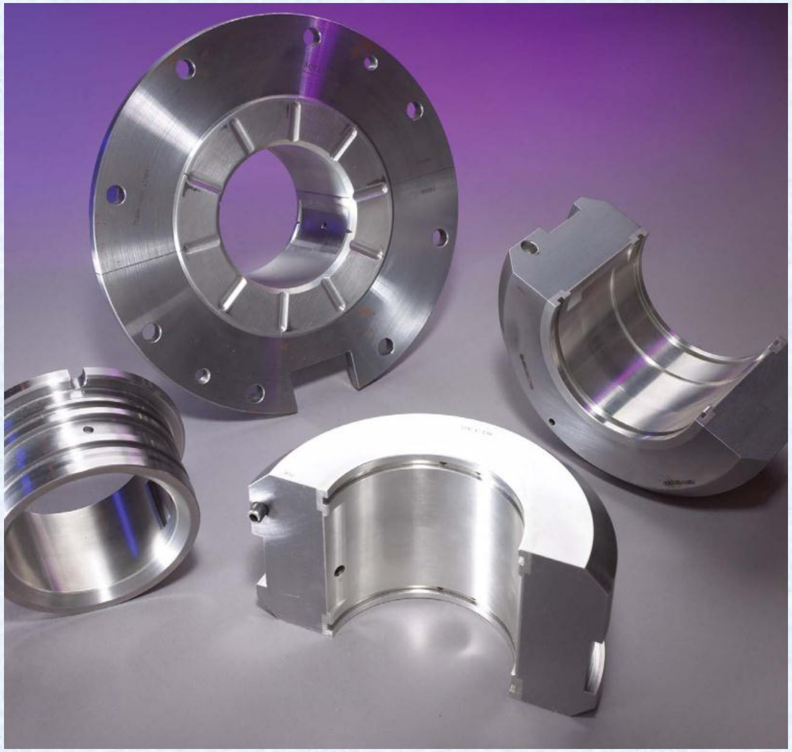
Bearings



Components of a hydrodynamic journal bearing. *Source:* Courtesy of the Kingsbury Co.

Getting wisdom is the most important thing you can do! And whatever else you do, get good judgment.

Proverbs 4:7

The Reynolds Equation

$$\underbrace{\frac{\partial}{\partial x}\left(\frac{\rho h^3}{12\eta}\frac{\partial p}{\partial x}\right)}_{\text{Poiseuille}} = \rho\left(\underbrace{w_a - w_b - u_a}_{\text{Squeeze}}\frac{\partial h}{\partial x}\right)$$

$$+ \underbrace{h\frac{\partial \rho}{\partial t}}_{\text{Local expansion}}$$

$$+ \overbrace{\frac{\partial}{\partial x}\underbrace{\left(\frac{\rho h\left(u_a + u_b\right)}{2}\right)}_{\text{Couette}}}$$

$$\underbrace{\frac{h\left(u_a + u_b\right)}{2}\frac{\partial \rho}{\partial x}}_{\text{Density wedge}} \qquad \frac{\rho h}{2}\underbrace{\frac{\partial}{\partial x}\left(u_a + u_b\right)}_{\text{Stretch}} \qquad \underbrace{\frac{\rho\left(u_a + u_b\right)}{2}\frac{\partial h}{\partial x}}_{\text{Physical wedge}}$$

Density Wedge Mechanism

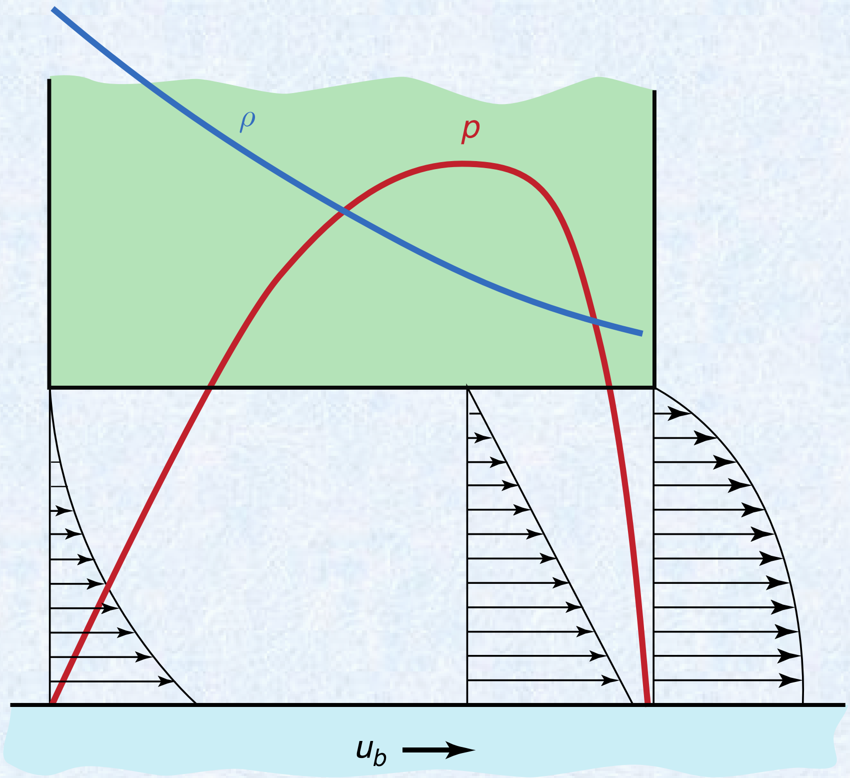


Figure 12.1: Density wedge mechanism.

Density wedge term:

$$\frac{h(u_a+u_b)}{2}\frac{\partial \rho}{\partial x}$$

Note: Not usually important.

Stretch Mechanism

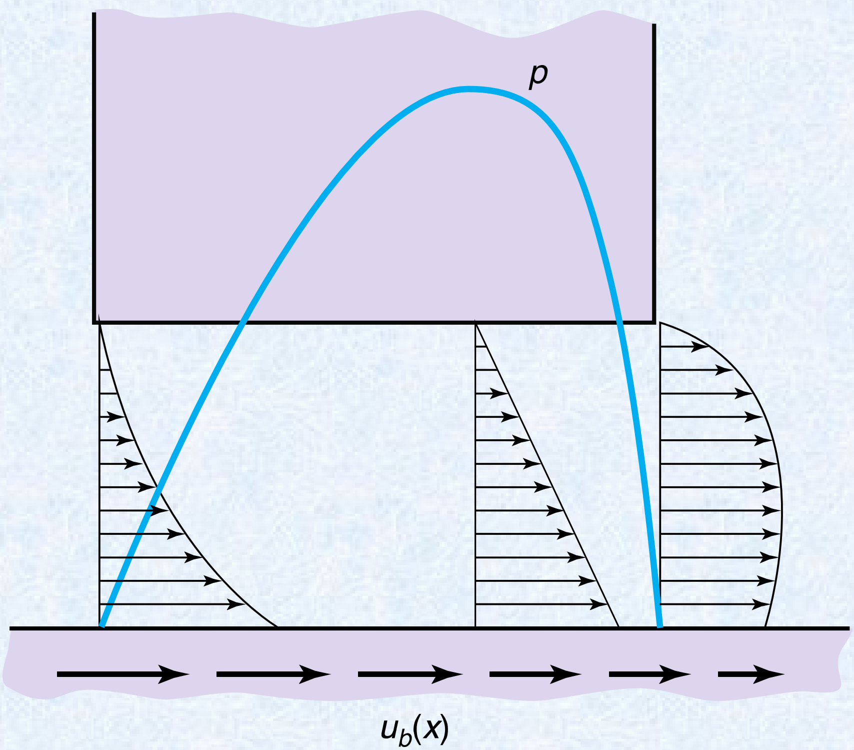


Figure 12.2: Stretch mechanism.

Stretch term:

$$\frac{\rho h}{2} \frac{\partial (u_a + u_b)}{\partial x}$$

Note: Not usually important in bearings, can be important in lubricant film breakdown in manufacturing.

Physical Wedge

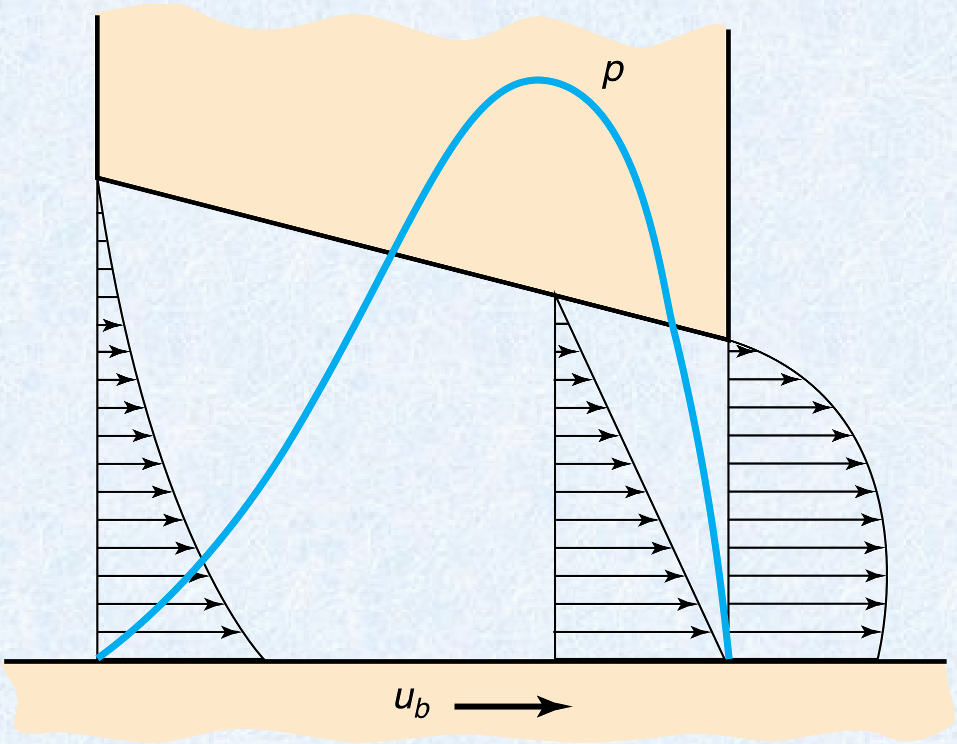


Figure 12.3: Physical wedge mechanism.

Physical wedge term:

$$\frac{\rho(u_a + u_b)}{2}\frac{\partial h}{\partial x}$$

Note: Very important mechanism for bearings, gears, cams, etc.

Normal Squeeze

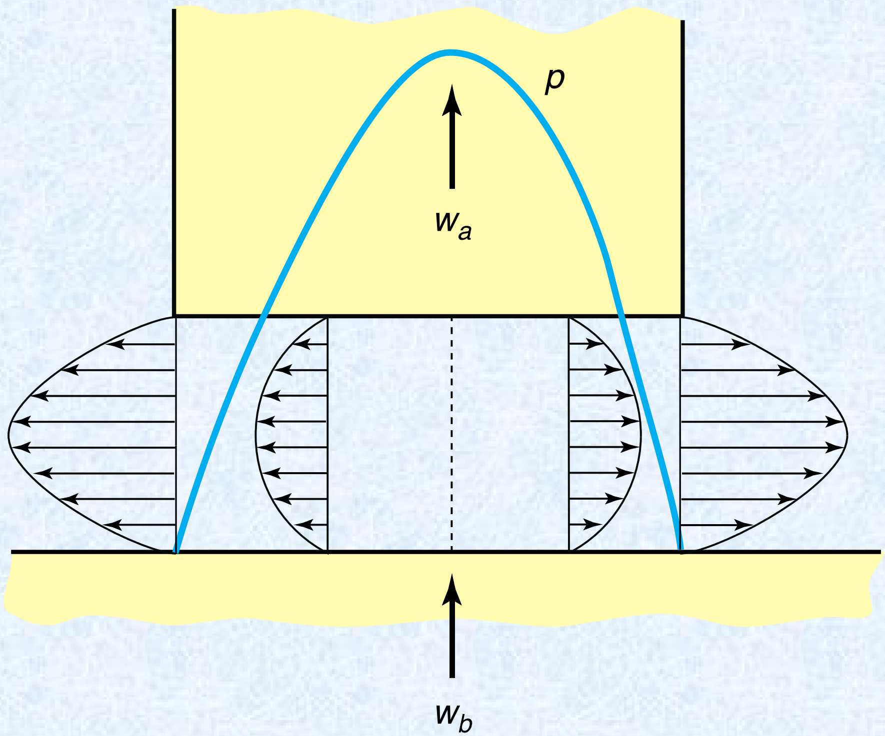


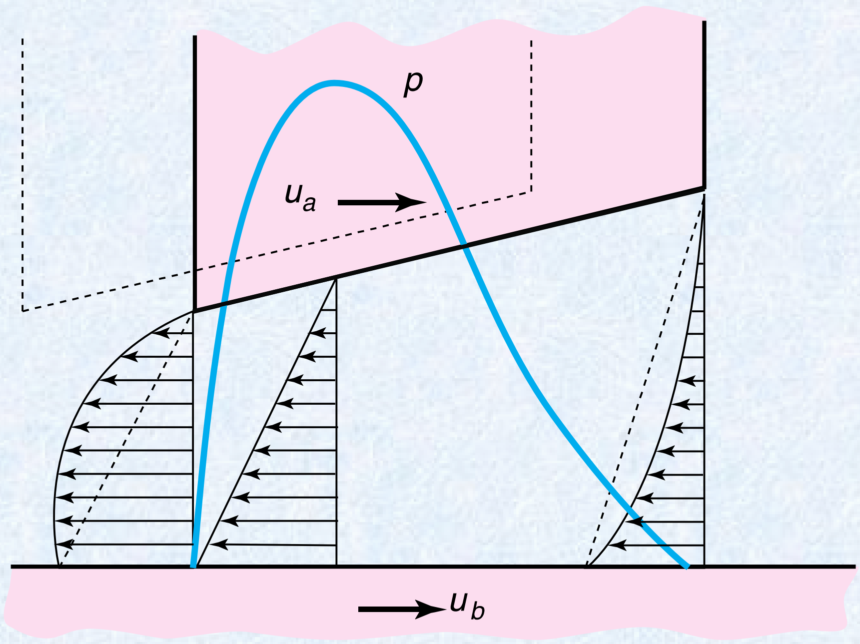
Figure 12.4: Normal squeeze mechanism.

