ME-340 EXAM #02 *November 12, 1998*

*NAME:*________________________ *10:00 AM*

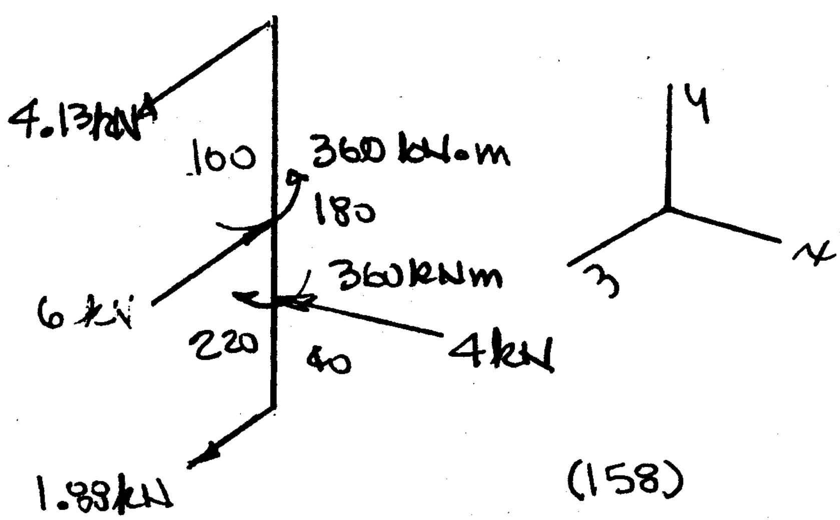
- *Read all problems carefully. Check the board for any additions or corrections.*
- *Show all work, which must be neat and orderly to be graded. That is, sloppy work will not be graded!*
- *Show no work on this page. Return this page with your solutions.*

(1) A sliding element bearing has a length of 25-mm, an L/d ratio of unity, and a clearance of 0.02-mm. The bearing must support a load of 1,250-N, corresponding to a rotational speed of 60- rev/second. The lubricant is SAE 40 oil with a supply temperature of 35C. The design engineer assumed a viscosity of 50mPa-s. Do you agree with that assumption (yes or no) and show the calculations required to support your decision. If you do not agree what should be done to correct the design engineer's error? Using the original data, determine the temperature rise across the bearing, the outlet oil temperature, coefficient of friction, eccentricity, total and side oil flows, maximum oil pressure, and the power lost in the bearing.

(2) For the design of problem (1) select a 200 series rolling element bearing which has a reliability of 95% and must last for 25,000 hours. The application is moderate impact.

(3) For the figure shown below, verify the free body diagram is correct.

(4) A bearing has been operated at 2,500,000 cycles at 5,000 pounds. Can this bearing be operated at 8,333 pounds for an additional 250,000 cycles? The bearing is an 02 series with a bore diameter of 80-mm.



(1) $l = 25mm$

$\frac{l}{d} = 1 \rightarrow \; = 25mm$

$W = 1250N$

$c = 0.02mm$

$N = 60 \, rev/sec$

SAE 40 oil

$T_i = 35°C$

$T_{AVG, ASSUMED} = 54°C \quad (\mu = 50 \, mPa \, sec)$

$\frac{r}{c} = \frac{12.5mm}{0.02mm} = 625$

$\rightarrow P = \frac{1250N}{.025m \times .025m} = 2 \times 10^6 \, N/m^2$

ASSUMED TEMPERATURE RISE

$\Delta T_{ASSUMED} = 2(T_{AVG,ASSUMED} - T_i) = 38°C$

$T_{out,ASSUMED} = T_i + \Delta T_{ASSUMED} = 73°C$

FIRST CALCULATE THE SOMMERFIELD NUMBER AT THE ASSUMED T_{AVG}

$$S = \left(\frac{r}{c}\right)^2 \left(\frac{\mu N}{P}\right) = (625)^2 \left(\frac{50 \times 10^{-3} \, Pa \, Sec \, (60 \, rev/sec)}{2 \times 10^6 \, Pa}\right) = 0.5859$$

NOW CALCULATE THE TEMPERATURE RISE AND SEE IF IT MATCHES

$$\Delta T_c = \frac{8.30 P}{\left(1 - .5\left(\frac{Q_s}{Q}\right)\right)} \frac{\left(\frac{r}{c}f\right)}{\left(\frac{Q}{rcNl}\right)} = \frac{8.30(2 \, MPa)}{1 - .5(0.3)} \frac{11.}{3.6} = 60°C$$

SINCE THE CALCULATED TEMPERATURE RISE IS MORE THAN 5% OFF THE ENGINEER'S ASSUMPTION IS INCORRECT. A NEW TEMPERATURE MUST BE GUESSED, AND THEN CHECKED VIA CALCULATION. SUGGEST NEXT GUESS AT $\Delta T_{ASSUMED} = 50°C$

USING THE ORIGINAL DATA (ASSUMPTIONS) FIND THE TEMPERATURE RISE ACROSS THE BEARING, THE OUTLET TEMPERATURE, COEFFICENT OF FRICTION, ECCENTRICITY, TOTAL AND SIDE OIL FLOWS, MAXIMUM OIL PRESSURE AND THE POWER LOST IN THE BEARING.

