ME 308
MACHINE ELEMENTS II

CHAPTER 4

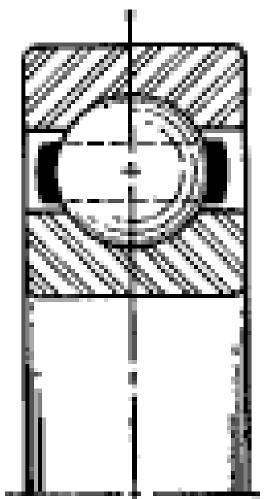
JOURNAL BEARINGS

(NO ROLLING ELEMENTS,
ONLY A SHAFT AND A HOLE AND
SOME LUBRICANT/OIL IN BETWEEN)

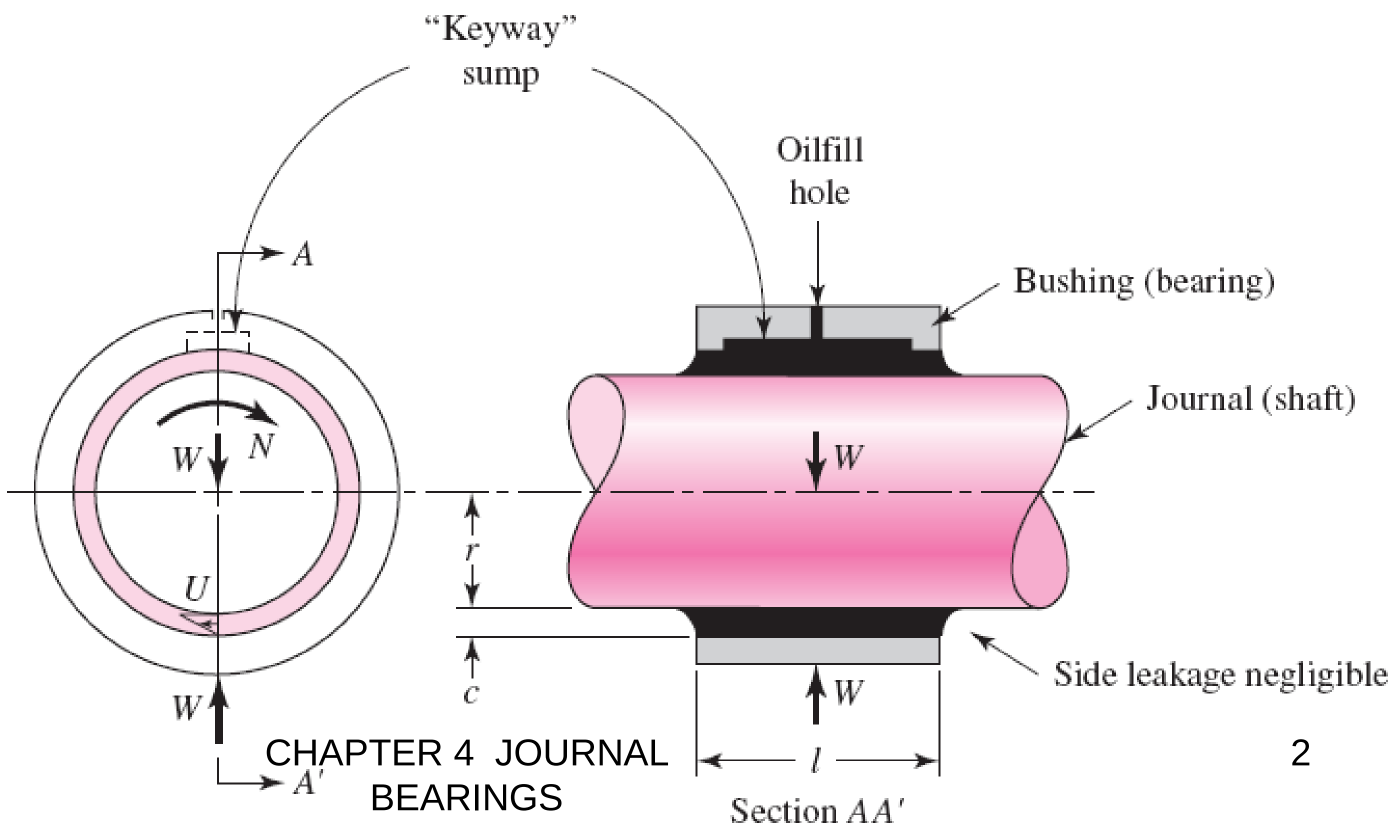
INTRODUCTION

- **Journal bearings** are the supporting units based on lubrication of two surfaces with almost no metal to metal contact during relative movement and no rolling elements in between.

The object of lubrication is to reduce friction, wear, and heating of machine parts that move relative to each other.

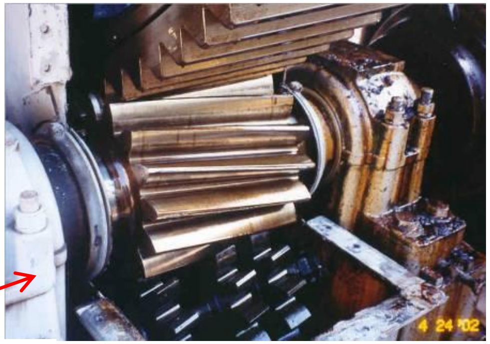


(a)
Deep groove



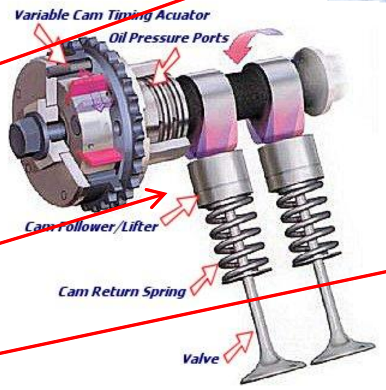
Most machine elements which are in motion or moves relative to other elements are lubricated to:

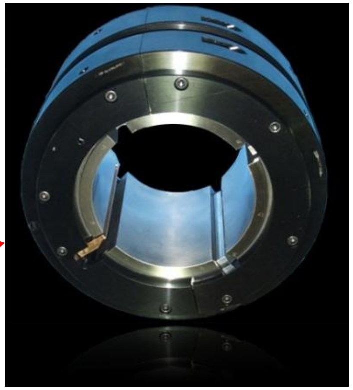
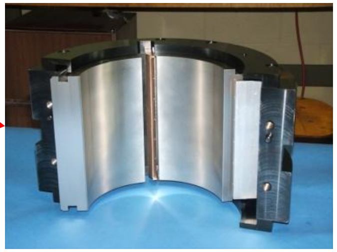
- to reduce friction between contacting parts,
- to reduce wear of the rubbing or contacting surfaces, and
- to reduce heating of the machine elements/ parts that move relative to each other.



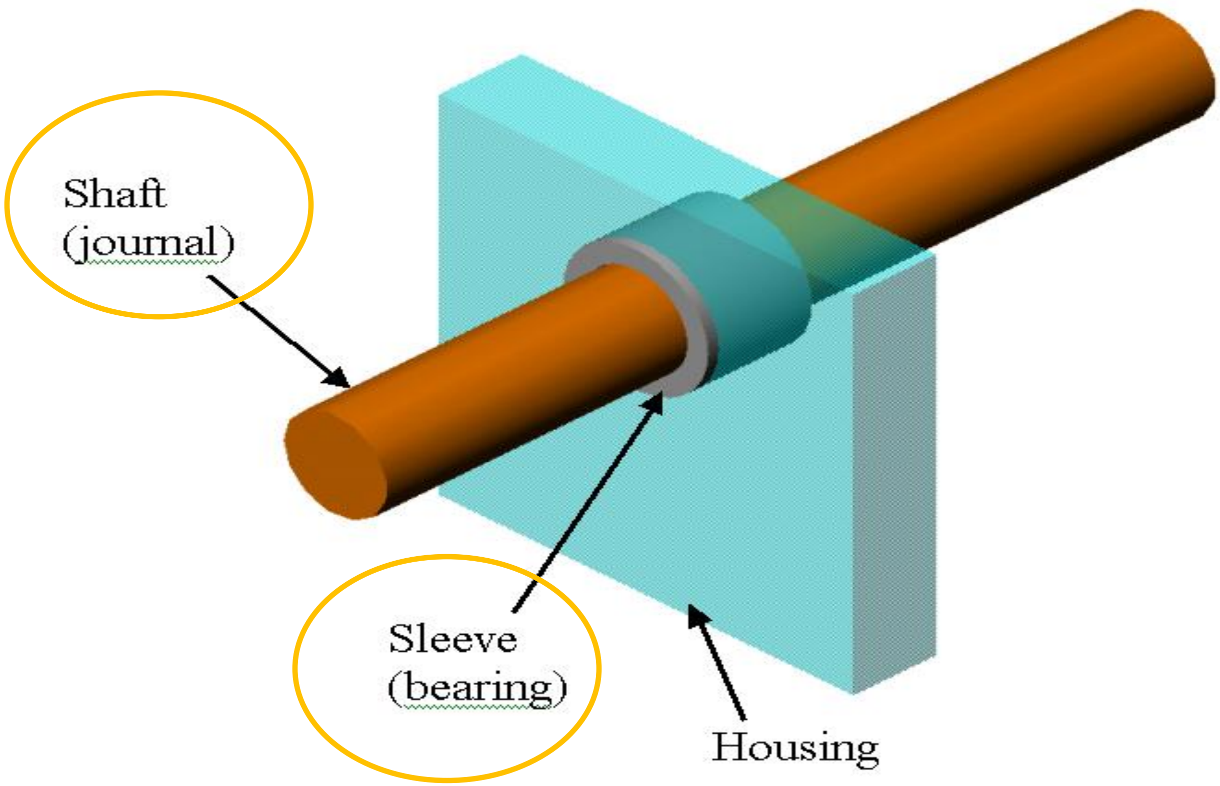
Lubrication, by means of oils or other low friction materials, is used in many instances such as:

- rolling contacting bearing (RCB's),
- gears meshing
- cam and follower mechanisms,
- piston moving in cylinder, and
- **Journal bearings(*).**

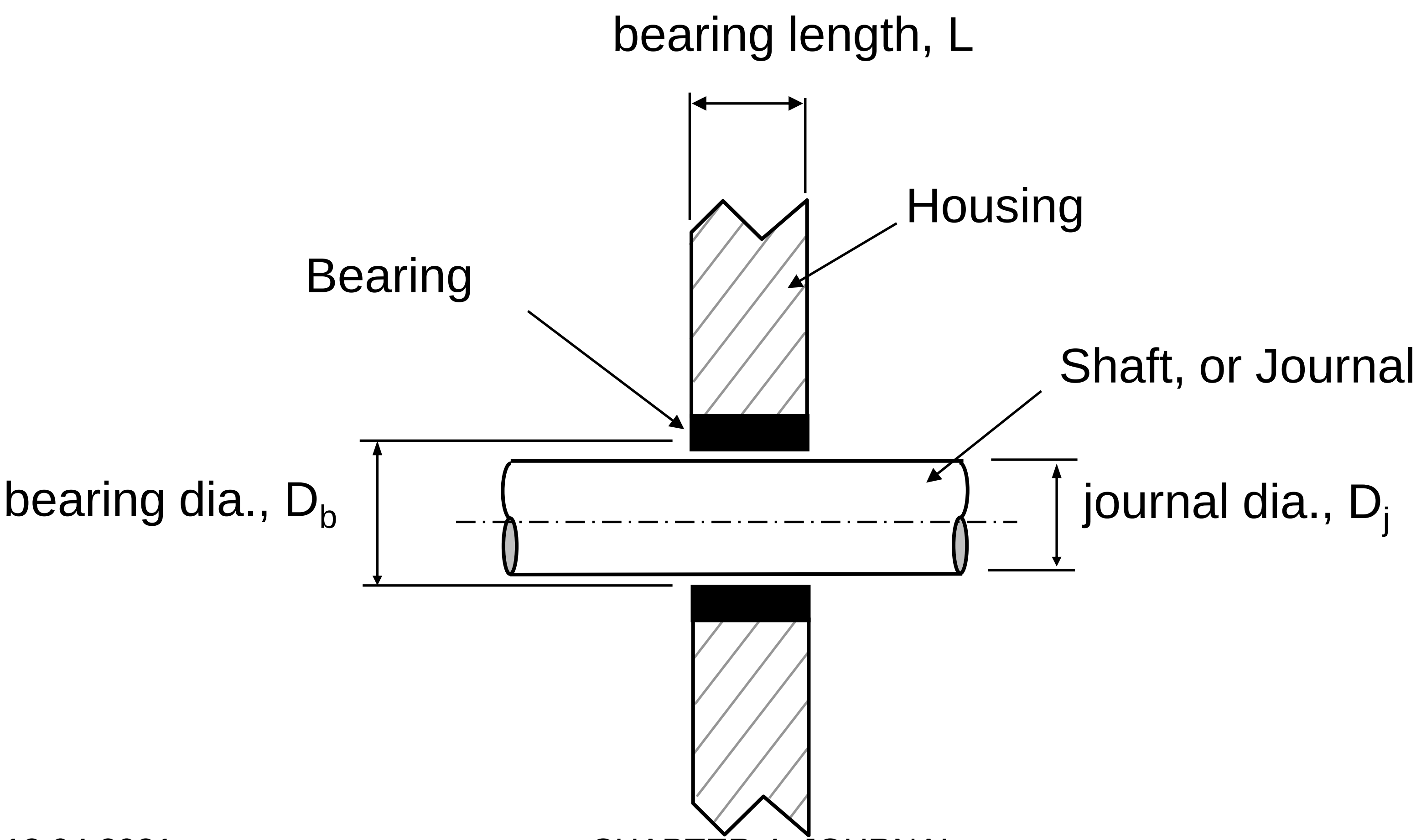


Bearings that allow two surfaces to slide on each other like the one below (with no rolling elements in between) are usually called journal bearings.



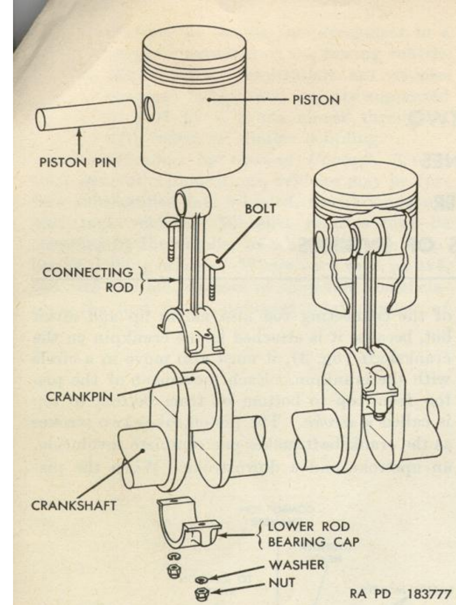
BEARING GEOMETRY



Journal bearings

Generally used in:

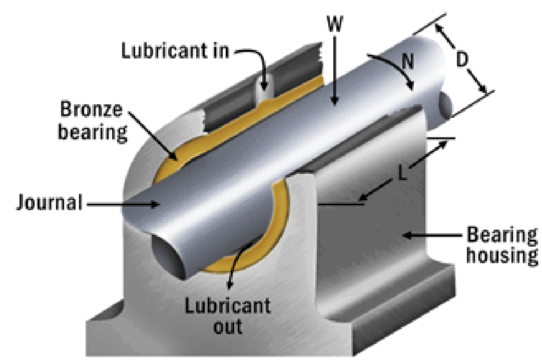
1. crank-shaft bearings,
2. connecting rod bearings,
3. steam turbine shaft bearings of power generation plants, and
4. other relatively light load and unimportant services.



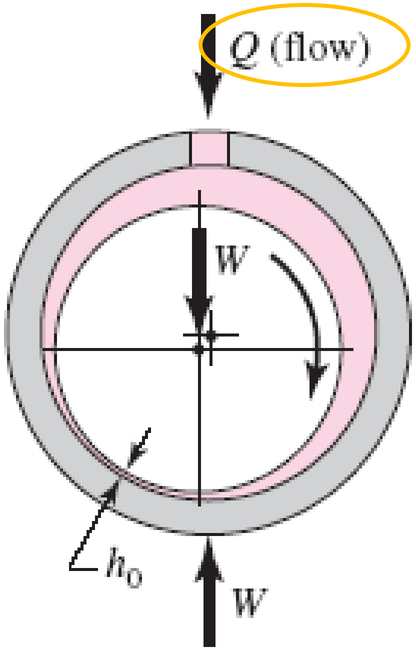
Journal Bearings have two main types:

1. <u>Ones with solid-film lubricant</u> such as:
   - graphite and
   - molybdenum disulphide

2. <u>Ones with liquid lubricant</u>
   a) Hydro-dynamic bearings,
   b) Hydro-static bearings.

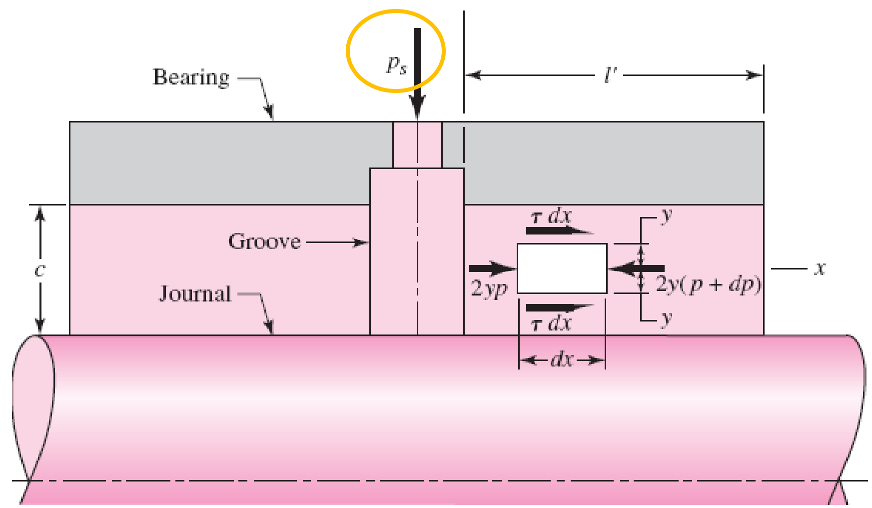


a) Hydro-dynamic bearings: lubrication (by oil) happens itself due to a large relative speed between journal and bearing.