Normal squeeze term:

$$\rho(w_a - w_b)$$

Note: Important for dynamically loaded bearings.

Translation Squeeze



Translation squeeze term:

$$-\rho u_a \frac{\partial h}{\partial x}$$

Note: Importance depends on interpretation; can be considered either an important mechanism or a correction for physical wedge.

Figure 12.5: Translation squeeze mechanism. Note that the velocity u_b is negative as shown, so that the pressure developed is positive.

Local Expansion

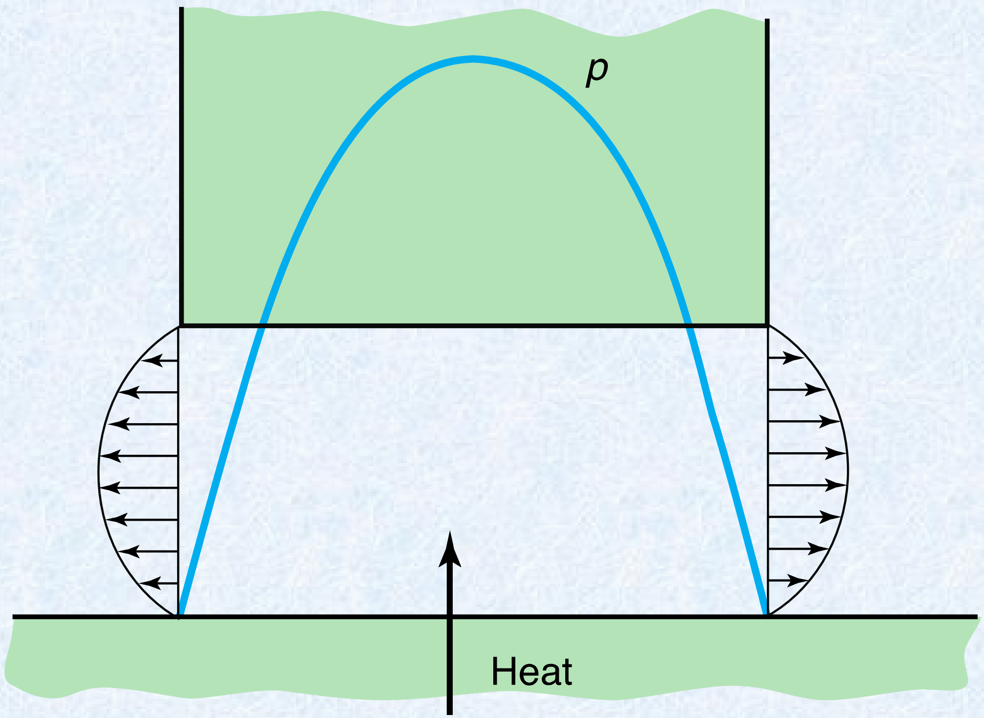


Figure 12.6: Local expansion mechanism.

Local expansion term:

$$h\frac{\partial \rho}{\partial t}$$

Note: Not usually important, but has some applications, such as lubrication by an evaporating liquid.

Reynolds Equation – Reduced Forms

Using only most important terms:

$$\frac{\partial}{\partial x}\left(\frac{\rho h^3}{\eta}\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial y}\left(\frac{\rho h^3}{\eta}\frac{\partial p}{\partial y}\right)=12\bar{u}\frac{\partial(\rho h)}{\partial x}$$

Where

$$\bar{u}=\frac{u_a+u_b}{2}=\text{Constant}$$

If side flow can be neglected (that is, bearings are wide):

$$\frac{\partial}{\partial x}\left(h^3\frac{\partial p}{\partial x}\right)=6\eta_o u_b\frac{\partial h}{\partial x}$$

Flow in Bearings

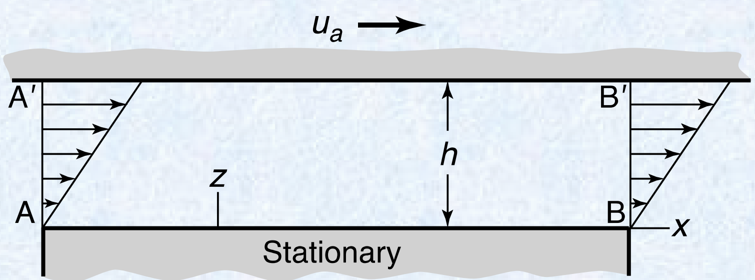


Figure 12.7: Velocity profiles in a parallel-surface slider bearing.

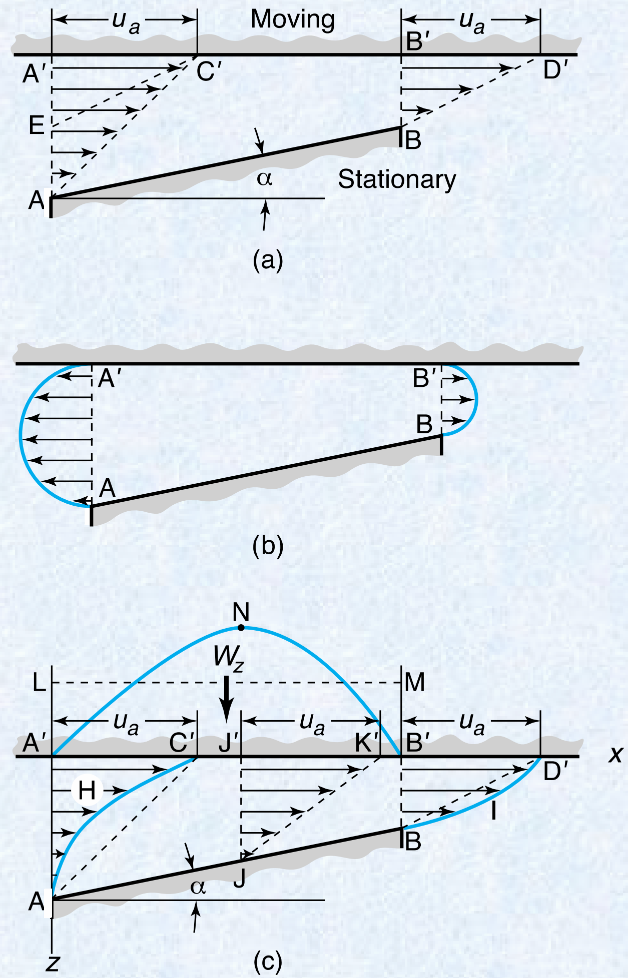


Figure 12.8: Flow within a fixed-incline slider bearing. (a) Couette flow; (b) Poiseuille flow; (c) resulting velocity profile.

Thrust Slider Bearings

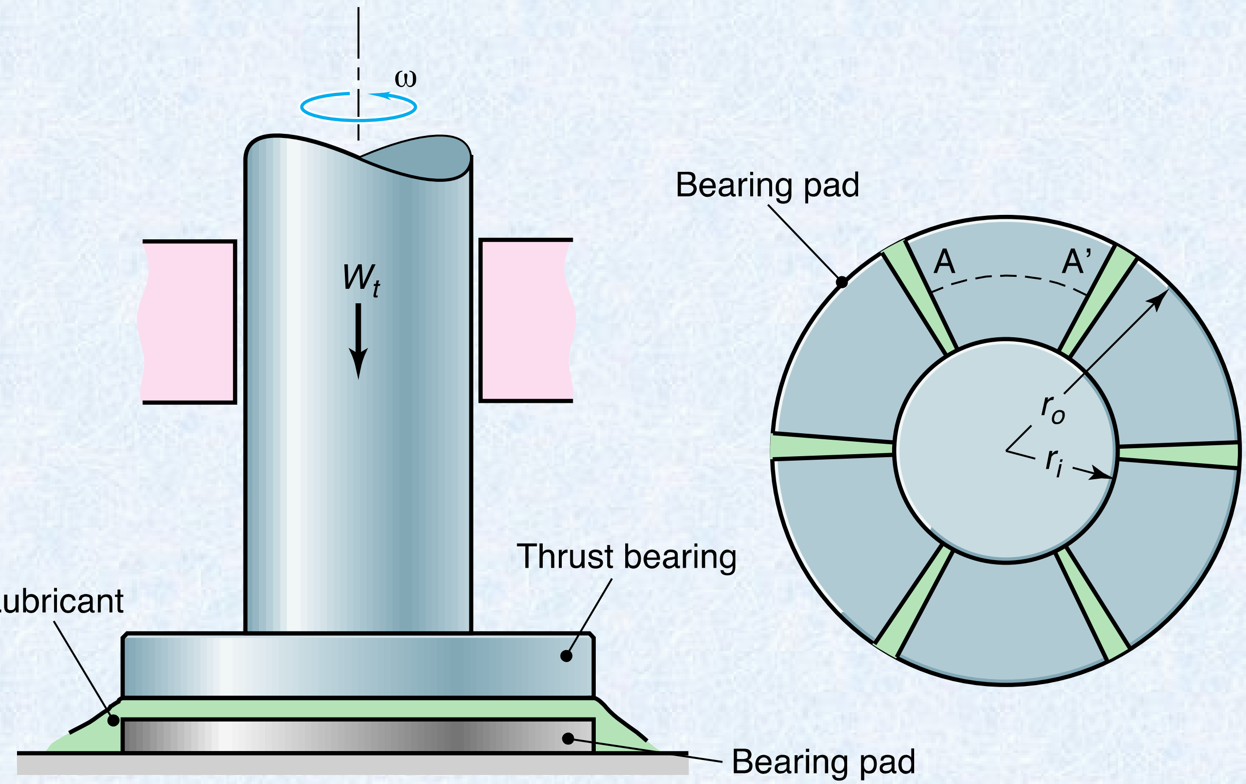


Figure 12.9: Thrust slider bearing geometry.

Thrust Slider Bearing Forces

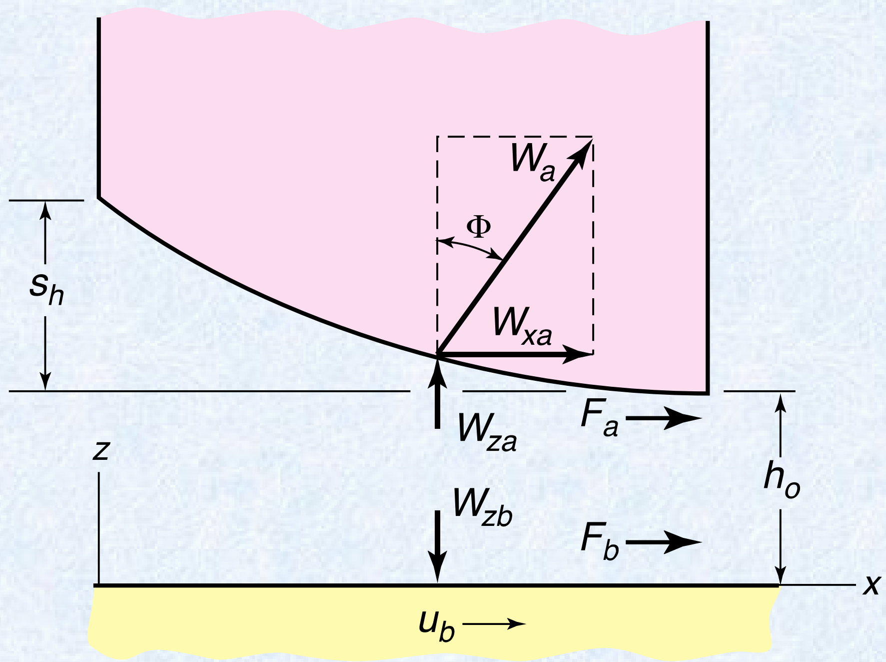


Figure 12.10: Force components and oil film geometry in a hydrodynamically lubricated thrust slider bearing.

