SOLUTION (13.1)
Known: A web site is given as **http://www.grainger.com**.

Find: Search for a self-aligning plastic bearing block with a 1 in. inside diameter. List the manufacturer, description, and price of the bearing.

Analysis: The web site lists:

Mfg. Name: Dayton
Description: Flange Block, UHMW-PE 2-Bolt Flange/Self-Aligning type
Price: $7.06

SOLUTION (13.2)
Known: A web site is given as **http://www.grainger.com**.

Find: Search for a two bolt rigid plastic bearing block with a 1 in. inside diameter. List the manufacturer, description, and price of the bearing.

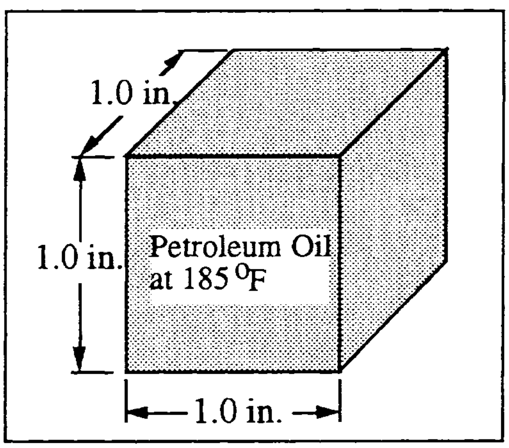
Analysis: The web site lists:

Mfg. Name: Dayton
Description: Plastic Bearing Block, 1"ID Solid UHMW-PE 2-Bolt Rigid-Type
Price: $18.50

SOLUTION (13.3)
Known: A petroleum oil is at 185 °F.

Find: Determine the weight per cubic inch.

Schematic and Given Data:



Assumption: Equation 13.6b is valid.

Analysis:

1. From Eq. 13.6b for T = 185 °F. Hence, $\rho = 0.89 - 0.00035(185 - 60) = 0.846$ g/cm^3

2. Writing ρ in lbm/in.^{3},

ρ = 0.846 g/cm^3 = 0.846 g/cm^3 (10^{-3} kg/g)(lbm/0.454 kg)(cm^3/10^{-6} m^3)(0.3048^3 m^3/ft^3)(ft^3/12^3 in.^{3}) = 0.0306 lbm/in.^{3} ■

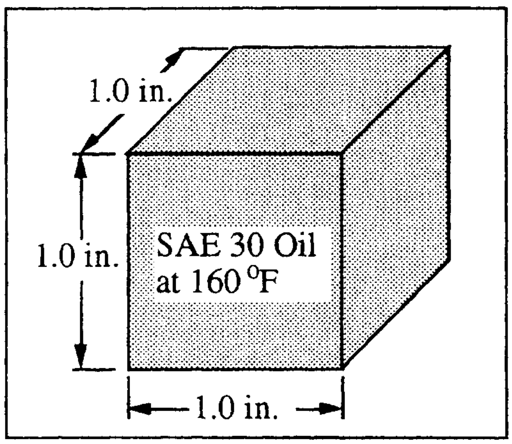
3. $\gamma = \frac{g}{g_c}\rho = \frac{32.2}{32.2}(0.0306) = 0.0306\ \frac{\text{lb}}{\text{in.}^3}$

SOLUTION (13.4)

Known: SAE 30 oil is at 160 °F.

Find: Determine the weight per cubic inch.

Schematic and Given Data:



Assumption: Equation 13.6b is valid.

Analysis:

1. From Eq. 13.6b for T = 160 °F. Hence, ρ = 0.89 - 0.00035(160 - 60) = 0.855 g/cm^3
2. Writing ρ in lbm/in.^{3},

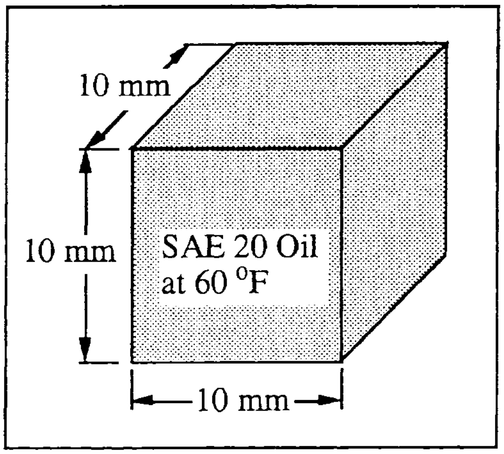
ρ = 0.855 g/cm^3 = 0.855 g/cm^3 (10^{-3} kg/g)(lbm/0.454 kg)(cm^3/10^{-6} m^3)(0.3048^3 m^3/ft^3)(ft^3/12^3 in.^{3}) = 0.0309 lbm/in.^{3} ■

3. $\gamma = \frac{g}{g_c}\rho = \frac{32.2}{32.2}(0.0309) = 0.0309\ \frac{\text{lb}}{\text{in.}^3}$

SOLUTION (13.5)
Known: SAE 20 oil is at 60 °F.

Find: Determine the density in gram per cubic centimeter.

Schematic and Given Data:



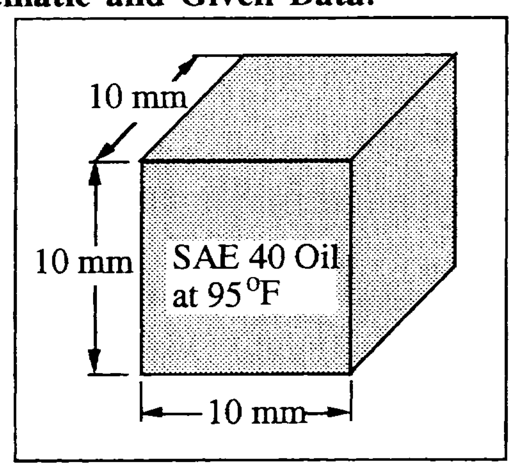
Assumption: Equation 13.6a is valid.

Analysis:
1. 60 °F = (60 - 32) (5/9) = 15.6 °C.
2. From Eq. 13.6a with T = 15.6 °C,
   $\rho = 0.89 - 0.00063(15.6 - 15.6) = 0.89 \text{ g/cm}^3$ ■

SOLUTION (13.6)
Known: SAE 40 oil is at 95 °F.

Find: Determine the density in gram per cubic centimeter.

Schematic and Given Data:



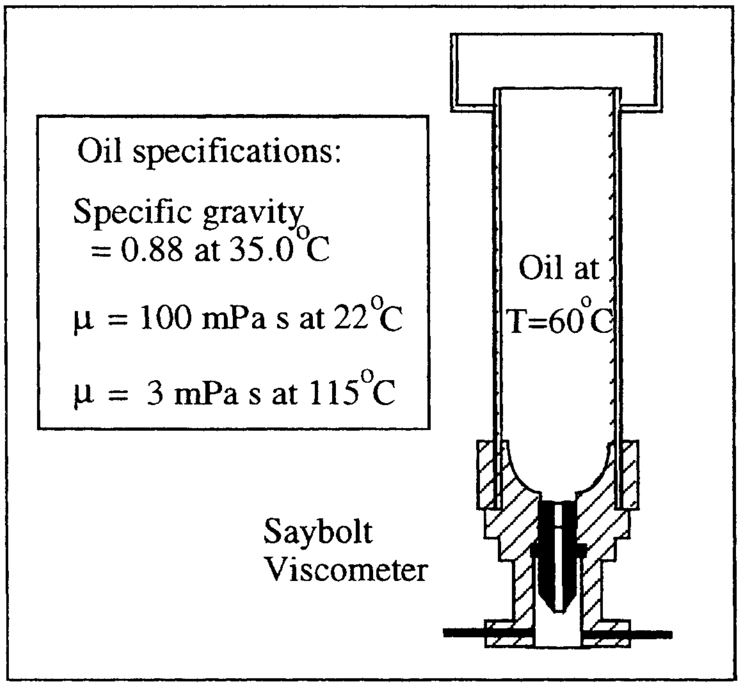
Assumption: Equation 13.6a is valid.

Analysis:
1. 95 °F = (95 - 32) (5/9) = 35 °C.
2. From Eq. 13.6a with T = 35 °C,
   $\rho = 0.89 - 0.00063(35 - 15.6) = 0.88\ g/cm^3$ ■

SOLUTION (13.7)
Known: An oil's viscosity is given at two different temperatures (10 °C and 100 °C).

Find: Determine the oil viscosity in mPa·s at 80 °C.

Schematic and Given Data:



Assumption: The absolute viscosity can be determined from Fig. 13.6 by interpolation.

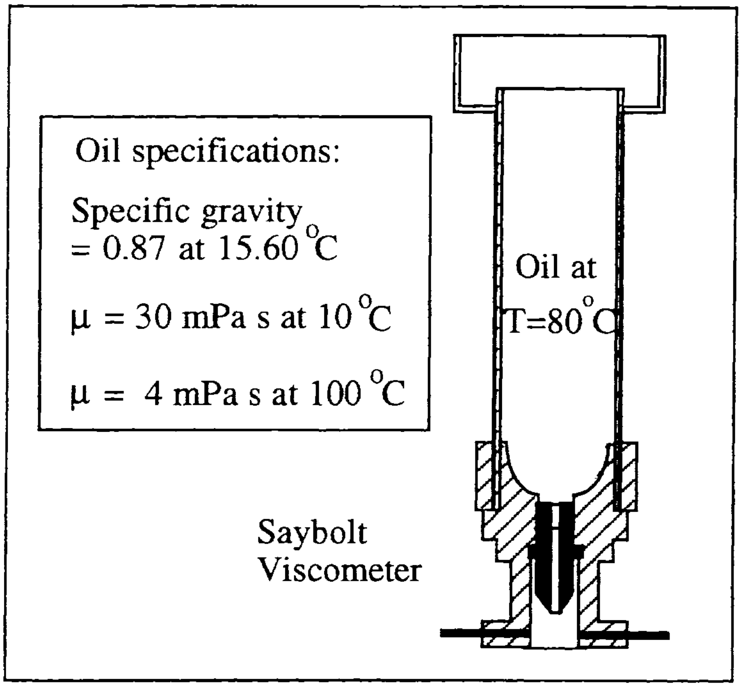
Analysis:
1. Mark the two given data point locations on Fig. 13.6.
2. Connect these two points with a straight line.
3. Use the straight line to determine the value of viscosity for T = 60 °C.
4. The viscosity is 13 mPa·s. ■

SOLUTION (13.8)

Known: An oil's viscosity is given at two different temperatures (10 °C and 100 °C).

Find: Determine the oil viscosity in mPa·s at 80 °C.

Schematic and Given Data:



Assumption: The absolute viscosity can be determined from Fig. 13.6 by interpolation.

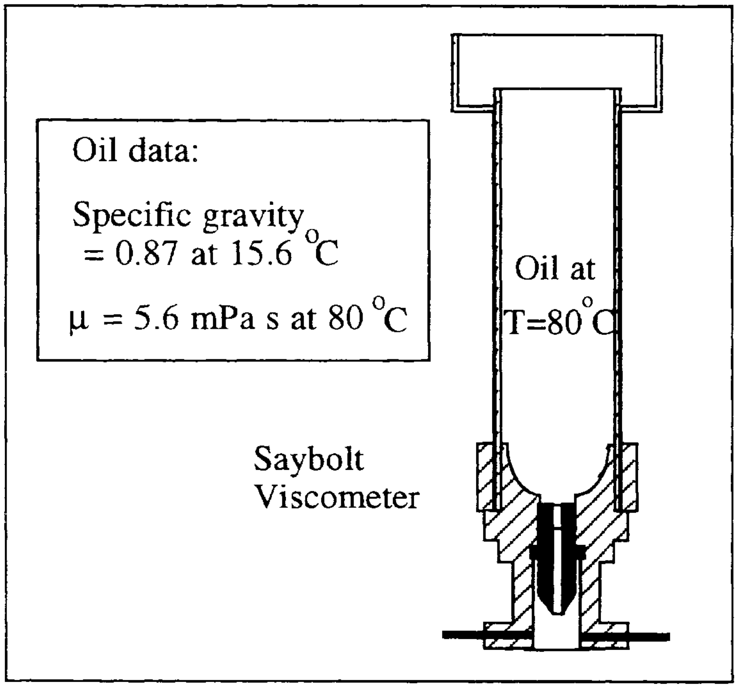
Analysis:

1. Mark the two given data point locations on Fig. 13.6.
2. Connect these two points with a straight line.
3. Use the straight line to determine the value of viscosity for T = 80 °C.
4. The viscosity is 5.6 mPa·s. ■

SOLUTION (13.9)
Known: Oil is at a known temperature of 80 °C.

Find: Determine the kinematic viscosity at 80 °C.

Schematic and Given Data:



Assumption: The absolute viscosity can be determined from Fig. 13.6 by interpolation.

