.

F_a/C_0	Y_2
0.056	1.71
0.0645	Y_2
0.070	1.63

$$\frac{Y_2 - 1.71}{1.63 - 1.71} = \frac{0.0645 - 0.056}{0.070 - 0.056} = 0.607$$

$$Y_2 = 1.71 + 0.607(1.63 - 1.71) = 1.661$$

$$F_e = 0.56(8) + 1.661(4) = 11.12 \text{ kN}$$

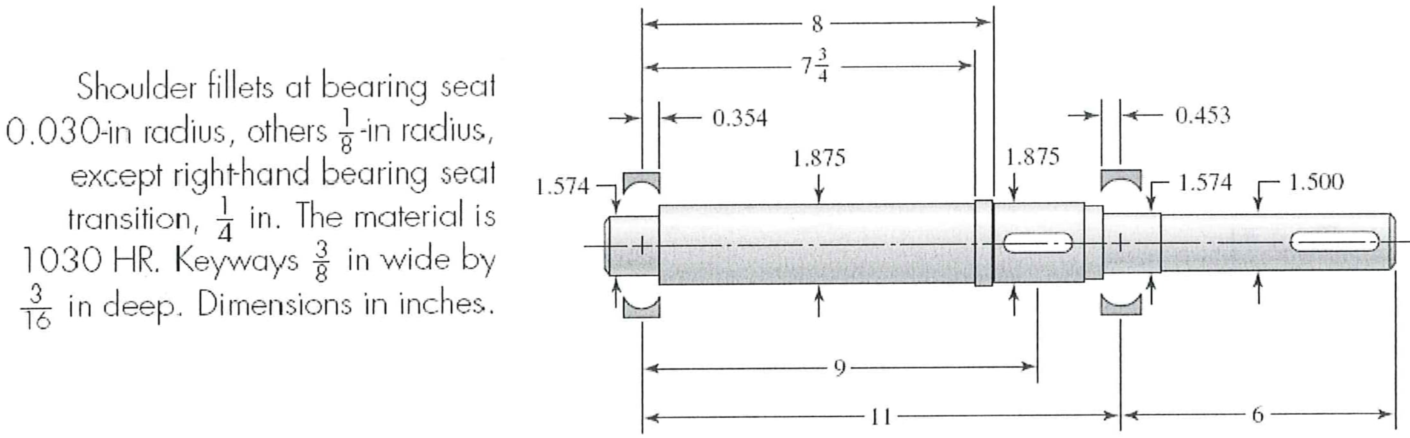
$$C_{10} = 11.12\left\{\frac{270}{0.02 + 4.439[\ln(1/0.96)]^{1/1.483}}\right\}^{1/3}$$

$$= 88.61 \text{ kN} < 95.6 \text{ kN}$$

Bearing is OK.

Decision: Specify a deep-groove 02-90 mm ball bearing. *Ans.*

2. In the figure is a proposed shaft design to be used for the input shaft a in Prob. 7–7. A ball bearing is planned for the left bearing, and a cylindrical roller bearing for the right. (a) Determine the fatigue factor of safety by evaluating at pinion seat keyway (assuming end mill keyway radius = 0.01"). Use ASME Elliptic fatigue failure criteria. Also ensure that the shaft does not yield in the first load cycle. (b) To avoid complex calculation, approximate the shaft as a uniform shaft with diameter of 1.875 in, and check the design for adequacy with respect to deformation, according to the recommendations in Table 7–2. Use 1030 HR.



Solution

Candidate critical locations for strength:

- Pinion seat keyway
- Right bearing shoulder
- Coupling keyway

Table A-20 for 1030 HR: $S_{ut} = 68$ kpsi, $S_y = 37.5$ kpsi, $H_B = 137$

Eq. (6-8): $S'_e = 0.5(68) = 34.0$ kpsi

Eq. (6-19): $k_a = 2.70(68)^{-0.265} = 0.883$

$k_c = k_d = k_e = 1$

Pinion seat keyway

See Table 7-1 for keyway stress concentration factors

$$\left.\begin{aligned} K_t &= 2.2 \\ K_{ts} &= 3.0 \end{aligned}\right\} \text{Profile keyway}$$

For an end-mill profile keyway cutter of 0.010 in radius,

From Fig. 6-20: $q = 0.50$

From Fig. 6-21: $q_s = 0.65$

Eq. (6-32):

$$\begin{aligned} K_{fs} &= 1 + q_s(K_{ts} - 1) \\ &= 1 + 0.65(3.0 - 1) = 2.3 \\ K_f &= 1 + 0.50(2.2 - 1) = 1.6 \end{aligned}$$

Eq. (6-20): $$k_b = \left(\frac{1.875}{0.30}\right)^{-0.107} = 0.822$$

Eq. (6-18): $$S_e = 0.883(0.822)(34.0) = 24.7 \text{ kpsi}$$

Eq. (7-11):

$$\frac{1}{n} = \frac{16}{\pi(1.875^3)}\left\{4\left[\frac{1.6(2178)}{24\,700}\right]^2 + 3\left[\frac{2.3(2500)}{37\,500}\right]^2\right\}^{1/2}$$

= 0.3 and thus n = 3.33.

σ'_{max} = 11,342 psi using d = 1.875" (see example 7.7 in the class note, depending on the q and q_s you used, σ'_{max} may be different. I used the same numbers as used in example 7.7)

For yield, $n = S_y/\sigma'_{max}$ = 37500/ 11342 = 3.31 ok

(b) One could take pains to model this shaft exactly, using say finite element software. However, for the bearings and the gear, the shaft is basically of uniform diameter, 1.875 in. The reductions in diameter at the bearings will change the results insignificantly. Use $E = 30(10^6)$ psi.

To the left of the load:

$$\theta_{AB} = \frac{Fb}{6EIl}(3x^2 + b^2 - l^2)$$

$$= \frac{1330 \quad (2)(3x^2 + 2^2 - 11^2)}{6(30)(10^6)(\pi/64)(1.825^4)(11)}$$

$$= 2.2143 \quad (10^{-6})(3x^2 - 117)$$

At $x = 0$: $\qquad \theta = -2.59 \quad (10^{-4})$ rad

At $x = 9$ in: $\qquad \theta = 2.79 \quad (10^{-4})$ rad

At $x = 11$ in:

$$\theta = \frac{1330 \quad (9)(11^2 - 9^2)}{6(30)(10^6)(\pi/64)(1.875^4)(11)}$$

$$= 3.98 \quad (10^{-4}) \text{ rad}$$

Obtain allowable slopes from Table 7-2.

Left bearing:

$$n_{fs} = \frac{\text{Allowable slope}}{\text{Actual slope}}$$

$$= \frac{0.001}{0.000259} = 3.86$$

Right bearing:

$$n_{fs} = \frac{0.0008}{0.000398} = 2.01$$

Gear mesh slope:

Table 7-2 recommends a minimum relative slope of 0.0005 rad. While we don't know the slope on the next shaft, we know that it will need to have a larger diameter and be stiffer. At the moment we can say

$$n_{fs} < \frac{0.0005}{0.000279} = 1.79$$

ME 208
Homework #5

1. The cold-drawn AISI 1040 steel bar is subjected to an axial fluctuating load. Estimate the factors of safety for fatigue (based on the Goodman line), and for yielding for the following three loading conditions. $S_{ut} =$ $S_y = 71$ kpsi

(a) 600 to 3200 lbf $\sigma_m = 1900$ lbf$/A$

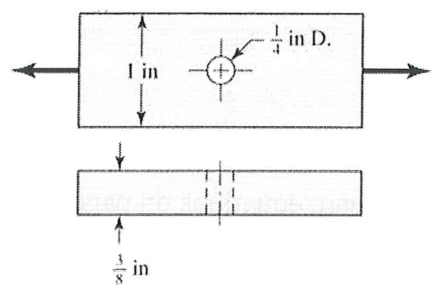
(b) -600 to 3200 lbf $\sigma_m = 1300$ lbf$/A$