Side View

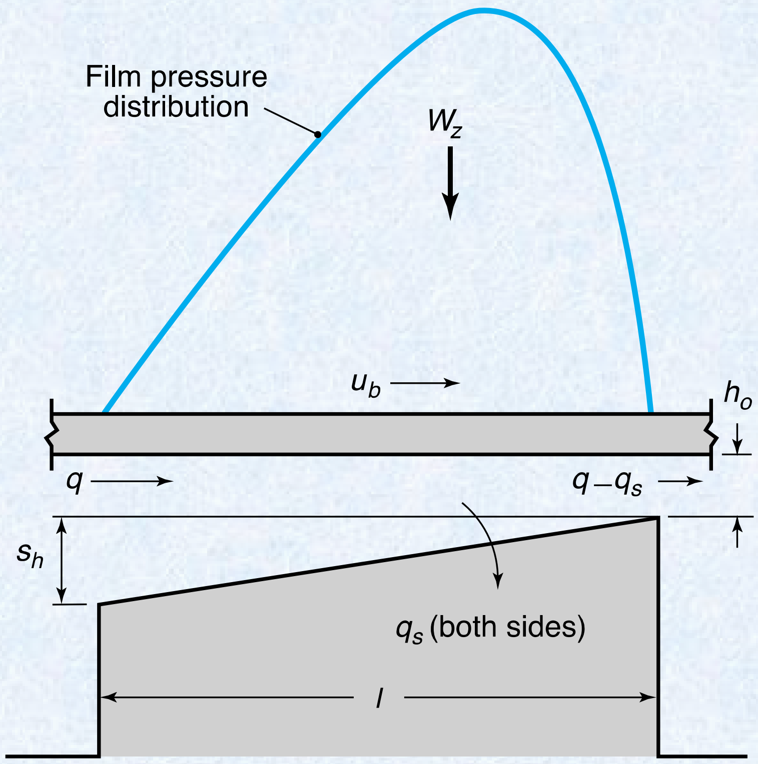


Figure 12.11: Side view of fixed-incline slider bearing.

Multiple Pad Slider Bearing

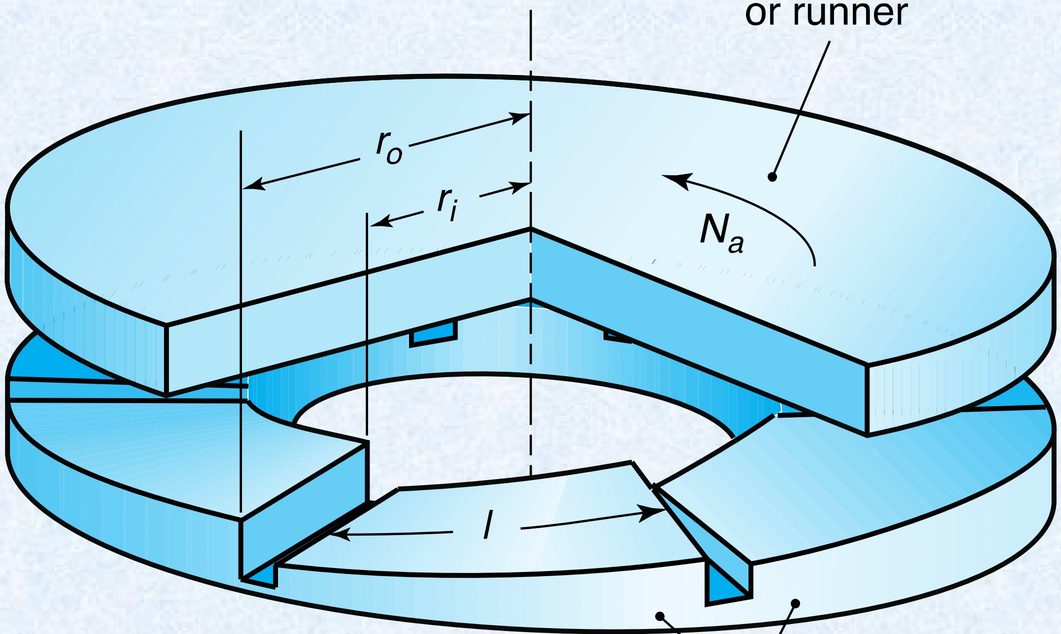


Figure 12.12: Configuration of multiple fixed-incline thrust slider bearing.

Design Procedure 12.1: Fixed-Incline Thrust Bearings

1. Choose a pad length-to-width ratio; a square pad ($\lambda = 1$) is generally considered to give good performance. If it is known whether maximum load or minimum power loss is more important in a particular application, the outlet film thickness ratio H_o can be determined from Fig. 12.13. If neither condition is a constraint, a value between these extremes is generally advisable.
2. Once λ and H_o are known, Fig. 12.14 can be used to obtain the bearing number B_t.
3. From Fig. 12.15, determine the temperature rise due to viscous shear-induced heating for a given λ and B_t. The volumetric specific heat $C_s = \rho C_p$, which is the dimensionless temperature rise parameter, is relatively constant for mineral oils and is approximately 1.36 x 10^6 N/(m^2-°C).
4. Determine the lubricant temperature. The mean temperature can be expressed as

$$t_m = t_{mi} + \frac{\Delta t_m}{2}$$

1. Where t_{mi} is the inlet temperature and is usually known beforehand. Once the mean temperature, t_m, is known, it can be used in Fig. 8.13 to determine the viscosity of SAE oils, or Fig. 8.13 or Table 8.8 can be used.

Design Procedure 12.1 (concluded)

5. Use Eqs. (12.32) and (12.66) to determine the outlet (minimum) film thickness, h_o, as

$$h_o = H_o l \sqrt{\frac{\eta_o u_b w_t}{W_z B_t}}$$

6. Once the outlet film thickness is known, the shoulder height, s_h, can be directly obtained from $s_h = h_o/H_o$. If the outlet film thickness is specified and either the velocity, u_b, or the normal applied load, W_z, is not known, Eq. (12.69) can be rewritten to establish u_b or W_z.
7. Check Table 12.1 to see if the outlet (minimum) film thickness is sufficient for the surface finish as manufactured. If the result from Eq.~(12.69) is greater than the recommendations in Table 12.1, go to step 7. Otherwise, consider one or both of the following steps:
   a. Increase the bearing speed.
   b. Decrease the load, improve the surface finish, choose a more viscous lubricant, or reduce the inlet temperature. Upon making this change, return to step 3.
8. Evaluate the remaining performance parameters. Once an adequate minimum film thickness and a proper lubricant temperature have been determined, the performance parameters can be evaluated. Specifically, the power loss, the coefficient of friction, and the total and side flows can be determined from Fig. 12.16.

Film Thickness in Thrust Sliders

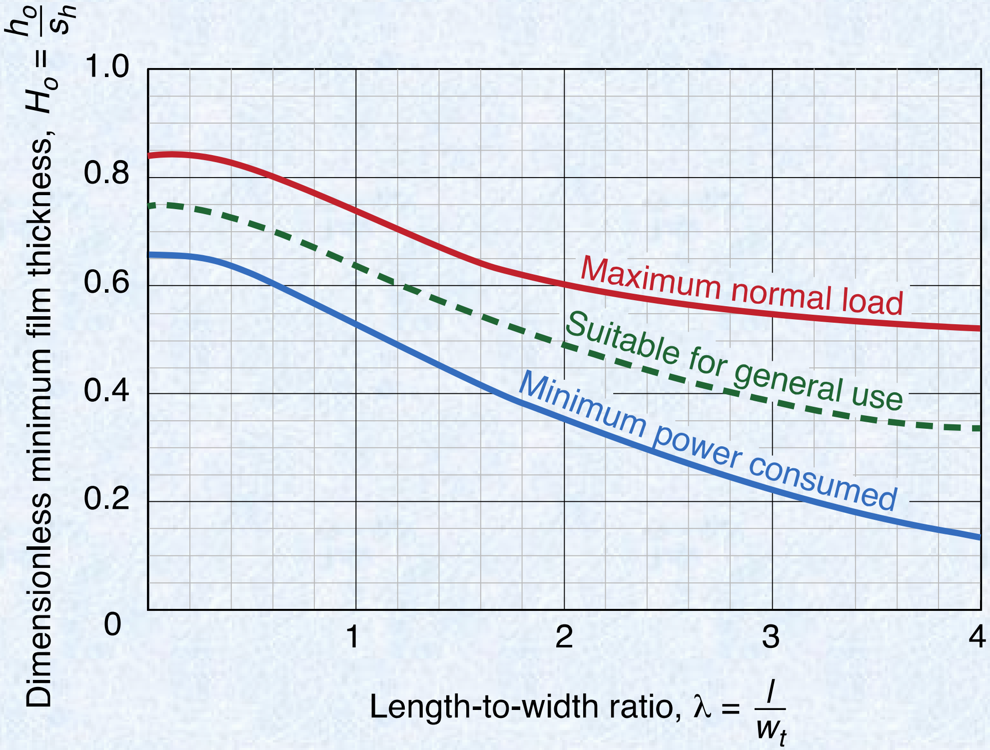


Figure 12.13: Chart for determining minimum film thickness corresponding to maximum load or minimum power-loss for various pad proportions in fixed-incline bearings. *Source:* From Raimondi and Boyd [1955].

Film Thickness – Fixed Incline Slider Bearings

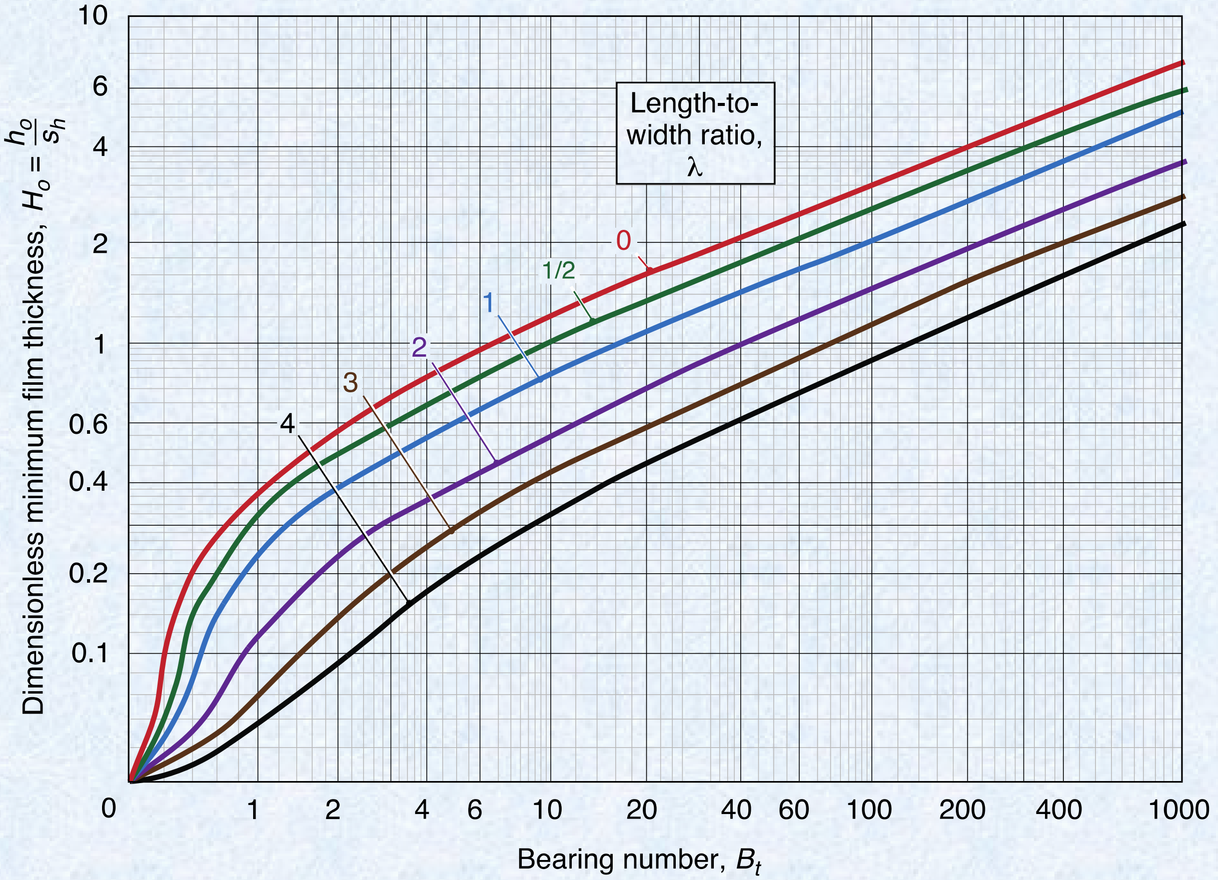


Figure 12.14: Chart for determining minimum film thickness for fixed-incline thrust bearings. *Source:* From Raimondi and Boyd [1955].

