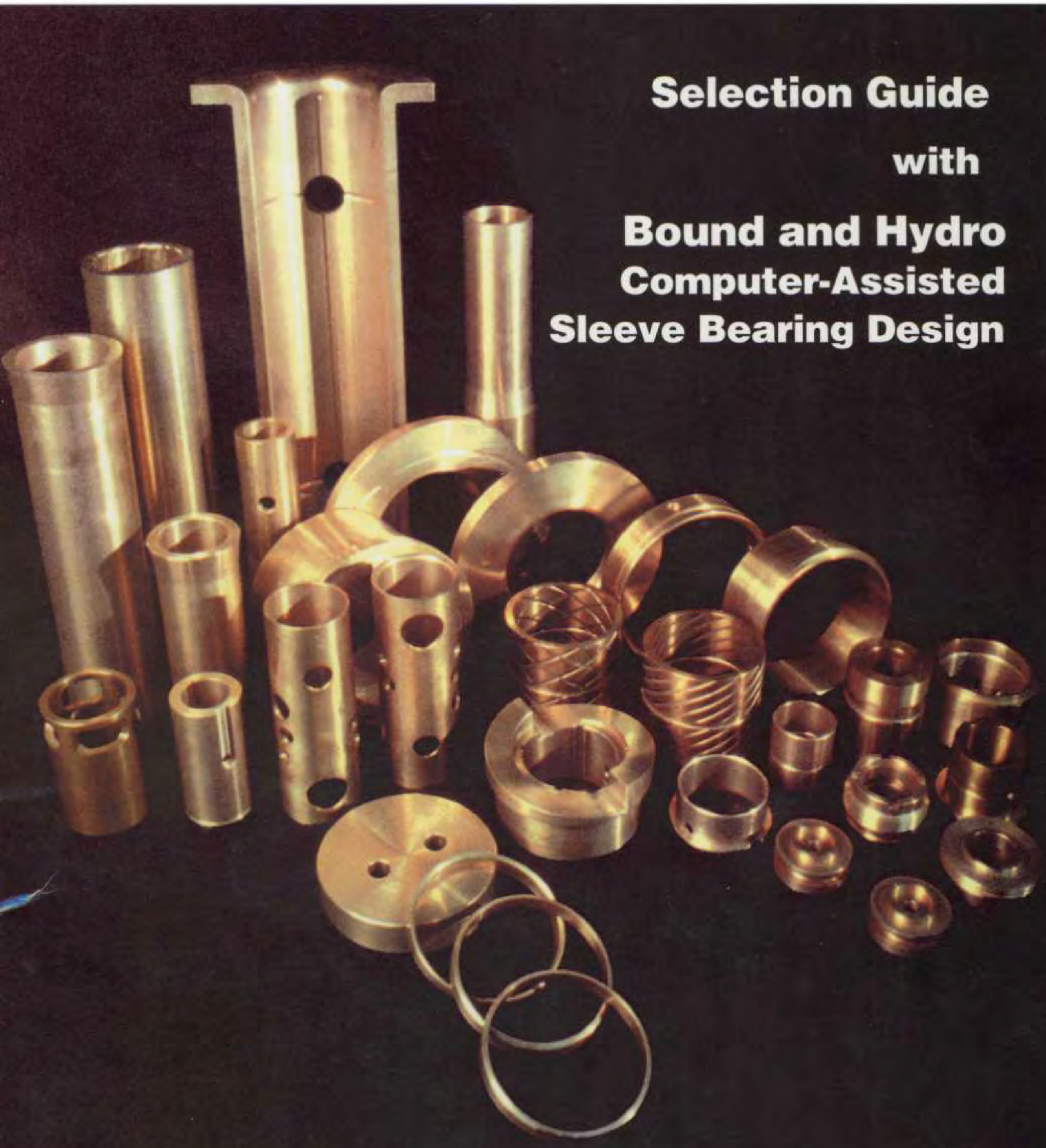
# CAST COPPER ALLOY SLEEVE BEARINGS




Non-Ferrous Founders' Society

CDA
Copper Development Association

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## INTRODUCTION TO LUBRICATED BRONZE BEARING DESIGN



## SELECTING CAST BEARING BRONZES




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## COMPUTER PROGRAMS



## REFERENCES AND APPENDICES

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# LIST OF SYMBOLS AND UNITS USED