(c) 600 and -3200 lbf $\sigma_m = -1300$ lbf$/A$

$k_a k_b k_c k_d k_e S_e'$

$k_c = .85$

k_d



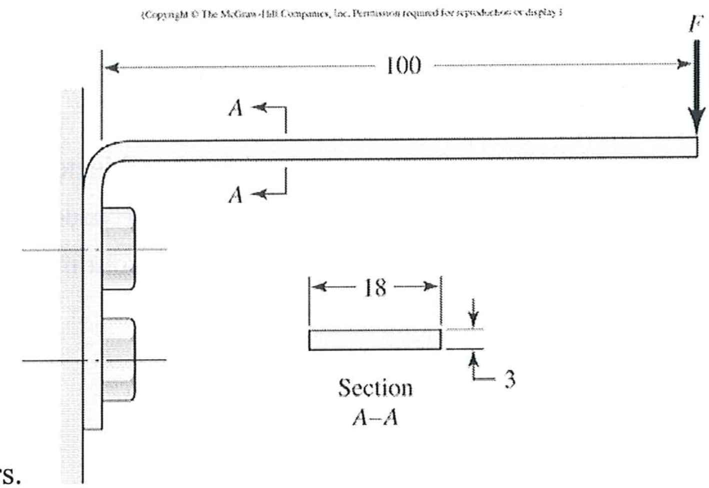
$n > 10^6$

$A = \frac{3}{8}\left(\frac{3}{4}\right) = .2813 \text{ in}^2$

$K_t \approx 2.41$

Partial Solutions: (a) $n_f = 1.98$; $n_y = 2.89$; (b) $n_f = 1.655$; $n_y = 2.89$; (c) $n_f = 2.137$; $n_y = 2.89$;

2. The figure shown below is a drawing of a 3- by 18-mm latching spring. A preload is obtained during assembly by shimming under the bolts to obtain an estimated initial deflection of 2 mm. The latching operation itself requires an additional deflection of exactly 4 mm, thus a total deflection of 6 mm. The material is ground high-carbon steel, bent then hardened and tempered to a minimum hardness of 490 Bhn (See equation 2-17 for relationship of hardness to strength). The radius of the bend is 3 mm. Estimate the yield strength to be 90 percent of the ultimate strength.
   (a) Find the maximum and minimum latching forces.
   (b) Find the fatigue factor of safety based on infinite life, using the modified Goodman criterion.
   (c) Find the static factor of safety.



Dimensions in millimeters.

Partial Solutions:

(a) $F_{min} = 50.3N$; $F_{max} = 150.9N$

(b) You need to use the curved beam equations on pages 112-115 to find the stresses in a curved beam:

Curved beam: $$r_n = \frac{h}{\ln(r_o/r_i)} = \frac{3}{\ln(6/3)} = 4.3281 \text{ mm}$$

$$r_c = 4.5 \text{ mm}, \qquad e = r_c - r_n = 4.5 - 4.3281 = 0.1719 \text{ mm}$$

$$\sigma_i = -\frac{Mc_i}{Aer_i} - \frac{F}{A} = -\frac{(0.1015F)(1.5 - 0.1719)}{54(0.1719)(3)(10^{-3})} - \frac{F}{54} = -4.859F \text{ MPa}$$

$$\sigma_o = \frac{Mc_o}{Aer_o} - \frac{F}{A} = \frac{(0.1015F)(1.5 + 0.1719)}{54(0.1719)(6)(10^{-3})} - \frac{F}{54} = 3.028F \text{ MPa}$$

$$(\sigma_i)_{min} = -4.859(150.9) = -733.2 \text{ MPa}$$

$$(\sigma_i)_{max} = -4.859(50.3) = -244.4 \text{ MPa}$$

$$(\sigma_o)_{max} = 3.028(150.9) = 456.9 \text{ MPa}$$

$$(\sigma_o)_{min} = 3.028(50.3) = 152.3 \text{ MPa}$$

....

S_e =605 Mpa

Inner radius: $n_f = 2.48$; $n_y = 2.58$

Outer radius: $n_f = 2.65$; $n_y = 4.44$

(c)

Inner radius: $n_y = 2.05$

Outer radius: $n_y = 3.29$

HW #5)

1) $k_a = a\,S_{ut}^{\,b} = 2.70(85\,\text{kpsi})^{-.265}$

table 6-2 notch sensitivity

$q = 0.82$

$k_b = 1$ axial

$k_c = 0.85$ axial

$K_f = 1 + q(K_t - 1) = 1 + .82(2.41 - 1)$

$k_d = 1$ temp not given

$K_f = 2.156$

$k_e = 1$ not included, K_f not included

$S_e' = 0.5(S_{ut})$ @ $N > 10^6$

$S_e' = 42.5$

$S_e = (.8319)(0.85)(S_e')$

$S_e = 30.05$ kpsi

a) $\sigma_{min} = \dfrac{600\ \text{lbf}}{.2813\ \text{in}^2} = 2133$ psi

$\sigma_{max} = \dfrac{3200\ \text{lbf}}{.2813\ \text{in}^2} = 11380$ psi

s.c. factor K_f Answers: 14.62

$\sqrt{}\ \sigma_m = 6.754\ \text{kpsi}\,(2.156) = 14.56$

$n_f = \dfrac{30.05\,\text{kpsi}}{14.56\ \text{kpsi}} =$

s.c. factor

$\sigma_a = \dfrac{\sigma_{max} - \sigma_{min}}{2} = \dfrac{11.38 - 2.133}{2} = 4.624\ \text{kpsi}\,(2.156) = 9.969$

$n_f = \left(\dfrac{\sigma_a}{S_e} + \dfrac{\sigma_m}{S_{ut}}\right)^{-1} = \left(\dfrac{9.969}{30.05} + \dfrac{14.56}{85}\right)^{-1}$ $\boxed{n_f = 1.988}$

$n_y = \dfrac{S_y}{|\sigma_{max}|} = \dfrac{71}{11.38(2.156)} = \boxed{n_y = 2.894}$

where do I get this

ME 208

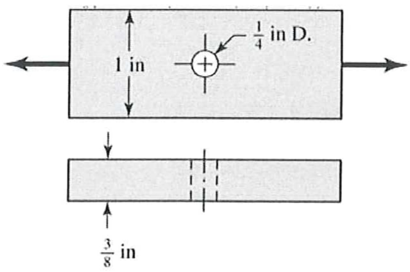
Homework #5

1. The cold-drawn AISI 1040 steel bar is subjected to an axial fluctuating load. Estimate the factors of safety for fatigue (based on the Goodman line), and for yielding for the following three loading conditions.

(a) 600 to 3200 lbf

(b) -600 to 3200 lbf

(c) 600 and -3200 lbf



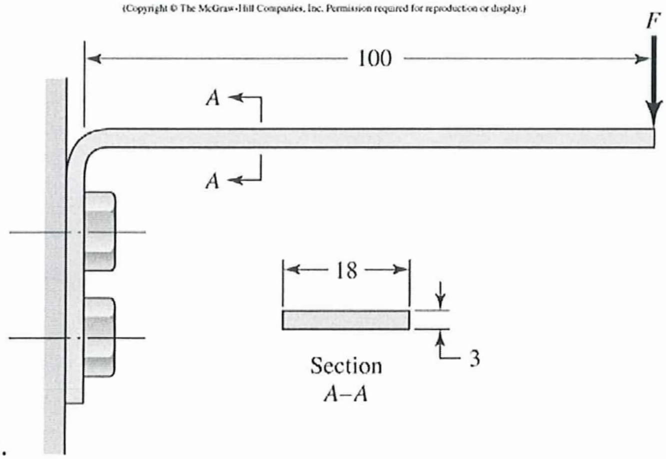
Partial Solutions: (a) $n_f = 1.98$; $n_y = 2.89$; (b) $n_f = 1.655$; $n_y = 2.89$; (c) $n_f = 2.137$; $n_y = 2.89$;

2. The figure shown below is a drawing of a 3- by 18-mm latching spring. A preload is obtained during assembly by shimming under the bolts to obtain an estimated initial deflection of 2 mm. The latching operation itself requires an additional deflection of exactly 4 mm, thus a total deflection of 6 mm. The material is ground high-carbon steel, bent then hardened and tempered to a minimum hardness of 490 Bhn (See equation 2-17 for relationship of hardness to strength). The radius of the bend is 3 mm. Estimate the yield strength to be 90 percent of the ultimate strength.

   (a) Find the maximum and minimum latching forces.

   (b) Find the fatigue factor of safety based on infinite life, using the modified Goodman criterion.

   (c) Find the static factor of safety.



Dimensions in millimeters.

