MECH 325
Homework Assignment #3
Due Nov. 12

Problem 1 (Question 12-26)

12–26 An Oiles SP 500 alloy brass bushing is 0.75 in long with a 0.75-in dia bore and operates in a clean environment at 70°F. The allowable wear without loss of function is 0.004 in. The radial load is 400 lbf. The shaft speed is 250 rev/min. Estimate the number of revolutions for radial wear to be 0.004 in.

12-26 Given: Oiles SP 500 alloy brass bushing, $L = 0.75$ in, $D = 0.75$ in, $T_\infty = 70°\text{F}$, $F = 400$ lbf, $N = 250$ rev/min, and $w = 0.004$ in.

Table 12-9: $K = 0.6(10^{-10})\ \text{in}^3\cdot\text{min}/(\text{lbf}\cdot\text{ft}\cdot\text{h})$

$$P = F/(DL) = 400/[0.75(0.75)] = 711 \text{ psi}$$

$$V = \pi DN/12 = \pi(0.75)250/12 = 49.1 \text{ ft/min}$$

Table 12-9: $PV_{max} = 46\,700$ psi·ft/min, $P_{max} = 3560$ psi, $V_{max} = 100$ ft/min

$$P_{max} = \frac{4}{\pi}\frac{F}{DL} = \frac{4}{\pi}\frac{400}{0.75^2} = 905 \text{ psi} < 3560 \text{ psi} \quad O.K.$$

$$PV = 711\,(49.1) = 34\,910 \text{ psi·ft/min} < 46\,700 \text{ psi·ft/min} \quad O.K.$$

Eq. (12-43) can be written as

$$w = K\frac{4}{\pi}\frac{F}{DL}Vt$$

Solving for t,

$$t = \frac{\pi DLw}{4KVF} = \frac{\pi(0.75)0.75(0.004)}{4(0.6)10^{-10}(49.1)400}$$

$$= 1500 \text{ h} = 1500(60) = 90\,000 \text{ min}$$

Cycles $= Nt = 250\,(90\,000) = 22.5\,(10^6)$ cycles *Ans.*

Problem 2 (Question 12-27)

12–27 Choose an Oiles SP 500 alloy brass bushing to give a maximum wear of 0.002 in for 1000 h of use with a 200 rev/min journal and 100 lbf radial load. Use $\hbar_{CR} = 2.7$ Btu/(h · ft^2 · °F), $T_{max} = 300$°F, $f_s = 0.03$, and a design factor $n_d = 2$. The bearing is to operate in a clean environment at 70°F. Table 12–12 lists the bushing sizes available from the manufacturer.

12-27 Given: Oiles SP 500 alloy brass bushing, $w_{max} = 0.002$ in for 1000 h, $N = 200$ rev/min, $F = 100$ lbf, $\hbar_{CR} = 2.7$ Btu/(h·ft^2·°F), $T_{max} = 300$ °F, $f_s = 0.03$, and $n_d = 2$.

Using Eq. (12-49) with $n_d F$ for F, $f_s = 0.03$ from Table 12-10, and $\hbar_{CR} = 2.7$ Btu/(h · ft^2 · °F), gives

$$L \geq \frac{720\ f_s n_d F N}{J\hbar_{CR}\left(T_f - T_\infty\right)} = \frac{720(0.03)2(100)200}{778(2.7)(300-70)} = 1.79 \text{ in}$$

From Table 12-12, the smallest available bushing has an ID = 1 in, OD = $1\frac{3}{8}$ in, and L = 2 in. With $L/D = 2/1 = 2$, this is inside of the recommendations of Eq. (12-44). Thus, for the first trial, try the bushing with ID = 1 in, OD = $1\frac{3}{8}$ in, and $L = 2$ in. Thus,

Eq. (12-42): $$P_{max} = \frac{4}{\pi}\frac{n_d F}{DL} = \frac{4}{\pi}\frac{2(100)}{1(2)} = 127.3 \text{ psi} < 3560 \text{ psi} \quad \text{(OK)}$$

$$P = \frac{n_d F}{DL} = \frac{2(100)}{1(2)} = 100 \text{ psi}$$

Eq. (12-40): $$V = \frac{\pi D N}{12} = \frac{\pi(1)200}{12} = 52.4 \text{ ft/min} < 100 \text{ ft/min} \quad \text{(OK)}$$

$$PV = 100(52.4) = 5240 \text{ psi} \cdot \text{ft/min} < 46\,700 \text{ psi} \cdot \text{ft/min} \quad \text{(OK)}$$

Eq. (12-43), with Table 12-9:

$w = K n_d F N t/(3L)$

$= 6(10^{-11})(2)(100)(200)(1000)/[(3)(2)] = 0.000\,400$ in < 0.001 in (OK)

Answer Select ID = 1 in, OD = $1\frac{3}{8}$ in, and $L = 2$ in.

