CHAPTER 2

5.2 Plain bearings

For plain bearings, some materials give much longer life than others. Some of the John Harrison clocks still operate after hundreds of years because of the *lignum vitae* wood employed in their construction, whereas his metal clocks are seldom run due to potential wear.

Flexure bearings

Flexure bearings rely on elastic properties of a material. Flexure bearings bend a piece of material repeatedly. Some materials fail after repeated bending, even at low loads, but careful material selection and bearing design can make flexure bearing life indefinite.

Short-life bearings

Although long bearing life is often desirable, it is sometimes not necessary. Harris 2001 describes a bearing for a rocket motor oxygen pump that gave several hours life, far in excess of the several tens of minutes life needed.[15]

Composite bearings

Depending on the customized specifications (backing material and PTFE compounds), composite bearings can operate up to 30 years without maintenance.

Oscillating bearings

For bearings which are used in oscillating applications, customized approaches to calculate L10 a

External factors[edit]

The service life of the bearing is affected by many parameters that are not controlled by the bearing manufacturers. For example, bearing mounting, temperature, exposure to external environment, lubricant cleanliness and electrical currents through bearings etc. High frequency PWM inverters can induce currents in a bearing, which can be suppressed by the use of ferrite chokes.

The temperature and terrain of the micro-surface will determine the amount of friction by the touching of solid parts.

Certain elements and fields reduce friction while increasing speeds.

Strength and mobility help determine the amount of load the bearing type can carry.

Alignment factors can play a damaging role in wear and tear, yet overcome by computer aid signaling and non-rubbing bearing types, such as magnetic levitation or air field pressure.

LIGHTLY LOADED BEARINGS

551. (a) A 3 x 3 – in. full bearing supports a load of 900 lb., $c_d/D = 0.0015$, $n = 400\,rpm$. The temperature of the SAE 40 oil is maintained at 140 °F. Considering the bearing lightly loaded (Petroff), compute the frictional torque, fhp, and the coefficient of friction. (b) The same as (a) except that the oil is SAE 10W.

Solution.

(a) $T_f = \frac{\mu \pi D L v_{ips}}{(c_d/2)}\left(\frac{D}{2}\right)$

$L = 3\,in$

$D = 3\,in$

$v_{ips} = \frac{\pi D n}{60} = \frac{\pi(3)(400)}{60} = 20\pi\ ips$

$c_d/D = 0.0015$

SAE 40 oil, 140 °F, Figure A16.

$\mu = 7.25\ \mu reyns$

$F = \frac{\mu \pi D L v_{ips}}{(c_d/2)} = \frac{(7.25\times10^{-6})(\pi)(3)(3)(20\pi)}{(0.0015/2)} = 17.173\,lb$

$T_f = F\left(\frac{D}{2}\right) = (17.173)\left(\frac{3}{2}\right) = 25.76\ in-lb$

$fhp = \frac{F v_m}{33{,}000}$

$v_m = \frac{\pi D n}{12} = \frac{\pi(3)(400)}{12} = 314.16\ fpm$

$fhp = \frac{F v_m}{33{,}000} = \frac{(17.173)(314.16)}{33{,}000} = 0.1635\ hp$

$f = \frac{F}{W} = \frac{17.173}{900} = 0.0191$

(b) SAE 10W oil, 140 °F, Figure A16.

$\mu = 2.2\ \mu reyns = 2.2\times10^{-6}\ reyn$

$F = \frac{\mu \pi D L v_{ips}}{(c_d/2)} = \frac{(2.2\times10^{-6})(\pi)(3)(3)(20\pi)}{(0.0015/2)} = 5.211\,lb$

$T_f = F\left(\frac{D}{2}\right) = (5.211)\left(\frac{3}{2}\right) = 7.817\ in-lb$

$fhp = \frac{F v_m}{33{,}000}$

$$v_m = \frac{\pi Dn}{12} = \frac{\pi(3)(400)}{12} = 314.16\ fpm$$

$$fhp = \frac{Fv_m}{33,000} = \frac{(5.211)(314.16)}{33,000} = 0.0496\ hp$$

$$f = \frac{F}{W} = \frac{5.211}{900} = 0.00579$$

553. The average pressure on a 6-in. full bearing is 50 psi, $c_d = 0.003\,in.$, $L/D = 1$. While the average oil temperature is maintained at 160 °F with $n = 300\ rpm$, the frictional force is found to be 13 lb. Compute the coefficient of friction and the average viscosity of the oil. To what grade of oil does this correspond?

Solution:

$$p = \frac{W}{LD}$$

$D = 6\ in.$

$L/D = 1$

$L = 6\ in.$

$W = pLD = (50)(6)(6) = 1800\ lb$

$F = 13\ lb$

Coefficient of Friction

$$f = \frac{F}{W} = \frac{13}{1800} = 0.0072$$

$$F = \frac{\mu\pi DLv_{ips}}{(c_d/2)}$$

$$v_{ips} = \frac{\pi Dn}{60} = \frac{\pi(6)(300)}{60} = 30\pi\ ips$$

$$F = \frac{\mu\pi DLv_{ips}}{(c_d/2)} = \frac{\mu(\pi)(6)(6)(30\pi)}{(0.003/2)} = 13\ lb$$

$\mu = 1.8\times10^{-6}\ reyn = 1.8\ \mu reyns$

Figure AF 16, 160 °F use SAE 10W or SAE 20W

<u>FULL BEARINGS</u>

554. The load on a 4-in. full bearing is 2000 lb., $n = 320\ rpm$; $L/D = 1$; $c_d/D = 0.0011$; operating temperature = 150 °F; $h_o = 0.00088\ in$. (a) Select an oil that will closely accord with the started conditions. For the selected oil determine (b) the frictional loss (ft-lb/min), (c) the hydrodynamic oil flow through the bearing, (d) the amount of end leakage,

(e) the temperature rise as the oil passes through the bearing, (f) the maximum pressure.

Solution:

(a) $D = 4\ in$
$L/D = 1$
$L = 4\ in$
$c_d = 0.0011D = 0.0011(4) = 0.0044\ in$
$h_o = 0.00088\ in$

$$\varepsilon = 1 - \frac{2h_o}{c_d} = 1 - \frac{2(0.00088)}{0.0044} = 0.6$$

Table AT 20
$\varepsilon = 0.6$, $L/D = 1$
Sommerfield Number

$$S = \frac{\mu n_s}{p}\left(\frac{D}{c_d}\right)^2$$

$$n_s = \frac{320}{60} = 5.333\ rps$$

$$p = \frac{W}{LD} = \frac{2000}{(4)(4)} = 125\ psi$$

$c_d/D = 0.0011$

$$0.121 = \frac{\mu(5.333)}{125}\left(\frac{1}{0.0011}\right)^2$$

$\mu = 3.4\times10^{-6}\ reyn = 3.4\ \mu reyns$
Figure AF-16, 150 ^{o}F, use SAE 30 or SAE 20 W
Select SAE 30, the nearest
$\mu = 3.9\times10^{-6}\ reyn$

(b) Table AT 20, $L/D = 1$, $\varepsilon = 0.6$

$$\frac{r}{c_r} f = 3.22$$

$$\frac{r}{c_r} = \frac{D}{c_d} = \frac{1}{0.0011}$$

$$\left(\frac{1}{0.0011}\right) f = 3.22$$

$f = 0.003542$
$F = fW = (0.003542)(2000) = 7.084\ lb$

$$v_m = \frac{\pi Dn}{12} = \frac{\pi(4)(320)}{12} = 335.1\ fpm$$

Frictional loss = $Fv_m = (7.084)(335.1)2374\ ft-lb/min$

(c) Table AT 20, $L/D = 1$, $\varepsilon = 0.6$

$$\frac{q}{rc_r n_s L} = 4.33$$

$$r = \frac{D}{2} = 2.0\ in$$

$$c_r = \frac{c_d}{2} = \frac{0.0044}{2} = 0.0022\ in$$

$$n_s = 5.333\ rps$$

$$L = 4\ in$$

$$q = 4.33 rc_r n_s L = 4.33(2.0)(0.0022)(5.333)(4) = 0.4064\ in^3/sec$$

(d) Table AT 20, $L/D = 1$, $\varepsilon = 0.6$

$$\frac{q_s}{q} = 0.680$$

$$q_s = 0.680q = 0.680(0.4064) = 0.2764\ in^3/sec$$

(e) Table AT 20, $L/D = 1$, $\varepsilon = 0.6$

$$\frac{\rho c \Delta t_o}{p} = 14.2$$

$\rho c = 112$, $p = 125\ psi$

$$\Delta t_o = \frac{14.2p}{\rho c} = \frac{14.2(125)}{112} = 15.85\ ^\circ F$$

(f) Table AT 20, $L/D = 1$, $\varepsilon = 0.6$

$$\frac{p}{p_{\max}} = 0.415$$

$$p_{\max} = \frac{125}{0.415} = 301.2\ psi$$

555. A 4-in., 360° bearing, with $L/D = 1.1$ (use table and chart values for 1), is to support 5 kips with a minimum film thickness 0.0008 in.; $c_d = 0.004\ in.$, $n = 600\ rpm$. Determine (a) the needed absolute viscosity of the oil .(b) Suitable oil if the average film temperature is 160 F, (c) the frictional loss in hp. (d) Adjusting only h_o to the optimum value for minimum friction, determine the fhp and compare. (e) This load varies. What could be the magnitude of the maximum impulsive load if the eccentricity ration ε becomes 0.8? Ignore "squeeze" effect.

Solution:

$D = 4\ in$

$L = 1.1D = 1.1(4) = 4.4\ in$

$p = \frac{W}{LD} = \frac{5000}{(4.4)(4)} = 284\ psi$

$h_o = 0.0008\ in$

$c_d = 0.004\ in.$

$\varepsilon = 1 - \frac{2h_o}{c_d} = 1 - \frac{2(0.0008)}{0.004} = 0.6$

$\eta = \frac{600}{60} = 10\ rps$

(a) Table AT20, $L/D = 1$, $\varepsilon = 0.6$

$S = 0.121$

$S = \left(\frac{r}{c_r}\right)^2 \frac{\mu n_s}{p} = \frac{\mu n_s}{p}\left(\frac{D}{c_d}\right)^2$

$0.121 = \frac{\mu(10)}{284}\left(\frac{4}{0.004}\right)^2$

$\mu = 3.4\times10^{-6}\ reyn$

(b) Figure AF16, 160 F

Use SAE 30, $\mu = 3.2\times10^{-6}\ reyn$

(c) Table AT 20, $L/D = 1$, $\varepsilon = 0.6$

$\frac{r}{c_r} f = 3.22$

$\frac{D}{c_d} f = 3.22$

$\left(\frac{4}{0.004}\right) f = 3.22$

$f = 0.00322$

$F = fW = (0.00322)(5000\ lb) = 16.1\ lb$

$v_m = \frac{\pi Dn}{12} = \frac{\pi(4)(600)}{12} = 628.3\ fpm$

$$fhp = \frac{Fv_m}{33{,}000} = \frac{(16.1)(628.3)}{33{,}000} = 0.3065\ hp$$

(d) adjusting h_o, $c_d = 0.004\ in.$

Table AT 20, $L/D = 1$

$h_o/c_r = 0.30$ optimum value for minimum friction

$$\frac{r}{c_r} f = 2.46$$

$$\frac{D}{c_d} f = 2.46$$

$$\left(\frac{4}{0.004}\right) f = 2.46$$

$$f = 0.00246$$

$$F = fW = (0.00246)(5000\ lb) = 12.3\ lb$$

$$v_m = \frac{\pi D n}{12} = \frac{\pi(4)(600)}{12} = 628.3\ fpm$$

$$fhp = \frac{Fv_m}{33{,}000} = \frac{(12.3)(628.3)}{33{,}000} = 0.234\ hp < fhp\ (c)$$

(e) $\varepsilon = 0.8$, Table AT 20, $L/D = 1$

$$S = 0.0446$$

$$S = \left(\frac{r}{c_r}\right)^2 \frac{\mu n_s}{p} = \frac{\mu n_s}{p}\left(\frac{D}{c_d}\right)^2$$

$$0.0446 = \frac{(3.2\times10^{-6})(10)}{p}\left(\frac{4}{0.004}\right)^2$$

$$p = 717.5\ psi$$

$$W = pDL = (717.5)(4)(4.4) = 12{,}628\ lb$$

556. For an 8 x 4 – in. full bearing, $c_r = 0.0075\ in.$, $n = 2700\ rpm$, average $\mu = 4\times10^{-6}\ reyn$. (a) What load may this bearing safely carry if the minimum film thickness is not to be less than that given by Norton, §11.14, *Text*? (b) Compute the corresponding frictional loss (fhp). (c) Complete calculations for the other quantities in Table AT 20, ϕ, q, q_s, Δt_o, p_{max}. Compute the maximum load for an optimum (load) bearing (d) if c_r remains the same, (e) if h_o remains the same.

