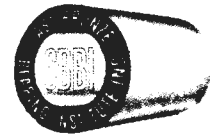


# CAST BRONZE BEARING DESIGN MANUAL

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# **Cast Bronze Bearing Design Manual**

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## PREFACE

PROPERLY DESIGNED and lubricated cast bronze sleeve bearings offer operating and wear performance second to none. But sleeve bearings have always suffered from lack of fast, simple design techniques.

Recognizing this need, a group of producers of cast bronze sleeve bearings recently formed the Cast Bronze Bearing Institute which undertook, as its first venture, the preparation of a Design Manual. The Friction and Lubrication Laboratory of The Franklin Institute Laboratories was engaged to accomplish this task.

Design and application of 360-degree bronze bearings is fully covered in this Design Manual for full-film, boundary, and mixed-film lubrication. New design methods and numerous graphs reduce actual calculations to a minimum for quick determination of bearing configurations.

Detailed explanations tell how to predict load-carrying capacity, power requirements, lubricant requirements, stability, and operating temperatures. Bearing clearances are recommended for various classes of machinery and shaft sizes. Special problems caused by shock loads and heavy loads at slow speeds are analyzed. The Manual also presents additional information on lubricating fluids, including a method for calculating viscosity. Methods are described for applying lubricant. Determining shape and dimensions of grooves in bearing surfaces is illustrated for each type of lubrication. Properties of cast bronze bearing materials are discussed.

Material contained in this Design Manual has been taken from many different sources. Almost all of it has been reworked to suit the particular needs of the Manual. Hence, CBBI is deeply indebted to the works of many experimenters and writers in the field of sleeve bearing design, analysis, and application. Published material most heavily drawn from is tabulated in the accompanying bibliography.

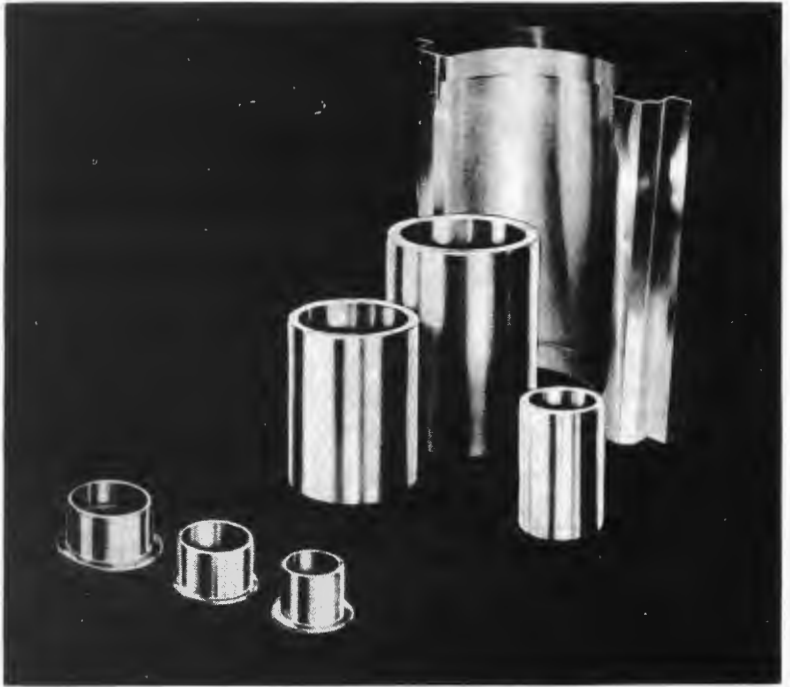
Participating members of the staff of the Friction and Lubrication Laboratory of The Franklin Institute Laboratories in this work included: J. G. Hinkle, A. M. Loeb, S. B. Malanoski, R. R. Pandolfi, S. A. Richardson, G. M. Robinson, W. W. Shugarts, C. H. Stevenson and, in particular, D. D. Fuller, Principal Scientist, under whose guidance the work was accomplished. Greatest credit, however, belongs to Harry C. Rippel, Senior Staff Engineer, Franklin Institute, whose conscientious and untiring efforts in preparing this Manual have brought it into reality. Mr. Rippel's simple, reliable, and straightforward approach will, it is believed, make this Manual the necessary text of every sleeve bearing designer and lubrication engineer in industry.

This Manual first appeared as a series of articles in *Machine Design*, a Penton publication. It is a second edition; others will follow as planned additional research and development bring to light new design data and information.\*

Comments concerning this Manual, and any other phase of cast bronze sleeve bearing design, are welcomed by CBBI and should be addressed to Chairman for Research and Development at the address below.

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\*See page 2 for list of all other CBBI Publications.



# Introduction to Cast Bronze

## Sleeve Bearings

**S**TATED simply, the designer's problem from the standpoint of bearings is: Provide suitable bearings to support the moving system in a trouble-free manner when operating at the given speeds and subjected to the given loads.

A typical "moving system" is shown schematically in Fig. 1. The system is to be supported by "suitable bearings." Type and speed of motion is usually dictated, as are the types and magnitudes of loads. In addition, diameter of the shaft in the vicinity of the bearings is also normally specified to meet certain strength or stiffness considerations within the system. Knowing speed, load, and size requirements, the designer can proceed to design trouble-free bearings. "Trouble-free" usually means long service life, low friction, and minimum maintenance.

At this stage in the development of suitable bearings, the present-day designer selects rather than designs such bearings. Actual design of bearings, such as the bronze sleeve bearing, is becoming a lost art. Most bronze bearings being designed today have, as the basis for their design, bearings used in previous, similar applications which proved successful. The experienced bronze-bearing designer relies heavily upon his past experience. This practice is both good and bad—good in that a workable de-

sign can be arrived at in a short time, and bad in that what's best in one case may not be best for a similar application.

The prime purpose of this Manual is to present sufficient engineering data to permit design and performance prediction of bronze sleeve bearings. On this basis, a bearing design can be accepted or rejected for use in a particular application.

### ► Description of a Sleeve Bearing

Sleeve bearings are quite probably the most used machine element in our civilization. They are of many sizes, shapes, and configurations, but they consist essentially of a band or sleeve of close-fitting material that encloses and supports a moving member. Fig. 2 shows a typical bronze sleeve bearing and the member being supported. Usually, the sleeve is stationary and is called the bearing. The moving member is usually referred to as the journal. Other names for sleeve bearings are journal bearings, since they support journals, and radial bearings, since all loads supported are in the radial direction or perpendicular to the axis of the journal.

Important physical dimensions of the sleeve bearing in Fig. 2 are journal diameter  $D$ , bearing length

#### Nomenclature

$C$	= Radial clearance, in.
$D$	= Journal diameter, in.
$F$	= Bearing friction force, lb
$f$	= Coefficient of friction = $F/W$
$L$	= Bearing length, in.
$N$	= Rotational speed of journal, rpm
$p$	= Projected area unit load, psi = $W/LD$
$W$	= Steady load to be supported, lb
$W_m$	= Dead weight of rotating parts, lb
$W_s$	= Shock load, lb
$Z$	= Lubricant absolute viscosity, centipoise

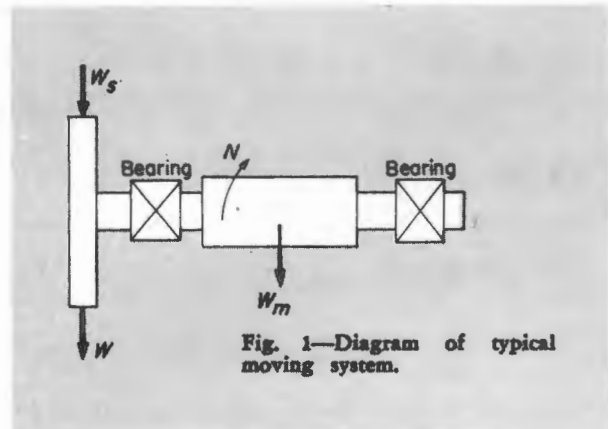


Fig. 1—Diagram of typical moving system.

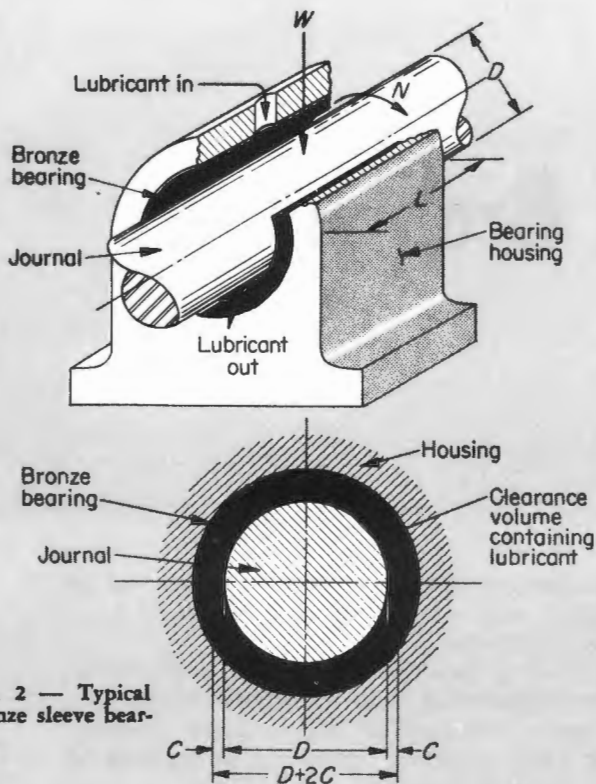


Fig. 2 — Typical bronze sleeve bearing.

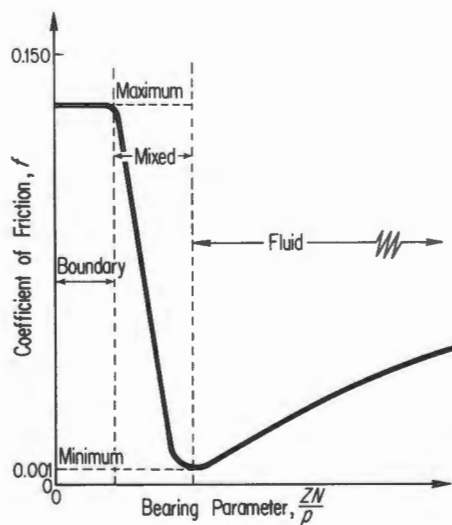


Fig. 3—Variation of coefficient of friction with  $ZN/p$  for bronze sleeve bearings.

$L$ , and radial clearance  $C$ . Of these three dimensions usually only one, the journal diameter, may be specified beforehand. The other two must be determined before the bearing can be made. Two useful and meaningful terms in sleeve-bearing work are:

$$\frac{2C}{D} = \text{Clearance ratio} \quad (1)$$

$$\frac{L}{D} = \text{Length-to-diameter ratio}$$

Lubricant of some sort is usually introduced to the bearing and eventually leaves at the ends of

the bearing. Consequently, lubricant must be continuously or periodically replaced. Some functions of the lubricant in a bearing are:

1. Reducing friction.
2. Removing some of the heat generated.
3. Minimizing wear of the rubbing parts.

Properties of the lubricant of most concern are absolute viscosity and ability to cling to bearing members.

## ► Types of Sleeve-Bearing Operation

There are three types or realms of sleeve-bearing operation. Proper classification and definition is important since the design and operating characteristics are different for each. The three types are:

1. Full-film or hydrodynamic lubrication. (Hydrostatic lubrication is also full-film.)
2. Complete boundary lubrication.
3. Mixed-film lubrication.

*Full-film lubrication* physically separates the journal from the bearing by a relatively thick (on the order of 0.001 in.), continuous film of self-pressurized lubricant with no metal-to-metal contact. This happy state of affairs is also termed "hydrodynamic lubrication." Low friction and infinitely long service life can be obtained provided a supply of clean lubricant of the right viscosity and sufficient quantity is continuously maintained.

Full-film is the ideal type of lubrication for bronze bearing operation and should be strived for when feasible in order to reap the benefits of low power loss, almost infinite life, and low cost. Sleeve-bearing coefficients of friction for full-film lubrication are on the order of 0.005. However, coefficients of friction of 0.001 or less can be obtained from bronze sleeve bearings in certain applications.

*Complete boundary lubrication* indicates that the bearing and journal surfaces are being rubbed together in the presence of an extremely thin film of lubricant which adheres to the surface of both the journal and the bearing. Unless the bearing is re-lubricated periodically, the thin film is eventually destroyed and intimate metal-to-metal contact results. For bronze bearings operating under conditions of complete boundary lubrication, the coefficient of friction may vary over a range of approximately 0.08 to 0.14. A bronze alloy with a high percentage of lead (15 to 25 per cent) is recommended for extreme boundary conditions.

*Mixed-film lubrication\** is, as the name implies, a combination of hydrodynamic and boundary lubrication. That is, part of the total load carried by the bearing is being supported by individual load-carrying pools of self-pressurized lubricant and the remaining part by the very thin contaminating film associated with boundary lubrication. Coefficients of friction in this realm of operation are in the range of 0.02 to 0.08.

\*Common current practice is to combine boundary film and mixed-film lubricated sleeve bearings under the one title of "boundary lubrication." However, in this Manual, they are treated separately.

**Lubrication and Friction:** Fig. 3 indicates in a very general way the picture of bronze sleeve-bearing performance. This slightly modified version of a diagram familiar to bearing designers consists of a plot of coefficient of friction versus bearing parameter  $ZN/p$ . The figure is separated into three distinct regions to indicate the three realms of sleeve-bearing operation.

At the extreme right of Fig. 3, which corresponds to the higher values of  $ZN/p$ , is the full-fluid-film or hydrodynamic lubrication region. Under full-film conditions, the coefficient of fluid friction is seen to be approximately proportional to viscosity and speed and inversely proportional to load. The coefficient attains a minimum value of approximately 0.001, which is also the minimum coefficient of friction for a good precision grade of rolling-element bearing.

At the extreme left-hand side of Fig. 3 the curve levels off at a high value for the coefficient of friction and remains constant. This portion of the curve is recognized as the region of true boundary friction where the coefficient of friction is independent of viscosity and rubbing speed. Thus, for small values of  $ZN/p$ , the coefficient of friction remains essentially constant. Its magnitude will normally lie between 0.08 and 0.14 depending upon the bearing materials and the lubricant used.

Between the boundary and full-film zones of lubrication (they may also be called zones of friction) is the zone where, with reduction in  $ZN/p$ , the coefficient of friction increases sharply. Evidence indicates that in this zone a combination of fluid friction and boundary friction exists; hence, it bears the title "mixed friction" or "mixed lubrication." The exact values of  $ZN/p$  at which the transition from complete boundary to mixed-film lubrication occurs, and also from mixed-film to full-film lubrication, is difficult to predict. These transition points depend upon such variables as:

1. Quantity of lubricant available.
2. Ability of the lubricant to adhere to bearing surfaces under adverse conditions.
3. Rigidity of bearing and journal.
4. Bearing and journal materials.
5. Operating temperature of bearing.
6. "Wearing-in" of bearing.

Bronze sleeve bearings in common use today are called upon to operate in one or more of these three types of lubrication. For instance, the sleeve bearings of a small fan motor may be squirted with oil at the beginning of summer so that on June 1 the bearings may have an abundance of lubricant. Because of light load and high speed, the bearings will probably operate on a full film until they lose their copious supply of lubricant. Thus, on June 2 the bearings probably will be operating under mixed-film conditions. Certainly by the time the fan is ready for winter storage, the bearings will be operating on a very thin boundary film with no ill effects. Small hand-tool and appliance motors fall into the same category. Usually the bearings for such motors are small in size and very lightly loaded. They are capable of years and years of service

with just the barest of lubrication.

In direct contrast are the large turbine-generator sleeve bearings on which are generated practically every kilowatt of electrical power and which are continuously operated only under full-film conditions. Elaborate precautions are taken to insure that such bearings receive a copious supply of clean lubricant of the right viscosity. Such bearings fulfill what is perhaps one of the most demanding tasks in our present civilization—100 per cent reliability. A classic example of a large-size sleeve bearing which operates in all three regions of lubrication is the journal bearing used for railroad freight cars.

**Full-Film Operation:** Conditions which are necessary to promote full-film or hydrodynamic operation are:

1. Bearing characteristic number should lie within a specified range.
2. Relative surface speed should be greater than approximately 25 fpm and continuous in one direction. Rotation in the opposite direction is also possible provided the motion is not oscillatory in nature. Exceptions are pulsating loads such as occur in engine wrist pins.
3. Lubricant should have the proper viscosity.
4. Lubricant at the proper rate should be continuously supplied to the bearing, and the flow must not be less than a specified minimum rate.
5. The bearing must be properly designed to promote and maintain full-film hydrodynamic lubrication.

Full-film lubrication under very high-load, very slow-speed conditions is possible by using hydrostatic lubrication. Extremely low coefficients of friction (zero at zero speed) can be realized using this type of lubrication. Also, starting and stopping under load can be easily accomplished. Since hydrostatic lubrication requires external pumps, bearings of this type are costly but offer advantages not found in any other type of bearing.

**Mixed-Film Operation:** Bronze bearings which usually operate in the mixed-film lubricated realm of sleeve bearing operation are:

1. All oil-lubricated bearings which are supplied a continuous but relatively small amount of lubricant if the oil-supply rate is less than some specified rate and the surface velocity is greater than 10 fpm.
2. Bearings supplied by drop-feed oilers, wicks, bottle oilers, mechanical oilers, and other types of low-feed-rate devices.
3. Bearings subjected to oscillatory motion if relative surface speeds are greater than 10 fpm.

**Complete Boundary Operation:** Bronze bearings which usually operate in the complete boundary lubricated realm of sleeve bearing operation are:

1. Grease-lubricated bearings.
2. Bearings which are periodically relubricated, as by hand oiling or greasing.
3. Bearings used for reciprocating motion applications (motion along the axis of the shaft).
4. Bearings used in very slow-speed applications where relative velocity between shaft and bearing is less than 10 fpm.



## CHAPTER 2

# Full-Film Lubrication

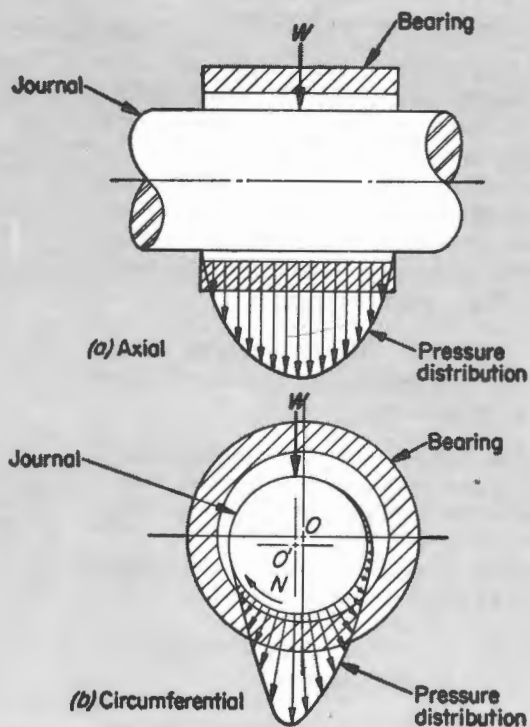


Fig 4—Pressure distributions in a full-film lubricated bronze bearing.

**P**RESENCE of hydraulic pressure in the oil film of a sleeve bearing was discovered in 1883. Later, it was explained and proved that full-film lubrication was the result of hydrodynamic action. In a properly lubricated sleeve bearing, lubricant adheres to both the journal and the bearing. As a result of journal rotation, lubricant is drawn into the converging region formed by the displaced journal in the bearing. Because of lubricant viscosity, fluid pressure is generated in the lubricant film separating the bearing members. This fluid pressure provides the load-carrying capacity in a bronze sleeve bearing.

General shape of the pressure distribution that exists in a bronze sleeve bearing operating on a full film of lubricant is indicated in Fig. 4. Both axial and circumferential pressure distributions are shown. Magnitudes and shapes of the fluid pressure distributions will vary depending upon such factors as load, speed, clearance ratio, and lubricant viscosity. How to determine magnitude and shape of the pressure distribution is described later.

Distinct steps in the formation of the fluid film in a bronze bearing are shown in Fig. 5. When the bearing is at rest, no lubricant separates the bearing members. When the journal begins to rotate, it first "climbs" the wall of the bearing in a direction opposite to rotation. Actually, it rolls up the wall because of friction between journal and bearing. As soon as the friction-force component in the direction of the load is overcome, the journal falls down the wall and crosses to the other side of the bearing. As speed increases, the journal draws more lubricant into the converging wedge until it is completely supported on a full film of lubricant with no metal-to-metal contact. Properly

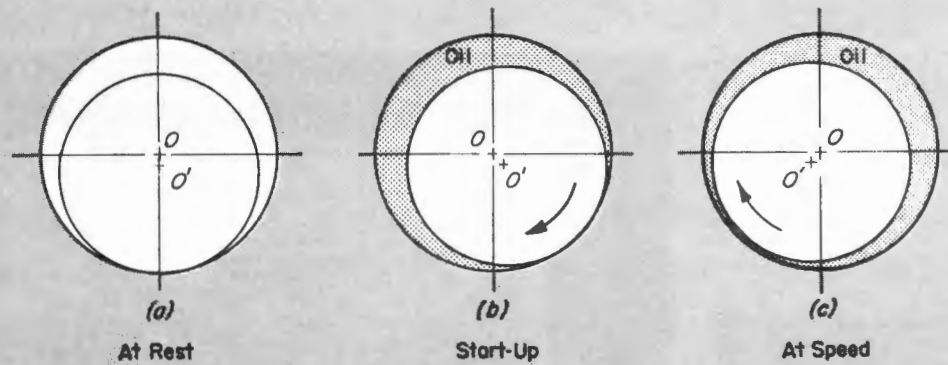


Fig. 5—Formation of full fluid film in a bronze bearing.

designed and continuously supplied with clean lubricant of the proper viscosity and quantity, the bronze bearing will operate forever with absolutely no wear and extremely low friction.

Such performance is the dream of every designer and is readily obtained by using the basic principles and recommendations which follow. After individual factors are discussed, a chart outlines the use of these factors in the actual design procedure. Given conditions are listed first, and other data are easily read from charts and tables. Preliminary calculations provide condensed factors for final computations. The actual ten-step method starts with an assumed bearing bore temperature. With this value, the remaining bearing information is determined and a temperature balance is obtained. If the final temperature does not agree with the initially assumed temperature, a different bore temperature is selected and the ten steps repeated until assumed and calculated temperatures are approximately the same.

Therefore, in the sections which follow, equations and charts are presented, discussed in detail, and ultimately combined into a simple work sheet for designing bronze sleeve bearings under full-film lubrication. A sample design then illustrates the method.

### ► Bearing Clearance Ratios

Often, some confusion results when specifying clearances for bronze sleeve bearings because of the various methods of expressing this clearance. For use in this Manual, terms regarding bearing clearance are defined as:

Diametral clearance,  $2C$ , equal to bearing bore diameter minus journal diameter.

Radial clearance,  $C$ , equal to bearing bore radius minus journal radius.

Clearance ratio,  $2C/D$ , equal to diametral clearance divided by journal diameter.

Because  $C$  is much, much smaller than  $D$ , clearance ratio is multiplied by 1000 to obtain numbers

### Nomenclature

$A$	= Bearing characteristic number
$C$	= Radial clearance, in.
$c_p$	= Specific heat of lubricant, Btu/lb-deg F
$D$	= Journal diameter, in.
$D_B$	= Bearing bore diameter, in.
$e$	= Eccentricity, or radial displacement of journal, in.
$h_o$	= Minimum film thickness of lubricant, in.
$K_1$	= Ventilation factor, Table 1
$k_c$	= Whirl speed factor
$k_f$	= Frictional power factor
$k_q$	= Side-leakage flow factor
$k_s$	= Shock load factor
$k_1, k_2, k_3, k_4, k_s$	= Factors used in Table 2
$L$	= Bearing length, in.
$m$	= Clearance factor $= 1000(2C)/D$
$N$	= Rotational speed of journal, rpm
$N_c$	= Half-frequency whirl speed of shaft, rpm
$O$	= Actual center location of bearing
$O'$	= Actual center location of displaced journal
$P_F$	= Frictional horsepower generated within full-film lubricated bearing, hp
$p$	= Projected area unit load, psi $= W/LD$
$Q$	= Side-leakage oil-flow, or oil-flow feed rate, gpm or drops per min
$Q'$	= Minimum oil-flow required for full-film lubrication, gpm or drops per min
$s$	= Lubrication factor, Table 1 $= (T_2 - T_3)/(T_3 - T_4)$
$T_1$	= Oil inlet temperature, F
$T_2$	= Lubricant film temperature for full-film conditions, or bearing bore temperature for mixed-film and boundary conditions, F
$T_3$	= Surface temperature of bearing housing, F
$T_4$	= Ambient atmosphere temperature, F
$t_B$	= Bearing bore "plus" tolerance, in.
$t_J$	= Journal diameter "minus" tolerance, in.
$\Delta t$	= Duration of shock load, sec
$W$	= Steady load to be supported, lb
$W_m$	= Dead weight of rotating parts, lb
$W_s$	= Shock load, lb
$Z$	= Lubricant absolute viscosity, centipoise
$\gamma$	= Lubricant density, lb per gal
$\epsilon$	= Journal eccentricity ratio
$\epsilon_o$	= Journal eccentricity ratio before shock load
$\epsilon_1$	= Journal eccentricity ratio at end of shock load
$\theta$	= Angle between direction of load and direction of journal displacement, deg

# Recommended Values of Clearance

Fig. 6—Precision spindles made of hardened, ground steel running on lapped cast-bronze bearings (8 to 16 microinch rms finish) when the product  $DN$  is less than 2000.

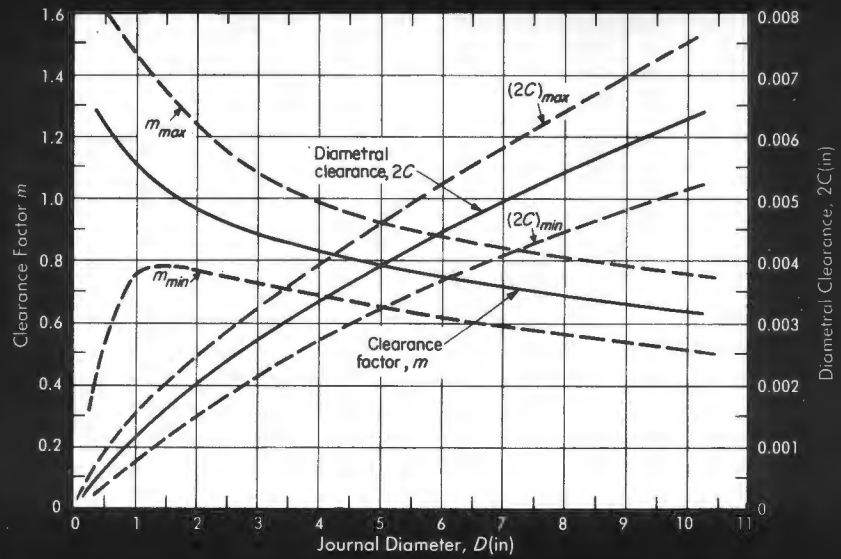


Fig. 7—Precision spindles made of hardened, ground steel running on lapped cast-bronze bearings (8 to 16 microinch rms finish) when the product  $DN$  is more than 2000.

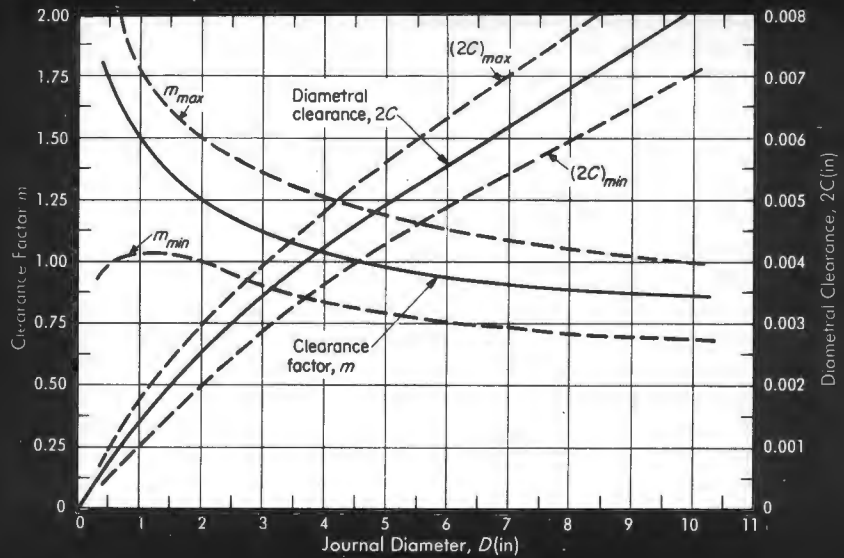
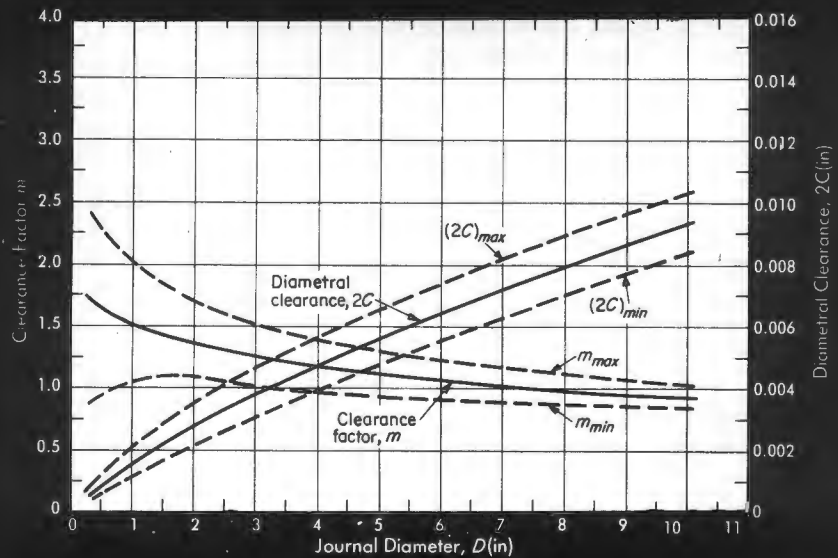


Fig. 8—Electric motors, generators, and similar types of machinery using ground journals in broached or reamed cast-bronze bearings (16 to 32 microinch rms finish).



# Factor m and Diametral Clearance 2C

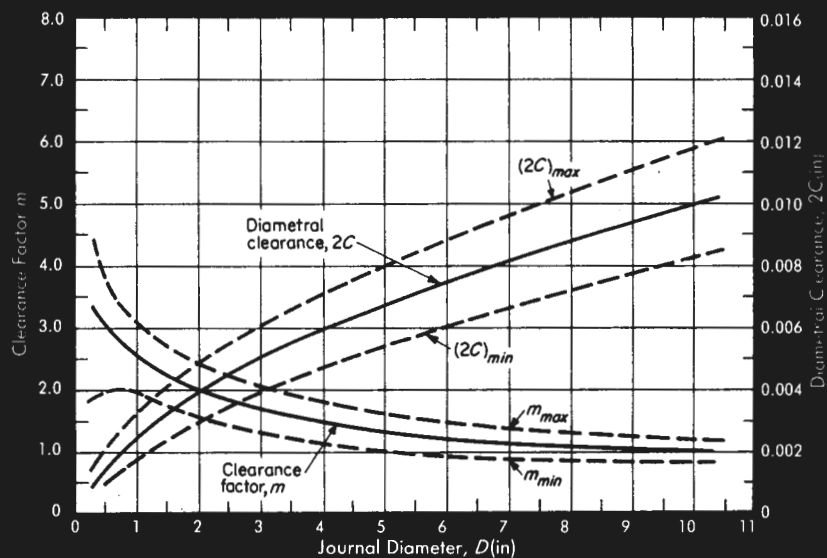


Fig. 9 — General machinery which continuously rotates or reciprocates and uses turned or cold-rolled steel journals in bored and reamed cast-bronze bearings (32 to 63 microinch rms finish).

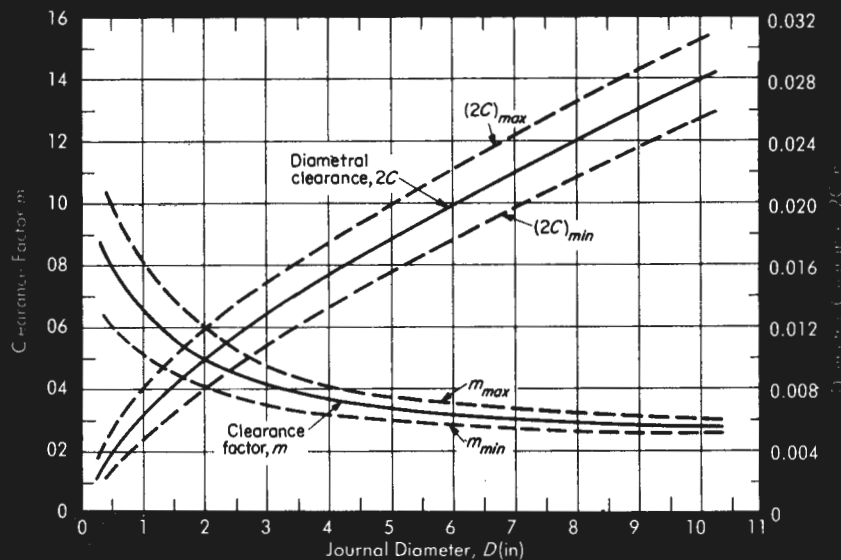


Fig. 10 — Rough-service machinery having turned or cold-rolled steel journals operating on cast-bronze bearings (63 to 125 rms finish).

more convenient to use. This modified ratio is termed "clearance factor" and is

$$m = 1000 \left( \frac{2C}{D} \right) \quad (2)$$

As a useful guide to determining bearing clearance factor  $m$ , first identify the particular application with one of the descriptions given for Fig. 6 through 10. Then, a recommended value for  $m$  can be obtained from the figure for that type of machinery. Two sets of curves are shown on each of the figures. One set indicates average recommended values for  $m$  and also shows the range within which the designer may obtain acceptable results. The second set shows similar curves for the resulting diametral clearances,  $2C$ , obtained when using the suggested values of  $m$ . In general, these clearance factor

values may also be used for mixed-film and complete boundary film operating conditions.