| Symbol | Explanation | Units |
|---|---|---|
| $A$ | apparent area of surface contact | $m^2$ |
| $A_B$ | outer surface area of bearing housing | $m^2$ |
| $A_j$ | inside bearing area ($\pi BD$) | $m^2$ |
| $A_r$ | real area of surface contact | $m^2$ |
| $B$ | bearing width | m |
| $C$ | radial clearance $(D_b\text{-}D_j)/2$ | m |
| $C_D$ | diametral clearance | m |
| $C_{eff}$ | effective radial clearance at $\theta_{eff}$ | m |
| $d$ | diametral wear | m |
| $d^*$ | critical wear depth | m |
| $D$, $D_b$ | bearing diameter | m |
| $D_j$ | journal diameter | m |
| $E$ | modulus of elasticity | $N/m^2$ |
| $e$ | bearing eccentricity | m |
| $F$ | bearing radial load | N |
| $f$ | friction coefficient | - |
| $F_n$ | applied radial load | N |
| $H$ | hardness | $N/m^2$ |
| $h_c$ | convection heat transfer coefficient | $W/m^2 \cdot °K$ |
| $h_{min}$ | minimum film thickness | m |
| $h_{2min}$ | recommended minimum film thickness | m |
| $h_r$ | radiant heat transfer coefficient | $W/m^2 \cdot °K$ |
| $K$ | wear coefficient | - |

## LIST OF SYMBOLS AND UNITS USED\continued

| Symbol | Explanation | Units |
|---|---|---|
| $k$ | conduction heat transfer coefficient | W/m • °K |
| $k_a$ | heat transfer coefficient to ambient | W/m • °K |
| $L$ | total sliding distance | m |
| $N$ | journal speed | rps |
| $N_e$ | equivalent journal rotation | rps |
| $N_j$ | original journal rotation | rps |
| $N_b$ | bearing rotation | rps |
| $N_F$ | load vector rotation. | rps |
| $N_T$ | transition journal speed | rps |
| $P$ | nominal bearing stress $F_n/B \cdot D$ | $N/m^2$ |
| $P_{ta}$ | heat flow to ambient | W |
| $R$ | journal radius, D/2 | m |
| Ra | arithmetic average of surface roughness | µm |
| $S$ | Sommerfeld number | - |
| $T$ | total operational time | s |
| $t$ | sliding time | s |
| $t_e$ | equivalent sliding time | t |
| $t_s$ | total sliding time | s |
| $T_{regrease}$ | regreasing interval | h |
| $U$ | sliding velocity | m/s |
| $V$ | wear volume | $m^3$ |
| $V_f$ | bearing grease consumption rate | $cm^3/h$ |
| $V_L$ | total grease volume in bearing | $m^3$ |
| $w$ | power loss | W |
| $\alpha$ | thermal expansion coefficient | m/m • °K |
| $\varepsilon$ | eccentricity ratio | - |
| $\varepsilon_T$ | transition eccentricity ratio | - |
| $\eta$ | dynamic fluid viscosity | Pa • s |
| $\eta_{eff}$ | effective viscosity at $\theta_{eff}$ | Pa • s |
| $\Lambda$ | bearing loading factor | - |
| $\nu$ | kinematic fluid viscosity | $m^2/s$ |
| $\theta_{amb}$ | ambient temperature | °C |
| $\theta_B$ | bearing temperature | °C |
| $\theta_{eff}$ | effective bearing temperature | °C |
| $\theta_{max}$ | maximum lubricant temperature | °C |
| $\rho$ | material density | $kg/m^3$ |
| $\sigma_y$ | material yield strength | Pa |

# Introduction to Lubricated Bronze Bearing Design

# I. INTRODUCTION

When two surfaces are in relative sliding motion, a normal load may be transmitted between them. Such a system is referred to as a sliding bearing. Under nominally dry conditions, this transmitted load is supported by small irregularities on the bearing surfaces, called asperities. These asperities are in direct contact. Under limited conditions of load and speed, such a bearing will work quite well and experience moderate wear. Materials such as high-leaded tin bronzes have self-lubricating properties that permit fairly high loads to be supported this way.