Solution:

$D \times L = 8 \times 4$

$L/D = 1/2$

$c_r = 0.0075\ in$

$r = D/2 = 4\ in$

$\mu = 4 \times 10^{-6}\ reyn$

(a) by Norton, $h_o = 0.00025D = 0.00025(8) = 0.002\ in$

$$\frac{h_o}{c_r} = \frac{0.002}{0.0075} = 0.27$$

Table AT 20, $L/D = 1/2$, $\frac{h_o}{c_r} = 0.27$

$S = 0.172$

$$S = \left(\frac{r}{c_r}\right)^2 \frac{\mu n_s}{p}$$

$$n_s = \frac{2700}{60} = 45\ rps$$

$$S = 0.172 = \left(\frac{4}{0.0075}\right)^2 \frac{(4 \times 10^{-6})(45)}{p}$$

$p = 298\ psi$

$W = pDL = (298)(8)(4) = 9536\ lb$

(b) Table AT 20, $L/D = 1/2$, $\frac{h_o}{c_r} = 0.27$

$\phi = 38.5^\circ$

$$\frac{r}{c_r} f = 4.954$$

$$\frac{D}{c_d} f = 4.954$$

$$\left(\frac{4}{0.004}\right) f = 4.954$$

$f = 0.0093$

$F = fW = (0.0093)(9536\ lb) = 88.7\ lb$

$$v_m = \frac{\pi Dn}{12} = \frac{\pi(8)(2700)}{12} = 5655\ fpm$$

$$fhp = \frac{Fv_m}{33{,}000} = \frac{(88.7)(5655)}{33{,}000} = 15.2\ hp$$

(c) Table AT 20, $L/D = 1/2$, $\frac{h_o}{c_r} = 0.27$

$\phi = 38.5^\circ$

$\frac{q}{rc_r n_s L} = 5.214$

$q = 5.214 rc_r n_s L = 5.214(4)(0.0075)(45)(4) = 28.2\ in^3/\sec$

$\frac{q_s}{q} = 0.824$

$q_s = 0.824(28.2) = 23.2\ in^3/\sec$

$\frac{\rho c \Delta t}{p} = 20.26$

$\Delta t = \frac{20.26(298)}{112} = 54\ ^\circ F$

$\frac{p}{p_{\max}} = 0.3013$

$p_{\max} = \frac{298}{0.3013} = 989\ psi$

To solve for maximum load, Table AT 20, $L/D = 1/2$, $\frac{h_o}{c_r} = 0.43$

$S = \left(\frac{r}{c_r}\right)^2 \frac{\mu n_s}{p} = 0.388$

(d) $c_r = 0.0075\ in$

$S = 0.388 = \left(\frac{4}{0.0075}\right)^2 \frac{(4\times10^{-6})(45)}{p}$

$p = 132\ psi$

$W = pDL = (132)(8)(4) = 4224\ lb$

(e) $h_o = 0.002\ in$

$\frac{h_o}{c_r} = 0.43$

$c_r = \frac{0.002}{0.43} = 0.00465\ in$

$S = 0.388 = \left(\frac{4}{0.00465}\right)^2 \frac{(4\times10^{-6})(45)}{p}$

$p = 343.3\ psi$

$W = pDL = (343.3)(8)(4) = 10{,}986\ lb$

557. A 6 x 6 – in full bearing has a frictional loss of $fhp = 11$ when the load is 68,500 lb. and $n = 1600\ rpm$; $c_r/r = 0.001$. (a) Compute the minimum film thickness. Is this in the vicinity of that for an optimum bearing? (b) What is the viscosity of the oil and a proper grade for an operating temperature of 160 F? (c) For the same h_o, but for the maximum-load optimum, determine the permissible load and the fhp.

Solution:

$L = 6\ in$

$D = 6\ in$

$L/D = 1$

$r = D/2 = 3\ in$

$c_r/r = 0.001$

$n = 1600\ rpm$

$$v_m = \frac{\pi D n}{12} = \frac{\pi(3)(1600)}{12} = 2513\ fpm$$

$$fhp = \frac{F v_m}{33{,}000}$$

$$F = \frac{33{,}000(11)}{2513} = 144.45\ lb$$

$$f = \frac{F}{W} = \frac{144.45}{68{,}500} = 0.00211$$

(a) $\frac{r}{c_r} f = \left(\frac{1}{0.001}\right)(0.00211) = 2.11$

Table AT 20, $L/D = 1$, $\frac{r}{c_r} f = 2.11$

Near the vicinity of optimum bearing

$c_r = 0.001 r = 0.001(3) = 0.003\ in$

$h_o = 0.254 c_r = 0.254(0.003) = 0.0008\ in$

(b) Table AT 20, $L/D = 1$, $\frac{r}{c_r} f = 2.11$

$S = 0.0652$

$$S = \left(\frac{r}{c_r}\right)^2 \frac{\mu n_s}{p} = 0.388$$

$$n_s = \frac{1600}{60} = 26.67\ rps$$

$$p = \frac{W}{LD} = \frac{68{,}500}{(6)(6)} = 1902.8\ psi$$

$$S = 0.0652 = \left(\frac{1}{0.001}\right)^2 \frac{(\mu)(26.67)}{1902.8}$$

$\mu = 4.7 \times 10^{-6}\ reyn$

Figure AF 16, 160 F, use SAE 40.

(c) Table AT 20, $L/D = 1$

optimum bearing, maximum load, $\frac{h_o}{c_r} = 0.53$

h_o the same, $c_r = \frac{h_o}{0.53} = \frac{0.0008}{0.53} = 0.0015\ in$

$\frac{h_o}{c_r} = 0.53$, $S = 0.214$, $\frac{r}{c_r} f = 4.89$

$$S = \left(\frac{r}{c_r}\right)^2 \frac{\mu n_s}{p}$$

$$S = 0.214 = \left(\frac{3}{0.0015}\right)^2 \frac{(4.7 \times 10^{-6})(26.67)}{p}$$

$p = 2343\ psi$

$W = pDL = (2343)(6)(6) = 84{,}348\ lb$

$$\frac{r}{c_r} f = 4.89$$

$$\left(\frac{3}{0.0015}\right) f = 4.89$$

$f = 0.00245$

$F = fW = 0.00245(84{,}348) = 206.65\ lb$

$v_m = 2513\ fpm$

$$fhp = \frac{Fv_m}{33{,}000} = \frac{(206.65)(2513)}{33{,}000} = 15.74\ hp$$

558. The maximum load on a 2.25 x 1.6875 in. main bearing of an automobile is 3140 lb. with wide-open throttle at 1000 rpm. If the oil is SAE 20W at 210 F, compute the minimum film thickness for a bearing clearance of (a) 0.0008 in. and (b) 0.0005 in. Which clearance results in the safer operating conditions? Note: Since a load of this order exists for only 20-25^{o} of rotation, the actual h_o does not reach this computed minimum (squeeze effect).

Solution:

$D \times L = 2.25 \times 1.6875\ in$

$$\frac{L}{D} = \frac{1.6875}{2.25} = 0.75$$

SAE 20 W at 210 °F

$\mu = 0.96 \times 10^{-6}\ reyn$

$W = 3140\ lb$

$n = 1000\ rpm$

$$p = \frac{W}{DL} = \frac{3140}{(2.25)(1.6875)} = 827\ psi$$

$$n_s = \frac{1000}{60} = 16.67\ rps$$

$$r = \frac{D}{2} = 1.125\ in$$

$$S = \frac{\mu n_s}{p}\left(\frac{r}{c_r}\right)^2$$

(a) $c_r = 0.0008\ in$

$$S = \frac{(0.96 \times 10^{-6})(16.67)}{827}\left(\frac{1.125}{0.0008}\right)^2 = 0.038$$

Table AT 20, $L/D = 3/4$, $S = 0.038$

L/D	h_o/c_r	S
1	0.2	0.0446
½	0.2	0.0923
¾	0.2	0.0685

L/D	h_o/c_r	S
1	0.1	0.0188
½	0.1	0.0313
¾	0.1	0.0251

At $L/D = 3/4$

$$\frac{h_o}{c_r} = \left(\frac{0.038 - 0.0251}{0.0685 - 0.0251}\right)(0.2 - 0.1) + 0.1 = 0.13$$

$$h_o = 0.13 c_r = 0.13(0.0008) = 0.0001\ in$$

(b) $c_r = 0.0005\ in$

$$S = \frac{(0.96 \times 10^{-6})(16.67)}{827}\left(\frac{1.125}{0.0005}\right)^2 = 0.098$$

Table AT 20, $L/D = 3/4$, $S = 0.098$

L/D	h_o/c_r	S
1	0.2	0.0446
½	0.2	0.0923
¾	0.2	0.0685

L/D	h_o/c_r	S
1	0.4	0.121
½	0.4	0.319
¾	0.4	0.220

At $L/D = 3/4$

$$\frac{h_o}{c_r} = \left(\frac{0.098 - 0.0685}{0.220 - 0.0685}\right)(0.4 - 0.2) + 0.2 = 0.239$$

$$h_o = 0.239c_r = 0.239(0.0005) = 0.00012\ in$$

use $c_r = 0.0005\ in$, $h_o = 0.00012\ in$

561. A 360° bearing supports a load of 2500 lb.; $D = 5\ in.$, $L = 2.5\ in.$, $c_r = 0.003\ in.$, $n = 1800\ rpm$; SAE 20 W oil entering at 100 F. (a) Compute the average temperature t_{av} of the oil through the bearing. (An iteration procedure. Assume μ; compute S and the corresponding Δt_o; then the average oil temperature $t_{av} = t_i + \Delta t_o/2$. If this t_{av} and the assumed μ do not locate a point in Fig. AF 16 on the line for SAE 20 W oil, try again.) Calculate (b) the minimum film thickness, (c) the fhp, (d) the amount of oil to be supplied and the end leakage.

Solution:

$D = 5\ in$

$L = 2.5\ in$

$$\frac{L}{D} = \frac{2.5}{5} = 0.5$$

$c_r = 0.003\ in$

(a) Table AT 20

Parameter, $\dfrac{\rho c \Delta t_o}{p}$, $\rho c = 112$

$$S = \left(\frac{r}{c_r}\right)^2 \frac{\mu n_s}{p}$$

$p = \frac{W}{DL} = \frac{2500}{(5)(2.5)} = 200\ psi$

$n_s = \frac{1800}{60} = 30\ rps$

$r = \frac{D}{2} = 2.5\ in$

$c_r = 0.003\ in$

Fig. AF 16, SAE 20 W, Table AT 20, $L/D = 0.5$, $t_i = 100\ ^\circ F$

Trial μ ($t\ ^\circ F$), reyns	S	$\frac{\rho c \Delta t_o}{p}$	$\Delta t_o\ ^\circ F$	$t_{av} = t_i + \Delta t_o/2\ ^\circ F$
3.5×10^{-6} (130 F)	0.365	36.56	65	132.5
3.2×10^{-6} (134 F)	0.333	34.08	61	130.5
3.4×10^{-6} (132 F)	0.354	35.71	64	132.0

Therefore, use $t_{av} = 132\ ^\circ F$, $S = 0.354$

(b) Table AT 20, $L/D = 0.5$, $S = 0.354$

$\frac{h_o}{c_r} = 0.415$

$h_o = 0.415(0.003) = 0.00125\ in$

(c) Table AT 20, $L/D = 0.5$, $S = 0.354$

$\frac{r}{c_r} f = 8.777$

$\left(\frac{2.5}{0.003}\right) f = 8.777$

$f = 0.0105$

$F = fW = 0.0105(2500) = 26.25\ lb$

$v_m = \frac{\pi D n}{12} = \frac{\pi(5)(1800)}{12} = 2356\ fpm$

$fhp = \frac{F v_m}{33{,}000} = \frac{(26.25)(2356)}{33{,}000} = 1.874\ hp$

(d) Table AT 20, $L/D = 0.5$, $S = 0.354$

$\frac{q}{r c_r n_s L} = 4.807$

$q = 4.807 r c_r n_s L = 4.807(2.5)(0.003)(30)(2.5) = 2.704\ in^3/\text{sec}$

$\frac{q_s}{q} = 0.7165$

$q_s = 0.7165(2.704) = 1.937\ in^3/\sec$

PARTIAL BEARINGS

562. A 2 x 2-in. bearing has a clearance $c_r = 0.001\,in$, and $h_o = 0.0004\,in.$, $n = 2400\ rpm$, and for the oil, $\mu = 3\times10^{-6}\ reyn$. Determine the load, frictional horsepower, the amount of oil to enter, the end leakage of oil, and the temperature rise of the oil as it passes through for : (a) a full bearing, partial bearings of (b) 180°, (c) 120°, (d) 90°, (e) 60°.

Solution:

$D = L = 2\ in$

$L/D = 1$

$c_r = 0.001\,in$

$r = D/2 = 1\ in$

$n = 2400\ rpm$

$n_s = 40\ rps$

$\mu = 3\times10^{-6}\ reyn$

$h_o = 0.004\ in.$

$$\frac{h_o}{c_r} = \frac{0.0004}{0.001} = 0.4$$

$$v_m = \frac{\pi D n}{12} = \frac{\pi(2)(2400)}{12} = 1257\ fpm$$

(a) Full bearing

Table AT 20, $L/D = 1$, $h_o/c_r = 0.4$

$S = 0.121$

$$\frac{rf}{c_r} = 3.22$$

$$\frac{q}{rc_r n_s L} = 4.33$$

$$\frac{q_s}{q} = 0.680$$

$$\frac{\rho c \Delta t_o}{p} = 14.2$$

$$\frac{p}{p_{\max}} = 0.415$$

Load W

$$S = \left(\frac{r}{c_r}\right)^2 \frac{\mu n_s}{p}$$

$$0.121 = \left(\frac{1}{0.001}\right)^2 \frac{(3\times 10^{-6})(40)}{p}$$

$p = 992\ psi$

$W = pDL = (992)(2)(2) = 3968\ lb$

fhp:

$F = fW$

$$\frac{rf}{c_r} = 3.22$$

$$\left(\frac{1}{0.001}\right)f = 3.22$$

$f = 0.00322$

$F = fW = (0.00322)(3968) = 12.78\ lb$

$$fhp = \frac{Fv_m}{33{,}000} = \frac{(12.78)(1257)}{33{,}000} = 0.4868\ hp$$

Oil flow, q

$$\frac{q}{rc_r n_s L} = 4.33$$

$$\frac{q}{(0.1)(0.001)(40)(2)} = 4.33$$

$q = 0.3464\ in^3/\sec$

End leakage

$$\frac{q_s}{q} = 0.680$$

$q_s = 0.68(0.3464) = 0.2356\ in^3/\sec$

Temperature rise, Δt_o

$$\frac{\rho c \Delta t_o}{p} = 14.2$$

$$\frac{(112)\Delta t_o}{992} = 14.2$$

$\Delta t_o = 126\ ^\circ F$

(b) 180° Bearing

Table AT 21, $L/D = 1$, $h_o/c_r = 0.4$

$S = 0.128$

$$\frac{rf}{c_r} = 2.28$$

$$\frac{q}{rc_r n_s L} = 3.25$$

$$\frac{q_s}{q} = 0.572$$

$$\frac{\rho c \Delta t_o}{p} = 12.4$$

Load W

$$S = \left(\frac{r}{c_r}\right)^2 \frac{\mu n_s}{p}$$

$$0.128 = \left(\frac{1}{0.001}\right)^2 \frac{(3\times10^{-6})(40)}{p}$$

$$p = 937.5\ psi$$

$$W = pDL = (937.5)(2)(2) = 3750\ lb$$

fhp:

$$F = fW$$

$$\frac{rf}{c_r} = 2.28$$

$$\left(\frac{1}{0.001}\right)f = 2.28$$

$$f = 0.00228$$

$$F = fW = (0.00228)(3750) = 8.55\ lb$$

$$fhp = \frac{Fv_m}{33{,}000} = \frac{(8.55)(1257)}{33{,}000} = 0.3257\ hp$$

Oil flow, q

$$\frac{q}{rc_r n_s L} = 3.25$$

$$\frac{q}{(0.1)(0.001)(40)(2)} = 3.25$$

$$q = 0.26\ in^3/\sec$$

End leakage

$$\frac{q_s}{q} = 0.572$$

$$q_s = 0.572(0.26) = 0.1487\ in^3/\sec$$

Temperature rise, Δt_o

$$\frac{\rho c \Delta t_o}{p} = 12.4$$
$$\frac{(112)\Delta t_o}{937.5} = 12.4$$
$$\Delta t_o = 104\ ^\circ F$$