**Journal and Bore Tolerances:** When dimensions and tolerances for journal and bore are specified, the following practice is recommended:

Journal diameter should be specified as

$$D = D_{nom} \begin{matrix} +0.0000 \\ -t_j \end{matrix} \quad (3)$$

and bearing bore diameter as

$$D_B = (D_B)_{nom} \begin{matrix} +t_B \\ -0.0000 \end{matrix} \quad (3a)$$

When the two diameters are specified as above, minimum diametral clearance will be

$$(2C)_{min} = (D_B)_{nom} - D_{nom} \quad (3b)$$

and maximum diametral clearance will be

$$(2C)_{max} = [(D_B)_{nom} - D_{nom}] + t_B + t_J \quad (3c)$$

The allowable tolerances may then be computed from

$$t_B + t_J = (2C)_{max} - (2C)_{min} \quad (4)$$

Both  $(2C)_{min}$  and  $(2C)_{max}$  may be obtained directly from Fig. 6 through 10, and values may be conveniently assigned to  $t_B$  and  $t_J$  by using Equation 4. If the nominal journal diameter is known, the required nominal bearing bore diameter may be determined directly by using Equation 3b.

**EXAMPLE:** The journal of a 3-in. diameter bearing for an electric motor is specified as 3.000, +0.0000, -0.0005. How should the bronze bearing bore be specified to insure a satisfactory value of  $m$ ?

From Fig. 8,

$$(2C)_{max} \text{ for a 3-in. diameter is } 0.0047 \text{ in.}$$

$$(2C)_{min} \text{ for a 3-in. diameter is } 0.0030 \text{ in.}$$

From Equation 4,

$$t_B + 0.0005 = 0.0047 - 0.0030$$

$$t_B = 0.0012 \text{ in.}$$

Nominal bearing bore diameter from Equation 3b is

$$(D_B)_{nom} = (2C)_{min} + D_{nom}$$

$$= 0.0030 + 3.0000$$

$$= 3.0030 \text{ in.}$$

Bore diameter may then be specified as

$$D_B = 3.0030^{+0.0012}_{-0.0000}$$

In this example, the tolerance assigned to both bearing members allows  $m$  to range between 1.0 and 1.5. Satisfactory results are obtained by using an average value for  $m$ . In this case, a value of 1.25 would be selected for  $m$ .

To maintain proper diametral clearance, the bore diameter of the bearing should be finish-machined to size after installation. This recommended practice eliminates errors caused by the accumulation of tolerances from bearing ID and OD and housing ID. The method also results, generally, in lower overall installation cost since only one close tolerance need be, and definitely can be, maintained.

## ► Bearing Characteristic Number

First major step in the design of a bronze bearing is to evaluate the bearing characteristic number,  $A$ . This number is determined as follows:

$$A = \frac{m^2 W}{D^2 Z N} \quad (5)$$

Note that this number is a slight modification of the parameter  $ZN/p$  used previously. Usually, desired operating speed  $N$  and total steady load  $W$  to be carried are known. Likewise, journal diameter  $D$  in the vicinity of the bronze bearing may also be specified to meet certain stiffness or deflection requirements. Determination of clearance factor  $m$  was discussed in the preceding section, *Bearing Clearance Ratios*.

Since the operating temperature of the lubricant film for a bronze bearing of unknown length is impossible to predict, the next best thing is to assume an operating temperature. Absolute viscosity  $Z$  can then be determined for the lubricant at the assumed temperature,  $T_2$ . Ordinarily, sufficient data from the manufacturer of the selected lubricant will permit simple determination of a value for  $Z$ . If complete information is not available, absolute viscosity can be calculated by a method described in Chapter 5. However obtained, the proper value of  $Z$  is then used in Equation 5.

A fair approximation of lubricant temperature rise may be made if the method of lubrication is known. For forced-feed or pressure lubrication, temperature rise  $T_2 - T_1$  will be in the neighborhood of 5 to 10 F. With less oil being supplied, the bearing will tend to run hotter since it is not being flushed by as much relatively cool inlet oil. Thus, for other lubricating techniques, such as oil bath, splash feed, and oil ring, lubricant temperature rise may range from 10 to 100 F. Knowing these relationships, the designer should be able to make a reasonable approximation of the lubricant film temperature and, hence, of lubricant viscosity.

With known and determined values substituted for  $m$ ,  $W$ ,  $D$ ,  $Z$ , and  $N$  in Equation 5, the bearing characteristic number can be evaluated. If  $A$  falls in the range between 0.0005 and 0.50, practical full-film lubrication is possible. With special care, the upper limit can easily be doubled to 1.0. However, most bronze sleeve bearings for rotating machinery operate within this given range of bearing characteristic number. If  $A$  is greater than 0.50, special types of lubrication discussed later may be required.

Even if the value of  $A$  falls within the normal

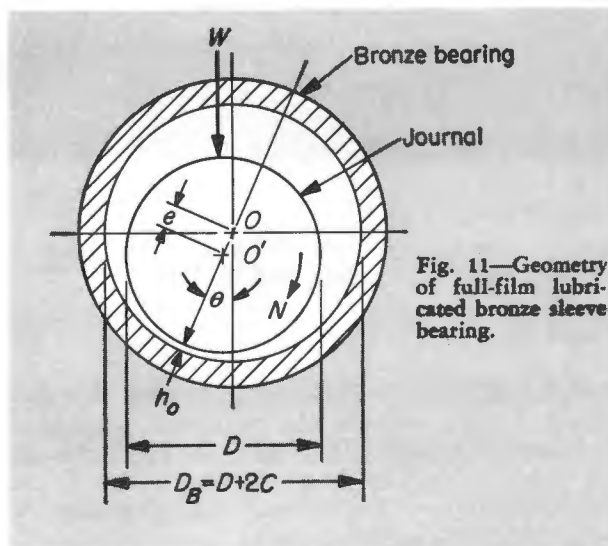
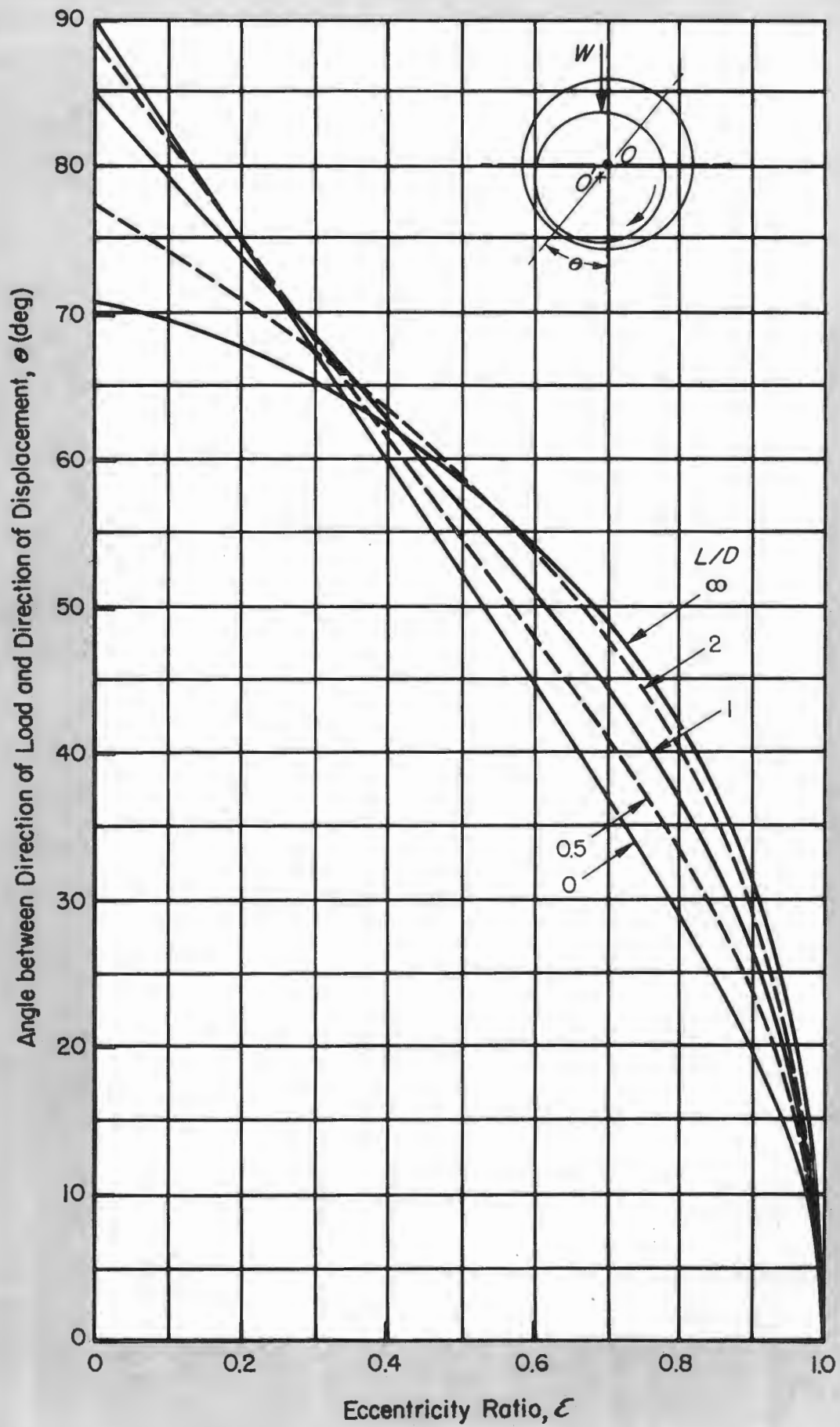
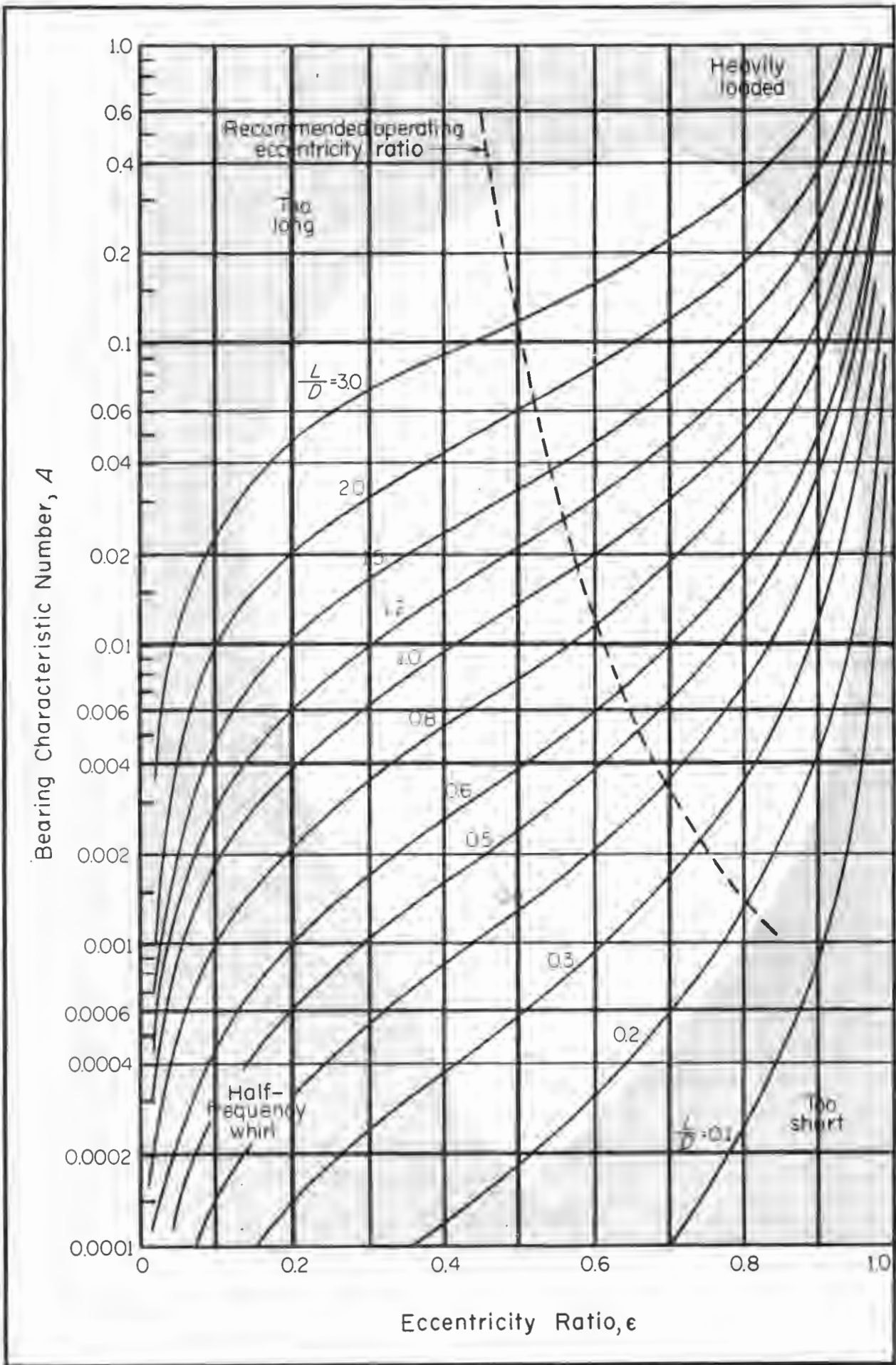


Fig. 11—Geometry of full-film lubricated bronze sleeve bearing.

Fig. 12—Angular location of displaced journal within a full-film lubricated bronze bearing. (Reference 3.)





range, full-film lubrication may not be possible for one or more of the following reasons:

1. Insufficient flow of lubricant to the bearing as a result of either ignorance or an attempt to reduce the flow to a value felt to be acceptable.
2. Misplaced oil-feed grooves.
3. Too low a lubricant viscosity.
4. Use of a porous bearing material which prevents formation of hydrodynamic pressures. Hence, separation of the journal from the bearing is not possible.
5. Severe misalignment.
6. Dirty lubricant.
7. Excessive heating within the bearing which is a result of poor heat dissipation and which reduces lubricant viscosity. Cast bronze bearings have excellent heat-dissipation qualities.

A large value for bearing characteristic number indicates a heavily loaded or relatively slow-speed bearing. Conversely, light loads and high speeds give very low bearing characteristic numbers. If desired, the numerical value of  $A$  can be adjusted up or down by varying the parameters  $m$ ,  $D$ , and  $Z$ . Also, duty cycle of the bearing should be investigated to determine the possible variation of  $A$  with changing speed, load, and, indirectly, viscosity.

Bear in mind that only steady loads in both magnitude and direction are considered for use in the bearing characteristic number. Likewise, only constant, unidirectional rotation of the shaft is applicable for continual full-film, hydrodynamic lubrication. Shock loads, rotating loads, and reciprocating or oscillating journals are considered in another section of this Manual. This treatment is valid only for fluid lubricants and does not apply to grease lubrication.

### ► Journal Eccentricity Ratio

An important concept in full-film lubrication is journal eccentricity ratio,  $\epsilon$ , which is determined

from

$$\epsilon = \frac{e}{C} \quad (6)$$

Fig. 11 illustrates the eccentricity of a bronze bearing operating with a full film of lubricant. In this cross section of the bearing, clearance has been purposely exaggerated. Equilibrium position for the center of the journal,  $O'$ , will be displaced from the center of the bearing,  $O$ , a distance equal to  $e$  and at an angle equal to  $\theta$ . Both  $e$  and  $\theta$  depend upon:

1. Magnitude and direction of load  $W$ .
2. Magnitude and direction of speed  $N$ .
3. Viscosity of lubricant in the film,  $Z$ .
4. Bearing clearance factor  $m$ .

When there is no load on the bearing ( $W = 0$ ), the journal will run virtually centered within the bearing, and eccentricity  $e$  will be zero. Thus, from Equation 6, the eccentricity ratio will likewise be zero. As the load increases, the journal moves eccentrically until a position is reached where hydrodynamic pressure distribution developed in the oil film balances the load. Additional loading requires that the journal move to an even more eccentric position.

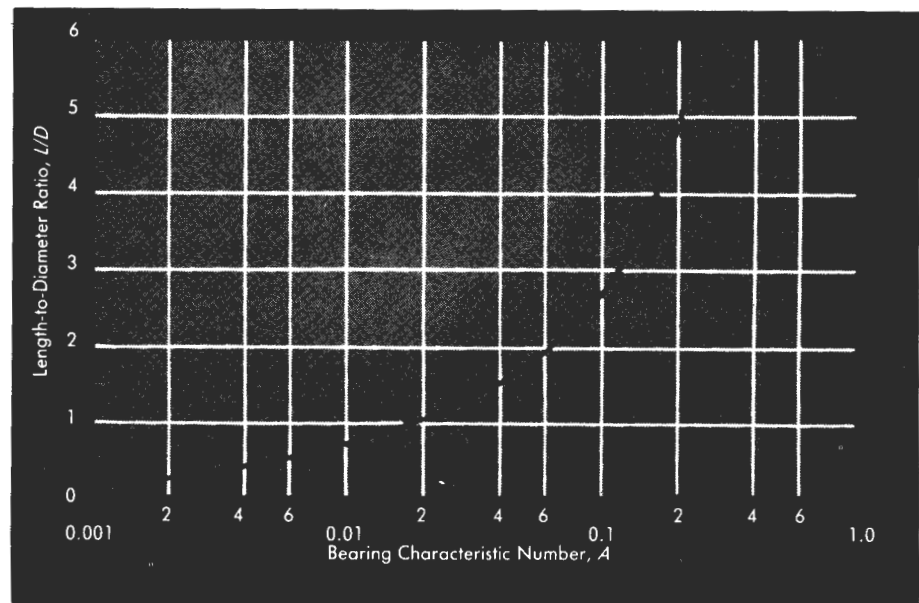
While eccentricity  $e$  is increasing, minimum film thickness  $h_o$  is decreasing. The equation for  $h_o$  is

$$h_o = C - e = C(1 - \epsilon) \quad (7)$$

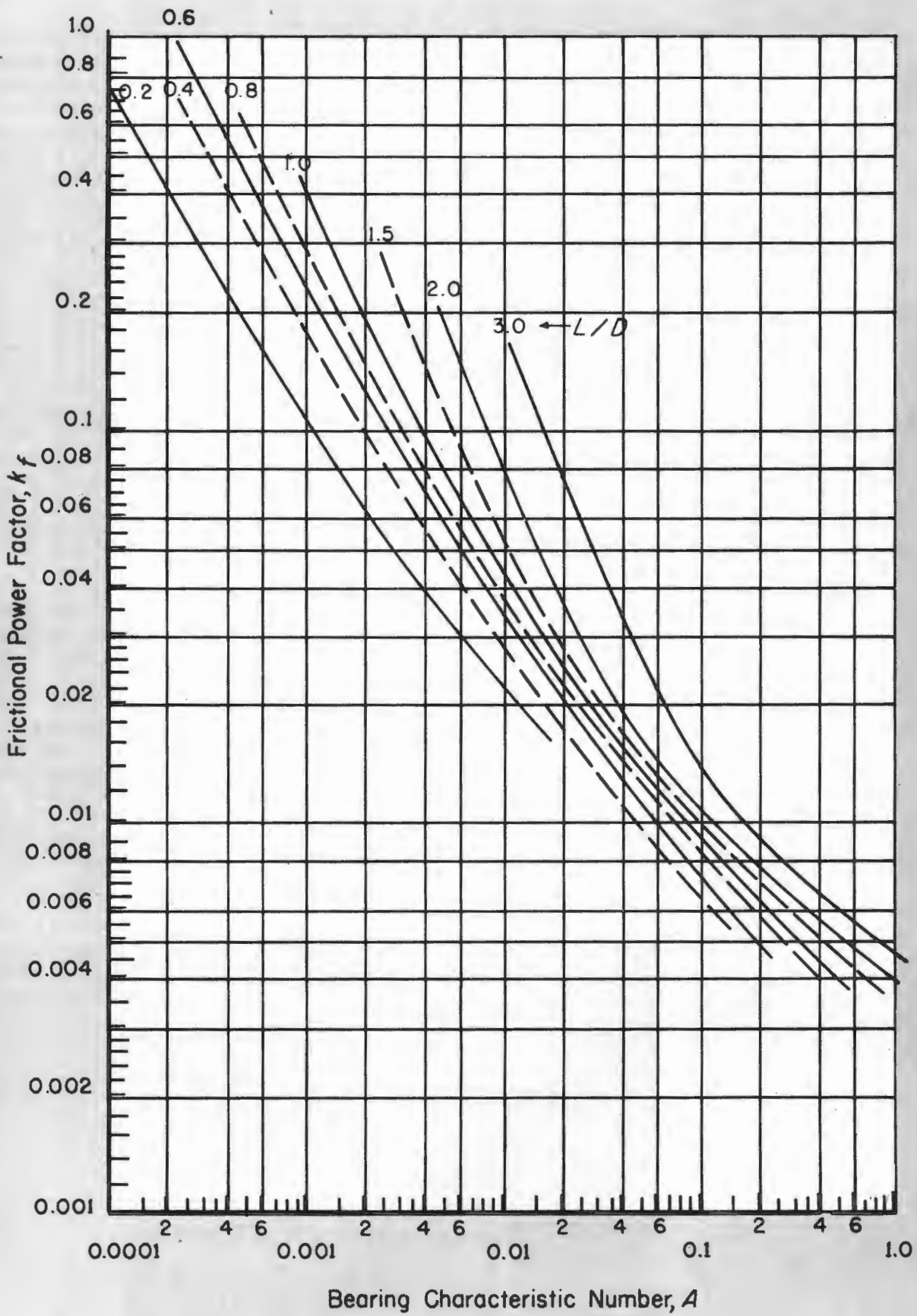
If the load becomes great enough, the journal may eventually touch the bearing. For this situation,  $\epsilon = C$ ,  $h_o = 0$ , and the eccentricity ratio becomes unity. Fig. 12 is a plot of angle  $\theta$  (between direction of load and direction of displacement of the journal) and eccentricity ratio  $\epsilon$  for various  $L/D$  ratios. Thus, for any eccentricity ratio, the exact location of the center of the journal and also the location and magnitude of minimum film thickness can be readily determined. This information is necessary for later considerations such as surface-finish requirements and oil grooving.

◀ Fig. 13—Bearing size selection chart for full-film lubricated 360-deg bronze sleeve bearings. (After Raimondi and Boyd.)

Fig. 14—Ratios of  $L/D$  required by recommended operating eccentricity ratios. The  $L/D$  curve is obtained from Fig. 13. ▶







## ► Bearing Length

Once the bearing characteristic number is computed, a suitable length for the bearing can be established which will insure satisfactory full-film performance. Fig. 13 permits determination of proper bearing proportions  $L/D$  and thereby calculation of  $L$  since  $D$  is known.

In Fig. 13 eccentricity ratio is plotted against bearing characteristic number for various  $L/D$  ratios. The unshaded area of the chart is the only region of interest. To use this chart, enter the vertical axis at the proper value of  $A$  and move horizontally across the graph to the dashed line marked *Recommended operating eccentricity ratio*. The required ratio of  $L/D$  to obtain this recommended eccentricity ratio for the particular value of  $A$  is obtained by interpolation of the  $L/D$  curves. As an example, suppose the computed bearing characteristic  $A$  equals 0.011. From the chart, this value of  $A$  gives  $L/D$  equal to 0.80 for the recommended operating eccentricity ratio of 0.60. Since  $D$  is known,  $L$  is easily calculated. Fig. 14 is a plot of  $L/D$  versus  $A$  for recommended operating eccentricity ratios.

Lengths of bronze bearings other than those dictated by the recommended operating eccentricity ratio can be used. For instance, for the case just mentioned where  $A = 0.011$ , an  $L/D$  value as low as 0.2 could be chosen, but the operating eccentricity ratio would be 0.95, which means that the journal would almost be touching the bearing. At the other extreme,  $L/D$  ratio of 1.7 gives an operating eccentricity ratio of 0.15, which means that the journal would be only slightly displaced within the bearing.

Any selection of  $L/D$  within the unshaded area of Fig. 13 can be made. Shorter bronze bearings are usually preferred from the standpoint of space, friction, and flow requirements. However, for the same bearing characteristic number, shorter bearings necessarily operate with greater eccentricity. There must be a compromise, and on this basis the recommended

operating eccentricity ratio curve in Fig. 13 was established. Also, some allowance is made to enable the bearing to carry higher loads than anticipated, since operating film thicknesses are still fairly large. In addition, if film thicknesses are large, slight contamination of the lubricant can be tolerated. Thus, if possible, bearing lengths for full-film lubrication should be computed as dictated by the recommended operating eccentricity ratios. Fig. 14 makes this computation very easy once the value of  $A$  has been calculated.

For the hypothetical value of  $A = 0.011$ , which gives an  $L/D$  ratio of 0.8, the curve of  $L/D = 0.8$  can be followed to observe the change of eccentricity ratio caused by a change in load and/or speed and/or viscosity. For instance, if the load doubles, the new value of  $A$  will be 0.022. According to Fig. 13, this value requires an eccentricity ratio of 0.73. In other words, once an  $L/D$  ratio has been decided upon, location of the journal within the bearing can be completely defined, with the help of Fig. 12 and 13, for varying speed, load, or viscosity.

## ► Power Requirements

Although the coefficient of friction for full-film lubricated bronze bearings is quite small, power required to drive the journal may be appreciable because of high load-carrying capacity and reasonably high speeds. Also, since frictional energy is dissipated in the form of heat, energy required to overcome friction must be evaluated to determine any cooling requirements for the bearing.

The equation which determines driving or input power is

$$P_F = k_f m D N W \times 10^{-6} \quad (8)$$

The value of frictional power factor  $k_f$  can be obtained from Fig. 15 if the value of the bearing characteristic number and the  $L/D$  ratio are known.

Fig. 15—Frictional power factors for 360-deg full-film lubricated bronze bearings having various  $L/D$  ratios. (After Raimondi and Boyd.)

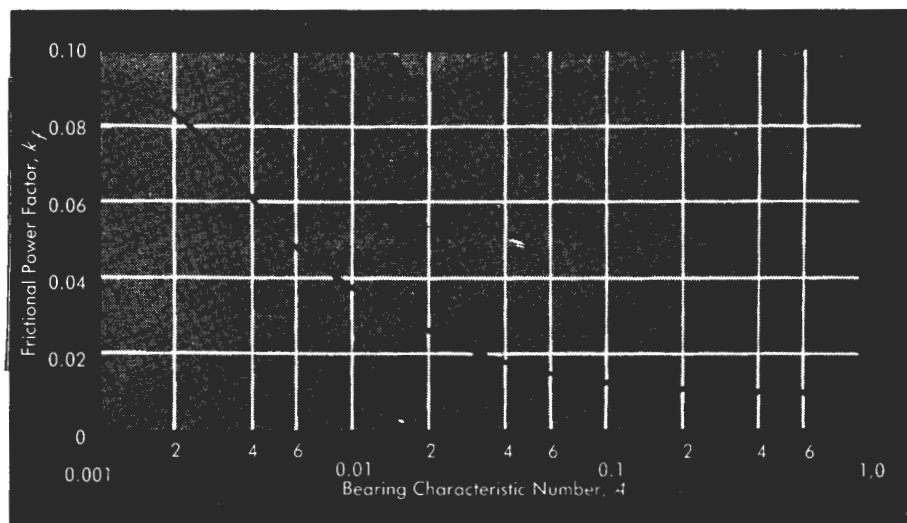
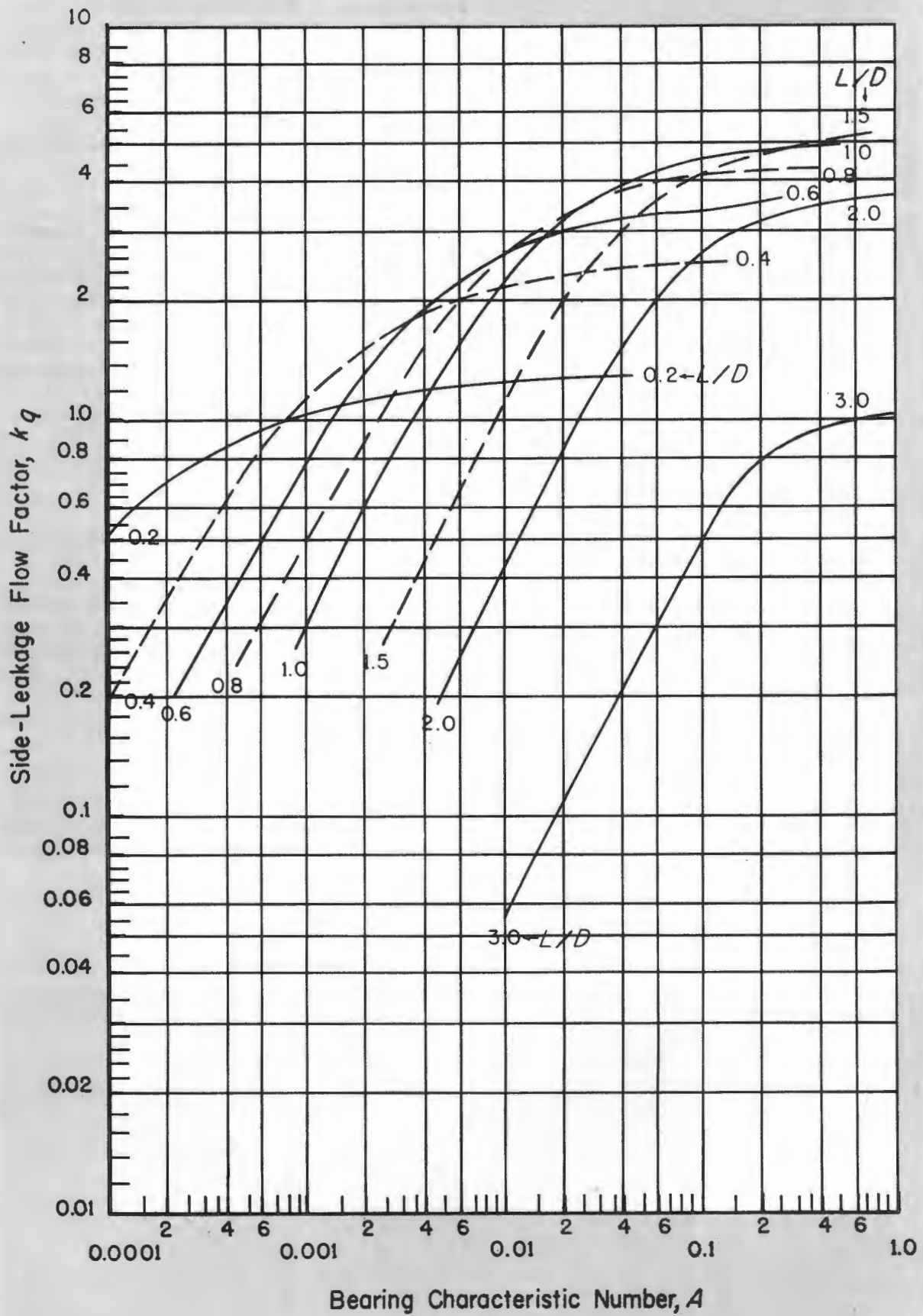


Fig. 16—Frictional power factors based upon recommended operating eccentricity ratios.



To illustrate with the previous example where  $A = 0.011$  and the  $L/D$  ratio is 0.80, Fig. 15 gives  $k_f = 0.036$ . Since the other terms in Equation 8 are known, the horsepower requirement for a full-film bronze sleeve bearing can be easily computed.

For a bearing of given  $L/D$  ratio, the variations in power requirement with change in load, speed, or viscosity can be determined, for a correspondingly different value of  $A$ , by picking from Fig. 15 the new value of  $k_f$  and substituting it into Equation 8. In the preceding section, *Bearing Length*, particular values of  $A$  were shown to have a recommended operating eccentricity ratio which, if adhered to, fixed the  $L/D$  ratio. If both  $A$  and ratio  $L/D$  are fixed,  $k_f$  is restricted to a single value also. Thus, a plot of  $k_f$  against  $A$  can be obtained based on a bearing designed to operate at the recommended operating eccentricity ratio. Such a plot is shown in Fig. 16.

### ► Oil-Feed Requirements

In a 360-deg bronze bearing, oil must enter the bearing at the same rate that it escapes from the ends of the bearing. End leakage in a sleeve bearing is caused by the high pressure that develops in the center of the bearing. This pressure forces lubricant toward the low-pressure areas at the ends of the bearing (see Fig. 4). If the journal is to maintain its position within the bearing, lubricant lost through end leakage must be continually replenished.

The theoretical oil-feed rate required to maintain the clearance space filled with oil while the bearing is operating is

$$Q = k_q m D^3 N \times 10^{-6} \quad (9)$$

The value of side-leakage flow factor  $k_q$  can be obtained from Fig. 17. For given values of  $A$  and  $L/D$ ,  $k_q$  can be quickly determined. For the example using  $A = 0.011$  and  $L/D = 0.8$ ,  $k_q$  is found to equal 2.8. As in similar charts, curves in Fig. 17 for given

values of  $L/D$  indicate the variation of  $k_q$  with changes in load, speed, and viscosity. When these values of  $k_q$  are used in Equation 9, the variations in flow requirement can be determined.

Surprisingly enough, if less than the computed rate of flow is delivered, operating eccentricity will be not far removed from what it would be if the required rate were delivered. The effect of reduced oil flow on operating eccentricity is almost negligible at the higher eccentricity ratios. How much less the oil-feed rate may safely be than the value calculated with  $k_q$  in Equation 9, while maintaining reasonably similar eccentricity ratios, is difficult to specify.

As a rule of thumb, when operating at eccentricity ratios less than 0.50, calculated oil-feed rate should be maintained if the expected operating eccentricity ratio is to be realized. For larger eccentricity ratios, feed rate may be somewhat less than calculated, gradually decreasing to about 80 per cent of the calculated flow at approximately 0.90 eccentricity ratio.

Consider the operation of a bronze bearing subjected to wide extremes of load. If enough oil is supplied to the bearing at high loads, the bearing will be oversupplied at low loads, which is not harmful. However, if the flow rate is designed for low loads, insufficient oil flow for high loads will force the bearing to operate on a thinner film of lubricant. Hence, the safe practice is to supply the highest oil-feed rate required to meet varying conditions.