Problem 3 (Question 11-3)

An angular-contact, inner ring rotating, 02-series ball bearing is required for an application in which the life requirement is 40 kh at 520 rev/min. The design radial load is 725 lbf. The application factor is 1.4. The reliability goal is 0.90. Find the multiple of rating life x_D required and the catalog rating C_{10} with which to enter Table 11–2. Choose a bearing and estimate the existing reliability in service.

11-3 For the angular-contact 02-series ball bearing as described, the rating life multiple is

$$x_D = \frac{L_D}{L_R} = \frac{60L_D n_D}{L_{10}} = \frac{60(40\,000)520}{10^6} = 1248$$

The design radial load is

$$F_D = 1.4(725) = 1015 \text{ lbf} = 4.52 \text{ kN}$$

Eq. (11-9):

$$C_{10} = 1015\left\{\frac{1248}{0.02 + (4.459 - 0.02)\left[\ln(1/0.9)\right]^{1/1.483}}\right\}^{1/3}$$

$$= 10\,930 \text{ lbf} = 48.6 \text{ kN}$$

Table 11-2: Select an 02-60 mm bearing with C_{10} = 55.9 kN. *Ans.*

Eq. (11-21): $$R = \exp\left\{-\left[\frac{1248(4.52/55.9)^3 - 0.02}{4.439}\right]^{1.483}\right\} = 0.945 \qquad \textit{Ans.}$$

Problem 4 (Question 11-23)

An 02-series single-row deep-groove ball bearing with a 30-mm bore (see Tables 11–1 and 11–2 for specifications) is loaded with a 2-kN axial load and a 5-kN radial load. The inner ring rotates at 400 rev/min.

(*a*) Determine the equivalent radial load that will be experienced by this particular bearing.

(*b*) Determine the predicted life (in revolutions) that this bearing could be expected to give in this application with a 99 percent reliability.

11-23 **(a)** $F_a = 2$ kN, $F_r = 5$ kN, $n_D = 400$ rev/min, $V = 1$

From Table 11-2, 30 mm bore, C_{10} = 19.5 kN, C_0 = 10.0 kN

$$F_a / C_0 = 2 / 10 = 0.2$$

From Table 11-1, $0.34 \le e \le 0.38$.

$$\frac{F_a}{VF_r} = \frac{2}{(1)(5)} = 0.4$$

Since this is greater than e, interpolating Table 11-1, with $F_a / C_0 = 0.2$, we obtain $X_2 = 0.56$ and $Y_2 = 1.27$.

Eq. (11-12): $F_e = X_i V F_r + Y_i F_a = (0.56)(1)(5) + (1.27)(2) = 5.34$ kN *Ans.*

$F_e > F_r$ so use F_e.

(b) Solve Eq. (11-10) for x_D.

$$x_D = \left(\frac{C_{10}}{a_f F_D}\right)^a \left[x_0 + (\theta - x_0)(1 - R_D)^{1/b}\right]$$

$$x_D = \left(\frac{19.5}{(1)(5.34)}\right)^3 \left[0.02 + (4.459 - 0.02)(1 - 0.99)^{1/1.483}\right]$$

$$x_D = 10.66$$

$$x_D = \frac{L_D}{L_R} = \frac{\mathsf{L}_D n_D (60)}{10^6}$$

$$\mathsf{L}_D = \frac{x_D (10^6)}{n_D (60)} = \frac{10.66(10^6)}{(400)(60)} = 444 \text{ h} \quad \textit{Ans.}$$

Rev= L_D n_D = 444 hours (60 min/hr)400rpm = 1.07×10^6 revolutions

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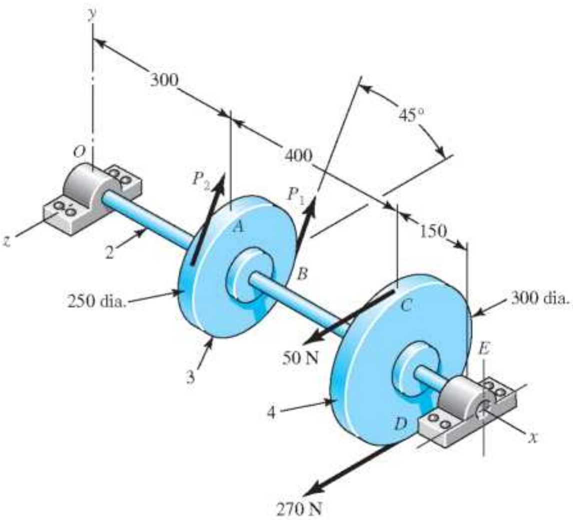
Problem 5 (Question 11-35)