Partial Solutions:

(a) $F_{min} = 50.3N$; $F_{max} = 150.9N$

(b) You need to use the curved beam equations on pages 112-115 to find the stresses in a curved beam:

Curved beam: $$r_n = \frac{h}{\ln(r_o/r_i)} = \frac{3}{\ln(6/3)} = 4.3281 \text{ mm}$$

$$r_c = 4.5 \text{ mm}, \quad e = r_c - r_n = 4.5 - 4.3281 = 0.1719 \text{ mm}$$

$$\sigma_i = -\frac{Mc_i}{Aer_i} - \frac{F}{A} = -\frac{(0.1015F)(1.5 - 0.1719)}{54(0.1719)(3)(10^{-3})} - \frac{F}{54} = -4.859F \text{ MPa}$$

$$\sigma_o = \frac{Mc_o}{Aer_o} - \frac{F}{A} = \frac{(0.1015F)(1.5 + 0.1719)}{54(0.1719)(6)(10^{-3})} - \frac{F}{54} = 3.028F \text{ MPa}$$

$$(\sigma_i)_{\min} = -4.859(150.9) = -733.2 \text{ MPa}$$

$$(\sigma_i)_{\max} = -4.859(50.3) = -244.4 \text{ MPa}$$

$$(\sigma_o)_{\max} = 3.028(150.9) = 456.9 \text{ MPa}$$

$$(\sigma_o)_{\min} = 3.028(50.3) = 152.3 \text{ MPa}$$

....

S_e =605 Mpa

Inner radius: $n_f = 2.48$; $n_y = 2.58$

Outer radius: $n_f = 2.65$; $n_y = 4.44$

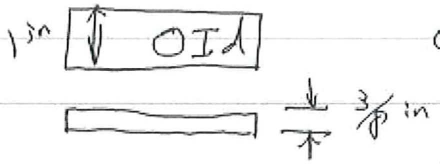
(c)

Inner radius: $n_y = 2.05$

Outer radius: $n_y = 3.29$

ME 208 HW #5

(1)



Given: $d = \frac{1}{4}$ in $\quad S_y = 71$ Kpsi

$S_{ut} = 85$ Kpsi

Find n_f, n_y

$S_e' = 0.5\, S_{ut} = 42.5$ Kpsi

$S_e = K_a K_b K_c K_d K_e K_f\, S_e'$

$K_a = a\, S_{ut}^{\,b}$ From 6-2 $\quad a = 2.70,\ b = -0.265$

$K_a = 2.7(85)^{-0.265} = 0.8319$

$K_b = 1$ for axial

$K_c = 0.85$ for axial

$K_d = 1$, $K_e = 1$

$K_f = 1 + q(K_t - 1)$ From 6-20, $q = 0.82$

From A-15-1 $\frac{d}{w} = \frac{0.25}{1}$, $K_t = 2.42$

$K_f = 1 + 0.82(2.4) - 1)$

$= 2.1644 \rightarrow$ multiply to σ

$K_f = 1$

$S_e = (0.8319)(1)(0.85)(1)(1)(42.5) = 30.05$ Kpsi

$\sigma = \frac{F}{A}$. $\quad A = (w-d)t = (1-0.25)\frac{3}{8} = 0.28125$

a) $\sigma_{min} = \dfrac{600}{0.28125} = 2.133$ Kpsi

$\sigma_{max} = \dfrac{3200}{0.28125} = 11.377$ Kpsi

$\sigma_m = \dfrac{\sigma_{max} + \sigma_{min}}{2} = \dfrac{11.377 + 2.133}{2} = 6.755$ Kpsi

After S.C. correction $\sigma_m = 6.755\, K_f = 6.755(2.1644)$

$= 14.62$

$$|\sigma_a| = \frac{\sigma_{max} - \sigma_{min}}{2} = 4.622 \text{ kpsi}.$$

After. S.C. correction $\sigma_a = (4.622)(2.1644)$

$$= 10.0$$

$$\frac{1}{n_f} = \frac{\sigma_a}{S_e} + \frac{\sigma_m}{S_{ut}} = \frac{4.622(2.1644)}{30.05} + \frac{14.62}{85}$$

$$= 0.5048 \Rightarrow n_f = 1.98$$

$$n_y = \frac{S_y}{|\sigma_{max}|} = \frac{71}{11.377(SC)} = \frac{6.24}{2.16} = 2.89$$

b)

$$\sigma_{min} = \frac{-600}{0.28125} = -2.133 \text{ kpsi}$$

$$\sigma_{max} = \frac{3200}{0.28125} = 11.377 \text{ kpsi}$$

After S.C. $\sigma_m = (2.1644)\frac{11.377 - 2.133}{2} = 10 \text{ kpsi}$

$$\sigma_a = (2.1644)\frac{11.377 - (-2.133)}{2} = 14.62 \text{ kpsi}.$$

$$\frac{1}{n_f} = \frac{\sigma_a}{S_e} + \frac{\sigma_m}{S_{ut}} = \frac{14.62}{30.05} + \frac{10}{85} = 0.604$$

$\therefore n_f = 1.655$

$$n_y = \frac{71}{11.377 \cdot 2.16} = \frac{6.24}{2.16} = 2.89$$

c) $\sigma_{min} = \dfrac{-3200}{0.28125} = -11.377\ Kpsi$

$\sigma_{max} = \dfrac{600}{0.28125} = 2.133\ Kpsi$

After S.C.

$\sigma_m = -10\ Kpsi$ (compression)

$\sigma_a = 14.62\ Kpsi$

$n_f = \dfrac{S_e}{\sigma_a} = \dfrac{30.05}{14.62} = 2.137$

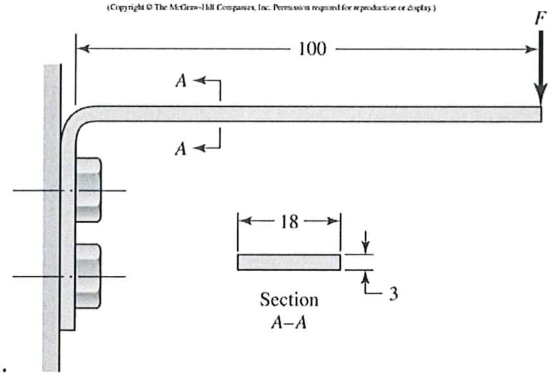
$n_y = \dfrac{71}{11.377 \cdot 2.16} \approx \dfrac{6.24}{2.16} = 2.89$

2. The figure is a drawing of a 3- by 18-mm latching spring. A preload is obtained during assembly by shimming under the bolts to obtain an estimated initial deflection of 2 mm. The latching operation itself requires an additional deflection of exactly 4 mm, thus a total deflection of 6 mm. The material is ground high-carbon steel, bent then hardened and tempered to a minimum hardness of 490 Bhn (See equation 2-17 for relationship of hardness to strength). The radius of the bend is 3 mm. Estimate the yield strength to be 90 percent of the ultimate strength.

   (a) Find the maximum and minimum latching forces.

   (b) Find the fatigue factor of safety based on infinite life, using the modified Goodman criterion.

   (c) Find the static factor of safety.



Dimensions in millimeters.

Solutions

(a)

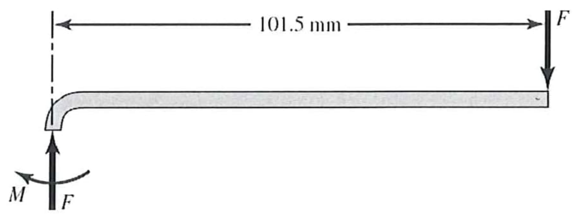
$$I = \frac{1}{12}(18)(3^3) = 40.5 \text{ mm}^4$$

$$y = \frac{Fl^3}{3EI} \quad \Rightarrow \quad F = \frac{3EIy}{l^3}$$

$$F_{\min} = \frac{3(207)(10^9)(40.5)(10^{-12})(2)(10^{-3})}{(100^3)(10^{-9})} = 50.3 \text{ N}$$

$$F_{\max} = \frac{6}{2}(50.3) = 150.9 \text{ N} \quad Ans.$$

(b)



$$M = 0.1015F \text{ N} \cdot \text{m}$$

$$A = 3(18) = 54 \text{ mm}^2$$

Curved beam: $$r_n = \frac{h}{\ln(r_o/r_i)} = \frac{3}{\ln(6/3)} = 4.3281 \text{ mm}$$

$$r_c = 4.5 \text{ mm}, \quad e = r_c - r_n = 4.5 - 4.3281 = 0.1719 \text{ mm}$$

$$\sigma_i = -\frac{Mc_i}{Aer_i} - \frac{F}{A} = -\frac{(0.1015F)(1.5 - 0.1719)}{54(0.1719)(3)(10^{-3})} - \frac{F}{54} = -4.859F \text{ MPa}$$

$$\sigma_o = \frac{Mc_o}{Aer_o} - \frac{F}{A} = \frac{(0.1015F)(1.5 + 0.1719)}{54(0.1719)(6)(10^{-3})} - \frac{F}{54} = 3.028F \text{ MPa}$$