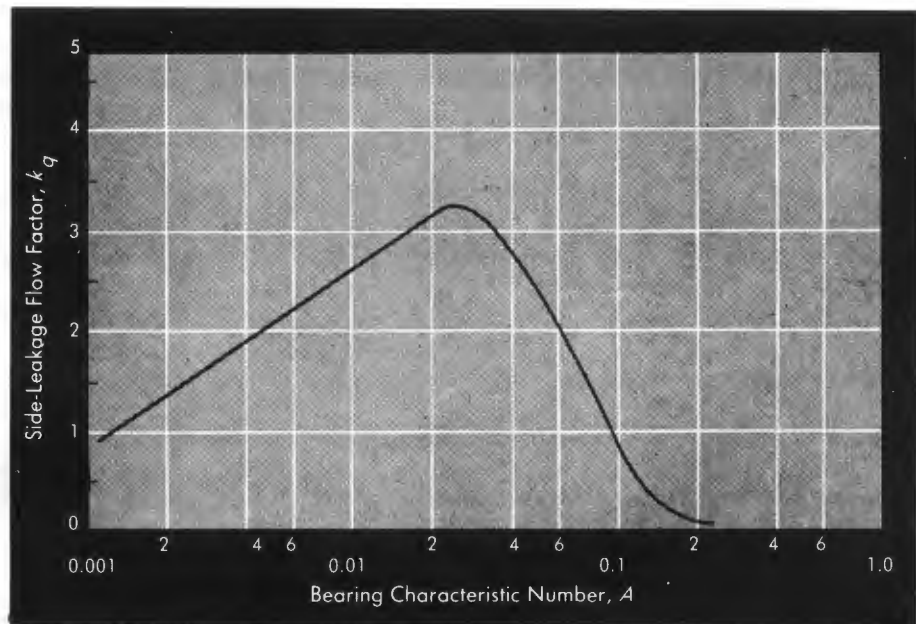
If  $A$  and  $L/D$  are fixed,  $k_q$  must also be restricted to a particular value. For a bearing designed to operate at the recommended operating eccentricity ratio,  $k_q$  can be plotted against  $A$  as shown in Fig. 18.

### ► Minimum Flow Requirements

Operating with a bearing characteristic number that indicates a full film of lubricant between journal

Fig. 17—Side-leakage flow factors for 360-deg full-film lubricated bronze bearings having various  $L/D$  ratios. (After Raimondi and Boyd.)

Fig. 18—Side-leakage flow factors based upon recommended operating eccentricity ratios.



and bearing, a bronze bearing will run at a prescribed eccentricity if supplied with sufficient lubricant. If the rate of flow to the bearing is reduced, the journal will move to a new, more eccentric position within the bearing because of insufficient fluid. Experiments have determined that fluid-film lubrication is not possible below a definite minimum lubricant flow rate.<sup>1</sup> The journal will begin to touch the bearing at the point of closest approach, and bearing friction will increase markedly if lubricant flow is below this minimum value.

Minimum flow required for full-film lubrication,  $Q'$ , can be determined approximately with the following equations:

$$Q' = 29.3 \times 10^{-9} \left( L + 0.0043 \frac{W}{D} \right) m D^2 N, \text{ gpm} \quad (10a)$$

$$Q' = 3.32 \times 10^{-3} \left( L + 0.0043 \frac{W}{D} \right) m D^2 N, \text{ drops per min} \quad (10b)$$

$$1 \text{ cu cm} = 30 \text{ drops}$$

Needless to say, supplying just the barest minimum of flow to the bearing to sustain full-film lubrication, especially in a new bearing which has had no "wearing in," is extremely dangerous practice. However,  $Q'$  can be used as a guide to determine what value the flow may be allowed to approach and still maintain the benefits of full-film lubrication.

## ► Oil-Film Temperature

To determine lubricant viscosity and, in turn, bearing characteristic number, oil film temperature  $T_2$  had to be assumed. Since means for calculating bearing length, frictional power, and flow through the bearing have been presented, temperature rise of the lubricant as it passes through the bearing can now be computed. A portion of the heat generated in the bearing is transmitted to the flowing lubricant, and some of the heat is also dissipated through the wall of the bearing to the ambient atmosphere. Of the many types of sleeve bearing materials available, solid cast bronze offers the best heat-dissipating qualities.

**Self-Contained Bearing System:** A self-contained bearing is one which requires no oil-circulating or cooling system. In this type of bearing practically all the heat is removed by conduction, convection,

<sup>1</sup>References are tabulated in Bibliography.

**Table 1—Ventilation and Lubrication Factors**

Ambient Air Condition	Ventilation Factor, $K_1$	Lubrication Factor, $s$		
		Oil-Bath Lubrication	Oil-Ring Lubrication	Waste-Pack Lubrication
Quiet Air	6	0.20	0.90	1.60
Moving Air (500 fpm)	16	0.63	1.30	4.4

and radiation to the surrounding atmosphere. Self-contained bearings are the conventional pillow-block and pedestal type found on most motors, generators, turbines, pumps, and similar equipment. Such bearings reach thermal equilibrium 1 to 3 hr after starting. Thereafter, radiation and convection from the bearing housing is sufficient to dissipate all the heat generated by friction in the bearing.

If the particular bearing is operating at high speeds and heavy loads, equilibrium may be reached at a high level of temperature—perhaps higher than 200 F. Temperatures in this range may not be acceptable for industrial use because some lubricants undergo a rather rapid deterioration which results in the formation of harmful acids. Therefore, the limit of acceptable equilibrium film temperatures for the usual industrial application range from 160 to 180 F. Operation beyond these temperatures usually requires auxiliary cooling such as a cooling coil in the oil sump. In this case, some of the heat is then removed directly from the oil itself.

Cast-bronze bearings can be effectively applied at elevated temperatures with the use of proper high-temperature lubricants. However, discussion in this Manual is concerned with bronze bearing operation at less than 200 F.

In a self-contained bearing system the assumption that all frictional heat generated within the bearing must be dissipated to the ambient atmosphere permits the resulting temperature difference to be expressed as<sup>2</sup>

$$T_2 - T_4 = \frac{(s + 1) P_F}{15 \times 10^{-6} D L K_1} \quad (11)$$

Values for  $s$  and  $K_1$  are simplifications of data in Fuller's text and may be obtained from Table 1.

The value of  $T_2$  can be determined from Equation 11 and compared with the assumed value used earlier to calculate the bearing characteristic number. In all likelihood they will not agree exactly. Hence, another assumption of  $T_2$  is made and the entire process repeated. Another value of  $T_2$  is calculated and is again compared with the assumed value. This process is continued until reasonable agreement is obtained between the assumed value of  $T_2$  and the calculated value of  $T_2$ . The actual process will be illustrated in the sample problem which follows later.

**Forced-Feed Lubricating System:** In bearings supplied with copious amounts of relatively cool lubricant, most of the heat of friction enters the oil and is thus removed from the bearing. Usually, the oil is returned to a sump where it is cooled before re-entering the bearing.

If the assumption is made that all heat generated by friction within the bearing is removed by the lubricant and that side-leakage flow through the bearing is as determined by Equation 9, then temperature rise of the lubricant can be expressed as

$$T_2 - T_1 = \frac{42.4 P_F}{\gamma c_p Q} \quad (12)$$

If oil inlet temperature  $T_1$  is known,  $T_2$  can readily

be determined from Equation 12 and compared with the assumed value of  $T_2$  used to determine the bearing characteristic number. Reasonable agreement indicates a solution has been found. Otherwise, the process, based on a new assumed value for  $T_2$ , is repeated until agreement between assumed and calculated values of  $T_2$  is obtained.

**Combined Heat Loss:** For those cases where heat is removed not only by the oil but also by conduction through the bearing, lubricant film temperature can be determined by the following equation, which is a combination of Equations 11 and 12:

$$T_2 = \frac{P_F + \left( \frac{15 \times 10^{-6} K_1}{s + 1} \right) (DL) T_4 + \left( \frac{\gamma c_p Q}{42.4} \right) T_1}{\left( \frac{15 \times 10^{-6} K_1}{s + 1} \right) (DL) + \left( \frac{\gamma c_p Q}{42.4} \right)},$$

degrees Rankine (13)

Since this equation does not deal with temperature differences, absolute temperatures in degrees Rankine (degrees F plus 460) must be used for  $T_1$ ,  $T_2$ , and  $T_4$ .

Once again, if  $T_2$  as computed from Equation 13 is different than the assumed value, a new assumption for  $T_2$  is made and all the calculations repeated until agreement is reached. Equation 13 should be used when there is any doubt concerning the distribution of heat losses.

### ► Simplified Design Method

To make the computational work required for actual bearing design more systematic, Table 2 is presented in three parts. Necessary entries and computations are indicated, and space is provided for given or determined values. Part A lists data needed to design the bearing. This information is usually given or known. Part B provides for preliminary calculations that reduce the number of terms involved and simplify final computations. Values for all symbols used are obtained from Part A. Part C is a step-by-step process for final calculations which determine a temperature balance. Results obtained from Step 10 indicate whether or not the bearing design is satisfactory, and if not, what adjustment is necessary before again proceeding through the ten steps.

### ► Heavily Loaded Bronze Bearings

A heavily loaded bronze bearing operating under full-film lubrication will have a high eccentricity ratio. Hence, resulting minimum film thickness  $h_0$  will be extremely small. The heavily loaded, or relatively slow-speed, region of operation is indicated as the extreme upper right portion of Fig. 13.

Theoretically, a bronze bearing will carry an infinitely large load before the journal touches the bearing. However, the actual maximum load that a sleeve bearing can support is far short of infinite

because of such things as surface roughness, journal deflection, bearing distortion, and out-of-roundness. Recommended maximum allowable eccentricity ratios for heavily loaded bronze bearings of various  $L/D$  ratios are shown on Fig. 13. If these values are exceeded, the journal may rupture the film of lubricant and touch the bearing.

When such contact does occur, friction within the bearing increases very markedly. Frictional increases of 1000 per cent are not uncommon for slight increases in load, or decreases in speed, in the transition from full-film to mixed-film lubrication. The increased friction generates more heat, which reduces lubricant viscosity, which promotes more metal-to-metal contact, which causes more friction, and the cycle is repeated again and again. If overloading is not too severe, this vicious cycle will continue until the bearing "wears-in" and some temperature equilibrium condition is established in the sleeve bearing.

On the other hand, operation of the bearing under severe overload can lead to rapid temperature rises, accelerated wear, and early failure. Thus, if a sleeve bearing is to be subjected to unavoidable overloading, the overloads should be of short duration and should occur no oftener than absolutely necessary. Also, if possible, the bearing should be well worn-in before being overloaded. The beneficial effects of wearing-in result from smoothing microscopic high spots.

If continuous operation is desired at speeds and steady loads requiring a bearing characteristic number greater than 0.50, full-film hydrodynamic operation should be abandoned in favor of hydrostatic lubrication. This method of lubrication is discussed later.

Reviewing briefly, operation beyond the maximum eccentricity ratios shown in Fig. 13 is to be avoided if at all possible. Occasionally, the bearing may be subjected to overloads that are not too severe and of short duration. For this type of service the bearing should be carefully worn-in before being overloaded. In any case, a maximum allowable bearing characteristic number is approximately 0.50 for steady (nonshock) loading.

### ► Shock-Loaded Bronze Bearings

In many applications, bronze sleeve bearings under full-film lubrication are subjected to pulsating or reciprocating loads. Because of high pressure and lack of continuous sliding or rotation, oil-film breakdown and severe wear should theoretically occur if the bearing characteristic number is used as a criterion. Strangely enough, bronze bearings properly designed for applications of this type show no signs of wear. Crank-pin and piston-pin bearings are examples of heavily loaded, slow-speed bronze bearings having varying loads. Relative motion in such bearings is zero at periodic intervals, but nevertheless, an oil film is maintained between bearing surfaces.

This load-carrying phenomenon can be explained by the fact that a viscous lubricant cannot be instantaneously squeezed out from between two sur-

## Table 2 – Design Sheet for Full-Film

### Part A--Known Data

	Symbol	Value	Units
Total steady load	$W$	---	lb
Journal speed	$N$	---	rpm
Journal diameter	$D$	---	in.
Clearance factor	$m$	---	---
Oil inlet temperature	$T_1$	---	F
	$T_1$	---	Rankine
Ambient atmosphere temperature	$T_A$	---	F
	$T_A$	---	Rankine
	$T_A$	---	in.
Radial clearance	$C$	---	---
Ventilation factor	$K_1$	---	---
Lubrication factor	$s$	---	lb/gal
Lubricant density	$\gamma$	---	Btu/lb-deg F
Lubricant specific heat	$c_p$	---	---
Dead weight supported	$W_m$	---	lb

### Part B--Simplified Factors

	Calculation	Value
$k_1$	$m^2 W/D^2 N$	---
$k_2$	$m D N W \times 10^{-6}$	---
$k_3$	$m D^3 N \times 10^{-6}$	---
$k_4$	$(15 \times 10^{-6} K_1)/(s+1)$	---
$k_5$	$\gamma c_p / 42.4$	---

### Part C--Final Calculation Form

Step NO.	Description	Values for First Assumption	Values for Second Assumption	Values for Third Assumption
1.	$T_2$ (assumed), deg Rankine	---	---	---
2.	$Z$ (known), centipoise	---	---	---
3.	$A = k_1 / Z$	---	---	---
4.	$L$ (Fig. 13), in.	---	---	---
5.	$\epsilon$ (Fig. 13)	---	---	---
6.	$h_o$ (Equation 7), in.	---	---	---
7.	$\theta$ (Fig. 12), deg	---	---	---
8.	$P_F = k_f k_2$ , hp	---	---	---
9.	$Q = k_q k_3$ , gpm	---	---	---
10.	$T_2 = (P_F + DL k_4 T_4 + k_5 Q T_1) / (DL k_4 + k_5 Q)$ , deg Rankine	---	---	---

# Lubricated Bronze Bearings

## ► Preliminary Steps

Part A: Fill in values dictated by the problem.

Select  $m$  from Fig. 6 through 10 for the specified class of machinery and operating speed.

Obtain  $K_1$  and  $s$  from Table 1.

Values of  $\gamma$  and  $c_p$  should be known. Average values for normal petroleum lubricants are  $\gamma = 7.5$  lb/gal and  $c_p = 0.48$  Btu/lb-deg F.

Part B: Make calculations indicated, obtaining necessary values from Part A, and record the answers.

## ► Final Calculations

Step 1. Choose a suitable value for  $T_2$ . Temperature  $T_2$  should not be so high as to weaken the bearing material or damage the lubricant.

Step 2. Determine  $Z$  for assumed temperature  $T_2$ .

Step 3. Calculate  $A$  by dividing factor  $k_1$  from Part B by  $Z$  from Step 2.

Step 4. Obtain  $L/D$  ratio from Fig. 13 using the curve for recommended operating eccentricity ratio and the value of  $A$  in Step 3. Then, calculate  $L$  by multiplying the  $L/D$  ratio times  $D$  from Part A.

Step 5. Obtain  $\epsilon$  from Fig. 13 for value of  $A$  in Step 3 and the curve for recommended operating eccentricity ratio.

Step 6. Solve Equation 7 for  $h_o$  using  $C$  from Part A and  $\epsilon$  from Step 5.

Step 7. Obtain angle  $\theta$  from Fig. 12 for  $\epsilon$  of Step 5 and the  $L/D$  ratio obtained for Step 4.

Step 8. Obtain  $k_f$  from Fig. 15 for value of  $A$  in Step 3 and the  $L/D$  ratio obtained for Step 4. Then, calculate  $P_F$  by multiplying  $k_f$  times factor  $k_2$  from Part B.

Step 9. Obtain  $k_q$  from Fig. 17 for value of  $A$  in Step 3 and the  $L/D$  ratio obtained for Step 4. Then, calculate  $Q$  by multiplying  $k_q$  times factor  $k_3$  from Part B.

Step 10. Calculate  $T_2$  with the formula in the table. Use factors  $k_4$  and  $k_5$  from Part B, temperatures  $T_1$  and  $T_2$  from Part A, frictional horsepower  $P_F$  from Step 8, and oil-flow rate  $Q$  from Step 9. If the value of  $T_2$  calculated in Step 10 does not agree with the assumed value in Step 1, another value is assumed for  $T_2$  and the process in Part C repeated until agreement is obtained.

## ► Numerical Example

Design a cast-bronze sleeve bearing for a precision spindle operating with full-film lubrication. A well-ventilated, oil-bath lubricating system with SAE 20 motor oil is to be used.

Known values for the Design Sheet, Part A, are:  $W = 500$  lb;  $N = 1000$  rpm;  $D = 1.5$  in.;  $m = 1.025$  from Fig. 6;  $T_1 = 130$  F = 590 Rankine;  $T_4 = 100$  F = 560 Rankine;  $C = 0.8 \times 10^{-3}$  in.;  $K_1 = 16$  from Table 1;  $s = 0.63$  from Table 1;  $\gamma = 7.5$  lb per gal;  $c_p =$

0.48 Btu/lb-deg F;  $W_m = 2$  lb.

Calculations for Part B are:  $k_1 = (1.025)^2(500)/(1.5)^2(1000) = 0.234$ ;  $k_2 = 1.025(1.5)(1000)(500)(10^{-6}) = 0.768$ ;  $k_3 = 1.025(1.5)^3(1000)(10^{-6}) = 3.46 \times 10^{-3}$ ;  $k_4 = 15 \times 10^{-6}(16)/(0.63 + 1) = 0.147 \times 10^{-3}$ ;  $k_5 = 7.5(0.48)/42.4 = 0.085$ .

Values for the complete design are tabulated for Part C.

Explanation: First entry for Step 1 in the table for Part C is an assumed value for  $T_2$ . Since the bearing will be oil-bath lubricated, a reasonable temperature rise would be 10 F, or  $T_2 = T_1 + 10 = 140$  F, or  $T_2 = 600$  Rankine.

First entry for Step 2 is obtained from the manufacturer of the SAE 20 oil for  $T_2$  in Step 1. A later article illustrates how  $Z$  can also be determined if complete lubricant information is not available from the manufacturer.

First entry for Step 3 is  $0.234/25 = 0.0094$ .

For Step 4, the  $L/D$  ratio is obtained from Fig. 13 for the value of  $A$  in Step 3 and the curve for recommended operating eccentricity ratio. This  $L/D$  value is noted in the table. Length is then  $0.7(1.5) = 1.05$ .

First entry for Step 5 is obtained from Fig. 13 for the value of  $A$  in Step 3 and the curve for recommended operating eccentricity ratio.

First entry for Step 6, calculated from Equation 7, is  $0.8 \times 10^{-3}(1 - 0.62) = 0.304 \times 10^{-3}$ . This is an acceptable minimum film thickness.

First entry for Step 7 is obtained from Fig. 12 for the  $L/D$  ratio noted in Step 4 and the value of  $\epsilon$  in Step 5.

For Step 8,  $k_f$  is obtained from Fig. 15 for the value of  $A$  in Step 3 and the  $L/D$  ratio noted in Step 4. Frictional horsepower is then  $0.036(0.768) = 0.0276$ .

For Step 9,  $k_q$  is obtained from Fig. 17 for the value of  $A$  in Step 3 and the  $L/D$  ratio noted in Step 4. Oil-flow rate is then  $2.60(3.46 \times 10^{-3}) = 9.00 \times 10^{-3}$ .

First entry for Step 10, calculated from the equation in Part C, Table 2 is  $(0.0276 + 0.147 \times 10^{-3} \times 1.05 \times 1.5 \times 560 + 0.085 \times 9.00 \times 10^{-3} \times 590)/(0.147 \times 10^{-3} \times 1.05 \times 1.5 + 0.085 \times 9.00 \times 10^{-3}) = 611$ .

Since this value for  $T_2$  does not agree with the assumed value of Step 1, a new assumption must be made. The second assumption,  $T_2 = 606$  Rankine, is entered for Step 1. Proceeding through the calculations as just described gives a calculated value of 608 Rankine for Step 10. A third assumption is made and the calculated value of 607 Rankine matches the assumed value of  $T_2$ . Hence, the third cycle represents a solution, and the results obtained for the other values are valid.

Bearing bore diameter  $D_B$  will be  $D + 2C = 1.5 + 2(0.8 \times 10^{-3}) = 1.5016$  in.

Center of the operating journal, from Equation 6, will be distance  $e$  from center of the bearing:  $e = 0.61(0.8 \times 10^{-3}) = 0.488 \times 10^{-3}$  in.

Theoretical oil-flow rate that will support the journal in its calculated position is 0.00934 g.p.m., or 1060 drops per min. Minimum oil-flow rate for full-film lubrication, from Equation 10b, is  $Q' = 3.32 \times 10^{-3}(1.20 + 0.0043 \times 500/1.5)(1.025)^2(1000) = 20.2$  drops per minute.

Thus, if the oil supply rate is reduced to approximately 20 drops per min., full-film lubrication may still be possible. However, the journal will nearly be touching the bearing, and  $T_2$  will be much larger than the assumed value of 607 Rankine because, in this problem, lubricant is used to remove most of the frictional heat generated within the bearing. As oil flow decreases, the bearing runs hotter. Increased operating temperatures will decrease the absolute viscosity of the lubricant, which, in turn, will cause the journal to operate eccentrically. Likewise, if oil-flow is reduced to a low rate, sufficient lubricant may not be available within the bearing to fill the clearance volume. Hence, the journal must operate more eccentrically.

Step No.		First Assumption	Second Assumption	Third Assumption
1.	$T_2$	600	606	607
2.	$Z$	25	22	21
3.	$A$	0.0094 ( $L/D = 0.7$ )	0.0106 ( $L/D = 0.77$ )	0.01114 ( $L/D = 0.80$ )
4.	$L$	1.05	1.155	1.20
5.	$\epsilon$	0.62	0.61	0.60
6.	$h_o$	$0.304 \times 10^{-3}$	$0.312 \times 10^{-3}$	$0.32 \times 10^{-3}$
7.	$\theta$	48	49	50
8.	$P_F$	0.0276	0.02611	0.02534
9.	$Q$	$9.00 \times 10^{-3}$	$9.20 \times 10^{-3}$	$9.34 \times 10^{-3}$
10.	$T_2$	611	608	607



faces that are approaching each other. Time is required for the load to force these surfaces to meet. During that interval, a pressure is developed because the lubricant resists extrusion, and the load is actually supported by the oil film. Thus, if the load is of short enough duration, such as a shock load or a rotating load, the two surfaces will not meet before the load is removed. When the load is removed or reversed, the oil film can often recover its thickness in time for the next load application if the bearing is designed to permit and assist this buildup. However, indiscriminate location of oil holes, oil grooves, and reliefs may interfere with restoration of the oil film and thus destroy the major portion of the load-carrying capacity of the bearing.

The name used to describe this type of lubrication is "squeeze film." Requirements for successful squeeze-film lubrication are:

1. Copious amounts of lubricant with sufficient viscosity to resist being squeezed out of the clearance space.
2. Short duration of shock loads to avoid metal-to-metal contact. A reciprocating load or rotating load fulfills this requirement.
3. Absence of oil grooves or holes in the load-carrying region to avoid reducing the squeeze-film pressure developed.
4. Bearing materials which readily conform to shock loads without deforming, thus distributing load and squeeze film over a greater area. Cast bronze is an ideal material for this type of application.

Development of a squeeze film does not depend on rotational speed, and hence squeeze films may be used to good advantage when there is no relative sliding velocity.

Design of a bronze bearing which requires squeeze-film lubrication, because of slow (or zero) relative speed or extremely large loads, can be attempted if magnitude of load  $W_s$  and its duration,  $\Delta t$ , are known. Both these quantities are assumed to be constant. If load varies, as it would for a rotating load, some average load magnitude and time must be determined. With duration and magnitude of load known, the required length of bronze bearing may be determined with

$$L = \frac{(\Delta t)m^2W_s}{k_s DZ} \quad (14)$$

Shock-load factor  $k_s$  depends upon lubricant film thickness both before and at the end of the shock load. Fig. 19 shows values for  $k_s$  as determined by journal eccentricity ratio before the shock load,  $\epsilon_0$ , and journal eccentricity ratio at the end of the shock load,  $\epsilon_1$ . Use of Fig. 19 is illustrated for a journal that operates with an initial eccentricity ratio of  $\epsilon_0 = 0$  and is required to have an eccentricity ratio of 0.60 at the end of the shock load. Enter the graph along the bottom at  $\epsilon_1 = 0.60$  and move up until curve  $\epsilon_0 = 0$  is intersected. Then move horizontally to the  $k_s$  scale and read 1.43. With  $k_s$  and the magnitude and duration of shock load known, the required length of bearing can be com-

puted using Equation 14.

Curves in Fig. 19 can also indicate whether or not a given bearing will carry a given shock load.

EXAMPLE: A bronze bearing with 1 in. diam, 1 in. length, and  $m = 1.0$  operates at a speed of 1000 rpm with a steady load of 100 lb and a lubricant viscosity of  $Z = 10$  centipoise. Determine lubricant film thickness at the end of a shock load of 500 lb applied in addition to the steady load for 0.05 second.

First, calculate the bearing characteristic number:

$$A = \frac{(1)^2(100)}{(1)^2(10)(1000)} = 0.01$$

From Fig. 13, for  $L/D = 1.0$ ,  $\epsilon$  has a value of 0.42. Let this value be  $\epsilon_0$ . Then, using Equation 14,

$$k_s = \frac{(\Delta t)m^2W_s}{DLZ} = \frac{(0.05)(1)^2(500)}{1 \times 1 \times 10} = 2.5$$

Enter Fig. 19 at  $k_s = 2.5$  and move horizontally to an interpolated curve for  $\epsilon_0 = 0.42$ . Then, directly below on the  $\epsilon_1$  scale, read  $\epsilon_1 = 0.83$ . Therefore, minimum film thickness will be

$$h_0 = 0.5 \times 10^{-3}(1 - 0.83) = 0.000085$$

This is an acceptable value for minimum film under shock loading.

Some error is introduced when short bearings with a small  $L/D$  ratio are considered because the assumption was made in the derivation of  $k_s$  that lubricant does not flow in the axial direction. However, shock load factors in Fig. 19 can be used with reasonable accuracy for bronze bearings having  $L/D$  ratios as low as 0.5.

## ► Lightly Loaded, High-Speed Bearings

An unstable condition sometimes experienced by lightly loaded, high-speed bronze sleeve bearings is known as "half-frequency whirl." This phenomenon is very serious and troublesome, especially with vertical guide bearings, and results from instability of the oil film. Half-frequency whirl occurs when the shaft starts to whirl around in the clearance space. If the condition becomes serious enough, shaft and bearing may destroy themselves. This type of whirl is defined as a condition in which the center of the journal rotates or orbits about the center of the bearing at a frequency equal to approximately one-half the rotational or spin speed of the shaft.

Where speed of the shaft is less than the first critical speed, half-frequency whirl will begin to occur at a shaft speed determined approximately by

$$N_c = k_c \times 10^3 \left[ \frac{DNZ}{m^3 W_m} \right]^{1/4} \quad (15)$$

Whirl-speed factor  $k_c$  is obtained from Fig. 20. Enter the chart at the bottom for the proper value of bearing characteristic number, proceed upward to the desired curve of  $L/D$ , and then move horizontally to read the value of  $k_c$  on the vertical scale.

If the computed value of  $N_c$  is greater than the contemplated operating speed, the journal will not suffer from half-frequency whirl. However, if  $N_c$  is less than the desired operating speed, the journal

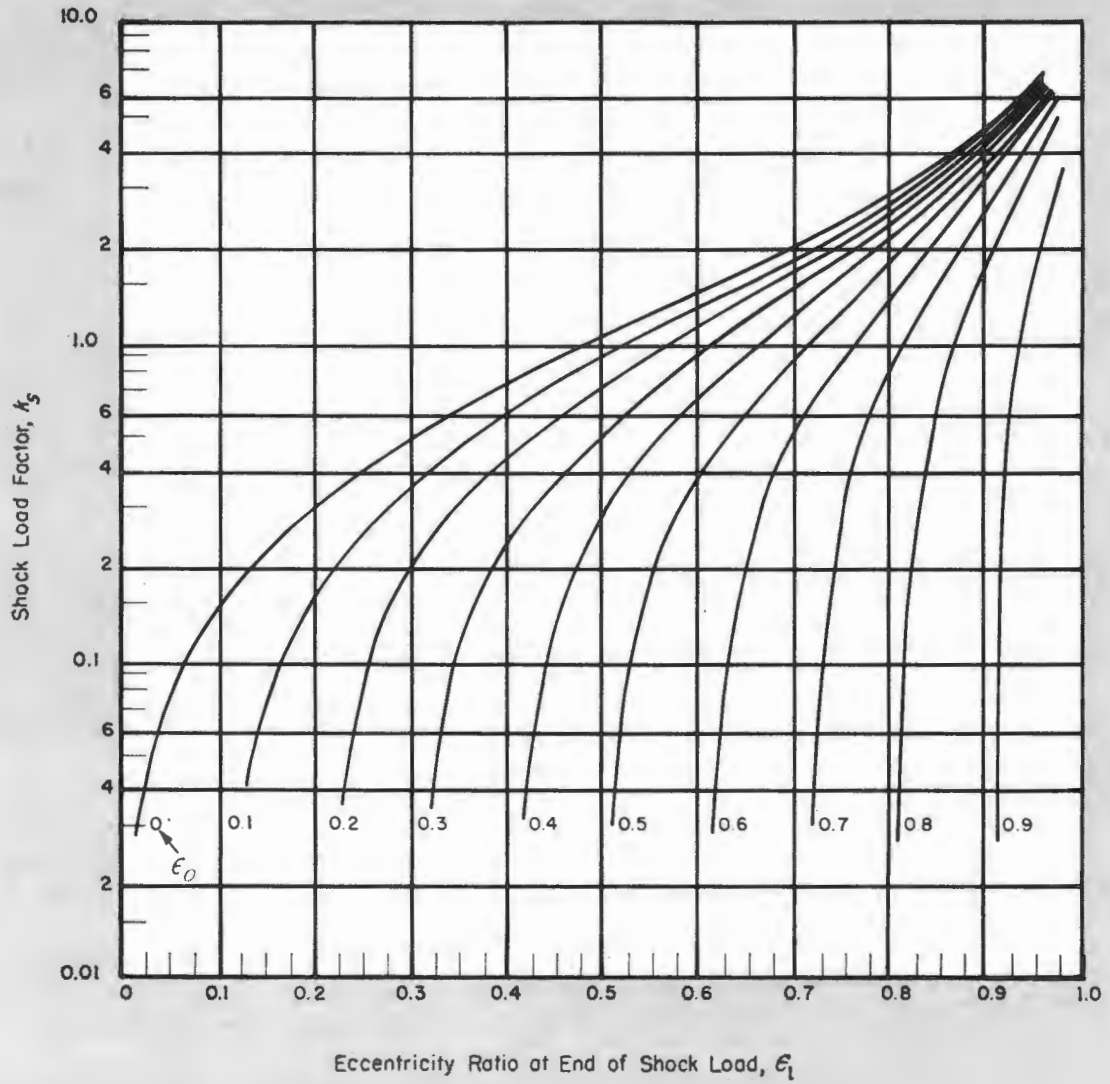
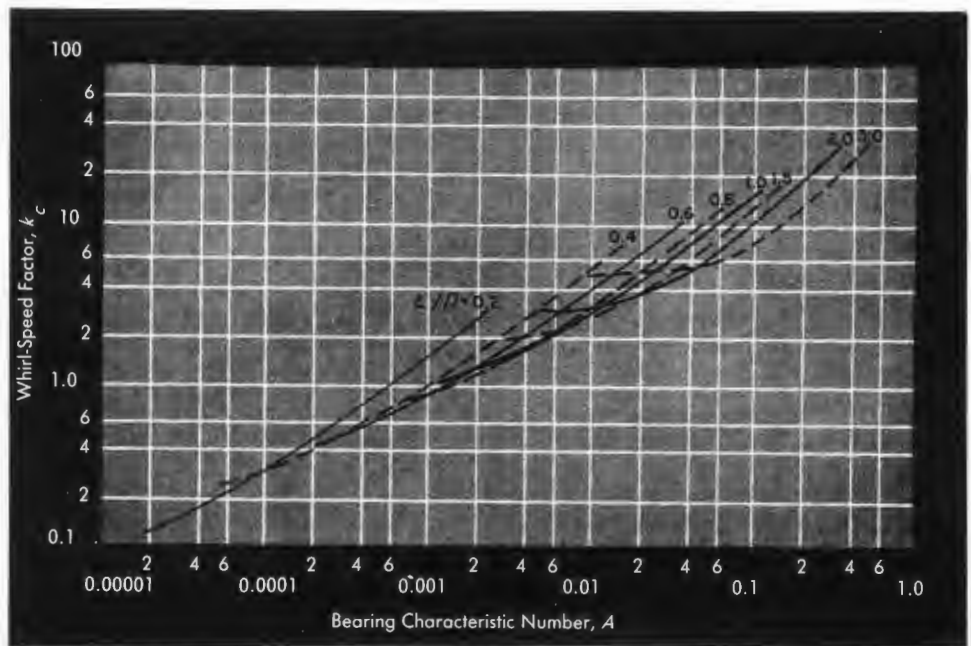


Fig. 19—Shock-load factors at various initial and final eccentricity ratios for 180 and 360-deg bronze bearings.

Fig. 20—Whirl-speed factors for 360-deg full-film lubricated bronze bearings having various  $L/D$  ratios.



will begin to whirl when the spin speed is approximately equal to  $N_c$ . Further increase in speed beyond  $N_c$  will cause the journal to whirl more severely with the result that stable operation at desired operating speed  $N$  will not be obtainable. If such is the case, the bearing will have to be redesigned and re-evaluated until  $N_c$  is sufficiently above the operating range. In general, if the shaded region in the lower left corner of Fig. 13 is avoided, the bearing should not be susceptible to half-frequency whirl.

Cast bronze bearings can readily be applied to high-speed applications. Properly designed, they are as efficient as, and in some cases more efficient than, other types of bearings.

## ► Hydrostatic Lubrication

The method of lubrication discussed up to this point has dealt with hydrodynamically lubricated bearings. In hydrodynamic bearings the fluid pressure needed to support the load is generated within the bearing by relative motion of the bearing members. Load-carrying capacity therefore depends upon relative shaft speed.

In applications where loads are high and speeds are low, hydrodynamic lubrication may be impossible. When this is true, and when full-fluid-film lubrication with no metal-to-metal contact is still desired, hydrostatic lubricating may be used.