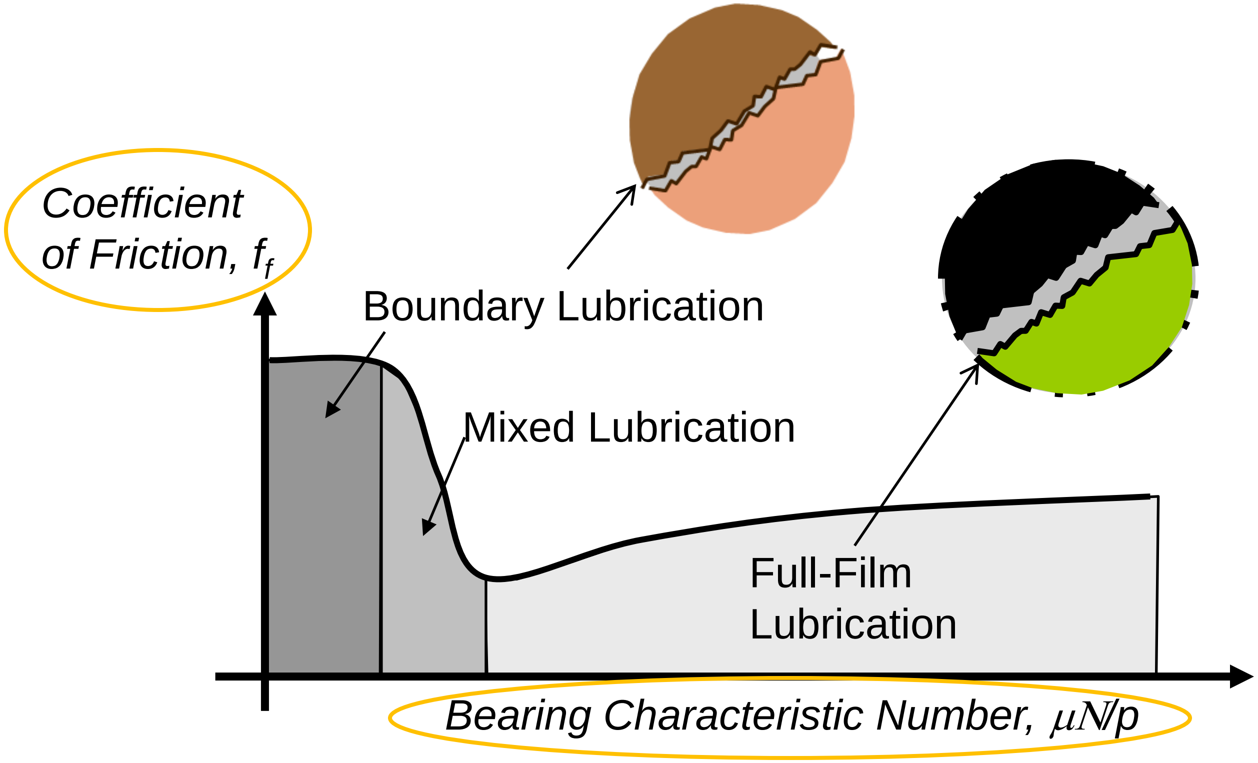
(b) Lubricated

b) Hydro-static bearings: relative speed between journal and bearing is low and lubricant is pumped at a certain pressure (Ps) inside the clearance to provide the lubrication.



TYPES OF LUBRICATION

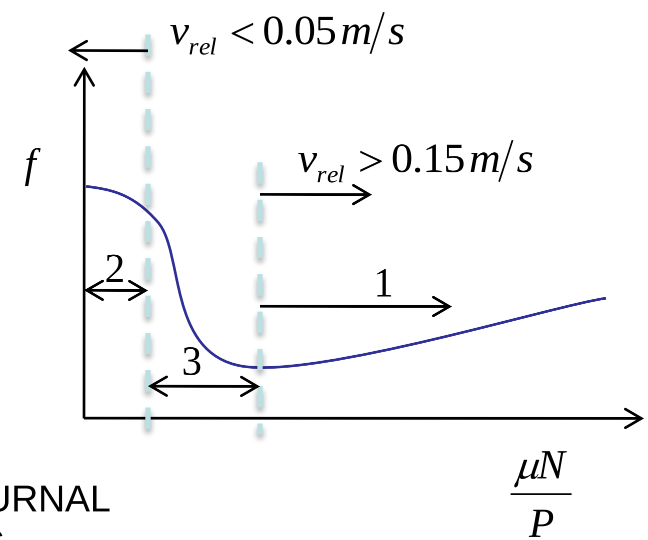
I. Hydrodynamic lubrication
II. Hydrostatic lubrication
III. Elastohydrodynamic lubrication
IV. Boundary lubrication
V. Solid film lubrication

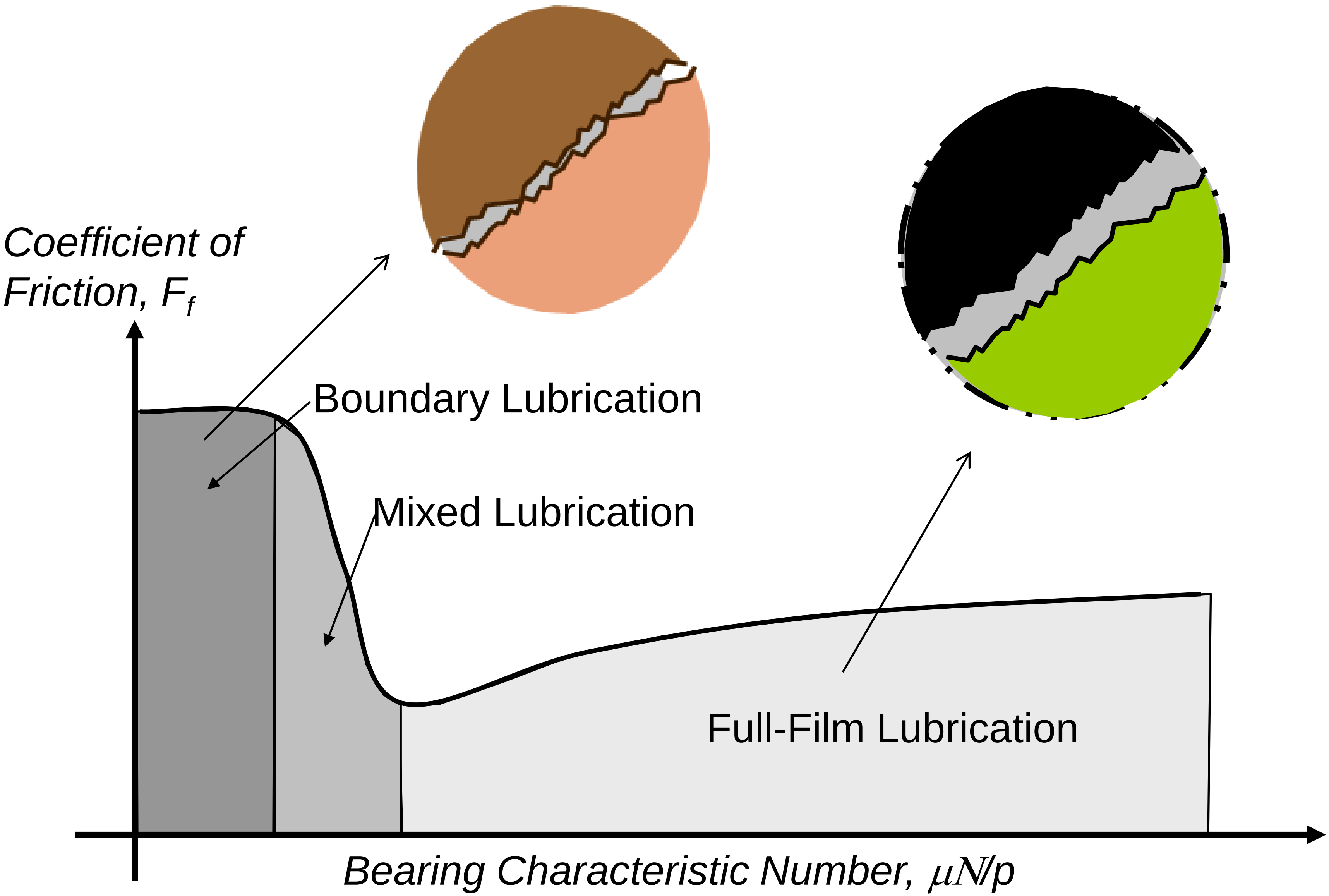


HYDRODYNAMIC LUBRICATION (HDJB)

3 types of lubrication happens in HDJB depending on the relative speed, load carried and the lubricant viscosity;

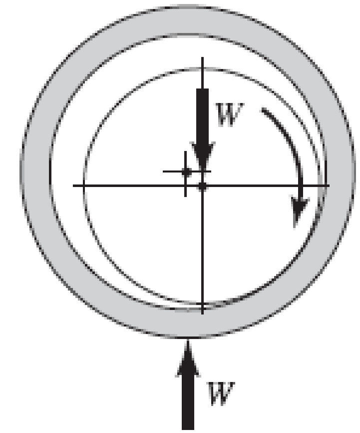
1. Thick or full-film (preferred with very low friction coefficient) — $f_1 = 0.001\text{-}0.01$ *a thick oil film, surfaces do not touch each other*
2. Thin film (boundary lubrication) (not preferred since friction coefficient is relatively high) — $f_2 = 0.08\text{-}0.14$ *a thin oil film therefore surface asperities touch each other*
3. Mixed film (Thick + Thin) (not preferred due to instability between thick and thin film)



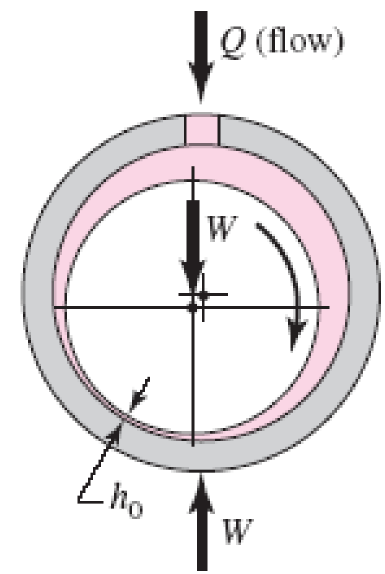

Coefficient of Friction, F_f
Boundary Lubrication
Mixed Lubrication
Full-Film Lubrication
Bearing Characteristic Number, $\mu N/p$

HYDRODYNAMIC LUBRICATION

- *Hydrodynamic lubrication means that the load-carrying surfaces of the bearing are* separated by a relatively thick film of lubricant, so as to prevent metal-to-metal contact,
- Hydrodynamic lubrication does not depend upon the introduction of the lubricant under pressure, though that may occur; but it does require the existence of an adequate supply at all times.
- The film pressure is created by the moving surface itself pulling the lubricant into a wedge-shaped zone at a velocity sufficiently high to create the pressure necessary to separate the surfaces against the load on the bearing.
- Hydrodynamic lubrication is also called *full-film or fluid lubrication.*



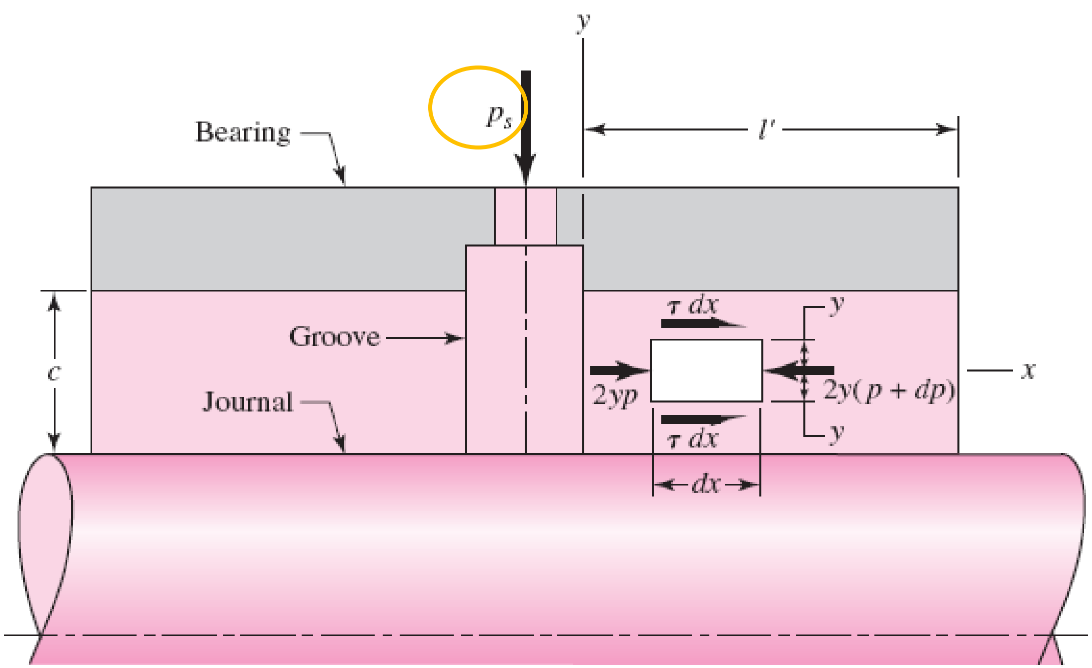
(a) Dry



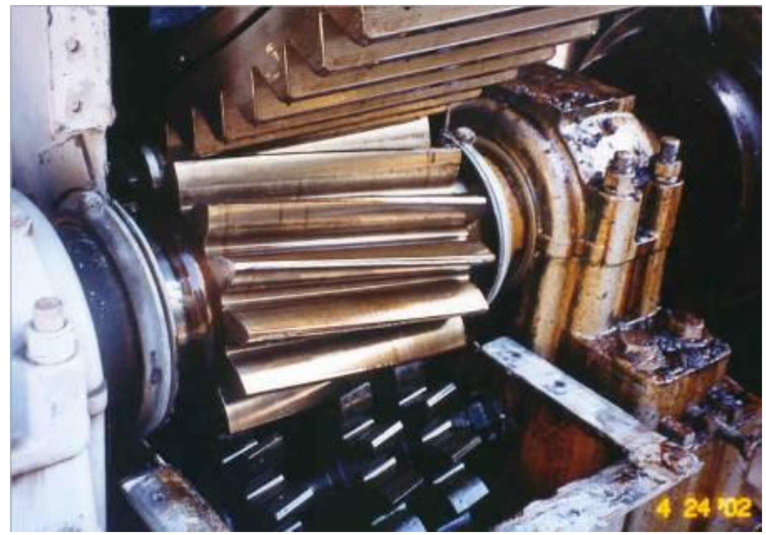
(b) Lubricated

HYDROSTATIC LUBRICATION

- *Hydrostatic lubrication is obtained by introducing the lubricant* into the load-bearing area at a pressure (Ps) high enough to separate the surfaces with a relatively thick film of lubricant.
- So, unlike hydrodynamic lubrication, this kind of lubrication does not require motion of one surface relative to another.



❖ *Elastohydrodynamic lubrication is the phenomenon that occurs when a lubricant is* introduced between surfaces that are in rolling contact, such as mating gears or rolling bearings.



❖The mathematical explanation requires the Hertzian theory of contact stress and fluid mechanics.

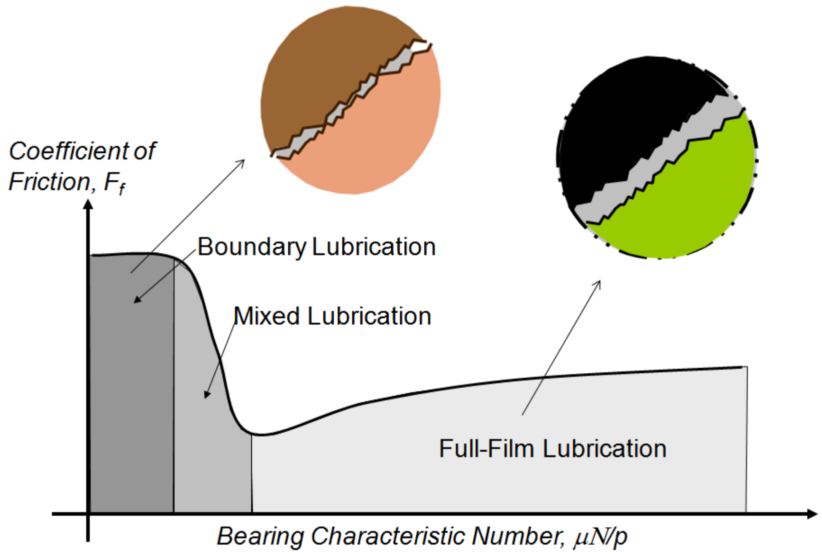
BOUNDRY LUBRICATION

- Insufficient surface area,
- a drop in the velocity of the moving surface,
- a lessening in the quantity of lubricant delivered to a bearing,
- an increase in the bearing load,
- or an increase in lubricant temperature resulting in a decrease in viscosity

any one of these above may prevent the buildup of a film thick enough for full-film lubrication.

- When this happens, the highest asperities may be separated by lubricant films only several molecular dimensions in thickness.
- This is called *boundary lubrication.*

- *The change* from hydrodynamic to boundary lubrication is not at all a sudden or abrupt one. It is probable that a mixed hydrodynamic- and boundary-type lubrication occurs first, and as the surfaces move closer together, the boundary-type lubrication becomes predominant.



4.2.4 SOLID-FILM LUBRICATION

- When bearings must be operated at extreme or high temperatures, a *solid-film lubricant* such as graphite or molybdenum disulfide must be used because the ordinary mineral oils are not satisfactory.

- Much research is currently being carried out to find composite bearing materials with low wear rates as well as small frictional coefficients.

4.3 **Viscosity**

In Fig. 4–1 let a plate *A* be moving with a velocity *U* on a film of lubricant of thickness *h.* We imagine the film as composed of a series of horizontal layers and the force *F* causing these layers to deform or slide on one another just like a deck of cards.

The layers in contact with the moving plate are assumed to have a velocity *U;* those in contact with the stationary surface are assumed to have a zero velocity. Intermediate layers have velocities that depend upon their distances *y* from the stationary surface. Newton’s viscous effect states that the shear stress in the fluid is proportional to the rate of change of velocity with respect to *y*. Thus

$$\tau = \frac{F}{A} = \mu \frac{du}{dy}$$

where *μ* is the constant of proportionality and defines absolute viscosity, also called dynamic viscosity.