$$(\sigma_i)_{\min} = -4.859(150.9) = -733.2 \text{ MPa}$$

$$(\sigma_i)_{\max} = -4.859(50.3) = -244.4 \text{ MPa}$$

$$(\sigma_o)_{\max} = 3.028(150.9) = 456.9 \text{ MPa}$$

$$(\sigma_o)_{\min} = 3.028(50.3) = 152.3 \text{ MPa}$$

Eq. (2-17) $$S_{ut} = 3.41(490) = 1671 \text{ MPa}$$

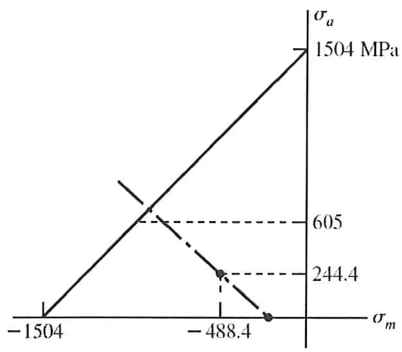
Per the problem statement, estimate the yield as $S_y = 0.9S_{ut} = 0.9(1671) = 1504$ MPa. Then from Eq. (6-8), $S'_e = 700$ MPa; Eq. (6-19), $k_a = 1.58(1671)^{-0.085} = 0.841$; Eq. (6-25) $d_e = 0.808[18(3)]^{1/2} = 5.938$ mm; and Eq. (6-20), $k_b = (5.938/7.62)^{-0.107} = 1.027$.

$$S_e = 0.841(1.027)(700) = 605 \text{ MPa}$$

At Inner Radius

$$(\sigma_i)_a = \left|\frac{-733.2 + 244.4}{2}\right| = 244.4 \text{ MPa}$$

$$(\sigma_i)_m = \frac{-733.2 - 244.4}{2} = -488.8 \text{ MPa}$$



Load line: $\sigma_m = -244.4 - \sigma_a$

Langer (yield) line: $\sigma_m = \sigma_a - 1504 = -244.4 - \sigma_a$

Intersection: $\sigma_a = 629.8$ MPa, $\sigma_m = -874.2$ MPa

(Note that σ_a is more than 605 MPa)

Yield: $$n_y = \frac{629.8}{244.4} = 2.58$$

Fatigue: $n_f = \dfrac{605}{244.4} = 2.48$ Thus, the spring is likely to fail in fatigue at the inner radius. *Ans.*

At Outer Radius

$$(\sigma_o)_a = \frac{456.9 - 152.3}{2} = 152.3 \text{ MPa}$$

$$(\sigma_o)_m = \frac{456.9 + 152.3}{2} = 304.6 \text{ MPa}$$

Yield load line: $\sigma_m = 152.3 + \sigma_a$

Langer line: $\sigma_m = 1504 - \sigma_a = 152.3 + \sigma_a$

Intersection: $\sigma_a = 675.9 \text{ MPa}, \quad \sigma_m = 828.2 \text{ MPa}$

$$n_y = \frac{675.9}{152.3} = 4.44$$

Fatigue line: $\sigma_a = \left(1 - \dfrac{\sigma_m}{S_{ut}}\right) S_e = \sigma_m - 152.3$

$$605\left(1 - \frac{\sigma_m}{1671}\right) = \sigma_m - 152.3$$

$$1.362\sigma_m = 757.3 \quad \Rightarrow \quad \sigma_m = 556.0 \text{ MPa}$$

$$\sigma_a = 556.0 - 152.3 = 403.7 \text{ MPa}$$

$$n_f = \frac{403.7}{152.3} = 2.65 \quad \textit{Ans.}$$

c) Statics

Inner:

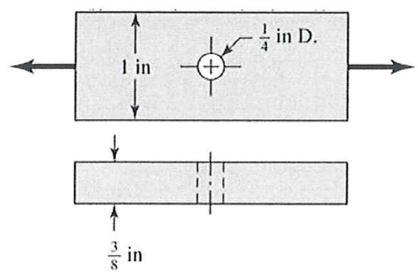
$$n_y = \frac{S_y}{|\sigma_{max}|} = \frac{1503.8}{733.2} = 2.05$$

Outer:

$$n_y = \frac{S_y}{|\sigma_{max}|} = \frac{1503.8}{456.9} = 3.29$$

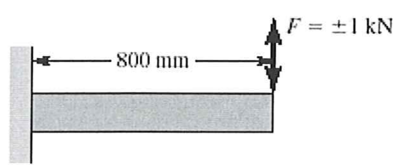
Homework #4
ME 208

1. The cold-drawn AISI 1040 steel bar shown in the figure is subjected to a completely reversed axial load fluctuating between 6200 lbs in compression to 6200 lbs in tension. Estimate the fatigue factor of safety based on achieving infinite life. If infinite life is not predicted, estimate the number of cycles to failure.



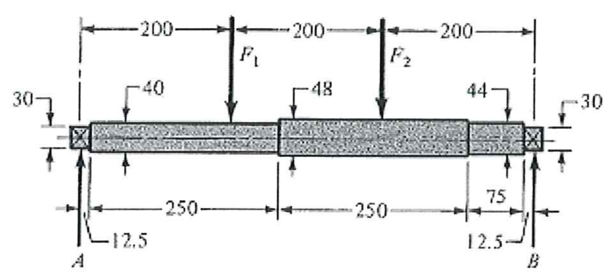
Partial Solutions: Se = 30.05; N ~ 27,000 cycles (this number may vary widely)

2. A solid round rod is cantilevered at one end. The rod is 0.8 m long and supports a completely reversing transverse load at the other end of ±1 kN. The material is AISI 1045 hot-rolled steel. Neglect any stress concentrations at the support end. If the rod must support this load for 10^4 cycles with a factor of safety of 1.5, what diameter should the rod be?



Partial Solutions: D ~ 32.9 mm

3. The steel shaft shown in the figure has a machined finish and a minimum tensile strength of 500 MPa and yield strength of 410 MPa. Bearing reactions are located at A and B. The bending forces are $F_1 = 3.0$ kN and $F_2 = 4.5$ kN. All fillets have 1.2 mm radius. Estimate the fatigue factor of safety based on achieving infinite life. If infinite life is not predicted, estimate the number of cycles to failure.



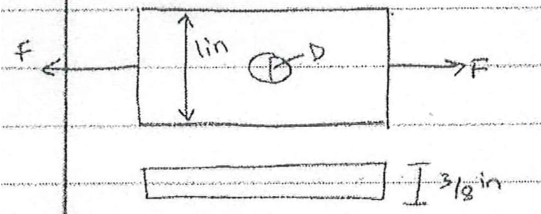
Partial Solutions: Se = 181.5 Mpa; N = 284,000 cycles

ME 208 Homework #4

(1)

Given: $D = 1/4$ $F = \pm 6200$ lbs

$S_{ut} = 85$ kpsi



Find: n_f or N

$\sigma_a = \frac{F}{A}$ $A = (w - D)t = (1 - .25)(3/8)$ $A = .2813 \text{ in}^2$

$\sigma_a = \frac{6200 \text{ lbs}}{.2813 \text{ in}^2}$ $\sigma_a = 22040.53$ psi $= 22.041$ kpsi

$S_e = k_a k_b k_c k_d k_e k_f S_e'$ $S_e' = .5 S_{ut}$ (6-8)

$S_e' = .5(85) = 42.5$ kpsi

$k_a = a S_{ut}^{\,b}$ From 6-2 $a = 2.70$ $b = -.265$

$k_a = 2.70(85)^{-.265}$ $k_a = .8319$

$k_b = 1$ for axial

$k_c = .85$ for axial

$k_d = 1$

$k_e = 1$

$S_e = (0.8319)(1)(0.85)(1)(1)(42.5) = 30.05$ kpsi

$K_f = 1 + q(K_t - 1)$ From 6-20, $q = 0.83$

From A-15-1, $d/w = 0.25/1 = 0.25 \Rightarrow K_t = 2.42$

$K_f = 1 + 0.83(2.42 - 1) = 2.1786$

$\sigma_a = (22.04)(2.1786) = 48.02$

$n_f = S_e / \sigma_a = 30.05 / 48.02 = 0.6257 \Leftarrow$ will fail

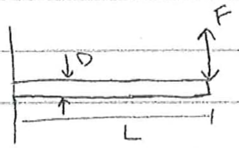
$N = \left(\sigma_a / a\right)^{1/b}$ From 6-18 $f = 0.868$