Analysis:

1. From Eq. 13.6a but with 0.89 replaced by 0.87,
   $\rho_{80\,^{\circ}C} = 0.87 - 0.00063(80 - 15.6) = 0.829\ g/cm^3$.
2. From Eq. 13.3,

$$\upsilon_{80\,^{\circ}C} = \frac{\mu_{80\,^{\circ}C}}{\rho_{80\,^{\circ}C}} = \frac{5.6\ (mPa\ s)}{0.829\ (g/cm^3)}$$

$$= \frac{5.6 \times 10^{-3}(N/m^2)\ s}{0.829\ (10^{-3}\ kg/10^{-6}\ m^3)}$$

$$= \frac{5.6 \times 10^{-3}(kg\ m/9.81\ sec^2 m^2)\ s}{0.829 \times 10^3 (kg/m^3)}$$

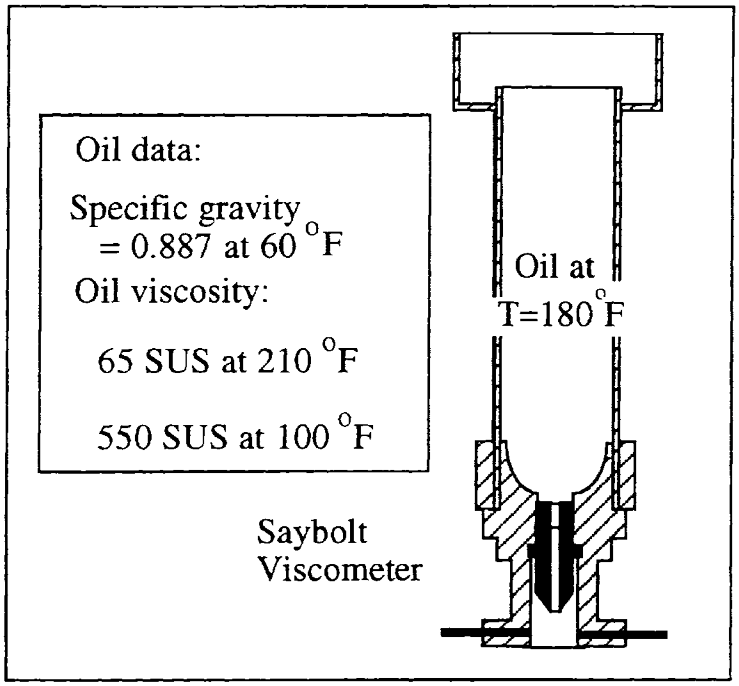
$$= 6.886 \times 10^{-7}\ m^2/s \text{ or } 6.886 \times 10^{-3}\ St.$$ ■

SOLUTION (13.10)

Known: An oil has a known specific gravity at 60 °F. Also we know its viscosity at two other known temperatures.

Find: Estimate the viscosity of this oil in μreyn at 180 °F.

Schematic and Given Data:



Assumptions:
1. Equation 13.5 is valid.
2. The absolute viscosity can be determined from Fig. 13.6 by interpolation.
3. Equation 13.6b is valid, but 0.89 is replaced by 0.887.

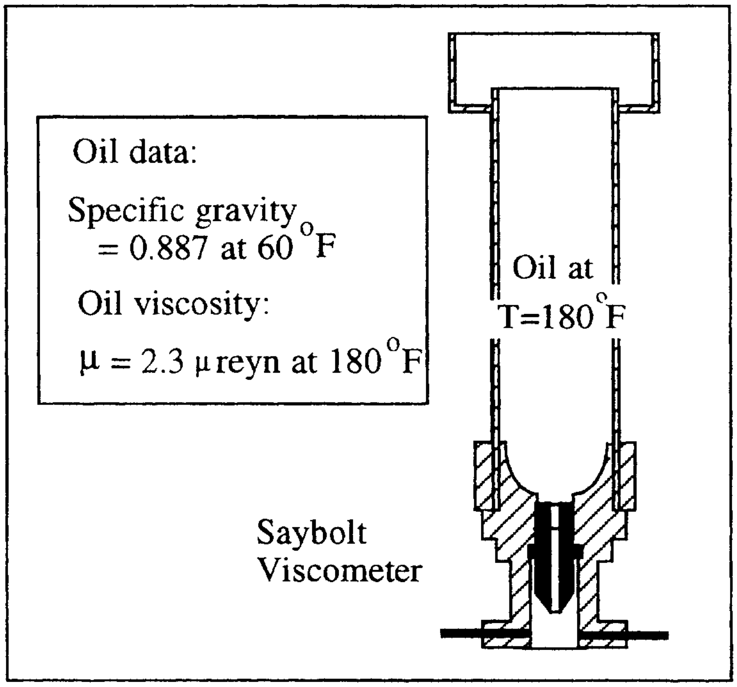
Analysis:
1. From Eq. 13.6b,
   $\rho_{210\,°F} = 0.887 - 0.00035\ (210 - 60) = 0.8345\ g/cm^3$
   $\rho_{100\,°F} = 0.887 - 0.00035\ (100 - 60) = 0.873\ g/cm^3$
2. From Eq. 13.5,
   $\mu_{210\,°F} = 0.145\ (0.22 \times 65 - 180/65) \times 0.8345 = 1.395\ \mu reyn$
   $\mu_{100\,°F} = 0.145\ (0.22 \times 550 - 180/550) \times 0.873 = 15.28\ \mu reyn$
3. With the interpolation method, we obtain from Fig. 13.6, $\mu_{180\,°F} = 2.3\ \mu reyn$ ■

SOLUTION (13.11)

Known: An oil has a known specific gravity at 60 $^{\circ}$F. Also we know its viscosity at two other known temperatures.

Find: Determine the kinematic viscosity at 180 $^{\circ}$F.

Schematic and Given Data:



Assumptions:

1. Equation 13.5 is valid.
2. The absolute viscosity can be determined from Fig. 13.6 by interpolation.
3. Equation 13.6b is valid, but 0.89 is replaced by 0.887.

Analysis:

1. From Eq. 13.5,

   $\rho_{180\,^{\circ}F} = 0.887 - 0.00035(180 - 60) = 0.845 \text{ g/cm}^3$

2. From Eq. 13.3,

$$\nu_{180\,^{\circ}F} = \frac{\mu_{180\,^{\circ}F}}{\rho_{180\,^{\circ}F}} = \frac{2.3\ \mu\text{reyn}}{0.845\ (\text{g/cm}^3)} = \frac{2.3 \times 10^{-6} \times 6890\ (\text{Pa·s})}{0.845\ (\text{g/cm}^3)}$$

$$= \frac{1.585 \times 10^{-2}\ (\text{N/m}^2)\text{s}}{0.845\ (10^{-3}\ \text{kg}/10^{-6}\text{m}^3)} = \frac{1.585 \times 10^{-2}\ (\text{kgm}/9.81\ \text{s}^2\text{m}^2)\text{s}}{0.845 \times 10^3(\text{kg/cm}^3)}$$

$$= 1.912 \times 10^{-6}\ \text{m}^2/\text{s} = 1.912 \times 10^{-2}\ \text{St.} = 2.96 \times 10^{-3}\ \text{in.}^2/\text{s}$$

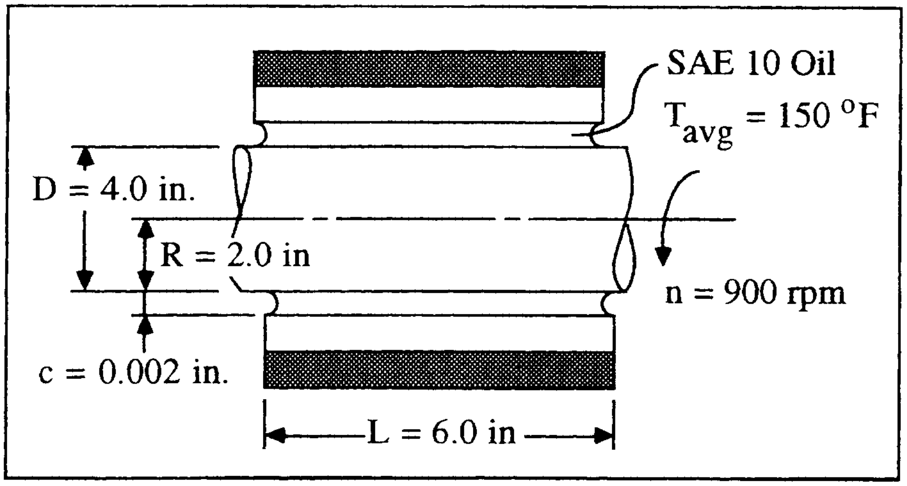
■

SOLUTION (13.12)

Known: A lightly loaded 360° journal bearing has a known diameter, length and radial clearance. The journal has a given rotational speed and is lubricated with a SAE 10 oil at a known average temperature.

Find: Determine the power loss and the frictional torque.

Schematic and Given Data:



Assumptions:

1. The effect of eccentricity between the journal bearing and the journal is negligible.
2. There is no lubricant flow in the axial direction.

Analysis:

1. From Fig. 13.6, $\mu = 1.6 \times 10^{-6}$ μreyn
2. From Petroff's equation,

$$T_f = \frac{4\pi^2 \mu n L R^3}{c} = \frac{4\pi^2(1.6 \times 10^{-6} \text{ reyn})(15 \text{ rev/s})(6 \text{ in.})(2 \text{ in.})^3}{(0.002 \text{ in.})}$$

$= 22.75$ lb·in. $= 1.90$ lb·ft. ■

3. Power loss, $\dot{W} = \dfrac{nT_f}{5252} = \dfrac{(900 \text{ rev/min})(1.90 \text{ lb·ft})}{5252\left(\dfrac{\text{lb·ft·rev}}{\text{min·hp}}\right)}$

$= 0.326$ hp/bearing ■

Comments:

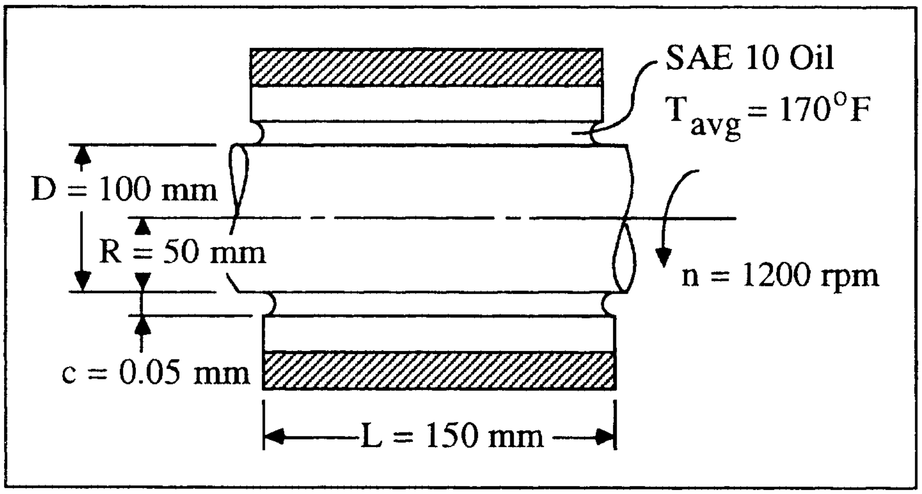
1. In an actual bearing, we would need to verify that while dissipating 0.326 hp/bearing the average oil temperature in the bearing would be consistent with the value of viscosity used in the calculation.
2. If we double the radius then the frictional torque is increased by 8. But increasing the bearing radius requires more clearance and eventually more oil must be supplied to prevent bearing failure.

SOLUTION (13.13)

Known: A Petroff bearing has a known diameter, length and radial clearance. The journal has a given rotational speed and is lubricated with a SAE 10 oil at a known average temperature.

Find: Estimate the power loss and the friction torque.

Schematic and Given Data:



Assumptions:
1. The effect of eccentricity between the journal bearing and the journal is negligible.
2. There is no lubricant flow in the axial direction.
3. The radial load is small.

Analysis:
1. From Fig. 13.6, μ = 7.3 mPa·s.
2. From Eq. (b) on page 485,

$$T_f = \frac{4\pi^2 \mu n L R^3}{c}$$

$$= \frac{4\pi^2\left(7.3 \times 10^{-3}\ \frac{Ns}{m^2}\right)(20\ rev/s)(0.15\ m)(0.05\ m)^3}{0.05 \times 10^{-3}\ m}$$

$$= 2.16\ N{\cdot}m$$

3. $$\dot{W} = \frac{nT}{9549} = \frac{(1200\ rev/min)(2.16\ Nm)}{9549\ \frac{N{\cdot}m{\cdot}rev}{min\ kW}}$$

$$= 0.271\ kW$$ ■

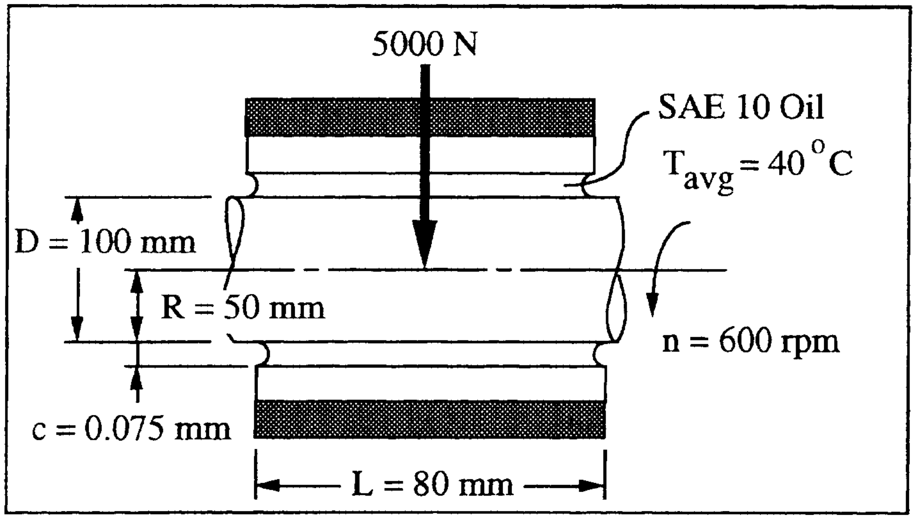
Comment: In an actual situation, we would need to verify that when dissipating 0.271 kW, the average oil temperature in the bearing would be consistent with the value of viscosity used in the calculation.

SOLUTION (13.14)

Known: A shaft with a known diameter, rotational speed and radial load is supported by an oil lubricated bearing of specific length and diametral clearance.

Find: Determine the bearing coefficient of friction and the power loss.