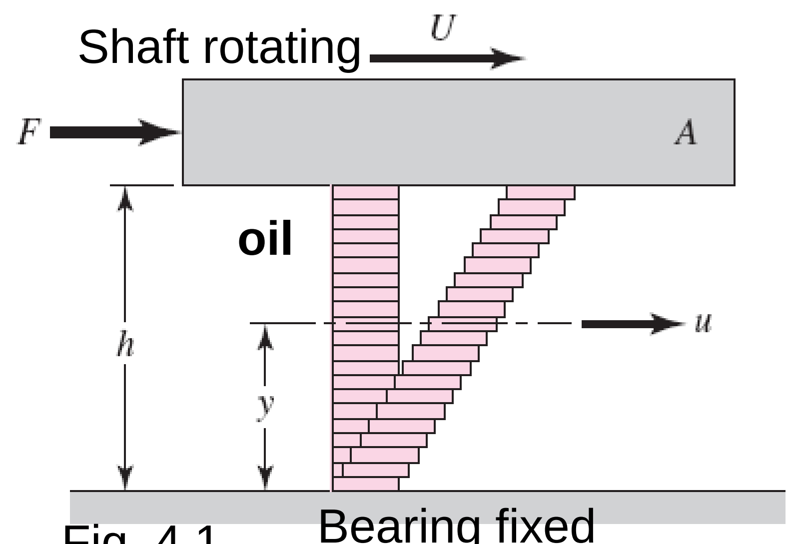


Fig. 4.1

$$\tau = \frac{F}{A} = \mu \frac{du}{dy}$$

- The derivative *du*/*dy* is the rate of change of velocity with distance and may be called the rate of shear, or the velocity gradient.
- The viscosity *μ* is thus a measure of the internal frictional resistance of the fluid.
- For most lubricating fluids, the rate of shear is constant, and *du*/*dy* = *U*/*h*. Thus, from Eq. (4–1)

$$\tau = \frac{F}{A} = \mu \frac{U}{h}$$

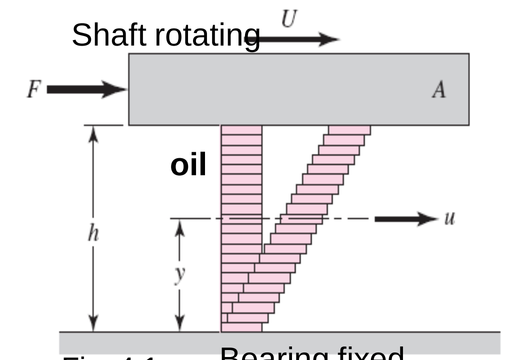


Fig. 4.1

where

τ: is shear stress between lubricant films.

μ : is the dynamic (absolute) viscosity of the lubricant with unit of

$$(Pa - \sec) \quad or \quad \frac{N}{m^2}\sec \quad or \quad \frac{kg}{m - \sec}$$

Fluids exhibiting this characteristic are said to be *Newtonian fluids.*

The unit of viscosity in the ips (English and American) system is seen to be the pound-force-second per square inch; this is the same as stress or pressure multiplied by time.

The ips unit of *viscosity* is called the *reyn, in honor* of Sir Osborne Reynolds who has worked the subject of viscosity.

The absolute viscosity is measured by the pascal-second (Pa · sec) in SI;.

$$(Pa-\text{sec}) \quad or \quad \frac{N}{m^2}\text{sec} \quad or \quad \frac{kg}{m-\text{sec}}$$

The conversion from ips units to SI is the same as for stress.
For example, multiply the absolute viscosity in reyns by 6890 to convert to units of Pa · sec.

The American Society of Mechanical Engineers (ASME) has published a list of cgs (centimeter, gram, seconds) units that are not to be used in ASME documents. This list results from a recommendation by the International Committee of Weights and Measures (CIPM) that the use of cgs units with special names be discouraged.

a unit of force called the *dyne (dyn),*
a unit of dynamic viscosity called the poise (P), and
a unit of kinematic viscosity called the *stoke (St).*

All of these units have been, and still are, used extensively in lubrication studies.

The poise is the cgs (centimeter, gram, seconds) unit of dynamic or absolute viscosity, and its unit is the dynesecond per square centimeter *(dyn · s/cm²)*.

It has been customary to use the centipoise (*cP*) in analysis, because its value is more convenient. When the viscosity is expressed in centipoises, it is designated by *Z. The* conversion from cgs units to SI and ips units is as follows:

$$\mu(Pa \cdot s) = (10)^{-3} Z(cP)$$

$$\mu(reyn) = \frac{Z(cP)}{6.89(10)^{6}}$$

$$\mu(mPa \cdot s) = 6.89\mu^{'}(\mu reyn)$$

In using ips units, the microreyn (μreyn) is often more convenient. The symbol *μ′* will be used to designate viscosity in μreyn such that
$\mu' = \mu/(10^{6})$.

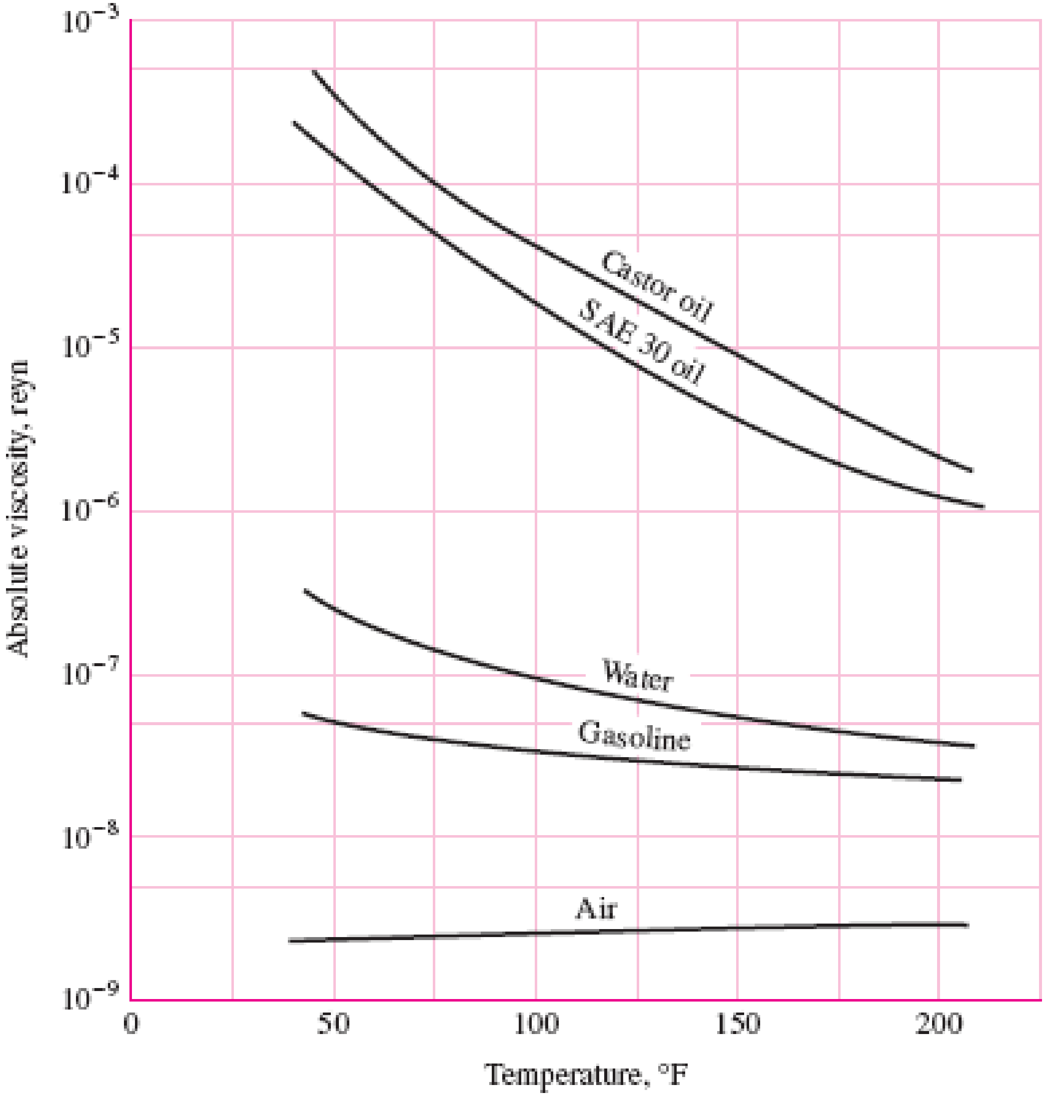


Figure 4–2 shows the absolute viscosity of a number of fluids often used for lubrication purposes and their variation with temperature.

Viscosity varies:

- **inversely with temperature.**
- **directly with pressure**

Both in non-linear fashion.

Fig. 4.2 A comparison of the viscosities of various fluids.

4.4 Petroff's Equation

The phenomenon of bearing friction was first explained by Petroff on the assumption that the shaft is concentric. Though we shall seldom make use of Petroff's method of analysis in the material to follow, it is important because it defines groups of dimensionless parameters and because the coefficient of friction predicted by this law turns out to be quite good even when the shaft is not concentric.

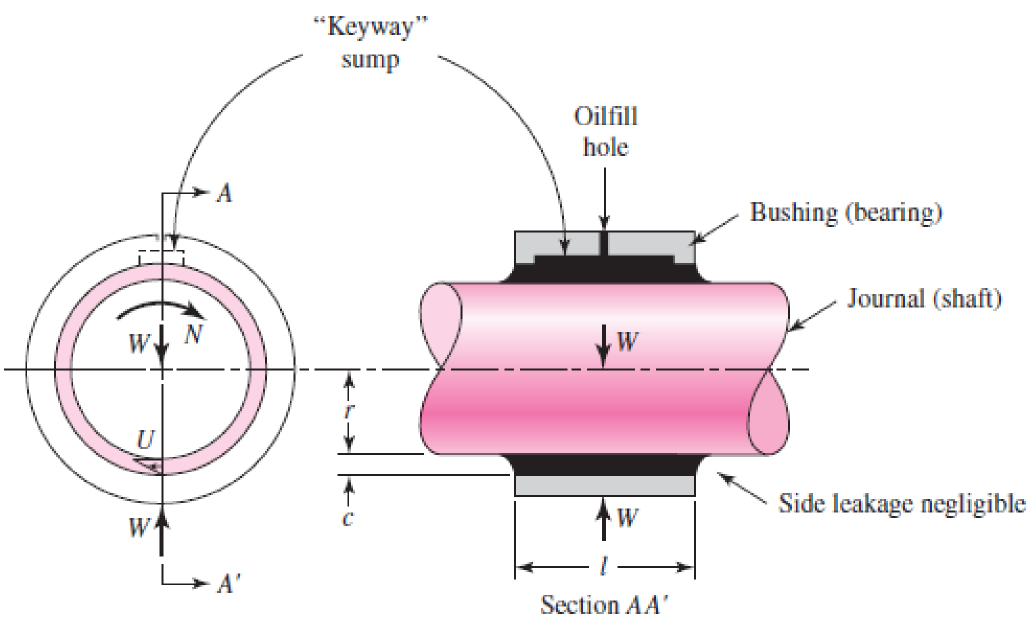
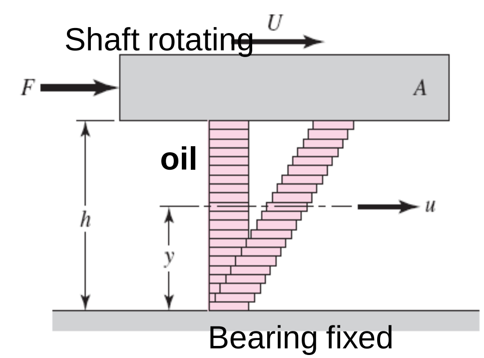


Fig. 4.3 Petroff's lightly loaded journal bearing consisting of a shaft journal and a bushing with an axial-groove internal lubricant reservoir.

The linear velocity gradient is shown in the end view. The clearance c (or h) is several thousandths of an inch and is grossly exaggerated for presentation purposes

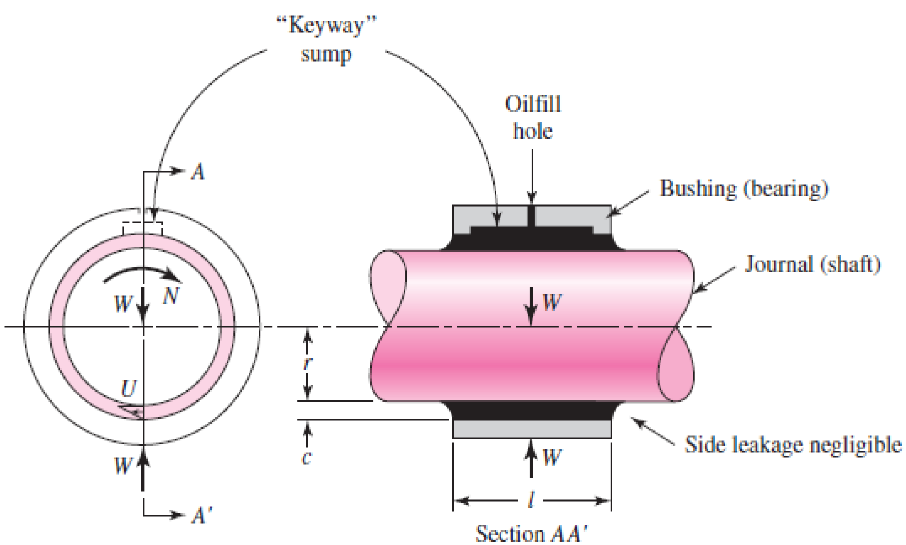


4.4 Petroff's Equation

Let us now consider a (massless) horizontal shaft rotating in a guide bearing. It is assumed that the bearing carries a very small load, that the clearance space is completely filled with oil, and that leakage is negligible

We denote the radius of the shaft by *r*, the radial clearance by *c*, and the length of the bearing by *l*, all dimensions being in meters.

If the shaft rotates at *N rev/s*, then its surface velocity is *U = πd *N in m/s*.



The shear stress in the oil

$$\tau = \frac{F}{A} = \mu\frac{U}{h} = \mu\frac{\pi dN}{c}$$

The force creating τ is $F = \tau A$

$$A = 2\pi r \times l$$

$$F = \mu\frac{\pi dN}{c} \times (\pi dl)$$

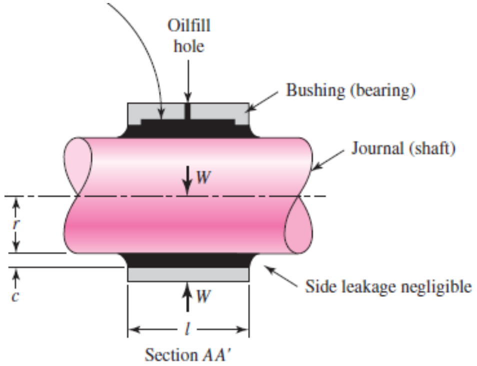
Torque creating shear force *F* (in oil) is : $T_s = F*r$

$$T_s = \left[\frac{\pi dN}{c}\mu(\pi dl)\right]*\frac{d}{2} \qquad T_s = \mu\frac{\pi^2 d^3 Nl}{2c} \quad or \quad T_s = \frac{4\pi^2 r^3 \mu Nl}{c}$$

Now If we designate a small force on the bearing by *W*,

There will be a friction force opposing the rotation $F_f = f * W$
where f is the friction coefficient in bearing.

The torque that is created by this friction force is



$$T_f = F_f \times r = (f * W) \times r$$

This is the same torque required to shear the lubricant within the bearing.

$T_s = T_f$

Thus $T_s = T_f$

$$\frac{4\pi^2 r^3 \mu N l}{c} = fWr$$

from here

$$f = \frac{4\pi^2 r^2 \mu N l}{cW} \quad or$$

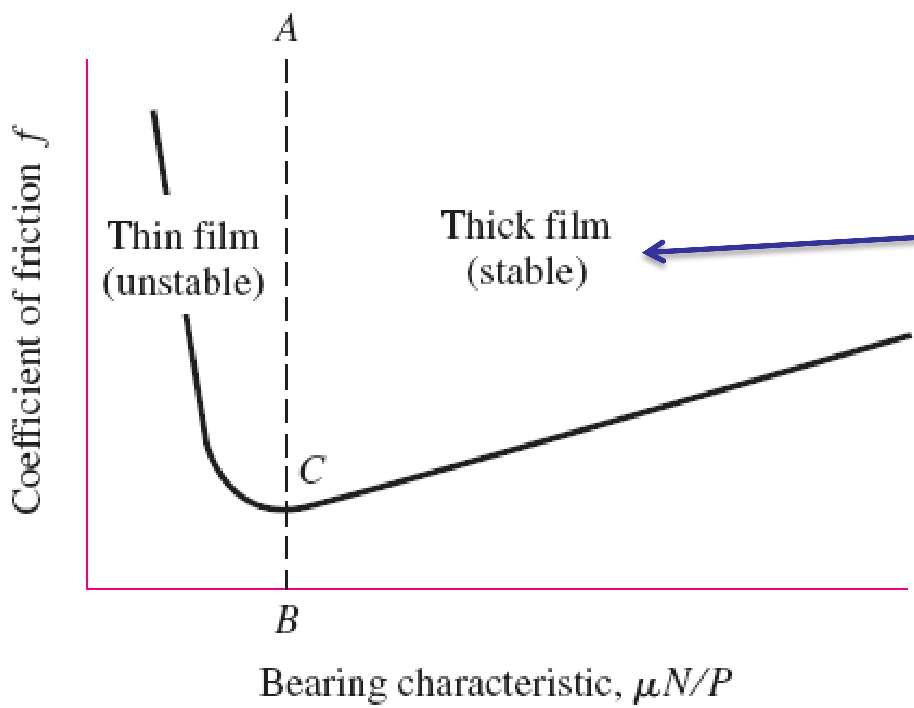
$$\cancel{f = 2\pi^2 \left(\frac{\mu N}{P}\right)\left(\frac{r}{c}\right)} \quad (4.6)$$

Petroff 's Law

where $P = \dfrac{W}{ld}$ bearing pressure (unit load)

- Equation (4–6) is called *Petroff 's equation* <u>*but this equation is not used to find the friction coefficient of the journal bearings*.</u>
- <u>*Actual friction coefficients of JB's are determined through charts given in following sections*</u>
- The two quantities μ*N*/*P* and *r*/*c* are very important parameters in lubrication. They are dimensionless hence *f* is dimensionless

$$f = 2\pi^2 \left(\frac{\mu N}{P}\right)\left(\frac{r}{c}\right)$$



This (thick film) is the section which will be studied in journal bearings.