$a = (f S_{ut})^2 / S_e = [0.868(85)]^2 / 30.05 = 181.15$

$b = -\frac{1}{3} \log\left(\frac{f S_{ut}}{S_e}\right) = -\frac{1}{3} \log\left(\frac{(0.868)(85)}{30.05}\right) = -0.13$

$N = \left(48.02 / 181.15\right)^{\frac{1}{-0.13}} = 27,259$ cycles

(2)



Given: $L = .8\,m$, $F = \pm 1\,kN$, $n_f = 1.5$

$N = 10^4$ cycles, $S_{ut} = 570$ MPa, $S_y = 310$ MPa

Find: D

$\sigma = \frac{My}{I}$ $M = FL = 1000(800\,mm)$ $M = 800{,}000$ N·M $y = \frac{D}{2}$

$I = \frac{\pi}{64} D^4$

$\sigma = \frac{8\times10^5 (\frac{D}{2})}{\frac{\pi}{64} D^4} = \frac{4\times10^5 D}{\frac{\pi}{64} D^4} = \frac{8.149\times10^6}{D^3}$

Use Yielding to estimate diameter

$\sigma_{max} = \frac{S_y}{FS} = \frac{310}{1.5} = 206.7$ $206.7 = \frac{8.149\times10^6}{D^3}$ $D = 34.03$ mm

$S_e = k_a k_b k_c k_d k_e k_f S_e'$ $S_e' = .5 S_{ut} = .5(570\,MPa) = 285$ MPa

$k_a = a S_{ut}^b$ From 6-2 $a = 57.7$ $b = -.718$

$k_a = 57.7(570)^{-.718}$ $\underline{k_a = .606}$

$k_b = 1.24\, d_{eq}^{-.107}$ $d_{eq} = .37D = .37(34.03)$ $d_{eq} = 12.59$ mm

$k_b = 1.24(12.59)^{-.107}$ $\underline{k_b = .9456}$

$\underline{k_c = 1}$ for bending

$k_d = 1$ $k_e = 1$ $k_f = 1$

$S_e = (.606)(.9456)(1)(285\,MPa)$ $S_e = 163.3$ MPa

$N = \left(\frac{1.5\sigma_a}{a}\right)^{1/b}$ $a = \frac{(f S_{ut})^2}{S_e}$ From 6-18 $f = .875$ $a = \frac{((.875)(570))^2}{163.3} = 1523.3$

$b = -\frac{1}{3}\log\left(\frac{f S_{ut}}{S_e}\right) = -\frac{1}{3}\log\left(\frac{(.875)(570)}{163.3}\right)$ $b = -.1616$

$10^4 = \left(\frac{1.5\sigma_a}{1523.3}\right)^{\frac{1}{-.1616}}$ $\sigma_a = 229.2$ MPa

$\sigma = \sigma_a \rightarrow 229.2 = \frac{8.149\times10^6}{D^3}$ $\boxed{D = 32.9\text{ mm}}$

Check k_b $k_b = 1.24\, d_{eq}^{-.107}$ $d_{eq} = .37D = .37(32.9) = 12.173$

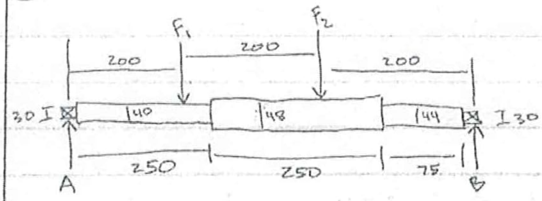
$k_b = 1.24(12.173)^{-.107}$ $k_b = .9490 \approx .9456$

(3)

Given: $S_{ut} = 500 MPa$ $S_y = 410 MPa$

$F_1 = 3.0 kN$ $F_2 = 4.5 kN$

$r = .0012 m$



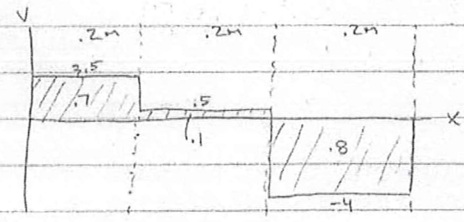
Find: FS or N

$\Sigma F_y = 0$ $\Sigma F_y = A + B - F_1 - F_2 = 0$ $A + B = 7.5 kN$

$\Sigma M_A = 0$ $\Sigma M_A = -F_1(.2m) - F_2(.4m) + B(.6m) = 0$

$= -3.0(.2) - 4.5(.4) + B(.6) = 0$ $B = 4 kN$

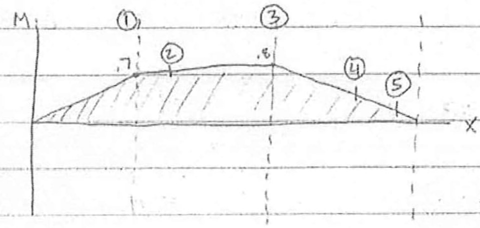
$A + 4 = 7.5$ $A = 3.5 kN$



$M_{max} = .8 kN \cdot m = 800 N \cdot m$ at F_2

$\sigma = \frac{My}{I} = \frac{M_{max}(\frac{D}{2})}{\frac{\pi}{64}D^4}$

(3) $\sigma = \frac{800(\frac{.048}{2})}{\frac{\pi}{64}(.048)^4}$ $\sigma_3 = 73.68 MPa$



(1) $\sigma = \frac{700(\frac{.040}{2})}{\frac{\pi}{64}(.040)^4}$ $\sigma_1 = 111.41 MPa$

* Neglect stress near A

(2) $\frac{800-700}{200} = \frac{M-700}{62.5}$ $M = 731.25 N \cdot m$ $\frac{r}{d} = \frac{.0012m}{.04m} = .03$

$\frac{D}{d} = \frac{.048}{.04} = 1.2$ From A-15-9 $K_t = 2.2$

From 6-20, $q = 0.7$ $K_f = 1 + q(K_t - 1) = 1 + 0.7(2.2$

$= 1.84$

$\sigma_2 = \frac{1.84(731.25)(\frac{0.04}{2})}{\frac{\pi}{64}(0.04)^4} = 214 Mpa$

(4) $\frac{800}{200} = \frac{M}{87.6}$ $M = 350 N \cdot m$ $\frac{r}{d} = \frac{0.0012}{0.044} = 0.0$

$D/d = 0.048/0.044 = 1.091$ $A-15-9 \Rightarrow K_t =$

$K_f = 1 + 0.7(1.97 - 1) = 1.679$

$\sigma_4 = \frac{1.679(350)(\frac{0.044}{2})}{\frac{\pi}{64}(0.044)^4} = 70.27 Mpa$

(5) $\frac{800}{200} = \frac{M}{12.5}$ $M = 50\ N \cdot m$ $\frac{r}{d} = \frac{.0012}{.03} = .04$ $\frac{D}{d} = \frac{.044}{.03} = 1.467$

From A-15-9 $K_t = 1.98$ $K_f = 1 + 0.7(1.98-1)$
$= 1.686$

$\sigma_5 = \frac{1.686(50)(\frac{.03}{2})}{\frac{\pi}{64}(.03)^4}$ $\sigma_5 = 31.8\ MPa$

$S_e' = .5 S_{ut}$ $S_e' = .5(500) = 250\ MPa$

$S_e = k_a k_b k_c k_d k_e k_f S_e'$

$k_a = a S_{ut}^{\ b}$ From 6-2 $a = 4.51$ $b = -.265$

$k_a = 4.51(500)^{-.265}$ $\underline{k_a = .8689}$

$k_b = 1.24\, d_{eq}^{-.107}$ $d_{eq} = d = 40\ mm$

$k_b = 1.24(40)^{-.107}$ $\underline{k_b = .8356}$

$k_c = 1$ for bending

$k_d = 1$ $k_e = 1$ $k_f = 1$

$S_e = (0.8689)(0.8356)250 = 181.51\ MPa$

$F_S = S_e/\sigma = 181.51/214 = 0.848 \rightarrow$ will fail

$N = \left(\frac{\sigma}{a}\right)^{1/b}$ $a = \frac{(f S_{ut})^2}{S_e}$

From Fig. 6-18, $f = 0.895$ $a = \frac{((0.895)(500))^2}{181.51}$

$a = 1103.3$ $b = -\frac{1}{3}\log\left(\frac{f S_{ut}}{S_e}\right)$

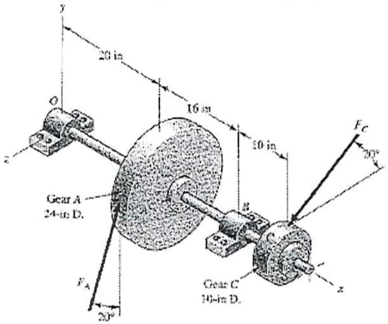
$= -\frac{1}{3}\log\left(\frac{(0.895)(500)}{181.51}\right)$

$= -0.1306$

$N = \left(\frac{214}{1103.3}\right)^{\frac{1}{-0.1306}} = 284,383$ cycles

ME 208
Homework #1

1. A gear reduction unit uses the countershaft shown in the figure. When the shaft is rotating at constant operating speed, the force transmitted to Gear A is $F_A = 300$ lbf.
   a) Find the magnitudes of the bearing reactions.
   b) Draw shear and moment diagrams for the shaft. Make one set for the horizontal plane and another set for the vertical plane.
   c) Considering the moment diagrams from both planes, find the location and magnitude of the maximum bending moment in the shaft.