Schematic and Given Data:



Assumptions:

1. There is no eccentricity between the bearing and the journal, and no lubricant flow in the axial direction.
2. The frictional drag force is equal to the product of the coefficient of the friction times the radial shaft load.

Analysis:

1. From Fig 13.6,

   $\mu = 32$ mPa·sec (For SAE 10 oil at 40 °C)

2. From Eq. 13.7,

$$f = 2\pi^2 \frac{\mu n}{P} \frac{R}{c}$$

$$f = 2\pi^2 \frac{(32 \times 10^{-3}\ \text{Pa·s})\left(\frac{600}{60}\ \text{rps}\right)}{\left(\frac{5000}{0.1 \times 0.08}\right)\frac{\text{N}}{\text{m}^2}} \frac{50 \times 10^{-3}\ \text{m}}{0.075 \times 10^{-3}\ \text{m}} = 6.74 \times 10^{-3}$$ ■

3. Power loss $= 2\pi T_f n$

$= 2\pi f(WD)n/2$

$= 6.74 \times 10^{-3}\ \pi\ (5000\ \text{N})(0.1\ \text{m})(10\ \text{rps})$

$= 105.9$ W ■

Comments:

1. If the clearance increases, then friction and power loss decrease but oil supply increases.
2. In an actual situation, we would need to verify when dissipating 105.9 W that the average oil temperature in the bearing would be consistent with the value of viscosity used in the calculations.

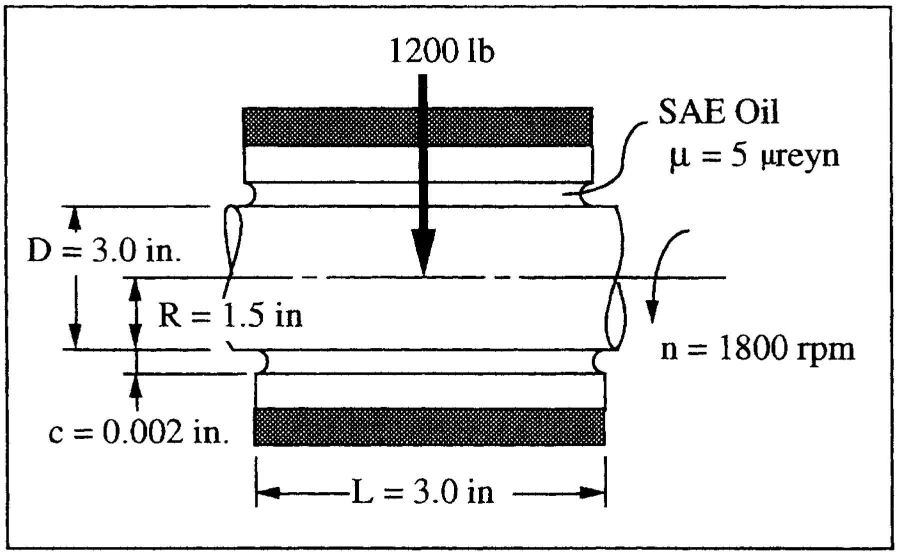
3. Petroff's equation can be checked by using Fig. 13.14 to calculate f. For L/D = 0.8 and $S = \left(\frac{R}{c}\right)^2 \frac{\mu n}{P} = 0.228$, we have $\frac{R}{c} f = 5.5$. Hence, f = 0.0083, which is 19% larger than 6.74×10^{-3}, so Petroff's equation is not very accurate for this problem.

SOLUTION (13.15)

Known: A shaft with known diameter, rotational speed, and radial load is supported by an oil lubricated bearing of specified length and diametrical clearance. The oil viscosity is known.

Find: Estimate the bearing coefficient of friction and the power loss.

Schematic and Given Data:



Assumptions:
1. The load creates a negligible eccentricity between journal and bearing.
2. There is negligible lubricant flow in the axial direction.
3. The Petroff's approach is suitable.

Analysis:
1. The load intensity,

$$P = \frac{W}{LD} = \frac{1200 \text{ lb}}{(3 \text{ in.})(3 \text{ in.})} = 133.3 \text{ psi}$$

2. From Eq. (13.7):

$$f = 2\pi^2 \frac{\mu n}{P} \frac{R}{c} = 2\pi^2 \frac{(5\times10^{-6} \text{ reyn})(30 \frac{\text{rev}}{\text{sec}})}{133.3 \frac{\text{lb}}{\text{in.}^2}} \left(\frac{1.5 \text{ in.}}{0.002 \text{ in.}}\right) = 0.0167$$ ■

3. $T_f = fWr$ = (0.0167)(1200 lb)(1.5 in.) = 29.98 in. lb = 2.5 ft lb

4. Power loss = $\dfrac{T_f(\text{ft lb})\, n(\frac{\text{rev}}{\text{min}})}{5252(\frac{\text{ft lb rev}}{\text{min hp}})} = \dfrac{(2.5 \text{ ft lb})(1800 \frac{\text{rev}}{\text{min}})}{5252(\frac{\text{ft lb rev}}{\text{min hp}})} = 0.856 \text{ hp}$ ■

Comment: Petroff's equation can be checked by using Fig. 13.14 to calculate f.

For $\frac{L}{D} = \frac{3}{3} = 1$ and $S = (\frac{R}{c})^2\frac{\mu n}{P} = 0.632$, we have $\frac{R}{c}f = 13$.

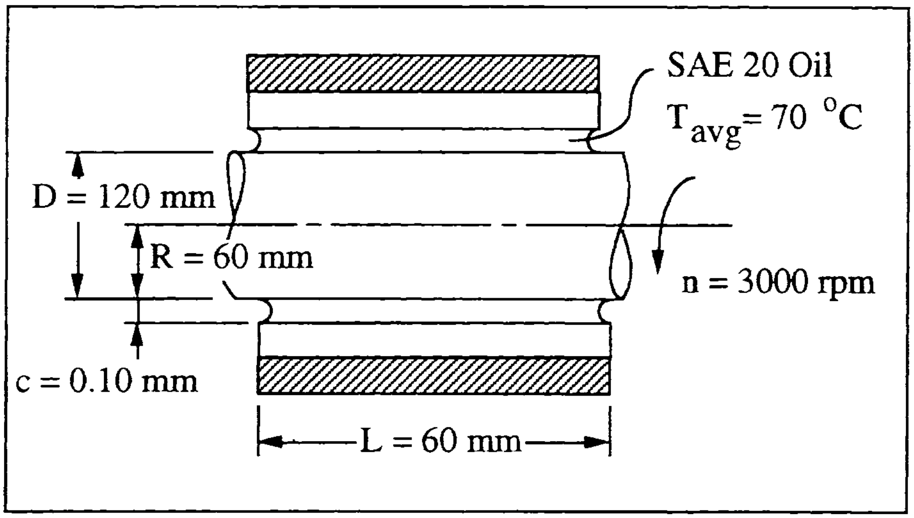
Hence, f = 0.01733 which is 4% larger than 0.01667.

SOLUTION (13.16)

Known: A journal bearing has a known diameter, length, and diametral clearance. The journal has a given rotational speed and is lubricated with a SAE 20 oil at a known average temperature.

Find: Determine the power loss and the friction torque.

Schematic and Given Data:



Assumptions:
1. The effect of eccentricity between the journal bearing and the journal is negligible.
2. There is no lubricant flow in the axial direction.
3. The radial load is small.

Analysis:
With the above assumptions, Petroff's equation would be applicable.
1. From Fig. 13.6, μ = 12.5 mPa·s = .0125 Pa·s

$$= 0.0125 \frac{\text{Ns}}{\text{m}^2}$$

2. From Eq. (b) on page 485, $T_f = \dfrac{4\pi^2 \mu n L R^3}{c}$

$$= \frac{4\pi^2(0.0125\ \frac{Ns}{m^2})(\ 50\ rev/s)(0.060m)(0.060\ m)^3}{0.00010\ m}$$

$T_f = 3.2$ N·m ■

3. From Eq. 1.2, $\dot{W} = \dfrac{nT}{9549}$

$$\dot{W} = \frac{3.2\ N{\cdot}m\ (3000\ rev/min)}{9549\ \frac{N{\cdot}m\ rev}{min\ kW}} = 1.01\ kW$$ ■

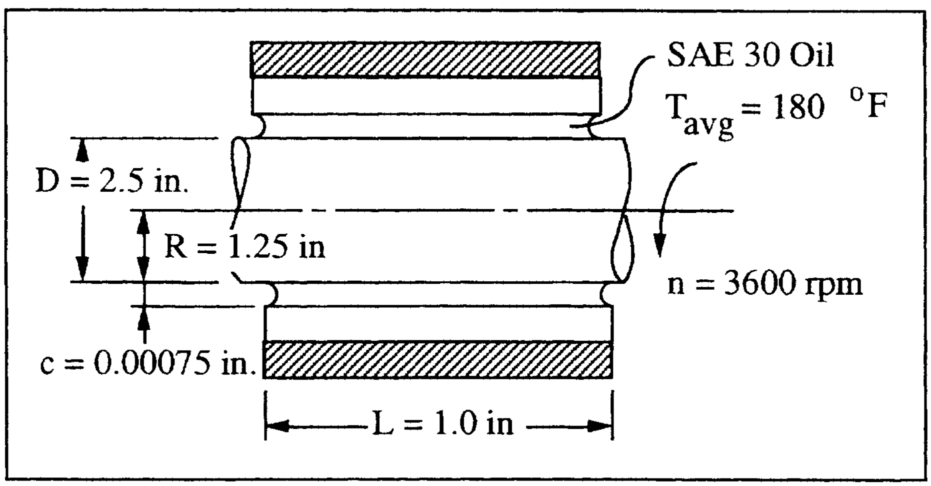
Comment: In an actual situation, we would need to verify that when dissipating 1.01 kW, the average oil temperature in the bearing would be consistent with the value of viscosity used in the calculations.

SOLUTION (13.17)

Known: A journal rotates at given speed and with SAE 30 oil at 180 °F in a journal bearing with known diameter, length, and diametral clearance.

Find: Determine the power loss.

Schematic and Given Data:



Assumptions:
1. The effect of eccentricity between the journal bearing and the journal is negligible.
2. There is no lubricant flow in the axial direction.
3. The radial load is small.

Analysis:
1. From Fig 13.6, for SAE 30 oil at 180 °F, $\mu = 1.87$ μreyn.

2. $$T_f = \frac{4\pi^2 \mu n L R^3}{c} = \frac{4\pi^2 (1.87 \times 10^{-6} \frac{\text{lb sec}}{\text{in}^2})(60 \frac{\text{rev}}{\text{s}})(1 \text{ in.})(1\frac{1}{4} \text{ in.})^3}{0.00075 \text{ in.}}$$

$= 11.54$ lb in. $= 0.962$ lb ft ■

3. $$\dot{W} = \frac{nT}{5250} = \frac{(3600 \text{ rev/min})(0.962 \text{ lb ft})}{5250 \frac{\text{lb ft rev}}{\text{min hp}}} = 0.66 \text{ hp/bearing}$$ ■

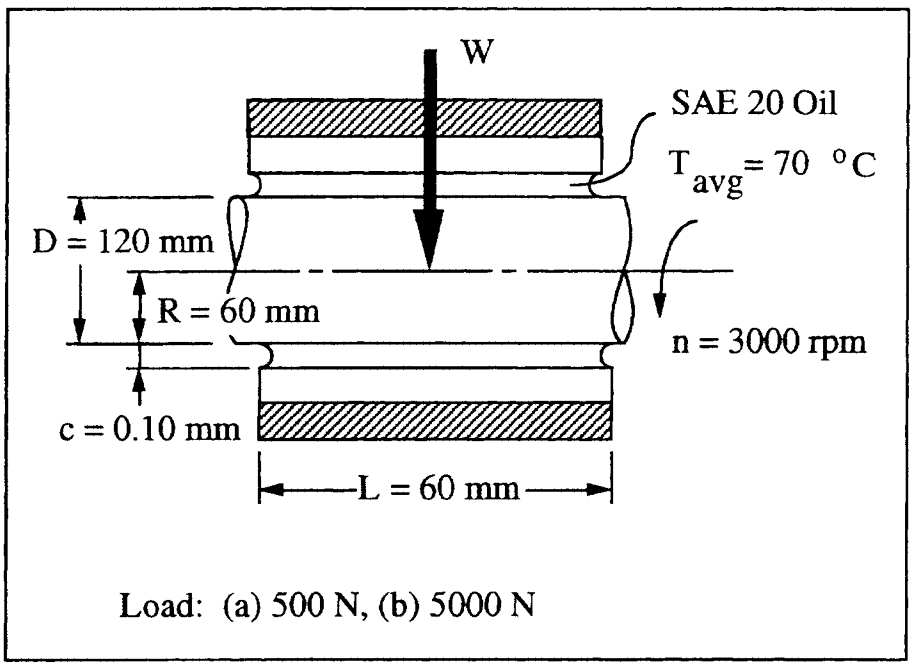
Comment: In an actual situation, we would need to verify that when the dissipating 0.66 hp/bearing, the average oil temperature in the bearing would be consistent with the value of viscosity used in the calculations.

SOLUTION (13.18)

Known: A journal bearing has a known diameter, length, and diametral clearance. The journal has a given rotational speed and is lubricated with a SAE 20 oil at a known average temperature. The bearing load is (a) 500 N and (b) 5000 N.

Find: Determine the power loss and the friction torque.

Schematic and Given Data:



Assumptions:
1. The lubricant is supplied to the bearing at atmospheric pressure.
2. The influence on flow rate of any oil holes or grooves is negligible.
3. Viscosity is assumed to be constant, and to correspond to the average temperature of the oil flowing to and from the bearing.