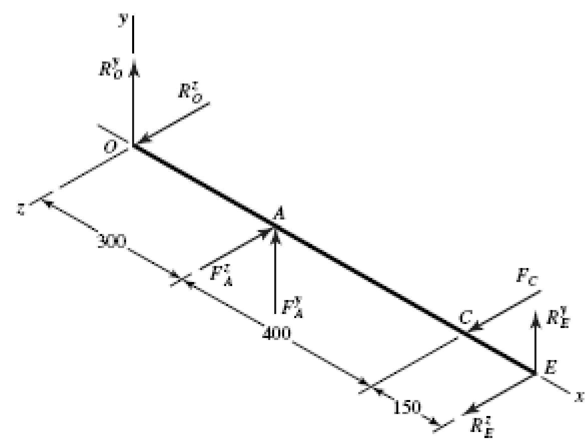
The figure is a schematic drawing of a countershaft that supports two V-belt pulleys. The countershaft runs at 1500 rev/min and the bearings are to have a life of 60 kh at a combined reliability of 0.98, assuming distribution data from manufacturer 2 in Table 11–6. The belt tension on the loose side of pulley *A* is 15 percent of the tension on the tight side. Select deep-groove bearings from Table 11–2 for use at *O* and *E,* using an application factor of unity.



Problem 11–35
Dimensions in millimeters.

11-35 The reliability of the individual bearings is $R = \sqrt{0.98} = 0.9899$

From statics,



$T = (270 - 50) = (P_1 - P_2)125$

$= (P_1 - 0.15\, P_1)125$

$P_1 = 310.6$ N,

$P_2 = 0.15\,(310.6) = 46.6$ N

$P_1 + P_2 = 357.2$ N

$F_A^y = 357.2 \sin 45^\circ = 252.6 \text{ N} = F_A^z$

$$\sum M_O^z = 850R_E^y + 300(252.6) = 0 \Rightarrow R_E^y = -89.2 \text{ N}$$
$$\sum F^y = 252.6 - 89.2 + R_O^y = 0 \Rightarrow R_O^y = -163.4 \text{ N}$$
$$\sum M_O^y = -850R_E^z - 700(320) + 300(252.6) = 0 \Rightarrow R_E^z = -174.4 \text{ N}$$
$$\sum F^z = -174.4 + 320 - 252.6 + R_O^z = 0 \Rightarrow R_O^z = 107 \text{ N}$$

$$R_O = \sqrt{(-163.4)^2 + 107^2} = 195 \text{ N}$$
$$R_E = \sqrt{(-89.2)^2 + (-174.4)^2} = 196 \text{ N}$$

The radial loads are nearly the same at O and E. We can use the same bearing at both locations.

$$x_D = \frac{60\,000(1500)(60)}{10^6} = 5400$$

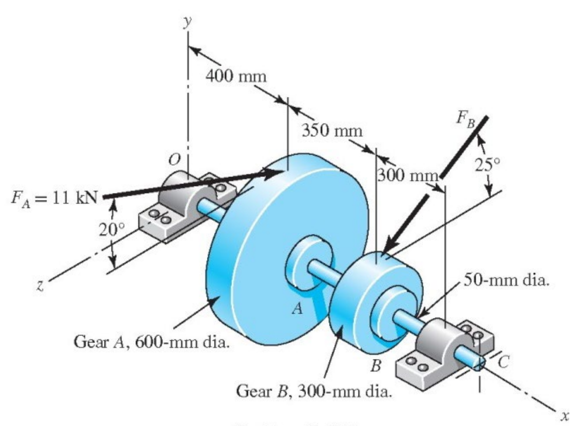
Eq. (11-9): $$C_{10} = 1(0.196)\left\{\frac{5400}{0.02 + 4.439\left[\ln(1/0.9899)\right]^{1/1.483}}\right\}^{1/3} = 5.7 \text{ kN}$$