A hydrostatically lubricated bronze bearing receives high-pressure lubricant from an external source. Lubricant is injected into a recess in the load zone of the bearing. Supply pressures required are usually far in excess of those normally used to supply lubricant to hydrodynamic bearings. Externally supplied pressure is sufficient to "float" the shaft with respect to the bearing. A thick, fluid film therefore separates shaft from bearing even at zero speed. In many cases where high starting loads exist, the load is hydrostatically supported until the unit is up to speed. External pressure is then removed and the load is supported hydrodynamically. At other times when speeds are low, hydrostatic pressure is applied continuously.

Usually, hydrodynamic bearings can depend on boundary and mixed-film lubrication for their starting periods, especially when loads are speed-dependent. If a bronze sleeve bearing is either very highly loaded at start-up or does not have sufficient speed to maintain full hydrodynamic film, high wear rates, high temperatures, and shortened bearing life may result.

Determination of the performance of hydrostatic bearings is a specialized aspect of lubrication. Full coverage of hydrostatic lubrication is presently beyond the scope of this Manual. However, many authors and investigators have done work in the field of hydrostatic bearings. Fuller<sup>2</sup> covers hydrostatic lubrication quite thoroughly. Loeb<sup>4</sup> determines characteristics of hydrostatic bearings by using an electric analog approach. Loeb and Rippel<sup>5</sup> describe methods for determining optimum proportions for bearings of this type.

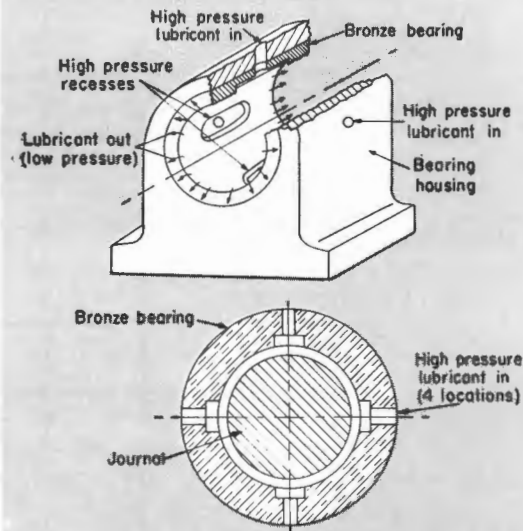


Fig. 21—Typical hydrostatic bronze bearing.

Some very distinct advantages that hydrostatic bronze bearings have over bearings of other types are:

1. High load-carrying capacity at low speed.
2. Extremely low running friction.
3. Zero starting friction (at zero speed).
4. High stiffness.
5. High reliability.
6. Predictable load-carrying capacity.
7. Almost infinite life.

Coefficients of friction much lower than the 0.001 of hydrodynamic and rolling-element bearings are possible with hydrostatic lubrication.

A typical hydrostatic bronze bearing with several high-pressure recesses is shown in Fig. 21. Lubricant under pressure is pumped into the bearing recesses and flows out across the fitted area of the bearing. All hydrostatic bearings may be analyzed by using two equations which relate load-carrying capacity, recess pressure, film thickness, viscosity, and flow. However, these equations contain constants which must be evaluated for each bearing size and configuration.

Hydrostatic bearings have been applied with a great deal of success in rolling mills, machine tools, radio telescopes, optical telescopes, and other heavily loaded, slow-moving equipment. However, specialized techniques, which include thorough knowledge of hydraulic components accompanying the bearing package are required. The designer is cautioned against pursuing the design of bearings of this type without a full knowledge of all aspects of the problem.

## CHAPTER 3

# Complete Boundary Lubrication

SINCE actual lubrication in boundary-lubricated sleeve bearings is practically nil, "boundary friction" might be a better name for this section. However, little as it may be, some lubrication is essential for the successful operation of a boundary-lubricated bearing. Before such a bearing can be designed, some understanding of the mechanism of boundary friction and lubrication must be gained.

Discussion which follows applies only to "boundary" friction, or the rubbing together of two surfaces without either a complete or a partial intervening oil film. This type of lubrication exists in one form or another in almost every kind of bearing. Even in well-designed and lubricated full-film bearings, some boundary friction must occur when starting and stopping.

Greases are very frequently used as boundary lubricants. Hence, the following discussion applies equally well to grease-lubricated sleeve bearings. Even though greases have a considerably higher viscosity than mineral oils, they do not allow a full film of lubricant to separate the journal from the bearing because the grease, too, is eventually forced from between the moving surfaces, and boundary friction will occur before reapplication of the grease.

### ► Mechanism of Boundary Friction

Successful boundary lubrication of sleeve bearings requires the use of a suitable lubricant with proper journal and bearing materials. The correct combination will produce one or both of these effects: 1. A strong affinity for metal surfaces such that those molecules adjacent to the metal hold their position and greatly resist being displaced. 2. Formation of a soap film that is bound to the metal surfaces by the chemical reaction which occurs between lubricant and bearing and journal metals.

**Fatty-Acid Lubricating Agents:** Animal, vegetable, and marine fats and oils are superior to plain mineral oil as boundary lubricants. The ingredient common to all good boundary lubricants is some kind of fatty acid which occurs in chemical combination with glycerine or other high-molecular-weight alcohol. These fatty acids are often called "oiliness" agents. The three most important fatty acids used to enhance boundary-lubricating ability of lubricants are:

1. Stearic acid, as contained in lard oil and beef and mutton tallow.
2. Palmitic acid, a principal ingredient of cottonseed oil, palm oil, and animal and marine oils.
3. Oleic acid, which is found in high percentages in almost all animal and vegetable oils.

When a small amount of one of these fatty acids is added to mineral oils, boundary-lubrication friction values decrease very markedly. Hence, if the bearing is to operate under conditions of complete boundary lubrication using oil, a fatty-acid type additive should be specified for the oil. If a grease is to be used, no fatty acid need be specified be-

### Nomenclature

- $D$  = Journal diameter, in.  
 $f_n$  = Coefficient of friction for complete boundary lubrication  
 $L$  = Bearing length, in.  
 $L_{max}$  = Maximum bearing length for complete boundary lubrication, in.  
 $N$  = Rotational speed of journal, rpm  
 $P_B$  = Frictional horsepower generated within complete boundary lubricated bearing, hp  
 $T_2$  = Lubricant film temperature for full-film lubrication, or bearing bore temperature for mixed-film and boundary lubrication, F  
 $T_4$  = Ambient atmosphere temperature, F  
 $W$  = Steady load to be supported, lb

cause greases normally contain fatty acids in their composition.

The action of a fatty acid in reducing friction under boundary conditions is generally attributed to molecular adherence. The fatty acid adheres to the metal surface with sufficient strength to resist being torn off when the rubbing surfaces slide over each other. A simplified explanation is that the fatty-acid molecules orient themselves on the metal surface such that they all stand up like the pile of a carpet, Fig. 22. The molecular layers actually isolate the two metal surfaces and friction, which would be high in the absence of lubricant, is substantially reduced.

Present research indicates that a chemical reaction occurs between the fatty acid and the metal involved. Product of this reaction is a soap film that is chemically bound to the metal surface. Thus, a fatty acid is most effective as a friction reducer when the nature of the metal permits a chemical reaction. Bronze is classed as a reactive material and readily combines with fatty acids to produce low-shear-strength metallic soaps.

When a fatty acid is used with a reactive metal, breakdown of the extremely thin lubricating film does not occur at the melting temperature of the fatty acid but at a considerably higher temperature. Actual breakdown temperature depends on the nature of the metal and on the load and speed of sliding. It corresponds approximately to the stage at which the metallic soap film softens or melts. When breakdown temperature is reached, the fatty acid loses its boundary-lubricating properties, and the coefficient of friction increases, as indicated by the "fatty acid" curve in Fig. 23. Breakdown temperature, of course, depends upon the particular metallic soap formed by the reaction of fatty acid and bearing metal.

Rapid transition to the no-lubricating-film condition at breakdown is not desirable, since a temperature variation of only a few degrees could seriously affect performance of the bearing. Hence, when a boundary-lubricated sleeve bearing is expected to operate at elevated temperature or at high speed with high load, probable breakdown temperature of the metallic soap must be considered. The best source for this sort of information, once

the particular lubricant and bronze bearing material are known, is the lubricant manufacturer. After returning to operating temperatures below the transition temperature, the fatty acid regains its boundary lubricating qualities.

**Extreme-Pressure Lubricants:** As has been stated, when bearing loads, speeds, and ambient temperature conditions combine to give surface temperatures ranging from 250 to 300 F, fatty acids and their products undergo thermal decomposition with a resulting increase in friction and surface damage. Therefore, under such conditions, some new kind of low-shear-strength film that is more stable must be formed.

The need for a suitable lubricant for high-speed hypoid gears led to the development of a class of lubricants containing extreme-pressure additives. These additives are called extreme-pressure lubricants, or just EP lubricants. Active chemicals that have proved to be effective in forming an organic film of low shear strength on the metal bearing surfaces are chlorine (chlorinated esters, etc.), sulfur (sulfurized lard oil, etc.), and phosphorus (tricresyl phosphate, etc.). In general, compounds containing these elements are used as additives to form, through reaction with metal surfaces, chlorides, sulfides, and phosphides. Such surfaces have relatively low shear strength so that rubbing between contacting surfaces occurs in the low-shear-strength surface film and thus protects the base metal. The surface films also have a relatively high melting point (iron sulfide, 2150 F; iron chloride, 1200 F) and will remain on the rubbing surfaces even at high contact temperatures.

However, if surface temperatures are below a 200 to 350 F range, the reaction of extreme-pressure additives with metal surfaces does not take place very rapidly. Consequently, as indicated in Fig. 23, until some temperature favorable to promoting the reaction is attained, these substances may prove relatively ineffective as boundary lubricants. For this reason, a small quantity of fatty acid is often included in the lubricant to provide effective lubrication at temperatures below the reaction temperature of the chemical additive. Friction characteristics of

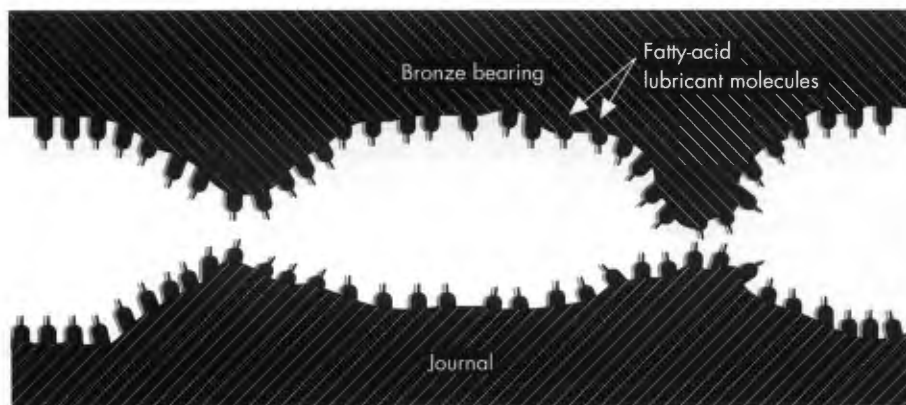
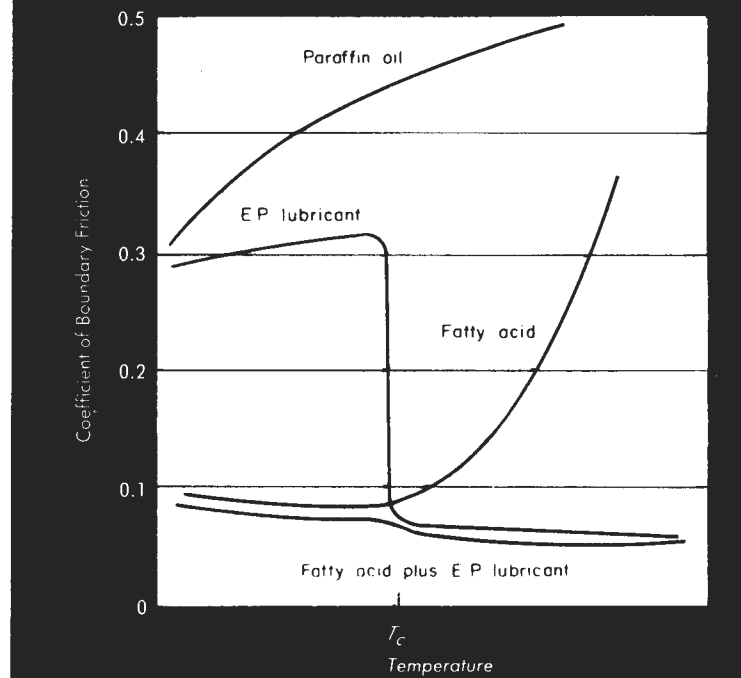


Fig. 22—Fatty-acid lubricant molecules adhering to bearing and journal surface.

Fig. 23—Variation of coefficients of friction with temperature for several lubricants under complete boundary-lubrication conditions. (After Bowden.)



this "ideal" lubricant and characteristics for normal paraffin oil are also shown in Fig. 23.

The curve in Fig. 23 for a fatty acid is typical. The acid reacts with the metal surface to form a metallic soap. At the melting point of the soap, friction increases. The EP additive in a lubricant reacts slowly below some critical temperature,  $T_c$ , so that up to this temperature, lubrication is poor. Above this temperature, a protective, low-shear-strength, high-melting-temperature film is formed, and effective boundary lubrication is provided up to a very high temperature. The lowest curve is an idealized result when some fatty acid is added to an EP lubricant. Good boundary lubrication is provided by the fatty acid below  $T_c$ . Above  $T_c$ , the effectiveness of boundary lubrication is attributed chiefly to the EP additive.

Care should be exercised in the type of extreme-pressure compound selected. If reaction rate between lubricant additive and metal surface is too rapid, more harm than good may result. Continuous fast-reaction rates will lead to chemical corrosion. Hence, the EP additive selected should give rise to chemical reaction only at temperatures or pressures where welding and tearing of bearing surfaces becomes so imminent that high wear and subsequent seizure are likely to result.

As a brief review, major facts of boundary friction and lubrication are:

1. Unmodified mineral oils are not good boundary lubricants.

2. Fatty acids are added to a lubricant to improve its boundary lubricating properties by: 1. Adhering very strongly to the surfaces, thereby forming very thin films which reduce friction and prevent metal-to-metal contact. 2. Combining chemically with the bearing materials to form low-shear-strength metallic soaps whose maximum operating temperature is governed by breakdown temperature

of the metallic salt.

3. Extreme-pressure additives are used to increase permissible surface temperatures for satisfactory boundary lubrication. Reaction rate should be controlled to prevent undue chemical corrosion.

### ► Designing for Complete Boundary Lubrication

Since design of complete boundary lubricated bronze bearings follows the same lines as the design of mixed-film lubricated bronze bearings, complete discussion of the procedure will be presented in the next section, *Mixed-Film Lubrication*.

Required length for complete boundary lubricated sleeve bearings is determined from\*

$$L_{max} = \frac{f_B N W}{15.28(T_2 - T_4)} \quad (16)$$

This equation yields satisfactory results where continuous bearing operation under conditions of complete boundary lubrication at given temperature, load, and speed is required. If operation is intermittent with regard to load and speed, shorter bearing lengths than indicated by Equation 16 may be used. Also, since diameter of the bearing does not enter into Equation 16, some limit on allowable  $L/D$  ratio should be set. In general,  $L/D$  ratios larger than 4 to 1 should be avoided.

Horsepower required to overcome friction in a complete boundary-lubricated sleeve bearing is obtained with\*

$$P_B = 7.87 \times 10^{-6} f_B D N W \quad (17)$$

Clearance ratios for boundary-lubricated bronze bearings are similar to those used for full-film lubricated bronze bearings.

\*Derivation and correct application of Equations 16 and 17 are explained in Chapter 4.

## Nomenclature

- $A$  = Bearing characteristic number  
 $A_1, A_2$  = Areas of heat dissipation  
 $C$  = Radial clearance, in.  
 $D$  = Journal diameter, in.  
 $F$  = Bearing friction force, lb  
 $f'$  = Minimum coefficient of friction for mixed-film lubrication  
 $f_B$  = Coefficient of friction for complete boundary lubrication  
 $f_F$  = Coefficient of friction for full-film lubrication  
 $f_M$  = Coefficient of friction for mixed-film lubrication  
 $K', K''$  = Coefficients of heat transfer  
 $k_L$  = Bearing length factor for mixed-film lubrication  
 $k'_L$  = Minimum bearing length factor for mixed-film lubrication  
 $k_6, k_7$  = Simplifying factors  
 $k_9, k_{10}, k_{11}$  = Factors used in Table 3  
 $L$  = Bearing length, in.  
 $L_{max}$  = Maximum bearing length for complete boundary lubrication, in.  
 $L_{min}$  = Minimum bearing length for fluid-film lubrication, in.  
 $m$  = Clearance factor  
 $= 1000(2C)/D$   
 $N$  = Rotational speed of journal, rpm  
 $P_M$  = Frictional horsepower generated within mixed-film lubricated bearing, hp  
 $p$  = Projected area unit load, psi  
 $Q$  = Side-leakage oil flow, or oil-flow feed rate, gpm or drops per min  
 $Q'$  = Minimum oil flow required for full-film lubrication, gpm or drops per min  
 $Q'_{min}$  = Minimum oil flow required for full-film lubrication when bearing length is  $L_{min}$ , gpm or drops per min  
 $T_2$  = Lubricant film temperature for full-film lubrication, or bearing bore temperature for mixed-film and boundary lubrication, F  
 $T_3$  = Surface temperature of bearing housing, F  
 $T_4$  = Ambient atmosphere temperature, F  
 $V$  = Shaft surface velocity, fpm  
 $W$  = Steady load to be supported, lb  
 $W_B$  = Load supported by lubricant film under complete boundary lubricating conditions, lb  
 $W_F$  = Load supported by lubricant film under full-film lubricating conditions, lb

**T**HOUSANDS of bronze sleeve bearings operate under mixed-film conditions, either by intent or by chance. Lubrication by frequent hand oiling, mechanical feed, wick feed, waste pack, or drop feed usually results in mixed-film operation. Bearings operate under mixed-film conditions because: 1. Load is too large. 2. Speed is too low. 3. Viscosity is too low. 4. Lubricant is restrained.

The first three reasons result in a large value for bearing characteristic number,  $A$ . For the fourth reason, rate of lubricant application is below minimum requirements, or  $Q < Q'$ .

### ► Mechanism of Mixed Friction

In a bronze sleeve bearing operating under conditions of mixed-film lubrication, part of the total load carried by the bearing is supported on a boundary film in the areas of closest approach between journal and bearing. The remaining portion of the load is supported by hydrodynamic pressure developed in the oil-filled, depressed regions of apparent contact area. Hence, total friction encountered when operating under conditions of mixed-film lubrication depends upon the coefficient of friction for boundary friction,  $f_B$ , and the coefficient of friction for fluid friction,  $f_F$ .

As previously mentioned in the discussion of Fig. 3, the value of  $f_B$  ranges from 0.08 to 0.14 depending upon the combination of lubricant and bearing materials. For the case of fluid friction, a safe assumption is that coefficient  $f_F$  will be on the order of 0.002. Minimum coefficient of mixed friction,  $f'$ , is approximately equal to 0.020, while the maximum coefficient of mixed friction approaches the coefficient of boundary friction. When all the load is supported on a boundary film because of either lack of lubricant or slow speed, both of which prevent build-up of hydrodynamic pressure to support a portion of the load, boundary friction constitutes the total friction encountered.

Based on the assumption that load is shared between a boundary film and many partial fluid films,

$$W = W_B + W_F$$

## CHAPTER 4

# Mixed-Film Lubrication

Total friction force will then be total load times coefficient of mixed friction and is the sum of boundary friction and fluid friction forces, or,

$$f_M W = f_B W_B + f_F W_F$$

By combining the two previous equations, the solution for coefficient of mixed-film lubrication becomes

$$f_M = f_F + (f_B - f_F) \frac{W_B}{W} \quad (18)$$

If all the load is carried on a boundary film, then  $W = W_B$ ,  $W_B/W = 1$ , and  $f_M = f_B$ . Conversely, if all the load is carried on a fluid film and there is no boundary friction,  $W_B = 0$  and  $f_M = f_F$ . Between these two end conditions, ratio  $W_B/W$  must be determined to evaluate  $f_M$ .

As discussed in Chapter 2, a minimum flow rate of lubricant,  $Q'$ , must be supplied to a given sleeve bearing subjected to a given load and speed to permit it to operate hydrodynamically. If less than this minimum flow rate is supplied, some boundary friction will occur and an immediate increase in bearing friction will signify the onset of mixed-film conditions. As oil-feed rate is further reduced, the logical assumption is that ratio  $W_B/W$  will increase since less lubricant is available. Hence, an approximate relationship between  $W_B/W$  and oil-feed rate  $Q$  can be established. A suitable equation for this relationship is

$$\frac{W_B}{W} = 1 - \frac{(f_B - f')}{(f_B - f_F)} \left( \frac{Q}{Q'} \right)^2 \quad (19)$$

If oil-feed rate is known, the coefficient of mixed friction can be obtained by substituting Equation 19 into Equation 18, or,

$$f_M = f_B - (f_B - f') \left( \frac{Q}{Q'} \right)^2 \quad (20)$$

Coefficient of boundary friction,  $f_B$ , for steel journals running on cast bronze ranges from 0.08 to 0.14, depending upon the lubricant. An observed approximate value for  $f'$  is 0.020, which becomes the approximate coefficient of mixed friction when oil-feed rate  $Q$  equals minimum oil flow  $Q'$ . Equation 10a or 10b is used to evaluate  $Q'$ .

One important variable not included in the preceding discussion is the effect of wearing-in of bearing members. If a sleeve bearing is given an opportunity to wear-in, the necessary value of  $Q'$  will probably decrease. Hence, the preceding analysis is on the safe side in that predicted coefficients of friction may be somewhat higher than actual values. Because this reduction in  $Q'$  cannot be evaluated, no attempt is made to include it in the equations.

Coefficient of mixed friction is plotted against oil-feed ratio  $Q/Q'$  for various values of coefficient of boundary friction in Fig. 24. Notice that the mini-

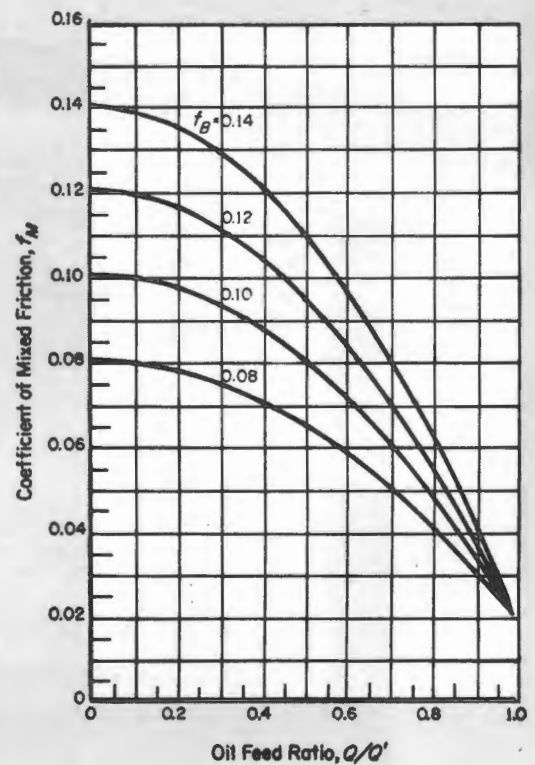


Fig. 24—Coefficients of mixed friction for bronze bearings operating with mixed-film lubrication.



### Table 3 – Design Sheet for Mixed

Part A--Known Data			
	Symbol	Value	Units
Total steady load	$W$	_____	lb
Journal speed	$N$	_____	rpm
Journal diameter	$D$	_____	in.
Clearance factor	$m$	_____	—
Bearing bore temperature	$T_2$	_____	F
Ambient atmosphere temperature	$T_4$	_____	F
Coefficient of boundary friction (estimated)	$f_B$	_____	—

Part B-- Simplified Factors		
	Calculation	Value
$k_8$	$NW/(T_2 - T_4)$	_____
$k_9$	$0.0043W/D$	_____
$k_{10}$	$(3.32 \times 10^{-3})mD^2N$	_____

Part C--Final Calculation Form				
Step No.		Original Values	Values for First Adjustment	Values for Second Adjustment
1.	$k_L$ (step 5)	$k_L = 0.00131$	_____	_____
2.	$L = k_L k_8$ , in.	$L'_{min} = 0.00131 k_8$ = _____	_____	_____
3.	$L + k_9$	_____	_____	_____
4.	$Q' = k_{10} \times$ step 3, drops per min	$Q'_{min} =$ _____	_____	_____
5.	$k_L$ (Fig. 25) for $Q =$ _____ drops per min	_____	_____	_____

# Film Lubricated Bronze Bearings

## ► Preliminary Steps

Part A: Fill in values dictated by the problem.

Select  $m$  from Fig. 6 through 10 for the specified class of machinery and operating speed.

Choose a suitable value for  $T_2$ . Temperature  $T_2$  should not be so high as to weaken the bearing material or damage the lubricant. For cast leaded bronze,  $T_2$  should never exceed 450 F if the lubricant can function properly at that temperature.

Estimate a value for  $f_B$ .

Part B: Make calculations indicated, obtaining necessary values from Part A, and record the answers.

## ► Final Calculations

Step 1. Use  $k_L = 0.00131$  in first column as indicated. Taken from Equation 23,  $k_L = f_M/15.28$ . When  $Q = Q'$ ,  $k_L = k'_L = 0.00131$  regardless of the value of  $f_B$ . Factor  $k_L$  is plotted in Fig. 25. In second and subsequent columns, copy  $k_L$  from Step 5 of preceding column.

Step 2. Calculate  $L'_{min}$  in first column as indicated. In subsequent columns, calculate  $L$ .

Step 3. Add the result of Step 2 and factor  $k_B$  from Part B.

Step 4. Multiply the result of Step 3 by factor  $k_{10}$  from Part B. Entry in first column equals  $Q'_{min}$ , which, for the given conditions, will tend to reduce boundary friction to a minimum for a bearing of length  $L'_{min}$ . If  $Q'_{min}$  is acceptable and can be continuously supplied to the bearing, and if  $L'_{min}$  is also acceptable, the design is completed. However, if bearing length is too long and will not fit the application, either full-film, hydrodynamic lubrication must be used, if possible, or bearing diameter must be varied. If oil-flow rate appears excessive, and if the  $L/D$  ratio is small and space is available for increased bearing length, compute a new value for  $k_L$  in Step 5.

Step 5. Determine a new value of  $k_L$  from Fig. 25 if permitted by conditions listed in Step 4. First select an acceptable value for  $Q$  and record it. This oil-flow rate is used to determine all subsequent new values of  $k_L$ . Then, entering Fig. 25 at the right-hand side for the selected value of  $Q$ , move horizontally left to the value of  $Q'$  obtained in Step 4. From this point, move vertically to the value of  $f_B$  estimated in Part A. Finally, move horizontally from the  $f_B$  value to the left-hand scale to obtain the new value

of  $k_L$ . Interpolate for curves  $Q'$  and  $f_B$  when necessary. Record the value of  $k_L$ .

## ► Numerical Example

Determine the length of a cast-bronze sleeve bearing for a precision spindle of hardened, ground steel. Design for mixed-film lubrication.

Known values for the Design Sheet, Part A, are:  $W = 1000$  lb;  $N = 200$  rpm;  $D = 1$  in.;  $m = 1.1$  from Fig. 6;  $T_2 = 250$  F;  $T_4 = 100$  F;  $f_B = 0.10$  estimated.

Calculations for Part B are:  $k_B = 200(1000)/(250 - 100) = 1333$ ;  $k_9 = 0.0043(1000)/1 = 4.3$ ;  $k_{10} = 3.32 \times 10^{-3}(1.1)(1)^2(200) = 0.726$ .

Values for the complete design are tabulated for Part C.

Explanation: First entry for Step 1 in the table for Part C is given for  $k'_L$ . First entry for Step 2 is  $0.00131(1333) = 1.756$ . First entry for Step 3 is the sum of 1.756 and 4.3. First entry for Step 4 is  $0.726(6.056) = 4.4$ .

Thus, if at least 4.4 drops of oil per minute are supplied to a bearing approximately  $1\frac{3}{4}$  in. long, satisfactory mixed-film operation should result. A flow rate less than 4.4 drops per min will result in more boundary friction and, hence, will require a greater bearing length.

For a complete illustration of the design procedure, suppose the flow rate can be no more than 3 drops per min. A new value of  $k_L$  is obtained from Fig. 25: Enter the graph at  $Q = 3$  and move horizontally to the location corresponding to 4.4 for  $Q'$  in Step 4. Move vertically to the curve for  $f_B = 0.10$ . Then move horizontally to the left-hand side of the graph and read  $k_L = 0.0040$ . This value is entered for Step 5 and also as the second entry for Step 1.

The process is then repeated using the new value of  $k_L$ . The new value for  $L$  is  $0.0040(1333) = 5.332$ . For Step 3,  $5.332 + 4.3 = 9.632$ , and for Step 4,  $Q' = 0.726(9.632) = 7.0$  drops per min.

Since the calculation process continues until no change occurs in two successive values of  $L$ , or, in effect, for  $k_L$ , a new value for  $k_L$  is again taken from Fig. 25: Enter the graph at the desired value of  $Q = 3$ , move horizontally to  $Q' = 7.0$ , move vertically to  $f_B = 0.10$ , and move horizontally to  $k_L = 0.0055$ .

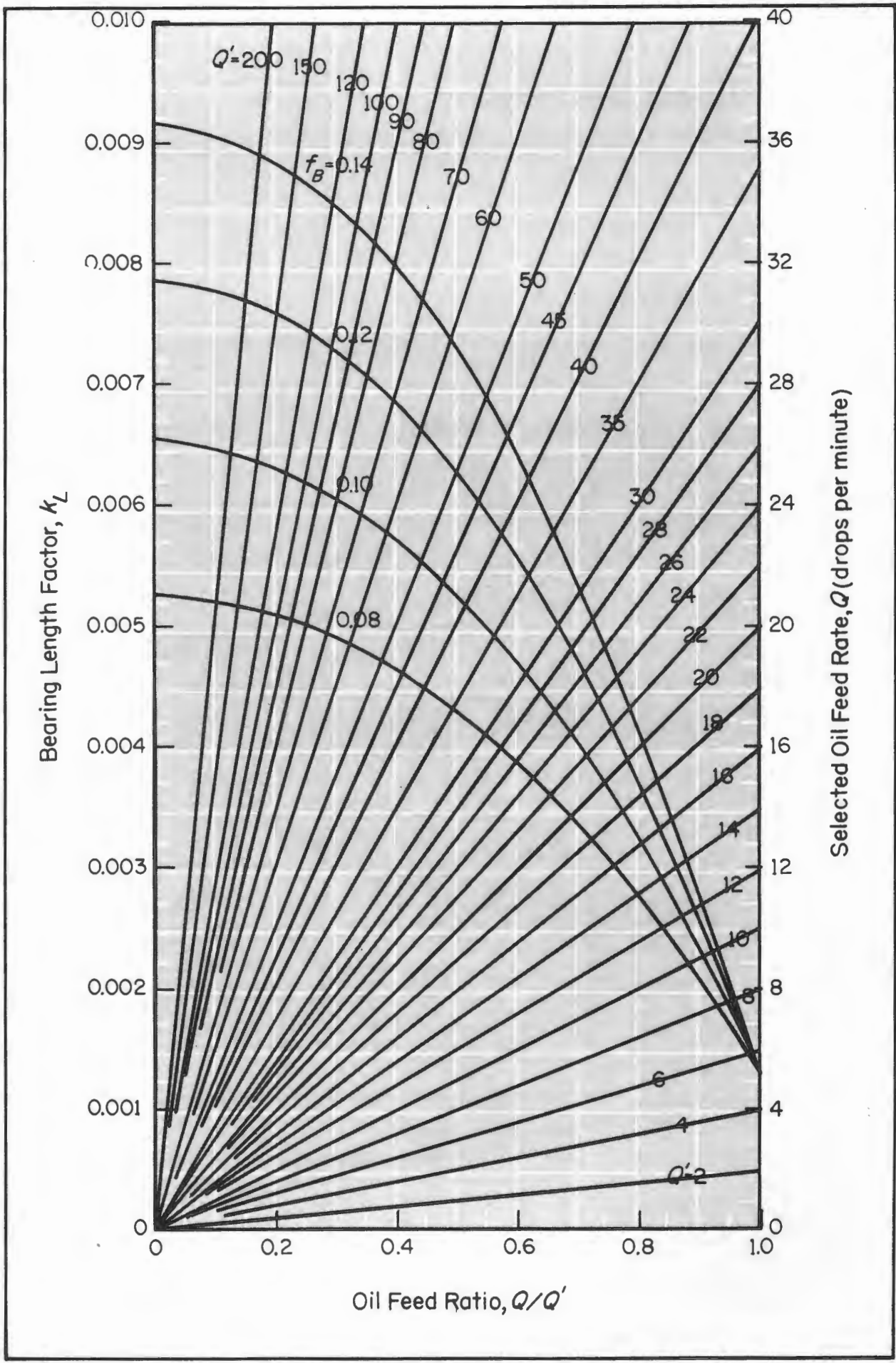
Because this last value of  $k_L$  does not equal the previously used value, 0.0040, the third set of entries must be calculated using  $k_L = 0.0055$ . Thus, entries are made in columns for the Second, Third, and Fourth Adjustment.

After the Second Adjustment is completed,  $k_L$  is 0.00585. The Third Adjustment gives  $k_L = 0.0059$ , which agrees very closely with the previous value of  $k_L$ . Value of  $L$  is also computed and is sufficiently close to the 7.79 value to indicate a solution.

Thus, a sleeve bearing of 1-in. diameter, supplied with lubricant at a rate of 3 drops per min, will give satisfactory mixed-film performance if its length is approximately 7.8 in. This length gives an  $L/D$  ratio of approximately 8, which is entirely too large. Hence, every effort should be made to reduce the length by supplying more lubricant, namely, the 4.4 drops per min. The shorter length is necessary if alignment difficulties are to be avoided. Ratios of  $L/D$  greater than 4.0 should be avoided.

This example illustrates, quite dramatically, the effect of "starving" the bearing—that is, supplying less lubricant than the minimum rate required to operate under full-film conditions. Needless to say, it is highly recommended that flow rate be at least equal to  $Q'_{min}$  if at all possible.