(c) 12° Bearing

Table AT 22, $L/D = 1$, $h_o/c_r = 0.4$
$$S = 0.162$$
$$\frac{rf}{c_r} = 2.16$$
$$\frac{q}{rc_r n_s L} = 2.24$$
$$\frac{q_s}{q} = 0.384$$
$$\frac{\rho c \Delta t_o}{p} = 15$$

Load W
$$S = \left(\frac{r}{c_r}\right)^2 \frac{\mu n_s}{p}$$
$$0.162 = \left(\frac{1}{0.001}\right)^2 \frac{(3\times10^{-6})(40)}{p}$$
$$p = 741\ psi$$
$$W = pDL = (741)(2)(2) = 2964\ lb$$

fhp:
$$F = fW$$
$$\frac{rf}{c_r} = 2.16$$
$$\left(\frac{1}{0.001}\right) f = 2.16$$
$$f = 0.00216$$
$$F = fW = (0.00216)(2964) = 6.4\ lb$$
$$fhp = \frac{Fv_m}{33{,}000} = \frac{(6.4)(1257)}{33{,}000} = 0.2438\ hp$$

Oil flow, q

$$\frac{q}{rc_r n_s L} = 2.24$$

$$\frac{q}{(0.1)(0.001)(40)(2)} = 2.24$$

$$q = 0.1792\ in^3/\sec$$

End leakage

$$\frac{q_s}{q} = 0.384$$

$$q_s = 0.384(0.1792) = 0.0688\ in^3/\sec$$

Temperature rise, Δt_o

$$\frac{\rho c \Delta t_o}{p} = 15$$

$$\frac{(112)\Delta t_o}{741} = 15$$

$$\Delta t_o = 99\ ^\circ F$$

(d) 60^o Bearing

$L/D = 1$, $h_o/c_r = 0.4$

$$S = 0.450$$

$$\frac{rf}{c_r} = 3.29$$

$$\frac{q}{rc_r n_s L} = 1.56$$

$$\frac{q_s}{q} = 0.127$$

$$\frac{\rho c \Delta t_o}{p} = 28.2$$

Load W

$$S = \left(\frac{r}{c_r}\right)^2 \frac{\mu n_s}{p}$$

$$0.450 = \left(\frac{1}{0.001}\right)^2 \frac{(3\times 10^{-6})(40)}{p}$$

$$p = 267\ psi$$

$$W = pDL = (267)(2)(2) = 1068\ lb$$

fhp:

$$F = fW$$

$\frac{rf}{c_r} = 3.29$

$\left(\frac{1}{0.001}\right) f = 3.29$

$f = 0.00329$

$F = fW = (0.00329)(1068) = 3.514\ lb$

$fhp = \frac{Fv_m}{33{,}000} = \frac{(3.514)(1257)}{33{,}000} = 0.1339\ hp$

Oil flow, q

$\frac{q}{rc_r n_s L} = 1.56$

$\frac{q}{(0.1)(0.001)(40)(2)} = 1.56$

$q = 0.1248\ in^3/\sec$

End leakage

$\frac{q_s}{q} = 0.127$

$q_s = 0.127(0.1248) = 0.0158\ in^3/\sec$

Temperature rise, Δt_o

$\frac{\rho c \Delta t_o}{p} = 28.2$

$\frac{(112)\Delta t_o}{267} = 28.2$

$\Delta t_o = 67\ ^\circ F$

563. A 2 x 2 in. bearing sustains a load of $W = 5000\ lb.$; $c_r = 0.001\ in.$; $n = 2400\ rpm$; $\mu = 3\times10^{-6}\ reyn$. Using Figs. AF 17 and AF 18, determine the minimum film thickness and the frictional loss (ft-lb/min.) for (a) a full bearing, and for partial bearings of (b) 180^o, (c) 120^o, (d) 90^o, (e) 60^o.

Solution:

$L = 2\ in$

$D = 2\ in$

$W = 5000\ lb$

$c_r = 0.001\ in.$

$n = 2400\ rpm$

$n_s = 40\ rps$

$\mu = 3\times10^{-6}\ reyn$

$r = D/2 = 1\ in$

$$p = \frac{W}{LD} = \frac{5000}{(2)(2)} = 1250 \ psi$$

$$S = \left(\frac{r}{c_r}\right)^2 \frac{\mu n_s}{p} = \left(\frac{1}{0.001}\right)^2 \frac{(3\times10^{-6})(40)}{1250} = 0.10$$

$$v_m = \frac{\pi D n}{12} = \frac{\pi(2)(2400)}{12} = 1257 \ fpm$$

Using Fig. AF 17 and AF 18

(a) Full Bearing

$$\frac{h_o}{c_r} = 0.346$$

$$\frac{r}{c_r} f = 2.8$$

$$h_o = 0.346(0.001) = 0.000346 \ in$$

$$\left(\frac{1}{0.001}\right) f = 2.8$$

$$f = 0.0028$$

$$F = fW = (0.0028)(5000) = 14 \ lb$$

$$Fv_m = (14)(1257) = 17{,}600 \ ft-lb/\min$$

(b) 180° Bearing

$$\frac{h_o}{c_r} = 0.344$$

$$\frac{r}{c_r} f = 2.0$$

$$h_o = 0.344(0.001) = 0.000344 \ in$$

$$\left(\frac{1}{0.001}\right) f = 2.0$$

$$f = 0.0020$$

$$F = fW = (0.0020)(5000) = 10 \ lb$$

$$Fv_m = (10)(1257) = 12{,}570 \ ft-lb/\min$$

(c) 120° Bearing

$$\frac{h_o}{c_r} = 0.302$$

$\frac{r}{c_r} f = 1.7$

$h_o = 0.302(0.001) = 0.000302\ in$

$\left(\frac{1}{0.001}\right) f = 1.7$

$f = 0.0017$

$F = fW = (0.0017)(5000) = 8.5\ lb$

$Fv_m = (8.5)(1257) = 10{,}685\ ft - lb/\min$

(d) 60^o Bearing

$\frac{h_o}{c_r} = 0.20$

$\frac{r}{c_r} f = 1.4$

$h_o = 0.20(0.001) = 0.0002\ in$

$\left(\frac{1}{0.001}\right) f = 1.4$

$f = 0.0014$

$F = fW = (0.0014)(5000) = 7\ lb$

$Fv_m = (7)(1257) = 8{,}800\ ft - lb/\min$

564. A 120^o partial bearing is to support 4500 lb. with $h_o = 0.002\ in.$; $L/D = 1$; $D = 4\ in.$; $c_d = 0.010\ in.$; $n = 3600\ rpm$. Determine (a) the oil's viscosity,(b) the frictional loss (ft-lb/min), (c) the eccentricity angle, (d) the needed oil flow, (e) the end leakage, (f) the temperature rise of the oil as it passes through, (g) the maximum pressure. (h) If the clearance given is the average, what approximate class of fit (Table 3.1) is it? (i) What maximum impulsive load would be on the bearing if the eccentricity ratio suddenly went to 0.8? Ignore "squeeze" effect.

Solution:

$W = 4500\ lb$

$h_o = 0.002\ in$

$L/D = 1$

$D = 4\ in$

$L = 4\ in$

$r = D/2 = 2\ in$

$c_d = 0.010\ in.$

$n = 3600\ rpm$

$$n_s = \frac{3600}{60} = 60\ rps$$

$$v_m = \frac{\pi D n}{12} = \frac{\pi(2)(3600)}{12} = 3770\ fpm$$

$$p = \frac{W}{LD} = \frac{4500}{(4)(4)} = 281.25\ psi$$

$$\frac{h_o}{c_r} = \frac{2h_o}{c_r} = \frac{2(0.002)}{0.010} = 0.4$$

Table AT 22, $L/D = 1$, $h_o/c_r = 0.4$

$$S = 0.162$$

$$\phi = 35.65^\circ$$

$$\frac{r}{c_r} f = 2.16$$

$$\frac{q}{r c_r n_s L} = 2.24$$

$$\frac{q_s}{q} = 0.384$$

$$\frac{\rho c \Delta t_o}{p} = 15.0$$

$$\frac{p}{p_{\max}} = 0.356$$

(a) $S = \left(\frac{r}{c_r}\right)^2 \frac{\mu n_s}{p}$

$$S = \left(\frac{D}{c_d}\right)^2 \frac{\mu n_s}{p}$$

$$0.162 = \left(\frac{4}{0.010}\right)^2 \frac{\mu(60)}{281.25}$$

$$\mu = 4.75 \times 10^{-6}\ reyn$$

(b) $\frac{r}{c_r} f = 2.16$

$$\frac{D}{c_d} f = 2.16$$

$$\left(\frac{4}{0.010}\right) f = 2.16$$

$$f = 0.0054$$

$$F = fW = 0.0054(4500) = 24.30\ lb$$

$Fv_m = (24.30)(3770) = 91{,}611\ ft - lb/\min$

(c) $\phi = 35.65^\circ$

(d) $\dfrac{q}{rc_r n_s L} = \dfrac{4q}{Dc_d n_s L} = 2.24$

$\dfrac{4q}{(4)(0.010)(60)(4)} = 2.24$

$q = 5.4\ in^3/\sec$

(e) $\dfrac{q_s}{q} = 0.384$

$q_s = 0.384(5.4) = 2.07\ in^3/\sec$

(f) $\dfrac{\rho c \Delta t_o}{p} = 15.0$

$\dfrac{(112)\Delta t_o}{281.25} = 15.0$

$\Delta t_o = 38\ ^\circ F$

(g) $\dfrac{p}{p_{\max}} = 0.356$

$p_{\max} = \dfrac{281.25}{0.356} = 790\ psi$

(h) $c_d = 0.010\ in$, $D = 4\ in$

Table 3.1

RC 8, Hole, average = + 0.0025

Shaft, average = - 0.00875

$c_d = 0.0025 + 0.00875 = 0.01125 \approx 0.010\ in$

Class of fit = RC 9

(i) $\varepsilon = 0.80$

Table AT 22, , $L/D = 1$

$S = 0.162$

$S = \left(\dfrac{D}{c_d}\right)^2 \dfrac{\mu n_s}{p}$

$0.0531 = \left(\dfrac{4}{0.010}\right)^2 \dfrac{(3\times 10^{-6})(60)}{p}$

$p = 542\ psi$

$W = pDL = (542)(4)(4) = 8672\ lb$

565. A 120o partial bearing is to support 4500 lb., $D = 3\ in.$, $c_d = 0.003\ in.$; $n = 3600\ rpm$; SAE 20W entering at 110 F. Calculate (a) the average temperature of the oil as it passes through,(b) the minimum film thickness, (c) the fhp, (d) the quantity of oil to be supplied. HINT: In (a) assume μ and determine the corresponding values of S and Δt_o; then $t_{av} = t_i + \Delta t_o/2$. If assumed μ and t_{av} do not locate a point in Fig. AF 16 that falls on line for SAE 20W, iterate.

Solution:
$W = 4500\ lb$
$D = 3\ in$
$L = 3\ in$
$L/D = 1$
$c_d = 0.003\ in.$

$$S = \left(\frac{D}{c_d}\right)^2 \frac{\mu n_s}{p}$$

$$n_s = \frac{3600}{60} = 60\ rps$$

$$p = \frac{W}{DL} = \frac{4500}{(3)(3)} = 500\ psi$$

$\frac{\rho c \Delta t_o}{p}$, (SAE 20W)

(a) Using Table AT22, $L/D = 1$, $\rho c = 112$, $t_i = 110^\circ F$

Trial μ	t, °F	S	$\frac{\rho c \Delta t_o}{p}$	Δt_o	$t_{av} = t_i + \Delta t_o/2$
3.5 x 10^{-6}	130	0.42	19.8	88	154
2.0 x 10^{-6}	160	0.24	15.4	68	144
2.6 x 10^{-6}	145	0.312	17.7	79	149.5
2.35 x 10^{-6}	150	0.282	17.2	76	148
2.4 x 10^{-6}	149	0.288	17.3	78	149

$\therefore$ Use $t_{av} = 149\ ^\circ F$

(b) Table AT 22, $L/D = 1$, $S = 0.288$

$$\frac{h_o}{c_r} = 0.513$$

$$\frac{2h_o}{c_d} = 0.513$$

$$2h_o = 0.513(0.003)$$

$$h_o = 0.00077\ in$$

(c) Table At 22, $L/D = 1$, $S = 0.288$

$$\frac{r}{c_r} f = 2.974$$

$$\frac{D}{c_r} f = \frac{3}{0.003} f = 2.974$$

$$f = 0.002974$$

$$F = fW = (0.002974)(4500) = 13.383\ lb$$

$$fhp = \frac{Fv_m}{33{,}000}$$

$$v_m = \frac{\pi Dn}{12} = \frac{\pi(3)(3600)}{12} = 2827\ fpm$$

$$fhp = \frac{Fv_m}{33{,}000} = \frac{(13.383)(2827)}{33{,}000} = 1.15\ hp$$

(d) Table At 22, $L/D = 1$, $S = 0.288$

$$\frac{q}{rc_r n_s L} = 2.528$$

$$\frac{4q}{Dc_d n_s L} = 2.528$$

$$\frac{4q}{(3)(0.003)(60)(3)} = 2.528$$

$$q = 1.024\ in^3/\text{sec}$$

566. The 6000-lb. reaction on an 8 x 4 –in., 180° partial bearing is centrally applied; $n = 1000\ rpm$; $h_o = 0.002\ in$. For an optimum bearing with minimum friction determine (a) the clearance, (b) the oil's viscosity, (c) the frictional horsepower. (d) Choose a c_d/D ratio either smaller or larger than that obtained in (a) and show that the friction loss is greater than that in the optimum bearing. Other data remain the same.

