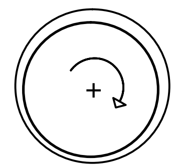
Plain Bearings or Sliding Bearings

A “bearing” is a contacting surface through which a load is applied.
When there is relative motion between the surfaces, you have a sliding bearing, and you need to deal with friction and wear.
This calls for some kind of lubrication.

- We are going to cover how to design a system of materials, geometry, and lubrication to provide a low-friction, durable, plain bearing.

Configuration Types:

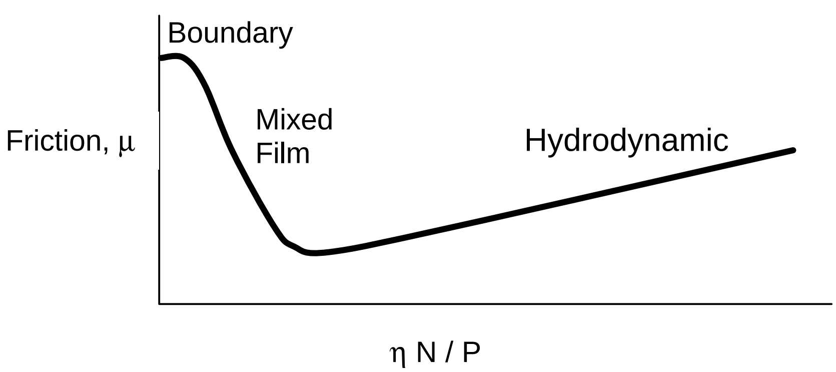


1. Journal / Sleeve — Partial (freight cars)
   Full (360°)
2. Thrust

Lubrication Types: (Sec. 8.4)

1. Hydrodynamic — Total bearing carried on “thick” film of lubrication. Depends on relative motion between surfaces. > Hydrostatic = Pressure fed; doesn’t need relative motion.
2. Elastohydrodynamic — Rolling contact, like gear tooth contact & rolling element bearings. Hamrock talks about Hard (metals & ceramics) and Soft (rubber & polymers)
3. Boundary / Mixed Film — Extensive direct surface contact
4. Solid film — Usually high temperature: Graphite or molydisulfide

Some of these modes are shown in a Stribeck curve:

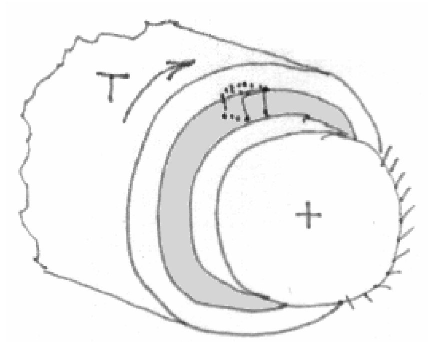


$$\eta N / P = \frac{\text{Viscosity * Rev/Sec}}{\text{Brg Unit Load (psi)}}$$

“High” loads, “low” speeds, and low viscosities will not permit hydrodynamic operation.

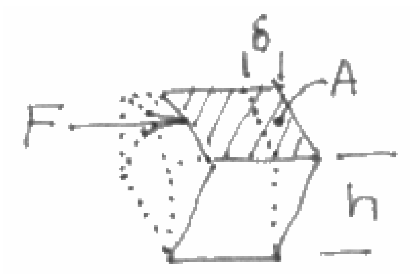
<u>VISCOSITY</u>

An Analogy



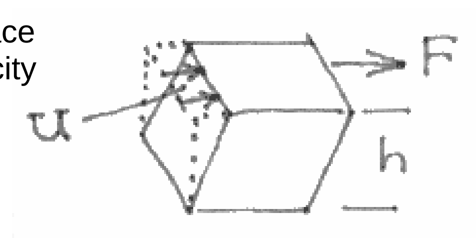
A shaft and collar with material in the gap, at equilibrium.