If a lubricant in the form of a fluid or grease is present between the sliding surfaces, a pressure may develop such that a significant portion of the load is transmitted by fluid pressure rather than metal to metal contact. When all the load is supported by fluid pressure, we speak of a full-film or hydrodynamic bearing. In a truly hydrodynamic bearing, the two surfaces are thus not in contact at all. Since fluids are easily sheared without damage, and generally have a lower shear resistance than most solids, relative motion between the two bearing surfaces takes place in the fluid film. This results in a lowering of friction and a reduction or elimination of wear. To achieve hydrodynamic or full-film lubrication in bearings requires special attention to several key variables that influence film formation. Most important among these are the geometry and motion of the bearing surfaces, the nature and consistency of the fluid employed, and the continual supply of this fluid.

The benefit of having the load supported by fluid pressure is very great indeed. It reduces wear and friction by several orders of magnitude. Unfortunately we cannot ensure that the correct conditions exist at all times. Contact between two bearing surfaces will invariably occur at some point during operation. When contact occurs, behavior of the bearing material is very important in the reliable operation of the bearing. Hence, besides the bearing geometry, bearing kinematics and lubricant, the bearing materials are of paramount importance. It is thus not difficult to see that the design of lubricated sliding bearings can be quite complex.

Complexity should not, however, be allowed to act as a deterrent to tackling a bearing design because the rewards for a good design are very rich indeed. Well designed bearings often have very long life expectancies and are very low cost from an operational and maintenance point of view. One of the first tilting-pad hydrodynamic bearings ever designed and put in service in this country is still operating today, after 80 years of almost continuous service. See Snow[1].

The intent of this publication is to describe the principles of lubricated bronze journal bearing design, including the many alloys now being successfully applied, and to introduce **Hydro** and **Bound,** CDA's bearing design software. Together, this text and the software can assist designers in choosing the optimum material and dimensions for anticipated service conditions.

## Historical

The problem of transporting heavy loads was significantly reduced by the invention of the wheel. The invention of the wheel was significant, but the bearing surface that supported the wheel must be regarded with equal importance. How and when the bearing and wheel were invented is not known, but it may be assumed that once invented, their use spread fairly rapidly. It is also fairly certain that soon it was realized that different types of wood gave different useful lives and that some materials reduced the amount of work required for hauling loads. It was probably also soon realized that smaller shaft bores gave lower friction but also had reduced life.

Today we would quickly understand these phenomena through the application of our vast knowledge base. In antiquity these bearing principles would have been determined through experimentation, just like the reduction in effort would have been noticed when wet conditions prevailed, or when animal greases were applied to the bearings. It was probably a combination of increased life and reduced effort that made the lubrication of wheel axles a regular practice.

Gradual improvements undoubtedly were made all the time to these bearing systems. Materials such as bronze and later, iron, would have increased life, but also increased the attention to be paid to greasing these bearings because of the relative preciousness of these materials. In addition to animal and vegetable fats, the use of pitch-tar, tree resins, and beeswax for bearing lubricants would have developed over time.

How and why bearings worked, and why the introduction of lubricants in them lowered the friction remained much of a mystery until the late 18th and into the 19th century. Most of the developments in bearings during this time were driven by the wheel bearing problems that occurred in the railways. The railways of the time were searching for ways to increase the load and speed capacity of cars and locomotives. The limit was always set by the problems encountered by bearing failures.

Matters were not helped along by the almost simultaneous introduction of plentiful, but somewhat deficient, mineral oils. Oiliness, the ability to prevent wear under metal-to-metal contact conditions, is almost absent in mineral oils, compared to the vegetable and animal oils and fats used up to that time. It was the development of full-film lubrication as we call it today, that gave the breakthrough into bearings capable of high loads and high speeds. Advances in both experimental techniques and mathematical analysis brought a lot of detail to light, fostering an understanding that still serves us quite well today.

The real breakthrough in experimental work, leading directly to the development of the hydrodynamic theory, should be credited to Beauchamp Tower. See Cameron[2]. Discussing his railroad journal bearing experiments, Tower indicated: "...interesting discovery when oil bath experiments were nearing completion. The bearing seized under increased load (625 psi), and the brass bearing was removed. While the bearing was removed, a 1/2-inch hole was drilled for a lubricator fitting through the cast iron bearing cap and the brass bearing. The bearing was reassembled and put in an oil bath lubrication. Oil was observed running out of the hole when the bearing was operated, making a mess. Attempts were made to plug the hole first with a cork and then with a wooden plug. Each one was forced out, indicating high pressure. A 200-psi pressure gage was screwed into the hole and the gage read the maximum gage capacity of 200 psi when the bearing was operated. The estimated bearing pressure was 100 psi."