Temperature Rise – Fixed Incline Slider Bearings

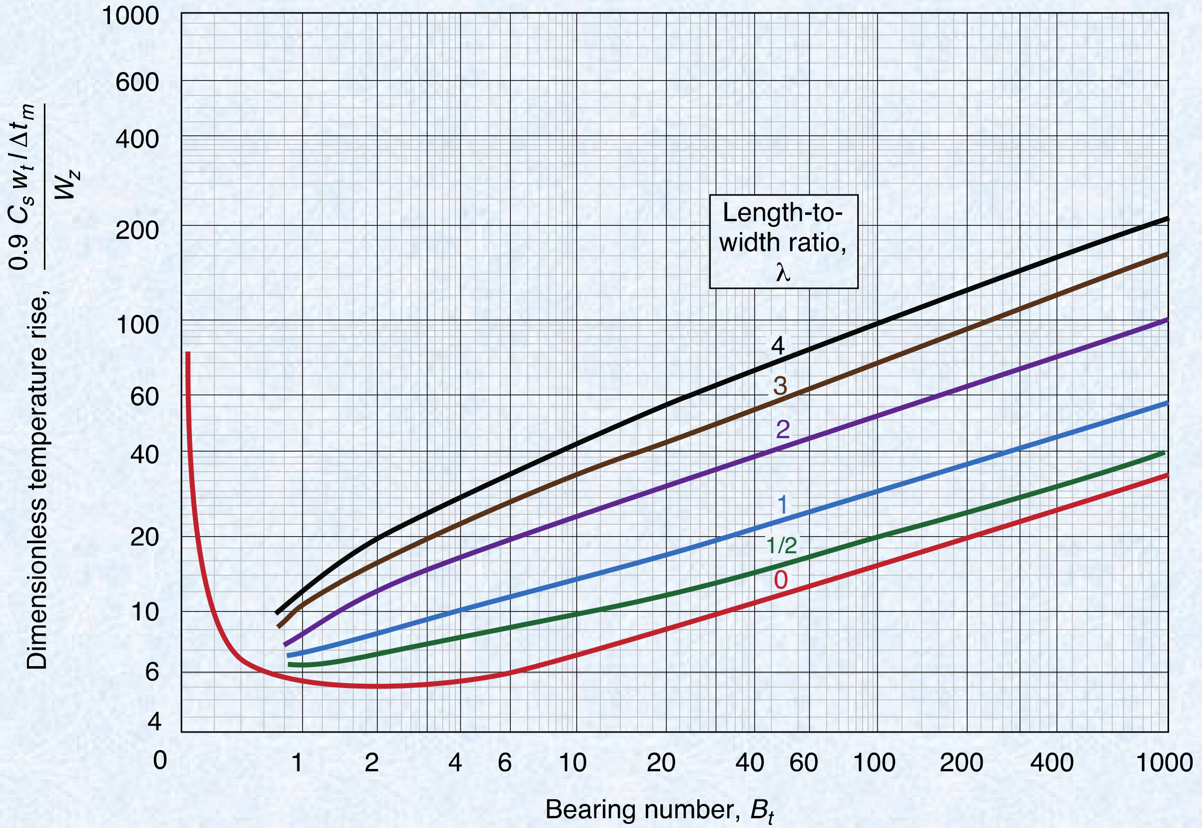


Figure 12.15: Chart for determining dimensionless temperature rise due to viscous shear heating of lubricant in fixed-incline thrust bearings. *Source:* From Raimondi and Boyd [1955].

Friction Coefficient – Fixed Incline Slider Bearings

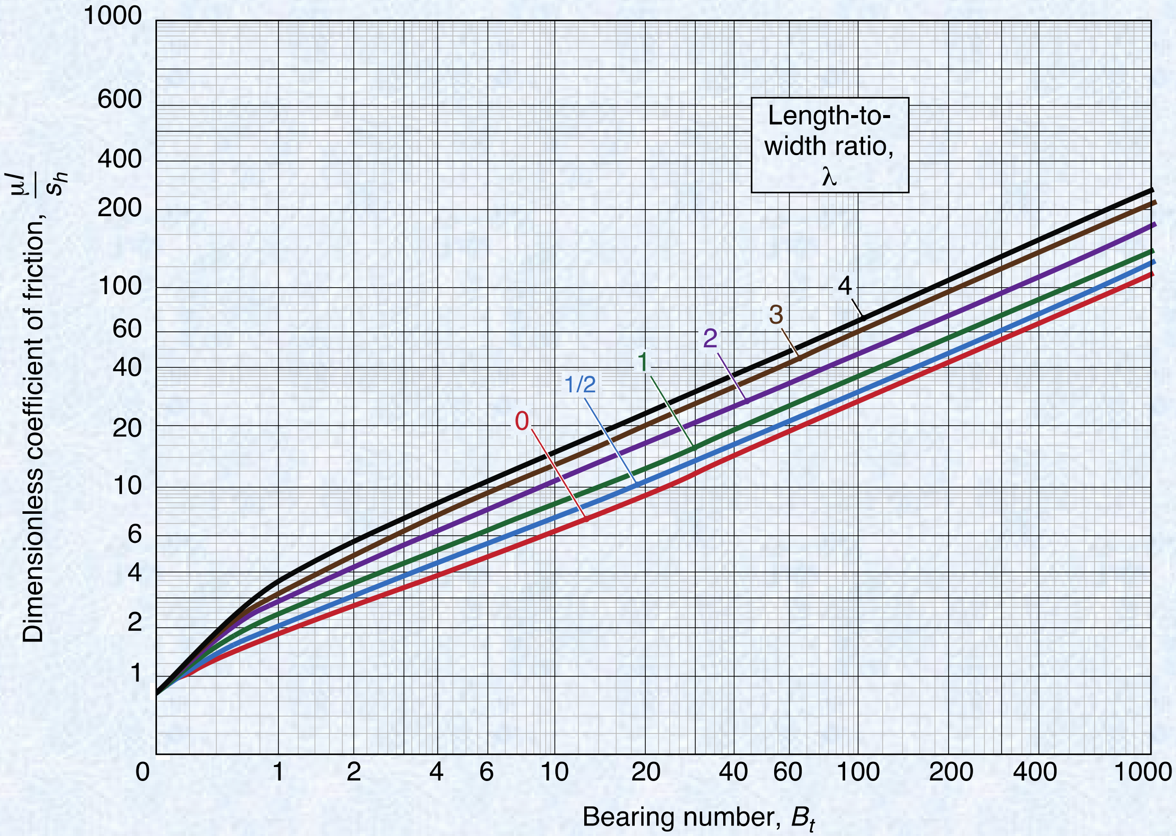


Figure 12.16: Chart for determining performance parameters of fixed-incline thrust bearings. (a) Friction coefficient;

Power Loss – Fixed Incline Slider Bearings

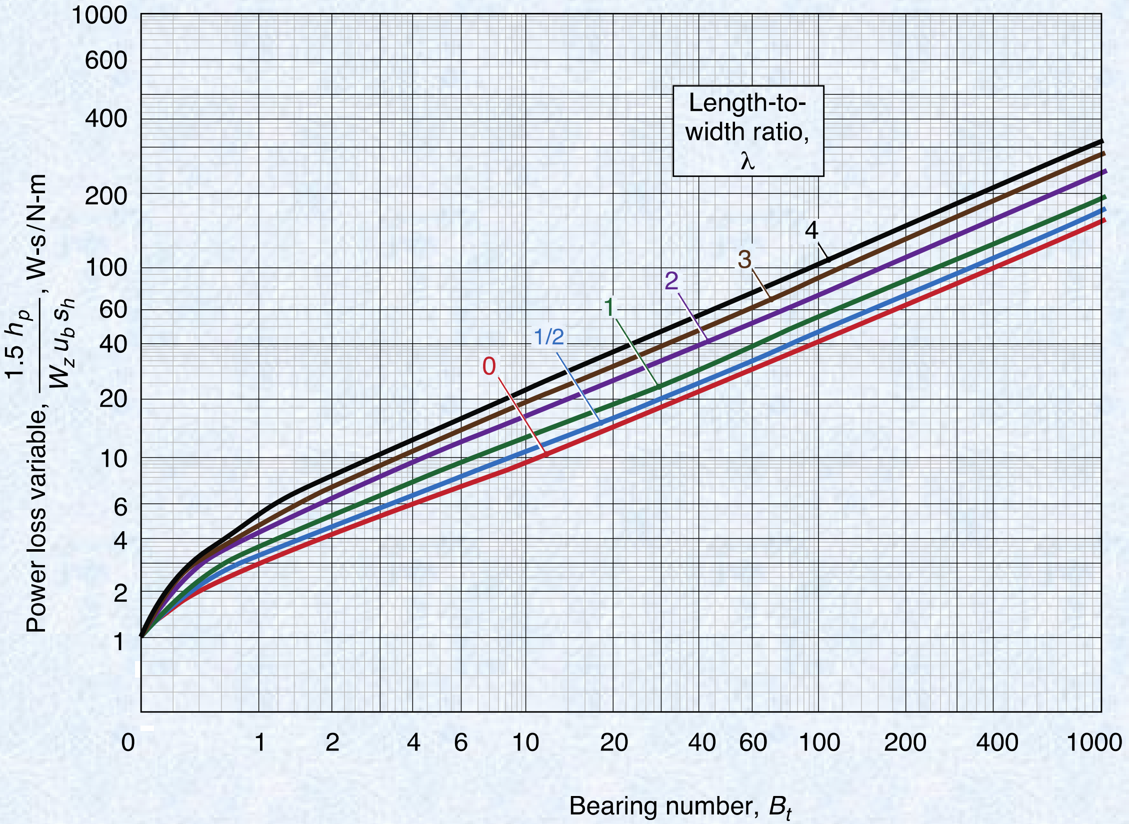


Figure 12.16: Chart for determining performance parameters of fixed-incline thrust bearings. (b) Power loss;

Lubricant Flow – Fixed Incline Slider Bearings

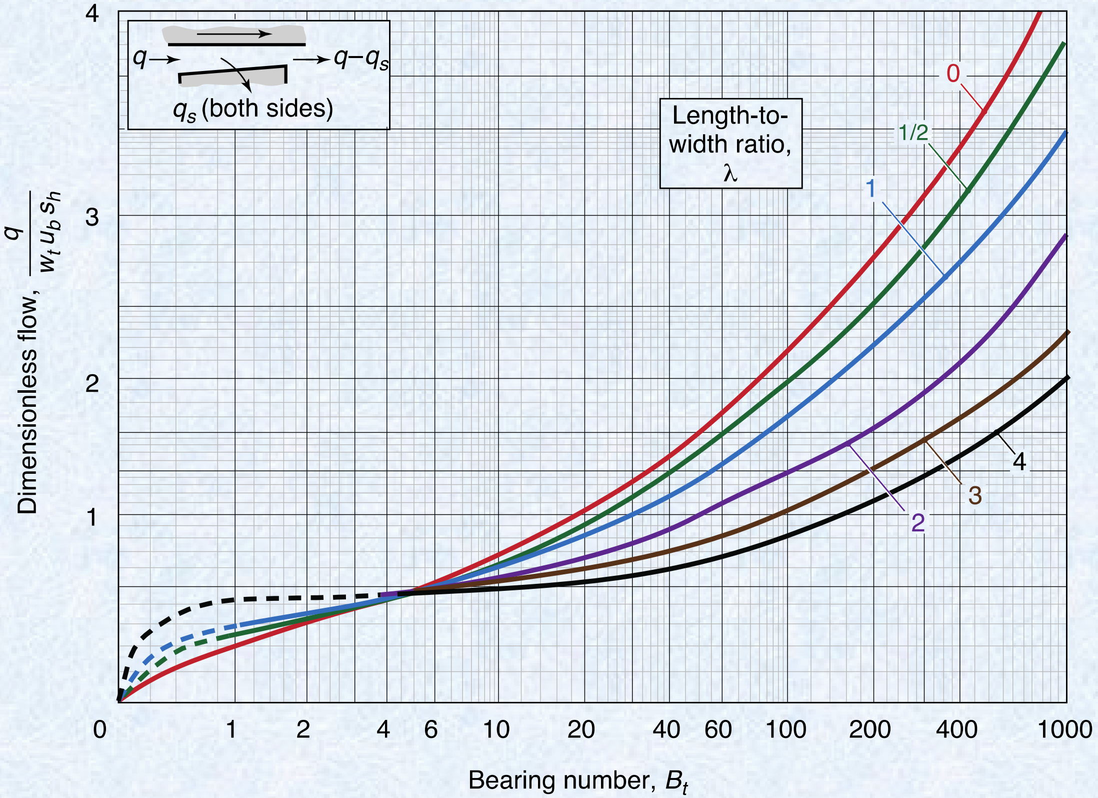


Figure 12.16: Chart for determining performance parameters of fixed-incline thrust bearings. (a) Lubricant flow;