Analysis:

(a) 500 N

1. $$S = \left(\frac{R}{c}\right)^2 \frac{\mu n}{P} = \frac{\left(\frac{60\text{ mm}}{0.1\text{ mm}}\right)^2 \left(\frac{0.0125\text{ Ns}}{\text{m}^2}\right)\left(50\ \frac{\text{rev}}{\text{s}}\right)}{\left(\frac{500\text{ N}}{(0.060)(0.120)\text{ m}^2}\right)} = 3.24$$

2. From Fig. 13.14, $\frac{R}{c} f = 65$, $f = 65\,\frac{c}{R} = 65\,\frac{0.1}{60} = 0.1083$

3. $T_f = WfR = 500\text{ N}\,(0.1083)(0.060\text{ m}) = 3.25\text{ N·m}$ ■

4. From Eq. (1.2), $\dot{W} = \frac{3.25\ (3000)}{9549} = 1.02\text{ kW}$ ■

5. From Fig. 13.13, $\frac{h_o}{c} = 0.86$, hence $h_o = 0.86(0.1) = 0.086$ mm. ■

(b) 5000 N

1. For W = 5000 N, S = 0.324, $\frac{R}{c} f = 8$ or $f = 8\,\frac{c}{R} = 8\,\frac{0.1}{60} = 0.0133$

2. $T_f = 5000\ (0.0133)\ (0.060) = 4.0\text{ N·m}$ ■

3. $\dot{W} = \frac{4\ (3000)}{9549} = 1.26\text{ kW}$ ■

4. From Fig. 13.13, $\frac{h_o}{c} = 0.41$, hence $h_o = 0.041$ mm ■

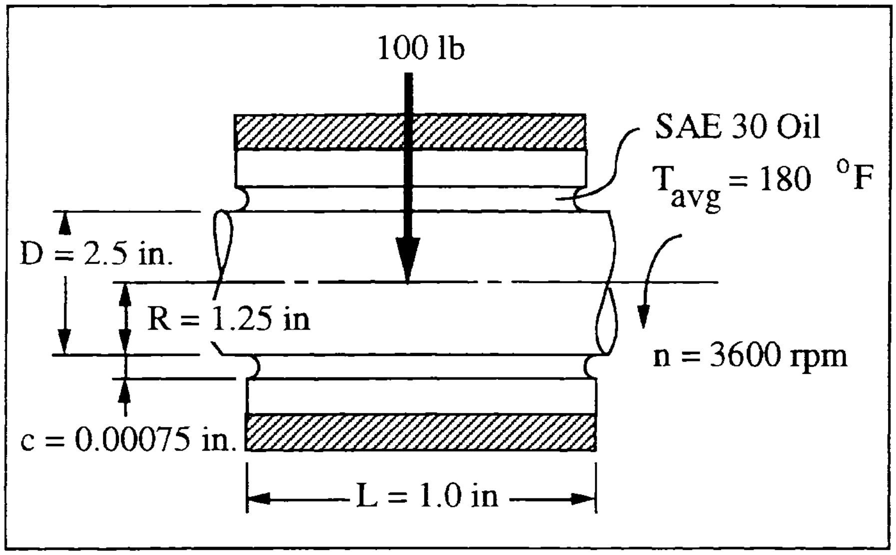
Comment: It is important to remember that the foregoing analysis involving the Raimondi and Boyd charts applies only to steady-state operation with a load that is fixed in magnitude and direction. Bearings subjected to rapidly fluctuating loads (as engine crank-shaft bearings) can carry much greater instantaneous peak loads than the steady-state analysis would indicate because there is not enough time for the oil film to be squeezed out before the load is reduced. This is sometimes called the squeeze-film phenomenon. It causes an apparent "stiffening" of the oil film as it is squeezed increasingly thin. The squeeze-film effect is the primary lubricating mechanism in engine wrist pin bearings (shown in Fig. 13.25) where the relative motion is oscillatory, over a small angle.

SOLUTION (13.19)

Known: A journal rotates at given speed and with SAE 30 oil at 180 °F in a journal bearing with known diameter, length, and diametral clearance.

Find: Determine the power loss.

Schematic and Given Data:



Assumptions:

1. The lubricant is supplies at atmospheric pressure.
2. There is no lubricant flow in the axial direction.
3. The influence on oil flow of oil holes or grooves is negligible.
4. The oil viscosity is assumed constant corresponding to the average temperature.

Analysis:

1. From Fig 13.6, for SAE 30 oil at 180 °F, μ = 1.6 μ reyn.

2. $P = W/LD = 100/(1)(2.5) = 40$ psi

3. $S = \left(\frac{R}{c}\right)^2 \left(\frac{\mu n}{P}\right)$

$$= \left(\frac{1.25 \text{ in.}}{0.00075 \text{ in.}}\right)^2 \frac{(1.6 \times 10^{-6} \text{ reyn})(60 \text{ rps})}{40 \text{ psi}} = 6.67$$

4. (a) From Fig. 13.13, $h_o/c = 0.88$ for L/D = 0.4, hence $h_o = 0.00066$ in. ■

5. From Fig. 13.14 for S = 6.67, $= \frac{R}{c} f = 124$, hence $f = 124 \frac{c}{R} = 0.0744$

6. $T_f = fWR = (0.0744)(100 \text{ lb})(1.25 \text{ in.}) = 9.3$ lb in. = 0.775 ft lb

7. $\text{Power loss} = \frac{Wf(\pi D/12)\, n}{33000} = \frac{100\,(.0744)(\pi)(2.5)(3600)}{12\,(33000)} = 0.53 \text{ hp}$ ■

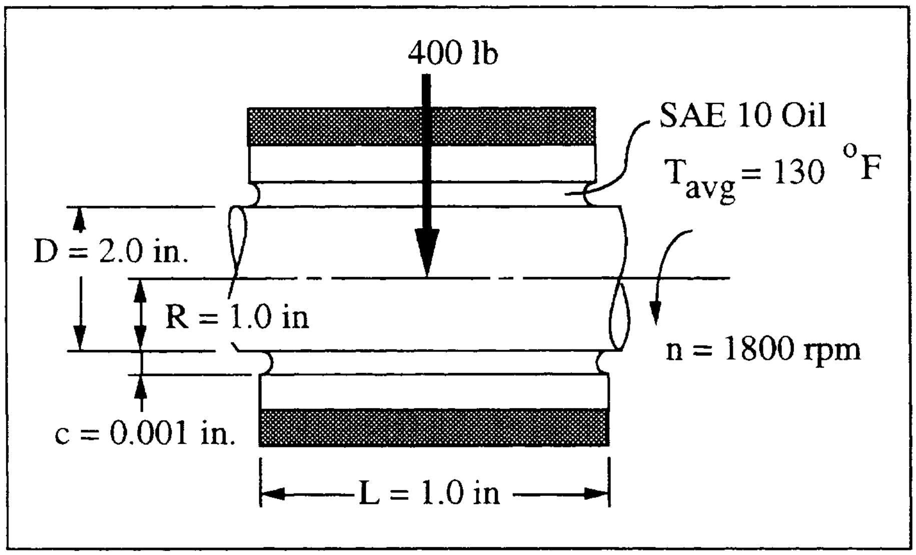
Comment: The bearing is not operating in the optimum zone. To operate in this zone, the amount of clearance should be increased.

SOLUTION (13.20)

Known: A journal rotates at a given speed with SAE 10 oil at 130 °F in a journal bearing with given diameter, and length, and having a known diametral clearance which supports a known load.

Find: Estimate (a) minimum oil film thickness, (b) coefficient of friction, (c) maximum film pressure, (d) angle between load direction and minimum film thickness, (e) angle between load direction and termination of film, (f) angle between load direction and maximum film pressure, (g) total circumferential oil flow rate, and (h) side or leakage flow rate. Also, the effect of clearance.

Schematic and Given Data:



Assumptions:

1. The lubricant is supplied to the bearing at atmospheric pressure.
2. The influence on flow rate of any oil holes or grooves is negligible.
3. Viscosity is assumed to be constant, and to correspond to the average temperature of the oil flowing to and from the bearing.

Analysis:

1. For SAE 10 oil at 130 °F, $\mu = 2.7 \times 10^{-6}$ reyn
2. $P = W/LD = 400/(2)(1) = 200$ psi
3. $S = \left(\frac{R}{c}\right)^2 \left(\frac{\mu n}{P}\right)$

$$= \left(\frac{1}{0.001}\right)^2 \frac{(2.7 \times 10^{-6})(30)}{200} = 0.41$$

4. (a) From Fig. 13.13, $h_o/c = 0.47$, hence $h_o = 0.00047$ in. ■

(b) From Fig. 13.14, $\frac{R}{c} f = 10$, hence $f = 10 \frac{0.001}{1}$

hence f = 0.01 ■

(c) From Fig. 13.15, $P/p_{max} = 0.39$, hence

$p_{max} = 200/0.39 = 513$ psi ■

(d) From Fig. 13.16, $\phi = 52^\circ$, ■

(e) From Fig. 13.17, $\theta_{p_o} = 70^\circ$,

(f) $\theta p_{max} = 17.5^\circ$ ■

(g) From Fig 13.18, $\frac{Q}{RcnL} = 4.65$,

$Q = 4.65\ (1)(0.001)(30)(1)$

$Q = 0.14$ in.^{3}/sec ■

(h) From Fig. 13.19, $Q_s/Q = 0.68$, hence

$Q_s = 0.68\ (0.14) = 0.09$ in.^{3}/sec. ■

A larger clearance is needed to be in the optimum zone. This would increase h_o with no increase in f.

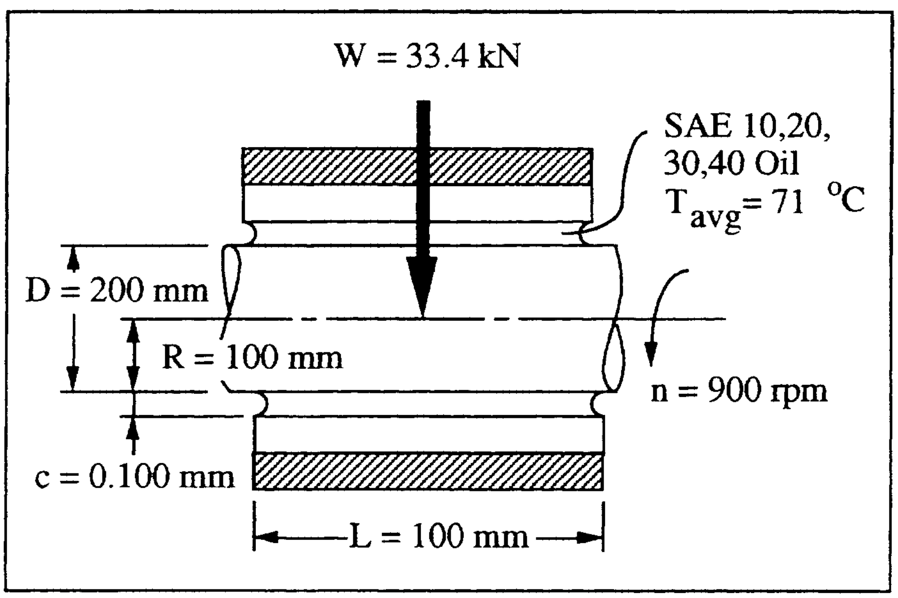
Comment: It is important to remember that the foregoing analysis involving the Raimondi and Boyd charts applies only to steady-state operation with a load that is fixed in magnitude and direction.

SOLUTION (13.21)

Known: A journal rotates at a given speed with specified SAE oils all at 71 °C in a journal bearing with known diameter, length and radial clearance.

Find: Plot friction power and h_o versus viscosity.

Schematic and Given Data:

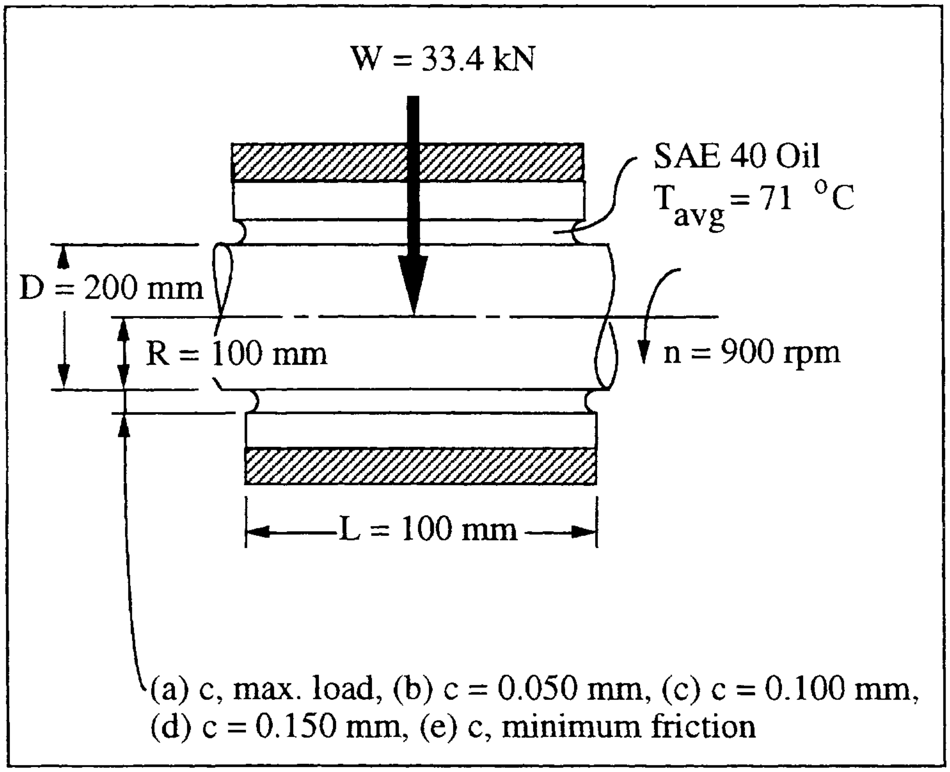


SOLUTION (13.22)

Known: A journal rotates at a given speed with a specified SAE oil at 71 °C in a journal bearing with known diameter and length and having specified radial clearances (a, b, c, d, and e).