Step No.	Original Values	First Adjustment	Second Adjustment	Third Adjustment	Fourth Adjustment
1.	$k_L$ $k'_L = 0.00131$	0.0040	0.0055	0.00585	0.0059
2.	$L$ $L'_{min} = 1.756$	5.332	7.34	7.79	7.80
3.	$L + k_B$ 6.056	9.632	11.64	12.09	
4.	$Q'$ $Q'_{min} = 4.4$	7.0	8.16	8.76	
5.	$k_L$ 0.0040 $Q = 3$	0.0055	0.00585	0.0059	



imum value of  $f_M$  for each value of  $f_B$  equals 0.020 when  $Q/Q' = 1$ . At the extreme left-hand side of the graph, when  $Q/Q' = 0$ ,  $f_M$  equals  $f_B$  in each case and indicates complete boundary-friction conditions. Between the two limits,  $f_M$  is dictated by the oil-feed ratio and  $f_B$ .

### ► Designing for Mixed-Film Lubrication

Regardless of whether a bronze bearing is operating on a full, mixed, or boundary film of lubricant, frictional energy is generated within the bearing. Frictional heat must be removed from the bearing at a rate equal to the rate of heat generation if some steady-state operating temperature is to be obtained. Since only small flow rates are associated with boundary and mixed-film lubricated bearings, practically all heat generated within the bearing must be conducted through the bearing walls and eventually dissipated to ambient atmosphere.

Frictional energy to be dissipated is the product of force required to overcome friction in the bearing and journal surface speed or  $FV$ . If steady-state temperature conditions are to be attained,

$$FV = K' A_1 (T_2 - T_3) = K'' A_2 (T_3 - T_4)$$

where  $K'$  and  $K''$  are heat-transfer coefficients and  $A_1$  and  $A_2$  are heat-dissipating areas.

Since  $F = f_M W$  and  $A_1$  and  $A_2$  are proportional to the product of bearing length and diameter ( $A_1 = k_6 DL$ ,  $A_2 = k_7 DL$ ), by substituting for  $F$ ,  $A_1$ , and  $A_2$ ,

$$f_M W V = K' k_6 DL (T_2 - T_3) = K'' k_7 DL (T_3 - T_4)$$

$$f_M V \left( \frac{W}{DL} \right) = k_6 K' (T_2 - T_3) = k_7 K'' (T_3 - T_4)$$

When the last equation is combined with information available on temperature rises for mixed-film conditions,

$$pV = \frac{k_7 (T_2 - T_4)}{f_M} \quad (21)$$

Value of  $k_7$  depends upon heat-dissipating qualities of the bearing, but to obtain a value for  $k_7$ ,

more empirical data are required. An observed fact is that maximum permissible value of  $pV$  for mixed-film lubrication is approximately 50,000 lb-ft per in.<sup>2</sup> per min. For  $pV$  to be maximum in Equation 21, temperature difference  $T_2 - T_4$  must be maximum, and the coefficient of mixed friction must be minimum. Factor  $k_7$  remains constant, since it is only a constant of proportionality. For continuous operation on cast bronze, an acceptable upper limit for  $T_2$  would be approximately 325 F. If  $T_4$  is chosen as 75 F, then  $(T_2 - T_4)_{max}$  equals 250 F. Also, from previous discussion and Fig. 24, minimum value of  $f_M$  is 0.020. By substituting these various values into Equation 21,  $k_7$  is found to equal 4. Thus,

$$pV = \frac{4(T_2 - T_4)}{f_M} \quad (22)$$

**Bearing Length:** To make Equation 22 more useful for design purposes,  $p$  can be replaced with  $W/DL$  and  $V$  with  $\pi DN/12$ . By rearranging,

$$L = \frac{f_M N W}{15.28 (T_2 - T_4)} \quad (23)$$

The only unknown quantity is  $L$ , since  $W$ ,  $N$ , and  $T_4$  are usually known for the problem. If a bearing-bore temperature compatible with lubricant and bearing materials is assigned to  $T_2$ , and a value of  $f_M$  is computed from Equation 20, a bearing length can be determined that will satisfy the design requirements. However, evaluation of  $f_M$  may require more manipulation than has been indicated.

With  $f' = 0.020$ , the coefficient of mixed friction for use in Equation 23 is obtained from Equation 20:

$$f_M = f_B - (f_B - 0.020) \left( \frac{Q}{Q'} \right)^2 \quad (20a)$$

In Equation 20a, minimum flow for full-film lubrication  $Q'$  is obtained from Equation 10b (repeated here for convenience):

$$Q' = 3.32 \times 10^{-3} \left( L + 0.0043 \frac{W}{D} \right) m D^2 N \quad (10b)$$

To evaluate Equation 10b requires a value for  $L$ , but  $L$  at this stage is unknown. However, from Equation 20a and Fig. 24, when  $Q/Q' = 1$ ,  $f_M$  equals 0.020, and when  $Q/Q' = 0$ ,  $f_M$  equals  $f_B$ . If these values are substituted into Equation 23 for the given speed, load, and temperature limits, maximum bearing length when  $Q = 0$ , and minimum bearing length when  $Q = Q'$  can be determined.

Thus, for complete boundary lubrication,

$$L_{max} = \frac{f_B N W}{15.28 (T_2 - T_4)} \quad (16)$$

and for the other condition, that of minimum length for fluid-film lubrication,

$$L_{min} = \frac{0.00131 N W}{T_2 - T_4} \quad (24)$$

The necessary oil-feed rate that will allow use of  $L_{min}$  is obtained from Equation 10b by substituting for  $L$  the value of  $L_{min}$  as computed from Equation 24, or,

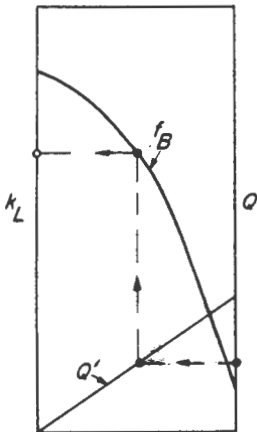


Fig. 25—Bearing-length factors for bronze sleeve bearings operating with mixed-film lubrication.

$$Q'_{min} = 3.32 \times 10^{-8} \left( L_{min} + 0.0043 \frac{W}{D} \right) mD^2N \quad (25)$$

If quantity  $Q'_{min}$  appears impractical or impossible to supply because of the high rate, an acceptable flow-rate value is assigned to  $Q$ . With this assigned flow rate, a repetitive process determines a length  $L$  that will give satisfactory service.

For initial calculations at minimum conditions,  $Q/Q' = Q/Q'_{min} = 1$ . Now, with  $Q$  selected and being smaller than  $Q'_{min}$ , ratio  $Q/Q'$  also becomes smaller, or less than 1. Consequently, as shown by Equation 20a and Fig. 24,  $f_M$  becomes larger than 0.020. Then, with  $f_M$  larger in Equation 23,  $L$  will be longer than  $L_{min}$ . For this longer value of  $L$ , Equation 10b shows that a new value for  $Q'$ , greater than the value obtained for  $Q'_{min}$ , is required.

With the assigned oil-feed rate still desired, the new ratio for  $Q/Q'$  becomes slightly smaller than the previous ratio because the new  $Q'$  is larger than the previous  $Q'$ , which, since this is the first cycle, is actually  $Q'_{min}$ . Thus, the second cycle starts with new values being determined for  $f_M$  and  $L$ . After several cycles, a solution is obtained when no further change occurs in length  $L$ .

### ► Simplified Design Method

Outlined in Table 3 is a systematic procedure for

designing bronze bearings under mixed-film conditions. All necessary entries and computations can be made in the spaces provided. Part A of Table 3 lists all data needed to compute the required bearing length. This information is known from conditions and requirements at the start of the design. Part B requires computation of factors that simplify final calculations. All information necessary to obtain these factors is taken from Part A. Part C indicates final calculations for determining bearing length  $L$ . In most cases several trials will be required before a satisfactory length is obtained, and space is indicated for initial cycles. Certain values required to complete the calculations can be easily determined from Fig. 25.

A sample problem is included that illustrates full use of the design method. Table 3 outline is used, and explanations are given throughout the procedure.

### ► Power Requirements

With required bearing length  $L$  and oil-feed ratio  $Q/Q'$  determined, expected coefficient of mixed friction can be obtained from either Equation 20a or Fig. 24. Then, power required to overcome mixed-film friction in the bearing is

$$P_M = 7.87 \times 10^{-6} f_M DNW \quad (26)$$

## CHAPTER 5

# Viscosity and Lubricants

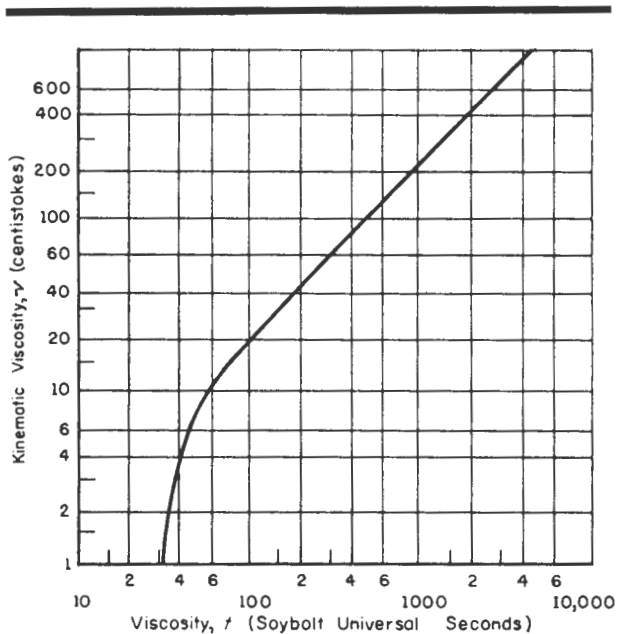


Fig. 26 — Conversion chart for kinematic viscosity and Saybolt viscosity.

### Nomenclature

$B$	= Constant for Equation 31 (Table 6)
$e$	= Base of natural logarithms = 2.718
$p$	= Pressure, psi
$T$	= Temperature, F
$t$	= Viscosity, Saybolt Universal Seconds (SUS)
$W$	= Steady load to be supported, lb
$Z$	= Lubricant absolute viscosity, centipoises
$\nu$	= Kinematic viscosity, centistokes
$\rho$	= Specific gravity, or mass density, of oil, gm per cu cm
$\rho_{API}$	= API gravity at 60 F, degrees

**T**HERE are so many methods of measuring viscosity of fluids that viscosity data can be quite confusing to a designer of bearings. To clarify the “mystery” surrounding viscosity information, simple definitions which follow present a clear picture of common terms and units. A step-by-step procedure shows how to calculate absolute viscosity, which is used for bronze sleeve bearing design.

Deciding which lubricant to use for full, mixed, or boundary-film lubrication can also be a problem. Discussion of factors to consider and a lubricant-selection chart help remove this obstacle.

Under certain conditions, oils may not provide satisfactory lubrication. Other lubricants must then be used. Descriptions of various greases and solid lubricants serve as a helpful reference when departing from conventional oils.

### ► Lubricant Viscosity

To the designer of a fluid-film bronze bearing, viscosity of the lubricant is its most important single physical property. Viscosity dictates load-carrying capacity, fluid-film thickness, operating temperature, and friction loss in the bearing.

Viscosity is defined as the internal frictional resistance offered by a fluid to any change in shape or relative motion of its parts. For example, because water pours more readily than oil, oil is said to be more viscous than water. Properly designed, bronze bearings can, and do, operate using “fluids” having viscosities that cover a range represented by

air and honey. Even a typical lubricating oil such as SAE 30 exhibits a 50-fold change in viscosity over its normal operating temperature range.

Various measures of viscosity and specific gravity are defined in Table 4.

**Viscosity Variation with Temperature:** If Saybolt viscometer data are available at two different temperatures, Saybolt viscosity at any other temperature can be determined from a plot of this information on special graph paper. Data for a typical SAE 30 oil are plotted in Fig. 27 on graph paper obtainable from American Society for Testing Materials (A.S.T.M. Chart No. D-341).

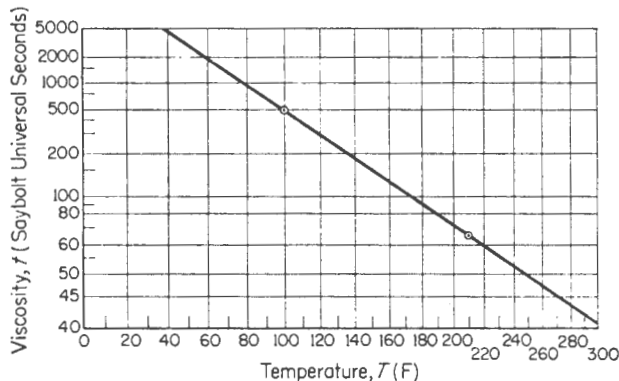


Fig. 27—Relationship of Saybolt viscosity and temperature for SAE 30 oil on sample of special ASTM graph paper. Line is determined by: 489 SUS at 100 F, and 65 SUS at 210 F. API gravity is 28.7 deg at 60 F.

A straight line drawn between the two data points gives a curve of  $t$  versus  $T$  for an appreciable temperature range. Saybolt viscosity  $t$  at any other temperature,  $T$ , can then be determined.

**Calculating Absolute Viscosity:** If specific gravity at a particular temperature and kinematic viscosity at the same temperature are known, absolute viscosity at that temperature can be computed. The sequence of operations is presented in Table 5 with given and calculated values indicated. Absolute viscosity can also be obtained from Fig. 28 if Saybolt viscosity and specific gravity have been determined. A chart similar to Fig. 28 for some other particular value of specific gravity can be plotted for Equation 30 if the designer finds it convenient.

Computations indicated in Table 5 were made for several SAE lubricants, and results are plotted in Fig. 29 for temperature  $T$ . These curves are only typical, and there may be (and, in fact, there is) considerable difference in viscosity-temperature relationship for the same SAE grade oil made by different companies. If sufficient data are available on the lubricant to be used, calculation of the absolute viscosity-temperature relationship outlined in Table 5 and a plot similar to Fig. 29 are recommended.

**Viscosity Index:** Quite often, the term "viscosity index" is used in lubrication practice to denote an arbitrary system of comparison used in evaluating

Table 4—Viscosity and Specific Gravity Definitions

**Specific Gravity:** Standard practice in the oil industry is to obtain a measure of specific gravity at 60 F on an arbitrary scale specified by American Petroleum Institute. As an example, API gravity may be expressed as "27.5 degrees at 60 F."

The relation between API degrees and specific gravity at 60 F is

$$\rho_{60} = \frac{141.5}{131.5 + \rho_{API}} \quad (27)$$

Specific gravity, or grams of mass per cubic centimeter, at some other temperature,  $T$ , is found from

$$\rho_T = \rho_{60} - 0.00035(T - 60) \quad (28)$$

Normal values of specific gravity for sleeve-bearing lubricants range from 0.75 to 0.95. Hence, if the API rating is not known, an assumed value of 0.85 may be used.

**Saybolt Viscosity:** A standard method established by American Society for Testing Materials is used by the oil industry to determine viscosity. The method requires determination of the time required for 60 cu cm of constant-temperature oil to flow through a tube 0.176 cm in diameter and 1.225 in. long. The instrument used to determine viscosity is the Saybolt Uni-

versal Viscometer, which may give such viscosity information as: 450 SUS (Saybolt Universal Seconds) at 100 F. This reading means the lubricant in question required 450 seconds to pass 60 cu cm through the Saybolt Universal Viscometer at 100 F.

**Kinematic Viscosity:** To an experienced lubricant man, SUS rating may be meaningful, but it is useless for computational purposes. A more meaningful measure of viscosity is obtained from

$$\nu = 0.22t - \frac{180}{t} \quad (29)$$

where  $\nu$  is the kinematic viscosity in centistokes and  $t$  is the number of seconds obtained from the Saybolt Universal Viscometer. For convenience, the curve of Equation 29 is plotted in Fig. 26.

**Absolute Viscosity:** Mass density of the oil must be introduced to obtain absolute viscosity:

$$Z = \rho\nu \quad (30)$$

Absolute viscosity is expressed in centipoises, with 1 centipoise being equivalent to 0.01 gram of mass per centimeter per second. Mass density in metric units is equal to specific gravity.

Step	Symbol		Calculation	Value
1.	$\rho_{API}$	Degrees API at 60F	Given	—, degrees API
2.	$\rho_{60}$	Specific gravity	$141.5 / (131.5 + \rho_{API})$	—, gm/cu cm
3.	$T$	Temperature	Selected	—, F
4.	$\rho_T$	Specific gravity	$\rho_{60} - 0.00035(T - 60)$	—, gm/cu cm
5.	$t$	Saybolt viscosity at T	Given	—, SUS
6.	$\nu_T$	Kinematic viscosity	$0.22 t - 180/t$	—, centistokes
7.	$Z_T$	Absolute viscosity	$\rho_T \nu_T$ , or from Fig. 28	—, centipoises

the relationship between viscosity and temperature. At one time it served the very useful purpose of identifying the source of an oil. Lately, it has fallen into disuse because modern developments have improved viscosity-temperature relationships. However, the index is still valuable, within limits, for expressing relative change of viscosity with temperature.

Graphical means for evaluating the viscosity index are provided by Fig. 30. Use of this chart is explained with the typical SAE 30 oil plotted in Fig. 27 as an example. Enter Fig. 30 on the horizontal scale for a Saybolt viscosity of 489 sec at 100 F. Move vertically to the Saybolt viscosity of 65 sec at 210 F, interpolating this point between the slanting lines.

Then move horizontally and read 106 as the VI rating on the vertical scale.

In general, oil having a high VI has a better temperature-viscosity relationship than oil with a low VI—better in the sense that rate of change of viscosity with temperature is less.

**Viscosity Variation with Pressure:** Many ordinary lubricants are known to undergo appreciable increase in viscosity when subjected to high pressure. In highly loaded bronze bearings, where  $W/LD > 1000$ , peak fluid pressures ranging from 10,000 to 20,000 psi are possible. Hence, the pressure effect on viscosity can be considerable. Fortunately for

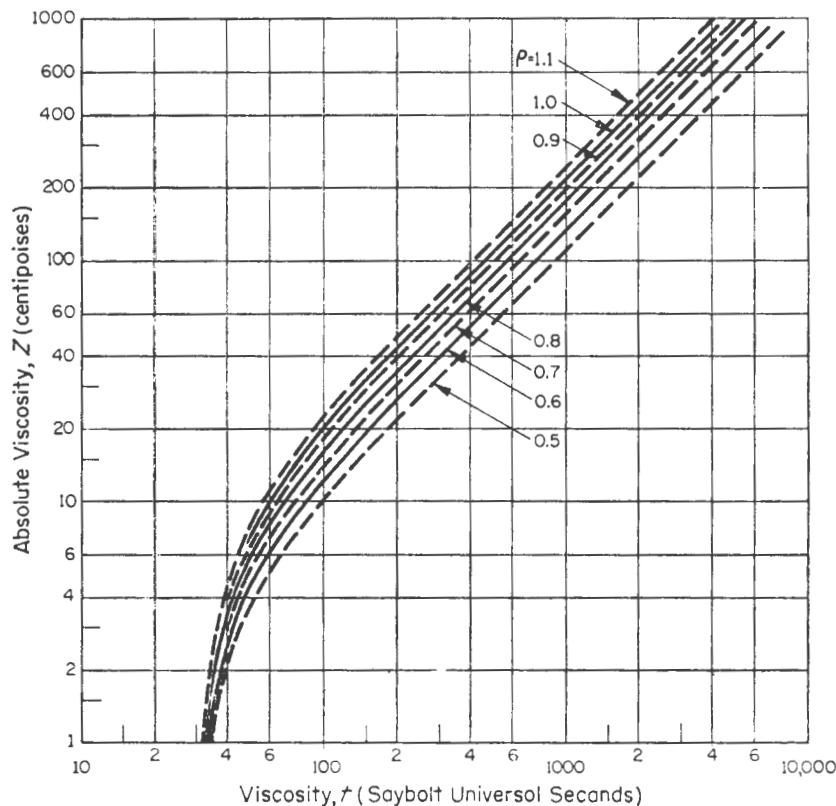


Fig. 28 — Conversion chart for absolute viscosity and Saybolt viscosity for different values of specific gravity.



the sleeve bearing, pressure causes an increase in viscosity such that load-carrying capacity of the bearing is increased.

An approximate expression that determines the new viscosity,  $Z_p$ , caused by pressure  $p$  is

$$Z_p = Ze^{Bp} \quad (31)$$

where  $e$  is a constant equal to 2.718 and  $Z$  is absolute viscosity at atmospheric pressure. Values of  $B$  for various lubricants are given in Table 6. However, the effect of increased viscosity as a result of pressure need not ordinarily be introduced.

### ► Lubricant Selection

Criteria for selecting bronze bearing lubricants are: 1. Journal speed. 2. Anticipated operation—that is, whether the bearing will operate on a full, mixed, or boundary film. 3. Unit load. As a guide to lubricant selection, Fig. 31 combines these factors with horizontal bands representing the recommended SAE lubricant.

For example, consider a lightly loaded journal operating under full-film conditions at 1000 rpm. Move vertically in Fig. 31 for a speed of 1000 rpm. The slanting line for light load, full-film lubrication

is intersected in the middle of the band for SAE 10 oil. If the journal were operating under a heavy load, an SAE 20 oil would be recommended by the chart.

In general, heavier oils—those with higher SAE numbers—are recommended for higher loads. Heavier oils should also be used for boundary and mixed-film operation. Grease lubrication is suggested when lubricants heavier than SAE 50 are required. When operating temperatures are high, oils heavier

Table 6—Values of  $B$

Lubricant	Viscosity (centipoises)	Temperature (F)	$B$
Steam cylinder oil	442	130	$1.69 \times 10^{-4}$
Light turbine oil	30	100	1.37
Kerosene	1.9	81	0.75
Castor oil	266	100	1.01
Glycerine	200	94	0.40
Olive oil	40	115	0.77
SAE 10	40	100	1.73
SAE 10	7	180	1.45
SAE 20 (high VI)	56	100	1.53
SAE 20 (high VI)	10.6	180	1.26
SAE 20 (low VI)	77	100	1.90
SAE 20 (low VI)	11.7	180	1.37
SAE 30	105	100	$2.2 \times 10^{-4}$

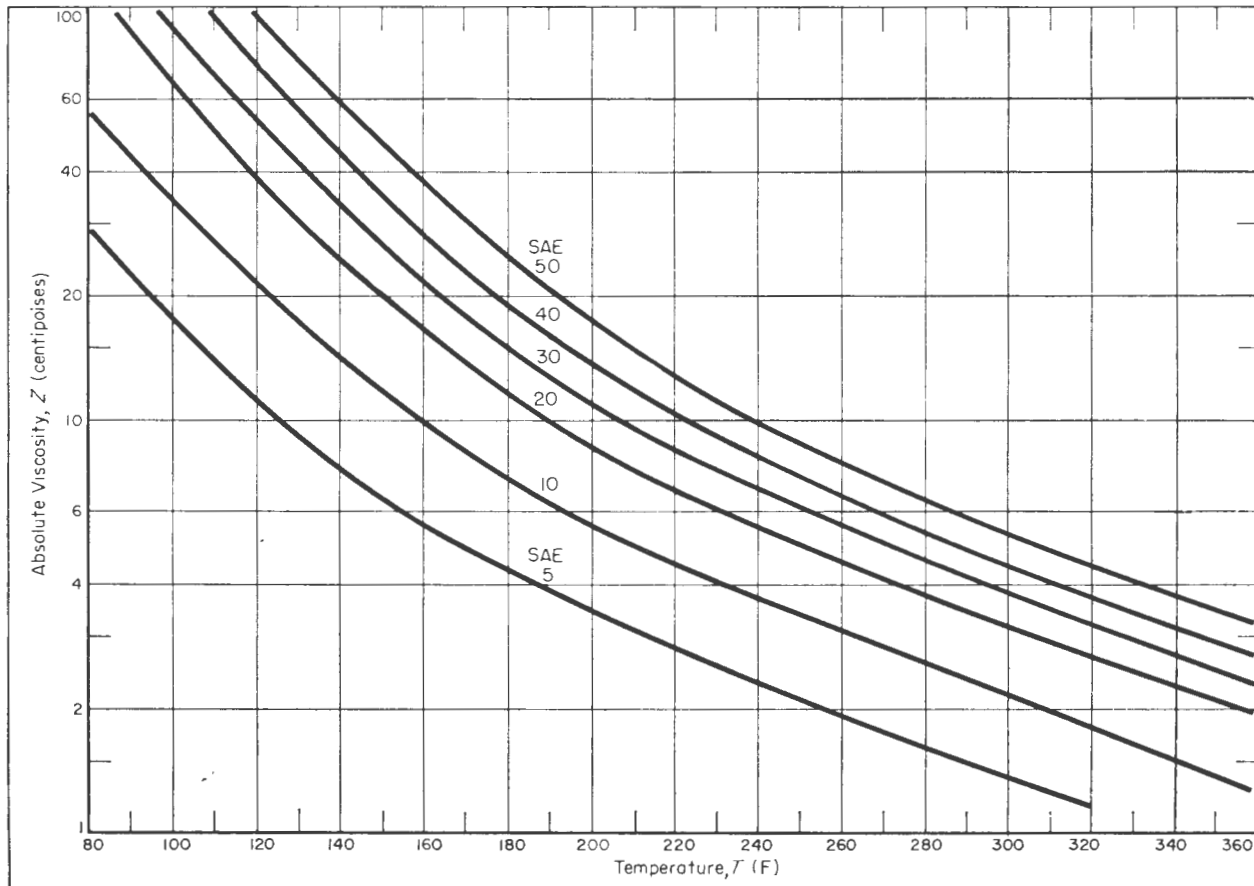
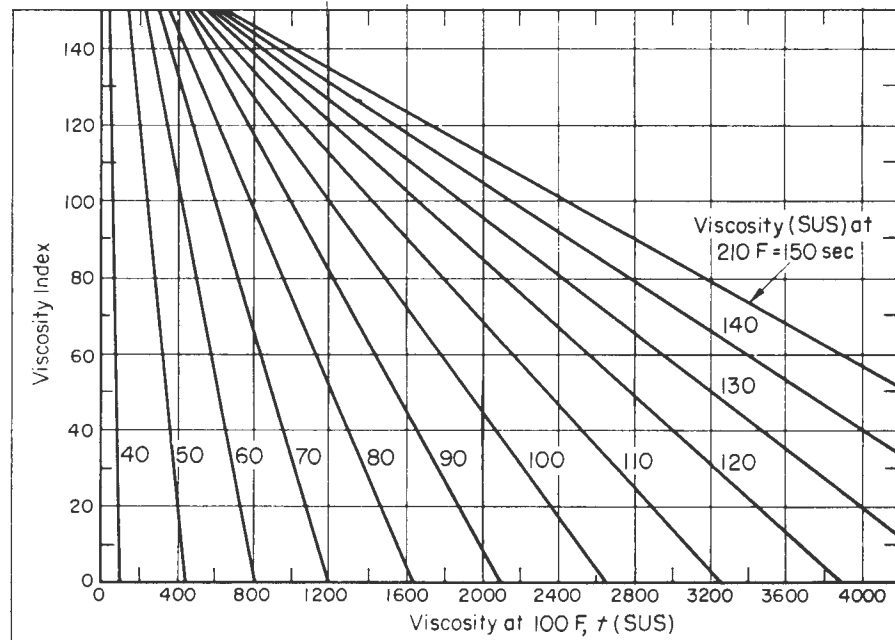


Fig. 29—Average absolute viscosities of typical SAE motor oils for normal temperature range.

Fig. 30—Viscosity index as determined by two Saybolt viscosities at standard temperatures of 100 and 210 F.



than indicated on the chart should be used, and for bearings more than 3 in. in diameter, lighter oils than indicated are recommended. Thus, although Fig. 31 is a useful guide, certain situations may require violation of the charted suggestions.

### ► Greases and Solid Lubricants

In addition to lubricating oils, many other commercial lubricants are available for use in bronze bearings. Reasons for using other lubricants are

varied, but several important reasons are to:

1. Lengthen the period between relubrication.
2. Avoid contaminating surrounding equipment or material with "leaking" lubricating oil.
3. Provide effective lubrication under extreme temperature ranges.
4. Provide effective lubrication in the presence of contaminating atmospheres.
5. Prevent intimate metal-to-metal contact under conditions of high unit pressures which might destroy boundary lubricating films.

Lubricants which can meet some or all of these con-

Table 7—Common Greases and Solid Lubricants

Lubricant Type	Appearance or Structure	Solubility in Water	Recommended Operating Temperature	Operating Loads	Comments
<b>Greases</b>					
Calcium soap, or lime soap	Smooth, buttery	Insoluble	160 F max	Moderate	—
Sodium soap, or soda base	Fibrous texture	Soluble	300 F max	Wide range	For wide speed range. Possible oil separation above 350 F.
Aluminum soap	Smooth, salve-like	Insoluble	180 F max	Moderate	—
Lithium soap base	Smooth	Semisoluble	300 F max	Moderate	Good for low temperatures.
Barium soap	Short fibers	Insoluble	350 F max	Wide range	Multipurpose grease. Similar to lithium soap base grease.
<b>Solid Lubricants</b>					
Graphite	Powder or flakes	Insoluble	1000 F max	Wide range	Chemically inert. Mixes readily with oil or grease.
Molybdenum disulfide	Powder	Insoluble	— 100 to 750 F	Wide range	Chemically inert. Resists attack by water, oil, alkalies, and most acids.

ditions fall into two major classifications—greases and solid lubricants.

**Greases:** Where full-film lubrication is not possible or is impractical for slow speed, fairly high-load applications, greases are widely used as bronze-bearing lubricants. Although full-film lubrication with grease is possible, an elaborate pumping system is required to continuously supply a prescribed amount of grease to the bearing. Bronze bearings supplied with grease are usually lubricated periodically. Hence, for this discussion, grease lubrication implies that the bearing will operate under conditions of complete boundary lubrication and should be designed accordingly.

Lubricating greases are essentially a combination of a mineral lubricating oil and a thickening agent, which is usually a metallic soap. When suitably mixed, they make excellent bronze-bearing lubricants. There are many different types of greases which, in general, may be classified according to the soap base used. Information on the most common greases is charted in Table 7.

*Synthetic greases* are composed of normal types of soaps but use synthetic hydrocarbons instead of normal mineral oils. Available in many consistency ranges in both water-soluble and insoluble types,

synthetic greases are capable of wide variations in operating temperature.

*Additives* for lubricating greases, such as oxidation inhibitors and extreme-pressure additives, are available. Greases can be fortified with fillers such as mica, lead, zinc, carbon black, or graphite to enhance their lubricating quality. Such fillers are of advantage under extremely heavy loads or intermittent motion. Because certain additives and fillers may adversely affect bearing or journal material, they should be selected with care. Cast bronzes are not affected by mild EP additives, but highly active additives require caution. Final recommendations on special-purpose greases should be obtained from the lubricant manufacturer.

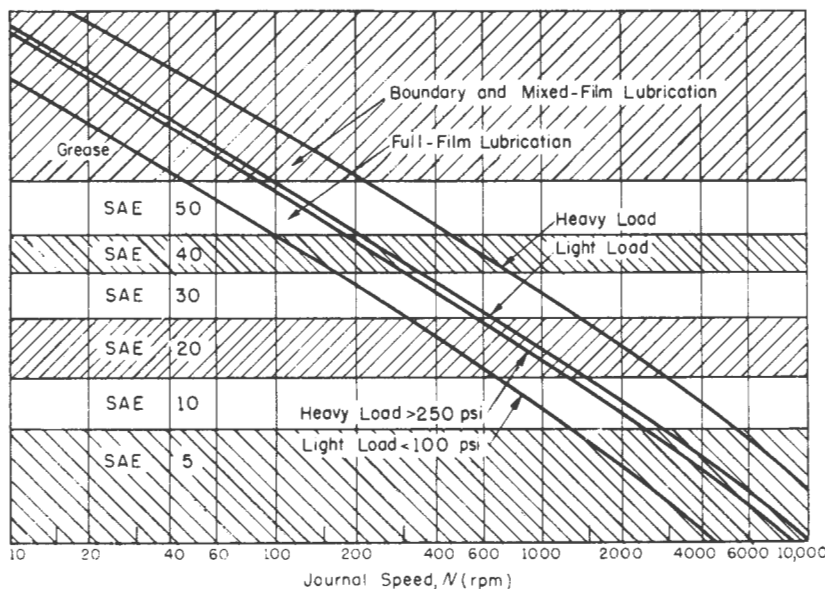
*Application of grease* is accomplished by one of several different techniques determined by grease consistency. National Lubricating Grease Institute has classified greases by their consistency and assigned NLGI consistency numbers to them. This classification is shown in Table 8 along with typical methods of application. Grooves for grease are generally greater in width, up to 1.5 times, than for oil.

*Coefficients of friction* for grease-lubricated bearings range from 0.08 to 0.16, depending upon consistency of the grease, frequency of lubrication, and type of greasc. An average value of 0.12 may be used for design purposes.

**Table 8—NLGI Consistency Numbers**

NLGI Consistency No.	Consistency of Grease	Typical Method of Application
0	Semifluid	Brush or gun
1	Very soft	Pin-type cup or gun
2	Soft	Pressure gun or centralized pressure system
3	Light cup grease	
4	Medium cup grease	Pressure gun or hand
5	Heavy cup grease	
6	Block grease	Hand, cut to fit

**Solid Lubricants:** The need for effective high-temperature lubricants led to development of several solid lubricants. Essentially, solid lubricants may be described as low-shear-strength solid materials. Their function within a bronze bearing is to act as an intermediary material between sliding surfaces. Since these solids have very low shear strength, they shear more readily than the bearing material and thereby allow relative motion. While solid lubricant remains between the moving sur-



**Fig. 31—General guide to lubricant selection for cast-bronze sleeve bearings operating between 60 and 140 F. (After B. Dunham)**

faces, effective lubrication is provided, and friction and wear are reduced to acceptable levels.

Two common solid lubricants currently used to provide effective lubrication of bronze bearings are listed in Table 7. Where high temperatures prevail, or where oil or grease contamination cannot be tolerated, solid lubricants are specified. Normal cast-bronze sleeve-bearing materials are not recommended above 550 F, but special bearing bronze alloys containing graphite plugs are available for temperatures up to 1000 F.

*Graphite*, in addition to its desirable high-temperature properties, will not react with bearing or journal materials. Mixed with oil or grease, graphite improves their boundary lubricating properties. Also, metal surfaces coated with graphite are more readily

wetted with oil or grease. Because dry graphite is difficult to apply, surfaces are usually coated with a solution of graphite in a volatile carrier or vehicle. After vaporizing, the vehicle leaves a thin film of graphite on the surfaces.