Side Flow – Fixed Incline Slider Bearings

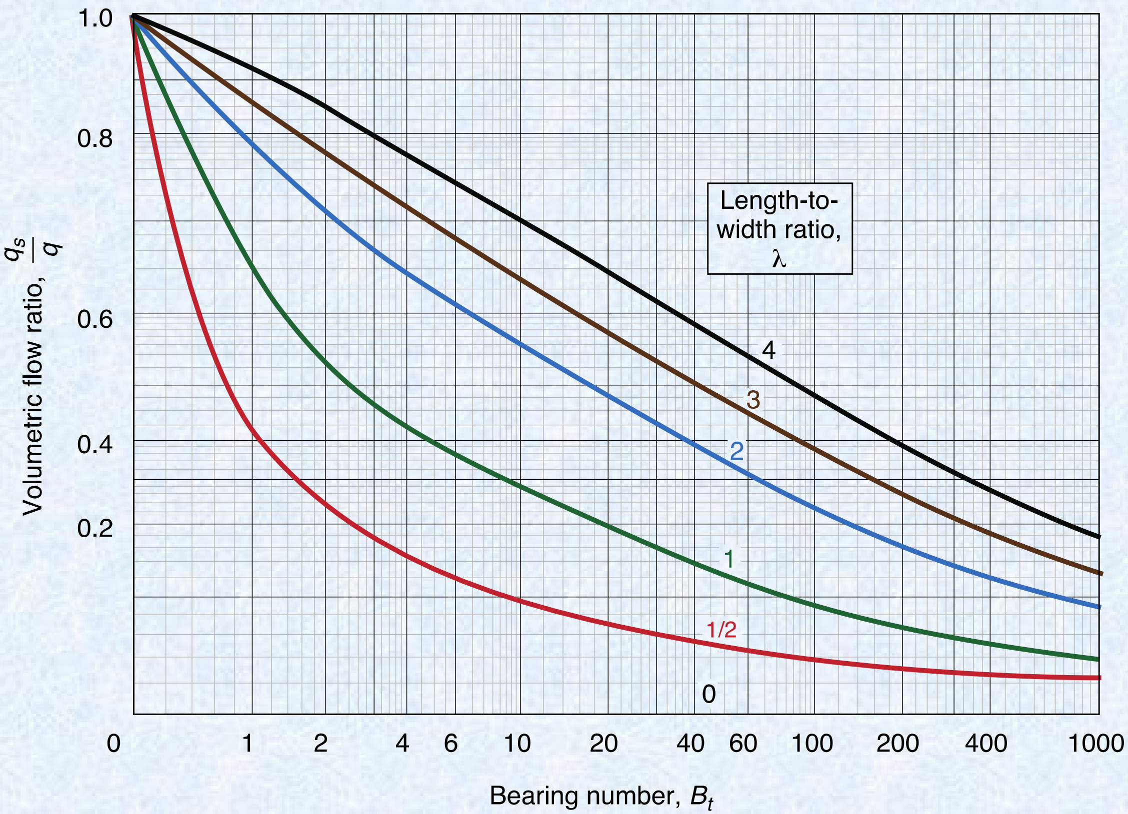


Figure 12.16: Chart for determining performance parameters of fixed-incline thrust bearings. (d) Lubricant side flow.

Allowable Film Thickness

Surface finish (centerline average), R_a μm	μin.	Description of surface	Examples of manufacturing methods	Approximate relative costs	Allowable outlet (minimum) film thickness[a], h_o μm	μin.
0.1-0.2	4-8	Mirror-like surface without tool marks; close tolerances	Grind, lap, and superfinish	17-20	2.5	100
0.2-0.4	8-16	Smooth surface without scratches; close tolerances	Grind and lap	17-20	6.2	250
0.4-0.8	16-32	Smooth surfaces; close tolerances	Grind, file, and lap	10	12.5	500
0.8-1.6	32-63	Accurate bearing surface without tool marks	Grind, precision mill, and file	7	25	1000
1.6-3.2	63-125	Smooth surface without objectionable tool marks moderate tolerances	Shape, mill, grind, and turn	5	50	2000

[a] The values of film thickness are given only for guidance. They indicate the film thickness required to avoid metal-to-metal contact under clean oil conditions with no misalignment. It may be necessary to take a larger film thickness than that indicated (e.g., to obtain an acceptable temperature rise).

Table 12.1: Allowable outlet (minimum) film thickness for a given surface finish.

Pressure Distribution – Journal Bearing

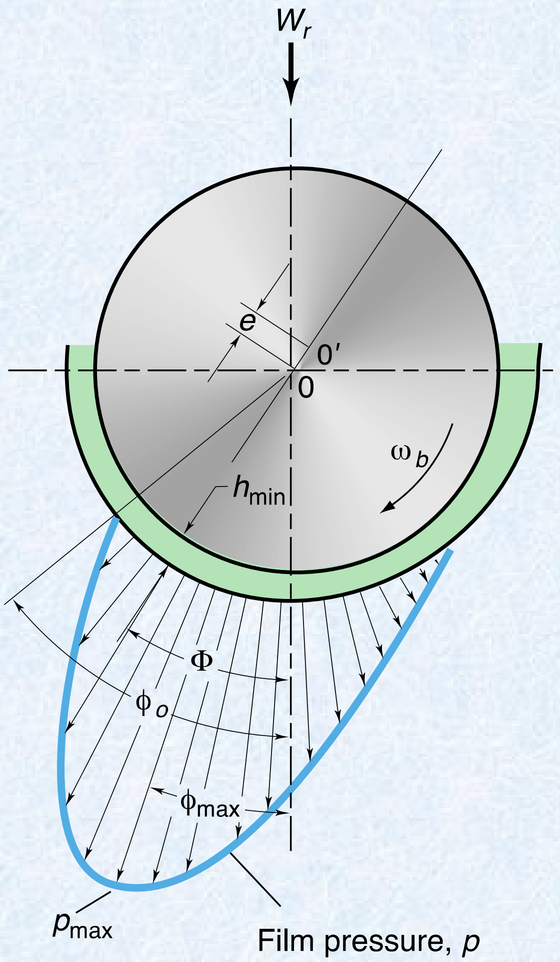


Figure 12.17: Pressure distribution around a journal bearing

Load per Area – Journal Bearings

Application	Average radial load per unit area, W_r' psi	MPa
Automotive engines:		
main bearings	600-750	4-5
connecting rod bearing	1700-2300	10-15
Diesel engines:		
main bearings	900-1700	6-12
connecting rod bearing	1150-2300	8-15
Electric motors	120-250	0.8-1.5
Steam turbines	150-300	1.0-2.0
Gear reducers	120-250	0.8-1.5
Centrifugal pumps	100-180	0.6-1.2
Air compressors:		
main bearings	140-280	1-2
crankpin	280-500	2-4
Centrifugal pumps	100-180	0.6-1.2

Table 12.2: Typical radial load per area W_r^* in use for journal bearings. *Source:* From Juvinall and Marshek [1991].

Concentric Journal Bearing

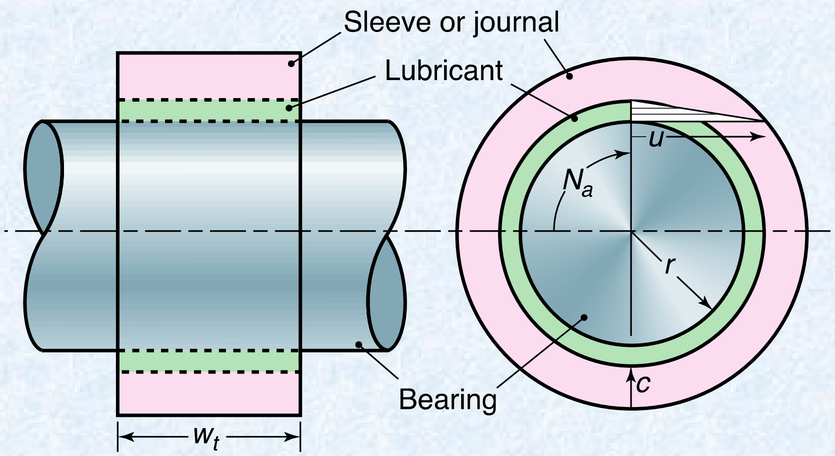


Figure 12.18: Concentric journal bearing.

Petrov's equation:

$$T = Fr = \frac{4\pi^2 \eta_o r^3 w_t N_a}{c} = \frac{2\pi \eta_o r^3 w_t \omega}{c}$$

Power loss:

$$h_p = \frac{8\pi^3 \eta_o r^3 w_t N_a^2}{c} = \frac{2\pi \eta_o r^3 w_t \omega^2}{c}$$

Bearing Surfaces

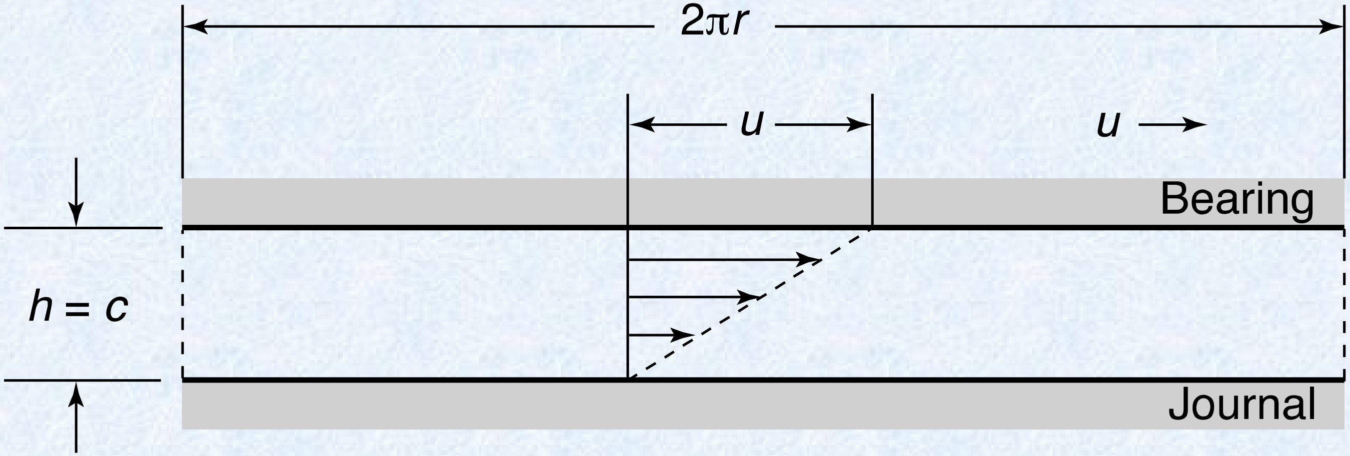


Figure 12.19: Developed journal and bearing surfaces for a concentric journal bearing.

Design Procedure 12.2: Journal Bearings

1. Usually, design problems are underconstrained. Diameter is usually predetermined from shaft size, and load and speed are design requirements. Table 12.2 can be used to select a diameter-to-width ratio if it has not been specified. Otherwise, Fig. 12.20 can be used to obtain the bearing number, B_j. The dimensionless film thickness, $h_{\min}/c$, can also be obtained from Fig. 12.20. Using a film thickness from Table 12.1 allows calculation of the radial clearance, c. Alternatively, a known value of c results in an estimate of film thickness that can be compared to Table 12.1 to evaluate the design and specify journal bearing and sleeve surface roughness and manufacturing approach. At this point, the required lubricant viscosity can be determined to produce the required bearing number. A review of Fig. 8.13 allows selection of a lubricant and operating temperature to produce this viscosity.
2. Performance parameters for the journal bearing can then be obtained from Figs. 12.21 through 12.26.

Design Procedure 12.2 (concluded)

3. The temperature rise within the journal bearing can then be obtained from Eq. (12.87). Temperature rise will indicate the adequacy of the design; the temperatures in Fig. 8.13 should in general not be exceeded within the bearing or else the lubricant may experience premature loss of viscosity and thermal degradation. Moderate temperature rises can provide useful information for design of lubricant cooling systems, if needed.
4. In some cases, a bearing is specified and its performance is to be evaluated. In this case, the lubricant viscosity is unknown, since the temperature rise and mean temperature are not known beforehand. The approach in this case is to assume a temperature rise, obtain performance parameters from Figs. 12.21 through 12.26, and calculate a temperature rise. If this temperature rise is close to that assumed, then this represents the operating condition of the journal bearing. If the assumed and calculated values differ by more than 10°C or so, a revised estimate can be made and the procedure repeated. Journal bearings are well-behaved, and a solution usually is obtained after one or two iterations.