Solution:

$W = 6000\ lb$

$D = 8\ in$
$L = 4\ in$
$n = 1000\ rpm$
$n_s = \frac{1000}{60} = 16.67\ rps$
$L/D = 1/2$
$h_o = 0.002\ in$

(a) Table AT 21, $L/D = 1/2$

Optimum value (minimum friction)
$h_o/c_r = 0.23$
$c_r = \frac{0.002}{0.23} = 0.0087\ in$

(b) Table AT 21, $L/D = 1/2$, $h_o/c_r = 0.23$

$S = 0.126$
$S = \left(\frac{r}{c_r}\right)^2 \frac{\mu n_s}{p}$
$p = \frac{W}{DL} = \frac{6000}{(4)(8)} = 187.5\ psi$
$r = \frac{D}{2} = 4\ in$
$S = 0.126 = \left(\frac{4}{0.0087}\right)^2 \frac{\mu(16.67)}{187.5}$
$\mu = 6.70\times10^{-6}\ reyn$

(c) Table AT 21, $L/D = 1/2$, $h_o/c_r = 0.23$

$\frac{r}{c_r} f = 2.97$
$\left(\frac{4}{0.0087}\right) f = 2.97$
$f = 0.00646$
$F = fW = (0.00646)(6000) = 38.76\ lb$
$v_m = \frac{\pi D n}{12} = \frac{\pi(8)(1000)}{12} = 2094\ fpm$

$$fhp = \frac{Fv_m}{33{,}000} = \frac{(38.76)(2094)}{33{,}000} = 2.46\ hp$$

For (a) $\frac{c_d}{D} = \frac{2c_r}{D} = \frac{2(0.0087)}{8} = 0.0022$

$\frac{c_d}{D} > 0.0022$

$\frac{c_d}{D} = 0.0030$

$c_d = 0.0030(8) = 0.0240\ in$

$c_r = 0.0120\ in$

$\frac{h_o}{c_r} = \frac{0.002}{0.012} = 0.1667$

Table AT 21, $L/D = 1/2$

$\frac{r}{c_r} f = 1.67$

$\left(\frac{4}{0.0016}\right) f = 1.67$

$f = 0.00668$

$F = fW = (0.00668)(6000) = 40.08\ lb$

$v_m = \frac{\pi Dn}{12} = \frac{\pi(8)(1000)}{12} = 2094\ fpm$

$fhp = \frac{Fv_m}{33{,}000} = \frac{(40.08)(2094)}{33{,}000} = 2.54\ hp > 2.46\ hp$

$\frac{c_d}{D} < 0.0022$

$\frac{c_d}{D} = 0.0020$

$c_d = 0.0020(8) = 0.0160\ in$

$c_r = 0.0080\ in$

$\frac{h_o}{c_r} = \frac{0.002}{0.008} = 0.25$

Table AT 21, $L/D = 1/2$

$\frac{r}{c_r} f = 3.26$

$\left(\frac{4}{0.0016}\right) f = 3.26$

$f = 0.00652$

$$F = fW = (0.00652)(6000) = 39.12\ lb$$
$$v_m = \frac{\pi Dn}{12} = \frac{\pi(8)(1000)}{12} = 2094\ fpm$$
$$fhp = \frac{Fv_m}{33,000} = \frac{(39.12)(2094)}{33,000} = 2.48\ hp > 2.46\ hp$$

567. A 120° partial bearing supports 3500 lb. when $n = 250\ rpm$; $D = 5\ in.$, $L = 5\ in.$; $\mu = 3\times10^{-6}\ reyn$. What are the clearance and minimum film thickness for an optimum bearing (a) for maximum load, (b) for minimum friction? (c) On the basis of the average clearance in Table 3.1, about what class fit is involved? Would this fit be on the expensive or inexpensive side? (d) Find the fhp for each optimum bearing.

Solution:

$$D = 5\ in.$$
$$L = 5\ in.$$
$$\frac{L}{D} = 1$$
$$n = 250\ rpm$$
$$n_s = \frac{250}{60} = 4.17\ rps$$
$$\mu = 3\times10^{-6}\ reyn$$
$$W = 3500\ lb$$
$$p = \frac{W}{DL} = \frac{3500}{(5)(5)} = 140\ psi$$

(a) Table AT 22, $\frac{L}{D} = 1$, max. load $\frac{h_o}{c_r} = 0.46$

$$S = 0.229$$
$$S = \left(\frac{r}{c_r}\right)^2 \frac{\mu n_s}{p}$$
$$r = \frac{D}{2} = 2.5\ in$$
$$S = 0.229 = \left(\frac{2.5}{c_r}\right)^2 \frac{(3.0\times10^{-6})(4.17)}{140}$$
$$c_r = 0.00156\ in$$
$$h_o = 0.46c_r = 0.46(0.00156) = 0.00072\ in$$

(b) Table AT 22, $\frac{L}{D}=1$, min. friction $\frac{h_o}{c_r}=0.40$

$$S = 0.162$$
$$S = \left(\frac{r}{c_r}\right)^2 \frac{\mu n_s}{p}$$
$$r = \frac{D}{2} = 2.5\ in$$
$$S = 0.162 = \left(\frac{2.5}{c_r}\right)^2 \frac{\left(3.0\times10^{-6}\right)(4.17)}{140}$$
$$c_r = 0.00186\ in$$
$$h_o = 0.46c_r = 0.40(0.00186) = 0.00074\ in$$

(c) $c_{d1} = 2(0.00156) = 0.00312\ in$

$c_{d2} = 2(0.00186) = 0.00372\ in$

Use Class RC4, ave. $c_d = 0.00320\ in$, expensive side

(d) Table AT 22, $\frac{L}{D}=1$, max. load $\frac{h_o}{c_r}=0.46$

$$\frac{r}{c_r} f = 2.592$$
$$\left(\frac{2.5}{0.00156}\right) f = 2.592$$
$$f = 0.00162$$
$$F = fW = (0.00162)(3500) = 5.67\ lb$$
$$v_m = \frac{\pi D n}{12} = \frac{\pi(5)(250)}{12} = 327.25\ fpm$$
$$fhp = \frac{F v_m}{33{,}000} = \frac{(5.67)(327.25)}{33{,}000} = 0.0562\ hp$$

For minimum friction, $\frac{h_o}{c_r}=0.40$

$$\frac{r}{c_r} f = 2.16$$
$$\left(\frac{2.5}{0.00186}\right) f = 2.16$$
$$f = 0.00161$$
$$F = fW = (0.00161)(3500) = 5.635\ lb$$

$$v_m = \frac{\pi D n}{12} = \frac{\pi(5)(250)}{12} = 327.25\ fpm$$

$$fhp = \frac{F v_m}{33,000} = \frac{(5.635)(327.25)}{33,000} = 0.0559\ hp$$

570. A 180° partial bearing is to support 17,000 lb. with $p = 200\ psi$, $n = 1500\ rpm$, $h_o = 0.003\ in$, $L/D = 1$. (a) Determine the clearance for an optimum bearing with minimum friction. (b) Taking this clearance as the average, choose a fit (Table 3.1) that is approximately suitable. (c) Select an oil for an average temperature of 150 F. (d) Compute fhp.

Solution:

$W = 17,000\ lb$

$p = 200\ psi$

$n = 1500\ rpm$

$$n_s = \frac{1500}{60} = 25\ rps$$

$L/D = 1$

$L = D$

$$p = \frac{W}{DL}$$

$$200 = \frac{17,000}{D^2}$$

$D = L = 9.22\ in$

$$r = \frac{D}{2} = \frac{9.22}{2} = 4.61\ in$$

(a) For optimum bearing with minimum friction

Table AT 21, $L/D = 1$, $h_o/c_r = 0.44$

$h_o/c_r = 0.44$

$$\frac{0.003}{c_r} = 0.44$$

$c_r = 0.00682\ in$

(b) Table 3.1, $D = 9.22\ in$

$c_d = 2c_r = 2(0.00682) = 0.01364\ in$

Use Class RC7, average $c_d = 0.01065\ in$

Or use Class RC8, average $c_d = 0.01575\ in$

(c) Table AT 21, $L/D = 1$, $h_o/c_r = 0.44$

$S = 0.158$

$$S = \left(\frac{r}{c_r}\right)^2 \frac{\mu n_s}{p}$$

$$0.158 = \left(\frac{4.61}{0.00682}\right)^2 \frac{\mu(25)}{200}$$

$\mu = 2.8\times 10^{-6}$ *reyn*
Fig. AF 16, at 150 F
Use Either SAE 20W or SAE 30.

(d) Table AT 21, $L/D = 1$, $h_o/c_r = 0.44$

$$\frac{r}{c_r} f = 2.546$$

$$\left(\frac{4.61}{0.00682}\right) f = 2.546$$

$$f = 0.00377$$

$$v_m = \frac{\pi D n}{12} = \frac{\pi(9.22)(1500)}{12} = 3621\ fpm$$

$$F = fW = (0.00377)(17{,}000) = 64.09\ lb$$

$$fhp = \frac{F v_m}{33{,}000} = \frac{(64.09)(3621)}{33{,}000} = 7.0\ hp$$

571. The reaction on a 120° partial bearing is 2000 lb. The 3-in journal turns at 1140 rpm; $c_d = 0.003\ in.$; the oil is SAE 20W at an average operating temperature of 150 F. Plot curves for the minimum film thickness and the frictional loss in the bearing against the ratio L/D, using $L/D =$ 0.25, 0.5, 1, and 2. (Note: This problem may be worked as a class problem with each student being responsible for a particular L/D ratio.)

Solution:

$W = 2000\ lb$

$D = 3\ in.$

$n = 1140\ rpm$

$$n_s = \frac{1140}{60} = 19\ rps$$

$c_d = 0.003\ in$

$c_r = 0.0015\ in$

For SAE 20W, 150 F

$\mu = 2.75\times 10^{-6}$ *reyn*

(a) $\frac{L}{D} = 0.25$

$L = 0.25D = 0.25(3) = 0.75\ in$

$p = \frac{W}{DL} = \frac{2000}{(3)(0.75)} = 889\ psi$

Table AT 22, $\frac{L}{D} = 0.25$

$r = \frac{D}{2} = 1.5\ in$

$S = \left(\frac{r}{c_r}\right)^2 \frac{\mu n_s}{p} = \left(\frac{1.5}{0.0015}\right)^2 \frac{(2.75\times10^{-6})(19)}{889} = 0.0588$

$\frac{h_o}{c_r} = 0.083$

$h_o = 0.083(0.0015) = 0.000125\ in$

$\frac{r}{c_r} f = 2.193$

$\left(\frac{1.5}{0.0015}\right) f = 2.193$

$f = 0.002193$

$F = fW = (0.002193)(2000) = 4.386\ lb$

$v_m = \frac{\pi D n}{12} = \frac{\pi(3)(1140)}{12} = 895\ fpm$

$fhp = \frac{F v_m}{33{,}000} = \frac{(4.386)(895)}{33{,}000} = 0.119\ hp$

(b) $\frac{L}{D} = 0.5$

$L = 0.5D = 0.5(3) = 1.5\ in$

$p = \frac{W}{DL} = \frac{2000}{(3)(1.5)} = 444\ psi$

Table AT 22, $\frac{L}{D} = 0.5$

$r = \frac{D}{2} = 1.5\ in$

$S = \left(\frac{r}{c_r}\right)^2 \frac{\mu n_s}{p} = \left(\frac{1.5}{0.0015}\right)^2 \frac{(2.75\times10^{-6})(19)}{444} = 0.1177$

$\frac{h_o}{c_r} = 0.2159$

$h_o = 0.2159(0.0015) = 0.000324\ in$

$\frac{r}{c_r} f = 2.35$

$$\left(\frac{1.5}{0.0015}\right)f = 2.35$$
$$f = 0.00235$$
$$F = fW = (0.00235)(2000) = 4.7\ lb$$
$$v_m = \frac{\pi D n}{12} = \frac{\pi(3)(1140)}{12} = 895\ fpm$$
$$fhp = \frac{F v_m}{33{,}000} = \frac{(4.7)(895)}{33{,}000} = 0.1275\ hp$$

(c) $\frac{L}{D} = 1$

$$L = D = 3\ in$$
$$p = \frac{W}{DL} = \frac{2000}{(3)(3)} = 222\ psi$$

Table AT 22, $\frac{L}{D} = 1$

$$r = \frac{D}{2} = 1.5\ in$$
$$S = \left(\frac{r}{c_r}\right)^2 \frac{\mu n_s}{p} = \left(\frac{1.5}{0.0015}\right)^2 \frac{(2.75\times10^{-6})(19)}{222} = 0.2354$$
$$\frac{h_o}{c_r} = 0.4658$$
$$h_o = 0.4658(0.0015) = 0.000699\ in$$
$$\frac{r}{c_r} f = 2.634$$
$$\left(\frac{1.5}{0.0015}\right)f = 2.634$$
$$f = 0.002634$$
$$F = fW = (0.002634)(2000) = 5.268\ lb$$
$$v_m = \frac{\pi D n}{12} = \frac{\pi(3)(1140)}{12} = 895\ fpm$$
$$fhp = \frac{F v_m}{33{,}000} = \frac{(5.268)(895)}{33{,}000} = 0.1429\ hp$$