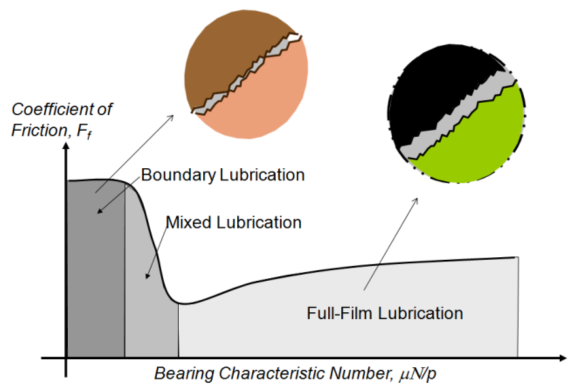


Fig. 4.4 The variation of the coefficient of friction *f with μN/P.*

The bearing characteristic number, or the **Sommerfeld number,** is defined by the equation.

$$S = \left(\frac{r}{c}\right)^2 \frac{\mu N}{P}$$

The Sommerfeld number is very important in lubrication analysis because it contains many of the parameters that are specified by the designer.

Note that it is also dimensionless.

The quantity *r/c* is called the radial clearance ratio.

4.5 Stable Lubrication

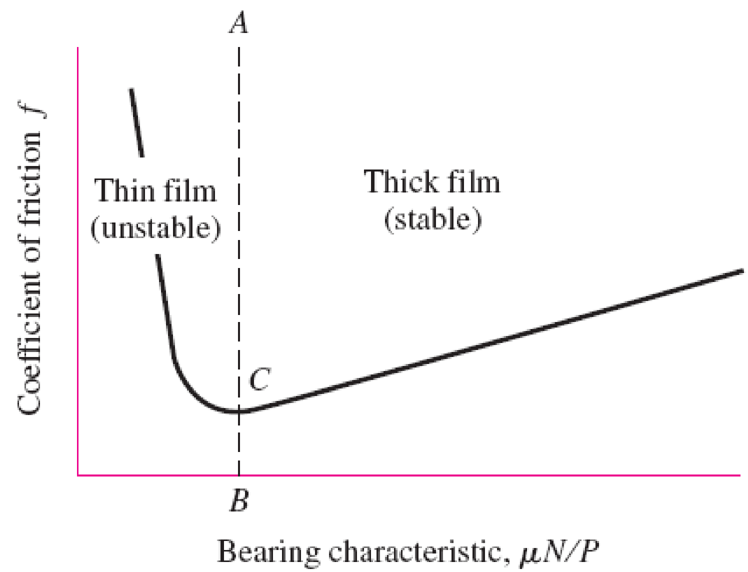
The difference between boundary and hydrodynamic lubrication can be explained by reference to Fig. 4–4.

This plot of the change in the coefficient of friction versus the bearing characteristic μ*N*/*P* is important because it defines stability of lubrication and helps us to understand **hydrodynamic (thick film) and boundary (or thin-film) lubrication**.

Suppose we are operating to the right of line *BA* and something happens, say, an increase in lubricant temperature. This results in a lower viscosity and hence a smaller value of μ*N*/*P*.

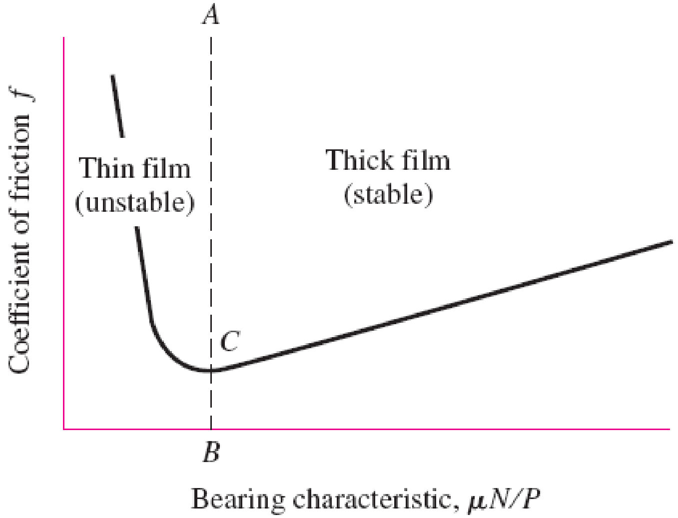
The coefficient of friction decreases, not as much heat is generated in shearing the lubricant, and consequently the lubricant temperature drops.

Thus the region to the right of line *B A* defines *stable lubrication* because variations are self-correcting



To the left of line *BA*, a decrease in viscosity would increase the friction. A temperature rise would ensue, and the viscosity would be reduced still more. The result would be compounded.

Thus the region to the left of line *BA* represents *unstable lubrication*.



It is also helpful to see that a small viscosity, and hence a small $\mu N/P$, means that the lubricant film is very thin and that there will be a greater possibility of some metal-to-metal contact, and hence of more friction. Thus, point *C* represents what is probably the beginning of metal-to-metal contact as $\mu N/P$ becomes smaller.

For a relatively large value of $\mu N/P$ *(to the right of line BA) we need:*

- *Either more viscous lubricant*
- *Or higher relative speed*
- *Or lower bearing pressure*
- *Or a combination of all together*

4.6 Thick-Film Lubrication

Let us now examine the formation of a lubricant film in a journal bearing. Figure 4–5*a* shows a journal that is just beginning to rotate in a clockwise direction. Under starting conditions, the bearing will be dry, or at least partly dry, and hence the journal will climb or roll up the right side of the bearing as shown in Fig. 4–5*a*.

Now suppose a lubricant is introduced into the top of the bearing as shown in Fig. 4–5*b. The action of the rotating journal is to pump the lubricant around the bearing* in a clockwise direction. The lubricant is pumped into a wedge-shaped space and forces the journal over to the other side. A *minimum film thickness h_0 occurs, not at the* bottom of the journal, but displaced clockwise from the bottom as in Fig. 4–5*b. This is* explained by the fact that a film pressure in the converging half of the film reaches a maximum somewhere to the left of the bearing center.

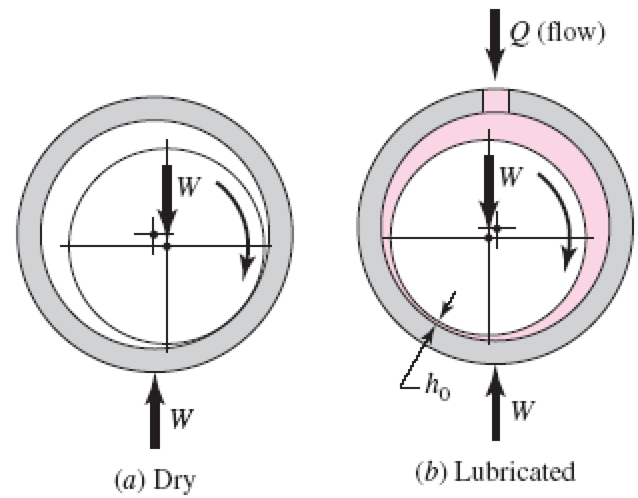


Fig. 4.5 Formation of a film.

4.6 Thick-Film Lubrication

Figure 4–5 shows how to decide whether the journal, under hydrodynamic lubrication, is eccentrically located on the right or on the left side of the bearing.

Visualize the journal beginning to rotate. Find the side of the bearing upon which the journal tends to roll. Then, if the lubrication is hydrodynamic, mentally place the journal on the opposite side.

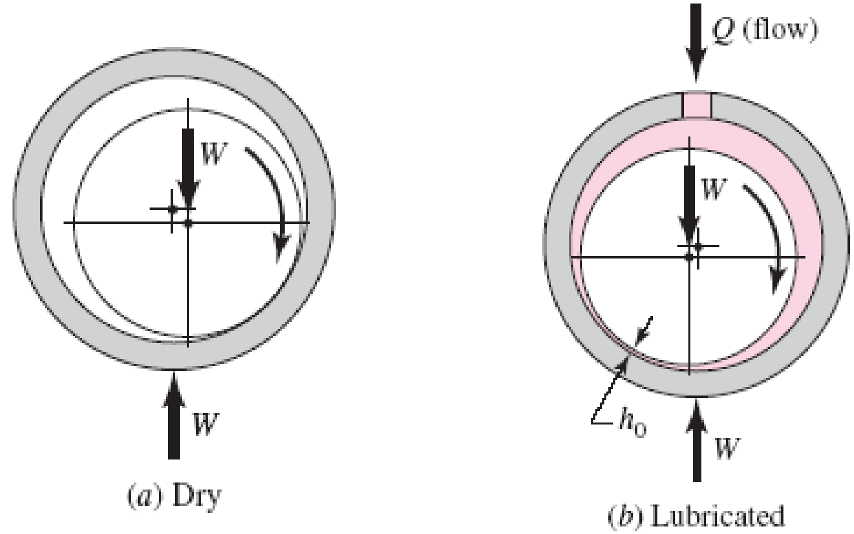
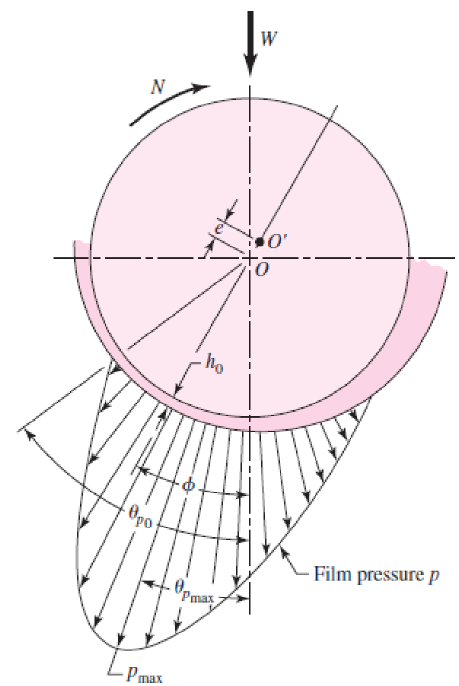


Fig. 4.5 Formation of a film.



4.6 Thick-Film Lubrication

The nomenclature of a journal bearing is shown in Fig. 4–6. The dimension c is the *radial clearance* and is the difference in the radii of the bushing and journal.

In Fig. 4–6 the center of the journal is at O and the center of the bearing at O'. The distance between these centers is the *eccentricity* and is denoted by e.

The *minimum film thickness* is designated by h_0, and it occurs at the line of centers. The film thickness at any other point is designated by h. We also define an *eccentricity ratio* ε as

$$\varepsilon = \frac{e}{c}$$

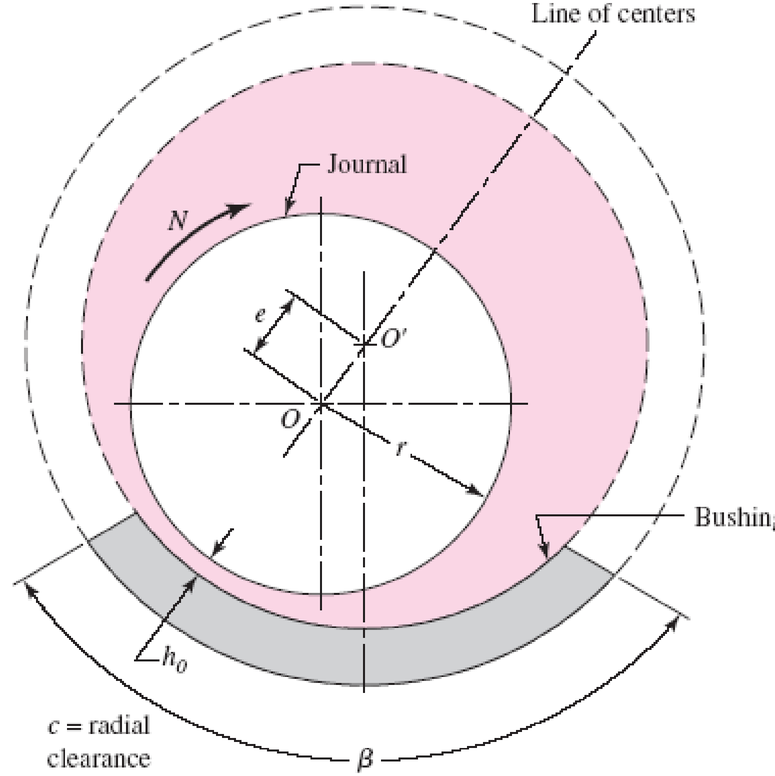


Fig. 4.6 Nomenclature of a partial journal bearing.

4.6 Thick-Film Lubrication

The bearing shown in the figure is known as a *partial bearing.* If the radius of the bushing is the same as the radius of the journal, it is known as a *fitted bearing*.

If the bushing encloses the journal, as indicated by the dashed lines, it becomes a *full bearing.*

The angle β describes the angular length of a partial bearing. For example, a 120° partial bearing has the angle β equal to 120° .

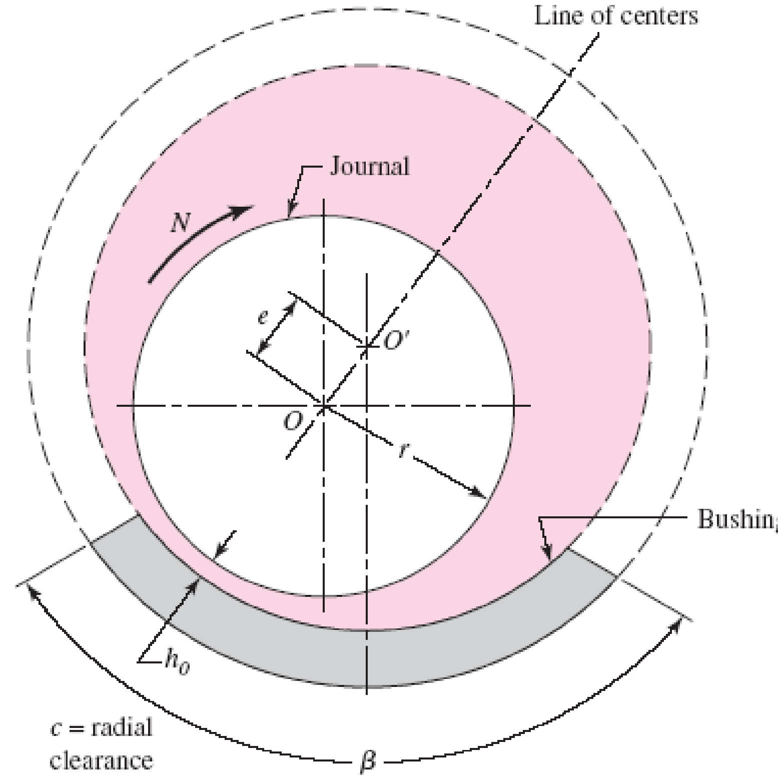


Fig. 4.6 Nomenclature of a partial journal bearing.

4.7 Hydrodynamic Theory

The present theory of hydrodynamic lubrication originated in the laboratory of Beauchamp Tower in the early 1880s in England. Tower had been employed to study the friction in railroad journal bearings and learn the best methods of lubricating them.

Figure 4–7 is a schematic drawing of the journal bearing that Tower investigated. It is a partial bearing, having a diameter of 4 in, a length of 6 in, and a bearing arc of 157°, and having bath-type lubrication, as shown. The coefficients of friction obtained by Tower in his investigations on this bearing were quite low, which is now not surprising.

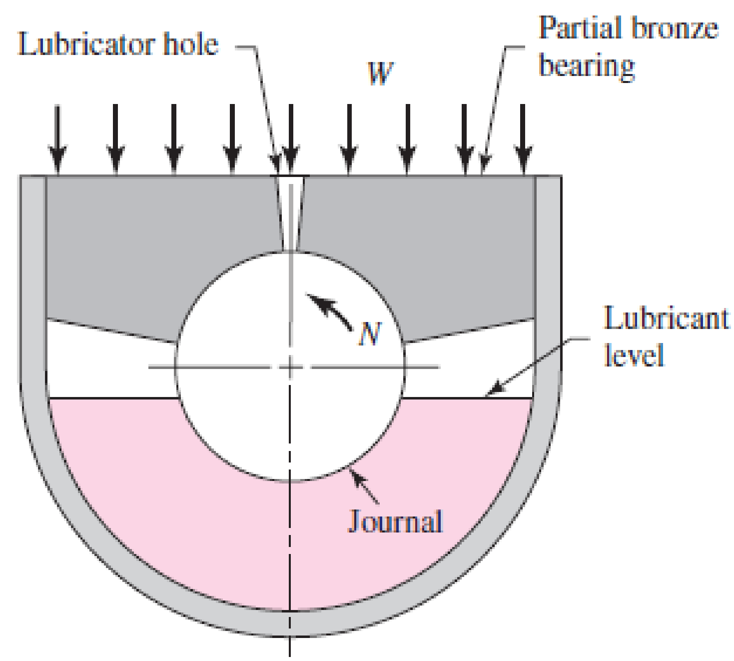
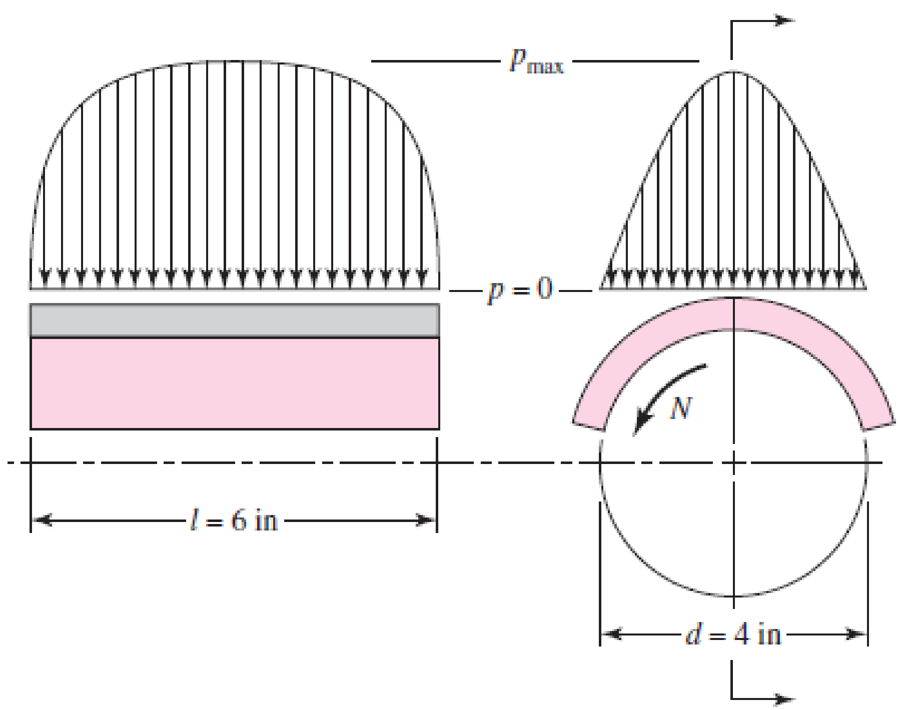


Fig. 4.7 Schematic representation of the partial bearing used by Tower.

