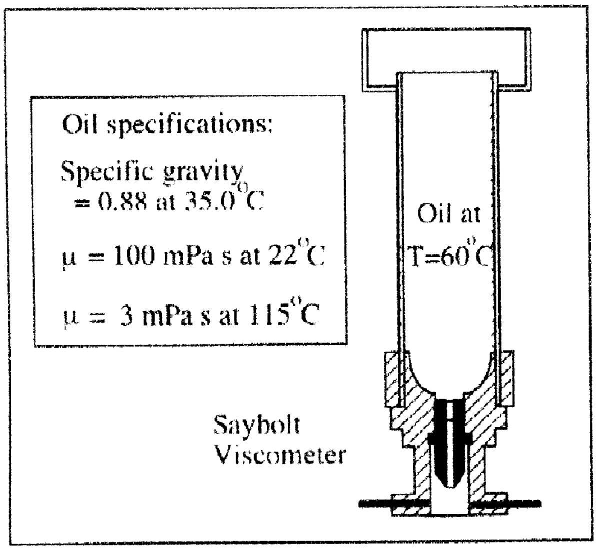
SOLUTION (13.9)

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:

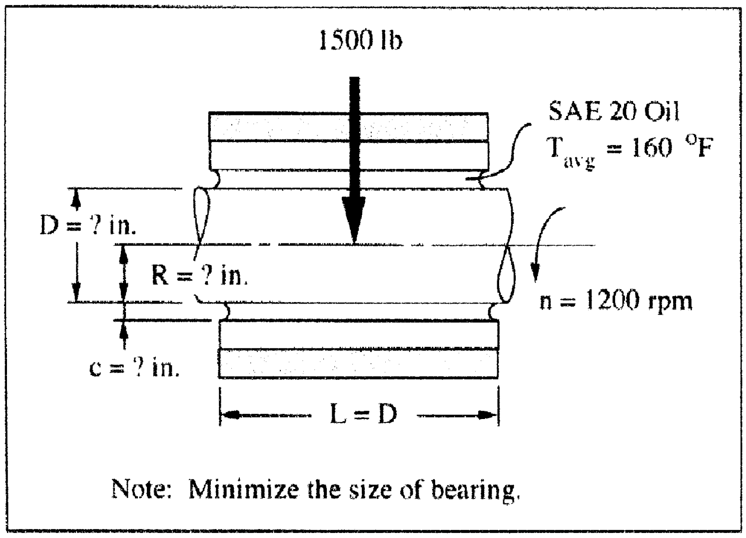
1. Mark the two given data point locations on Fig. 13.6.
2. Connect these two points with a straight line.
3. Use the straight line to determine the value of viscosity for T = 60 °C.
4. The viscosity is 13 mPa·s. ■

SOLUTION (13.28D)

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, μ = 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:

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

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): $h_0 \geq 0.0002 + 0.00004\,(2.5) = 3 \times 10^{-4}$ 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 $h_0/c = 0.19$ or $h_0 = 0.19\,(1.6 \times 10^{-3}\text{ in.}) = 3.04 \times 10^{-4}$ in.
4. Since 3.04×10^{-4} in. > 3.0×10^{-4} in., the criterion is satisfied. ■

Comments:

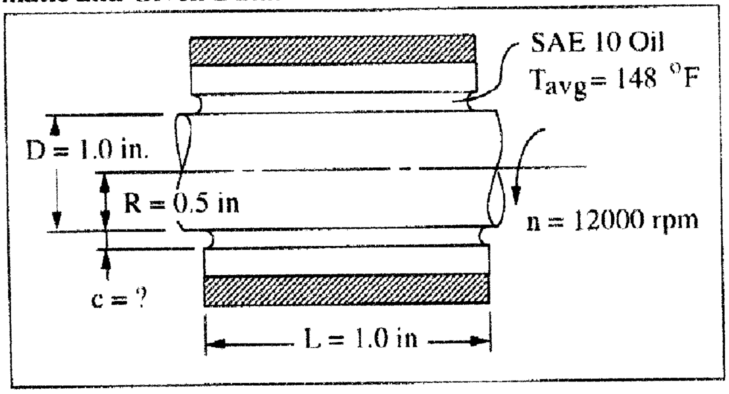
1. With the gravity force of the rotor loading the bearing only at the bottom, oil should be admitted and distributed at the top. Axial oil distribution could be accomplished with a groove, as in Fig. 13.23. Since the entire top of the bearing is never loaded, this groove could be very wide, perhaps 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 (13.25)

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 $h_o/c = 0.535$, then $c = h_o/0.535 = 0.0003/0.535 = 0.000561$, 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

$$0.21 = \left(\frac{0.5}{0.000561}\right)^2\left(\frac{1.8\times10^{-6}\times 200}{P}\right).$$ 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 $f = 4.8\,\frac{c}{R} = 4.8\,\frac{0.000561}{0.5} = 0.0054$

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

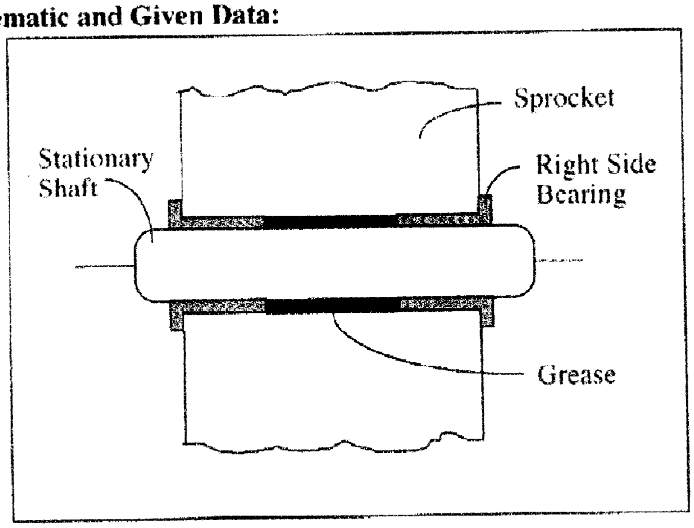
Comment: The foregoing analysis involving the Raimondi and Boyd charts applies only to steady-state operation with a load that is fixed in magnitude and direction.

SOLUTION (13.35D)

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. $$n = \frac{4000 \text{ mm/s}}{\pi (122.3) \text{ mm}} = 10.4 \text{ Hertz}$$

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

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

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) Other considerations. 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.