(d) $\frac{L}{D} = 2$

$$L = 2D = 2(3) = 6\ in$$
$$p = \frac{W}{DL} = \frac{2000}{(3)(6)} = 111\ psi$$

Table AT 22, $\frac{L}{D} = 2$

$$r = \frac{D}{2} = 1.5\ in$$

$$S = \left(\frac{r}{c_r}\right)^2 \frac{\mu n_s}{p} = \left(\frac{1.5}{0.0015}\right)^2 \frac{(2.75\times10^{-6})(19)}{111} = 0.47$$

$$\frac{h_o}{c_r} = 0.718$$

$$h_o = 0.718(0.0015) = 0.00108\ in$$

$$\frac{r}{c_r} f = 3.8118$$

$$\left(\frac{1.5}{0.0015}\right) f = 3.8118$$

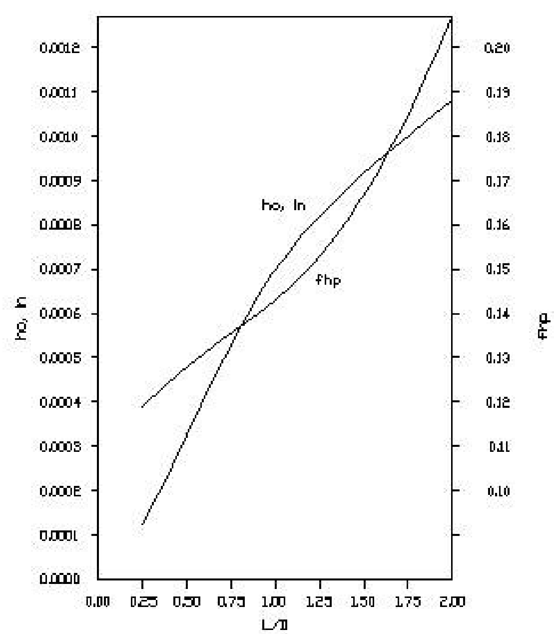
$$f = 0.003812$$

$$F = fW = (0.003812)(2000) = 7.624\ lb$$

$$v_m = \frac{\pi D n}{12} = \frac{\pi(3)(1140)}{12} = 895\ fpm$$

$$fhp = \frac{F v_m}{33{,}000} = \frac{(7.624)(895)}{33{,}000} = 0.2068\ hp$$

$\frac{L}{D}$	h_o, *in*	*fhp*
0.25	0.000125	0.119
0.5	0.000324	0.128
1.0	0.000699	0.143
2.0	0.001080	0.207



STEADY-STATE TEMPERATURE

572. A 180° partial bearing is subjected to a load of 12,000 lb.; $D \times L = 8 \times 8\ in.$, $c_r / r = 0.0015$, $h_o \approx 0.0024\ in.$, $n = 500\ rpm$. The air speed about the bearing is expected to be in excess of 1000 fpm (on moving vehicle) and the effective radiating area is $20DL$. Determine: (a) the eccentricity factor, (b) μreyns, (c) the frictional loss (ft-lb/min), (d) the estimated temperature of oil and bearing (a self-contained oil-bath unit) for steady-state operation, and a suitable oil.(e) Compute Δt_o of the oil passing through the load-carrying area, remark on its reasonableness, and decide upon whether some redesign is desirable.

Solution:

$D = 8\ in.$

$L = 8\ in.$

$L/D = 1$

$W = 12{,}000\ lb$

$r = \frac{D}{2} = 4\ in$

$c_r = 0.0015r = 0.0015(4) = 0.0060\ in$

$\frac{h_o}{c_r} = \frac{0.0024}{0.0060} = 0.4$

$n = 500\ rpm$

$n_s = \frac{500}{60} = 8.33\ rps$

Table AT 21, $h_o/c_r = 0.4$, $L/D = 1$

$S = 0.128$

$\frac{r}{c_r} f = 2.28$

$\frac{\rho c \Delta t_o}{p} = 12.4$

$p = \frac{W}{DL} = \frac{12{,}000}{(8)(8)} = 187.5\ psi$

(a) $\varepsilon = 1 - \frac{h_o}{c_r} = 1 - 0.4 = 0.6$

(b) $S = \left(\frac{r}{c_r}\right)^2 \frac{\mu n_s}{p}$

$S = 0.128 = \left(\frac{4}{0.0060}\right)^2 \frac{\mu (8.33)}{187.5}$

$\mu = 6.5 \times 10^{-6}\ reyn$

(c) $\frac{r}{c_r} f = 2.28$

$\left(\frac{4}{0.0060}\right) f = 2.28$

$f = 0.00342$

$F = fW = (0.00342)(12{,}000) = 41.04\ lb$

$v_m = \frac{\pi D n}{12} = \frac{\pi(8)(500)}{12} = 1047\ fpm$

$fhp = \frac{F v_m}{33{,}000} = \frac{(41.04)(1047)}{33{,}000} = 1.302\ hp$

Frictional loss = 43,000 ft-lb/min

(d) $Q = h_{cr} A_b \Delta t_b$ ft-lb/min

$Q = 43{,}000\ ft - lb/\min$

$h_{cr} = h_c + h_r$

$h_r = 0.108\ ft - lb/\min - sq.in. - F$

$h_c = 0.017 \frac{v_a^{0.6}}{D^{0.4}}$, $v_a \geq 1000\ fpm$

$h_c = 0.017 \frac{(1000)^{0.6}}{(8)^{0.4}} = 0.467\ ft - lb/\min - sq.in. - F$

$h_{cr} = 0.467 + 0.108 = 0.575\ ft - lb/\min - sq.in. - F$

$A_b = 20DL = 20(8)(8) = 1280\ sq.in.$

$Q = h_{cr} A_b \Delta t_b$

$43{,}000 = (0.575)(1280)(\Delta t_b)$

$\Delta t_b = 58.42\ F$

Oil-bath, 1000 fpm

$\Delta t_{oa} \approx (1.2)(1.3)(\Delta t_b)$

$\Delta t_{oa} = (1.2)(1.3)(58.42) = 91.1\ F$

assume 100 F ambient temperature

$t_b = 100 + 58.42\ F = 158.42\ F$

$t_b = 100 + 91.1\ F = 191.1\ F$

(c) $\frac{\rho c \Delta t_o}{p} = 12.4$

$\frac{(112)\Delta t_o}{187.5} = 12.4$

$\Delta t_o = 20.8\ F$

Solve for t_{o2}

$t_{o1} + t_{o2} = 2(191.1) = 382.2\ F$

$t_{o1} = 382.2 - t_{o2}$

$t_{o2} - t_{o1} = 20.8\ F$

$t_{o2} - 362.2 + t_{o2} = 20.8$

$t_{o2} = 201.5\ F \approx 200\ F$

$\therefore$ not reasonable since the oil oxidizes more rapidly above 200 F, a redesign is desireable.

573. A 2 x 2-in. full bearing (ring-oiled) has a clearance ratio $c_d/D = 0.001$. The journal speed is 500 rpm, $\mu = 3.4\times10^{-6}\ reyn$, and $h_o = 0.0005\ in.$ The ambient temperature is 100 F; $A_b = 25DL$, and the transmittance is taken as $h_{cr} = 2\ Btu/hr - sq.ft. - F$. Calculate (a) the total load for this condition; (b) the frictional loss, (c) the average temperature of the oil for steady-state operation. Is this temperature satisfactory? (d) For the temperature found, what oil do you recommend? For this oil will h_o be less or greater than the specified value? (e) Compute the temperature rise of the oil as it passes through the bearing. Is this compatible with other temperatures found? (f) What minimum quantity of oil should the ring deliver to the bearing?

Solution:

$L = 2\ in.$

$D = 2\ in.$

$c_d/D = 0.001$

$c_d = (0.001)(2) = 0.0020\ in$

$\mu = 3.4\times10^{-6}\ reyn$

$h_o = 0.0005\ in.$

$c_r = 0.0010\ in$

$h_o/c_r = 0.0005/0.0010 = 0.5$

Table AT 20, $L/D = 1$, $h_o/c_r = 0.5$, Full Bearing

$S = 0.1925$

$$\frac{r}{c_r} f = 4.505$$

$$\frac{q}{r c_r n_s L} = 4.16$$

$$\frac{\rho c \Delta t_o}{p} = 19.25$$

(a) $S = 0.1925$

$$S=\left(\frac{r}{c_r}\right)^2\frac{\mu n_s}{p}$$
$$r=\frac{D}{2}=1\,in$$
$$n_s=\frac{500}{60}=8.33\,rps$$
$$S=0.1925=\left(\frac{1}{0.0010}\right)^2\frac{(3.4\times10^{-6})(8.330)}{p}$$
$$p=147\ psi$$
$$W=pDL=(147)(2)(2)=588\,lb$$

(b) $\frac{r}{c_r}f=4.505$

$$\left(\frac{1}{0.001}\right)f=4.505$$
$$f=0.004505$$
$$F=fW=(0.004505)(588)=2.649\,lb$$
$$v_m=\frac{\pi Dn}{12}=\frac{\pi(2)(500)}{12}=261.8\ fpm$$
$$U_f=Fv_m=(2.649)(261.8)=693.5\,ft-lb/\min$$

(c) $Q=h_{cr}A_b\Delta t_b$

$$h_{cr}=2Btu/hr-sq.ft.-F=0.18\,ft-lb/\min-sq.in.-F$$
$$A_b=25DL=25(2)(2)=100\,sq.in.$$
$$Q=U_f$$
$$(0.18)(100)(\Delta t_b)=693.5$$
$$\Delta t_b=38.53\,F$$
$$\Delta t_{oa}=2\Delta t_b=2(38.53)=77\ F$$
$t_o=77+100=177\ F$, near 160 F

∴ satisfactory.

(d) $t_o=177\ F$, $\mu=3.4\times10^{-6}\ reyn$

Figure AF 16

Use SAE 40 oil, $\mu=3.3\times10^{-6}\ reyn$

$$S=\left(\frac{r}{c_r}\right)^2\frac{\mu n_s}{p}$$

$$S=\left(\frac{1}{0.0010}\right)^2\frac{(3.3\times10^{-6})(8.33)}{147}=0.187$$

Table AT 20, $L/D=1$, $S=0.187$

$h_o/c_r=0.4923$

$h_o=0.4923(0.0010)=0.00049\ in<h_o(=0.0005\ in)$

(e) $\dfrac{\rho c\Delta t_o}{p}=19.25$

$\dfrac{(112)\Delta t_o}{147}=19.25$

$\Delta t_o=25.3\ F$

$\Delta t_{o1}+\Delta t_{o2}=2(177)=354\ F$

$\Delta t_{o2}-\Delta t_{o1}=25.3\ F$

$2\Delta t_{o2}=354+25.3$

$\Delta t_{o2}=190\ F<200\ F$

$\therefore$ compatible.

(f) $\dfrac{q}{rc_r n_s L}=4.16$

$\dfrac{q}{(1)(0.001)(8.33)(2)}=4.16$

$q=0.0693\ in^3/\sec$

574. An 8 x 9-in. full bearing (consider $L/D=1$ for table and chart use only) supports 15 kips with $n=1200\ rpm$; $c_r/r=0.0012$; construction is medium heavy with a radiating-and-convecting area of about $18DL$; air flow about the bearing of 80 fpm may be counted on (nearby) pulley; ambient temperature is 90 F. Decide upon a suitable minimum film thickness. (a) Compute the frictional loss and the steady state temperature. Is additional cooling needed for a reasonable temperature? Determine (b) the temperature rise of the oil as it passes through the load-carrying area and the grade of oil to be used if it enters the bearing at 130 F, (c) the quantity of oil needed.

Solution:

$D=8\ in.$

$L=9\ in.$

$W=15{,}000\ lb.$

$n=1200\ rpm.$

$n_s=\dfrac{1200}{60}=20\ rps$

$c_r/r=0.0012$

$r = D/2 = 4\ in$

$c_r = 0.0012(4) = 0.0048\ in$

By Norton: $h_o = 0.00025D = 0.00025(8) = 0.002\ in$

$$\frac{h_o}{c_r} = \frac{0.002}{0.0048} = 0.4$$

Table AT 20, $L/D = 1$, $h_o/c_r = 0.4$

$S = 0.121$

$$\frac{r}{c_r} f = 3.22$$

$$\frac{q}{rc_r n_s L} = 4.33$$

$$\frac{\rho c \Delta t_o}{p} = 14.2$$

(a) $\frac{r}{c_r} f = 3.22$

$$\left(\frac{4}{0.0048}\right) f = 3.22$$

$f = 0.003864$

$F = fW = (0.003864)(15{,}000) = 57.96\ lb$

$$v_m = \frac{\pi D n}{12} = \frac{\pi(8)(1200)}{12} = 2513\ fpm$$

$U_f = Fv_m = (57.96)(2513) = 145{,}654\ ft-lb/\text{min}$

$Q = h_{cr} A_b \Delta t_b$

$h_r = 0.108\ ft-lb/\text{min}-sq.in.-F$

$$h_c = 0.017\frac{v_a^{0.6}}{D^{0.4}}\ ft-lb/\text{min}-sq.in.-F$$

$$h_c = 0.017\frac{(80)^{0.6}}{(8)^{0.4}} = 0.103\ ft-lb/\text{min}-sq.in.-F$$

$h_{cr} = h_c + h_r = 0.103 + 0.108 = 0.211\ ft-lb/\text{min}-sq.in.-F$

$A_b = 18DL = 18(8)(9) = 1296\ sq.in.$

$U_f = Q$

$145{,}654 = (0.211)(1296)\Delta t_b$

$\Delta t_b = 533\ F$, very high, additional cooling is necessary.

(b) $\frac{\rho c \Delta t_o}{p} = 14.2$

$$p = \frac{W}{DL} = \frac{15{,}000}{(8)(9)} = 208\ psi$$

$$\frac{(112)\Delta t_o}{208} = 14.2$$

$$\Delta t_o = 26\ F$$

$$t_i = 130\ F$$

$$t_o = 156\ F$$

$$t_{ave} = \tfrac{1}{2}(130 + 156) = 143\ F$$

$$S = \left(\frac{r}{c_r}\right)^2 \frac{\mu n_s}{p}$$

$$S = 0.121 = \left(\frac{4}{0.0048}\right)^2 \frac{\mu(20)}{208}$$

$$\mu = 1.8 \times 10^{-6}\ reyn$$

Figure AF 16, $\mu = 1.8\ \mu reyns$, 143 F
Use SAE 10W

(c) $\dfrac{q}{rc_r n_s L} = 4.33$

$$\frac{q}{(4)(0.0048)(20)(9)} = 4.33$$

$$q = 14.96\ in^3/\sec$$

575. A 3.5 x 3.5-in., 360° bearing has $c_r/r = 0.0012$; $n = 300\ rpm$; desired minimum $h_o \approx 0.0007\ in$. It is desired that the bearing be self-contained (oil-ring); air-circulation of 80 fpm is expected; heavy construction, so that $A_b \approx 25DL$. For the first look at the bearing, assume $\mu = 2.8 \times 10^{-6}\ reyn$ and compute (a) the frictional loss (ft-lb/min), (b) the average temperature of the bearing and oil as obtained for steady-state operation, (c) Δt_o as the oil passes through the load-carrying area (noting whether comparative values are reasonable). (d) Select an oil for the steady-state temperature and decide whether there will be any overheating troubles.