This indicated to Tower that the shaft was floating on a film of oil whose pressure was more than double the mean bearing pressure. Beauchamp Tower's publications on his landmark bearing experiments stimulated Osborne Reynolds to develop the "physical wedge concept" for hydrodynamic lubrication in 1888.

Once the law of viscous flow was well known, and it was recognized that bearings developed their own pressurized lubricant films, the possibility of mathematical prediction of bearing behavior was realized. Petroff developed an equation in 1883 for estimating the power loss in a journal bearing, knowing the viscosity, diameter, speed and load. He assumed constant viscosity and a shaft centered in the bearing. Reynolds, in England, formulated the now famous Reynolds equation in 1886, and actually proceeded to solve it, using the results obtained by Tower in later experiments as an experimental verification of his methods. Around the beginning of the 20th century, Stribeck and Sommerfeld in Germany advanced the hydrodynamic lubrication theory toward engineering application. Stribeck established the minimum friction point and experimentally developed the "Stribeck Curve." Sommerfeld produced an analytical solution to the Reynolds equation for journal bearings, making possible the development of a Stribeck curve from theory.

## What Must Bearings Do?

Bearings are multifunctional devices. In order to operate efficiently and provide long service life, bearings often have to satisfy several requirements simultaneously. These include:

- Position and support a shaft or journal and permit motion with minimum energy consumption;
- Support a fixed load and be able to withstand occasional shock loads;
- Run quietly and suppress externally generated vibrations;
- Act as a guide to support reci procating or oscillating motion;
- Withstand temperature excursions;
- Accommodate some degree of shaft misalignment;

- Accommodate dirt particles trapped in the lubricant;
- Resist corrosion — under normal service conditions as well as during storage or extended down-time; and
- Provide easy maintenance.

As with all engineering endeavors, selection of the type of bearing and the bearing material usually involves some compromise among the often competing design requirements. Bronze bearing alloys cannot satisfy all needs at all times, but they do offer the broadest range of properties among today's sleeve bearing materials and can perform well under a very wide range of operating conditions.

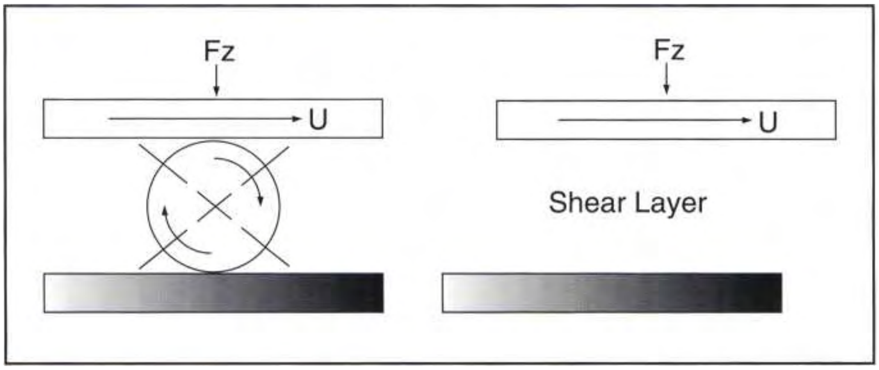


Figure 1.1. Load Carried by Rolling (l) and Sliding (r) Motion.

## Different Bearing Classes

When we speak of bearings, we mean a device that can transmit a load from one surface to another, when these surfaces experience a relative motion. This may be achieved by either some form of rolling motion or by some form of sliding action, see **Figure 1.1.**

Based on this fundamental difference we divide bearings into two broad classes as either: Rolling-Element Bearings (REB) or Sliding-Surface Bearings (SSB). These two classes are then further subdivided according to different forms of surface geometry and different types of surface interactions, see **Figure 1.2.**