*Molybdenum disulfide* has an affinity for metal. When rubbed on a metal surface, it forms a thin, durable film of solid lubricant. Correctly applied, it provides good protection against galling and seizing over a wide temperature range. Molybdenum disulfide has good extreme-pressure characteristics and can be combined with oils and greases to enhance their boundary lubricating qualities. It is also available in spray or paste form for more convenient application.

## CHAPTER 6

# Bearing Bronze Selection

**S**ELECTION of a sleeve-bearing material would be no problem if bearing and journal surfaces were absolutely smooth and always separated by a full film of lubricant, if the materials did not experience elastic or thermal distortions, and if the lubricant contained no abrasive particles. Under such ideal conditions, a bearing material of adequate strength becomes the only consideration. However, perfect operation cannot be realized because:

1. The bearing can distort.
2. The shaft can deflect and may rub at the ends of the bearing.
3. Bearing and journal surfaces cannot be made perfectly smooth; consequently, peaks of surface roughness may periodically puncture the lubricant film.
4. The bearing may occasionally be "starved" of lubricant.
5. Lubricant viscosity may be too low.

When these undesirable conditions exist, solid con-

tact is permitted between journal and bearing. It inevitably results in wear and reduced bearing life. Hence, choice of good bearing materials, in conjunction with bearing design and a suitable lubricant, is essential in obtaining trouble-free sleeve bearings.

### ► General Requirements

Experience shows that a good bearing material is:

1. Score resistant.
2. High in compressive strength.
3. High in fatigue strength.
4. Deformable.
5. Corrosion resistant.
6. Low in shear strength.
7. Structurally uniform.
8. Inexpensive.
9. Readily available.

This list of requirements is evidence that selection of a bearing material for a particular application is almost always a compromise. No single material

Table 10—Nominal Physical Properties of Cast Bronze Bearing Materials

Alloy	Tensile Strength (1000 psi)	Yield Strength (1000 psi)	Elongation (% in 2 in.)	Reduction of Area (per cent)	Brinell Hardness Number (500 kg load)	Modulus of Elasticity (psi)	Impact Strength, Izod (ft-lb)	Density (lb/cu in.)
SAE 40	35	18	28	25	60	$13.5 \times 10^6$	6-12	0.318
SAE 62	45	22	25	20	65	12.5	4-16	0.314
SAE 620	40	20	30	25	68	14.0	9-22	0.315
SAE 63	40	20	25	25	70	13	3-11	0.317
SAE 64	35	18	20	10	63	10.5	2-8	0.321
SAE 660	35	20	15	20	60	12.0	4-12	0.316
SAE 67	30	17	15	15	55	9.5	4-6	0.332
AMS 4840	25	15	15	10	48	9.0	3-6	0.326
ASTM B 148-52-9C	90	37	15	15	195	$18 \times 10^6$	10-15	0.272

Table 9—Composition of Cast Bronze Bearing Materials

Alloy	Common Name	Elements (per cent by weight)									
		Copper	Tin	Lead	Zinc	Iron	Nickel	Phosphorus	Aluminum	Silicon	Antimony
SAE 40	Leaded red brass; Ounce metal; Composition metal	84-86	4-6	4-6	4-6	0-0.30	0-1.0	.....	0-0.005	0-0.005	.....
SAE 62	Gun metal	86-89	9-11	0-0.3	1-3	0-0.15	0-1.0	.....	0-0.005	.....	.....
SAE 620	Navy "G"	86-89	7.5-9	0-0.3	3-5	0-0.15	0-1.0	.....	0-0.005	.....	.....
SAE 63	Leaded gun metal	86-89	9-11	1-2.5	0-0.75	0-0.15	0-1.0	0-0.25	0-0.005	.....	.....
SAE 64	Phosphor bronze	78-82	9-11	8-11	0-0.75	0-0.15	0-0.50	0-0.25	0-0.005	0-0.005	0.50
SAE 660	Bronze bearings	81-85	6.25-7.5	6-8	2-4	0-0.20	0-0.50	0-0.15	0-0.005	0-0.005	0.35
SAE 67	Semiplastic bronze	76.5-79.5	5-7	14-18	0-1.5	0-0.40	0-0.75	0-0.05	0-0.005	0-0.005	0.50
AMS 4840	High-leaded tin bronze	67.5-72.5	4.5-6	23-26	0-0.5	0-0.30	.....	0-0.05	.....	.....	0-0.20
ASTM B 148-52-9C	Aluminum bronze	83.0-87.0	....	.....	....	3.00-5.00	0-2.50	.....	10-11.5	.....	(Manganese, 0-0.50)

has all of these characteristics in a sufficiently acceptable level. For example, a material having good fatigue strength generally has poor deformability. The order of importance for these general requirements depends upon the particular application.

Score resistance is that quality of a bearing material which prevents damage to the journal during boundary or mixed-film operation or when starting and stopping. Antiweld or antiseizure characteristics are sometimes referred to as score resistance.

Compressive strength, a fundamental requirement, is the ability of the material to carry the imposed load without extrusion or disintegration.

Fatigue strength is the ability of the material to

give satisfactory service life when subjected to variable stresses.

Deformability is that quality of the material which permits it to yield, if necessary, to deformation under operation without causing failure.

Conformability is desirable when the journal touches the end of the bearing as a result of load deflection in both journal and bearing. The bearing should adjust itself by wearing or wiping away without developing a high-temperature condition.

Embeddability is important when dirt enters the clearance space. If a dirt particle cannot embed itself in a relatively soft bearing material, it will jam itself between a hard bearing material and the shaft and eventually cut a groove in both.

Thermal Coefficient of Expansion (in./in./deg F)	Thermal Conductivity (Btu-ft/hr/sq ft/deg F)	Maximum Operating Temperature (F)	Max Unit-Pressure Load (psi)	Compressive Strength Data					Alloy
				Permanent Set (in.)	Compressive Strength (1000 psi) for Sample Thickness (in.) of . . .				
					0.125	0.250	0.500	1.000	
10.9 × 10 <sup>-6</sup>	48	450	3500	0.001	28	17	14	13	SAE 40
				0.005	42	23	18	16	
10.0	37	500+	4000	0.001	26	23	22	22	SAE 62
				0.005	56	33	28	26	
10.0	45	500	4000+	0.001	28	25	23	21	SAE 620
				0.005	51	37	28	25	
9.9	..	500	4000+	0.001	36	24	22	19	SAE 63
				0.005	57	36	33	32	
10.2	27	450+	4000	0.001	22	19	18	17	SAE 64
				0.005	29	27	23	20	
10.0	27	450+	4000	0.001	30	26	21	18	SAE 660
				0.005	44	32	25	21	
10.7	..	450	3000+	0.001	20	17	14	13	SAE 67
				0.005	35	27	21	16	
10.3	..	400+	3000+	0.001	17	15	13	11	AMS 4840
				0.005	26	22	17	14	
9.0 × 10 <sup>-6</sup>	32	500+	4500+	0.001	45	30	25	18	ASTM B 148-52-9C
				0.005	95	48	39	31	

**Corrosion resistance** is that quality of a material which reduces or eliminates wearing away caused by various acids formed during oxidation and deterioration of lubricants. The danger of bearing corrosion can be greatly reduced by using inhibited oils.

**Shear strength** is the ability of the material to resist movement of one layer of the material with respect to another layer. Experiments indicate that the most desirable bearing metals have a soft, low-melting constituent in which actual shearing occurs.

**Structural uniformity** is that quality of the material which results in a sound, nonporous surface necessary to promote and maintain full-film lubrication.

**Low cost and availability** often play a major role in selection of a bearing material, especially if large quantities are involved.

### ► Cast Bronze Materials

Bronze offers a favorable compromise with respect to the general requirements for a good bearing material. Cast bronzes, in general, exhibit fair deformability; good score resistance, structural uniformity, shear properties, and compressive strength; excellent fatigue strength and corrosion resistance; and the lowest cost in the field.

Trying to be specific about bronzes is difficult because of the many different compositions and modifications of these compositions. Bronze bushings are suitable for a wide range of applications. They have great strength, yet they can be easily and economically produced in a variety of forms. Even when only one or two bearings are required, the cost is low. "Machined in" bearing tolerances are also an advantage cost-wise in comparison with close-tolerance housings, subassemblies, etc.

Bronze sleeve bearings can be made as a single,

solid unit with no bond or liner to fail. Hence, they are extremely reliable. Also, relatively thick walls provide an adequate reserve for wear.

The relatively high thermal conductivity of cast bronze is an advantage in removing heat from bearing surfaces, thus tending to prevent "hot such as may occur in clad-type or steel-backed rings with only a few thousandths of bronze or Dabbutt for a bearing surface.

Soft bronze alloys containing large amounts of lead and small amounts of tin have relatively low strength but good frictional characteristics. At the other extreme are high-tin alloys with very little lead. These have high strength and hardness but less-favorable friction characteristics. Intermediate alloys contain various amounts of tin and lead with the addition of small amounts of other elements such as zinc, silicon, phosphorus, nickel, and aluminum. Nine of the most popular cast-bronze bearing materials, their common alloy names, and their chemical composition are listed in Table 9.

Important physical properties of the nine popular cast bronzes are given in Table 10. Heretofore, a cast-bronze bearing material was generally selected for the tensile properties of the alloy. The vast majority of bearings in operation today, however, are in compression only. In the table, compression data are tabulated for a permanent set of 0.001 and 0.005 in. on a 1 sq in. area for thicknesses of 0.125, 0.250, 0.500, and 1.000 in. This information should not only help the designer select the most suitable alloy but also indicate necessary wall thickness for the bearing. Typical tensile properties, also listed in the table, are seen to have no definite relationship to the compression values for the alloys.

Selection of a cast bronze alloy for a bearing requires careful consideration of the important material properties described as deformability and conformability. Alloys with high physical properties are not always the best. Leaded bronzes, although they have lower physical properties, are generally the best bearing alloys.

**Table 11—Equivalent Government and Society Specifications\***

SAE	ASTM	AMS	Navy	Military	Federal	AAR
40	B 62-52 B 145-52-4A	4855B	46-B-23e	MIL-B-16444	QQ-B-691b-2	.....
62	B 22-52-D B 143-52-1A	4845D	.....	.....	.....	.....
620	B 143-52-1B	....	46-M-61	MIL-B-16576	QQ-B-691b-5	.....
63	.....	....	.....	.....	.....	.....
64	B 144-52-3A	4842A	.....	.....	.....	M-503-48 (Phosphor Bronze)
660	B 144-52-3B	....	46-B-22g-VI	MIL-B-16261-VI	QQ-B-691b-12	.....
67	B 66-52 (Hard Bronze) B 144-52-3D	4825A	46-B-22g-IV	MIL-B-16261-IV	QQ-B-691a-7	M-503-48 (Hard Bronze)
...	B 66-52 (Soft Bronze) B 144-52-3E	4840	46-B-22g-V	MIL-B-16261-V	.....	M-503-48 (Soft Bronze)
...	B 148-52-9C	....	46-B-18d-III	MIL-B-16033-III	QQ-B-671b-3	.....

**Key:** SAE, Society of Automotive Engineers; ASTM, American Society for Testing Materials; AMS, Aeronautical Material Specifications; Navy, U. S. Navy Department; Military, U. S. Dept. of

Defense (MIL); Federal, General Services Administration; AAR, Association of American Railroads.  
\*In some cases, compositions are similar but not exactly equivalent.

Table 12—Characteristics and Uses

**SAE 40**

(85 Cu, 5 Sn, 5 Pb, 5 Zn)

**Characteristics:** Reasonable strength; excellent thermal conductivity; sound material structure; reasonable corrosion resistance to sea water and brine; good machining and casting properties—sand casting of sections larger than 1-in. thick is not recommended.

**Uses:** Light loads and low to medium speeds; primarily for bearing shells and backing for babbitt-lined bearings; corn-picker snapping roll bushings; deep-well pump line-shaft bearings; automotive-transmission thrust washers; low-pressure valve bearings; manifold bushings for earth-moving machinery; pump sleeves; propeller bushings; spring bushings for farm and automotive equipment.

**SAE 62**

(88 Cu, 10 Sn, 2 Zn)

**Characteristics:** Hard; strong; good corrosion resistance, especially to sea water; wear resistant; high resistance to pounding; difficult to broach but can be bored or reamed; requires good, reliable lubrication and a shaft hardness of 300-400 Bhn.

**Uses:** Heavy loads and low speeds; piston-pin bushings; valve guides; rolling-mill bearings; rocker-shaft bushings for internal-combustion engines; worm bearings; pilot bearings; roll bushings, and linkage bushings for machine tools.

**SAE 620**

(88 Cu, 8 Sn, 4 Zn)

**Characteristics:** Hard; strong; ductile; corrosion resistant to sea water and brine; wear resistant; good resistance to pounding; good machining properties—similar to SAE 62 but easier to work in foundry; requires reliable lubrication, good alignment, and a shaft hardness of 300 to 400 Bhn.

**Uses:** Heavy loads and low speeds; bushings for aircraft landing gear; bridge bearings; machine-tool bearings; trunnion bearings; trolley wheels; wrist-pin bushings for air compressors.

**SAE 63**

(88 Cu, 10 Sn, 2 Pb)

**Characteristics:** Hard; strong; good corrosion resistance; wear resistant; good machining and fair casting properties; requires good lubrication and a shaft hardness of 300 to 400 Bhn.

**Uses:** Heavy loads, low speeds, and severe working conditions; applications requiring strength in addition to bearing properties; bearings for earth-moving machines; locomotive bearings; trunnion bearings; gear bushings and connecting rods for farm equipment; mechanical linkage for farm equipment and packag-

ing machinery; automotive spindle bushings.

**SAE 64**

(80 Cu, 10 Sn, 10 Pb)

**Characteristics:** Good strength; low thermal conductivity compared to high-tin bronzes; generally high physical properties combined with fairly good bearing qualities; corrosion resistant to mild acids as found in mine water, good resistance to mineral waters or paper-mill sulfite liquor; excellent wear resistance under conditions of high speed, heavy pressure, shock, and vibration; low friction; good machining and fair casting properties; can be used where lubrication is doubtful; requires hardened shaft; maximum shaft surface speed can be 800 to 1000 fpm.

**Uses:** Heavy loads and high speeds; backing for babbitt-lined bearings; wrist-pin and valve rocker-arm bushings; fuel and water-pump bushings; aircraft control bushings; electric-motor bushings; steering-knuckle bushings; track-roller bushings; crankshaft bearings for outboard motors, power lawn mowers, and garden tractors; gear bushings for farm equipment, reducers, and motorcycles; cam bushings for mechanical devices; farm equipment, and diesel engines; crankshaft, spindle, and connecting-rod bushings for farm equipment; piston pins for packaging equipment and diesel engines; linkage bushings for presses; roll-neck bearings; lathe bearings; babbitt-lined armature bearings for electrical locomotives; railroad-car bearings; rolling-mill bearings; trunnion bearings.

**SAE 660**

(83 Cu, 7 Sn, 7 Pb, 3 Zn)

**Characteristics:** Good hardness, strength, and wear resistance; excellent antifrictional qualities; good casting and excellent machining properties; can be readily broached or reamed.

**Uses:** Currently the standard cast-bronze bearing material; general utility applications for medium loads and speeds; electric-motor bushings; spring-shackle bushings; torque-tube bushings; bushings for automotive generators, distributors, and starters; motorcycle-engine bearings; sleeve bushings for cranes and draglines; track-roller bushings for crawler tractors; guide bushings for valves, rams, and piston rods; camshaft bushings for tractor engines; roller bushings for conveyors; main bearings for presses; mechanical-linkage bushings for farm and material-handling equipment; spindle bushings for farm equipment and trucks.

**SAE 67**

(78 Cu, 6 Sn, 16 Pb)

**Characteristics:** Fair strength; corrosion resistant to sea water and brine, resist-

ant to certain concentrations of sulfuric acid and acids in mine waters; fair wear resistance; excellent antifrictional qualities; nonseizing; good machining and casting properties; can be used where lubrication is doubtful.

**Uses:** General-service bearing material for moderate pressures and high speeds; general-purpose wearing material for rod bushings, freight and streetcar bearings, and backs for lined bearings; bushings for gasoline pumps; hydraulic glands; seals; deep-well pump-bowl bushings; general bearings for locomotive tenders and passenger and freight cars; rolling-mill bearings; brass and copper rolling-mill neck bearings; gas-engine bearings; diesel-engine bearings; kingpin bushings for earth-moving machinery; drum bushings for cranes and earth-moving machinery; bearings and spacer bushings for pumps.

**AMS 4840**

(70 Cu, 5 Sn, 25 Pb)

**Characteristics:** Fair strength; fair wear resistance; low pounding resistance; excellent antifrictional qualities; good conformability—will not score soft shafts; good machining and fair casting properties—requires care in broaching; can operate at high speeds under adverse lubrication conditions.

**Uses:** Light to moderate loads and high speeds; unsuitable for extremely heavy compressive and shock loads; suitable for bearing liners required to deform locally to conform to irregularities of motion or bad fit; under-water service; water-lubricated bushings; aircraft-carburetor bearings; fire-pump bushings; railroad-car bearings; used extensively for locomotive bearing parts; rod bushings and main bearings for refrigeration compressors; hydraulic pump bushings.

**ASTM B 148-52-9C**

(86 Cu, 3.5 Fe, 10.5 Al)

**Characteristics:** Very hard; abrasion resistant; excellent strength; physical properties remain good at elevated temperatures—compressive strength at 500 F is the same as for tin bronzes at room temperature; wear resistant; excellent resistance to repeated, severe impacts; relatively poor antiseizure properties; relatively poor conformability and embeddability; difficult to broach; requires reliable full-film lubrication to prevent metal-to-metal contact and possible scoring; requires shaft hardness of 550 to 600 Bhn to prevent damage to shaft; requires surface finish of 15 to 20 microinches rms on both bearing and shaft.

**Uses:** Bushings for power shovels and heavy earth-moving equipment; roll-neck bearings; guide-post bushings; turntable bushings; boring-bar guide bushings; machine-tool bearings.

Specifications for the nine cast bronzes are cross-referenced in Table 11. Important characteristics and typical applications are listed in Table 12.

Tin bronzes, which include SAE 62, SAE 620, and SAE 63, are generally used in those applications which require high loads at low speeds. Because of their hardness, these bronzes require adequate and reliable lubrication and will tolerate very little misalignment.

Leaded bronzes, which include SAE 40, SAE 64, SAE 660, SAE 67 and AMS 4840, are ordinarily used for applications requiring intermediate loads at relatively high speeds. Bearing properties of leaded bronzes are enhanced by the additional lead content which fulfills the requirement that a good bearing material contain a soft, low-melting constituent. SAE 67 and AMS 4840 are high-leaded bronzes which handle intermediate loads at slow to rela-

tively high speeds. Their excellent conformability enables them to correct minor misalignment, and their high lead content offers excellent shaft protection in case the lubricant supply should momentarily fail. Physical properties of SAE 64 and SAE 660, the two most widely used bronzes, offer the best compromise to all requirements. They are particularly suited for medium loads at medium to relatively high speeds. As a substitute for the other leaded bronzes, SAE 40 presents certain advantages. In addition to its ability to do a good job at the lighter loads, it is also somewhat lower in cost than the other bronzes.

Aluminum bronze, ASTM B 148-52-9C, is not a "true" bearing material because it lacks some of the common bearing characteristics such as antiseizure, embeddability, and conformability. Main advantages are high strength and retention of favorable characteristics at temperatures in excess of 500 F.



## CHAPTER 7

# Grooving Specifications

**S**PECIFICATION of grooves for a sleeve bearing is equally as important as any other aspect of bearing design. Many cast-bronze sleeve bearings, properly designed otherwise, have been handicapped by improper or overzealous application of grooves. In some cases, the only reasons a grooved bearing functions at all are choice of a cast bronze as the bearing material and removal of excessive heat by the lubricant.

The primary purpose of grooving within a sleeve bearing is to expedite and insure distribution and maintenance of an efficient film of lubricant, either partial or complete, between moving surfaces of journal and bearing. How this purpose may best be achieved depends mainly upon the anticipated mode

of operation—that is, whether the bearing is designed to operate under conditions of full-film, mixed, boundary, or hydrostatic lubrication. Other necessary considerations are:

1. Type of lubricant.
2. Lubrication system.
3. Nature of the load.
4. Relative motion of bearing members.

### ► Grooving for Full-Film Lubrication

Full-film lubrication can be achieved with either oil or grease. However, greases are seldom used for this purpose since it is a chore to continually re-

**Table 13—Uses of Grooving for Full-Film Lubrication**

Type of Groove	When To Use	Comments
Single oil hole	When oil flow from single oil-inlet hole can provide sufficient lubricant.	Hole location depends on direction of load and may be in either bearing or shaft.
Straight-axial	For more effective axial distribution of lubricant, especially in long bearings.	Groove and inlet-hole locations depend on direction of load and may be in either bearing or shaft.
Feeder plus straight-axial	In large-diameter bearings to permit correct location of axial groove.	Sequence of groove elements, in direction of rotation, must be: 1. Oil hole. 2. Feeder groove. 3. Axial groove.
Circular	When direction of load varies such that a consistent low-pressure region cannot be located on either bearing or shaft.	Groove is usually pressure-fed.
Straight-axial circular	When circular groove cannot be pressure-fed.	Axial groove must be located in low-pressure region.

plenish the grease which escapes from the bearing. Hence, only grooving for normal lubricating fluid—oil—will be considered.

From previous discussions, full-film lubrication was shown to be possible under favorable conditions of load and speed, provided, of course, that the bearing is properly designed to promote separation of the bearing members. An approximation of oil-flow rate required to sustain hydrodynamic lubrication has also been discussed. Next, where to introduce the necessary lubricant must be determined. The obvious and only location for introducing lubricant to the bearing is in a region of low pressure.

Theoretical distribution of pressure within an ungrooved, full-film lubricated sleeve bearing is shown

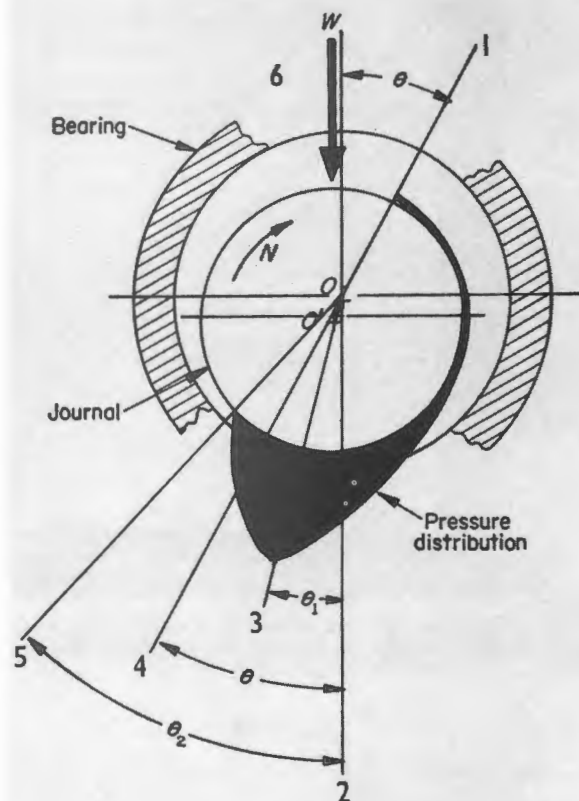


Fig. 32—Reference locations for pressure distribution within an ungrooved bronze bearing operating with full-film lubrication are: 1. Maximum film thickness; beginning of pressure formation. 2. Direction of load. 3. Maximum film pressure. 4. Minimum film thickness. 5. End of pressure film. 6. Unloaded side of bearing.

### Nomenclature

$A$	= Bearing characteristic number
$a$	= Groove width, in.
$b$	= Groove depth, in.
$D$	= Journal diameter, in.
$D_B$	= Bearing bore diameter, in.
$L$	= Bearing length, in.
$N$	= Rotational speed of journal, rpm
$O$	= Actual center location of bearing
$O'$	= Actual center location of displaced journal
$W$	= Steady load to be supported, lb
$w$	= Wall thickness of bearing, in.
$e$	= Journal eccentricity ratio
$\theta$	= Angle between direction of load and direction of journal displacement, deg
$\theta_1$	= Angle between direction of load and point of maximum film pressure, deg
$\theta_2$	= Angle between direction of load and point where film pressure ends, deg

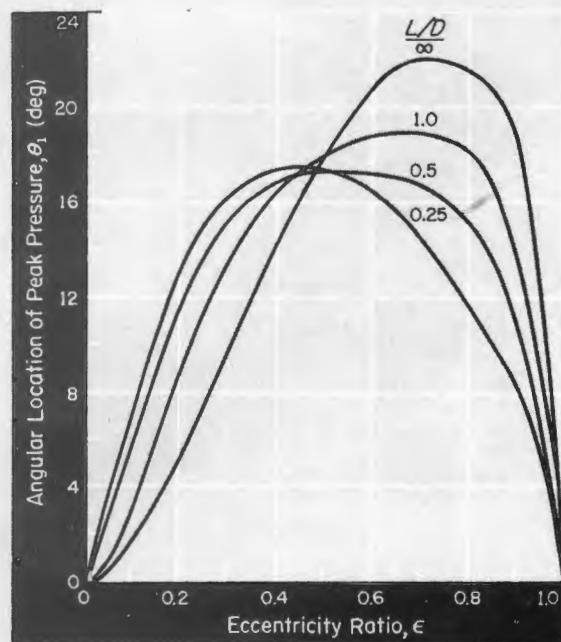


Fig. 33—Location of peak pressure, position 3 in Fig. 32, for full-film lubricated bearing.

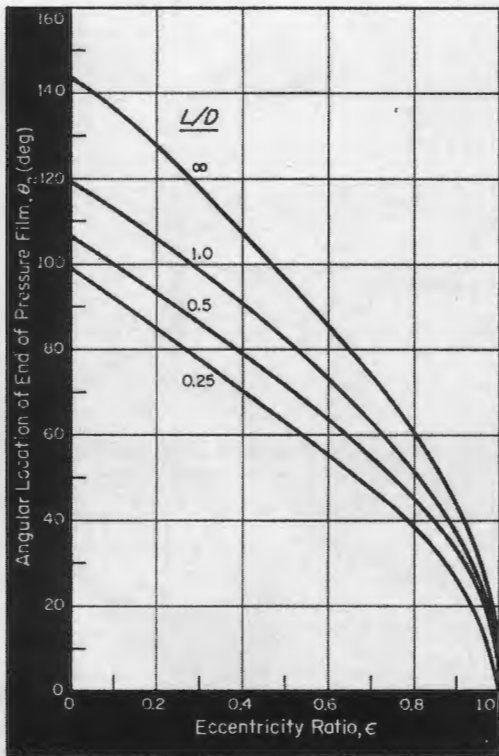


Fig. 34—Location of end of pressure film, position 5 in Fig. 32, for full-film lubricated bearing.

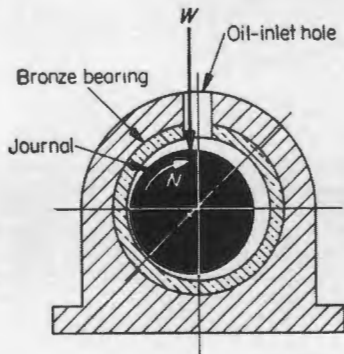


Fig. 35—Single oil-inlet hole located on unloaded side of bearing.

in Fig. 32. The numbered locations around the bearing are of interest and are referred to throughout this discussion. Pressure is generated, in a clockwise direction, from region 1 to 5, and the low-pressure region is from 5 to 1. Location 1 is the region of maximum film thickness and is diametrically opposite location 4, which is the region of minimum film thickness. Location 4 can be determined from the curves in Fig. 12 (in Reference 1). Location 2 is on a line through the bearing center and parallel to the direction of load. Peak pressure at position 3 and zero pressure at position 5 are located using position 2 as a reference. Fig. 33 and 34 can be used to determine locations 3 and 5 if the operating eccentricity ratio is known.

With three points on the pressure distribution curve known—zero at 1, maximum at 3, and zero again at 5—the pressure distribution can be approximated by connecting these three points as in Fig. 32. Location 6 is on the unloaded side of the bearing opposite location 2. If the journal were rotating in the opposite direction, the geometry would be reversed and could be obtained by looking at Fig. 32 in a mirror. Also, if the load were acting in some direction other than that shown, the proper picture could be obtained by rotating the figure until load  $W$  is in the proper angular position.

For the simple case where load is constant and always in one direction and the speed is also constant, the low-pressure region from 5 to 1 is always in the same location with respect to the bearing. Hence, the oil-inlet hole may be located somewhere between location 5 and, in the clockwise direction, location 1—the closer to 1 the better. Usual practice is to place the oil-inlet hole at location 6, Fig. 35. In horizontal bearings, the inlet hole should be centered axially; for vertical bearings, it should be closer to the top end.

In many cases a single oil-inlet hole located in the unloaded region of the bearing will suffice. However, if the bearing is of appreciable length, lubricant will not be distributed uniformly over the en-

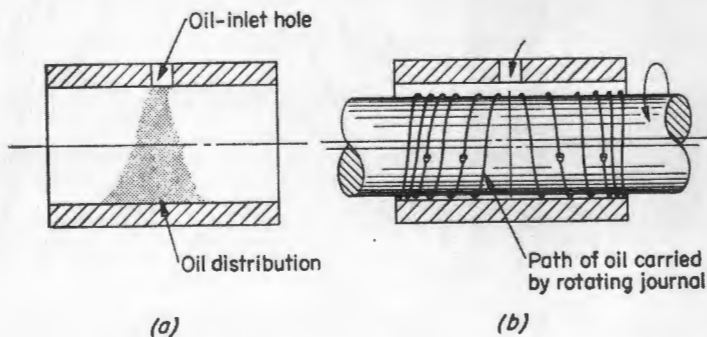


Fig. 36—Possible oil distribution in a long bearing using only a single oil-inlet hole.

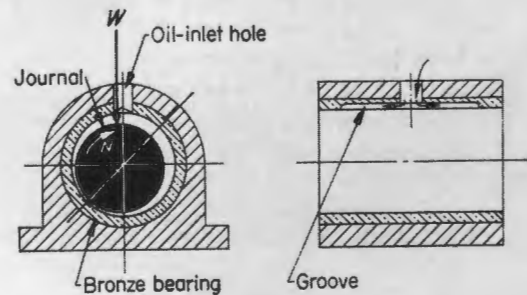


Fig. 37—Straight-axial oil distribution groove.

tire length of the bearing. Fig. 36a illustrates this condition. Possible route of travel of lubricant through the bearing is shown in Fig. 36b to further illustrate the point. To correct this situation, a straight-axial groove is often placed in the bearing, Fig. 37, usually at the same location as the inlet hole (position 6). When this type of grooving is used, the oil can be distributed axially along the groove before being picked up by the shaft and carried through the bearing. A much more uniform distribution of lubricant is thus made possible over the length of the bearing. This type of grooving shall be designated a "straight" groove.

If the bearing is fairly large in diameter, the straight-axial distribution groove should be located closer to position 1. This location, Fig. 38, allows the lubricant to be picked up by the shaft closer to the point where it is needed. If it is impossible to position the oil-inlet hole directly above the axial distribution groove at position 1, the inlet hole may be connected to the distribution groove by a feeder groove, Fig. 39. Remember, however, that the oil-inlet hole must be located only in the low-pressure region. Orientation of the grooves with respect to direction of shaft rotation is also important. That is, a point on the shaft should arrive first at the oil-inlet hole, then at the partial circumferential-feeder groove, and finally at the straight-axial distribution groove.

Thus far, only unidirectional loading and rotation have been considered, and required grooving is very simple. For such operating conditions, all that is required is to establish the location of the low-pressure region of the bearing and use either a single oil hole or a straight grooving arrangement in this region. The pressure distribution can be approximated by first evaluating the probable eccentricity ratio from Fig. 13 (Chapter 2). Position 4 and, hence, position 1 can then be determined from Fig. 12 (Chapter 2). The other two points of interest on the pressure distribution curve are points 3 and 5, which are determined from Fig. 33 and 34 for the same eccen-

tricity ratio.

Location of the generated pressure within a full-film lubricated bearing depends upon bearing characteristic number  $A$  and the directions of load and speed. Any change in operating conditions means a change in pressure distribution which must move within the bearing for continued satisfactory operation. Consequently, it may not be possible to locate a region of the bearing which is always at low pressure. For example, in a bearing where load may fall in any direction, a given location may alternately be in a high-pressure region and then in a low-pressure region, depending upon instantaneous load direction. To allow for this condition, a circular or annular groove is usually machined completely around the bearing, Fig. 40.

The circular type of groove divides the bearing into two shorter bearings, each of which is lubricated from one end. Other conditions being equal, two short bearings whose combined length is equal

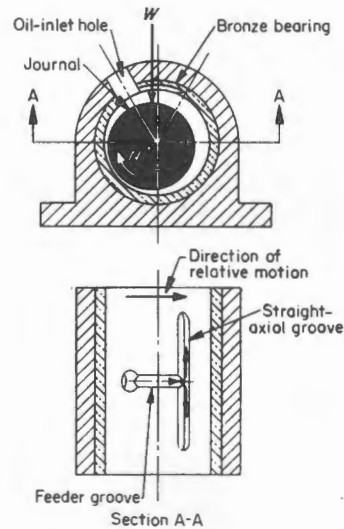


Fig. 39 — Oil-inlet hole joined to properly located straight-axial groove by a feeder groove. To aid machining, feeder groove may extend slightly beyond the straight groove or inlet hole.

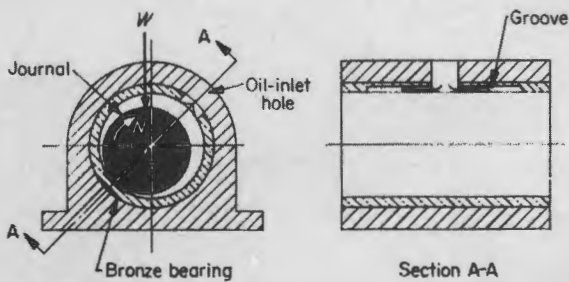


Fig. 38—Straight-axial groove and inlet hole located at beginning of generated-pressure zone for better oil distribution.