Solution:

$$D = 3.5\ in.$$

$$L = 3.5\ in.$$

$$c_r/r = 0.0012$$

$$r = D/2 = 1.75\ in.$$

$$c_r = (0.0012)(1.75) = 0.0021\ in$$

$$h_o \approx 0.0007\ in$$

$h_o/c_r = 0.0007/0.0021 = 0.333$

Table AT 20, 360° Bearing, $L/D = 1$, $h_o/c_r = 0.333$

$S = 0.0954$

$\frac{r}{c_r} f = 2.71$

$\frac{\rho c \Delta t_o}{p} = 12.12$

(a) $S = \left(\frac{r}{c_r}\right)^2 \frac{\mu n_s}{p}$

$n_s = \frac{300}{60} = 5\ rps$

$\mu = 2.8\times10^{-6}\ reyn$

$S = 0.0954 = \left(\frac{1.75}{0.0021}\right)^2 \frac{(2.8\times10^{-6})(5)}{p}$

$p = 102\ psi$

$W = pDL = (102)(3.5)(3.5) = 1250\ lb$

$\frac{r}{c_r} f = 2.71$

$\left(\frac{1.75}{0.0021}\right) f = 2.71$

$f = 0.00325$

$F = fW = (0.00325)(1250) = 4.0625\ lb$

$v_m = \frac{\pi D n}{12} = \frac{\pi(3.5)(300)}{12} = 275\ fpm$

$U_f = F v_m = (4.0625)(275) = 1117\ ft-lb/\min$

(b) $Q = h_{cr} A_b \Delta t_b$

$h_r = 0.108\ ft-lb/\min - sq.in. - F$

$h_c = 0.017 \frac{v_a^{0.6}}{D^{0.4}}\ ft-lb/\min - sq.in. - F$

$h_c = 0.017 \frac{(80)^{0.6}}{(3.5)^{0.4}} = 0.143\ ft-lb/\min - sq.in. - F$

$h_{cr} = h_c + h_r = 0.143 + 0.108 = 0.251\ ft-lb/\min - sq.in. - F$

$A_b = 25DL = 25(3.5)(3.5) = 306.25\ sq.in.$

$U_f = Q$

$1117 = (0.251)(306.25)\Delta t_b$

$\Delta t_b = 14.5\ F$

$\Delta t_{oa} = 2\Delta t_b = 2(14.5) = 29\ F$

assume ambient temperature of 100 F

$t_b = 114.5\ F$

$t_o = 129\ F$

(c) $\dfrac{\rho c \Delta t_o}{p} = 12.12$

$\dfrac{(112)\Delta t_o}{102} = 12.12$

$\Delta t_o = 11\ F$

$t_{o1} + t_{o2} = 2(129) = 258\ F$

$t_{o2} - t_{o1} = 11\ F$

$2t_{o2} = 269\ F$

$t_{o2} = 135\ F < 140\ F$

$\therefore$ reasonable

(d) $t_o = 129\ F$, $\mu = 2.8\times 10^{-6}\ reyn$

use SAE 10W

Figure AF 16, $t_o = 126\ F$

$\Delta t_{oa} = 126 - 100 = 26\ F$

$\Delta t_{oa} = 2\Delta t_b$

$\Delta t_b = \dfrac{26}{2} = 13\ F$

$Q = h_{cr} A_b \Delta t_b = (0.251)(306.25)(13) = 999\ ft-lb/\min < U_f$

$\therefore$ there is an overheating problem.

576. A 10-in. full journal for a steam-turbine rotor that turns 3600 rpm supports a 20-kip load with $p = 200\ psi$; $c_r/r = 0.00133$. The oil is to have $\mu = 2.06\times 10^{-6}\ reyn$ at an average oil temperature of 130 F. Compute (a) the minimum film thickness (comment on its adequacy), (b) the fhp, (c) the altitude angle, the maximum pressure, and the quantity of oil that passes through the load-carrying area (gpm).(d) At what temperature must the oil be introduced in order to have 130 F average? (e) Estimate the amount of heat lost by natural means from the bearing (considered oil bath) with air speed of 300 fpm. If the amount of oil flow computed above is cooled back to the entering temperature, how much heat is removed? Is this total amount of heat enough to care for frictional loss? If not, what can be done (§11.21)?

Solution:

$D = 10\ in.$

$n = 3600\ rpm$

$n_s = \frac{3600}{60} = 60\ rps$

$W = 20{,}000\ lb$

$p = 200\ psi$

$p = \frac{W}{DL}$

$200 = \frac{20{,}000}{10L}$

$L = 10\ in$

$L/D = 1$

$r = \frac{D}{2} = 5\ in$

$c_r/r = 0.00133$

$c_r = 0.00133(5) = 0.00665\ in$

$\mu = 2.06\times10^{-6}\ reyn$

$t_{ave} = 130\ F$

$S = \left(\frac{r}{c_r}\right)^2 \frac{\mu n_s}{p}$

$S = \left(\frac{5}{0.00665}\right)^2 \frac{(2.06\times10^{-6})(60)}{200} = 0.35$

Table AT 20, $L/D = 1$, $S = 0.35$

$h_o/c_r = 0.647$

$\phi = 65.66^\circ$

$\frac{r}{c_r} f = 7.433$

$\frac{q}{r c_r n_s L} = 3.90$

$\frac{p}{p_{max}} = 0.495$

$\frac{\rho c \Delta t_o}{p} = 30.8$

$\frac{q_s}{q} = 0.446$

(a) $h_o = 0.647 c_r = 0.647(0.00665) = 0.00430\ in$

Norton's recommendation = $0.00025D = 0.00025(10) = 0.00250\ in < 0.00430\ in$
$\therefore$ adequate

(b) $\frac{r}{c_r} f = 7.433$

$$\left(\frac{5}{0.00665}\right) f = 7.433$$
$$f = 0.0099$$
$$F = fW = (0.0099)(20{,}000) = 198\ lb$$
$$v_m = \frac{\pi D n}{12} = \frac{\pi(10)(3600)}{12} = 9425\ fpm$$
$$fhp = \frac{F v_m}{33{,}000} = \frac{(198)(9425)}{33{,}000} = 56.55\ hp$$

(c) $\phi = 65.66^\circ$

$$p_{\max} = \frac{p}{0.495} = \frac{200}{0.495} = 404\ psi$$
$$q = 3.90 r c_r n_s L$$
$$q = 3.90(5)(0.00665)(60)(10) = 77.805\ in^3/\sec$$
$$q = (77.805\ in^3/\sec)(1\ gpm/231\ in^3)(60\ \sec/\min) = 0.21\ gpm$$

(d) $\frac{\rho c \Delta t_o}{p} = 30.8$

$$\frac{(112)\Delta t_o}{200} = 30.8$$
$$\Delta t_o = 55\ F$$
$$t_{ave} = t_i + \frac{\Delta t_o}{2}$$
$$130 = t_i + \frac{55}{2}$$
$$t_i = 102.5\ F$$

(e) $Q = h_{cr} A_b \Delta t_b$

$$h_r = 0.108\ ft - lb/\min - sq.in. - F$$
$$h_c = 0.017 \frac{v_a^{0.6}}{D^{0.4}}\ ft - lb/\min - sq.in. - F$$
$$h_c = 0.017 \frac{(300)^{0.6}}{(3.5)^{0.4}} = 0.207\ ft - lb/\min - sq.in. - F$$
$$h_{cr} = h_c + h_r = 0.207 + 0.108 = 0.315\ ft - lb/\min - sq.in. - F$$

Assume

$A_b = 25DL = 25(10)(10) = 2500\ sq.in.$

$\Delta t_{oa} = 130 - 100 = 30\ F$

$\Delta t_{oa} = 1.3\Delta t_b$

$\Delta t_b = \dfrac{30}{1.3} = 23\ F$

$Q = (0.315)(2500)(23) = 18{,}113\ ft-lb/\text{min}$

$Q_r = \rho c(q - q_s)\Delta t_o\ in - lb/\sec$

$Q_r = (112)(1 - 0.446)(77.805)(55)(1/12)(60) = 1{,}327{,}602\ ft - lb/\text{min}$

$Q_T = Q + Q_r = 18{,}113 + 1{,}327{,}602 = 1{,}345{,}735\ ft - lb/\text{min}$

$U_f = Fv_m = (198)(9425) = 1{,}866{,}150\ ft - lb/\text{min} > Q_T$

not enough to care for frictional loss, use pressure feed (§11.21).

DESIGN PROBLEMS

578. A 3.5-in. full bearing on an air compressor is to be designed for a load of 1500 lb.; $n = 300\ rpm$; let $L/D = 1$. Probably a medium running for would be satisfactory. Design for an average clearance that is decided by considering both Table 3.1 and 11.1. Choose a reasonable h_o, say one that gives $h_o/c_r \approx 0.5$. Compute all parameters that are available via the Text after you have decided on details. It is desired that the bearing operate at a reasonable steady-state temperature (perhaps ring-oiled medium construction), without special cooling. Specify the oil to be used and show all calculations to support your conclusions. What could be the magnitude of the maximum impulsive load if the eccentricity ration ε becomes 0.8, "squeeze" effect ignored?

Solution:

$L/D = 1$

$D = 3.5\ in$

$L = 3.5\ in$

$W = 1500\ lb$

$n = 300\ rpm$

$n_s = \dfrac{300}{60} = 5\ rps$

$p = \dfrac{W}{DL} = \dfrac{1500}{(3.5)(3.5)} = 122.45\ psi$

Table 3.1, medium running fit,

$D = 3.5\ in$

RC 5 or RC 6

Use RC 6

Average $c_d = 0.0052\ in$

Table 11.1, air-compressor
General Machine Practice
Average $c_d = 0.0055\ in$
Using $c_d = 0.0055\ in$
$c_r = 0.00275\ in$
$h_o = 0.5c_r = 0.5(0.00275) = 0.001375\ in$
Table AT 20, $L/D = 1$, $h_o/c_r = 0.5$
$\varepsilon = 0.5$
$S = 0.1925$
$\phi = 56.84^\circ$

$$\frac{r}{c_r}f = 4.505$$

$$\frac{q}{rc_r n_s L} = 4.16$$

$$\frac{\rho c \Delta t_o}{p} = 19.25$$

$$\frac{p}{p_{\max}} = 0.4995$$

Specifying oil:
$Q = h_{cr}A_b\Delta t_b$
$U_f = Fv_m$

$$\frac{r}{c_r}f = 4.505$$

$$\left(\frac{1.75}{0.00275}\right)f = 4.505$$

$f = 0.00708$
$F = fW = (0.00708)(1500) = 10.62\ lb$

$$v_m = \frac{\pi Dn}{12} = \frac{\pi(3.5)(300)}{12} = 275\ fpm$$

$U_f = Fv_m = (10.62)(275) = 2921\ ft-lb/\min$
$Q = h_{cr}A_b\Delta t_b$
Assume $h_{cr} = 0.516\ ft-lb/\min-sq.in.-F$
Medium construction
$A_b = 15.5DL = 15.5(3.5)(3.5) = 189.875\ sq.in.$
Oil-ring bearing
$\Delta t_{oa} = 2\Delta t_b$
$Q = U_f$
$(0.516)(189.875)(\Delta t_b) = 2921$

$\Delta t_b = 30\ F$

$\Delta t_{oa} = 2\Delta t_b = 2(30) = 60\ F$

assume ambient temperature = 90 F

$t_o = 150\ F$

$$S = \left(\frac{r}{c_r}\right)^2 \frac{\mu n_s}{p}$$

$$S = 0.1925 = \left(\frac{1.75}{0.00275}\right)^2 \frac{\mu(5)}{122.45}$$

$\mu = 11.6\times10^{-6}\ reyn$

Figure AF 16, 150 F, $\mu \approx 11.6\times10^{-6}\ reyn$

Use SAE 70 oil

Maximum load, W with $\varepsilon = 0.8$

Table AT 20, $L/D = 1$

$S = 0.0446$

$$S = \left(\frac{r}{c_r}\right)^2 \frac{\mu n_s}{p}$$

$$S = 0.0446 = \left(\frac{1.75}{0.00275}\right)^2 \frac{(11.6\times10^{-6})(5)}{p}$$

$p = 527\ psi$

$W = pDL = (527)(3.5)(3.5) = 6456$

580. A 2500-kva generator, driven by a water wheel, operates at 900 rpm. The weight of the rotor and shaft is 15,100 lb. The left-hand, 5 –in, full bearing supports the larger load, $R = 8920\ lb$. The bearing should be above medium-heavy construction (for estimating A_b). (a) Decide upon an average clearance considering both Table 3.1 and 11.1, and upon a minimum film thickness ($h_o/c_r \approx 0.5$ is on the safer side). (b) Investigate first the possibility of the bearing being a self-contained unit without need of special cooling. Not much air movement about the bearing is expected. Then make final decisions concerning oil-clearance, and film thickness and compute all the parameters given in the text, being sure that everything is reasonable.