Bearing Number vs. Film Thickness

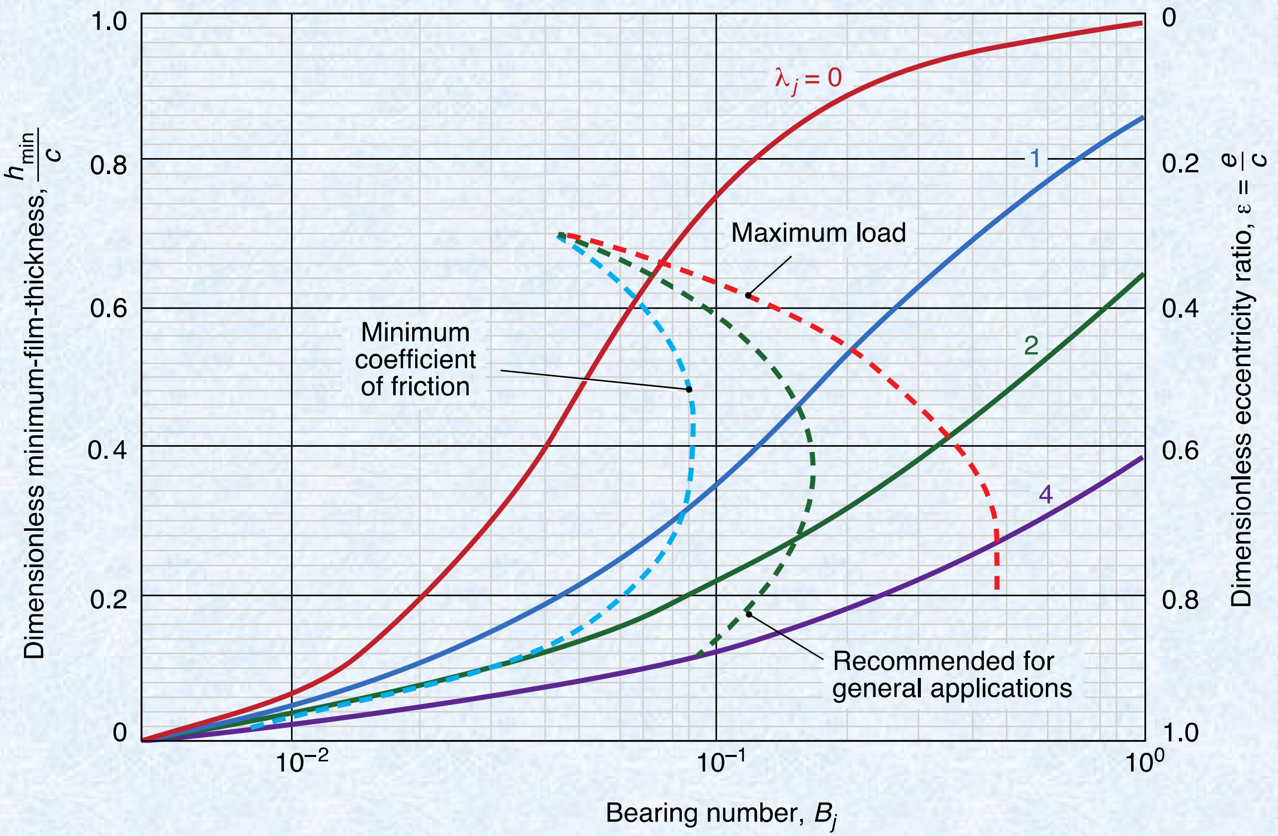


Figure 12.20: Effect of bearing number on minimum film thickness for four diameter-to-width ratios. The shaded area is the most common operating range for well-designed journal bearings. *Source:* From Raimondi and Boyd [1958].

Attitude Angle – Journal Bearings

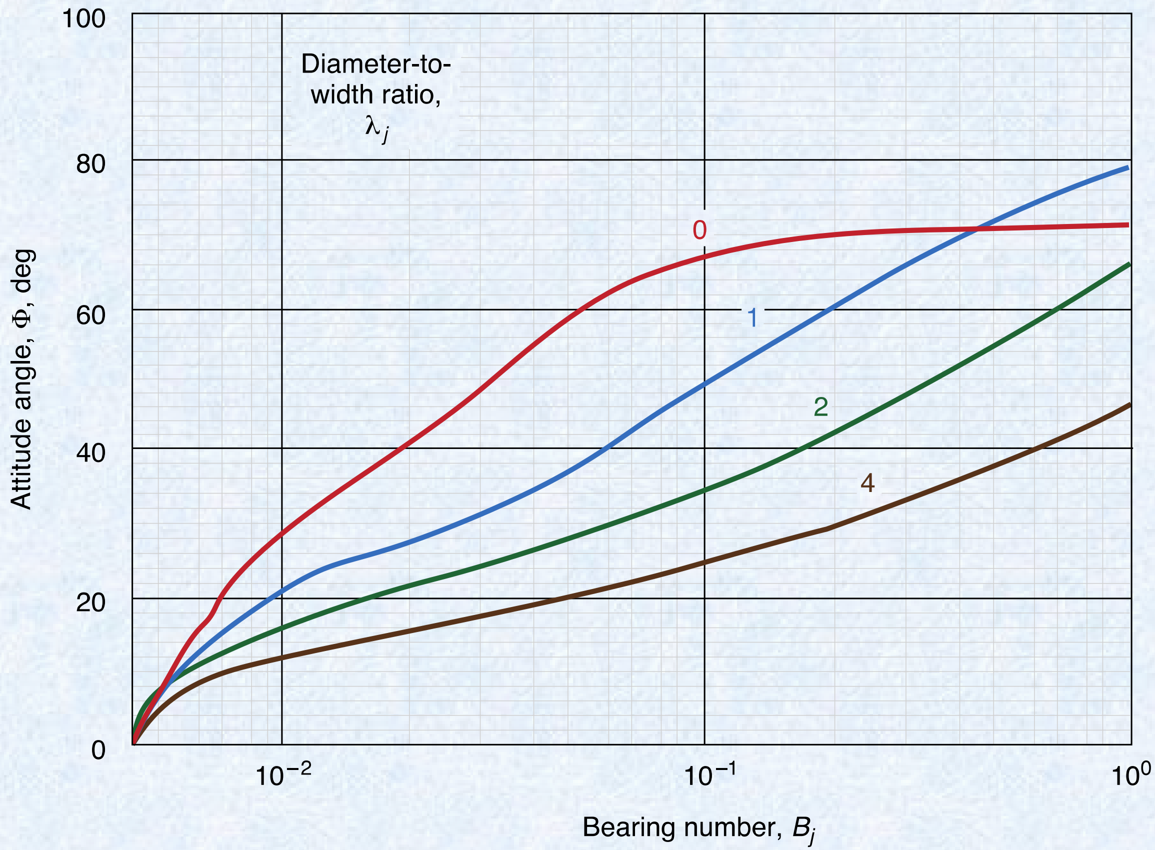


Figure 12.21: Effect of bearing number on attitude angle for four diameter-to-width ratios. *Source:* From Raimondi and Boyd [1958].

Friction – Journal Bearings

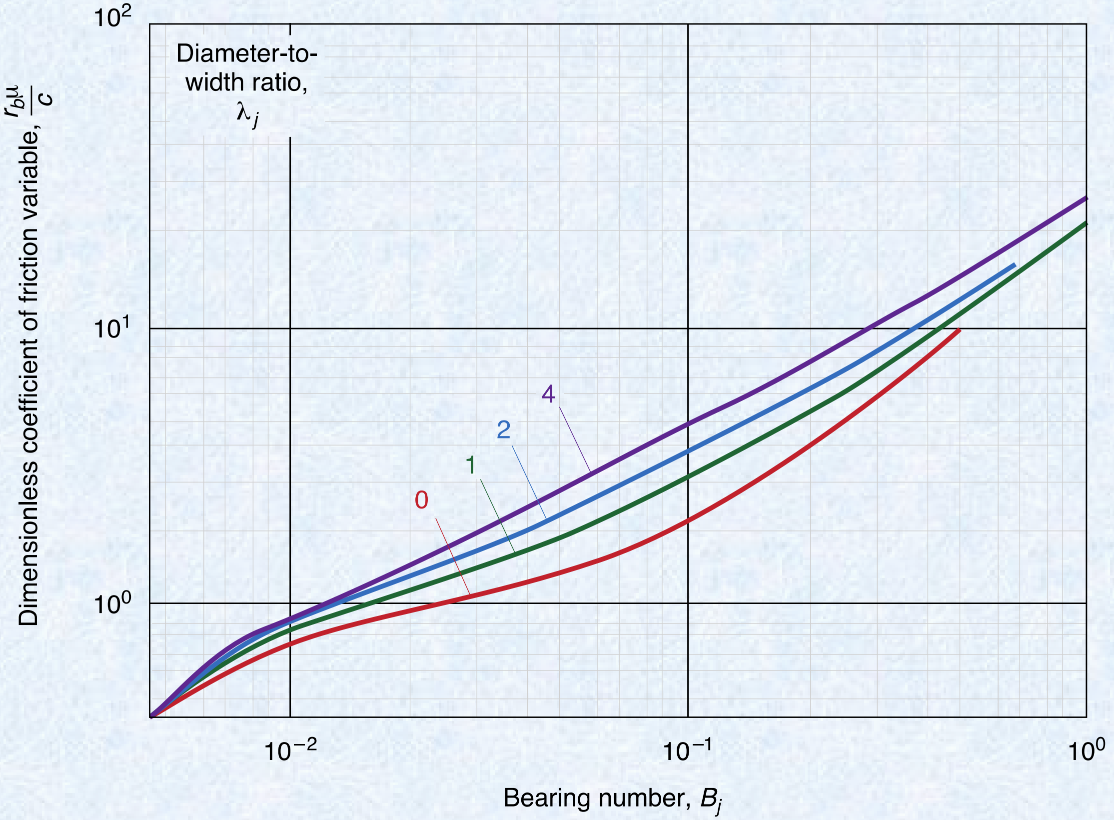


Figure 12.22: Effect of bearing number on coefficient of friction for four diameter-to-width ratios. *Source:* From Raimondi and Boyd [1958].

Flow Rate – Journal Bearings

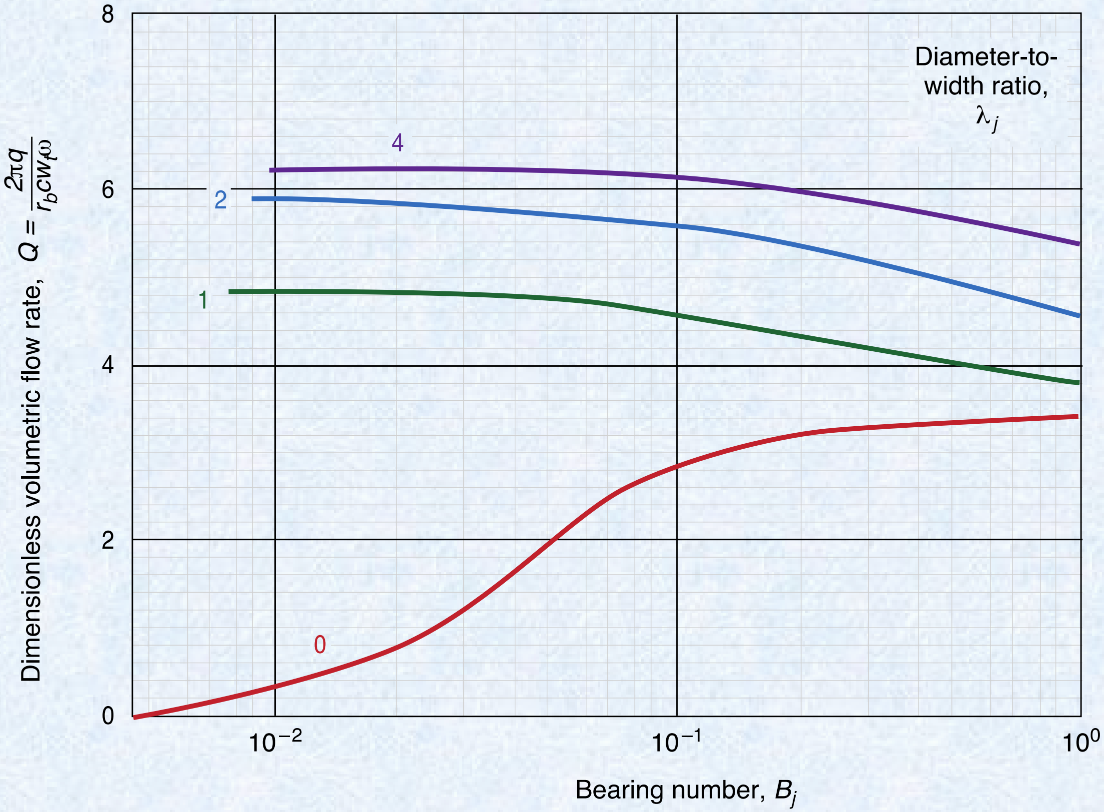


Figure 12.23: Effect of bearing number on dimensionless volumetric flow rate for four diameter-to-width ratios. *Source:* From Raimondi and Boyd [1958].