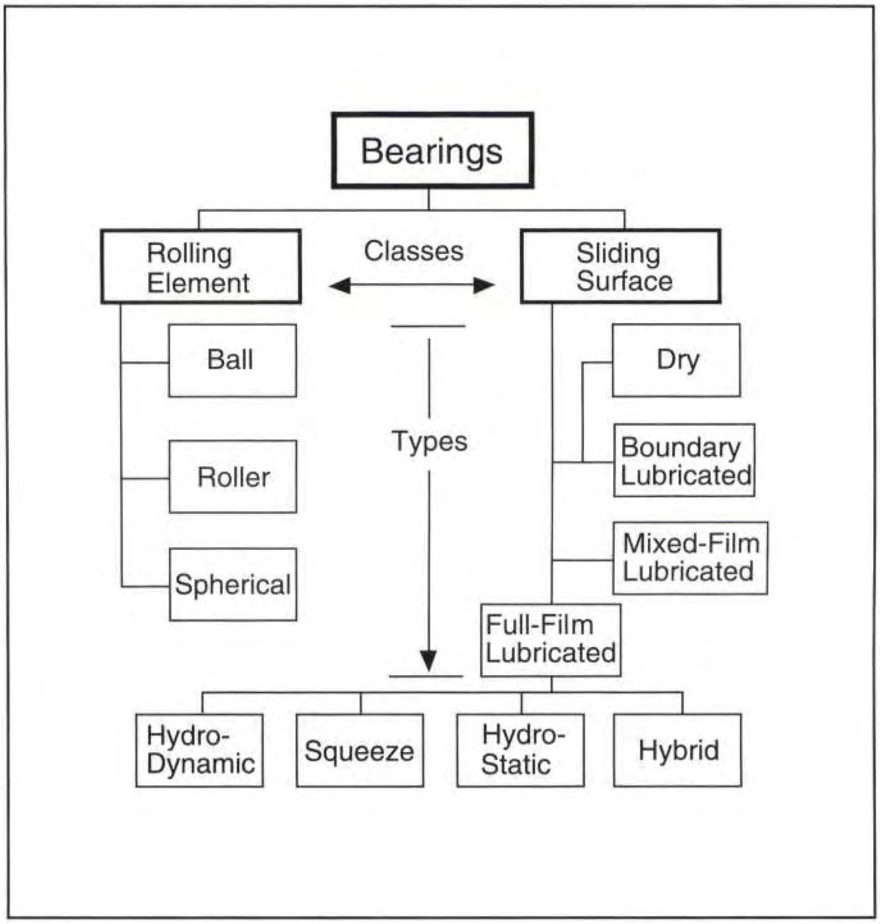


Figure 1.2. Bearing Classes and Different Types within Each Class.

## Differences Between Sliding and Rolling Bearings

There are important differences between the two classes of bearings that are worth noting. Many of these differences will favor the use of one type of bearing over the other.

As may be observed in **Table 1** on page 10, there are so many inherent advantages in sliding bearings that one should make every available effort to use them. Sliding bearings are used because they are simple, reliable, compact, durable, and because they can be custom-made economically for an almost limitless variety of applications—including and exceeding all those to which rolling-element bearings can lay claim.

| Sliding Bearings | Rolling Element Bearings |
|---|---|
| Need to be purposefully designed and manufactured. | Off-the-shelf finished bearings available. |
| Bearing, *per se,* usually less costly. Overall cost is less for small bearing systems. | Lower overall cost for larger bearings. Small bearings may have higher relative cost. |
| Service life not predictable, but can be capable of providing unlimited life. | Statistically predictable finite life. |
| Offer more design flexibility and versatility and broader spectrum of materials. | Limited by available standard sizes, configurations and materials. |
| More tolerant of abuse. May require periodic maintenance. | Very sensitive to abuse. Minimal maintenance, short of replacement. |
| Can provide high levels of beneficial damping. | Minimal damping characteristics. |
| Sometimes higher friction. Wear, under favorable conditions, can approach zero. | Finite, generally low friction and wear. |
| Little outside help available. | Lots of outside help available (suppliers). |

**Table 1. Differences between Sliding and Rolling Bearings.**

The use of rolling element bearings in every conceivable application is based more on the lack of suitable design skills in sliding bearing design, rather than an inherent superiority of rolling element bearings *per se.*

## Types of Sliding Bearings

Within the sliding bearing category there are various types of bearings broadly categorized as follows:

- Dry bearings and boundary lubricated bearings,
- Mixed-film bearings, and
- Full-film bearings.

### Dry Bearings and Boundary Lubricated Bearings

In the dry bearings, the lubricant is provided by solid particles contained within the bulk material. Besides lead, typical solid lubricants include graphite, molybdenum disulfide ($MoS_2$) and PTFE. A very successful line of dry bearings uses a sintered bronze material on a steel backing. The voids in the sintered material are then filled with a mix of lead and PTFE.