300

Partial Solutions: Max Bending occurs at point B: 7200in-lb

$\Sigma M_{B,Y'} = 0$

$-F_A \sin 20 \, (16'') + F_{O,Z} \, (36'')$

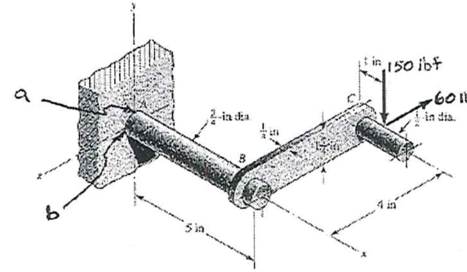
$F_{O,Z} = 45$ lbf

$\Sigma M_{B,Z'} = 0$

$-F_A \cos 20° \, (16) + F_{O,Y} (36) = 0$

$F_{O,Y} = 125.29$ lbf

2. Analyze the stress situation in beam AB by obtaining the following information.
   a) Compute all stresses acting at points *a* (top), and *b* (front) on the surface of the beam at the wall. (Transverse shear may be neglected if you can justify this decision.) Sketch the stress elements indicating the magnitudes and directions of all stresses.
   b) Determine the location of the critical stress element. (Not necessarily at *a* or *b*.) Sketch the critical stress element if it is not at *a* or *b*.
   c) For the critical stress element, find the principal stresses and the maximum shear stress. You may want to use Mohr's circle.



Partial Solutions:
(b) location angle of critical stress element = 111.8°
(c) τ_{max} = 13759 psi

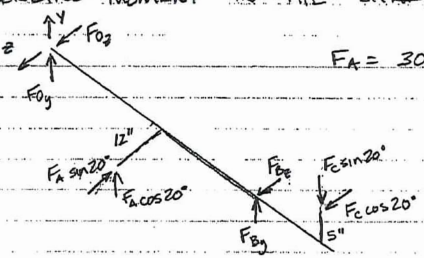
HW1

①

(a) FIND THE MAGNITUDE OF THE BEARING REACTIONS

(b) DRAW SHEAR AND MOMENT DIAGRAMS FOR THE SHAFT. MAKE ONE SET FOR THE HORIZONTAL PLANE AND ANOTHER SET FOR THE VERTICAL PLANE.

(c) FIND THE LOCATION AND MAGNITUDE OF THE MAX. BENDING MOMENT IN THE SHAFT.



$F_A = 300$ lbf.

(a)

$\Sigma M_x = 0$

$-F_A \cos 20°(12") + F_C \cos 20°(5") = 0$

$F_C = 720$ lbf.

$\Sigma M_{O_y} = 0$

$F_A \sin 20°(20") - F_{B_z}(36") - F_C \cos 20°(46") = 0$

$F_{B_z} = -807.5$ lbf.

ΣM_{O_z}

$F_A \cos 20°(20") + F_{B_y}(36") - F_C \sin 20°(46") = 0$

$F_{B_y} = 158.0$ lbf.

$\Sigma F_y = 0$

$F_{Oy} + F_A \cos 20° + F_{By} - F_C \sin 20° = 0$

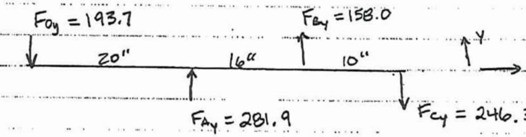
$F_{Oy} = -193.7$ lbf.

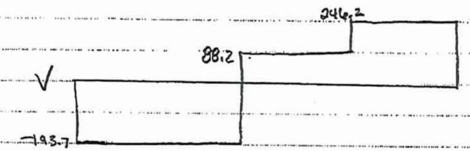
$\Sigma F_z = 0$

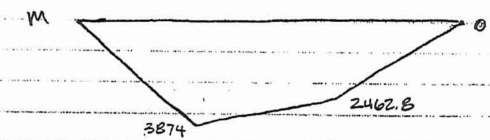
$F_{Oz} - F_A \sin 20° + F_{Bz} + F_C \cos 20° = 0$

$F_{Oz} = 233.5$ lbf.

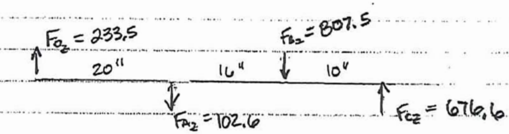
(b) x-y plane

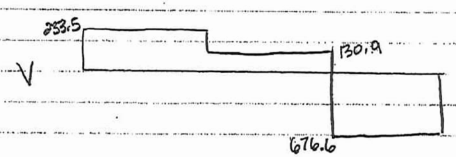


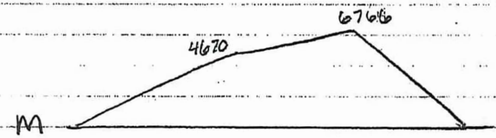




(c) x-z plane







(d) CHECK POINTS A + B FOR MAXIMUM BENDING MOMENT

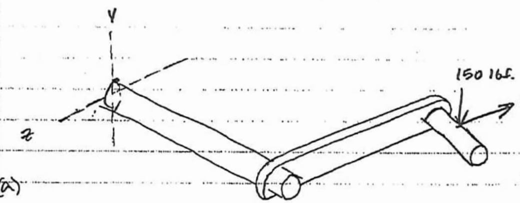
$$M_A = \sqrt{(3874)^2 + (4670)^2} = 6067.7 \text{ in-lb}$$

$$M_B = \sqrt{(2462.8)^2 + (6766)^2} = 7200.3 \text{ in-lb}$$

NOTE: MAXIMUM BENDING COULD OCCUR AT POINTS OTHER THAN A + B.

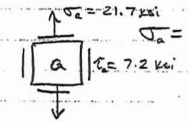
$$\Rightarrow \boxed{M_{MAX} = 7200 \text{ in-lb}} \text{ @ point B}$$

② (a) COMPUTE ALL STRESSES AT a AND b AT THE WALL. SKETCH THE STRESS ELEMENTS.
(b) DETERMINE THE LOCATION OF THE CRITICAL STRESS ELEMENT.
(c) FIND PRINCIPAL STRESSES + MAXIMUM SHEAR STRESS ON THE CRITICAL STRESS ELEMENT.



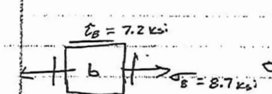
NOTE: TRANSVERSE SHEAR IS NEGLECTED. See note at end.