Side Flow – Journal Bearings

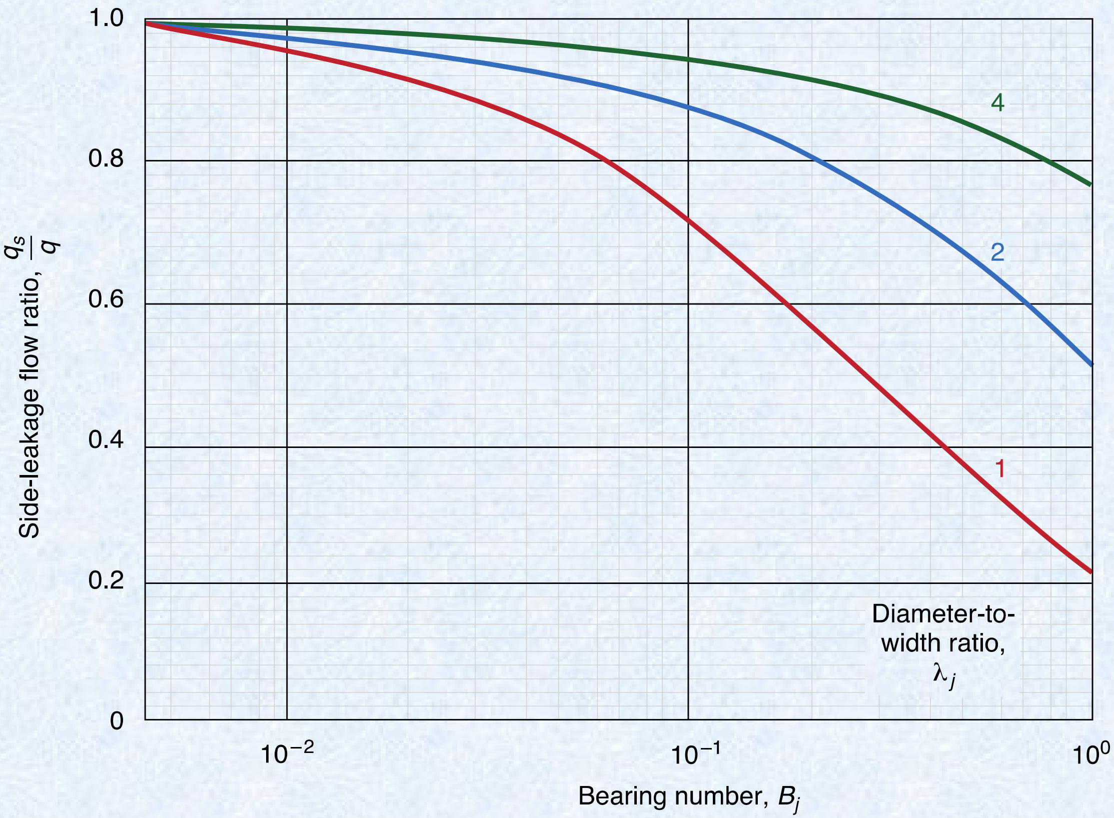


Figure 12.24: Effect of bearing number on side-leakage flow ratio for four diameter-to-width ratios. *Source:* From Raimondi and Boyd [1958].

Maximum Pressure – Journal Bearings

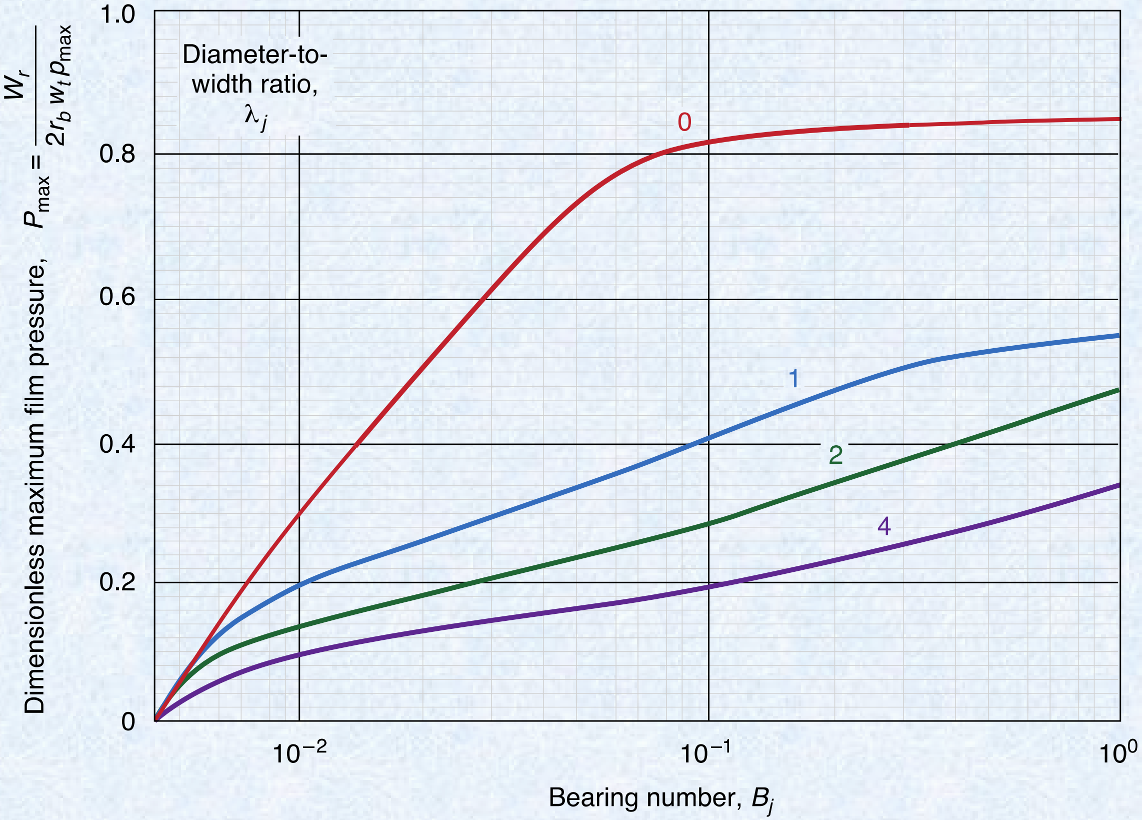


Figure 12.25: Effect of bearing number on dimensionless maximum film pressure for four diameter-to-width ratios. *Source:* From Raimondi and Boyd [1958].

Terminating Pressure – Journal Bearings

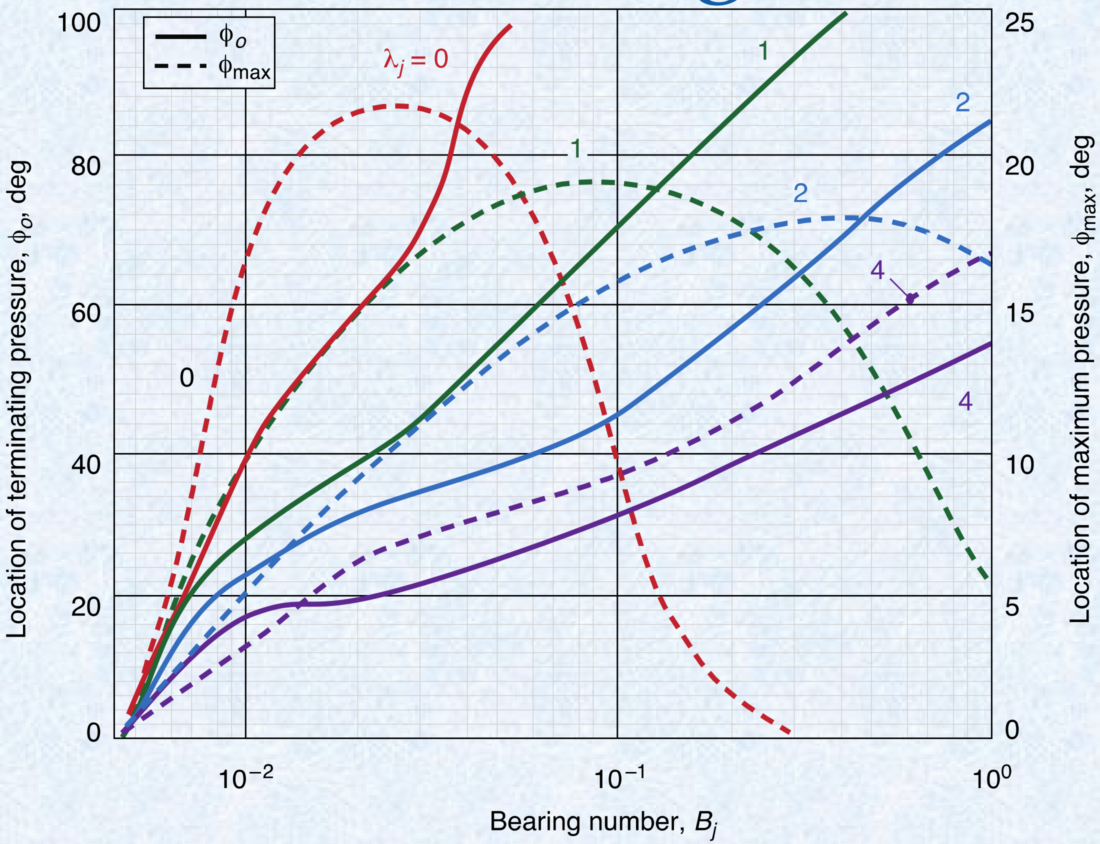


Figure 12.26: Effect of bearing number on location of terminating and maximum pressures for four diameter-to-width ratios. *Source:* From Raimondi and Boyd [1958].

Performance Optimization

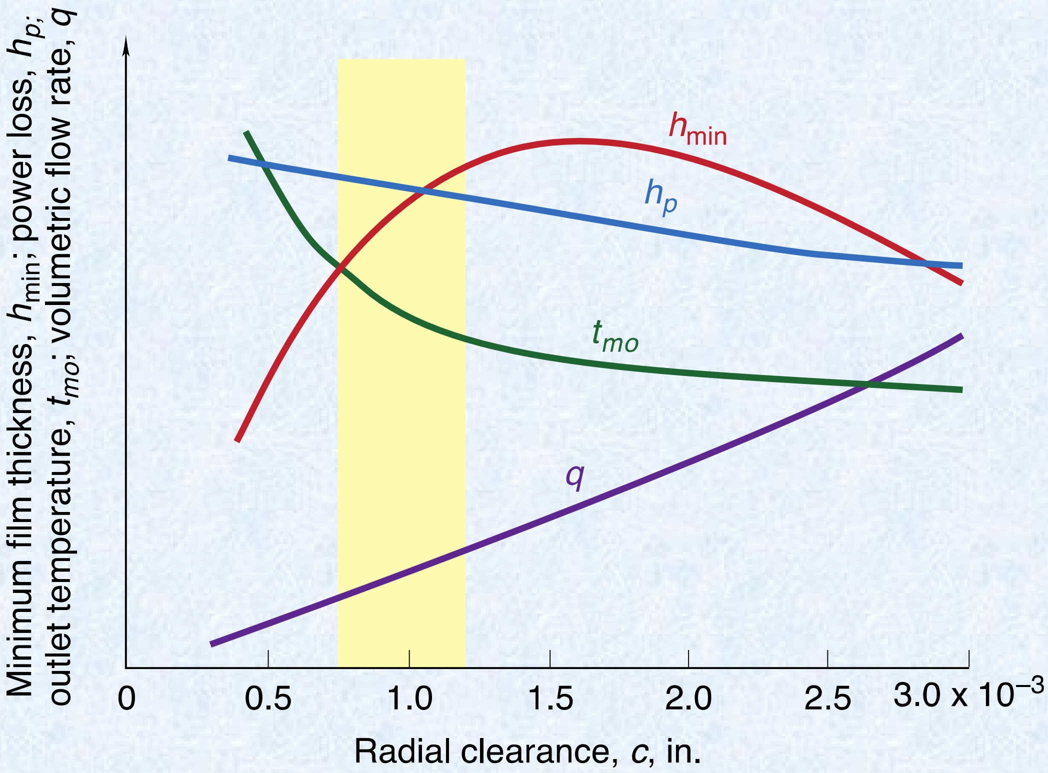


Figure 12.27: Effect of radial clearance on some performance parameters for a particular case.