4.7 Hydrodynamic Theory

After testing this bearing, Tower later drilled a 1/2 -in-diameter lubricator hole through the top. But when the apparatus was set in motion, oil flowed out of this hole. In an effort to prevent this, a cork stopper was used, but this popped out, and so it was necessary to drive a wooden plug into the hole. When the wooden plug was pushed out too, Tower, at this point, undoubtedly realized that he was on the verge of discovery.



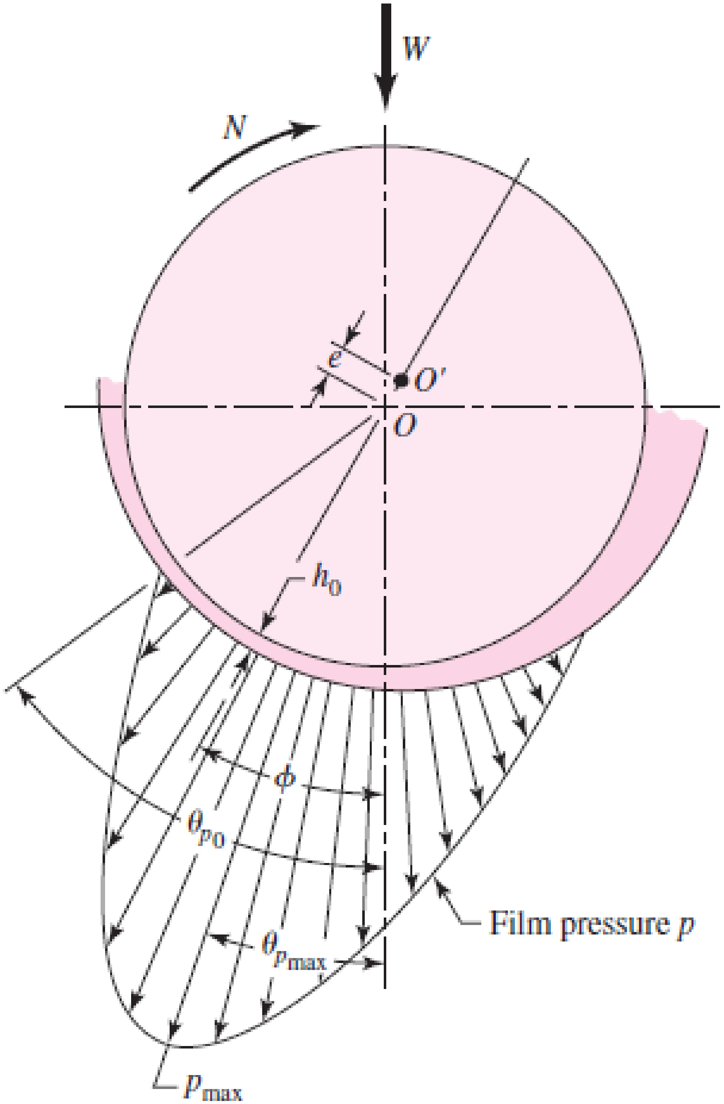
A pressure gauge connected to the hole indicated a pressure of more than twice the unit bearing load. Finally, he investigated the bearing film pressures in detail throughout the bearing width and length and reported a distribution similar to that of Fig. 4–8.

Fig. 4.8 Approximate pressure distribution curves obtained by Tower.

4.7 Hydrodynamic Theory

The results obtained by Tower had such regularity that Osborne Reynolds concluded that there must be a definite equation relating the friction, the pressure, and the velocity. The present mathematical theory of lubrication is based upon Reynolds' work following the experiment by Tower. The original differential equation, developed by Reynolds, was used by him to explain Tower's results. The solution is a challenging problem that has interested many investigators ever since then, and it is still the starting point for lubrication studies.

Reynolds pictured the lubricant as adhering to both surfaces and being pulled by the moving surface into a narrowing, wedge-shaped space so as to create a fluid or film pressure of sufficient intensity to support the bearing load. One of the important simplifying assumptions resulted from Reynolds' realization that the fluid films were so thin in comparison with the bearing radius that the curvature could be neglected. This enabled him to replace the curved partial bearing with a flat bearing, called a *plane slider bearing.*



h_o is the minimum film thickness during operation of the bearing at a speed of *N (rps),*

e is the eccentricity of the shaft relative to bearing's center during operation,

P_{max} *is the maximum pressure value in the bearing surface,*

Θ_{pmax} *shows the angular position of* P_{max} *in reference to shaft (journal) center line.*

P_o *is zero pressure at outlet side of* the bearing

Θ_{Po} *is the angular position zero pressure,*

P is the pressure at minimum film position,

Θ_p *is the angular position minimum film.*

Fig. 4.9 Polar diagram of the film–pressure distribution showing the notation used.

4.8 Design Considerations of JB's

a) given variables

We may distinguish between two groups of variables in the design of sliding bearings. In the first group are those whose values either are **given** or are **under the control of the designer**. These are:

1) ***μ*** **: viscosity**
2) ***P :*** **load per unit of projected bearing area, *W/ld***
3) ***N ,*** **speed,**
4) ***r, c, β, and l :*** **bearing dimensions**

Of these four variables, the designer usually has no control over the speed, because it is specified by the overall design of the machine. Sometimes the viscosity is specified in advance, as, for example, when the oil is stored in a sump and is used for lubricating and cooling a variety of bearings. The remaining variables, and sometimes the viscosity, may be controlled by the designer and are therefore the *decisions the designer makes.* In other words, when these four decisions are made, the design is complete.

4.8 Design Considerations of JB's

b) dependent variables

In the second group are the **dependent variables**. The designer cannot control these except indirectly by changing one or more of the first group. These are:

1) ***f* :** **coefficient of friction**
2) **Δ*T*:** **temperature rise**
3) ***Q*:** **volume flow rate of oil**
4) ***h_0*** **: minimum oil film thickness**

Should be minimized

Should be maximized

This group of variables tells us how well the bearing is performing, and hence we may regard them as *performance factors*. Certain limitations on their values must be imposed by the designer to ensure satisfactory performance. These limitations are specified by the characteristics of the bearing materials and of the lubricant. The fundamental problem in bearing design, therefore, is to define satisfactory limits for the second group of variables and then to decide upon values for the first group such that these limitations are not exceeded.

For example: minimizing *f* value and keeping h_0 in certain limits.

4.9 The Relations of the Variables in full JB's

Before proceeding to the problem of design, it is necessary to establish the relationships between the variables.

Albert A. Raimondi and John Boyd, of Westinghouse Research Laboratories, used an iteration technique to solve Reynolds' equation on the digital computer. This is the first time such extensive data have been available for use by designers, and consequently we shall employ them in this course

The relations hereafter and the design procedures are given for the full (β= 360º) bearing.

4.9 The Relations of the Variables in full JB's

The Raimondi and Boyd papers were published in three parts and contain 45 detailed charts and 6 tables of numerical information. In all three parts, charts are used to define the variables for length-diameter (*l/d) ratios of 1:4, 1:2, and 1 and for beta* angles of 60 to 360º. Under certain conditions the solution to the Reynolds equation gives negative pressures in the diverging portion of the oil film. Since a lubricant cannot usually support a tensile stress, Part III of the Raimondi-Boyd papers assumes that the oil film is ruptured when the film pressure becomes zero. Part III also contains data for the infinitely long bearing; since it has no ends, this means that there is no side leakage.

The charts appearing in this book are from Part III of the papers, and are for full journal bearings (β = 360º) only. Space does not permit the inclusion of charts for partial bearings. This means that you must refer to the charts in the original papers when beta angles of less than 360º are desired.

The bearing characteristic number, or the **Sommerfeld number,** is defined by the equation.

$$S = \left(\frac{r}{c}\right)^2 \frac{\mu N}{P}$$

The Sommerfeld number is very important in lubrication analysis because it contains many of the parameters that are specified by the designer.

Note that it is also dimensionless.

The quantity *r/c* is called the radial clearance ratio.

Sommerfeld number is the main parameter to determine the other design and performance parameters of JB such as :

- Minimum film thickness
- Friction coefficient
- Temperature rise
- Oil film thickness
- Oil flow rate
- Oil film pressure
- Power lost (due to friction)
- etc

In $$S = \left(\frac{r}{c}\right)^2 \frac{\mu N}{P}$$

The normal or recommended values of P and c values are given in some tables and figures to start with for a design.

FIGURE 12-27. Recommended radial clearances for cast-bronze bearings. The curves are identified as follows:

A—precision spindles made of hardened ground steel, running on lapped cast-bronze bearings (0.2- to 0.8-μm rms finish), with a surface velocity less than 3 m/s

B—precision spindles made of hardened ground steel, running on lapped cast-bronze bearings (0.2- to 0.4-μm rms finish), with a surface velocity more than 3 m/s

C—electric motors, generators, and similar types of machinery using ground journals in broached or reamed cast-bronze bearings (0.4- to 0.8-μm rms finish)

D—general machinery which continuously rotates or reciprocates and uses turned or cold-rolled steel journals in bored and reamed cast-bronze bearings (0.8- to 1.6-μm rms finish)

E—rough-service machinery having turned or cold-rolled steel journals operating on cast-bronze bearings (1.6- to 3.2-μm rms finish)

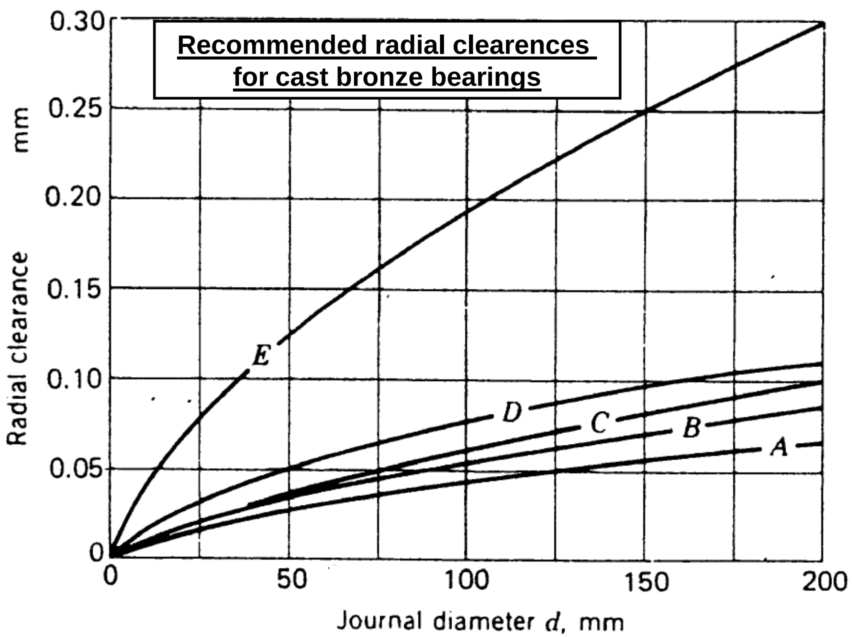


FIGURE 12-27. Recommended radial clearances for cast-bronze bearings.

<u>"c" is determined</u> in fig. 12.27 for "cast-bronze" bearings, and (r/c) range is recommended for different bearings in table 12-4

Table 12-4 SOME CHARACTERISTICS OF BEARING ALLOYS

Alloy name	Thickness, mm	Clearance ratio r/c	Load capacity	Corrosion resistance
Tin-base babbit	0.559	600–1000	1.0	Excellent
Lead-base babbit	0.559	600–1000	1.2	Very good
Tin-base babbit	0.102	600–1000	1.5	Excellent
Leaded bronze	Solid	500–1000	3.3	Very good
Copper-lead	0.559	500–1000	1.9	Good
Aluminum alloy	Solid	400–500	3.0	Excellent
Silver plus overlay	0.330	600–1000	4.1	Excellent
Cadmium (1.5% Ni)	0.559	400–500	1.3	Good

Table 12-3 RECOMMENDED UNIT LOADS FOR SLEEVE BEARINGS

Application	Unit load, kPa	Application	Unit load, kPa
Air compressors:		Diesel engines:	
Main bearings	1 000– 2 000	Main bearings	6 000–12 000
Crankpin	2 000– 4 000	Crankpin	8 000–15 000
Automotive engines:		Wristpin	14 000–15 000
Main bearings	4 000– 5 000	Electric motors	800– 1 500
Crankpin	10 000–15 000	Gear reducers	800– 1 500
Centrifugal pumps	600– 1 200	Steam turbines	800– 1 500

Table 12-3, based on $P=W/ld$, l and d are determined in designs.

Another recommended maximum value $(PV)_{max}$ for different bearing materials is given in Table 12.5 for boundary lubricated bearings which is the case <u>when hydrodynamically lubricated bearings are starting or stopping or journal is not rotating in one direction but reciprocating forward and backward.</u>

Table 12-5 SOME MATERIALS FOR BOUNDARY-LUBRICATED BEARINGS AND THEIR OPERATING LIMITS

Material	Maximum pressure, MPa	Maximum temperature, °C	Maximum speed, m/s	Maximum *PV* value, kPa·m/s
Cast bronze	31.0	165	7.5	1750
Porous bronze	31.0	65	7.5	1750
Porous iron	55.0	65	4.0	1750
Phenolics	41.0	95	13.0	530
Nylon	7.0	95	5.0	100
Teflon	3.5	260	0.5	35
Reinforced teflon	17.0	260	5.0	350
Teflon fabric	410	260	0.3	900
Delrin	7.0	80	5.0	100
Carbon-graphite	4.2	400	13.0	530
Rubber	0.4	65	20.0	...
Wood	14.0	65	10.0	530

$$S = \left(\frac{r}{c}\right)^2 \frac{\mu N}{P}$$

μ value in S is the absolute viscosity <u>at an average temperature</u> of

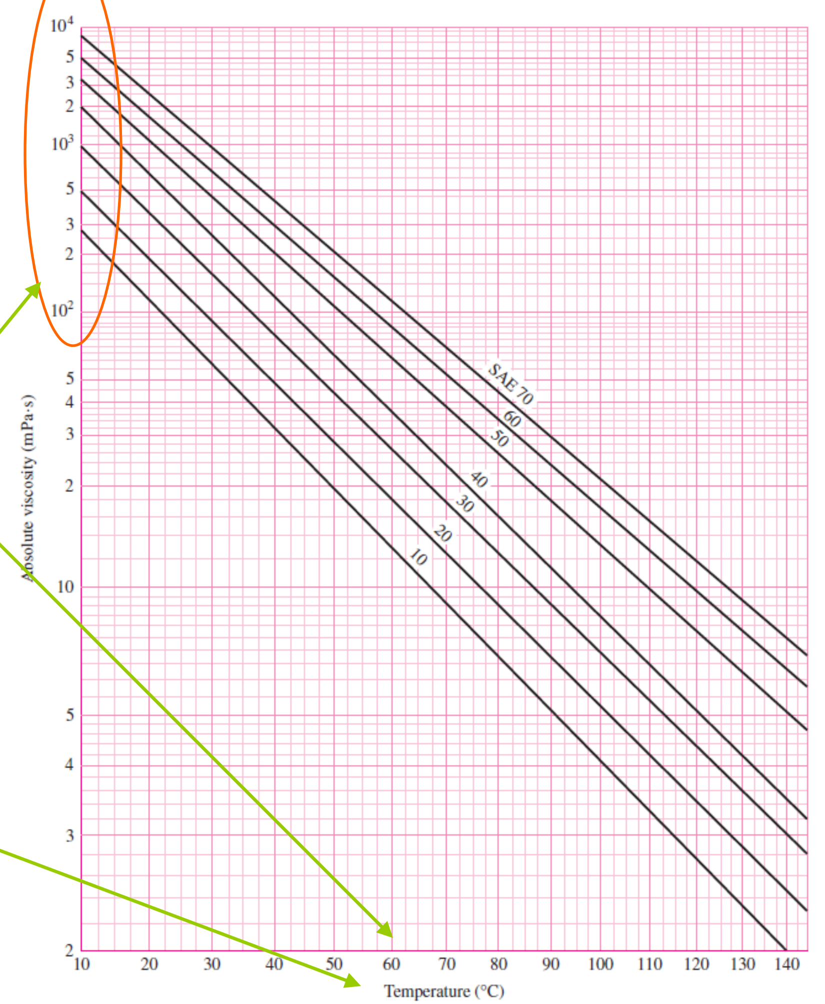
$$T_{ave} = \frac{T_{in} + T_{out}}{2} \quad or \quad T_{ave} = T_{in} + \frac{\Delta T}{2}$$

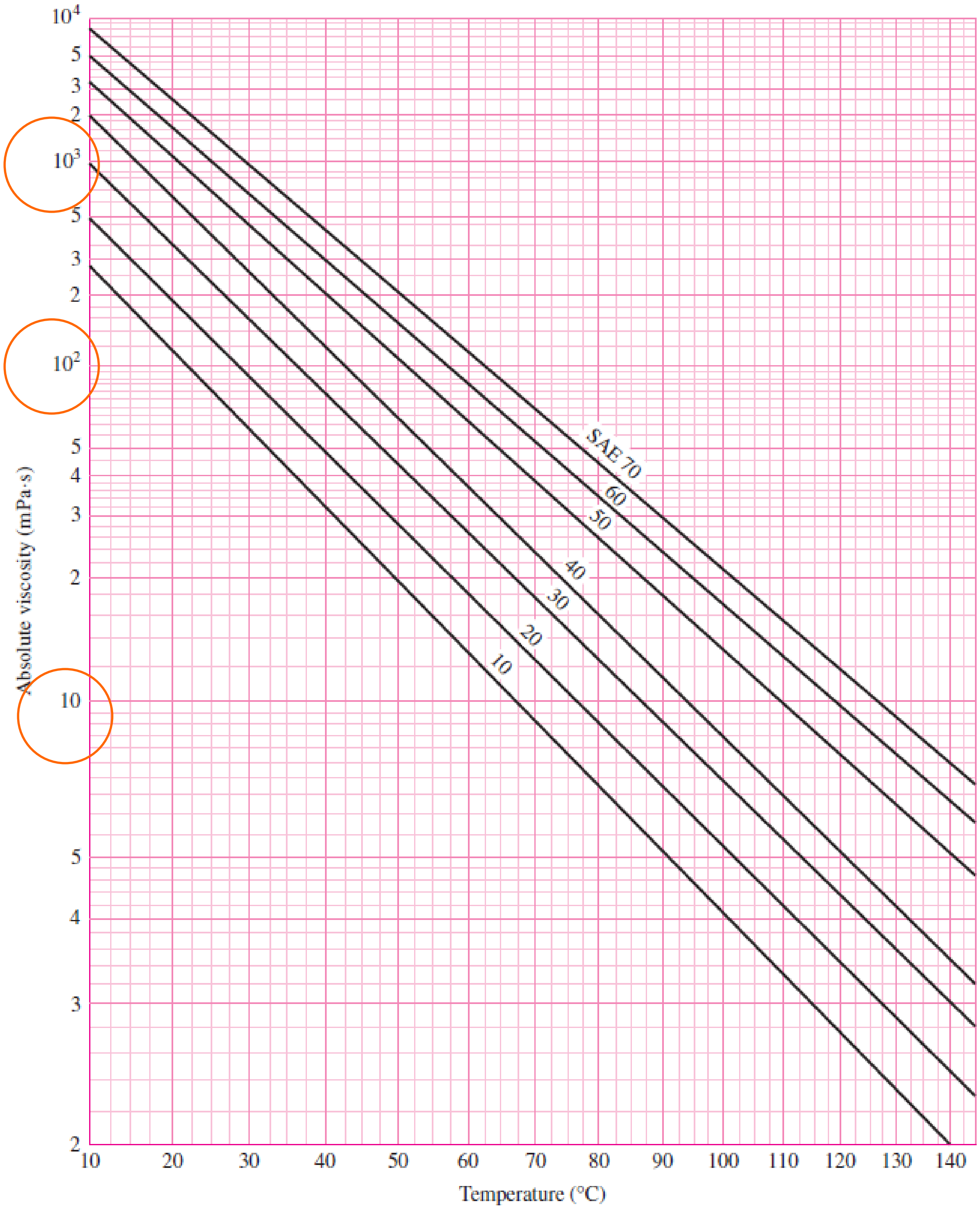
not at T_i or T_{out} but at T_{ave}

Absolute viscosity of different lubricants is determined by using following figures.