Find: Plot friction power and h versus radial clearance.

Schematic and Given Data:



Assumptions:

1. The lubricant is supplied to the bearing at atmospheric pressure.
2. The influences on flow rate of any oil holes or grooves is negligible.
3. Viscosity is assumed to be constant, and to correspond to the average temperature of the oil flowing to and from the bearing.

Analysis:

1. This problem is a continuation of Problem 13.21:

   μ = 22.5 mPa·S, P = 1.67 MPa, L/D = 0.5, $\frac{\mu n}{P} = 0.202 \times 10^{-6}$

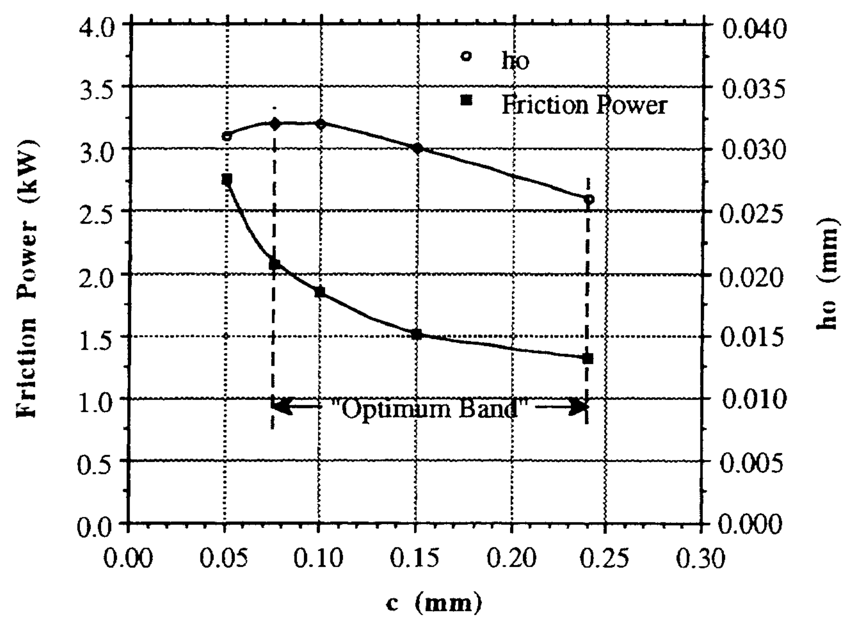
2. $\dot{W} = 314.8 f$ (As shown in Problem 13.21)

3. A table is constructed:

c mm	R/c	S $(\frac{R}{c})^2\frac{\mu n}{P}$	$\frac{R}{c}f$	f	Friction Power kW	h_o/c	h_o mm
0.050	2000	0.808	17.5	0.0088	2.77	0.610	0.031
0.100	1000	0.202	5.8	0.0058	1.84	0.320	0.032
0.150	667	0.090	3.2	0.0048	1.51	0.197	0.030
0.240	416	* 0.035	1.75	0.0042	1.32	0.110	0.026
0.076	1316	* 0.350	8.7	0.0066	2.08	0.425	0.032

* "Optimum Band"

4. The values in the table are plotted.



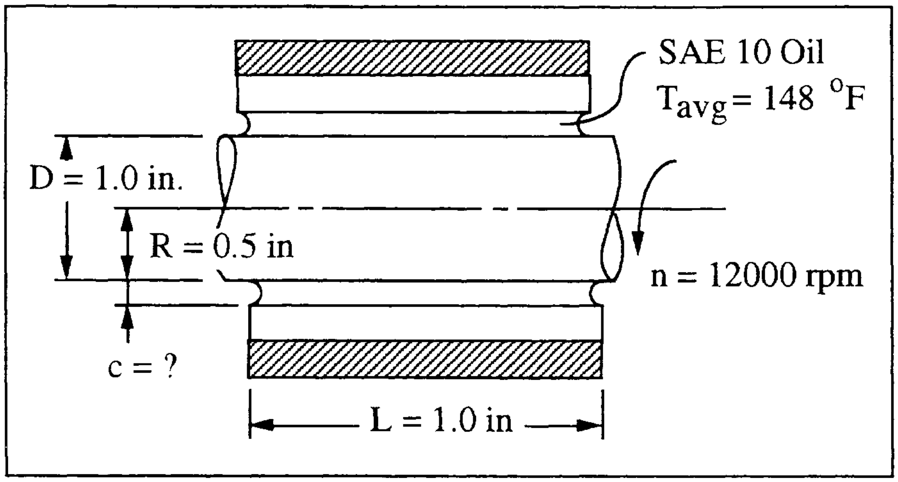
Comment: It is important to remember that the foregoing analysis involving the Raimondi and Boyd charts applies only to steady-state operation with a load that is fixed in magnitude and direction.

SOLUTION (13.23)

Known: A journal rotates at a given speed and with SAE 10 oil at 148 °F in a journal bearing with known diameter, length and minimum oil film thickness.

Find: Determine (a) the diametral clearance giving the greatest load carrying capacity and the corresponding (b) load capacity, and (c) friction power loss.

Schematic and Given Data:



Assumptions:

1. Excellent oil filtration is provided and the journal roughness is less than 32 micro-inches rms allowing a minimum oil film thickness of 0.0003 in.
2. The lubricant is supplied to the bearing at atmospheric pressure.
3. The influence on flow rate of any oil holes or grooves is negligible.
4. Viscosity is assumed to be constant, and to correspond to the average temperature of the oil flowing to and from the bearing.

Analysis:

(a) Select the "max load" point on Fig 13.13, where $h_0/c = 0.535$, then $c = h_0/0.535 = 0.0003/0.535 = 0.000561$, and diametral clearance $= 2c = 0.0011$ in. ■

(b) From Fig. 13.13, $S = 0.21 = \left(\frac{R}{c}\right)^2\left(\frac{\mu n}{P}\right)$. From Fig. 13.6, $\mu = 1.8 \times 10^{-6}$ reyn. With n = 200 rps and R = 0.5 in., we have

$$0.21 = \left(\frac{0.5}{0.000561}\right)^2\left(\frac{1.8 \times 10^{-6} \times 200}{P}\right).$$ Hence, P = 1362 psi. Since $P = \frac{W}{(1)(1)}$

we have W = 1362 lb, or rounding to a normal value, W = 1360 lb ■

(c) From Fig. 13.14, $\frac{R}{c} f = 4.8$, hence $f = 4.8 \frac{c}{R} = 4.8 \frac{0.000561}{0.5} = 0.0054$

$$\text{Friction power} = \frac{Wf(\pi D/12)\, n}{33000} = \frac{1362\,(.0054)(\pi)(12000)}{12\,(33000)} = 0.70 \text{ hp}$$ ■

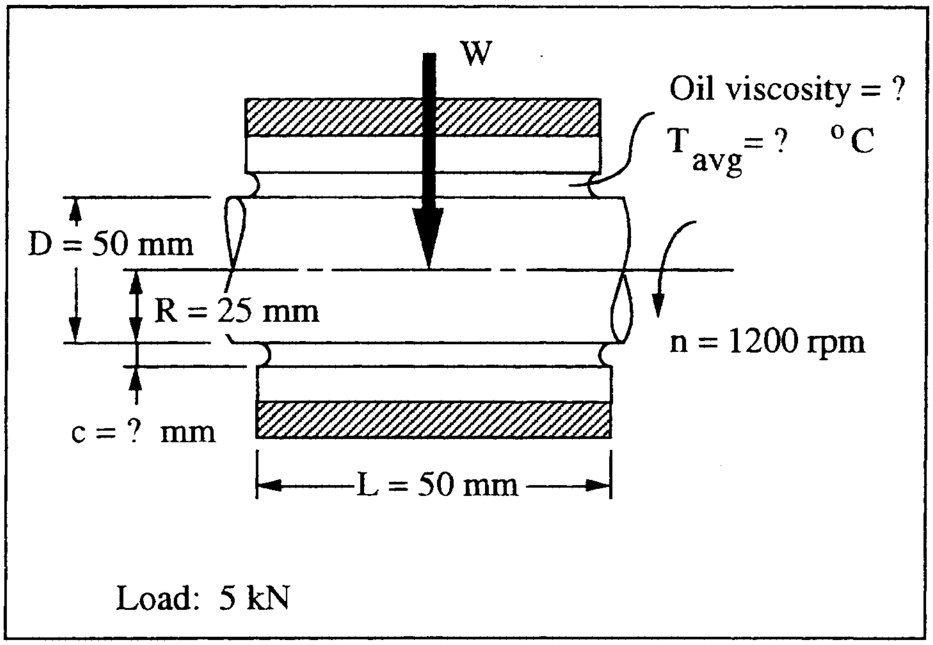
Comment: The foregoing analysis involving the Raimondi and Boyd charts applies only to steady-state operation with a load that is fixed in magnitude and direction.

SOLUTION (13.24)

Known: A journal rotates at a given speed and supports a known radial load producing a minimum friction in a journal bearing with known diameter, length and minimum film thickness.

Find: Determine the radial clearance, oil viscosity, coefficient of friction and friction power loss.

Schematic and Given Data:



Assumptions:

1. The lubricant is supplied to the bearing at atmospheric pressure.
2. The influence on flow rate of any oil holes or grooves is negligible.
3. Viscosity is assumed to be constant, and to correspond to the average temperature of the oil flowing to and from the bearing.

Analysis:

1. From Fig. 13.13, the bearing is operating at $S = 0.08$, $h_o/c = 0.3$. Since $h_o = 0.025$ (given), $c = 0.025/0.3 = 0.083$ mm. ■

2. $$S = 0.08 = \left(\frac{R}{c}\right)^2 \frac{\mu n}{P} = \left(\frac{25}{0.083}\right)^2 \frac{\mu(20)}{5000/(0.050)^2}$$

   Hence, $\mu = 0.88 \frac{\text{Ns}}{\text{m}^2} = 0.088$ Pa·s = 88 mPa·s ■

3. From Fig. 13.14, $\frac{R}{c}f = 2.4$, hence $f = 2.4\,\frac{0.083}{25} = 0.0080$ ■

4. Power loss = $fW(2\pi Rn)$ = (0.008)(5000 N) [2π(0.025 m)(20 rev/sec)] = 126 W (or 0.17 hp) ■

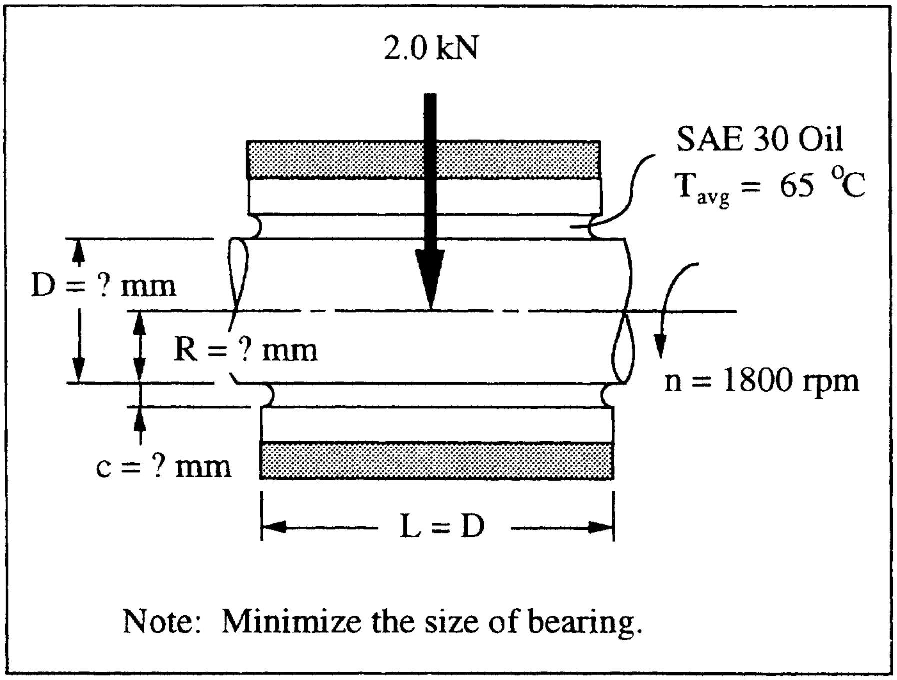
Comment: The foregoing analysis involving the Raimondi and Boyd charts applies only to steady-state operation with a load that is fixed in magnitude and direction.

SOLUTION (13.25D)

Known: A journal shaft has a given rotational speed and is lubricated with a SAE 30 oil at a known average film temperature. The journal bearing is to carry a specified load and have L/D = 1.

Find: Determine: (a) values of L and D, (b) values of c for the two edges of the optimum zone in Fig. 13.13, (c) whether the value of c for minimum friction satisfies Trumpler's criterion.

Schematic and Given Data:



Assumptions:
1. Bearing conditions are at steady state with the radial load fixed in magnitude and direction.
2. The lubricant is supplied to the bearing at atmospheric pressure.
3. The influence on flow rate of any oil holes or grooves is negligible.
4. Viscosity is assumed to be constant and to correspond to the average temperature of the oil flowing to and from the bearing.

Decisions:
1. From the 0.8 to 1.5 MPa range representative unit sleeve bearing loads for current practice given for gear reducer bearings in Table 13.2, select the unit load P = 1.5 MPa to provide the smallest bearing consistent with current practice.
2. Bearing parameters are selected so as to operate in the optimum operating range.