Solution:

$n = 900\ rpm$

$$n_s = \frac{900}{60} = 15\ rps$$

$D = 5\ in$

$W = R = 8920\ lb$

(a) Table 3.1, $D = 5\ in$
RC 5, average $c_d = 0.0051\ in$
$c_r = 0.00255\ in$
$h_o = 0.5c_r = 0.5(0.00255) = 0.00128\ in$

(b) Use $L/D = 1$
$L = 5\ in$
$r = \frac{D}{2} = 2.5\ in$
$p = \frac{W}{DL} = \frac{8920}{(5)(5)} = 356.8\ psi$
Table AT 20, $L/D = 1$, $h_o/c_r = 0.5$
$S = 0.1925$
$\frac{r}{c_r} f = 4.505$
$\frac{q}{rc_r n_s L} = 4.16$
$\frac{\rho c \Delta t_o}{p} = 19.25$
$S = \left(\frac{r}{c_r}\right)^2 \frac{\mu n_s}{p}$
$S = 0.1925 = \left(\frac{2.5}{0.00255}\right)^2 \frac{\mu(15)}{356.8}$
$\mu = 4.8 \times 10^{-6}\ reyn$
$\frac{r}{c_r} f = 4.505$
$\left(\frac{2.5}{0.00255}\right) f = 4.505$
$f = 0.00460$
$F = fW = (0.00460)(8920) = 41.032\ lb$
$v_m = \frac{\pi D n}{12} = \frac{\pi(5)(900)}{12} = 1178\ fpm$
$U_f = F v_m = (41.032)(1178) = 48{,}336\ ft-lb/\min$
$Q = h_{cr} A_b \Delta t_b$
Medium-Heavy
$A_b = 20.25DL = 20.25(5)(5) = 506.25\ sq.in.$
Assume $h_{cr} = 0.516\ ft-lb/\min - sq.in. - F$
$Q = U_f$

$(0.516)(506.25)(\Delta t_b) = 48{,}336$

$\Delta t_b = 185\ F$, very high

Therefore, special cooling is needed.

$$\frac{\rho c \Delta t_o}{p} = 19.25$$

$$\frac{(112)\Delta t_o}{356.8} = 19.25$$

$\Delta t_o = 61\ F$

Assume $t_i = 100\ F$

$$t_{ave} = 100 + \frac{61}{2} \approx 130\ F$$

Figure AF 16, $\mu = 4.8\ \mu reyns$, 130 F

Select SAE 30 oil. $\mu = 6.0\ \mu reyns$

$$S = \left(\frac{r}{c_r}\right)^2 \frac{\mu n_s}{p}$$

$$S = \left(\frac{2.5}{0.00255}\right)^2 \frac{(6.0\times10^{-6})(15)}{356.8} = 0.242$$

Table AT 20, $L/D = 1$, $S = 0.242$

SAE 30 oil at 130 F

$$\frac{h_o}{c_r} = 0.569$$

$\phi = 61.17^\circ$

$$\frac{r}{c_r} f = 5.395$$

$$\frac{q}{r c_r n_s L} = 4.04$$

$$\frac{\rho c \Delta t_o}{p} = 22.75$$

$$\frac{p}{p_{\max}} = 0.4734$$

Oil, SAE 30

$c_r = 0.00255\ in$

$h_o = 0.569(0.00255) = 0.00145\ in$

PRESSURE FEED

581. An 8 x 8-in. full bearing supports 5 kips at 600 rpm of the journal; $c_r = 0.006\ in.$; let the average $\mu = 2.5\times10^{-6}\ reyn$. (a) Compute the frictional loss U_f. (b) The

oil is supplied under a 40-psi gage pressure with a longitudinal groove at the point of entry. Assuming that other factors, including U_f, remain the same and that the heat loss to the surrounding is negligible, determine the average temperature rise of the circulating oil.

Solution:

$L = 5\ in$

$D = 5\ in$

$W = 5000\ lb$

$n = 600\ rpm$

$$n_s = \frac{600}{60} = 10\ rps$$

$c_r = 0.006\ in$

$\mu = 2.5\times10^{-6}\ reyn$

$L/D = 1$

$$p = \frac{W}{DL} = \frac{5000}{(8)(8)} = 78.125\ psi$$

$$S = \left(\frac{r}{c_r}\right)^2 \frac{\mu n_s}{p}$$

$$S = \left(\frac{4}{0.006}\right)^2 \frac{(2.5\times10^{-6})(10)}{78.125} = 0.1422$$

(a) Table AT 20, $L/D = 1$, $S = 0.1422$

$$\frac{r}{c_r} f = 3.6,\ \varepsilon = 0.57$$

$$\left(\frac{4}{0.006}\right) f = 3.6$$

$f = 0.0054$

$F = fW = (0.0054)(5000) = 27\ lb$

$$v_m = \frac{\pi D n}{12} = \frac{\pi(8)(600)}{12} = 1257\ fpm$$

$U_f = F v_m = (27)(1257) = 33{,}940\ ft-lb/\min$

(b) Longitudinal Groove.

$$q = 2.5\frac{c_r^3 p_i}{3\mu}\left(\tan^{-1}\frac{2\pi r}{L}\right)(1+1.5\varepsilon^2)\ in^3/\sec$$

$p_i = 40\ psi$

$$q = 2.5\frac{(0.006)^3(40)}{3(2.5\times10^{-6})}\left[\tan^{-1}\frac{2\pi(4)}{8}\right]\left[1+1.5(0.57)^2\right] in^3/\sec$$

$q = 5.41\, in^3/\sec$

$U_f = \rho c q \Delta t_o$

$(33{,}940\, ft-lb/\min)(12\, in/ft)(1\min/60\sec) = (12)(5.41)\Delta t_o$

$\Delta t_o = 11.2\, F$

583. A 4-in. 360° bearing, with $L/D = 1$, supports 2.5 kips with a minimum film of $h_o = 0.0008\, in.$, $c_d = 0.01\, in.$, $n = 600\, rpm.$ The average temperature rise of the oil is to be about 25 F. Compute the pressure at which oil should be pumped into the bearing if (a) all bearing surfaces are smooth, (b) there is a longitudinal groove at the oil-hole inlet. (c) same as (a) except that there is a 360° circumferential groove dividing the bearing into 2-in. lengths.

Solution:

$D = 4\, in$

$L = 4\, in$

$r = 2\, in$

$W = 2500\, lb$

$c_d = 0.010\, in$

$c_r = 0.005\, in$

$n = 600\, rpm$

$n_s = \dfrac{600}{60} = 10\, rps$

$\Delta t_o = 25\, F$

$p = \dfrac{W}{DL} = \dfrac{2500}{(4)(4)} = 156.25\, psi$

$h_o = 0.00080\, in$

$\dfrac{h_o}{c_r} = \dfrac{0.0008}{0.005} = 0.16$

Table AT 20, $L/D = 1$, $h_o/c_r = 0.16$

$\dfrac{r}{c_r} f = 1.44$, $\varepsilon = 0.84$

$\left(\dfrac{2}{0.005}\right) f = 1.44$

$f = 0.0036$

$F = fW = (0.0036)(2500) = 9\, lb$

$v_m = \dfrac{\pi D n}{12} = \dfrac{\pi(4)(600)}{12} = 628\, fpm$

$U_f = F v_m = (9)(628) = 5652\, ft-lb/\min = 1130\, in-lb/\sec$

$S = 0.0343$

$$S=\left(\frac{r}{c_r}\right)^2\frac{\mu n_s}{p}$$

$$S=0.0343=\left(\frac{2}{0.005}\right)^2\frac{\mu(10)}{156.25}$$

$$\mu=3.35\times10^{-6}\ reyn$$

$$U_f=\rho cq\Delta t_o$$

$$1130=(112)(q)(25)$$

$$q=0.404\ in^3/\sec$$

(a) Smooth

$$q=\frac{c_r^3p_i}{3\mu}\left(\tan^{-1}\frac{2\pi r}{L}\right)\left(1+1.5\varepsilon^2\right)in^3/\sec$$

$$0.404=\frac{(0.005)^3(p_i)}{3(3.35\times10^{-6})}\left[\tan^{-1}\frac{2\pi(2)}{4}\right]\left[1+1.5(0.84)^2\right]in^3/\sec$$

$$p_i=12.5\ psi$$

(b) Longitudinal groove

$$q=\frac{2.5c_r^3p_i}{3\mu}\left(\tan^{-1}\frac{2\pi r}{L}\right)\left(1+1.5\varepsilon^2\right)in^3/\sec$$

$$0.404=\frac{2.5(0.005)^3(p_i)}{3(3.35\times10^{-6})}\left[\tan^{-1}\frac{2\pi(2)}{4}\right]\left[1+1.5(0.84)^2\right]in^3/\sec$$

$$p_i=5\ psi$$

(c) Circumferential groove

$$q=\frac{2\pi rc_r^3p_i}{3\mu L}\left(1+1.5\varepsilon^2\right)in^3/\sec$$

$$0.404=\frac{2\pi(2)(0.005)^3(p_i)}{3(3.35\times10^{-6})(4)}\left[1+1.5(0.84)^2\right]in^3/\sec$$

$$p_i=5\ psi$$

<u>BEARING CAPS</u>

584. An 8-in. journal, supported on a 150° partial bearing, is turning at 500 rpm; bearing length = 10.5 in., $c_d=0.0035\ in.$, $h_o=0.00106\ in.$ The average temperature of the SAE 20 oil is 170 F. Estimate the frictional loss in a 160° cap for this bearing.

Solution:

$h_o = 0.00106\ in$
$c_d = 0.0035\ in$
$c_r = 0.00175\ in$

$$h_{av} = c_r\left[1 + 0.74\left(1 - \frac{h_o}{c_r}\right)^2\right] in$$

$$h_{av} = (0.00175)\left[1 + 0.74\left(1 - \frac{0.00106}{0.00175}\right)^2\right] = 0.00195\ in$$

For SAE 20, 170 F

$\mu = 1.7\times10^{-6}\ reyn$

$$F = \frac{\mu A v_{ips}}{h_{av}}$$

$$A = \frac{1}{2}\theta DL$$

$D = 8\ in$
$L = 10.5\ in$

$$\theta = 160^o = \frac{160}{180}\pi = \frac{8\pi}{9}$$

$$A = \frac{1}{2}\left(\frac{8\pi}{9}\right)(8)(10.5) = 117.3\ sq.in.$$

$$v_{ips} = \pi D n_s = \pi(8)\left(\frac{500}{60}\right) = 209.5\ ips$$

$$F = \frac{(1.7\times10^{-6})(117.3)(209.5)}{0.00195} = 21.424\ lb$$

$$v_m = \frac{\pi D n}{12} = \frac{\pi(8)(500)}{12} = 1047\ fpm$$

$$U_f = F v_m = (21.424)(1047) = 22{,}430\ ft-lb/\min = 1130\ in-lb/\sec$$

585. A partial 160° bearing has a 160° cap; $D = 2\ in.$, $L = 2\ in.$, $c_d = 0.002\ in.$, $h_o = 0.0007\ in.$, $n = 500\ rpm$, and $\mu = 2.5\times10^{-6}\ reyn$. For the cap only, what is the frictional loss?

Solution:

$c_d = 0.002\ in$
$c_r = 0.001\ in$
$h_o = 0.0007\ in$

$$\frac{h_o}{c_r} = \frac{0.0007}{0.001} = 0.7$$

$$\varepsilon = 1 - \frac{h_o}{c_r} = 1 - 0.7 = 0.3$$

$$h_{av} = c_r\left(1 + 0.74\varepsilon^2\right) = (0.001)\left[1 + 0.74(0.3)^2\right] = 0.001067\ in$$

$$F = \frac{\mu A v_{ips}}{h_{av}}$$

$$v_m = \frac{\pi D n}{12} = \frac{\pi(2)(500)}{12} = 261.8\ fpm$$

$$v_{ips} = (261.8)\left(\frac{12}{60}\right) = 52.36\ ips$$

$$A = \frac{1}{2}\left(\frac{160}{180}\right)\pi DL = \frac{1}{2}\left(\frac{160}{180}\right)(\pi)(2)(2) = 5.585\ sq.in.$$

$$F = \frac{\left(2.5\times10^{-6}\right)(5.585)(52.36)}{0.001067} = 0.685\ lb$$

$$U_f = Fv_m = (0.685)(261.8) = 179.3\ ft-lb/\min$$

586. The central reaction on a 120° partial bearing is 10 kips; $D = 8\ in.$, $L/D = 1.$, $c_r/r = 0.001$. Let $n = 400\ rpm$ and $\mu = 3.4\times10^{-6}\ reyn$. The bearing has a 150° cap. (a) For the bearing and the cap, compute the total frictional loss by adding the loss in the cap to that in the bearing. (b) If the bearing were 360°, instead of partial, calculate the frictional loss and compare.

Solution:

$$S = \left(\frac{r}{c_r}\right)^2 \frac{\mu n_s}{p}$$

$$n_s = \frac{400}{60} = 6.67\ rps$$

$$p = \frac{W}{DL} = \frac{10{,}000}{(8)(8)} = 156.25\ psi$$

$$S = \left(\frac{1}{0.001}\right)^2 \frac{\left(3.4\times10^{-6}\right)(6.67)}{156.25} = 0.145$$

(a) Table AT 22, $L/D = 1$, $S = 0.145$

$$\frac{r}{c_r} f = 2.021$$

$$\varepsilon = 0.6367$$

$$\frac{r}{c_r} f = 2.021$$

$$\left(\frac{1}{0.001}\right) f = 2.021$$

$$f = 0.002021$$

$F = fW = (0.002021)(10{,}000) = 20.21\,lb$

$v_m = \frac{\pi Dn}{12} = \frac{\pi(8)(400)}{12} = 838\ fpm$

$U_{f1} = Fv_m = (20.21)(838) = 16{,}936\ ft-lb/\min$

CAP:

$h_{av} = c_r(1+0.74\varepsilon^2)$

$c_r = 0.001r$

$r = \frac{D}{2} = 4\ in$

$c_r = 0.001(4) = 0.004\ in$

$h_{av} = c_r(1+0.74\varepsilon^2) = (0.004)\left[1+0.74(0.6367)^2\right] = 0.0052\ in$

$F = \frac{\mu A v_{ips}}{h_{av}}$

$v_{ips} = (838)\left(\frac{12}{60}\right) = 167.6\ ips$

$A = \frac{1}{2}\left(\frac{150}{180}\right)\pi DL = \frac{1}{2}\left(\frac{150}{180}\right)(\pi)(8)(8) = 83.78\ sq.in.$

$F = \frac{(3.4\times10^{-6})(83.78)(167.6)}{0.0052} = 9.18\ lb$

$U_{f2} = Fv_m = (9.18)(838) = 7693\ ft-lb/\min$

Total Frictional Loss

$= U_{f1} + U_{f2} = 16{,}936 + 7693 = 24{,}629\ ft-lb/\min$

(b) 360° Bearing, $L/D = 1$, $S = 0.145$

$\frac{r}{c_r}f = 3.65$

$\varepsilon = 0.5664$

BEARING:

$\left(\frac{1}{0.001}\right)f = 3.65$

$f = 0.00365$

$F = fW = (0.00365)(10{,}000) = 36.5\ lb$

$v_m = \frac{\pi Dn}{12} = \frac{\pi(8)(400)}{12} = 838\ fpm$

$U_{f1} = Fv_m = (36.5)(838) = 30{,}587\ ft-lb/\min$

CAP:

$h_{av} = c_r(1+0.74\varepsilon^2)$

$h_{av} = c_r(1+0.74\varepsilon^2) = (0.004)\left[1+0.74(0.5664)^2\right] = 0.00495\ in$

$$F = \frac{\mu A v_{ips}}{h_{av}}$$

$$F = \frac{(3.4\times10^{-6})(83.78)(167.6)}{0.00495} = 9.645\ lb$$

$$U_{f2} = Fv_m = (9.645)(838) = 8083\ ft-lb/\min$$

Total Frictional Loss

$= U_{f1} + U_{f2} = 30{,}587 + 8083 = 38{,}670\ ft-lb/\min$

587. The central reaction on a 120° partial bearing is a 10 kips; $D = 8\ in.$, $L/D = 1$, $c_r/r = 0.001$; $n = 1200\ rpm$. Let $\mu = 2.5\times10^{-6}\ reyn$. The bearing has a 160° cap. (a) Compute h_o and fhp for the bearing and for the cap to get the total fhp. (b) Calculate the fhp for a full bearing of the same dimensions and compare. Determine (c) the needed rate of flow into the bearing, (d) the side leakage q_s.