Horizontal axis is the "<u>average" temperature of the lubricant</u>.

Note the scale of vertical axis.





NOTE: vertical axis is logarithmic

Horizontal axis is the “average” temperature of the lubricant

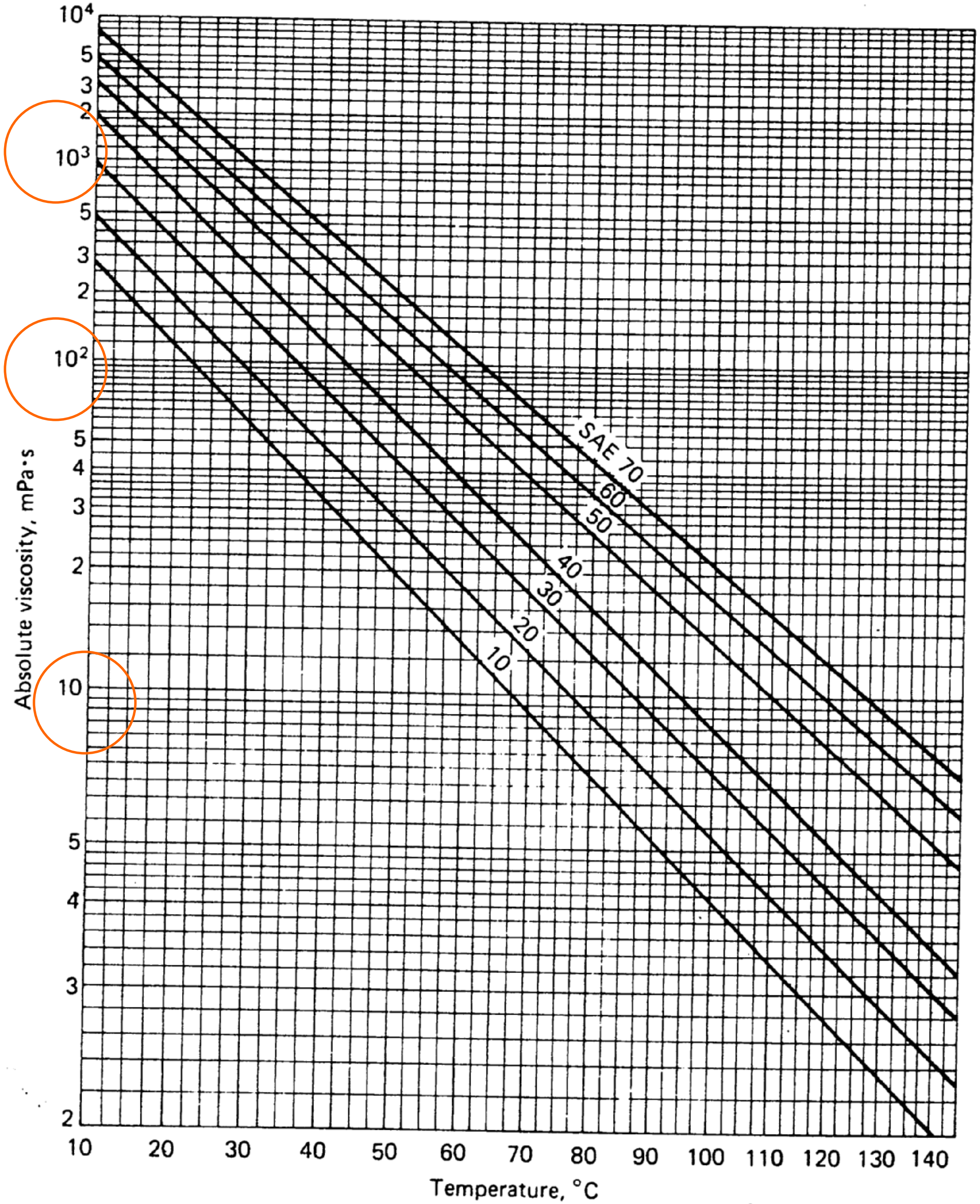


FIGURE 12-10 Viscosity-temperature chart.

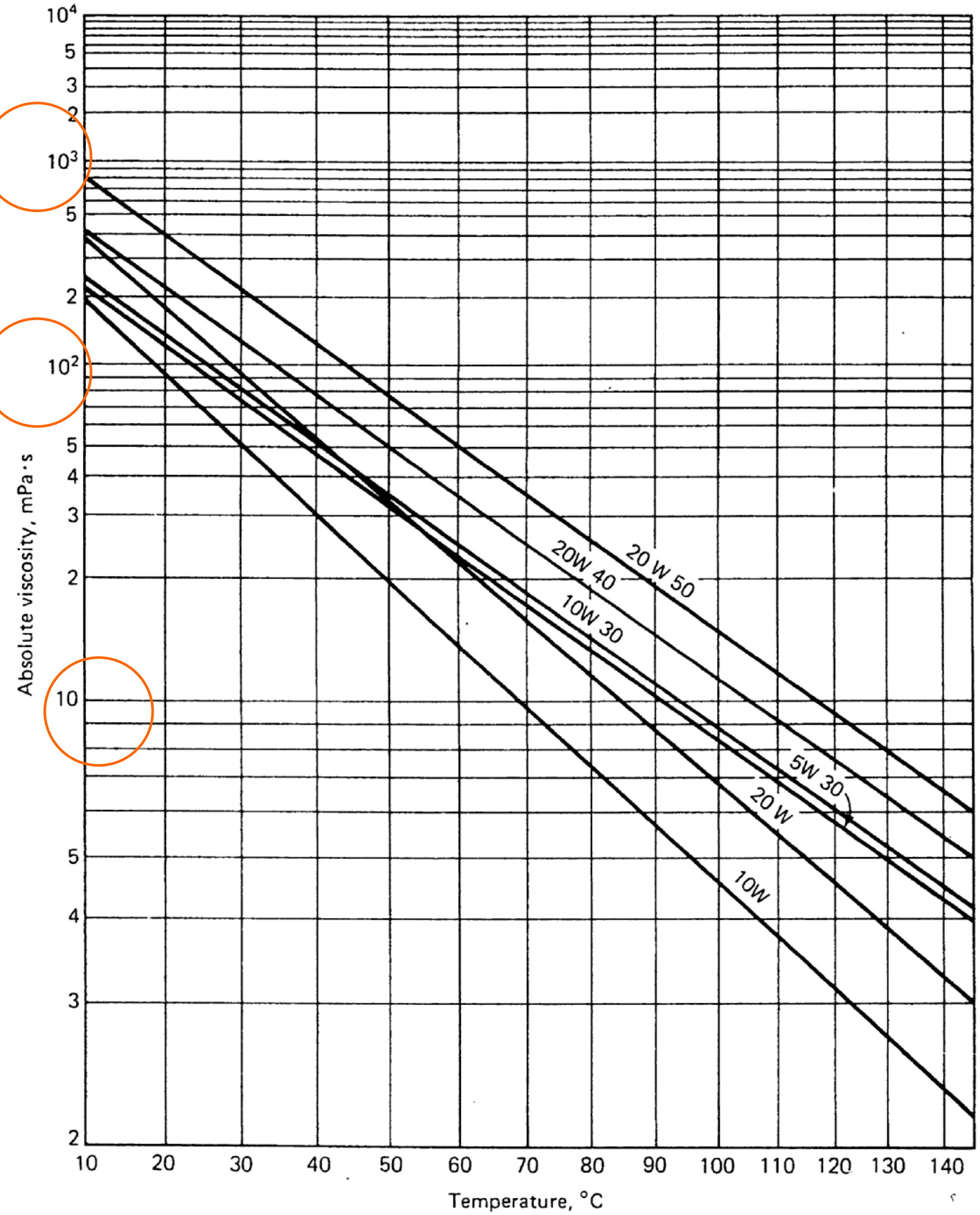


FIGURE 12-11 Viscosity-temperature chart for multiviscosity lubricants. This chart was derived from known viscosities at two points, 40 and 100°C, and the results are believed to be correct for other temperatures.

Table 12-3 RECOMMENDED UNIT LOADS FOR SLEEVE BEARINGS

Application	Unit load, kPa	Application	Unit load, kPa
Air compressors:		Diesel engines:	
Main bearings	1 000– 2 000	Main bearings	6 000–12 000
Crankpin	2 000– 4 000	Crankpin	8 000–15 000
Automotive engines:		Wristpin	14 000–15 000
Main bearings	4 000– 5 000	Electric motors	800– 1 500
Crankpin	10 000–15 000	Gear reducers	800– 1 500
Centrifugal pumps	600– 1 200	Steam turbines	800– 1 500

Table 12-4 SOME CHARACTERISTICS OF BEARING ALLOYS

Alloy name	Thickness, mm	Clearance ratio r/c	Load capacity	Corrosion resistance
Tin-base babbit	0.559	600–1000	1.0	Excellent
Lead-base babbit	0.559	600–1000	1.2	Very good
Tin-base babbit	0.102	600–1000	1.5	Excellent
Leaded bronze	Solid	500–1000	3.3	Very good
Copper-lead	0.559	500–1000	1.9	Good
Aluminum alloy	Solid	400–500	3.0	Excellent
Silver plus overlay	0.330	600–1000	4.1	Excellent
Cadmium (1.5% Ni)	0.559	400–500	1.3	Good

FIGURE 12-27. Recommended radial clearances for cast-bronze bearings. The curves are identified as follows:

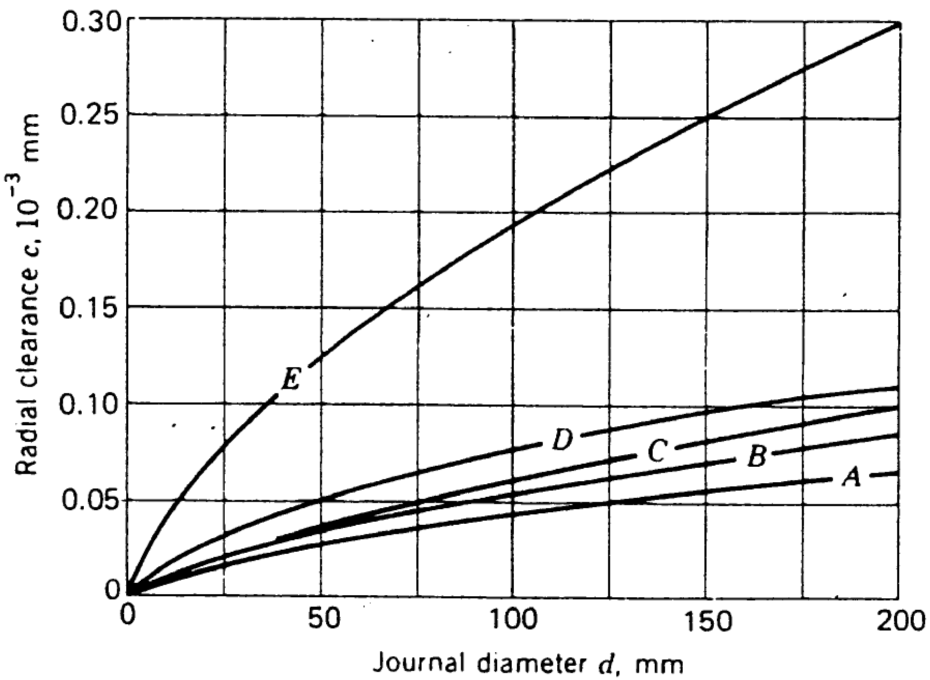
A—precision spindles made of hardened ground steel, running on lapped cast-bronze bearings (0.2- to 0.8-μm rms finish), with a surface velocity less than 3 m/s

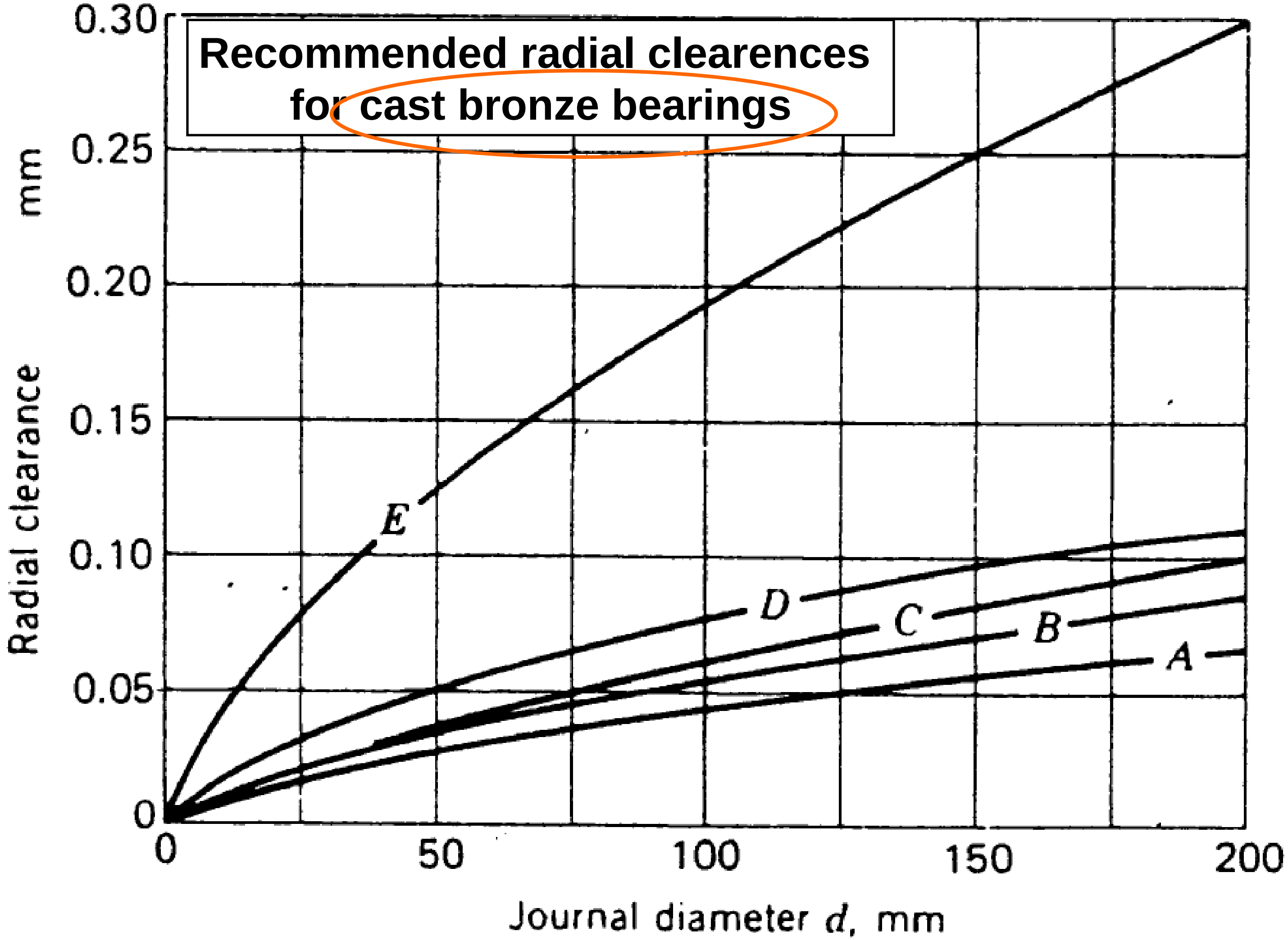
B—precision spindles made of hardened ground steel, running on lapped cast-bronze bearings (0.2- to 0.4-μm rms finish), with a surface velocity more than 3 m/s

C—electric motors, generators, and similar types of machinery using ground journals in broached or reamed cast-bronze bearings (0.4- to 0.8-μm rms finish)

D—general machinery which continuously rotates or reciprocates and uses turned or cold-rolled steel journals in bored and reamed cast-bronze bearings (0.8- to 1.6-μm rms finish)

E—rough-service machinery having turned or cold-rolled steel journals operating on cast-bronze bearings (1.6- to 3.2-μm rms finish)