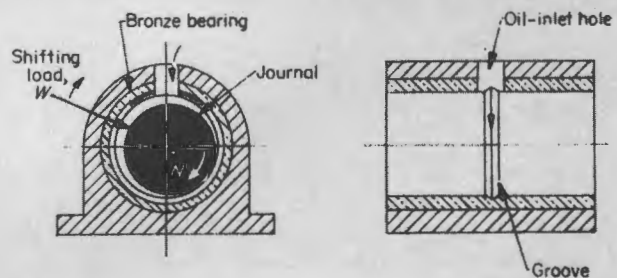


Fig. 40—Circular oil-distribution groove for load with constantly changing direction.

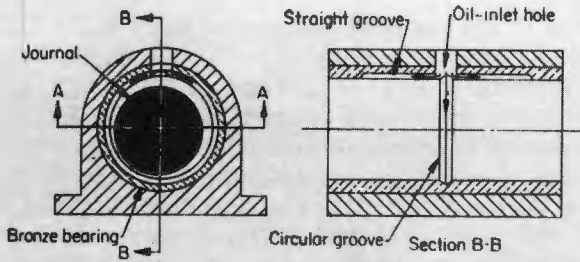


Fig. 41 — Nonpressure-fed bearing using a circular groove plus a straight groove located in a low-pressure region.

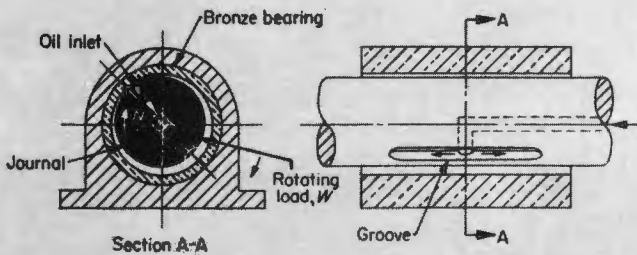


Fig. 42—Oil inlet and straight-axial groove located in low-pressure region of shaft for synchronous rotating load.

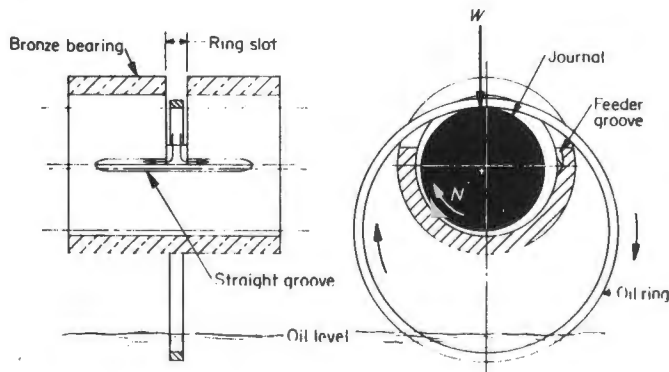


Fig. 43—Ring-oiled bearing using feeder and straight-axial grooves to distribute oil picked up by ring.

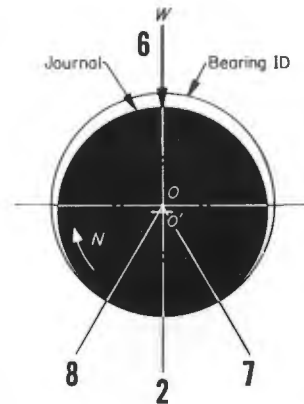


Fig. 44 — Geometry of completely boundary-lubricated sleeve bearing.

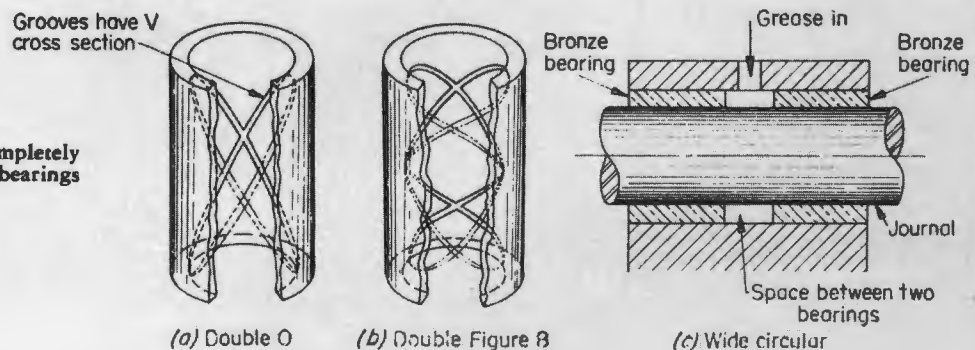


Fig. 45 — Grooving for completely boundary-lubricated bronze bearings using grease.

to that of a single bearing do not carry the same load as the single bearing. Thus, each half of the bearing has to be treated separately to evaluate load-carrying capacity, eccentricity ratio, etc. The oil-inlet hole can be at any angular position around the bearing. Actually, the circular groove could just as easily be put in the shaft if shaft strength permits. Also, the oil-inlet hole could be put in the shaft if supplying lubricant through the shaft is desired.

Circular-grooved bearings are usually pressure-fed. However, they may be used in nonpressure-fed applications in combination with a straight-axial groove if the straight groove is placed in a relatively lightly loaded region of the bearing, Fig. 41. If the lightly loaded region is fixed with respect to the shaft, the straight groove may be machined in the shaft. This type of grooving shall be designated a combination circular and straight-axial groove.

For the special case in which load rotates with the shaft, either a single hole or a straight groove or both may be used in the shaft. Lubricating through the shaft is possible with a synchronized rotating load because the pressure distribution is fixed with respect to the shaft. Here again the oil-inlet hole and groove in Fig. 42 must be oriented to lie between positions 6 and 1 just as was necessary when the inlet hole and groove were in the bearing. This type of grooving is also recommended when the bronze bearing rotates around a stationary shaft.

For ring or chain-oiled bearings, a slot at the

top of the bearing allows the ring or chain to touch the rotating journal. The slot is usually connected to a straight distribution groove by a feeder groove, Fig. 43. Orientation of the grooving should be as indicated. Recommended total clearance between ring and slot is  $\frac{1}{16}$  to  $\frac{1}{8}$  in.

Preferred types of grooving for full-film, hydrodynamically lubricated sleeve bearings are summarized in Table 13. In general, for vertical bearings the oil-inlet hole is displaced from the axial center of the bearing to a position closer to the top of the bearing. Circular grooves, when used, are also usually displaced toward the top of the bearing.

In some cases a sleeve bearing may receive lubricant from one end of the bearing. To provide effective lubrication, the oil must first enter and then pass through the bearing. Small clearances and the internal pressures generated normally prevent axial flow through the bearing. However, one way of promoting flow through the bearing is to provide a straight-axial groove which is located in the low-pressure region and which is open at the lubricant end of the bearing.

Performance of some full-film lubricated bronze bearings can at times be enhanced by some form of specialized or "exotic" grooving different from the types already covered. However, the designer should consider carefully before specifying such forms. Not only are special grooves expensive to fabricate, but more important, they may defeat their purpose by crossing regions of the bearing which would otherwise generate load-carrying pressure. If the necessary pressure distribution for full-film lubrication can be determined, special grooving may be used in the low-pressure region to suit a particular application. However, specialized grooving should be considered only if one of the previously discussed types cannot be used or is not adequate.

One final and important requirement is that the edges of all grooves be broken or chamfered at assembly after the bore of the bearing is finished or sized. This precaution eliminates the possibility of having burrs on the grooves which would act as oil scrapers.

### ► Grooving for Complete Boundary Lubrication

By definition of complete boundary lubrication, none of the load on the bearing is supported by lubricant. Consequently, the question might be asked: Is grooving necessary? The answer is "yes" because every advantage should be taken to insure that what little amount of lubricant is provided reaches the loaded region of the bearing to effectively boundary lubricate the moving surfaces.

Normally, sleeve bearings which are lubricated only periodically will at some time be operating under boundary lubrication conditions. For such applications, grease is the more usual lubricant although hand-oiled bearings fall into a similar category. Since grease does not flow readily, it must be pumped into the bearing. Inside the bearing, grooves are necessary to allow free passage of the grease to the loaded region and to provide a reservoir for the grease. When oil is used for completely bound-

ary-lubricated bearings, grooving allows easy passage of the lubricant to the region where it is required. Grooving also acts as a trap for the oil and retards its eventual escape from the bearing.

Thus, the little lubricant that is supplied under boundary lubricating conditions remains within the bearing longer than it would if there were no grooving, provided, of course, that the grooves do not break through the ends of the bearing. One other beneficial effect of grooving is that a convenient trap is provided for wear particles and other foreign matter which might otherwise enter the bearing.

Geometry of a completely boundary-lubricated journal operating within a sleeve is shown in Fig. 44. Under action of applied load, the journal will always bear against the bearing in the direction of load regardless of the magnitude of load and speed or the direction of motion of the shaft. Thus, the loaded region would be at position 2 as shown in Fig. 44. Load-carrying area, or area of contact, will lie between positions 7 and 8. Since the bearing cannot generate hydrodynamic film pressures under complete boundary lubrication because a continuous supply of lubricant is lacking, no harm can be done by grooves which pass through the load zone. However, removal of too much bearing material should be avoided to prevent excessive unit pressures in the load zone.

Two types of grooves acceptable for completely boundary-lubricated bearings using grease are shown in Fig. 45a and b. These grooves, while appearing to be complex, are easily generated on a grooving machine—an important fabricating consideration. Similar groove configurations may be used when oil is the lubricant. Notice that these grooves do not break through the ends of the bearing.

Another simple but good groove for grease lubrication is the arrangement in Fig. 45c. The wide circular groove can be obtained merely by allowing a gap to exist between two bearings in the same housing to provide a generous grease reservoir. The same effect could be achieved by cutting a wide circular groove in the shaft.

### ► Grooving for Mixed-Film Lubrication

Geometry of a mixed-film lubricated bronze bearing is similar to that of a complete boundary-lubricated bearing in that the journal bears against the bearing in the direction of load, Fig. 46. However, for this situation a portion of applied load is supported by fluid pressures generated in local areas of the load zone. Likewise, some metal-to-metal contact occurs in region 7 to 8. When mixed-film conditions exist, this area of contact is somewhat smaller than that for a completely boundary-lubricated sleeve bearing. In fact, contact area is quite small unless some wearing-in has occurred.

Previous definition of mixed-film conditions states that the bearing must be supplied continuously with lubricant and that the lubricant must be picked up by the shaft and carried to the load zone before escaping from the bearing. Use of grooves in the load zone should also be avoided for this type of

lubrication. Recommended grooves for mixed-film operation are the same as for full-film lubrication except that the oil should be picked up by the shaft at a point closer to the region of minimum film thickness, location 4 in Fig. 32. If load is continually changing direction, an axial groove, or even just a single hole, may be used in a relatively lightly loaded region.

### ► Groove Dimensions

Common groove cross sections are shown in Fig. 47. The V-groove is generally recommended from a fabrication standpoint. Specifying chamfered corners at the bearing surface, as indicated on the V-groove in Fig. 47, is an important detailing practice. This step is necessary in all grooving so that lubricant which adheres to the rotating shaft is not wiped off by burrs or projections at groove corners. However, chamfered corners are not as critical for boundary-lubricated bearings. Inside corners at the bottom of grooves should be rounded to avoid local stress concentration.

**Width and Depth:** Recommended groove widths and depths for various bore diameters are given in Fig. 48 and are the same for all oil grooving regardless of groove configuration. Width curve  $a$  is plotted in 1/32-in. increments since groove width is not critical and need not be specified to a closer tolerance. Also, groove depth  $b$  need not be specified any closer than the nearest 1/64 in.

Before groove depth from Fig. 48 is specified, value  $b$  should be compared with 1/3 the wall thickness of the bearing material. If the value of  $w/3$  is less than  $b$ , then  $w/3$  should be specified as the groove depth. On the other hand, if  $b$  is less than 1/3 the wall thickness, then the value of  $b$  should be specified. If groove depth is always kept less than 1/3 the wall thickness, the bearing is not unduly weakened at the location of the groove.

When grease is used as the lubricant, the groove should be widened somewhat to provide free passage for the less-mobile grease. For this purpose, groove width may be increased up to 1.5 times the values recommended for oil in Fig. 48.

**Configuration:** Dimensioning of groove configuration, or plan view, will depend upon type of grooving used and size of the bearing. As a general rule, grooves should not break out the ends of the bearing unless the bearing is to be lubricated from one end. However, grooving may approach the bearing ends. Recommended distance between groove and end of the bearing should be more than 0.05 times the length of the bearing, Fig. 49a and b. For short bearings, minimum distance from end of the bearing to grooving should be 1/8 in.

Oil-inlet holes should, as has been stated, be located in the low-pressure region of the bearing. If used in conjunction with grooving, inlet holes should be centered about the groove. Size of the

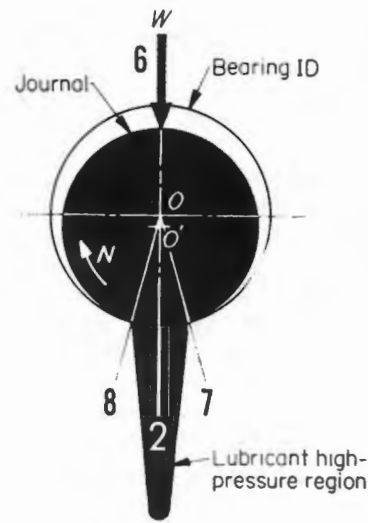


Fig. 46—Geometry of mixed-film lubricated sleeve bearing.

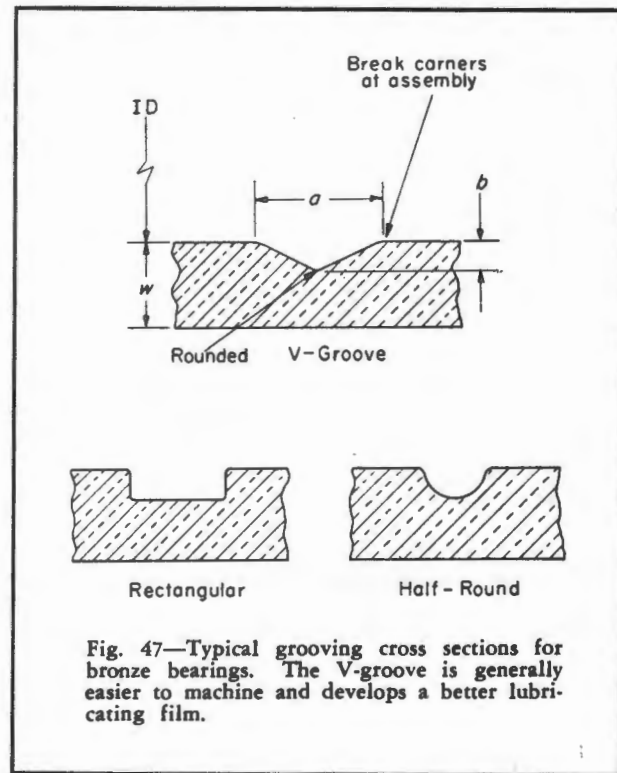


Fig. 47—Typical grooving cross sections for bronze bearings. The V-groove is generally easier to machine and develops a better lubricating film.

inlet hole should be at least as large as the width of the groove it is supplying and preferably 1.5 times groove width for the smaller size bearings. For vertically mounted bearings, the oil-inlet hole should be located closer to the top to help equalize oil flow. In nonpressure-fed applications, the inlet hole should be located approximately within the range shown in Fig. 49c.

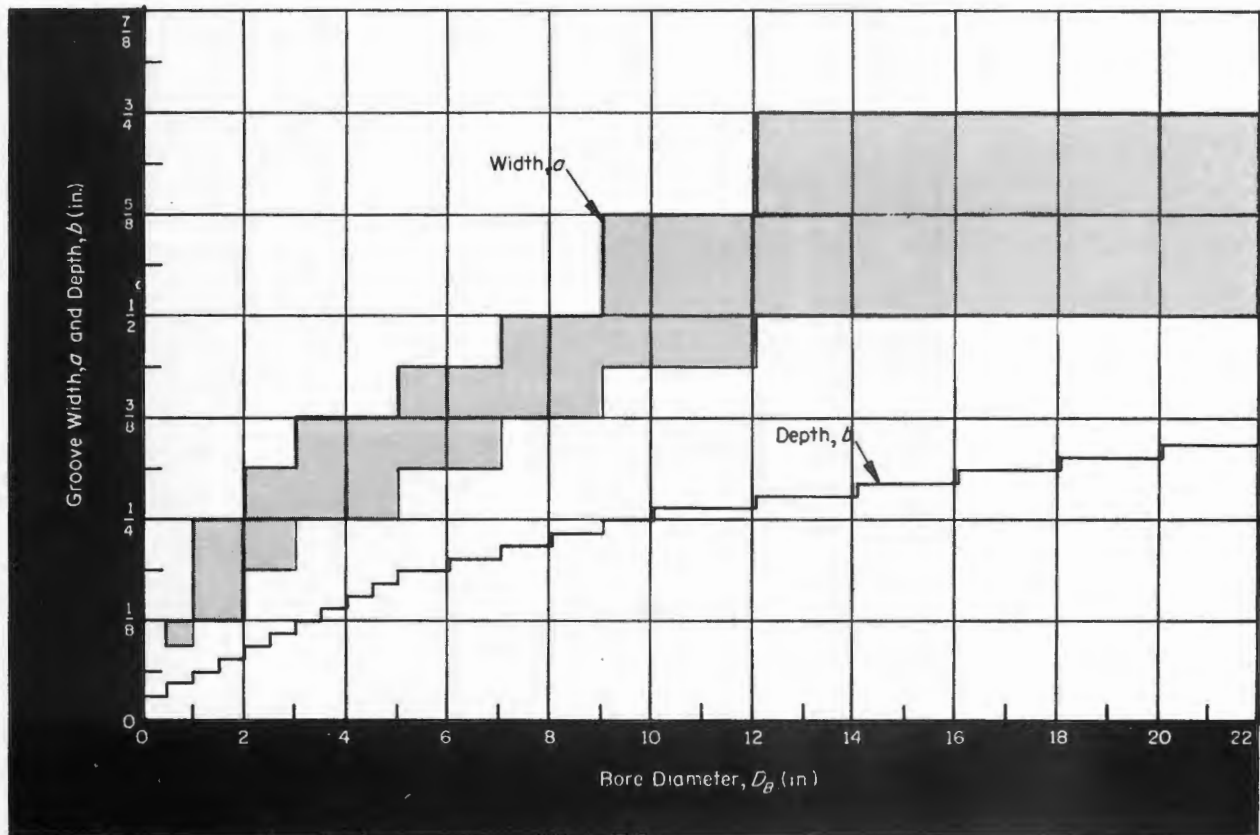


Fig. 48—Recommended groove dimensions for oil-lubricated bronze bearings. For grease-lubricated bearings, increase groove width up to 1.5 times values given.

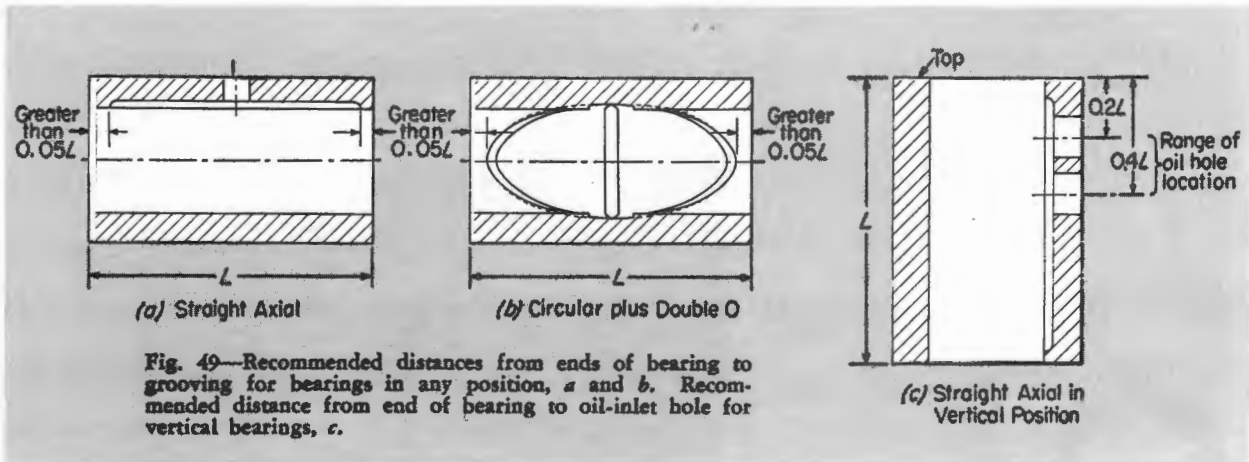


Fig. 49—Recommended distances from ends of bearing to grooving for bearings in any position, *a* and *b*. Recommended distance from end of bearing to oil-inlet hole for vertical bearings, *c*.



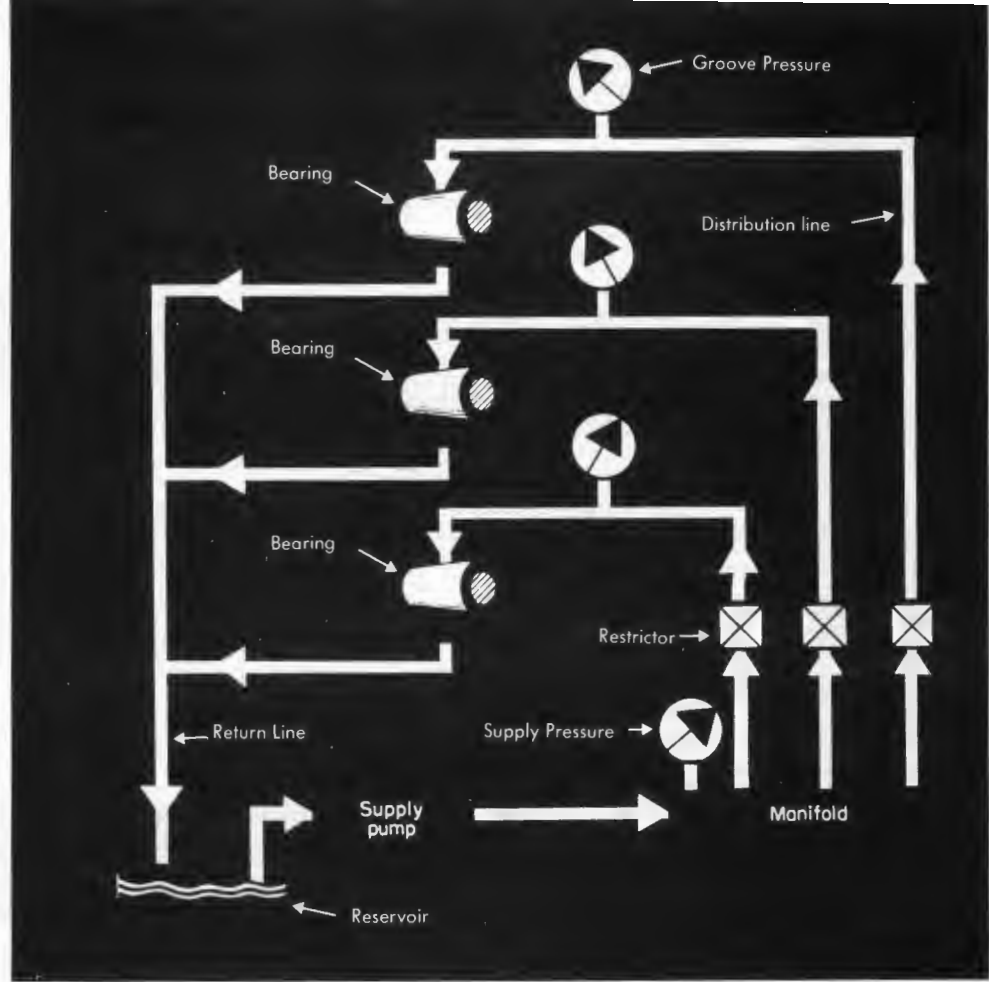


Fig. 50—Pressurized lubricant supply system containing three different bronze bearings.

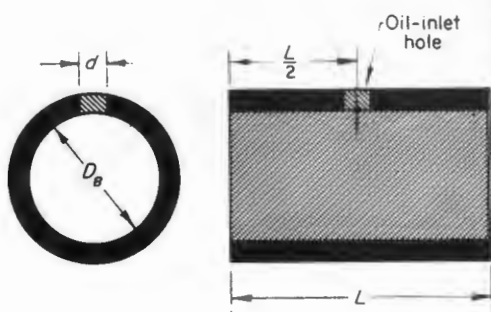
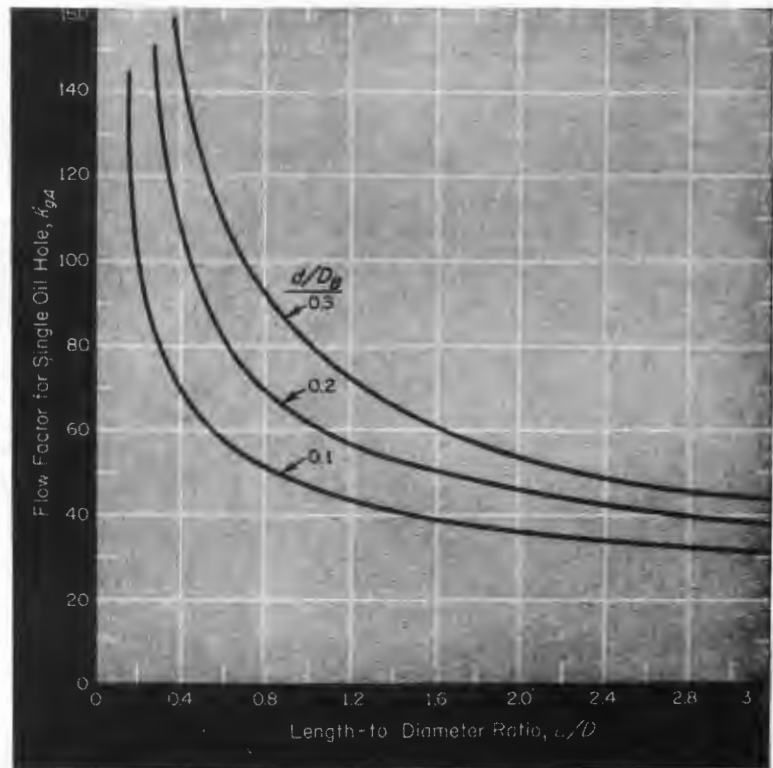


Fig. 51—Flow factors for centrally located single-hole oil groove with pressure lubrication ( $D_g \approx D$ ).



## CHAPTER 8

# Techniques for Supplying Lubricant

ONCE the actual bearing configuration is defined, there remains the problem of getting lubricant to the bearing in quantities sufficient to permit the desired type of operation. Does the application require a pressurized system to insure an adequate flow rate? Or will one of the simpler nonpressure methods be satisfactory?

This chapter will attempt to answer these questions by presenting, where possible, charts and equations for determining probable oil flow. With this information, the conditions required by the flow-rate equations for full-film and mixed-film (not boundary) lubrication can be satisfied.

In Chapter 2, theoretical flow rate for full-film lubrication is found from

$$Q = k_q m D^3 N \times 10^{-6} \quad (9)$$

with side-leakage flow factor  $k_q$  obtained from Fig. 17. If the amount of oil supplied is less than  $Q$ , the journal will operate more eccentrically within the bearing than indicated in Fig. 13.

Also from Chapter 2, the equation for minimum flow rate for full-film lubrication is

$$Q' = 29.3 \times 10^{-9} \left( L + 0.0043 \frac{W}{D} \right) m D^3 N \quad (10a)$$

Hence, to insure full-film conditions the bearing

### Nomenclature

- $C$  = Radial clearance, in.
- $D$  = Journal diameter, in.
- $D_B$  = Bearing bore diameter, in.
- $d$  = Diameter of oil-inlet hole (Fig. 51), in.
- $h$  = Wicking distance from top of oil level to journal, in.
- $k_g$  = Groove flow factor
- $k_{qA}$  = Flow factor for pressure-lubricated single oil-inlet hole
- $k_{qB}$  = Flow factor for pressure-lubricated straight-axial distribution groove
- $k_{qC}$  = Flow factor for pressure-lubricated circular groove
- $k_q$  = Side-leakage flow factor
- $L$  = Bearing length, in.
- $L'$  = Length from centrally located circular groove to end of bearing (Fig. 53), in.
- $L'_1, L'_2$  = Respective lengths from noncentrally located circular groove to each end of bearing (Fig. 53), in.
- $l$  = Length of straight-axial distribution groove (Fig. 52), in.
- $m$  = Clearance factor  
=  $1000 (2C)/D$
- $N$  = Rotational speed of journal, rpm
- $p_g$  = Pressure developed in groove of pressure-fed bearing, psi
- $Q$  = Side-leakage oil flow, or oil-flow feed rate, gpm
- $Q'$  = Minimum oil flow required for full-film lubrication, gpm
- $Q_g$  = Oil flow in grooved, pressure-fed bearing, gpm
- $W$  = Steady load to be supported, lb
- $Z$  = Lubricant absolute viscosity, centipoises
- $e$  = Journal eccentricity ratio

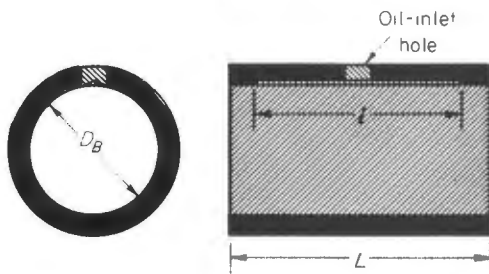
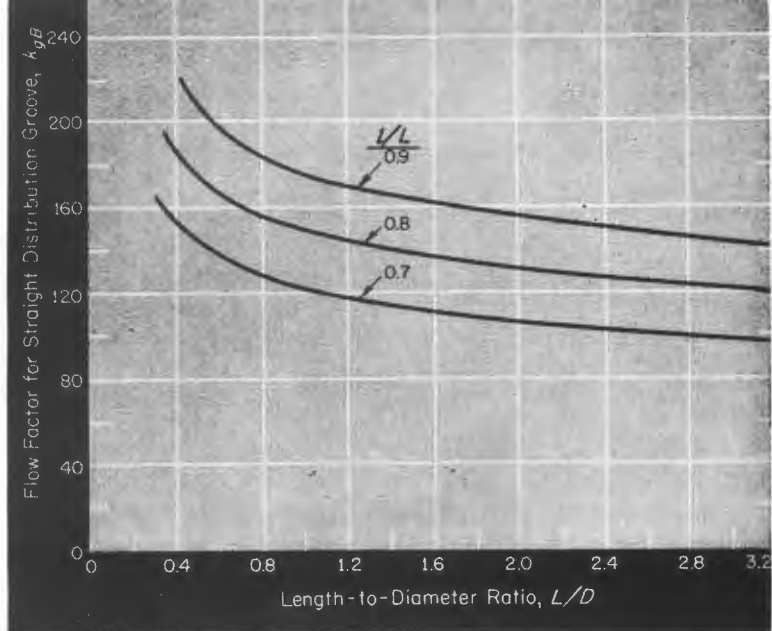


Fig. 52—Flow factors for straight-axial distribution groove with pressure lubrication ( $D_B \approx D$ ).



must be supplied continuously with a flow rate at least equal to  $Q'$  and preferably equal to  $Q$ . If oil flow is less than  $Q'$ , the bearing will operate under mixed-film conditions.

### Pressure-Fed Bearings

With pressure feeding, lubricant is supplied to the bronze bearing under external pressure. Depending upon bearing requirements, pressurized lubricant may be supplied continuously for full-film lubrication or intermittently for mixed-film lubrication. The intermittent method is used extensively in centralized lubricating systems to supply prescribed amounts of lubricant to many different bearings at periodic intervals.

When pressurized lubricant is continually supplied, pressure is generated within the bearing groove because the narrow clearance between bearing members restricts oil flow. Actual flow through the bearing depends upon lubricant pressure in the groove, bearing diameter, clearance, bearing length, lubricant viscosity, and groove configuration. When the journal is concentric within the bearing bore, flow through the bearing is

$$Q_g = \frac{k_g m^3 p_g D^3}{Z} \times 10^{-6} \quad (32)$$

Under load, the journal becomes eccentric with respect to the bearing and flow increases accordingly. Equation for the oil-flow rate of a loaded bearing is

$$Q_g = \frac{k_g m^3 p_g D^3}{Z} \left( 1 + \frac{3e^2}{2} \right) \times 10^{-6} \quad (33)$$

Values for groove flow factor  $k_g$  in the following discussion are identified as  $k_{gA}$ ,  $k_{gB}$ , etc., but can be substituted directly into Equation 33. Actually,  $D_B$  should be used in Equations 32 and 33 instead of  $D$ ,

but the numerical error is negligible. Use of shaft diameter agrees with previous flow equations and makes original  $L/D$  ratio valid for graphs which follow.

**Oil-Groove Pressure:** If only a single bearing is supplied with pressurized lubricant from a pump, groove pressure will be the same as the discharge pressure of the pump, assuming no line losses. If several bearings are supplied with pressurized lubricant from the same source, and again assuming no line losses, groove pressure developed within each bearing will be the same. Thus, if all bronze bearings in a system have identical configurations and operate under similar conditions, flow requirements for each bearing will be the same.

However, flow requirements are usually different for each bearing in a system and consequently require different groove pressures. In these situations, recommended practice is to install a restrictor in the distribution line to each bearing. Restrictors are merely devices inserted in the oil line to promote a pressure drop between manifold and bearing. Such devices include orifices, capillary tubes, needle valves, and flow-control valves. In addition to making the distribution line for each bearing independent of the others, restrictors also make full supply pressure immediately available to any bearing if required.

A schematic diagram of a lubricant supply system feeding three bronze bearings is shown in Fig. 50. Each bearing has a different groove pressure and requires an individual restrictor. Supply pressure is the same for all bearings. Proper "sizing" of restrictors to obtain desired flow rates requires some knowledge of pressure-flow relationships for the particular type of restrictor used.

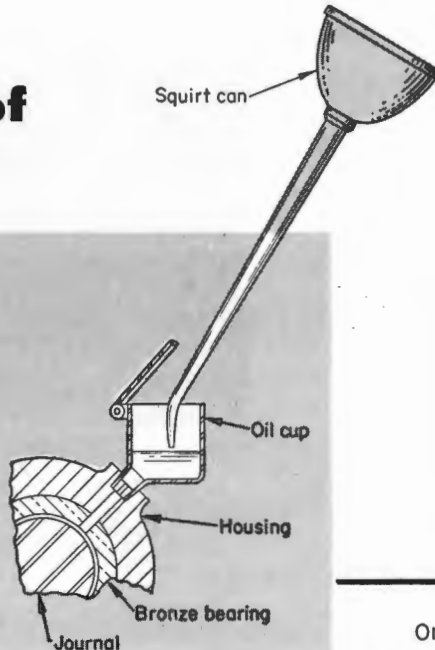
Thus, it is important to remember that  $p_g$  in Equation 33 is the groove pressure developed within the bearing. With  $p_g$  correctly determined, probable oil-flow rates can be predicted.