Boundary lubrication exits in liquid (or grease) lubricated bearings when there is no load carrying film and the lubricant serves mainly to keep friction fairly low. Cast bronze bearings are used extensively for boundary lubricated applications. We will return to this topic later.

### Mixed-Film Bearings

Oil-impregnated porous metal bearings are the most commonly used lubricated bearings where insufficient film exists to carry all the load. Only under very favorable conditions will the complete load be supported by the fluid film developed. These bearings contain voids within which a liquid lubricant can be maintained and provided in sufficient quantity for successful operation. Capillary action within the network of voids helps supply the surface with lubricant. Additional lubricant may be available to the bearing in the form of storage in felt reservoirs.

### Full-Film Bearings

In these types of bearings the lubricant is applied to the working surface from an external source. Lubricants typically used are oils or greases. They may also be other fluids such as water, process fluids or even gases. Full-film bearings fall into one of four categories:

- ***Hydrodynamic,*** where the shaft or bearing rotation generates a thin load carrying lubricant film,
- *Squeeze-film bearings,* where the normal motion of the surfaces generates a thin lubricant film ( connecting rods in internal combustion engines),
- ***Hydrostatic,*** where lubricant is supplied under very high pressure sufficient to separate the metal surfaces, and
- ***Hybrid bearings,*** where both hydrostatic and hydrodynamic features are used.

## Principal Bearing Geometries

Sliding bearings may be further categorized by the direction of load support that they provide. Broadly speaking this would be the journal bearing and the thrust bearing, see **Figure 1.3.**

The journal bearing, also known as a sleeve bearing, a bushing or a radial bearing, can support a radial load on a shaft. This radial load may act in any orientation, and may vary with time. Each of the different lubrication modes mentioned above is possible with a journal bearing. In its simplest form, a journal bearing consists of a shaft that fits through a clearance hole in a plate. This simple geometry is such that a full-film lubrication of the shaft and bearing may be possible under certain circumstances. The clearance space between the shaft and bearing form a natural wedge, and no complex machining techniques are needed to make a simple hydrodynamic journal bearing.

A thrust bearing is intended to support an axial thrust load on a shaft. It may consist of a simple flat washer of suitable material that rides against a thrust collar on the shaft. This simple form of thrust bearing will only be able to operate under boundary lubricating conditions because no natural wedge action takes place. Hence, similar to journal bearings, thrust bearings benefit greatly from the boundary lubrication characteristics of cast bronzes.

To obtain full-film lubrication with a thrust bearing requires surface features that need to be machined into the thrust washer surface. They may consist of simple grooves or dimples, or very sophisticated shapes such as wedges and spiral groove patterns. Because of the additional complexity in designing hydrodynamic thrust bearings, we will limit this discussion to journal bearing designs.

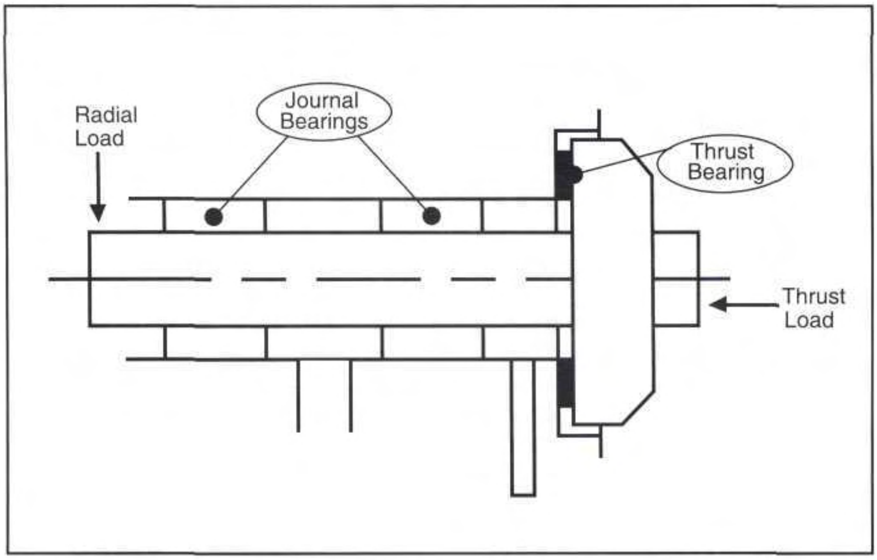


**Figure 1.3. Machine Member with Journal Bearings and a Thrust Bearing.**

## Bearing Operating Characteristics

Sliding bearings can operate in three fundamental modes, and it is important to understand that, for a given set of system requirements, their operating modes can be controlled by design. A good basis for the understanding of the fundamental phenomena in sliding bearings is the modified Stribeck diagram shown in **Figure 1.4.** This diagram presents three lubricated bearing performance parameters that are critical in slider bearing design. They are:

- ***Friction coefficient,*** a measure of the effort needed to move the surfaces relative to each other,
- ***Wear coefficient,*** a measure of the amount of material loss during the sliding and a determining factor in bearing durability, and
- ***Local bearing temperature,*** a measure of the likelihood of sudden failure that may be experienced in the system.

Variations of these design parameters are shown as a function of a so-called hydrodynamic variable that contains velocity, viscosity and load as important variables. In essence, the hydrodynamic variable determines how a load is supported in a sliding bearing.

Hence, the load transmitted between two bearing surfaces may be carried entirely by a continuous hydrodynamic lubricant film, entirely by the surface roughness peaks on the two opposing surfaces, or by a combination of the two. Thus, depending on the method by which the load is carried between two surfaces, we may define three lubrication and corresponding wear regimes as:

- ***Boundary lubrication.*** Large-scale asperity contact occurs. Load is carried by asperities and wear is moderate to heavy, depending on the nature of the surfaces and the materials involved.
- ***Mixed-film lubrication.*** *A* lubricant film is present, but intermittent contact occurs. Load is carried partially by fluid pressure and partially by asperity contact. Wear is moderate to light, depending on the chemical activity of the lubricant and bearing

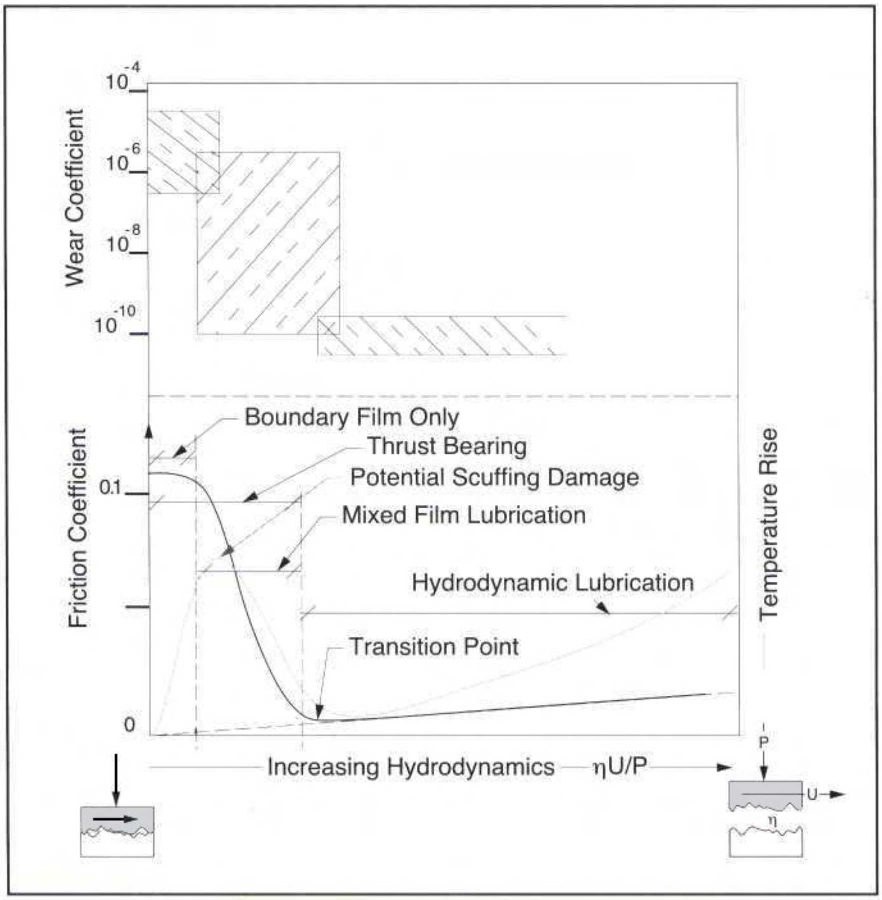


**Figure 1.4. Modified Stribeck Curve for Lubricated Sliding Surfaces.**

materials and on the contacting surfaces.