Design Analysis:

(a) From Table 13.2, select $P = 1.5\text{ MPa} = \frac{W}{LD} = \frac{2000\text{ N}}{L^2}$. Hence, L = 36.5 mm.

Using L = D = 37 mm. ■

(b)

1. From Fig. 13.6, for SAE 30 oil at 65 °C, μ = 22 mPa·s.
2. From Fig. 13.13, optimum zone edges are at S = 0.082 and S = 0.21.
3. This corresponds for minimum friction to:

$$0.082 = \left(\frac{R}{c}\right)^2(\mu n/P) = \left(\frac{18.5}{c}\right)^2\left(\frac{0.022\text{ Pa·s}\cdot 30\text{ rps}}{1.5\times 10^6\text{ Pa}}\right)\text{ or } c = 0.043\text{ mm}$$ ■

4. and for maximum load to: $0.21 = \left(\frac{18.5}{c}\right)^2\left(\frac{0.022\cdot 30}{1.5\times 10^6}\right)$, or c = 0.029 mm ■

(c)

1. Check the minimum clearance (c = 0.029 mm) with Eq. (13.15):
   $h_o \geq 0.005 + 0.00004\,(37) = 0.0065$ mm
2. Trumpler suggests SF = 2, meaning W = 2.0 × 2 = 4 kN.
3. Doubling P reduces S by half, giving S = 0.04. Then $h_o/c = 0.19$ or $h_o = 0.19\,(0.043) = 0.0082$ mm.
4. Since 0.0082 > 0.0065, the criterion is satisfied. ■

Comments:

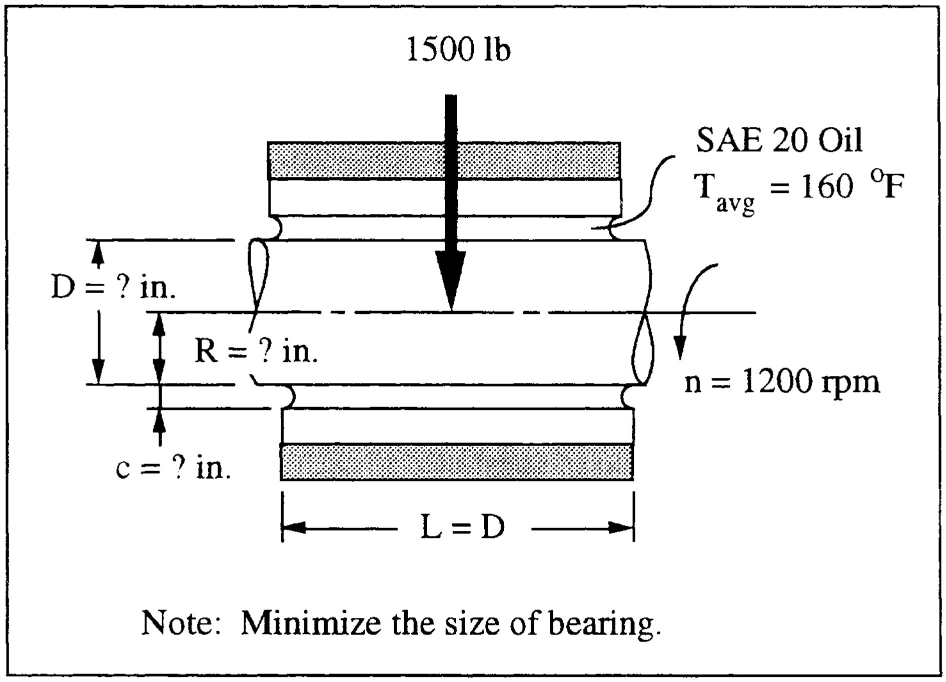
1. With the gravity force of the rotor loading the bearing only at the bottom, oil should be admitted and distributed at the top. Axial oil distribution could be accomplished with a groove, as in Fig. 13.23. Since the entire top of the bearing is never loaded, this groove could be very wide, perhaps encompassing the entire top 180°. This would give a 180° partial bearing, with the advantage of reducing viscous drag at the top. Special Raimondi and Boyd curves for partial bearings would then apply.
2. It is especially important that all oil passages be clean at the time of assembly. An appropriate oil filter should be provided.
3. It is unfortunate for the bearing that its load at rest and during starting and stopping is as great as the running load. However, since this load is under 2 MPa, and assuming neither frequent nor prolonged operation at low speed is anticipated, this should be an acceptable situation.
4. Some large turbines use hydrostatic bearings to avoid boundary lubrication during starting and stopping. In some cases the high-pressure pump can be turned off at operating speed, and hydrodynamic lubrication allowed to take over. (A low-pressure pump would normally remain on to provide a positive oil supply, as specified in the sample problem.)

SOLUTION (13.26D)

Known: A minimum size journal shaft has a given rotational speed and is lubricated with a SAE 20 oil at a known average film temperature. The journal bearing is to carry a specified load and have L/D = 1.

Find: Determine: (a) values of L and D, (b) values of c for the two edges of the optimum zone in Fig. 13.13, (c) whether the value of c for minimum friction satisfies Trumpler's criterion.

Schematic and Given Data:



Assumptions:

1. Bearing conditions are at steady state with the radial load fixed in magnitude and direction.
2. The lubricant is supplied to the bearing at atmospheric pressure.
3. The influence on flow rate of any oil holes or grooves is negligible.
4. Viscosity is assumed to be constant and to correspond to the average temperature of the oil flowing to and from the bearing.

Decisions:

1. From the 120 to 250 psi range representative unit sleeve bearing loads for current practice given for gear reducer bearings in Table 13.2, select the unit load P = 250 psi to provide the smallest bearing consistent with current practice.
2. Bearing parameters are selected so as to operate in the optimum operating range.

Design Analysis:

(a) From Table 13.2, select $P = 250 \text{ psi} = \frac{W}{LD} = \frac{1500 \text{ lb}}{L^2}$. Hence, L = 2.45 in.

Using L = D = 2.5 in. ■

(b)

1. From Fig. 13.6, for SAE 20 oil at 160 °F, $\mu = 1.7$ μreyn.
2. From Fig. 13.13, optimum zone edges are at S = 0.082 and S = 0.21.
3. This corresponds for minimum friction to:

$$0.082 = \left(\frac{R}{c}\right)^2(\mu n/P) = \left(\frac{1.25}{c}\right)^2\left(\frac{1.7 \times 10^{-6}\ \text{reyn} \cdot 20\ \text{rps}}{250\ \text{psi}}\right) \text{ or } c = 1.6 \times 10^{-3}\ \text{in.}$$ ■

4. and for maximum load to: $0.21 = \left(\frac{1.25}{c}\right)^2\left(\frac{1.7 \times 10^{-6} \cdot 20}{250}\right)$, or $c = 1 \times 10^{-3}$ in. ■

(c)

1. Check the minimum clearance ($c = 1 \times 10^{-3}$ in.) with Eq. (13.15):
   $h_o \geq 0.0002 + 0.00004\ (2.5) = 3 \times 10^{-4}$ in.
2. Trumpler suggests SF = 2, meaning W = 1500 lb × 2 = 3000 lb.
3. Doubling P reduces S by half, giving S = 0.041. Then $h_o/c = 0.19$ or
   $h_o = 0.19\ (1.6 \times 10^{-3}\ \text{in.}) = 3.04 \times 10^{-4}$ in.
4. Since 3.04×10^{-4} in. > 3.0×10^{-4} in., the criterion is satisfied. ■

Comments:

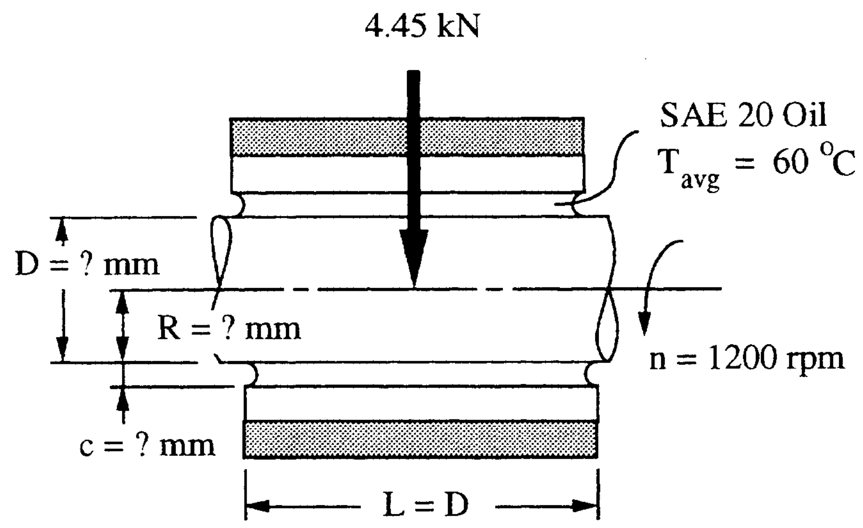
1. With the gravity force of the rotor loading the bearing only at the bottom, oil should be admitted and distributed at the top. Axial oil distribution could be accomplished with a groove, as in Fig. 13.23. Since the entire top of the bearing is never loaded, this groove could be very wide, perhaps encompassing the entire top 180°. This would give a 180° partial bearing, with the advantage of reducing viscous drag at the top. Special Raimondi and Boyd curves for partial bearings would then apply.
2. It is especially important that all oil passages be clean at the time of assembly. An appropriate oil filter should be provided.
3. It is unfortunate for the bearing that its load at rest and during starting and stopping is as great as the running load. However, since this load is under 300 psi, and assuming neither frequent nor prolonged operation at low speed is anticipated, this should be an acceptable situation.
4. Some large turbines use hydrostatic bearings to avoid boundary lubrication during starting and stopping. In some cases the high-pressure pump can be turned off at operating speed, and hydrodynamic lubrication allowed to take over. (A low-pressure pump would normally remain on to provide a positive oil supply, as specified in the sample problem.)

SOLUTION (13.27)

Known: A journal shaft has a given rotational speed and is lubricated with a SAE 20 oil at a known average film temperature. The journal bearing is to carry a specified load and have L/D = 1.

Find: Determine: (a) values of L and D, (b) values of c for the two edges of the optimum zone in Fig. 13.13, (c) whether the value of c for minimum friction satisfies Trumpler's criterion.

Schematic and Given Data:



Assumptions:

1. Bearing conditions are at steady state with the radial load fixed in magnitude and direction.
2. The lubricant is supplied to the bearing at atmospheric pressure.
3. The influence on flow rate of any oil holes or grooves is negligible.
4. Viscosity is assumed to be constant and to correspond to the average temperature of the oil flowing to and from the bearing.

Decisions:

1. From the 0.8 to 1.5 MPa range representative unit sleeve bearing loads for current practice given for gear reducer bearings in Table 13.2, select the unit load P = 1.5 MPa to provide the smallest bearing consistent with current practice.
2. Bearing parameters are selected so as to operate in the optimum operating range.

Design Analysis:

(a) From Table 13.2, select $P = 1.5\text{ MPa} = \frac{W}{LD} = \frac{4450\text{ N}}{L^2}$. Hence, L = 54.5 mm.

Using L = D = 55 mm. ■

(b)

1. From Fig. 13.6, for SAE 20 oil at 60 °C, μ = 18 mPa·s.
2. From Fig. 13.13, optimum zone edges are at S = 0.082 and S = 0.21.
3. This corresponds to;

$$0.082 = \left(\frac{R}{c}\right)^2(\mu n/P) = \left(\frac{27.5}{c}\right)^2\left(\frac{0.018\text{ Pa·s}\cdot 20\text{ s}^{-1}}{1.5\times 10^6\text{ Pa}}\right) \text{ or } c = 0.047\text{ mm}$$ ■

4. and to; $0.21 = \left(\frac{27.5}{c}\right)^2 \left(\frac{0.018 \cdot 20}{1.5 \times 10^6}\right)$, or c = 0.029 mm ■

(c)

1. Check the minimum clearance (c = 0.029 mm) with Eq. (13.15):
   $h_o \geq 0.005 + 0.00004\ (55) = 0.0072$ mm
2. Trumpler suggests SF = 2, meaning W = 4.45 × 2 = 8.9 kN.
3. Doubling P reduces S by half, giving S = 0.041. Then $h_o/c = 0.19$ or $h_o = 0.19\ (0.047) = 0.0089$ mm.
4. Since 0.0089 > 0.0072, the criterion is satisfied. ■

Comments:

1. With the gravity force of the rotor loading the bearing only at the bottom, oil should be admitted and distributed at the top. Axial oil distribution could be accomplished with a groove, as in Fig. 13.23. Since the entire top of the bearing is never loaded, this groove could be very wide, perhaps encompassing the entire top 180°. This would give a 180° partial bearing, with the advantage of reducing viscous drag at the top. Special Raimondi and Boyd curves for partial bearings would then apply.
2. It is especially important that all oil passages be clean at the time of assembly. An appropriate oil filter should be provided.
3. It is unfortunate for the bearing that its load at rest and during starting and stopping is as great as the running load. However, since this load is under 2 MPa, and assuming neither frequent nor prolonged operation at low speed is anticipated, this should be an acceptable situation.
4. Some large turbines use hydrostatic bearings to avoid boundary lubrication during starting and stopping. In some cases the high-pressure pump can be turned off at operating speed, and hydrodynamic lubrication allowed to take over. (A low-pressure pump would normally remain on to provide a positive oil supply, as specified in the sample problem.)