From Table 11-2, select an 02-12 mm deep-groove ball bearing with a basic load rating of 6.89 kN. *Ans.*

Problem 6 (Question 7-40)

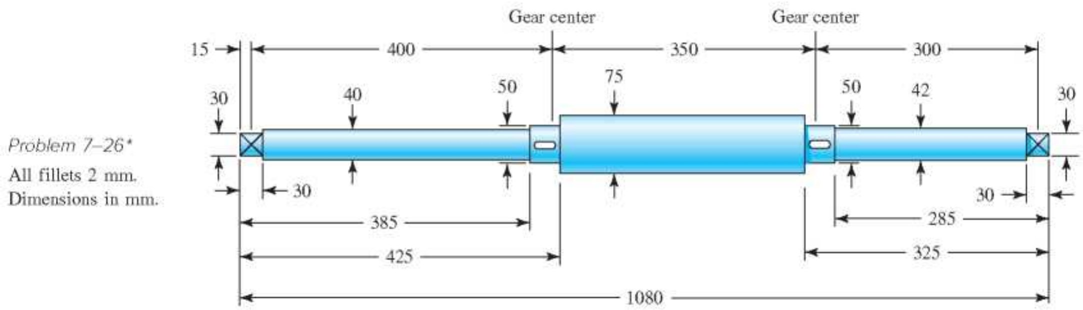
A gear reduction unit uses the countershaft shown in the figure. Gear *A* receives power from another gear with the transmitted force F_A applied at the 20° pressure angle as shown. The power is transmitted through the shaft and delivered through gear *B* through a transmitted force F_B at the pressure angle shown.

(*a*) Determine the force F_B, assuming the shaft is running at a constant speed.
(*b*) Find the bearing reaction forces, assuming the bearings act as simple supports.
(*c*) Draw shear-force and bending-moment diagrams for the shaft. If needed, make one set for the horizontal plane and another set for the vertical plane.
(*d*) At the point B, determine the maximum bending moment, the bending stress and the torsional shear stress.



Problem 3–84*

The shaft shown in the figure below is proposed for the application above. The material is AISI 1018 cold-drawn steel. The gears seat against the shoulders and have hubs with setscrews to lock them in place. The effective centers of the gears for force transmission are shown. The keyseats are cut with standard endmills. The bearings are press-fit against the shoulders.



(e) Specify a square key for gear B, using a factor of safety of 1.1.

Answer

(a)

$$\sum T = 0 = -11\,000(\cos 20^\circ)(300) + F_B(\cos 25^\circ)(150)$$

$$F_B = 22\,810 \text{ N} \qquad \textit{Ans.}$$

(b)

$$\sum M_{Oz} = 0 = -11\,000(\sin 20^\circ)(400) - 22\,810(\sin 25^\circ)(750) + R_{Cy}(1050)$$

$$R_{Cy} = 8319 \text{ N} \qquad \textit{Ans.}$$

$$\sum F_y = 0 = R_{Oy} - 11000(\sin 20^\circ) - 22\,810\sin(25^\circ) + 8319$$

$$R_{Oy} = 5083 \text{ N} \qquad \textit{Ans.}$$

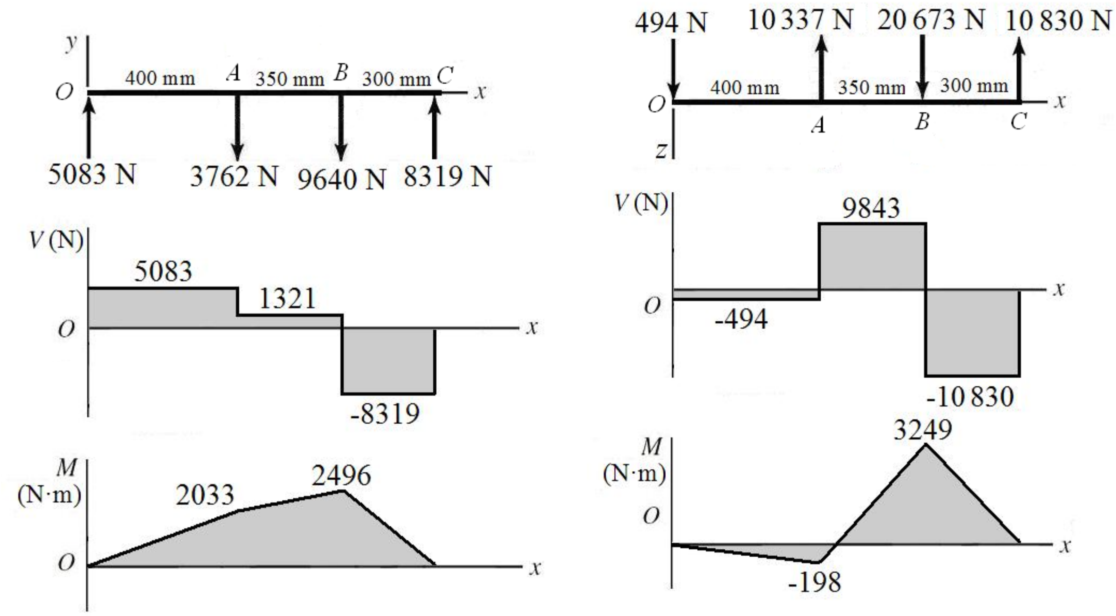
$$\sum M_{Oy} = 0 = 11\,000(\cos 20^\circ)(400) - 22\,810(\cos 25^\circ)(750) - R_{Cz}(1050)$$