(e) the temperature rise of the oil in the bearing both by equation (o), §11.13, Text, and by Table AT 22. (f) What is the heat loss from the bearing if the oil temperature is 180 F? Is the natural heat loss enough to cool the bearing? (g) It is desired to pump oil through the bearing with a temperature rise of 12 F. How much oil is required? (h) For the oil temperature in (f), what is a suitable oil to use?

Solution:

$$S = \left(\frac{r}{c_r}\right)^2 \frac{\mu n_s}{p}$$

$$n_s = \frac{1200}{60} = 20\ rps$$

$$p = \frac{W}{DL} = \frac{10{,}000}{(8)(8)} = 156.25\ psi$$

$$S = \left(\frac{1}{0.001}\right)^2 \frac{(2.5\times10^{-6})(20)}{156.25} = 0.32$$

(a) Table AT 22, $L/D = 1$, $S = 0.32$

$\varepsilon = 0.5417$

$$\frac{h_o}{c_r} = 0.4583$$

$$\frac{r}{c_r} f = 3.18$$

$$\frac{q}{r c_r n_s L} = 2.60$$

$\dfrac{q_s}{q} = 0.305$

$\dfrac{\rho c \Delta t_o}{p} = 17.834$

$\dfrac{p}{p_{\max}} = 0.38434$

$h_o = 0.4583 c_r = 0.4583(0.001)(4) = 0.00183\ in$

BEARING:

$\dfrac{r}{c_r} f = 3.18$

$\left(\dfrac{1}{0.001}\right) f = 3.18$

$f = 0.00318$

$F = fW = (0.00318)(10{,}000) = 31.8\ lb$

$v_m = \dfrac{\pi D n}{12} = \dfrac{\pi(8)(1200)}{12} = 2513\ fpm$

$U_{f1} = F v_m = (31.8)(2513) = 79{,}913\ ft-lb/\min$, $2.42\ hp$

CAP:

$h_{av} = c_r\left(1 + 0.74\varepsilon^2\right)$

$c_r = 0.001 r$

$r = \dfrac{D}{2} = 4\ in$

$c_r = 0.001(4) = 0.004\ in$

$h_{av} = c_r\left(1 + 0.74\varepsilon^2\right) = (0.004)\left[1 + 0.74(0.5417)^2\right] = 0.00487\ in$

$F = \dfrac{\mu A v_{ips}}{h_{av}}$

$v_{ips} = (2513)\left(\dfrac{12}{60}\right) = 503\ ips$

$A = \dfrac{1}{2}\left(\dfrac{160}{180}\right)\pi D L = \dfrac{1}{2}\left(\dfrac{160}{180}\right)(\pi)(8)(8) = 89.36\ sq.in.$

$F = \dfrac{\left(2.5\times 10^{-6}\right)(89.36)(5036)}{0.00487} = 23.1\ lb$

$U_{f2} = F v_m = (23.1)(2513) = 58{,}050\ ft-lb/\min$, $1.76\ hp$

Total Frictional Loss

$= U_{f1} + U_{f2} = 79{,}913 + 58{,}050 = 137{,}963\ ft-lb/\min$

$fhp = \dfrac{U_f}{33{,}000} = \dfrac{137{,}963}{33{,}000} = 4.18\ hp$

(b) Full Bearing, $L/D = 1$, $S = 0.32$

Table AT 20

$$\frac{h_o}{c_r} = 0.6305$$

$$\frac{r}{c_r} f = 6.86$$

$$\varepsilon = 0.3695$$

$$h_o = 0.6305(0.004) = 0.002522\ in$$

BEARING:

$$\frac{r}{c_r} f = 6.86$$

$$\left(\frac{1}{0.001}\right) f = 6.86$$

$$f = 0.00686$$

$$F = fW = (0.00686)(10{,}000) = 68.6\ lb$$

$$U_{f1} = Fv_m = (68.6)(2513) = 172{,}392\ ft-lb/\min,\ 5.224\ hp$$

CAP:

$$h_{av} = c_r\left(1 + 0.74\varepsilon^2\right)$$

$$h_{av} = c_r\left(1 + 0.74\varepsilon^2\right) = (0.004)\left[1 + 0.74(0.3695)^2\right] = 0.00440\ in$$

$$F = \frac{\mu A v_{ips}}{h_{av}}$$

$$F = \frac{\left(2.5\times10^{-6}\right)(89.36)(503)}{0.00440} = 25.54\ lb$$

$$U_{f2} = Fv_m = (25.54)(2513) = 64{,}182\ ft-lb/\min,\ 1.946\ hp$$

Total Frictional Loss

$$= U_{f1} + U_{f2} = 172{,}392 + 64{,}182 = 236{,}574\ ft-lb/\min$$

$$fhp = \frac{U_f}{33{,}000} = \frac{236{,}574}{33{,}000} = 7.17\ hp$$

(c) 120° Bearing

$$\frac{q}{rc_r n_s L} = 2.60$$

$$\frac{q}{(4)(0.004)(20)(8)} = 2.60$$

$$q = 6.656\ in^3/\sec$$

(d) $\frac{q_s}{q} = 0.305$

$$\frac{q_s}{6.656} = 0.305$$

$q_s = 2.03\ in^3/\sec$

(e) Equation(o)

$U_{f1} = \rho c q \Delta t_o$

$U_{f1} = 79{,}913\ ft-lb/\min = 79{,}913\left(\frac{12}{60}\right) in-lb/\sec = 15{,}983\ in-lb/\sec$

$U_{f1} = 15{,}983 = (112)(6.656)\Delta t_o$

$\Delta t_o = 21.4\ F$

Table 22.

$\frac{\rho c \Delta t_o}{p} = 17.834$

$\frac{112 \Delta t_o}{156.25} = 17.834$

$\Delta t_o = 24.9\ F$

(f) $Q = h_{cr} A_b \Delta t_b$

assume $h_{cr} = 0.516\ ft-lb/\min-sq.in.-F$

$A_b = 25DL = 25(8)(8) = 1600\ sq.in.$

$\Delta t_b = \frac{\Delta t_{oa}}{2}$

assume ambient = 100 F

$\Delta t_b = \frac{180-100}{2} = 40\ F$

$Q = (0.516)(1600)(40) = 33{,}024\ ft-lb/\min < U_{f1}$

Therefore not enough to cool the bearing.

(g) $Q_r + Q = U_{f1} + U_{f2}$

$Q_r + 33{,}024 = 137{,}963$

$Q_r = 104{,}939\ ft-lb/\min$

$Q_r = 20{,}988\ in-lb/\sec$

$Q_r = \rho c q \Delta t_o$

$20{,}988 = (112)q(12)$

$q = 15.62\ in^3/\sec$

(h) Fig. AF 16, 180 F, $\mu = 2.5\times10^{-6}\ reyn$

use SAE 30 oil

IMPERFECT LUBRICATION:

588. A 0.5 x 0.75-in. journal turns at 1140 rpm. What maximum load may be supported and what is the frictional loss if the bearing is (a) SAE Type I, bronze base, sintered bearing, (b) nylon (Zytel) water lubricated, (c) Teflon, with intermittent use, (d) one with carbon graphite inserts.

Solution:

(a) $f = 0.12$

$$v_m = \frac{\pi D n}{12} = \frac{\pi(0.5)(1140)}{12} = 149.23\ fpm$$

$pv_m = 50{,}000$

$p(149.23) = 50{,}000$

$p = 335\ psi$

$W = pDL = (335)(0.5)(0.75) = 126\ lb$

$F = fW = (0.12)(126) = 15.12\ lb$

$U_f = Fv_m = (15.12)(149.23) = 2256\ ft-lb/\min$

(b) $f = 0.14 \sim 0.18$, use $f = 0.16$

$pv_m = 2500$, water

$p(149.23) = 2500$

$p = 16.75\ psi$

$W = pDL = (16.75)(0.5)(0.75) = 6.28\ lb$

$F = fW = (0.16)(6.28) = 1.005\ lb$

$U_f = Fv_m = (1.005)(149.23) = 150\ ft-lb/\min$

(c) $v_m > 100\ fpm$

$f = 0.25$

$pv_m = 20{,}000$, intermittent

$p(149.23) = 20{,}000$

$p = 134\ psi$

$W = pDL = (134)(0.5)(0.75) = 50.25\ lb$

$F = fW = (0.25)(50.25) = 12.5625\ lb$

$U_f = Fv_m = (12.5625)(149.23) = 1875\ ft-lb/\min$

(d) $pv_m = 15{,}000$

$p(149.23) = 15{,}000$

$p = 100.5\ psi$

$W = pDL = (100.5)(0.5)(0.75) = 37.69\ lb$

assume $f = 0.20$

$F = fW = (0.20)(37.69) = 7.54\ lb$
$U_f = Fv_m = (7.54)(149.23) = 1125\ ft-lb/\min$

590. A bearing to support a load of 150 lb at 800 rpm is needed; $D = 1\ in.$; semi-lubricated. Decide upon a material and length of bearing, considering sintered metals, Zytel, Teflon, and graphite inserts.

Solution:
$$v_m = \frac{\pi Dn}{12} = \frac{\pi(1)(800)}{12} = 209.44\ fpm$$
assume, $L = D = 1\ in$
$$p = \frac{W}{DL} = \frac{150}{(1)(1)} = 150\ psi$$
$pv_m = (150)(209.44) = 31,416$
Use sintered metal, limit $pv_m = 50,000$

<u>THRUST BEARINGS</u>

592. A 4-in. shaft has on it an axial load of 8000 lb., taken by a collar thrust bearing made up of five collars, each with an outside diameter of 6 in. The shaft turns 150 rpm. Compute (a) the average bearing pressure, (b) the approximate work of friction.

Solution:

(a) $p = \frac{4W}{\pi D_o^2} = \frac{4(8000)}{\pi(6)^2} = 283\ psi$

(b) assume $f = 0.065$, average
$F = fW = (0.065)(8000) = 520\ lb$
$$v_m = \frac{\pi Dn}{12} = \frac{\pi(3)(150)}{12} = 117.81\ fpm$$
$U_f = nFv_m = (5)(520)(117.81) = 306,306\ ft-lb/\min$

593. A 4-in. shaft, turning at 175 rpm, is supported on a step bearing. The bearing area is annular, with a 4-in. outside diameter and a 3/4 –in. inside diameter. Take the allowable average bearing pressure as 180 psi. (a) What axial load may be supported? (b) What is the approximate work of friction?

Solution:
$$v_m = \frac{\pi Dn}{12}$$

$D = \frac{1}{2}(4 + 0.75) = 2.375\ in$

$v_m = \frac{\pi D n}{12} = \frac{\pi(2.375)(175)}{12} = 108.81\ fpm$

assume $f = 0.065$, average

(a) $p = \frac{4W}{\pi(D_o^2 - D_i^2)}$

$W = \frac{\pi}{4}\left[(4)^2 - \left(\frac{3}{4}\right)^2\right](180) = 2182\ lb$

(b) $U_f = fWv_m = (0.065)(2182)(108.81) = 15{,}433\ ft-lb/\min$

- end -

601. The radial reaction on a bearing is 1500 lb.; it also carries a thrust of 1000 lb.; shaft rotates 1500 rpm; outer ring stationary; smooth load, 8-hr./day service, say 15,000 hr. (a) Select a deep-groove ball bearing. (b) What is the rated 90 % life of the selected bearing? (c) For $b = 1.34$, compute the probability of the selected bearing surviving 15,000 hr.

Solution:

$F_x = 1500\ lb$

$F_y = 1000\ lb$

$B_{10} = (15{,}000)(60)(1500)(10^{-6}) = 1350\ mr$

$F_e = 0.56 C_r F_x + C_t F_z$

$C_r = 1.2$, outer ring stationary

assume $C_t = 1.8$

$F_e = 0.56(1.2)(1500) + (1.8)(1000) = 2808\ lb$

$$F_r = \left(\frac{B_{10}}{B_r}\right)^{\frac{1}{3}} F_e = (1350)^{\frac{1}{3}}(2808) = 31{,}034\ lb$$

(a) Table 12.3

use 321, $F_r = 31{,}800\ lb$

$F_s = 32{,}200\ lb$

To check:

$$\frac{F_z}{F_s} = \frac{1000}{32{,}000} = 0.03125$$

Table 12.2, $C_t = 1.96$, $Q = 0.2246$

$$\frac{F_z}{C_r F_x} = \frac{1000}{(1.2)(1500)} = 0.556 > Q$$

$F_e = 0.56 C_r F_x + C_t F_z$

$F_e = 0.56(1.2)(1500) + (1.96)(1000) = 2968\ lb$

$$F_r = \left(\frac{B_{10}}{B_r}\right)^{\frac{1}{3}} F_e = (1350)^{\frac{1}{3}}(2968) = 32{,}803\ lb$$

3.2 % higher than 31,800 lb. Safe.

Therefore use Bearing 321, Deep-Groove Ball Bearing.

(b) $F_r = 31{,}800\ lb$

$F_e = 2968\ lb$

$$31{,}800 = \left(\frac{B_{10}}{1\ mr}\right)^{\frac{1}{3}}(2968)$$

$B_{10} = 1230\ mr$

$B_{10} = (HR)(60)(1500)(10^{-6}) = 1230$

$HR \approx 13{,}700\ hr$