SOLUTION (13.28D)

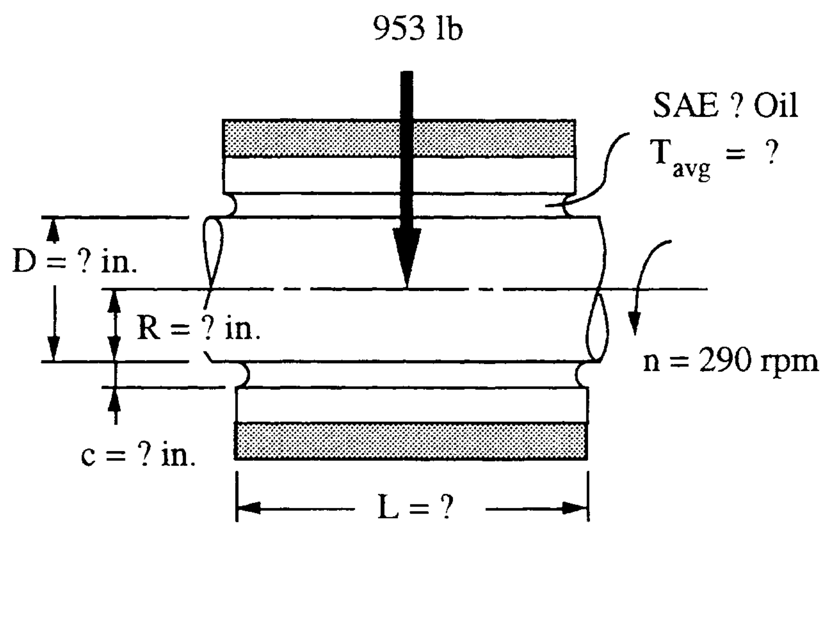
Known: A journal has a given rotational speed and supports a known radial load.

Find:

(a) Determine the bearing geometry, material, lubricant, and average oil temperature.

(b) Plot f, h_o, ΔT, Q and Q_s versus radial clearance, c.

Schematic and Given Data:



Note: Minimize the size of bearing.

Assumptions/Decisions:

1. The maximum clearance is 0.001 in. greater than the minimum clearance.
2. Choose L/D = 1, to minimize the use of radial space--see Section 13.13.
3. Choose c/R = 0.0015 as suggested in Section 13.13
4. The bearing will operate in the mid-optimum range of Fig. 13.13.
5. Select an average oil film temperature of 125 °F, and SAE 40 oil. The temperature can be controlled by an external oil cooler.
6. Select babbitt or bronze material for the bearing.

Design Analysis:

Given: W = 953 lb, n = 290/60 = 4.83 rps

1. We assumed that L/D = 1.
2. From Table 13.2, for gear reducers we use a unit load 120 < P < 250 psi.
3. The unit load, $P = \frac{W}{LD} = \frac{953}{L^2}$ should be between 120 and 250, hence L should be

   between 1.95 in. and 2.82 in.

4. Choose L = D = 2.5 in., which gives $P = \frac{953}{(2.5)^2} = 152$ psi

5. As assumed, choose $\frac{c}{R} = 0.0015$, hence c = 0.001875 in.

6. Find the viscosity for assumed mid-optimum range:

$$S = 0.14 = \left(\frac{1}{0.0015}\right)^2 \frac{\mu(4.83)}{152}. \text{ Hence } \mu = 9.99 \times 10^{-6}$$

7. As assumed, the average oil film temp = 125 °F.

8. As assumed, use SAE 40 oil, for which $\mu = 8 \times 10^{-6}$ reyn.

9. $$S = \left(\frac{R}{c}\right)^2 \frac{\mu n}{P} = \left(\frac{1.25}{c}\right)^2 \frac{(8 \times 10^{-6})(4.83)}{152} = \frac{3.97 \times 10^{-7}}{c^2}$$

10. Construct a table to investigate bearing performance for a range of radial clearances.

1	2	3	4	5	6	7	8	9	10	11
c	S	h_o/c	h_o	$\frac{R}{c}f$	f	$\frac{Q}{Rcn L}$	Q	Q_s/Q	Q_s	ΔT, deg F
.0005	1.59	.91	.000455	32.0	.0128	3.33	.0251	.12	.0030	1404.0
.001	.397	.69	.00069	8.2	.0066	3.8	.0573	.39	.022	99.0
.002	.0993	.345	.00069	2.7	.0043	4.4	.133	.725	.096	14.7
.004	.0248	.13	.00052	1.22	.0039	4.7	.284	.90	.256	5.0
.008	.0062	.03	.00024	0.53	.0034	4.8	.579	.965	.559	2.0
.0022	.08	.30	.00066	2.4	.0042	4.47	.148	.76	.112	12.3
.0013	.21	.54	.00070	4.8	.0050	4.1	.0804	.55	.044	37.4

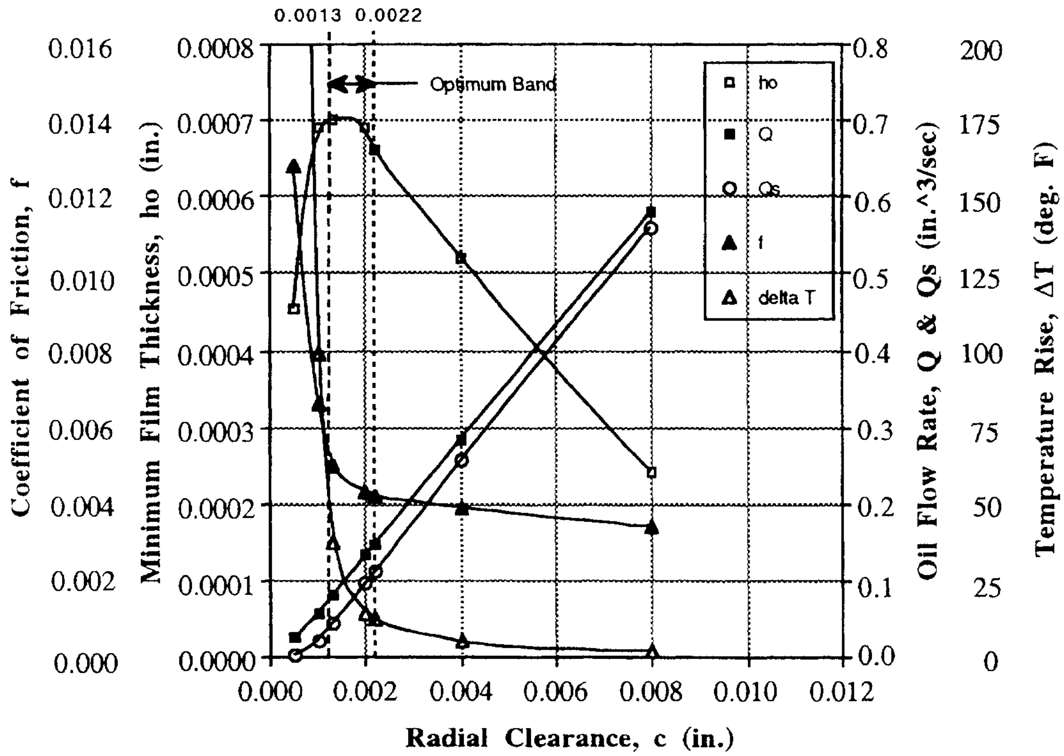
<u>Column</u>

1 Arbitrarily chosen
2 $S = 3.97 \times 10^{-7}/c^2$
3 Fig. 13.13
4 Col.3 × Col.1
5 Fig. 13.14
6 (Col.5) × (Col.1)/1.25
7 Fig. 13.18
8 (Col.7) × (Col.1) × (1.25) × (4.83) × (2.5)
9 Fig. 13.19
10 Col.8 × Col.9
11 -Heat Generated = $2\pi f W R N$
-Heat Absorbed by Oil = $Q_s \Delta T (c\rho)$
where, $c\rho = 110$ lb/(in.^{2} °F) (given on page 501)
-Equating: $\Delta T = 2\pi f W R N / Q_s c\rho$

$$\Delta T = \frac{2\pi(\text{Col.6})(953 \text{ lb})(1.25 \text{ in.})(4.83 \text{ rps})}{(\text{Col.10 in}^3/\text{s})(110 \frac{\text{lb}}{\text{in}^2 \text{ °F}})}$$

$\Delta T = 329 \times (\text{Col.6})/(\text{Col.10})$

11. Plot the tabular results versus radial clearance.



12. The proposed solution:
L = D = 2.5 in. ■
c = .0013 in./.0023 in. ■
Babbitt or bronze material ■
SAE 40 oil (assume 125 °F avg. temp.) ■

13. The power loss for the worst case of c = .0013 in., f = 0.005;

$$\text{Power loss} = \frac{(953\ \text{lb})(.005)(\frac{2.5}{12}\ \pi\ \text{ft})\ 290\ \text{rpm}}{33000} = .027\ \text{hp}$$

14. Check Trumpler's criterion for acceptable h_o, even after wear increases c to 0.0033 in.. Then, for W = 2 X 953 lb, c = 0.0033 in., S = 0.0182, $h_o/c = 0.1$. Hence $h_o = 0.00033$ in.

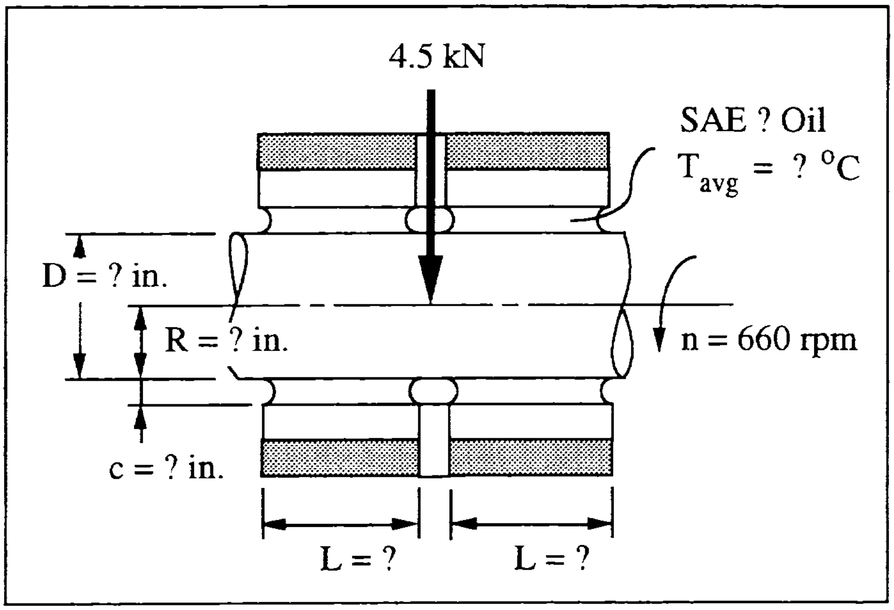
15. Trumpler requirement = 0.0002 + 0.00004 (2.5) = 0.0003 in. ($h_{o\ min}$). Hence the Trumpler criterion is satisfied.

SOLUTION (13.29D)

Known: A ring-oiled bearing journal has a given rotational speed and supports a known radial load.

Find: Determine the bearing geometry, material, lubricant, and surface finish.

Schematic and Given Data:



Assumptions:
1. The lubricant oil is pumped into the bearing at 1 atmosphere of pressure.
2. The oil viscosity is constant and equal to the viscosity of the oil at the average oil temperature.
3. The presence of the ring groove in the ring-oil bearing creates a system operating like two identical bearings.
4. The ring-oil bearing can be analyzed as two identical normal bearings each supporting half the total load.
5. The bearing is subjected to a relatively steady load.
6. The journal surface is fine ground with a peak-to-valley roughness not exceeding 0.005 mm.

Decisions:
1. Choose SAE 30 oil and an average oil film temperature of 50 °C.
2. Choose L/D = 1 and have a ring that divides the bearing into halves as shown in Fig. 13.22 and 13.24, each half having L/D = 0.5, and supporting a load of 2.25 kN.
3. Select operation at the middle of the optimum zone in Fig. 13.13.
4. Select a c/R of 0.0017.

Design Analysis:
1. From Fig. 13.6 with SAE 30 oil and the assumed average oil film temperature of 50 °C, we have μ = 44 mPa·s.
2. From Fig. 13.13 with L/D = 1 selecting nominal operation, we have S = 0.16 (middle of optimum zone).

3. With the selected value of c/R of 0.0017, we have:

$$S = \left(\frac{R}{c}\right)^2 \frac{\mu n}{P} \quad \text{or} \quad 0.16 = \left(\frac{1}{0.0017}\right)^2 \frac{(0.044 \ \text{Ns/m}^2)(11 \ \text{rev/s})}{2250 \ \text{N} / 0.5 \ D^2} \quad \text{from which,}$$

D = 0.0656 m, or D = 66 mm ■

4. L(each half-bearing) = 33 mm ■

5. $c = 0.0017 \frac{66}{2} = 0.056$ mm (nominal value) ■

6. From Fig. 13.13, $h_o/c = 0.28$, hence $h_o = 0.016$ mm ■

7. But to check Eq. (13.15), we use doubled normal load which gives S = 0.16/2 = 0.08, $h_o/c = 0.185$, and $h_o = 0.0104$ mm

8. We compare this with

   $h_o \geq 0.005 + (0.00004)\ 66 = 0.0076$ mm. Since 0.0104 > 0.0076, the criterion is satisfied.

9. The specified peak-to-valley roughness ≤ 0.005 mm is acceptable. ■

10. Bearing unit load, $P = \frac{W}{LD} = \frac{4500 \ \text{N}}{(66 \ \text{mm})^2} = 1.03$ MPa. This value is within the range given in Table 13.2. Hence, the bearing geometry is acceptable.