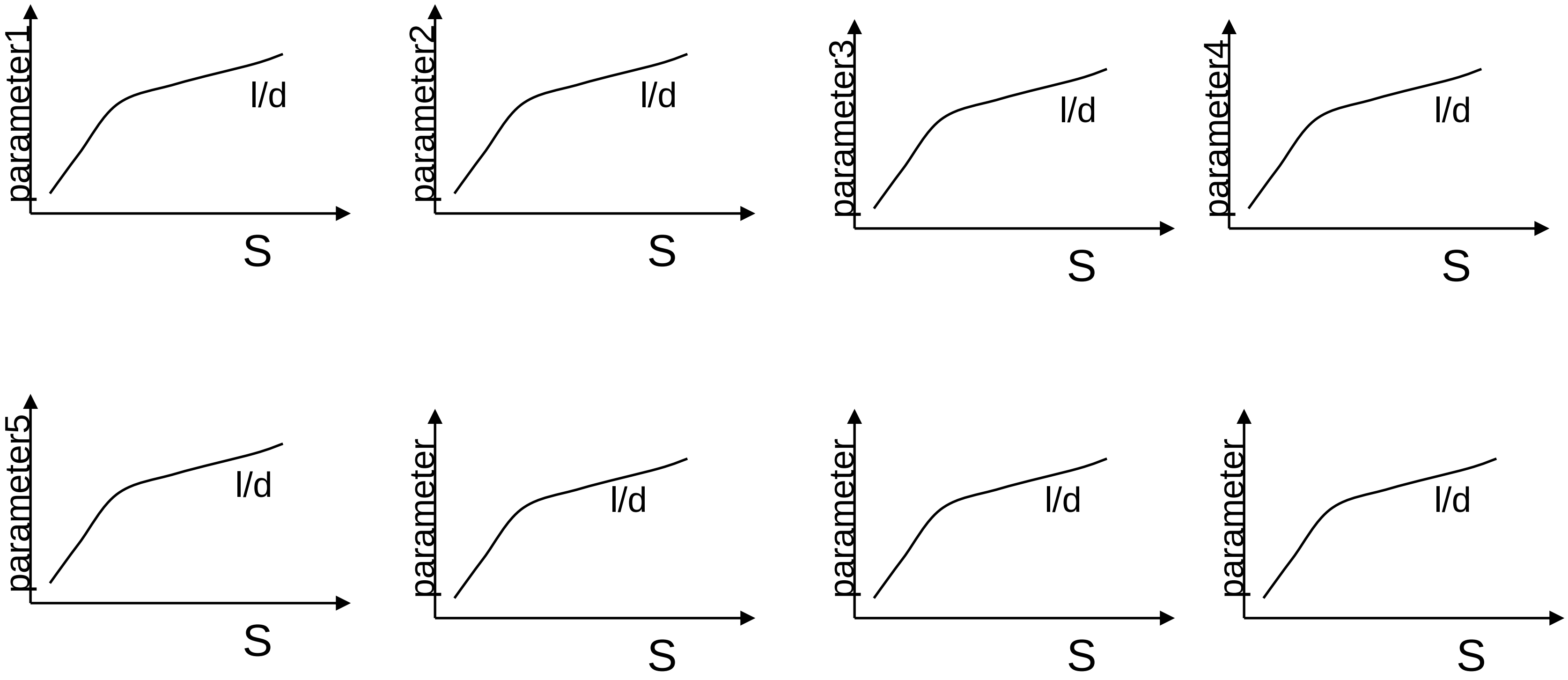

Recommended radial clearences
for cast bronze bearings
Radial clearance mm
0.30
0.25
0.20
0.15
0.10
0.05
0
E
D
C
B
A
0
50
100
150
200
Journal diameter d, mm

- For a JB usually some parameters such as bearing dimensions, bearing load, oil type and oil inlet temp are known before the run.
- Whereas some other parameters such as oil temp increase, friction coefficient, oil flow rate, min film thickness, oil film max pressure etc are not known before the run.
- Since most of these parameters are the performance indicators they should somehow be determined before the run to verify the correct design of JB.
- For example temp increase of oil is given by eqn.

$$\Delta T_{^oC} = \frac{8.30P}{\left[1 - \frac{1}{2} \times \frac{Q_s}{Q}\right]} \times \frac{\frac{r}{c} f}{\frac{Q}{rcNl}}$$

In this eqn. There are lots of parameters already unknown

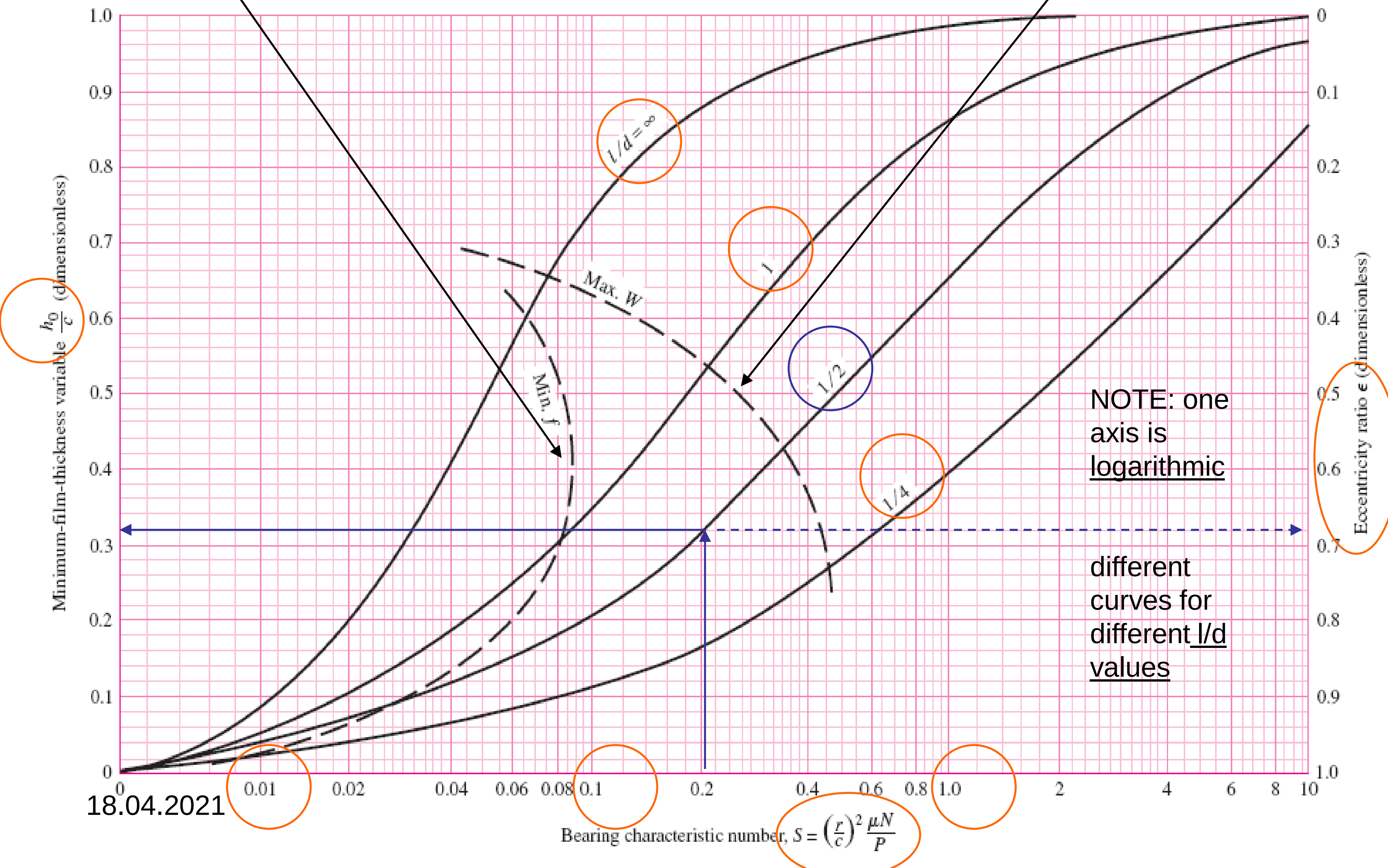
Luckily Riamond&Boyd have already studied the subject to have some relations/charts between the factor “S” and other parameters through different diagrams and curves as seen below.



Once “S” value and “l/d” ratio are known, then all other parameters (1,2,3,etc) can be determined through charts/curves.

Let S=0.2 and l/d=1/2 and then try determining other parameters from charts

This is the chart for <u>minimum film-thickness variable</u> and <u>eccentricity ratio</u>. The left boundary(hidden line) of the zone defines the optimal h_0 *for* ***minimum friction****;* *the right boundary is optimum* h_0 *for* ***max. Load capacity****.*



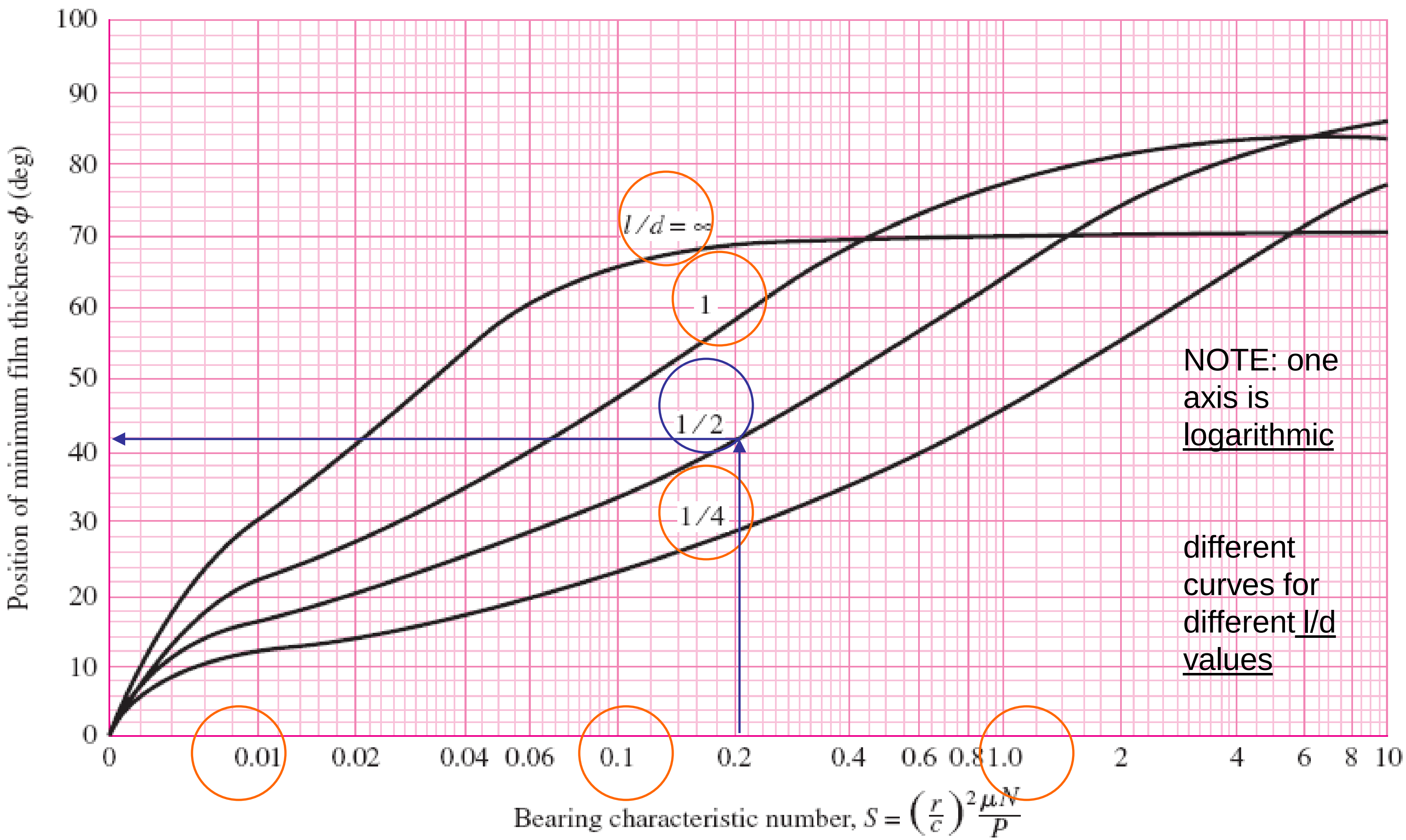
Once ho/c is read from chart (0.32)

then “ho” is calculated as ho=0.32 * c

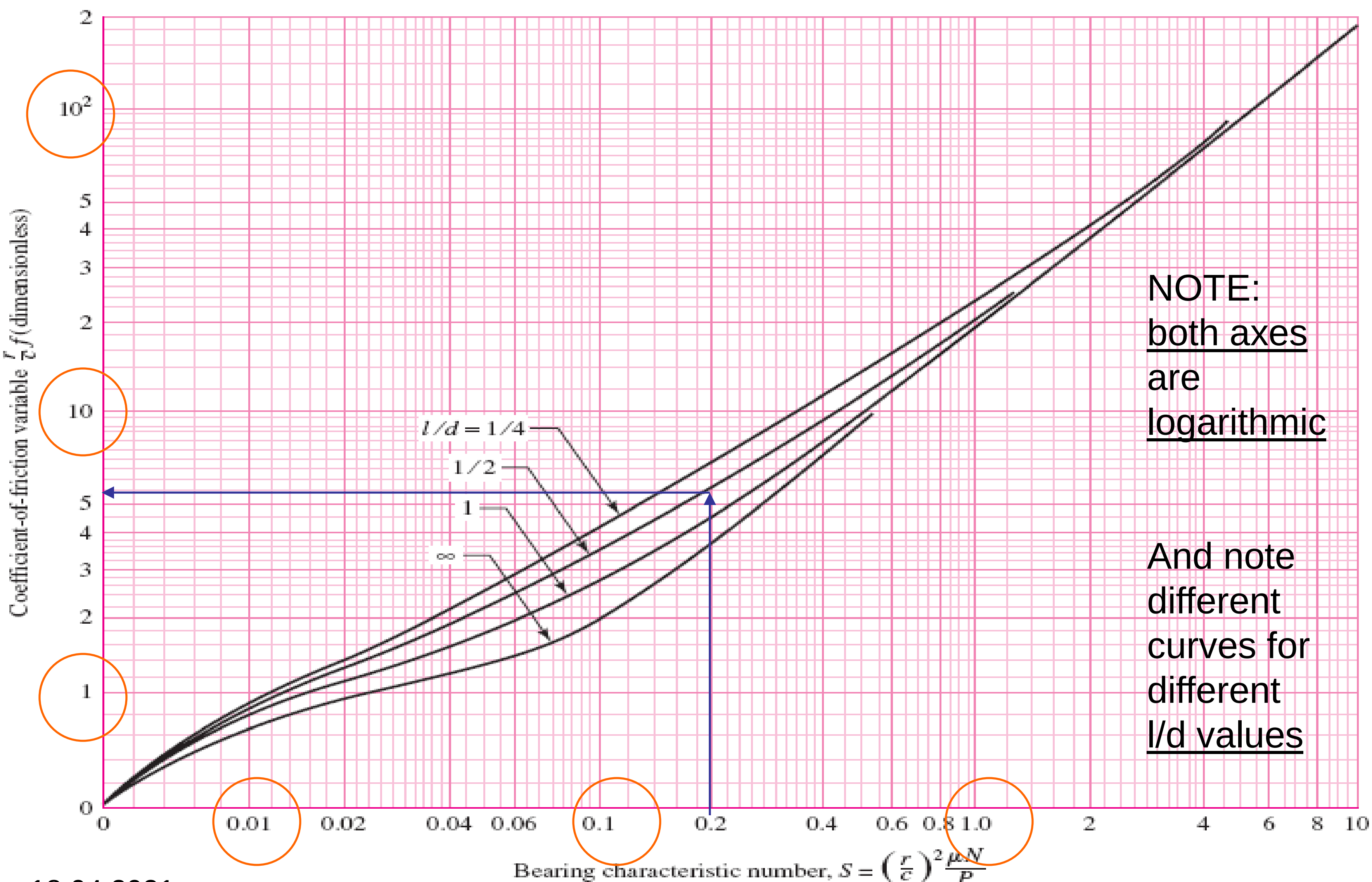
e/c=0.68 is also read an the right hand side,

Thus e=0.68*c

This is the chart for determining the position of the minimum film thickness h_0



This is the chart for coefficient-of-friction variable;



NOTE:
<u>both axes</u>
are
<u>logarithmic</u>

And note
different
curves for
different
<u>l/d values</u>

Once r*f/c is read from chart (5.5)

then “f” is calculated as f=5.5 / (r/c)

And then other parameters calculated as

$$F_{fric} = W \times f$$

$$T_{loss} = F_{fric} \times r$$

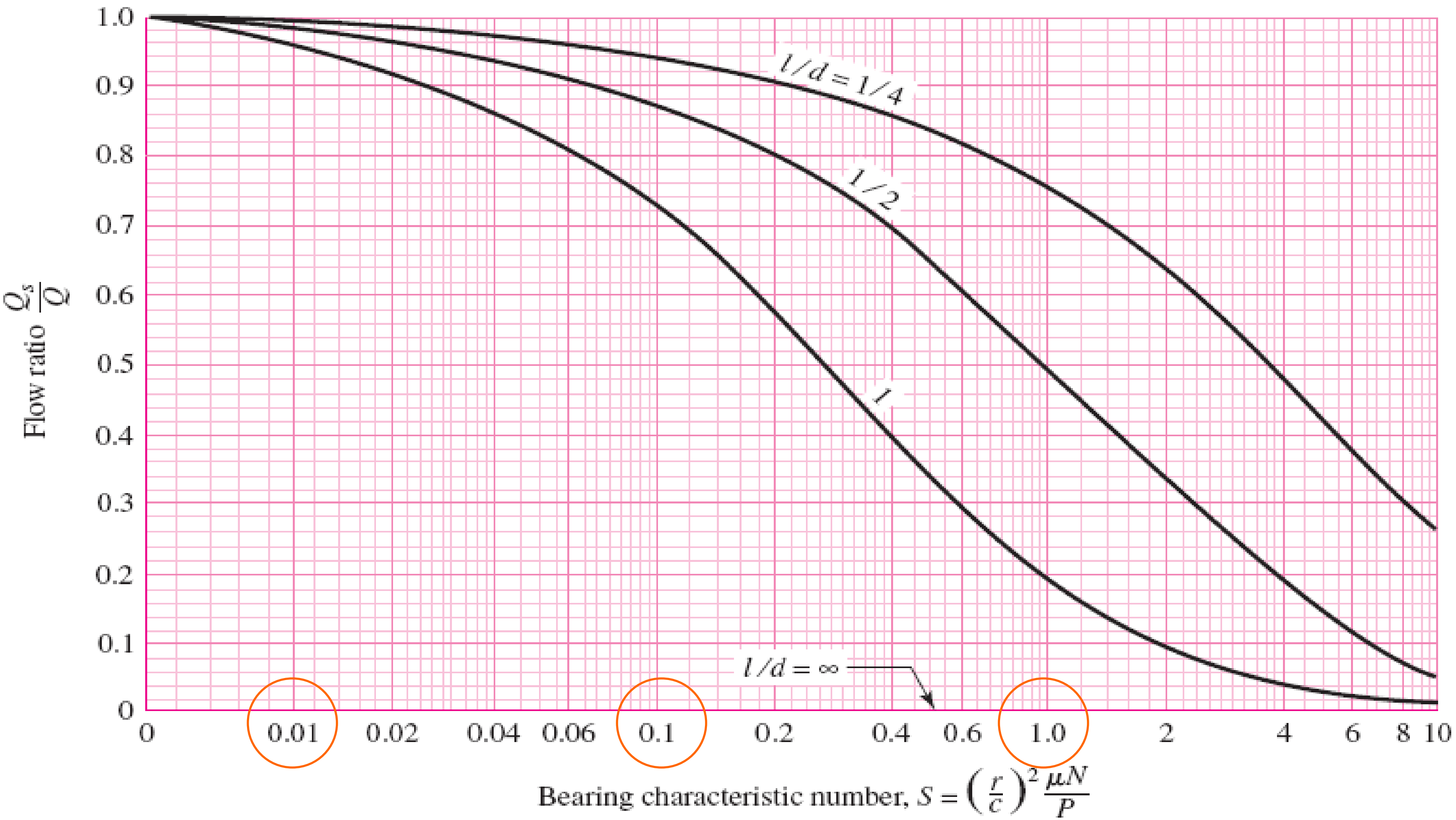
$$P_{loss} = T_{loss} \times w \frac{rad}{\sec}$$

This is the chart for flow variable.