**Single Oil-Inlet Hole:** Flow factors for a single oil

# Nonpressure Methods of Applying Lubricant

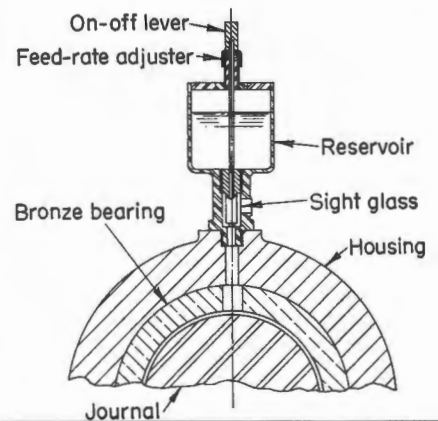
## Hand Oiling

Many bronze bearings operating under boundary and mixed-film lubrication conditions are adequately lubricated by periodic hand oiling with a squirt can—provided oil application is not too infrequent. The bearing to be oiled is usually provided with a dustproof cover, and the lubricant, of course, should be clean. Actual oil-feed rate varies from copious shortly after oiling to nothing when the reservoir runs dry. Hence, if hand oiling is to be used, the bearing should be designed for complete boundary lubrication.



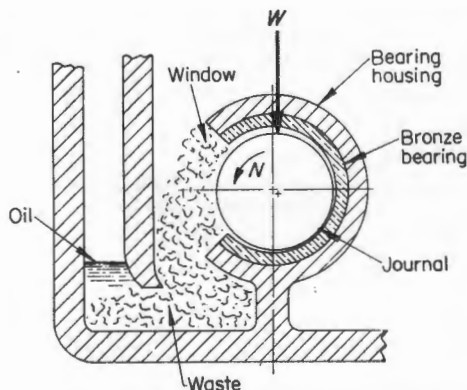
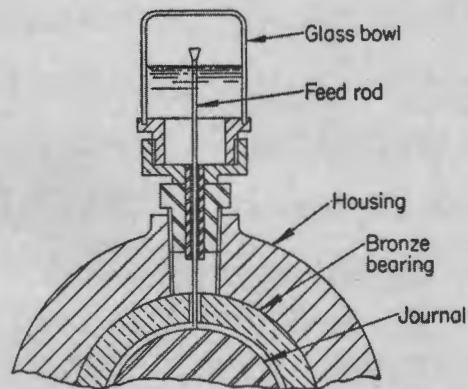
## Drop-Feed Oilers

A big improvement over hand oiling is the use of drop-feed oilers. Flow rate can be adjusted and essentially held constant if oil level and temperature are not allowed to vary appreciably. Dropping rates ranging from 0 to 100 drops per minute are possible. Oil flow is normally started and stopped by hand, although such oilers can be made automatic by electrical or pressure devices.



## Bottle Oilers

A typical bottle oiler consists essentially of a clear "bottle" containing an extension tube. The bottle is inverted over the bearing and the extension tube rides the journal. Movement of the rotating journal causes the extension tube to move up and down, thus pumping oil from the bottle to the bearing. Delivery rate of oil depends upon the amount of motion available for pumping the lubricant. Oil flow is also very sensitive to temperature, which is an advantage because more lubricant will flow at higher temperatures when it is required. Another advantage is that oil is pumped only when the bearing is in operation. Because the lubricant supply is sealed, bottle oilers are widely used in dusty atmospheres.



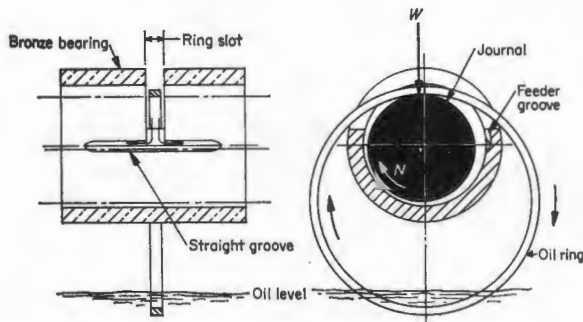
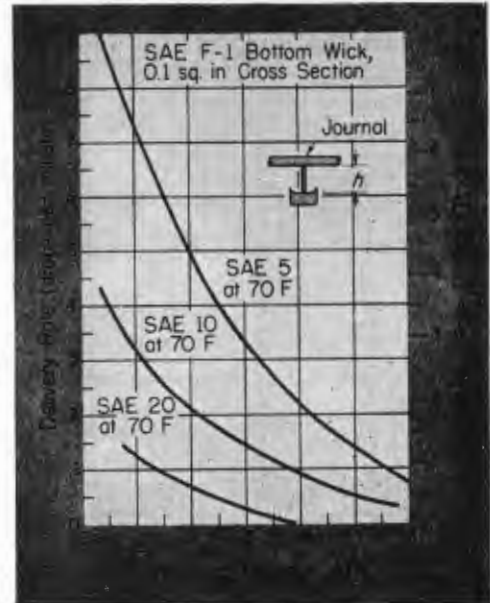
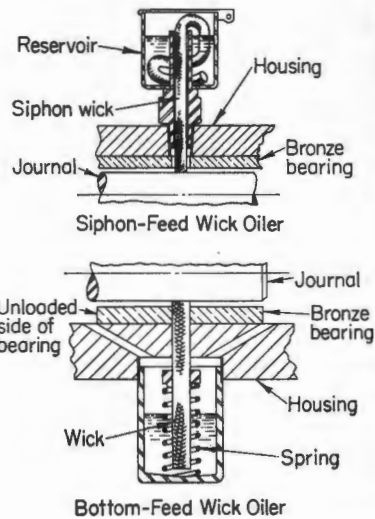
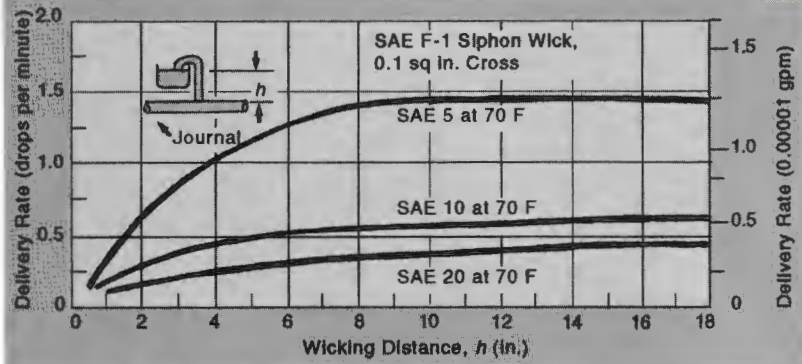
## Waste and Pad Oilers

Capillary action accounts for the lubricating effectiveness of oilers using waste or pads which contact an oil reservoir at one end and the rotating shaft at the other end. Needless to say, oil-feed rates are difficult to predict. Maintaining oil in the reservoir is important and permits the waste to supply copious amounts of lubricant. Such oilers are used extensively for railroad freight-car journal bearings. The waste, in this case, provides effective lubrication under adverse conditions.

## Wick Oilers

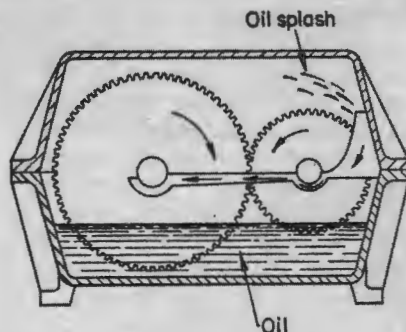
Use of a wick to transport lubricant from reservoir to bearing is a tried and true method. Lubricant moves through the wick due to capillary action. Although the method may appear to be quite crude, wick-feed oilers are efficient when properly designed. Flow rates possible with SAE F-1 felt for siphon and bottom type wicks may be estimated from the curves. Data presented are for wick areas of 0.10 sq in. for SAE lubricants at 70 F. Oil delivery rate is directly proportional to total cross-sectional area of the wick and inversely proportional to viscosity of the lubricant. At higher temperatures, oil flow will increase since lubricant viscosity decreases. Wicks also serve as effective filters to insure that only clean oil reaches the bearing.

Also in use are wooden wicks in which oil is drawn through the wood by capillary action. Normally, wooden wicks are built into the bearing along with a reservoir that is machined in the housing or bearing. Thus, all external wicks and reservoirs are eliminated.



## Ring and Chain Oilers

Copious amounts of lubricant can be supplied by the relatively simple means of hanging a ring or chain on the journal in a horizontal bronze bearing. At low speeds, the ring moves with the shaft. If the bottom of the ring is immersed in oil, it will, as it rotates, pick up and deliver oil to the bearing. If the oil is allowed to return to the reservoir to maintain the oil level, a constant supply of lubricant will be delivered to the bearing. Ring diameters are about 1½ to 2 times the diameter of the bearing. Minimum oil level should cover at least a 30-deg section of the ring. Chains can supply more lubricant than rings and require less space. However, they are not suitable for high speed because they tend to swing around the shaft. Oil supply rates of from ¼ to 1 gpm are easily obtained with rings or chains.



## Splash and Bath Oiling

Often effective lubrication can be provided by splashing lubricant into suitable channels which then direct the lubricant to the bearings. Splashing may be accomplished by parts of the machine moving through a constantly maintained level of oil within the machine. In this case cleanliness of the inside of the machine is important to avoid contaminating the lubricant supplied to the bearing. Bronze sleeve bearings may also be operated in a bath of lubricant.

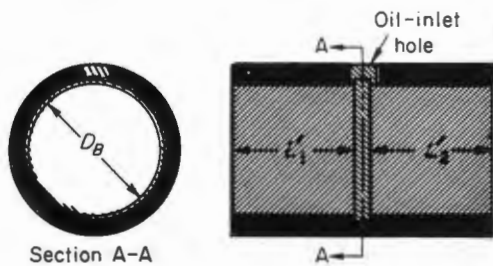
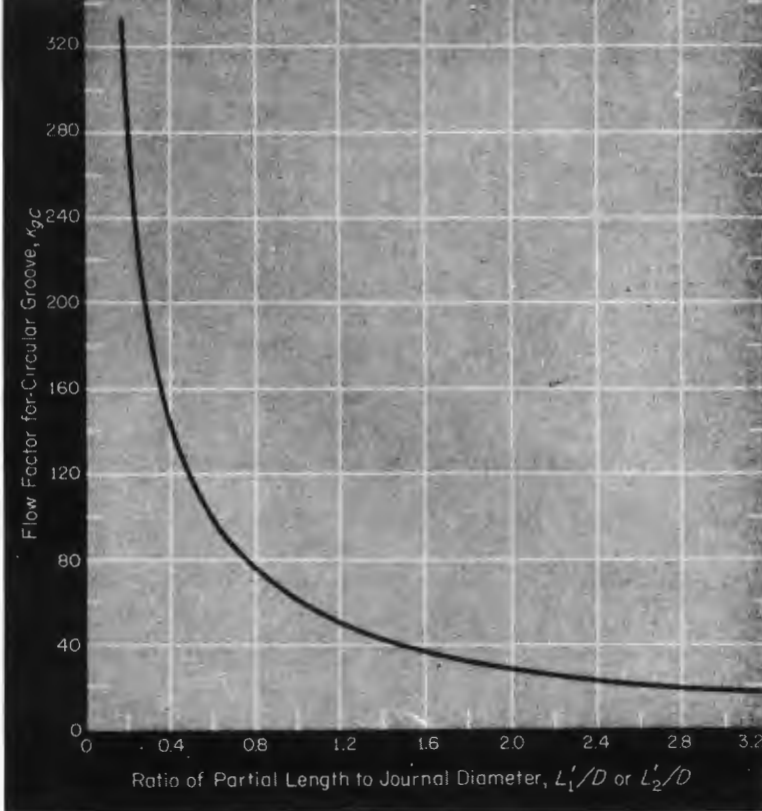


Fig. 53—Flow factors for circular groove with pressure lubrication ( $D_B \approx D$ ).



hole, considered as a groove, can be obtained from Fig. 51 for various ratios of inlet-hole diameter to bore diameter. Oil flow can then be determined from Equation 33.

If calculated oil flow for the pressurized lubricant is more than that obtained from Equation 9 for full-film lubrication, the bearing should operate at the specified eccentricity ratio. If calculated flow is less than full-film requirements, the journal will be more eccentric within the bearing than anticipated. To increase the flow, either  $d/D_B$  ratio or pressure may be increased. Equation 33 and Fig. 51 may also be used to determine pressure required at the hole to yield the proper flow when hole size is fixed.

**Straight-Axial Groove:** Flow factors for straight-axial distribution grooves can be obtained from Fig. 52 for several ratios of groove length to bearing length. Ratios of  $l/L$  between 0.7 and 0.9 are recommended. To evaluate flow from a pressurized axial groove first requires determination of  $k_{gB}$  from Fig. 52 for particular  $l/L$  and  $L/D$  ratios. Since the other quantities are known, substituting  $k_{gB}$  in Equation 33 yields the flow rate.

If a pressurized feeder is used with a straight-axial distribution groove, oil flow will be somewhat more than for a plain, straight groove. However, use of flow factors from Fig. 52 for straight grooves is sug-

gested when evaluating flow from a combination feeder and straight groove.

**Circular Groove:** Flow factors for a centrally located circular groove can be obtained from Fig. 53. In this curve the abscissa is  $L'/D$  with  $L'$  being the distance from edge of the groove to end of the bearing. After the value of  $k_{g0}$  is determined, substitution in Equation 33 permits evaluation of flow through one end of the bearing. Total flow will be twice this value if the groove is centered within the bearing.

When a circular groove is not axially centered within the bearing, two different values of  $L'$  exist:  $L'_1$  and  $L'_2$ . Hence, the flow factor for each end of the bearing will be different. Therefore, Equation 33 must be used twice to evaluate the flow for each section. Total oil flow is then the sum of these two values.

## Nonpressure-Fed Bearings

Bronze bearings are supplied with lubricant by many methods that do not use external pressure. Seven of the more common ways to apply lubricant are shown and discussed. Where possible, information on flow rates is furnished for the individual method.

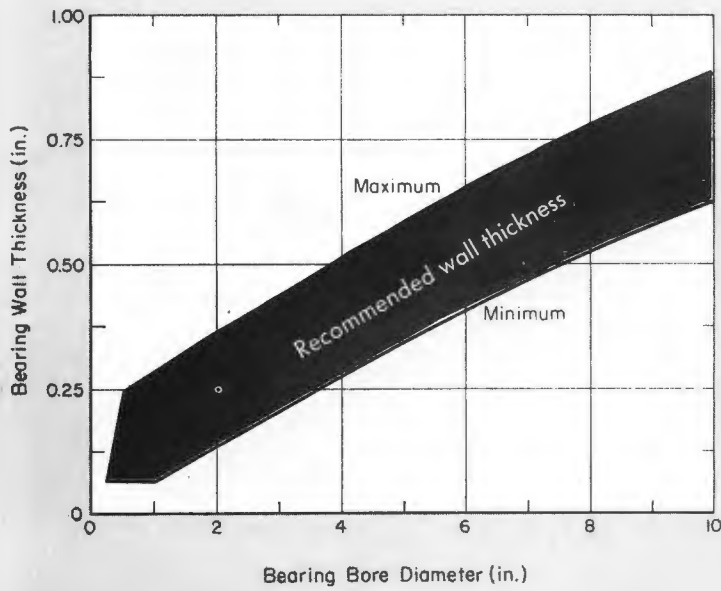


Fig. 54—Range of recommended wall thicknesses for cast bronze sleeve bearings as determined by bearing bore diameter. For bearings larger than 10-in. diameter, specific manufacturers should be consulted.

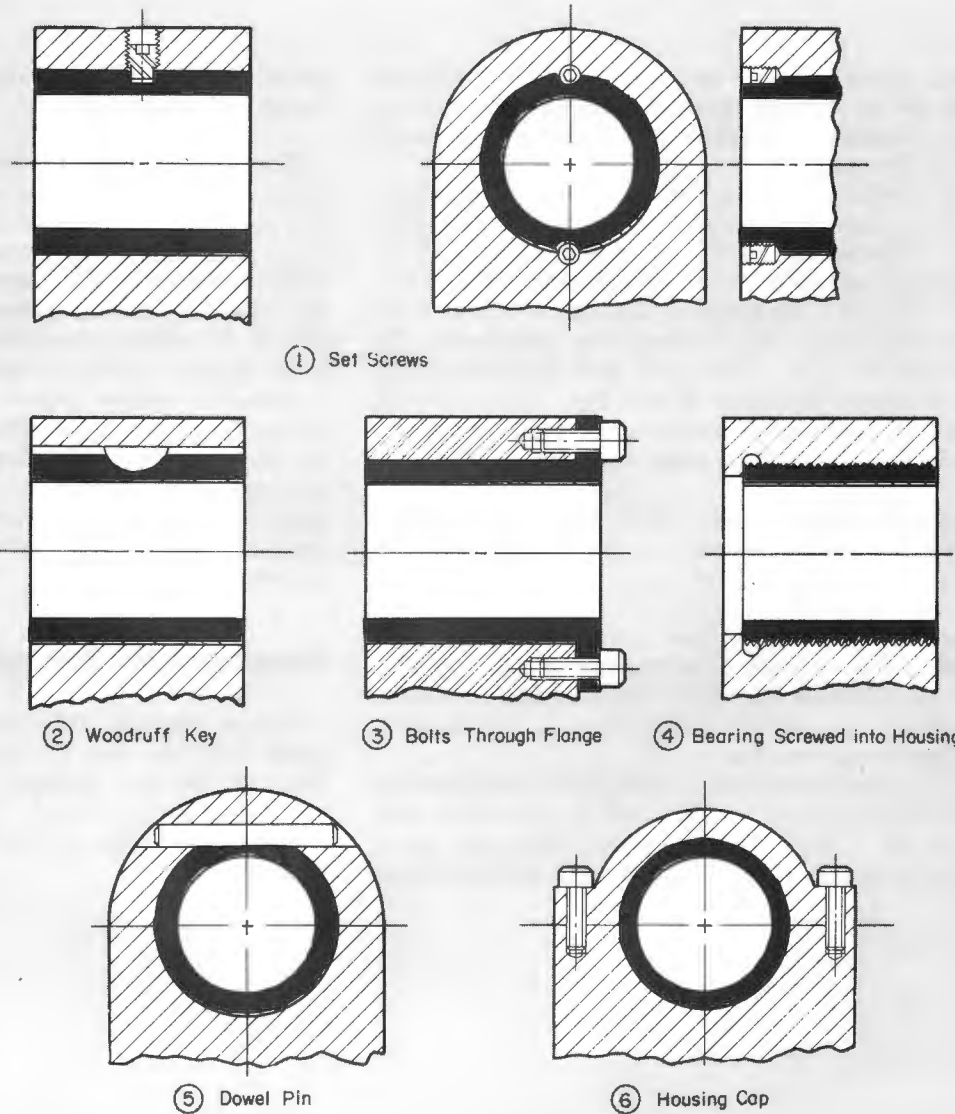


Fig. 55—Common methods used to prevent movement of bearing within the housing.

## CHAPTER 9

# Bearing Fabrication and Assembly

**L**IKE any other engineering problem, the major efforts in designing a bearing must be concluded with small but common-sense details. Without such practical considerations as proper installation or compatibility with other operating parts, even a perfect bearing could fail.

Wall thickness of the bearing, the type of housing material, and methods of retaining the bearing within the housing are some of the final, and often inter-related, points to be specified. Proper relationship between bearing and journal hardnesses must be established to prevent rapid wear and expensive replacement. Surface finish is also important in controlling wear. And after the bearing has been assembled and placed in service, maintenance personnel will need information about care of the bearing and its lubricating accessories and about replacement when it becomes necessary.

**Bearing Wall Thickness:** Bronze sleeve bearings are

normally contained within a housing or shell. Criteria for wall thickness is that the bearing, when installed in its housing, should provide adequate strength to support the imposed loads without elastic or thermal distortions which would destroy the "built-in" geometry of the bearing. In general, increasing wall thickness is required for increasing bore diameter. But ratio of wall thickness to bore diameter decreases with increasing bore diameter.

Thin bearing walls and heavy housings provide more strength than the opposite arrangement, but heat transfer from the bearing may be impeded. If severe wear of bearing material is permissible and expected, adequate material thickness should be provided. Anticipated temperature rise is another consideration when wall thickness is specified. Large variations in temperature, and different coefficients of thermal expansion for bearing and housing materials, combine to produce expansion and contraction forces that make bearing dimensions and fits diffi-

**Table 14—Recommended Tolerances for Nominal Housing Bores Using Press or Shrink Fits**

Nominal Bearing OD and Housing ID (in.)	Tolerance on Nominal Bearing OD (in.)	Tolerance on Nominal Housing ID (in.)	Range of Interference Fit (in.)	Average Interference Fit (in.)
0 to 0.5	+0.002 to +0.003	+0.0015 to +0.0020	zero to 0.0015	0.00075
0.5 to 1	+0.002 to +0.003	+0.0010 to +0.0015	0.0005 to 0.0020	0.00125
1 to 1.5	+0.002 to +0.003	+0.0005 to +0.0010	0.0010 to 0.0025	0.00175
1.5 to 2	+0.002 to +0.003	+0.0000 to +0.0010	0.0010 to 0.0030	0.0020
2 to 2.5	+0.002 to +0.003	+0.0000 to +0.0010	0.0010 to 0.0030	0.0020
2.5 to 3	+0.002 to +0.003	-0.0005 to +0.0005	0.0015 to 0.0035	0.0025
3 to 4	+0.003 to +0.005	+0.0005 to +0.0015	0.0015 to 0.0045	0.0030
4 to 5	+0.003 to +0.005	+0.0000 to +0.0010	0.0020 to 0.0050	0.0035
5 to 6	+0.003 to +0.005	-0.0005 to +0.0005	0.0025 to 0.0055	0.0040
6 to 8	+0.003 to +0.005	-0.0010 to +0.0000	0.0030 to 0.0060	0.0045



cult to maintain. Varying clearance within the bearing can result.

When a bearing is pressed or shrunk into the housing, unequal expansions can also cause stressing of both members. If the bearing material yields, cooling may change the original interference fit and result in a loose bearing. However, these difficulties may be minimized by proper design.

A final requirement, mentioned in Chapter 7, is that wall thickness should be at least three times the depth of any grooving used in the bearing. Suitable wall thicknesses and the amount of thickness variation allowed for cast bronze bearings are recommended in Fig. 54.

**Methods of Retaining Bearings:** Many different techniques are used to insure that a bronze bearing stays put within the housing. The method used depends upon the particular application but requires first that the unit lend itself to convenient assembly and disassembly. One goal to keep in mind is that the bearing wall should be uniformly thick to prevent introduction of weak points in the construction which might lead to elastic or thermal distortion.

**Press or Shrink Fit:** One common and satisfactory technique for retaining the bearing is to press or shrink the bearing in the housing with an interference fit. Uniform wall thickness over the entire bearing is easily maintained by either of these processes.

Standard stock bushings with finished inside and outside diameters are available in sizes up to approximately 5 in. ID. Stock bushings are commonly provided with 0.002 to 0.003 in. over nominal on OD sizes of 3 in. or less. For ODs greater than 3 in., actual OD is 0.003 to 0.005 in. over nominal. Since these tolerances are built into standard bush-

ings, the amount of press fit is controlled by housing bore. Fits recommended for general applications are listed in Table 14.

For high-temperature work, interference fits in Table 14 may need adjusting to account for expansion differences between bearing and housing and to avoid yielding of the bearing material. Thin-walled housings or housings other than steel or cast iron require somewhat lighter press fits. For fabricated bearings, tolerances of both bearing OD and housing bore should be specified to produce the recommended interference fits.

As a result of a press or shrink fit, the bore of the bearing material "closes in" by some amount. In general, this diameter decrease is approximately 70 to 100 per cent of the interference fit. Any attempt to accurately predict the amount of close-in, in an effort to avoid final clearance machining, should be avoided.

Shrink fits may be accomplished by chilling the bearing in dry-ice and alcohol, or in liquid air. These methods are easier than heating the housing and are preferred. Dry ice in alcohol has a temperature of  $-110$  F and liquid air boils at  $-310$  F.

When a bearing is pressed into the housing, the driving force should be uniformly applied to the end of the bearing to avoid upsetting of the bearing. Also important are the mating surfaces, which must be clean, smoothly finished, and free of machining imperfections.

**Keying Methods:** Many different ways are used to fix the position of the bearing with respect to its housing by "keying" the two together. Possible keying methods shown in Fig. 55 include:

1. Set screws.
2. Woodruff keys.
3. Bolted bearing flanges.
4. Threaded bearing OD.
5. Dowel pins.
6. Housing caps.

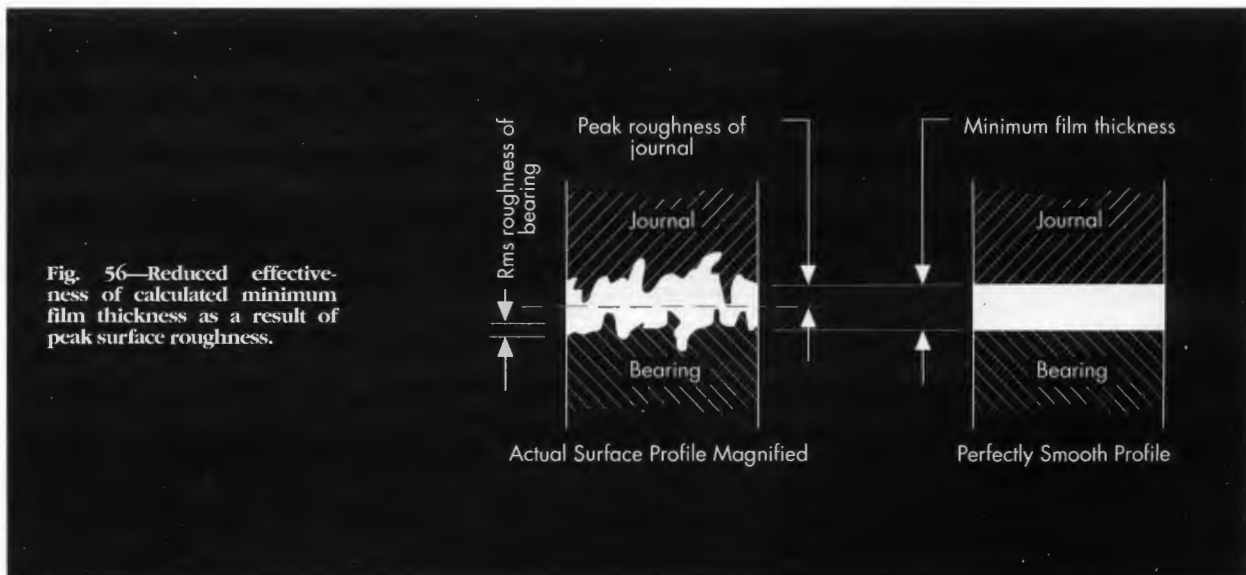


Fig. 56—Reduced effectiveness of calculated minimum film thickness as a result of peak surface roughness.

Factors to keep in mind when selecting one of these methods are:

1. Maintain uniform wall thickness of the bearing material if at all possible, especially in the load-carrying region of the bearing.
2. Provide as much contact area as possible between bearing and housing. Mating surfaces should be clean, smooth, and free from imperfections to facilitate heat transfer.
3. Prevent any local deformation of the bearing, if possible, that might result from the keying method. Machining after keying is recommended.
4. Investigate possibilities of bearing distortion resulting from the effect of temperature on the particular keying method.

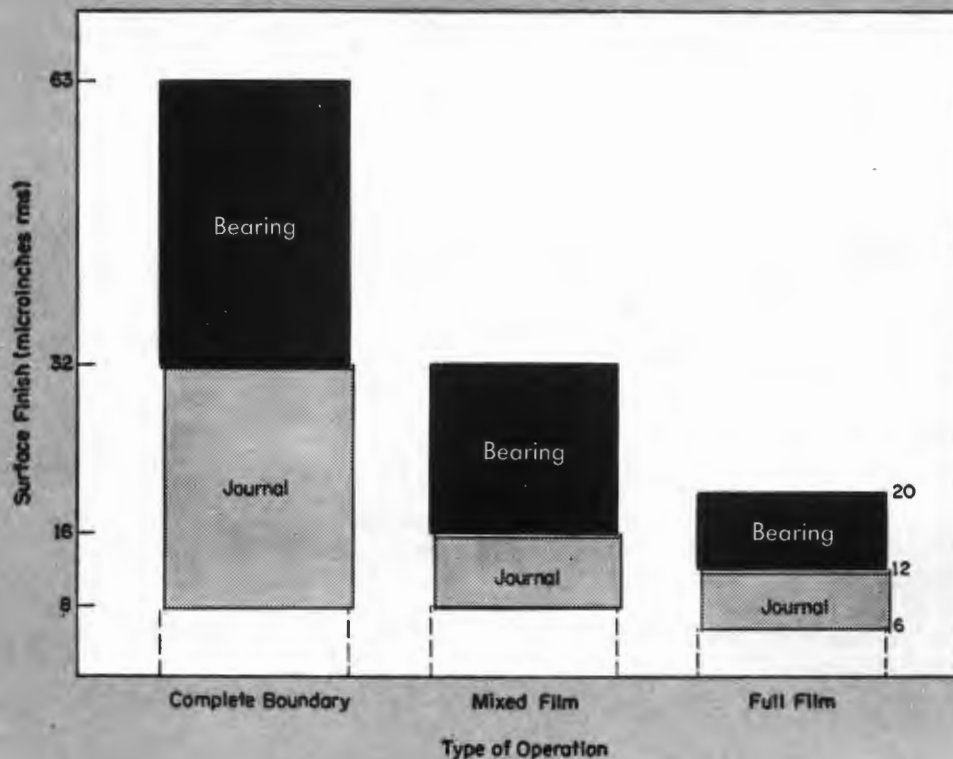
**Bearing and Journal Hardness:** Even in well-lubricated full-film sleeve bearings, momentary contact between journal and bearing may occur under such conditions as starting, stopping, or overloading. In mixed-film and boundary-film lubricated sleeve bearings, continuous metal-to-metal contact occurs. Hence, to allow for any necessary wearing-in, the

journal is usually made harder than the bearing material. This practice allows the effects of scoring or wearing to be inflicted on the more easily replaced bearing rather than on the more expensive shaft. As a general rule, recommended Brinell hardness of the journal is at least 100 points harder than the bearing material.

The softer cast bronzes are those with high lead content and very little tin. Such bronzes give adequate service in boundary and mixed-film applications where full advantage is taken of their excellent "bearing" characteristics. High-tin, low-lead content cast bronzes are the harder bronzes which have higher ultimate load-carrying capacity. Accordingly, higher journal hardnesses are required with these bearing bronzes. Aluminum bronze requires a journal hardness in the range of 550 to 600 Bhn.

Harder bearing materials, in general, require better alignment and more reliable lubrication to minimize local heat generation if and when the journal touches the shaft. Also, abrasives which find their way into the bearing are a problem for the harder bearing

Fig. 57—Recommended surface finishes for the three types of sleeve bearing operation. When proper wearing in occurs, a satisfactory finish will usually develop in the bearing.



materials because they cannot readily embed themselves in the bearing as they would in a softer material. Hence, lubricant cleanliness is also a more important consideration for the harder materials. However, clean lubricant is important for the best performance of any material.

**Surface Finish:** Whether bearing operation is complete boundary, mixed film, or fluid film, surface finish of journal and bearing must receive careful attention. In applications where operation is hydrodynamic or full film, peak surface variations should be less than the expected minimum film thickness. Otherwise, peaks on the journal surface will contact peaks on the bearing surface, Fig. 56, with resulting high friction and temperature rise. Ranges of surface roughness obtained by various finishing methods are listed in Table 15.

In general, better surface finishes are required for full-film bearings operating at high eccentricity ratios because full-film lubrication must be maintained with small clearances, and metal-to-metal contact must be avoided. Also, harder materials generally require better surface finishes. Surface finish requirements for boundary and mixed-film applications may be somewhat relaxed since bearing wear-in will smooth the surfaces if given an opportunity.

Fig. 57 is a general guide to the ranges required for bearing and journal surface finishes. Specifying a particular surface finish in each range can be simplified by following the general rule that smoother finishes are required for the harder materials, for high loads, and for high speeds.

**Maintenance and Replacement:** After a suitable design has been completed and the bearing has been properly fabricated and installed, successful trouble-free sleeve bearing operation lies in the hands of those

responsible for any bearing maintenance that may be required. Although this Manual is not intended to fully explore the problem of sleeve-bearing maintenance,

**Table 15—Surface and Peak Roughness**

Finishing Method	Surface Roughness	
	Range (microinches rms)	Peak Roughness (microinches)
Boring, broaching, reaming, finish turning	32-63	—
Commercial grinding	16-63	—
Precision grinding	4-16	4.5 × rms roughness
Honing and lapping	4-16	6.5 × rms roughness
Polishing and buffing	2-16	7 × rms roughness
Superfinishing	1-8	7 × rms roughness

no bearing design is entirely complete without consideration of particular maintenance requirements. Definite instructions regarding bearing maintenance should be issued to cover such items as:

1. Number and location of all bearings.
2. Specification of recommended lubricant.
3. Lubrication and inspection schedule and procedure.
4. Overhaul instructions applying to bearings.
5. Operating and maintenance instructions for accessory lubricating equipment if required.
6. Bearing replacement instructions emphasizing clearance requirements and proper installation procedure.

Properly designed and applied bronze sleeve bearings encounter difficulty for one of two reasons—either they are overworked with regard to speed and load, or they receive inadequate or faulty lubrication. Sleeve bearings, like humans, suffer when overworked, neglected, or treated badly. With reasonable care and attention, bronze sleeve bearings can, and do, fulfill a vital need in a highly mechanized age.

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\* This reference is highly recommended in that it contains a rather complete breakdown of many special areas of sleeve bearings and their lubrication. Specific references dealing with particular subjects are given and abstracted. Many of the references cited on this list are abstracted therein.

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