11. As decided, the bearing material could be babbitt or a copper alloy such as cast bronze. ■

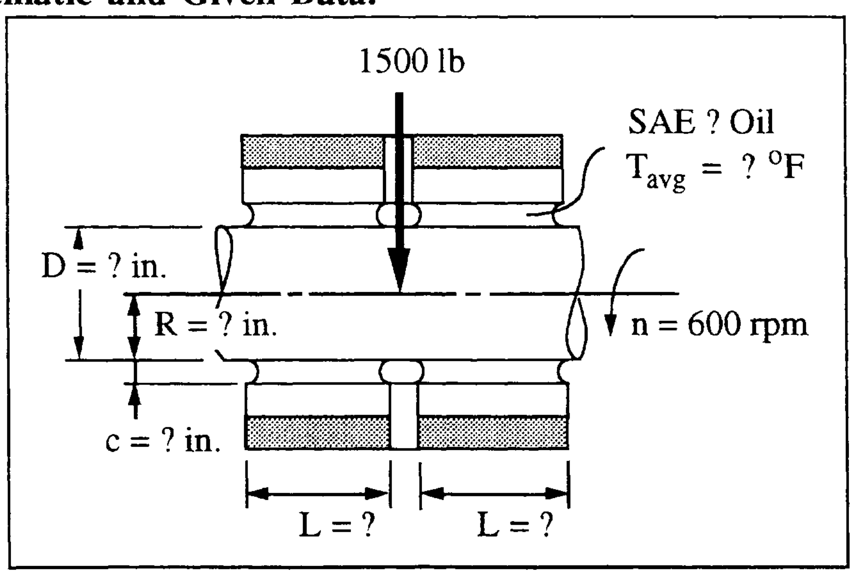
Comment: The foregoing analysis involving the Raimondi and Boyd charts applies only to steady-state operation with a load that is fixed in magnitude and direction.

SOLUTION (13.30D)

Known: A journal for a ring-oiled bearing has a given rotational speed and supports a known radial load.

Find: Determine the bearing geometry, material, lubricant, and surface finish.

Schematic and Given Data:



Assumptions:
1. The lubricant oil is pumped into the bearing at 1 atmosphere of pressure.
2. The oil viscosity is constant and equal to the viscosity of the oil at the average oil temperature.
3. The presence of the ring groove in the ring-oil bearing creates a system operating like two identical bearings.
4. The ring-oil bearing can be analyzed as two identical normal bearings each supporting half the total load.
5. The bearing is subjected to a relatively steady load.

Decisions:
1. Choose SAE 40 oil and an average oil film temperature of 140 °F.
2. Choose L/D = 1 to minimize the use of radial space, but since the bearing is an oil-ring type bearing having a ring that divides the bearing into halves, each half will have L/D = 0.5, and should support a load of 750 lb.
3. Select operation at the middle of the optimum zone in Fig. 13.13.
4. Select a c/R of 0.0015, as suggested in Section 13.13.
5. The journal surface is fine ground with a peak-to-valley roughness not exceeding 0.0002 in.
6. Choose babbitt or cast bronze for the bearing material.

Design Analysis:
1. From Fig. 13.6 with SAE 40 oil and the assumed average oil film temperature of 140 °F, we have $\mu = 5\ \mu$ reyn.
2. From Fig. 13.13 with L/D = 0.5 selecting nominal operation, we have S = 0.16 (middle of optimum zone).
3. With the selected value of c/R of 0.0015, we have:

$$S = \left(\frac{R}{c}\right)^2 \frac{\mu n}{P} \quad \text{or} \quad 0.16 = \left(\frac{1}{0.0015}\right)^2 \frac{(5 \times 10^{-6}\ \text{reyn})(10\ \text{rev/s})}{750\ \text{lb} / 0.5\ D^2}$$

   Hence, D = 3.286 in. ■
4. L(each half-bearing) = 1.643 in. ■
5. c = 0.0015(1.643) = 0.0025 in. (nominal value) ■
6. From Fig. 13.13, $h_o/c = 0.28$, hence $h_o = 0.0007$ in. ■
7. But to check Trumpler's criterion Eq. (13.15), we use doubled normal load which gives S = 0.16/2 = 0.08, $h_o/c = 0.185$, and $h_o = 0.00046$ in.
8. We compare this with
   $h_o \geq 0.0002 + (0.00004)(3.286\text{ in.}) = 0.00033$ in. Since 0.00046 > 0.00033, the criterion is satisfied.
9. The specified peak-to-valley roughness ≤ 0.0002 in. is acceptable. ■
10. Bearing unit load, $P = \dfrac{W}{LD} = \dfrac{1500\ \text{lb}}{(3.286\ \text{in.})^2} = 138.9$ psi. This value is within the range given in Table 13.2 for most applications having relatively steady loads. Hence, the bearing geometry is acceptable.

11. As decided, the bearing material could be babbitt or a copper alloy such as cast bronze. ■

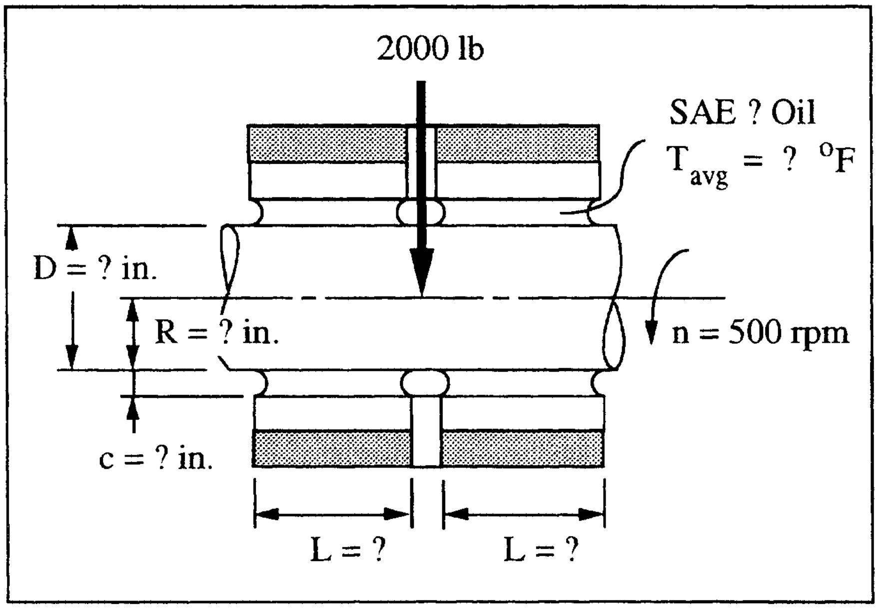
Comment: The foregoing analysis involving the Raimondi and Boyd charts applies only to steady-state operation with a load that is fixed in magnitude and direction.

SOLUTION (13.31D)

Known: A journal for a ring-oiled bearing has a given rotational speed and supports a known radial load.

Find: Determine the bearing geometry, material, lubricant, and surface finish.

Schematic and Given Data:



Assumptions:

1. The lubricant oil is pumped into the bearing at 1 atmosphere of pressure.
2. The oil viscosity is constant and equal to the viscosity of the oil at the average oil temperature.
3. The presence of the ring groove in the ring-oil bearing creates a system operating like two identical bearings.
4. The ring-oil bearing can be analyzed as two identical normal bearings each supporting half the total load.
5. The bearing is subjected to a relatively steady load.

Decisions:

1. Choose SAE 40 oil and an average oil film temperature of 140 °F.
2. Choose L/D = 1 to minimize the use of radial space, but since the bearing is an oil-ring type bearing having a ring that divides the bearing into halves, each half will have L/D = 0.5, and should support a load of 1000 lb.
3. Select operation at the middle of the optimum zone in Fig. 13.13.
4. Select a c/R of 0.0015, as suggested in Section 13.13.
5. The journal surface is fine ground with a peak-to-valley roughness not exceeding 0.0002 in.

6. Choose babbitt or cast bronze for the bearing material.

Design Analysis:

1. From Fig. 13.6 with SAE 40 oil and the assumed average oil film temperature of 140 ^oF, we have $\mu = 5\ \mu$ reyn.
2. From Fig. 13.13 with L/D = 0.5 selecting nominal operation, we have S = 0.16 (middle of optimum zone).
3. With the selected value of c/R of 0.0015, we have:

$$S = \left(\frac{R}{c}\right)^2 \frac{\mu n}{P} \quad \text{or} \quad 0.16 = \left(\frac{1}{0.0015}\right)^2 \frac{(5 \times 10^{-6} \text{ reyn})(8.33 \text{ rev/s})}{1000 \text{ lb} / 0.5\, D^2}$$

Hence, D = 4.158 in. ■

4. L(each half-bearing) = 2.079 in. ■
5. c = 0.0015(2.079 in.) = 0.0035 in. (nominal value) ■
6. From Fig. 13.13, $h_o/c = 0.28$, hence $h_o = 0.001$ in. ■
7. But to check Eq. (13.15), we use doubled normal load which gives S = 0.16/2 = 0.08, $h_o/c = 0.185$, and $h_o = 0.00065$ in.
8. We compare this with
   $h_o \geq 0.0002 + (0.00004)(4.158 \text{ in.}) = 0.00037$ in. Since 0.00065 > 0.00037, the criterion is satisfied.
9. The specified peak-to-valley roughness ≤ 0.0002 in. is acceptable. ■
10. Bearing unit load, $P = \frac{W}{LD} = \frac{2000 \text{ lb}}{(4.158 \text{ in.})^2} = 115.7$ psi. This value is not representative of unit sleeve bearing loads in current practice as it is at the low end of the range suggested by Table 13.2 for most applications having relatively steady loads. Although the bearing geometry is not optimal, however, it is acceptable.
11. As decided, the bearing material could be babbitt or a copper alloy such as cast bronze. ■

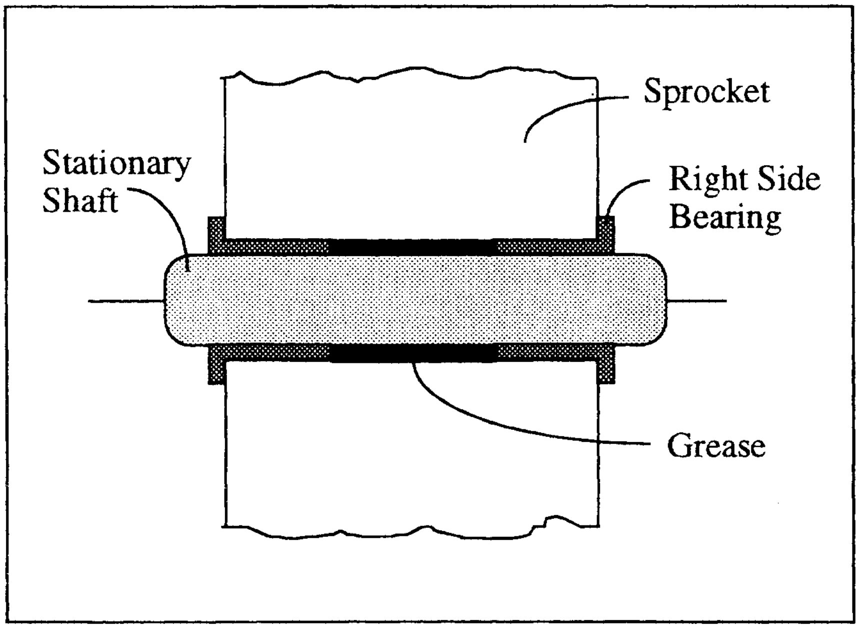
Comment: The foregoing analysis involving the Raimondi and Boyd charts applies only to steady-state operation with a load that is fixed in magnitude and direction.

SOLUTION (13.32D)

Known: Two journal bearings in a sprocket bore rotate at a given speed on a stationary shaft and support a known load.

Find: (a) Establish a satisfactory combination of bearing length, diameter and material, (b) Discuss factors which might influence a final choice of bearing geometry.

Schematic and Given Data:



Decisions/Assumptions:

1. Porous bronze bearings (Table 13.4) should be selected.
2. For a conservative design select PV = 0.70 MPa m/s.
3. Select a limiting dynamic P maximum of 5 MPa.
4. Use a value of L/D of 1.5.
5. Boundary lubrication exists because of the grease lubrication provided through the grease fitting.
6. The friction forces in the idler can be ignored.

Design Analysis:

(a) Determine suitable geometry.

1. To transmit 3.7 kW at 4 m/s, chain tension,

$$T = \frac{3700\ \text{W}}{4\ \text{m/s}} = 925\ \text{N}$$

2. This tension is present on both sides of the idler; hence, each of the two bearings (right and left sides) carries W = 925 N.

3. $n = \dfrac{4000\ \text{mm/s}}{\pi\,(122.3)\ \text{mm}} = 10.4$ Hertz
4. The figure shows a grease fitting; thus boundary lubrication is implied.
5. As decided, select porous bronze bearings (Table 13.4).
6. As decided, for a very conservative design, let PV at rated operation be 0.70 MPa·m/s (Table 13.4).

7. $PV = \frac{W}{LD}\pi Dn = \frac{925}{L}\pi\,(10.4 \text{ Hertz}) = 0.7 \times 10^6$. Hence, L = 0.043 m, or L = 43 mm ■

8. From Table 13.4, the limiting $P_{max} \approx 14$ MPa

9. Applying a generous SF, try P = 5 MPa, then

$$P = \frac{W}{LD};\quad 5 = \frac{925}{43\,D},\ \text{or, } D = 4.3 \text{ mm.}$$

10 While this might be adequate, a more reasonable value of L/D would be 1.5, giving D = 28.7 mm.

11. Arbitrarily rounding up, choose D = 30 mm.

(b) <u>Other considerations.</u> Other factors which might influence a final choice of bearing geometry would include:

(1) Shaft diameter for strength
(2) Shaft diameter for rigidity
(3) Use of standard shaft size
(4) Standardization of size with other bearings used in the machine.