(a)



$$\sigma_a = \frac{My}{I} = \frac{(150)(6")(3/8)}{\frac{\pi(3/4")^4}{64}} = 21730 \text{ psi}$$

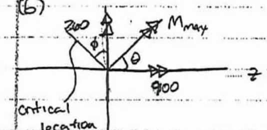
$$\tau_a = \frac{Tr}{J} = \frac{(150)(4")(3/8")}{\frac{\pi(3/4)^4}{32}} = 7243 \text{ psi}$$



$$\sigma_B = \frac{My}{I} = \frac{(60)(6")(3/8")}{\frac{\pi(3/4")^4}{64}} = 8692 \text{ psi}$$

$$\tau_B = \frac{Tr}{J} = \tau_A = 7243 \text{ psi}$$

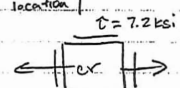
(b)



$$M_{max} = \sqrt{900^2 + 360^2} = 969 \text{ in-lbf.}$$

$$\theta = \tan^{-1}\left(\frac{360}{900}\right) = 21.8°$$

$$\phi = 90 + \theta = 111.8°$$



$$\sigma = \frac{(969)(3/8")}{\frac{\pi(3/4)^4}{64}} = 23.4 \text{ ksi}$$

(c)
$$\sigma_1, \sigma_2 = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

$$= \frac{23396 + 0}{2} \pm \sqrt{\left(\frac{23396 + 0}{2}\right)^2 + (7243)^2}$$

$$\boxed{\sigma_1 = 25457 \text{ psi} \quad \sigma_2 = -2061 \text{ psi} \quad \sigma_3 = 0}$$

$$\tau_{1/2} = \frac{\sigma_1 - \sigma_2}{2} = \frac{25457 + 2061}{2} = 13759 \text{ psi}$$

$$\tau_{2/3} = \frac{\sigma_2 - \sigma_3}{2} = \frac{-2061 + 0}{2} = -1031 \text{ psi}$$

$$\tau_{1/3} = \frac{\sigma_1 - \sigma_3}{2} = \frac{25457 - 0}{2} = 12729 \text{ psi}$$

$$\boxed{\tau_{max} = \tau_{1/2} = 13759 \text{ psi}}$$

(d)
$$\sigma_{oct} = \frac{1}{3}(\sigma_1 + \sigma_2 + \sigma_3)$$

$$= 1/3\,(25457 + (-2061) + 0)$$

$$\boxed{\sigma_{oct} = 7798.7 \text{ psi}}$$

$$\tau_{oct} = 2/3\left[\tau_{1/2}^2 + \tau_{2/3}^2 + \tau_{1/3}^2\right]^{1/2}$$

$$= 2/3\left[(13759)^2 + (-1031)^2 + (12729)^2\right]^{1/2}$$

$$\boxed{\tau_{oct} = 12514.9 \text{ psi}}$$

NOTE ABOUT TRANSVERSE SHEAR.

The transverse shear is often neglected for problems that are clearly dominated by bending and/or torsion. In this problem the length to height ratio is $6/(3/4) = 8$. A ratio over 4 generally indicates a small effect from transverse shear. A ratio over 10 generally indicates negligible effect, particularly in a design problem.

If transverse shear were to be included, at point "a" the transverse shear due to the vertical 150 lbf is zero, since the profile of $\frac{VQ}{It}$ is zero at the extremes from the neutral axis & maximum at the neutral axis (See Fig 3-20). The transverse shear at point "a" due to the horizontal 60 lbf is at its maximum value since it is on the neutral axis for the horizontal shear. Rather than making the cumbersome $\frac{VQ}{It}$ calculation, the maximum value can be obtained from Table 3-2 as $\frac{4V}{3A}$. Thus, at point "a", the transverse shear is $\tau_{trans} = \frac{4}{3}\frac{V}{A} = \frac{4}{3}\left(\frac{60}{\pi(3/8)^2}\right) = 181$ psi

This transverse shear is about 2.5% of the torsional shear, and could be added to the torsional shear to give a total shear at point "a" of $\tau = 7424$ psi.

At point "b", the transverse shear due to the 150 lbf vertical force is at its maximum value of

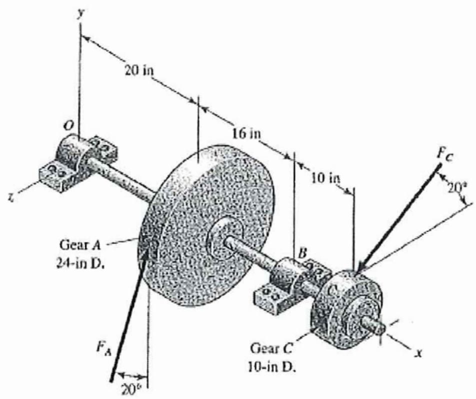
$$\tau_{trans} = \frac{4}{3}\frac{V}{A} = \left(\frac{4}{3}\right)\frac{150}{\pi(3/8)^2} = 453 \text{ psi}$$

This transverse shear is in the opposite direction from the torsional shear at point "b", since the transverse shear is trying to shear the element from the wall in the negative y direction, and the torsional shear is trying to shear the beam from the wall in the positive y direction. Thus, at point "b", the total shear is $\tau = 7243 - 453 = 6790$ psi.

At the critical stress element, neither the horizontal or vertical shear are at their maximum or minimum. Each one will have a magnitude determined by VQ/It. When the two components are calculated, it will be found that they sum to zero. This is because the total transverse shear is due to the vector addition of the 150 lbf + 60 lbf, which is at an angle 21.8° from the vertical. Thus, the critical stress element is located where the transverse shear is zero.

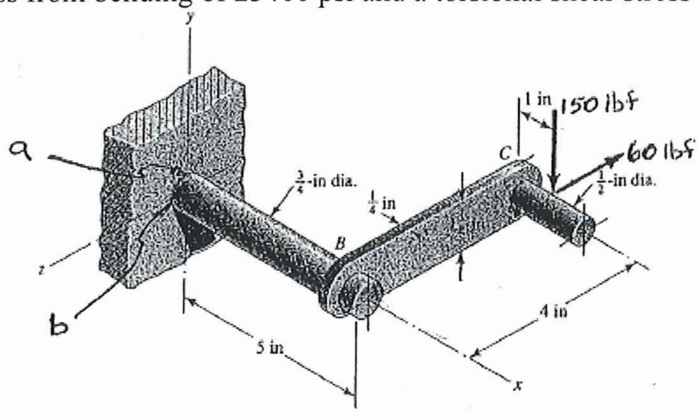
Homework 2
ME 208

1. For the gear-reduction unit from Homework 1, assuming a constant diameter shaft, use the Maximum Shear Stress failure theory to specify an appropriate shaft diameter to provide a yielding factor of safety of 3.5, using 1010 cold drawn steel. (Note: From Homework 1, you should have found the critical stress location at point B, with total bending moment of 7200 lbf-in, and torque of 3383 lbf-in.)



Partial Solutions: d = 1.86"

2. For the cantilevered beam AB from Homework 1, use the Distortion Energy failure theory to determine the yielding factor of safety at the critical stress element. Assume the material is 1010 cold drawn steel. (Note: From Homework 1, you should have found the critical stress element to be 21.8° CCW from point *a*, with a normal stress from bending of 23400 psi and a torsional shear stress of 7243 psi.)



Partial Solutions: $n_y = 1.66$

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Homework 2 ME 208

(1) From Hw # 1 $M_B = 7200 \text{ lb-in}$

$T_B = 3383 \text{ lb-in}$

Note: Bending stress at point B is much greater than transverse shear stress. Transverse shear stress is zero where bending stess is maximum + thus is neglected.

Bending

$$\sigma = \frac{My}{I} = \frac{(7200 \text{ lb-in})\left(\frac{d \text{ in}}{2}\right)}{\frac{\pi}{64}(d \text{ in})^4} = \frac{73{,}338.6 \text{ lb-in}}{(d \text{ in})^3} = \frac{73.3386 \text{ Kip-in}}{(d \text{ in})^3}$$

Torsion

$$\tau = \frac{Tr}{J} = \frac{(3383 \text{ lb-in})\left(\frac{d \text{ in}}{2}\right)}{\frac{\pi}{32}(d \text{ in})^4} = \frac{17{,}229.5 \text{ lb-in}}{(d \text{ in})^3} = \frac{17.2295 \text{ Kip-in}}{(d \text{ in})^3}$$

For 1010 CD steel $S_y = 44$ Ksi (From Table A-20 on pg. 1020)

MSS Theory $n = \frac{S_{sy}}{\tau_{max}} = \frac{S_y}{2\tau_{max}} = 3.5 \Rightarrow \tau_{max} = \frac{44 \text{ Ksi}}{2(3.5)} = 6.286 \text{ Ksi}$

$$\tau_{max} = \sqrt{\left(\frac{\sigma}{2}\right)^2 + \tau^2} = \sqrt{\left(\frac{73.3386 \text{ Kip-in}}{2(d \text{ in})^3}\right)^2 + \left(\frac{17.2295 \text{ Kip-in}}{(d \text{ in})^3}\right)^2}$$

$$(6.286 \text{ Ksi})^2 = \frac{1344.6 \text{ Kip}^2\text{-in}^2 + 296.9 \text{ Kip}^2\text{-in}^2}{(d \text{ in})^6} = \frac{1641.5 \text{ Kip}^2\text{-in}^2}{(d \text{ in})^6}$$

$$\boxed{d = \sqrt[6]{\frac{1641.5 \text{ Kip}^2\text{-in}^2}{(6.286 \text{ Ksi})^2}} = 1.86 \text{ in}}$$

(2)