Squeeze Film Bearing

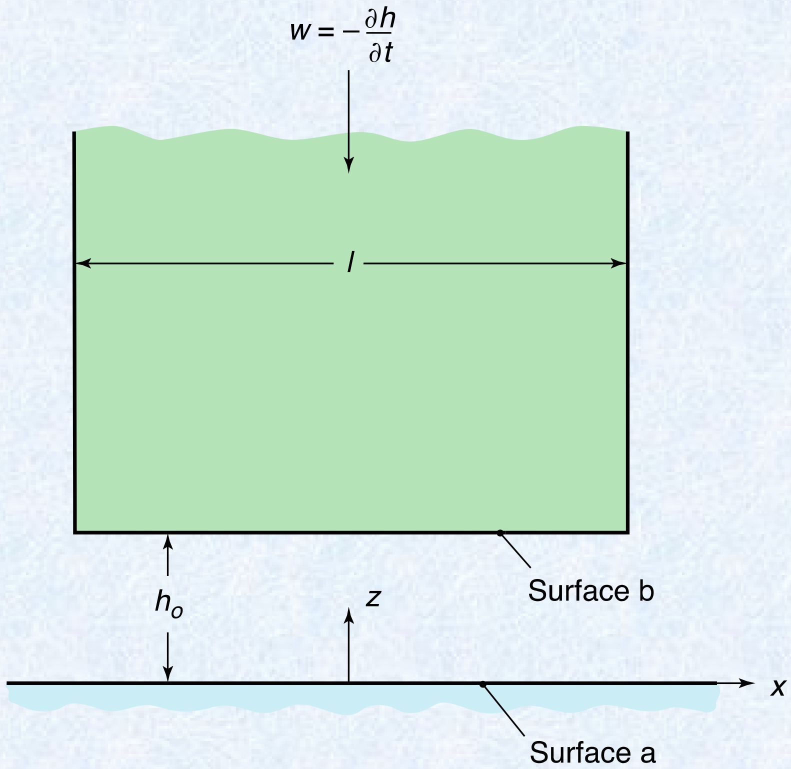


Figure 12.28: Parallel-surface squeeze film bearing.

Film thickness as a function of time:

$$h_{o,2} = \frac{h_{o,1}}{\left[1 + \left(2W_z' \Delta t h_{o,1}^2 / \eta_o l^3\right)\right]^{1/2}}$$

Hydrostatic Film Formation

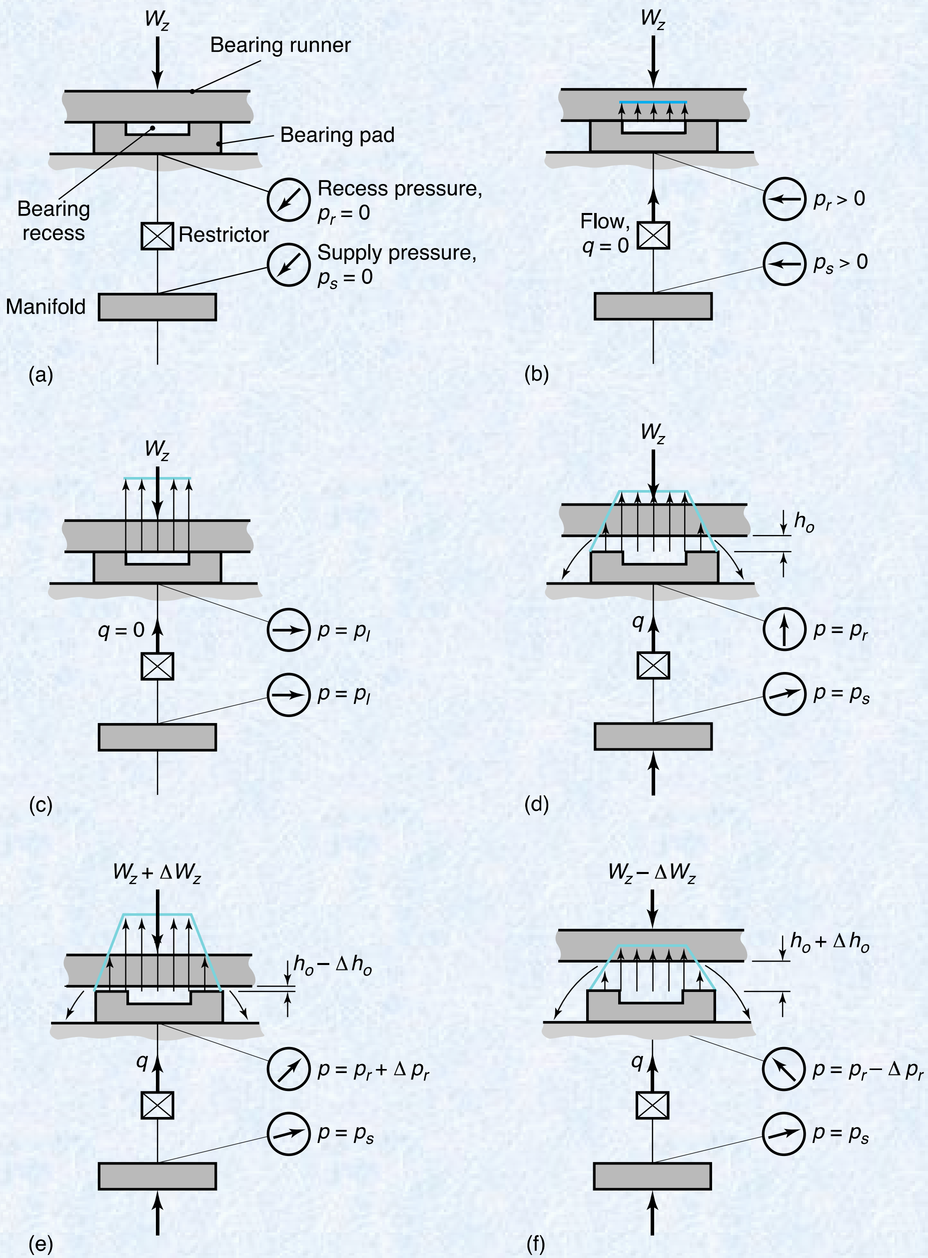


Figure 12.29: Formation of fluid film in hydrostatic bearing system. (a) Pump off; (b) pressure build up; (c) pressure times recess area equals normal applied load; (d) bearing operating; (e) increased load; (f) decreased load. *Source:* From Rippel [1963].

Hydrostatic Bearing Configurations

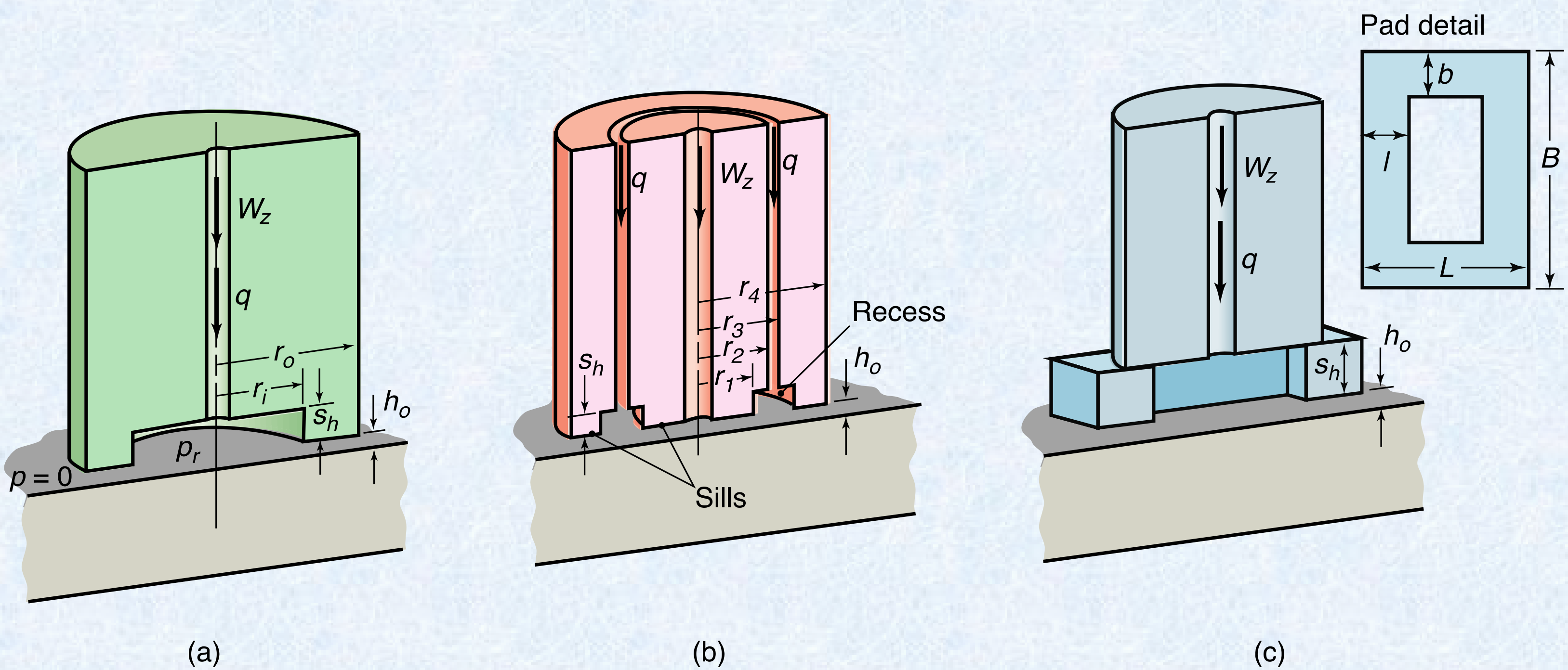


Figure 12.30: Hydrostatic thrust bearing configurations. (a) Radial-flow with circular step pad; (b) radial flow with annular pad; (c) rectangular pad.

Hydrodynamic Bearings in Engines

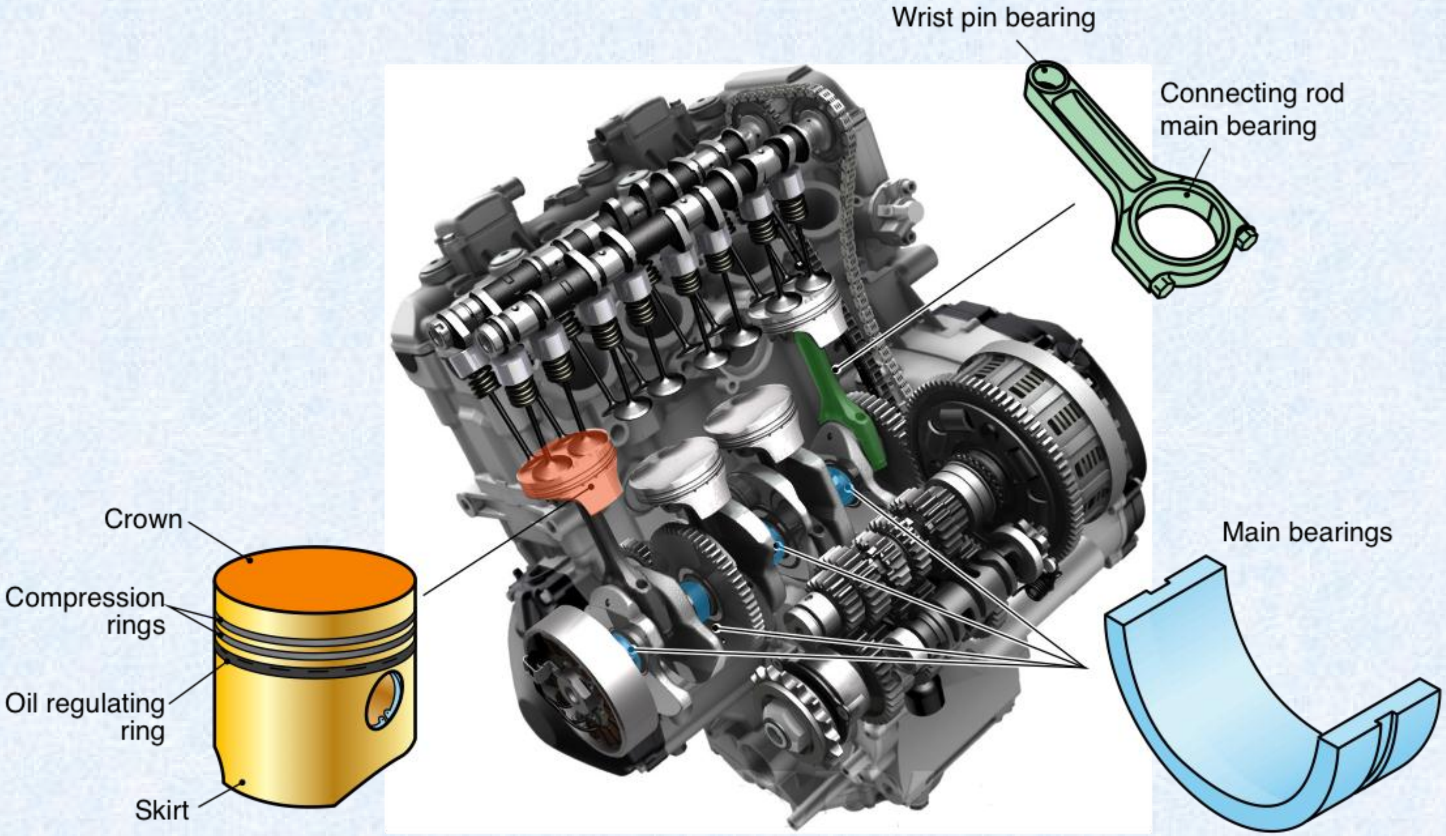


Figure 12.31: Illustration of an internal combustion engine with selected hydrodynamic bearings.