$$R_{Cz} = -10\,830 \text{ N} \qquad \textit{Ans.}$$

$$\sum F_z = 0 = R_{Oz} - 11\,000(\cos 20^\circ) + 22\,810(\cos 25^\circ) - 10\,830$$

$$R_{Oz} = 494 \text{ N} \qquad \textit{Ans.}$$

(c)



(d) From the bending moment diagrams, it is clear that the critical location is at B where both planes have the maximum bending moment. Combining the bending moments from the two planes,

$$M = \sqrt{(2496)^2 + (3249)^2} = 4097 \text{ N}\cdot\text{m}$$

The torque transmitted through the shaft from A to B is

$$T = 11\,000\cos(20^\circ)(0.3) = 3101 \text{ N}\cdot\text{m}.$$

For a stress element on the outer surface where the bending stress and the torsional stress are both maximum,

$$\sigma = \frac{Mc}{I} = \frac{32M}{\pi d^3} = \frac{32(4097)}{\pi(0.050)^3} = 333.9(10^6) \text{ Pa} = 333.9 \text{ MPa} \qquad \textit{Ans.}$$

$$\tau = \frac{Tr}{J} = \frac{16T}{\pi d^3} = \frac{16(3101)}{\pi(0.050)^3} = 126.3(10^6) \text{ Pa} = 126.3 \text{ MPa} \qquad \textit{Ans.}$$

The torque to be transmitted through the key from the gear to the shaft is T = 3101 N·m. The nominal shaft diameter supporting the gear is 50 mm. To determine an appropriate key size for the shaft diameter, we can either convert to inches and use Table 7-6, or we can look up standard metric key sizes from the internet or a machine design handbook. It turns out that the recommended metric key for a 50 mm shaft is 14 x 9 mm. Since the problem statement specifies a square key, we will use a 14 x 14 mm key. For comparison, using Table 7-6 as a guide, for d = 50 mm = 1.97 in, a 0.5 in square key is appropriate. This is equivalent to 12.7 mm. A 14 x 14 mm size is conservative, but reasonable after rounding up to standard sizes.
The force applied to the key is

$$F = \frac{T}{r} = \frac{3101}{0.050/2} = 124\left(10^3\right)\ \text{N}$$

Selecting 1020 CD steel for the key, with S_y = 390 MPa, and using the distortion-energy theory, $S_{sy} = 0.577\ S_y = 0.577(390) = 225$ MPa.

Failure by shear across the key:

$$\tau = \frac{F}{A} = \frac{F}{tl}$$

$$n = \frac{S_{sy}}{\tau} = \frac{S_{sy}}{F/(tl)} \qquad \Rightarrow \qquad l = \frac{nF}{tS_{sy}} = \frac{1.1(124)\left(10^3\right)}{(0.014)(225)\left(10^6\right)} = 0.0433\ \text{m} = 43.3\ \text{mm}$$

Failure by crushing:

$$\sigma = \frac{F}{A} = \frac{F}{(t/2)l}$$

$$n = \frac{S_y}{\sigma} = \frac{S_y}{2F/(tl)} \qquad \Rightarrow \qquad l = \frac{2Fn}{tS_y} = \frac{2(124)\left(10^3\right)(1.1)}{(0.014)(390)\left(10^6\right)} = 0.0500\ \text{m} = 50.0\ \text{mm}$$

Select 14 mm square key, 50 mm long, 1020 CD steel. *Ans.*