A. Gap material is
Solid Elastic Element



B. Gap material is
Viscous Fluid Element

Surface
Velocity



$$d = \frac{Fh}{AG}$$

G = Shear Modulus

$$U = \frac{Fh}{Ah}$$

h = Absolute Viscosity, aka Dynamic Viscosity

$$h = \frac{Fh}{AU} \quad \frac{LB.\,IN.}{IN^2\ IN/S} = \frac{LB.\,SEC}{IN^2} = reyns$$

$$SI:\ \frac{Newton \cdot Sec}{m^2} = Pascal \cdot Sec$$

1 reyn = 6890 PaS (same as Pascal to PSI conversion)

Units are usually microreyns and milliPaS.
Previous metric unit was Poise or centipoises, cP, where 1 cP = 1 mPaS

1 μreyn = 6.89 cP

VISCOMETER

Viscosity is commonly measured with a Saybolt Universal Viscometer [SUV]:
It measures the time for 60ml (~1/4 cup) of lubricant at constant temperature to run through a tube 17.6 mm Ø (0.7”) x 12.25 mm (0.5”) long.
This is

$$\text{“Kinematic Viscosity”} = \frac{\text{Absolute Viscosity}}{\text{Mass Density}} = h_k$$

Units are length^2 / time

Usually cm^2 / second = Stoke, St

Or mm^2 / second = Centistoke, cSt

To get Absolute Viscosity from SUV time (= Saybolt Universal Seconds, SUS)

$$h[mreyns] = 0.145(0.22S - \frac{180}{S})\,r$$

0.145 = 1000 / 6896

Where S = Seconds (SUS from SUV)

ρ = Mass density in grams/cm^3

= specific gravity (ratio of its density to density of water = 1 gm/cm^3)

Example: Petroleum Oil at 60°F, ρ = 0.89 gm/cm^3

- Generally, viscosity decreases with increasing temperature. See Fig. 8.16 on page 347.
  - Note that viscosity of gases increases with temperature.
- Newtonian fluids have linear velocity gradients.
  - Some non-Newtonian fluids are tomato paste, chocolate, peanut butter, and shampoo.
- Liquid viscosity increases slightly with pressure
  - Eqn. 8.32

$$h = h_0 e^{x\,p}$$

  - Since from Table 8.5, $x \approx$ 1 x 10^{-8}, at 100 psi = 689,500 Pa,

$$e^{(1\times10^{-8})(0.689\times10^{6})} = 1.007$$

so the viscosity only goes up 0.7%.

Petroff's Equation   § 12.4.1

- Concentric journal bearing
- Apply small radial load, W_r
- Produces "average" or projected film pressure:

$$P = \frac{W_r}{2rw_t} \quad or \quad W_r = 2 P r w_t$$

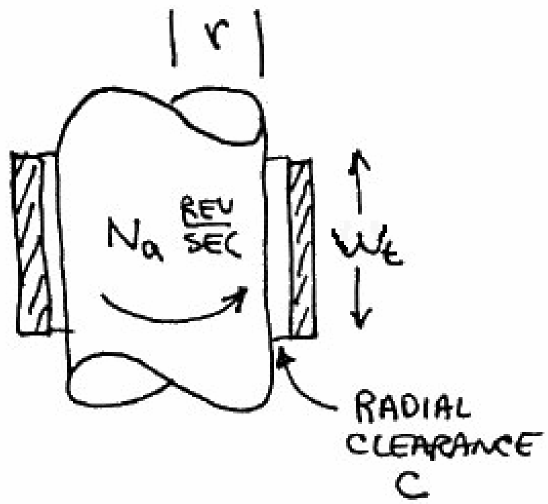


Figure 12.18

The coefficient of friction is

$$m = \frac{4p^2 h_0 r^2 w_t N_a}{cW_r} \qquad \text{Eqn. 12.78}$$

$$m = \frac{4p^2 h_0 r^2 w_t N_a}{2cPrw_t} = 2p^2 \left(\frac{h_0 N_a}{P}\right)\left(\frac{r}{c}\right)$$

$\left(\frac{h_0 N_a}{P}\right)$: Stribeck Curve

$\left(\frac{r}{c}\right)$: Clearance Ratio

> Petroff's Equations are ONLY for an unloaded journal bearing, so **DO NOT** use these equations for actual, side-loaded journal bearings.