Note: Not to be used for pressure-fed bearings.

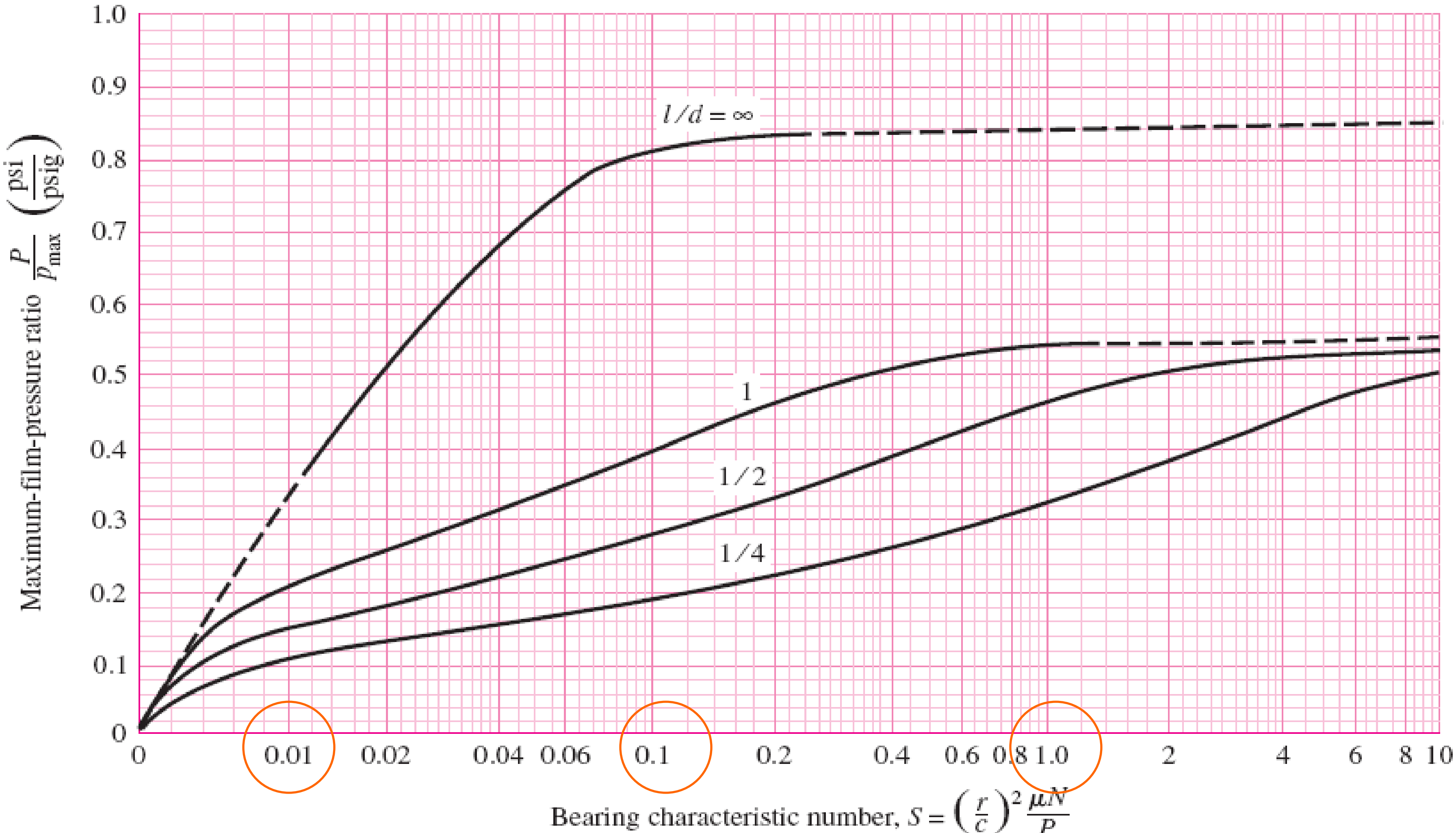
This is the chart for determining the ratio of side flow to total flow.

Note: Not to be used for pressure-fed bearings.

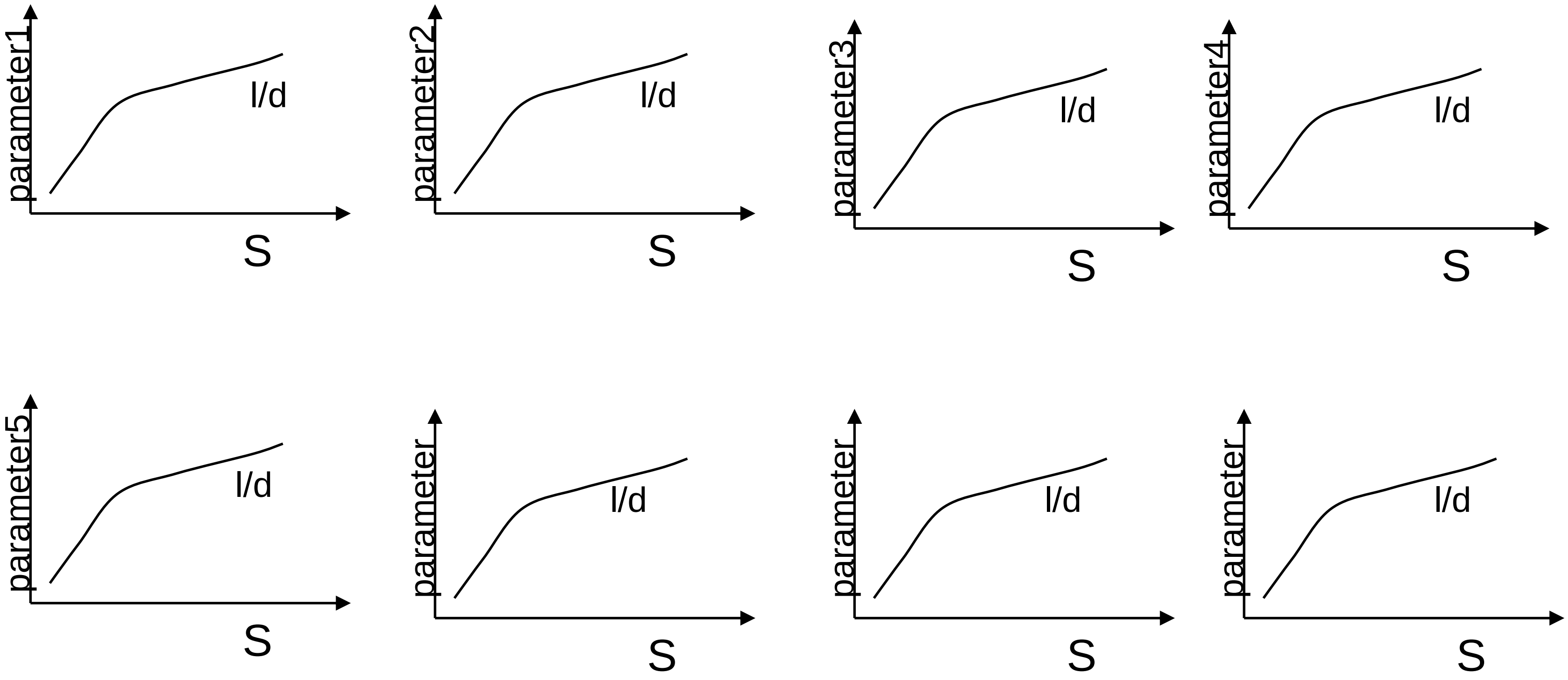


This is the chart for determining the maximum film pressure.

Note: Not to be used for pressure-fed bearings**.**



This is the chart for finding the terminating position of the lubricant film and the position of maximum film pressure



Once “S” value and “l/d” ratio are known, then all other parameters (1,2,3,etc) can be determined through charts/curves.

Almost all parameters required in following eqn. can be determined from previous charts.

$$\Delta T_{^oC} = \frac{8.30P}{\left[1 - \frac{1}{2} \times \frac{Q_s}{Q}\right]} \times \frac{\frac{r}{c} f}{\frac{Q}{rcNl}}$$

Thus temp. increase can be determined <u>as long as “S” value and “l/d” ratios are known.</u>

Temp. increase could also be determined through another eqn.by using another chart (fig12-12in following page) as before.

One of the objectives of lubrication analysis is to determine the oil outlet temperature when the oil and its inlet temperature are specified. This is a trial-and-error type of problem.

The dimensionless temperature-rise variable is

$$P = \frac{W}{l \times d}$$

$$T(\text{var}) = \frac{\gamma C_H \, \Delta T}{P} \qquad \text{or} \qquad \Delta T_{o_C} = T_{\text{var}} \frac{P}{\gamma \; C_H} \qquad (12\text{-}14)$$

where γ is the mass per unit volume of the lubricant. At an average specific gravity of 0.86, $\gamma = 861 \text{ kg/m}^3$. We shall use this value in our calculations.

The term C_H is the specific heat of the lubricant. An average value for general use is 1760 J/kg °C.

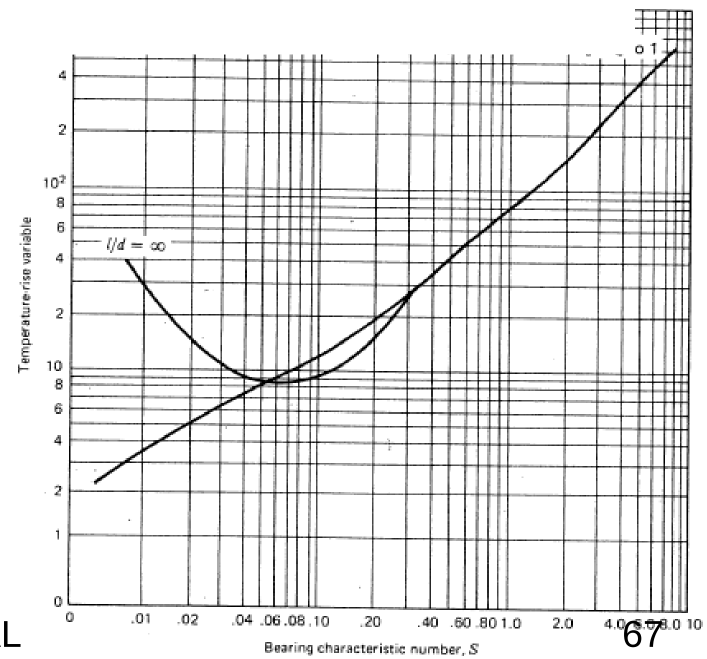


FIGURE 12-12 Chart for temperature-rise variable $T(\text{var}) = \gamma C_H \, \Delta T/P$. In plotting this chart it was found that the curves for $l/d = \frac{1}{4}, \frac{1}{2}$, and 1 were so close together that they could not be distinguished from a single curve.

T_{var}=20

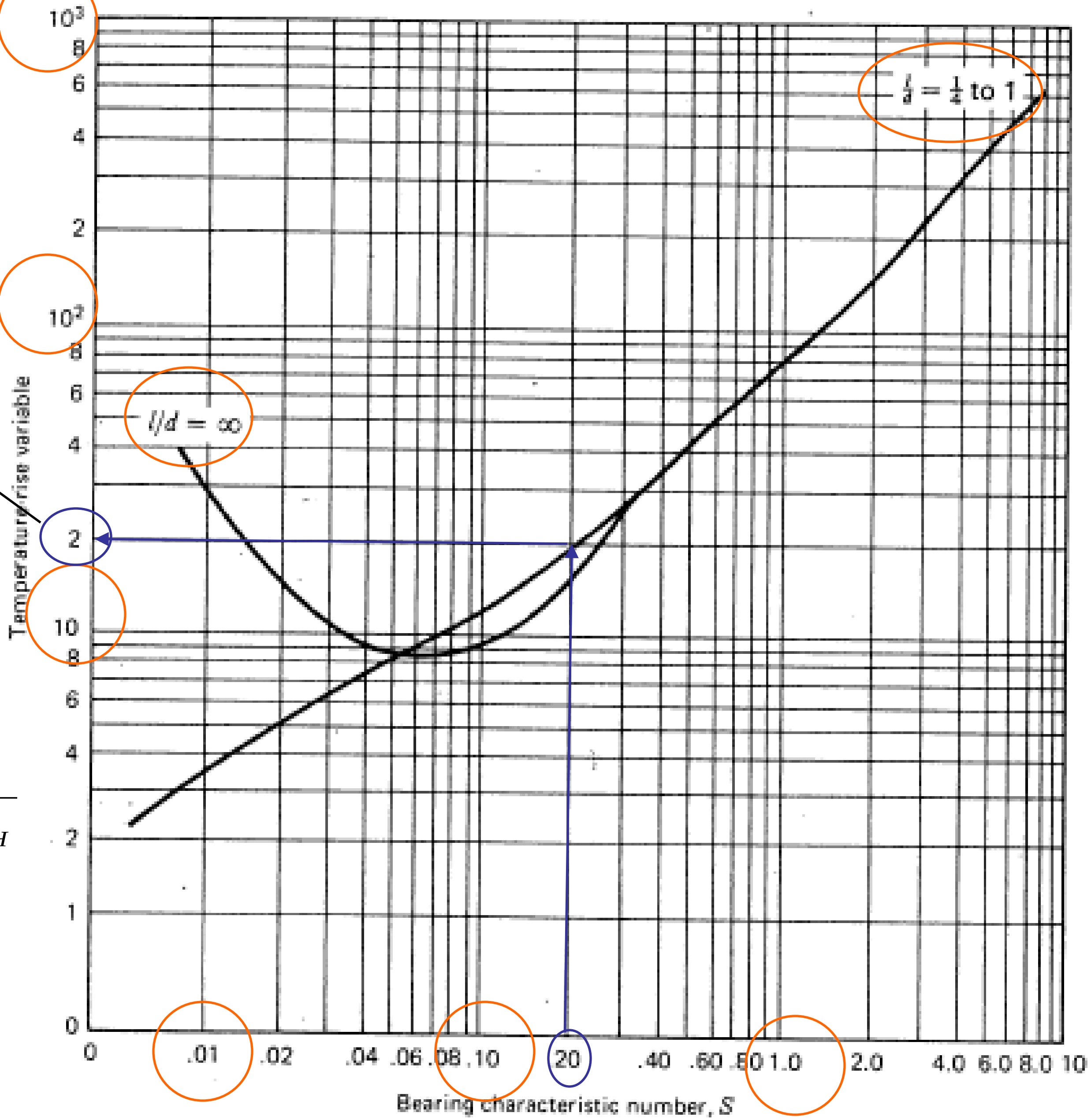


FIGURE 12-12 Chart for temperature-rise variable $T(\text{var}) = \gamma C_H\, \Delta T/P$. In plotting this chart it was found that the curves for $l/d = \frac{1}{4}, \frac{1}{2}$, and 1 were so close together that they could not be distinguished from a single curve.

$$\Delta T_{o_C} = T_{\text{var}} \frac{P}{\gamma\ C_H}$$

NOTE: both axes are <u>logarithmic</u>

And note different curves for different <u>l/d values</u>

For S=0.2 and l/d= ½ : $\longrightarrow$ Tvar=20 is read from chart

$$\Delta T_{o_C} = T_{\mathrm{var}} \frac{P}{\gamma\ C_H}$$

$$\Delta T_{o_C} = 20 * \frac{P}{\gamma\ C_H}$$

$$\Delta T_{o_C} = \frac{8.30P}{\left[1-\frac{1}{2}\times\frac{Q_s}{Q}\right]} \times \frac{\frac{\frac{r}{c}f}{Q}}{rcNl}$$

or

$$\Delta T_{o_C} = T_{\mathrm{var}} \frac{P}{\gamma \; C_H}$$

$$S = \left(\frac{r}{c}\right)^2 \frac{\mu N}{P}$$

$$T_{ave} = T_{in} + \frac{\Delta T}{2}$$

Ir-regardless of which equation is used to calculate temp increase *ΔT* we need to know the value of parameter 'S' to determine other parameters from charts like T_{var} etc.

Neverthless parameter 'S' is already dependent on the parameter *μ and μ is determined at average temp Tave which is also dependent on both Ti and ΔT.*

We see that determining ΔT requires knowing *ΔT.* !!!!! (needs a trial-and-error method as in other machine elements like springs & RCB's)