From HW # 1 critical stress element is 21.8° CCW from point a

$\sigma = 23.4$ Ksi $\tau = 7.243$ Ksi

DET for biaxial stress $\sigma_1 = \sqrt{\sigma_x^2 + \sigma_y^2 - \sigma_x \sigma_y + 3\tau^2}$

$$\sigma_1 = \sqrt{(23.4\text{ Ksi})^2 + 3(7.243\text{ Ksi})^2} = 26.55\text{ Ksi}$$

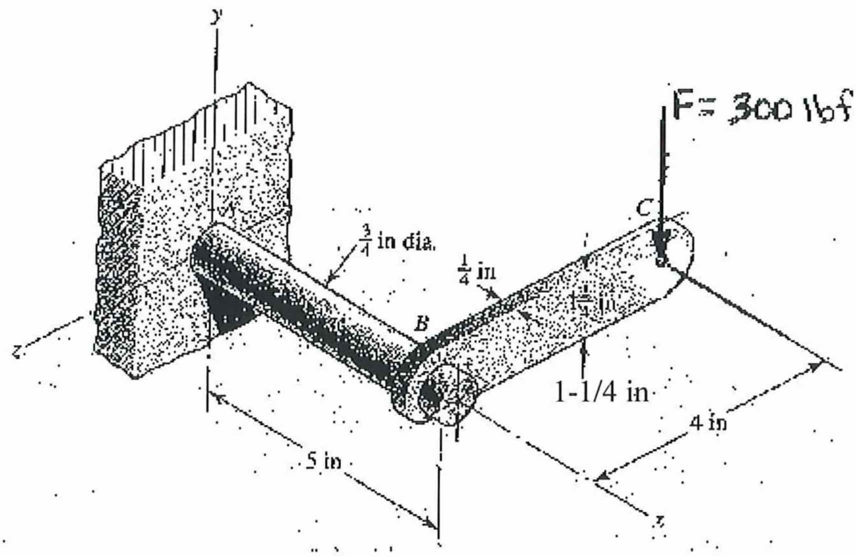
Table A-20 pg 1020

For 1010 CD steel $S_y = 44$ Ksi

$$\boxed{\eta = \frac{S_y}{\sigma_1} = \frac{44\text{ Ksi}}{26.55\text{ Ksi}} = 1.66}$$

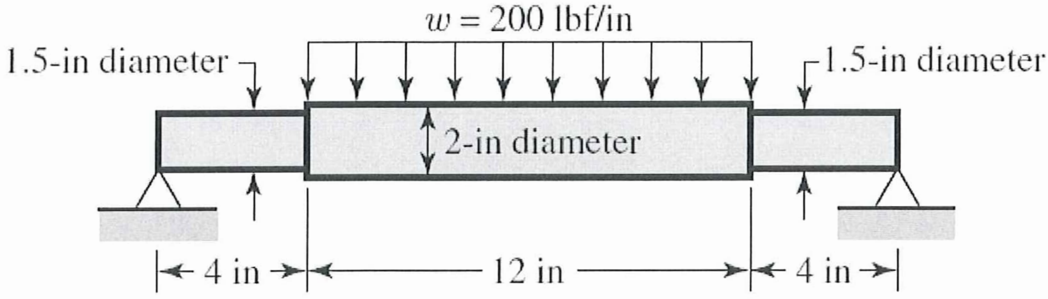
Homework #3
ME 208

1. The cantilevered handle is made from two pieces of mild steel welded together.
   (a) Determine the vertical deflection at point C using Castigliano's method.
   (b) Determine the vertical deflection at point C using traditional methods (other than Castigliano's method) learned in a mechanics of materials course.



Partial Solutions: $\delta = 0.1"$

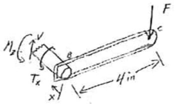
2. Determine the maximum deflection for the steel beam shown using Castigliano's theorem.



Partial Solutions: $\delta = -0.01673$ in

HW 3 Soln's 1 of 1

(1) a) deflection at C is due to bending + twisting of AB and bending of BC. (Neglect Shear)



Bending in AB

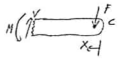
$M_z = Fx \qquad \frac{\partial M_z}{\partial F} = x$

$\delta = \frac{1}{EI}\int (M_z)\left(\frac{\partial M_z}{\partial F}\right)dx$

Twisting in AB

$T_x = 4F \qquad \frac{\partial T_x}{\partial F} = 4$

$\delta = \frac{1}{JG}\int (T_x)\left(\frac{\partial T_x}{\partial F}\right)dx$



Bending in BC

$M = Fx \qquad \frac{\delta M}{\partial F} = x$

$\delta = \frac{1}{EI}\int (M)\left(\frac{\partial M}{\partial F}\right)dx$

$\boxed{\begin{array}{l}\text{Mild steel}\\ E = 30\times10^6\ psi\\ G = 11.5\times10^6\ psi\end{array}}$

for AB

$I = \frac{\pi}{64}d^4 = 0.0155\ in^4$

$J = \frac{\pi}{32}d^4 = 0.0311\ in^4$

For BC

$I = \frac{bh^3}{12} = 0.0407\ in^4$

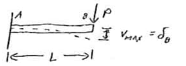
$$\delta_c = \frac{1}{EI_{AB}}\int_0^5 Fx(x)dx + \frac{1}{JG}\int_0^5 4F(4)dx + \frac{1}{EI_{BC}}\int_0^4 Fx(x)dx$$

$$\delta_c = \frac{300}{(30\times10^6)(0.0155)}\frac{1}{3}x^3\Big|_0^5 + \frac{(16)300}{(11.5\times10^6)(0.0311)}x\Big|_0^5 + \frac{300}{(30\times10^6)(0.0407)}\frac{1}{3}x^3\Big|_0^4$$

$$\delta_c = 0.0269 + 0.067 + 0.0052 = \boxed{\delta_c = 0.1\ in}$$

b.) Bending in AB

Find the deflection at B using beam tables:

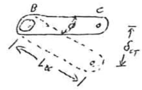


$\delta_B = \frac{FL_{AB}^3}{3EI_{AB}}$

Twisting in AB

Find deflection at C due to twisting of AB:

$\phi = \frac{TL_{AB}}{J_{AB}G}$



$\delta = L_{BC}\sin(\phi)$

for small ϕ,

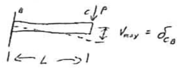
$\delta_{CT} \approx L_{BC}\phi$

$T = FL_{BC}$

$\delta_{CT} = \frac{(FL_{BC})L_{AB}}{J_{AB}G}L_{BC}$

Bending in BC

Find deflection at C due to bending of BC (beam tables):



$\delta_{CB} = \frac{FL_{BC}^3}{3EI_{BC}}$

$E = 30\times10^6\ psi$

$G = 11.5\times10^6\ psi$

$I_{AB} = 0.0155\ in^4 \qquad J_{AB} = 0.0311\ in^4$

$I_{BC} = 0.0407\ in^4$

$$\delta_c = \delta_B + \delta_{CT} + \delta_{CB} = \frac{FL_{AB}^3}{3EI_{AB}} + \frac{FL_{AB}L_{BC}^2}{J_{AB}G} + \frac{FL_{BC}^3}{3EI_{BC}}$$

$\boxed{\delta_c = 0.1\ in}$

2.

$I_1 = \frac{\pi}{64}(1.5^4) = 0.2485\ \text{in}^4 \qquad I_2 = \frac{\pi}{64}(2^4) = 0.7854\ \text{in}^4$

$R_1 = \frac{200}{2}(12) = 1200\ \text{lbf}$

For a dummy load $\uparrow Q$ at the center

$0 \le x \le 10\ \text{in} \qquad M = 1200x - \frac{Q}{2}x - \frac{200}{2}(x-4)^2, \qquad \frac{\partial M}{\partial Q} = \frac{-x}{2}$

$$y|_{x=10} = \frac{\partial U}{\partial Q}\Big|_{Q=0}$$

$$= \frac{2}{E}\left\{\frac{1}{I_1}\int_0^4 (1200x)\left(-\frac{x}{2}\right)dx + \frac{1}{I_2}\int_4^{10}[1200x - 100(x-4)^2]\left(-\frac{x}{2}\right)dx\right\}$$

$$= \frac{2}{E}\left[-\frac{200(4^3)}{I_1} - \frac{1.566(10^5)}{I_2}\right]$$

$$= -\frac{2}{30(10^6)}\left(\frac{1.28(10^4)}{0.2485} + \frac{1.566(10^5)}{0.7854}\right)$$

$= -0.01673$ in *Ans.*