Real Journal with Eccentricity

Bearing characteristics are a function of the dimensionless Sommerfeld number, B_j, also known as the Bearing Number, or Bearing Characteristic Number.

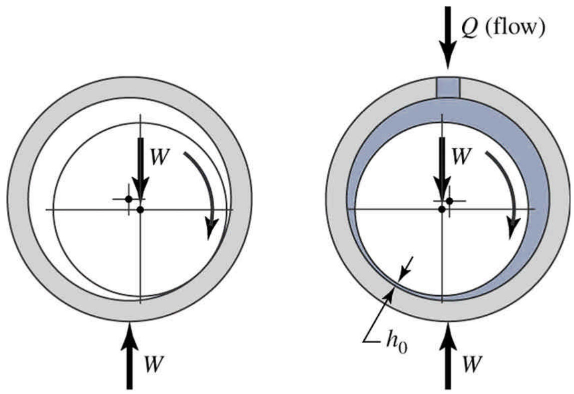
$$B_j = \frac{\eta_0 \omega_b r_b w_t}{\pi W_r}\left(\frac{r_b}{c}\right)^2 \qquad \text{Eqn. 12.84}$$

$$also = \frac{\eta_0 N_a}{P}\left(\frac{r_b}{c}\right)^2$$

Where h_0 is in reyns,
N_a is in revs/sec, and
P is in psi.

HYDRODYNAMIC THEORY

Hydrodynamic theory was driven by railroad car bearing research by Beauchamp Tower in 1883, Reynolds in 1886, and Sommerfeld in 1904.



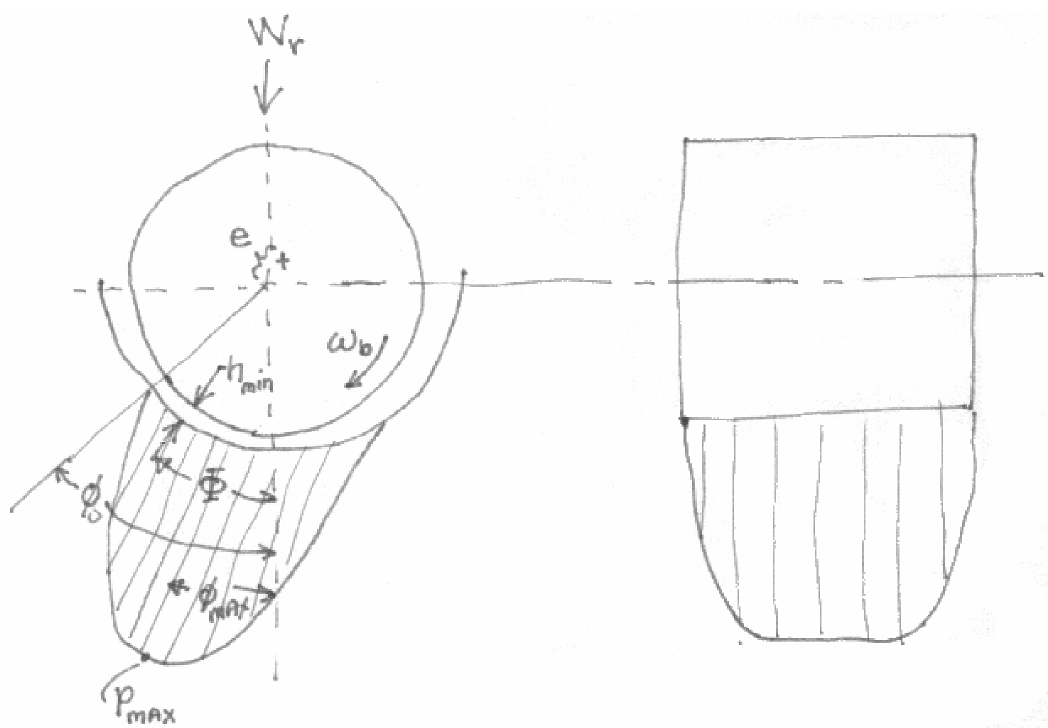
DRY LUBRICATED

Hydrodynamics requires:

1. Relative motion of surfaces
2. “Wedging action” given by shaft eccentricity (NOT like Petroff)
3. Suitable fluid

It is basically like waterskiing.

In the 1950’s, two guys at Westinghouse [Raimondi & Boyd] used a computer to solve Reynold’s equations for a series of bearing parameters versus the Sommerfeld number.



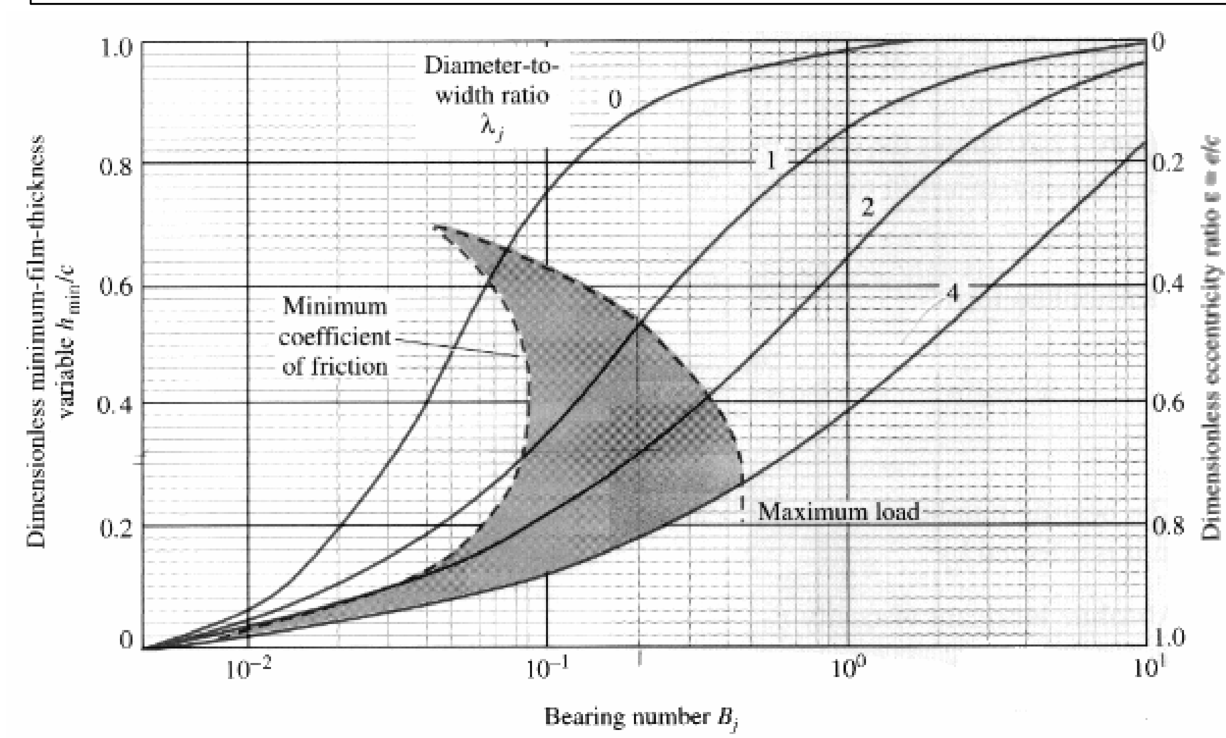
Raimondi & Boyd Charts (Full Journal)

- All are versus Sommerfeld number
- All have journal $\frac{Diameter}{Length} = \frac{2r_b}{w_t}$ as parameter (l_j)