FOR THE CALCULATED TEMP RISE SEE ABOVE (60°C)

ASSUMING $\Delta T = 60°C$, $T_{out} = T_i + \Delta T = 35°C + 60°C = 95°C$

① CONT'D

FIND COEFFICENT OF FRICTION (USE S FROM ORIGINAL ASSUMPTION) (CHART)

$$\frac{r}{c}f = 11 \; ; \; f = 11\frac{c}{r} = \frac{11}{625} = 0.0176$$

[SOMMERFIELD # = 0.5859]

FIND ECCENTRICITY

$$\varepsilon = 0.22 \rightarrow e = \varepsilon c = 0.22(0.02\,mm) = 0.0044\,mm$$

FIND TOTAL & SIDE OIL FLOWS.

$$\frac{Q}{rCNl} = 3.6 \; ; \; Q = 3.6(12.5mm)(0.02mm)60\,\tfrac{rev}{sec}(25mm) = 1350\,mm^3/sec$$
$$= 1.35\,cc/sec = 1.35 \times 10^{-6}\,m^3/sec$$

$$\frac{Q_s}{Q} = 0.3 \; ; \; Q_s = 0.3(1.35\,cc/sec) = 0.405\,cc/sec$$

FIND MAXIMUM OIL PRESSURE

$$\frac{W}{ld} \approx \text{NOMINAL PRESSURE} = 2 \times 10^6\,Pa = P = 290\,psi$$

$$\frac{P}{P_{max}} = 0.52 \; ; \; P_{max} = \frac{P}{0.52} = \frac{290psi}{0.52} = 558\,psig = 573\,psi$$
$$= 4Mpa$$

FIND POWER LOST IN BEARING

$$T = fWr = 0.0176(1250\,N)(.0125m) = 0.275\,Nm$$

$$H = 2\pi NT = 2\pi(60\,rev/sec)(0.275\,Nm) = 104\,w$$

(2) 200 SERIES BRG

RELIABILITY = 95% → $K_R = 0.62$

25,000 hr → $25{,}000\,\text{hr} \times 60\,\frac{\text{rev}}{\text{sec}} \times \frac{60\,\text{sec}}{\text{min}} \times \frac{60\,\text{min}}{\text{hr}} = 5.4\times10^9$ CYCLES

MODERATE IMPACT → $K_a = 1.75$

$$C_{REQ} = K_a F_e \left(\frac{L}{K_R L_R}\right)^{0.3} = 1.75(1.25\text{kN})\left[\frac{5.4\times10^9}{.62(90\times10^6)}\right]^{0.3} = 8.6\,\text{kN}$$

CHOOSE 40mm bore RADIAL BALL → 208 } OR LARGER
OR 40 mm bore ANGULAR BALL → 208 } BUILD UP SHAFT!

(3) THIS FIGURE IS NOT IN EQUILIBRIUM (NOT CORRECT), BECAUSE THERE IS NO X-DIRECTION FORCE (OR FORCES) TO OPPOSE THE -4kN FORCE.

(4) 2,5E6 CYCLES @ 5,000 lbs
8,333 lbs FOR + 250,000 CYCLES? } 02, 80 mm bore

$$\frac{l_1}{L_1} + \frac{l_2}{L_2} = 1 \rightarrow l_2 = \left(1 - \frac{l_1}{L_1}\right)L_2$$

$l_1 = 2{,}5$ E6 CYCLES

$l_2 = .25$ E6 CYCLES

DEEP GROOVE
C = 70,2 kN
= 15,782 lb

$$L_1 = 1E6\left(\frac{15{,}782\,\text{lb}}{5{,}000\,\text{lb}}\right)^{3.33} = 45E6$$

$$L_2 = 1E6\left(\frac{15{,}782\,\text{lb}}{8{,}333\,\text{lb}}\right)^{3.33} = 8.4E6$$

$l_2 = 7{,}9$ E6 CYCLES, SO YOU CAN INCREASE LOAD

ANGULAR CONTACT
C = 80,6 kN
= 18,120 lb

$$L_1 = 1E6\left(\frac{18{,}120}{5{,}000}\right)^{3.33} = 73E6$$

$$L_2 = 1E6\left(\frac{18{,}120}{8{,}333}\right)^{3.33} = 13E6$$

$l_2 = 12{,}5$ E6 CYCLES, SO YOU CAN INCREASE LOAD