Here is a summary of the charts:

1	Fig. 12.20	Minimum film thickness variable	$h_{\min}/c$
2	Fig. 12.21	Angle of minimum film thickness	F
3	Fig. 12.22	Coefficient of friction variable	$r_b m/c$
4	Fig. 12.23 *	Volumetric flow rate variable	$2pq/(r_b\, c w_t w_b)$ *
5	Fig. 12.24	Side-leakage flow ratio	q_s/q
6	Fig. 12.25 *	Maximum film pressure	$W_r/(2r_b w_t p_{\max})$ *
7	Fig. 12.26	Angles of max. and terminating pressures	$j_{\max}$, j_0

* Note that Hamrock has left out the " / " in the vertical axis labels for these two charts.



<u>Selectable</u> Design Variables:

1. Viscosity, h_0
2. Radial Load, W_r
3. Speed, w_b
4. Bearing Dimensions, r_b, c, w_t

Dependent Variables:

1. Coefficient of friction, μ
2. Temperature rise, Δt_m (Eq. 12.90)
3. Oil flow, q
4. Minimum film thickness, h_{min}

Adjust these
to tune these

Heat Management

Temperature rise in journal bearings, Eqn. 12.90

SI:
$$\Delta t_m = \frac{8.3 W_r^* (r_b/c)m}{Q(1 - 0.5 q_s / q)} \,{}^\circ C \qquad W_r^* = \frac{W_r}{2 r_b w_t} \quad MPa$$

English
$$\Delta t_m = \frac{0.103 W_r^{**} (r_b/c)m}{Q(1 - 0.5 q_s / q)} \,{}^\circ F \qquad W_r^{**} = \frac{W_r}{2 r_b w_t} \quad psi$$

Notice that $(r_b/c)m$ & q_s / q are directly readable off of charts.

> **NOTE**: The Sommerfeld number is most correctly calculated using the mean bearing temperature, which is the inlet lubricant temperature plus half of the temperature rise. This is important because of the strong sensitivity of the viscosity to temperature.

Power Loss = Torque x Angular Velocity

Power = ($\mu W_r r_b$) w_b in Watts, when W_r is in newtons and r_b in meters.

Note again that this is different from the Petroff equation for power (Eqn. 12.80) because this is for a real journal with a real side load and therefore not running concentrically!

Clearance Management

- Design is very sensitive to clearance
- Clearances are often very small
- They are hard to control accurately
- Wear will increase clearance
- See Fig. 12.27

Materials

- Bronze and Babbitt (tin alloy), good to 300°F
- Plastics
  - Most good to 200°F
  - PTFE (Teflon) good to 500°F
- Carbon graphite, good to 750°F
- Favorable characteristics:
  - Embeddability (Safely holds contaminates)
  - Corrosion resistance
  - Resistance to seizing (See Table 8.7)

EXAMPLE: Full Journal Bearing

Given:
D = 32 mm Journal diameter
L = 32 mm = w_t Journal length
W_r = 1800 N Side Load
N = 1150 RPM Speed
C = 4 x 10^{-5} m Radial Clearance
Oil = SAE 30 @ 40°C Inlet Temperature

Find:
- Minimum oil film thickness
- Total oil flow
- Max film pressure
- Temperature rise

Fig. 8.17a: h_0 = 0.075 N sec/m^2
r_b = 0.016 m
r_b/c = 16/0.04 = 400
N_a = 1150/60 = 19.2 rev/sec
w_b = 2π N_a = 120.4 Rad/sec

$$B_j = \frac{\eta_0 \omega_b r_b w_t}{\pi W_r}\left(\frac{r_b}{c}\right)^2 = \frac{(0.075)(120.4)(0.016)(0.032)}{\pi(1800)}(400)^2$$

B_j = 0.131 (dimensionless)
l_j = D/L = 1.0

	Figure		
1	12.20	$h_{\min}/c$ = 0.4	$h_{\min} = (0.4)(4\times10^{-5}) = 1.6\times10^{-5}\ m = 0.00063\ in.$
2	12.23	Q= $2pq/(r_b c w_t w_b)$ = 4.4	$q = \dfrac{4.4(.016)(4\times10^{-5})(.032)(120.4)}{2p} = 1.73\times10^{-6} m^3/s = 0.106\ in^3/s$
3	12.25	$P_{\max} = W_r/(2r_b w_t p_{\max})$ = 0.42	$p_{\max} = \dfrac{1800}{(0.42)(2)(.016)(.032)} = 4.185\ MPa = 607.4\ psi$
4	12.24	q_s/q = 0.66	
5	12.22	$r_b m/c$ = 4	
6		$W_r^* = \dfrac{W_r}{2r_b w_t} = \dfrac{1800}{(.032)(.032)} = 1.76\times10^6\ N/m^2 = 1.76\ MPa$	
7			$\Delta t_m = \dfrac{(8.3)(1.76)(4)}{(4.4)(1-0.66/2)} = 19.8°\,C \times \dfrac{9}{5} = 35.64°\,F$

So the mean lubricant temperature would be 40° + 19.8/2 = 49.9°C. This would make the viscosity 0.04, which would make B_j = 0.070, so we really should run through the calculations again, possibly several times…

Boundary Lubricated Bearings

P is projected load pressure = W_r / (2 r_b w_t) in MPa or ksi
V is surface speed at contact radius = 2 π r_b ω_b in m/sec or ft/min (NOT in/sec)

- PV is a measure of the bearing's ability to absorb energy without overheating.
- Design for about ½ of the rated PV value.

Operating Limits of Boundary-Lubricated Porous Metal Bearings [3]

	Static *P*		Dynamic *P*		*V*		*PV*	
Material	**MPa**	**(ksi)**	**MPa**	**(ksi)**	**m/s**	**(fpm)**	**MPa·m/s**	**(ksi·fpm)**
Bronze	55	(8)	14	(2)	6.1	(1200)	1.8	(50)
Lead–bronze	24	(3.5)	5.5	(0.8)	7.6	(1500)	2.1	(60)
Copper–iron	138	(20)	28	(4)	1.1	(225)	1.2	(35)
Hardenable copper–iron	345	(50)	55	(8)	0.2	(35)	2.6	(75)
Iron	69	(10)	21	(3)	2.0	(400)	1.0	(30)
Bronze–iron	72	(10.5)	17	(2.5)	4.1	(800)	1.2	(35)
Lead–iron	28	(4)	7	(1)	4.1	(800)	1.8	(50)
Aluminum	28	(4)	14	(2)	6.1	(1200)	1.8	(50)

Operating Limits of Boundary-Lubricated Nonmetallic Bearings [3]

	P		Temperature		*V*		*PV*	
Material	**MPa**	**(ksi)**	**°C**	**(°F)**	**m/s**	**(fpm)**	**MPa·m/s**	**(ksi·fpm)**
Phenolics	41	(6)	93	(200)	13	(2500)	0.53	(15)
Nylon	14	(2)	93	(200)	3.0	(600)	0.11	(3)
TFE	3.5	(0.5)	260	(500)	0.25	(50)	0.035	(1)
Filled TFE	17	(2.5)	260	(500)	5.1	(1000)	0.35	(10)
TFE fabric	414	(60)	260	(500)	0.76	(150)	0.88	(25)
Polycarbonate	7	(1)	104	(220)	5.1	(1000)	0.11	(3)
Acetal	14	(2)	93	(200)	3.0	(600)	0.11	(3)
Carbon (graphite)	4	(0.6)	400	(750)	13	(2500)	0.53	(15)
Rubber	0.35	(.05)	66	(150)	20	(4000)	—	—
Wood	14	(2)	71	(160)	10	(2000)	0.42	(12)

Ref. 3: "Mechanical Drives," *Machine Design Reference Issue*, Penton/IPC, Cleveland, 18June1981.