- ***Load carried by Hydrodynamic action only.*** When conditions of full hydrodynamics exist, the surfaces are completely separated by a lubricant film, and the load is carried by fluid pressure only. Wear in this regime is very low to non-existent.

Transition from boundary to hydrodynamic lubrication occurs when the mean film thickness separating the surfaces is greater than the local system roughness. Low roughness tends to promote early transition to hydrodynamics. It is always the desired goal in bearing design to work to the right of this transition. However several constraints may preclude operation in this domain and the bearing may have to operate for considerable periods of time in the boundary lubricated regime.

## Bearing Friction, Wear and Temperature

Examination of the behavior of bearing friction, wear and temperature as a function of the bearing operating regimes gives us some important insight into bearing materials selection and overall bearing design.

**Bearing Friction** (solid line in the lower half of **Figure 1.4.**) remains almost constant as we start to move from low to high hydrodynamics, and then experiences a rather sudden drop in the "mixed-film lubrication" region. The lowest friction occurs at the "transition point," and after this the friction rises slowly. The explanation for this friction behavior lies in the fact that friction at low hydrodynamics is almost completely governed by surface interaction. At high hydrodynamics, it is almost exclusively fluid friction (dashed line).

Asperity friction may be due to adhesive or abrasive contact. The friction level depends on the bearing material and its chemical interaction with the lubricant. Friction level is nearly velocity independent. To achieve low friction in the boundary lubricated region is thus a materials selection issue.

In the fluid friction region the nature of the bearing surfaces is not relevant because there is no asperity contact. The direct interaction properties of the bearing and shaft materials are not important here; however, the material must have sufficient plastic flow strength (indicated by hardness) and fatigue resistance to avoid gross dimensional changes due to high fluid pressures.

**Bearing Wear** (upper half of **Figure 1.4.**) decreases drastically as we move from low to high hydrodynamics, showing a rather sudden reduction in the "mixed film lubrication" region. This sudden drop occurs for the same reason that the friction drops, namely the reduction in asperity interaction due to load support by the lubricant film.

Bearing wear can occur due to several mechanisms. In sliding bearings, we typically encounter wear due to adhesion, two-body abrasion, or three-body abrasion. Adhesive wear occurs when interacting materials form solid bonds at the points of contact.

Sliding of the surfaces relative to each other causes the breakage of these bonds and results in removal of surface material. Two-body abrasion occurs when metal surfaces are in direct contact and where asperities from a hard surface penetrate into the other softer surface. Relative motion between the two now causes material removal of the softer surface. Both adhesive wear and two-body abrasive wear occur in sliding bearings when the surface separation is less than the mean operating roughness.

Adhesive bearing wear can be combated by selecting bearing materials with low solubility into each other. Materials such as indium, cadmium, lead, tin, silver and bismuth have low solubility with steel and are thus excellent choices as bearing materials. Note all the materials mentioned are very soft. It is also possible to add substances to a lubricant to contaminate the likely bonding sites.

Two-body abrasion can be minimized by selecting harder bearing materials or by reducing the surface roughness on the harder of the two surfaces. Increasing hardness of the bearing material may be detrimental to some other important aspect of the bearing.

Three-body abrasion is caused by hard particles in the lubricant. They may originate from sources outside the bearing such as dust and sand, or they may be generated within the bearing system itself due to adhesive or two-body abrasion.

There are several ways in which three-body abrasion can be combated. If the source of particles is external, we may employ lubricant filters and seals to reduce the concentration in the bearing. Third-body abrasion due to self-generated wear particles can be reduced by increasing the lubricant flushing action. We could also use harder, more wear resistant bearing materials. To avoid extensive bearing damage due to very large hard abrasive particles, an accommodation mechanism is incorporated by providing a fairly thick soft layer. Large, hard particles embed themselves into this layer. This however can lead to the two-body abrasion of the harder (normally shaft) material.

**Bearing Temperature** (dotted line in **Figure 1.4.),** which results from the product of friction and sliding velocity, is low at low sliding speed, reaches a peak, and then reduces due to a drop in friction. Further increases in speed produces a minimum temperature, and then it starts to rise again. The initial rise in temperature is almost exclusively the result of solid surface friction, while at high hydrodynamics it is completely due to fluid friction.

There are several important observations that can be made about the bearing temperature. The initial rapid rise is the result of high friction coupled with the velocity difference between the two surfaces. Remember, this is the boundary lubrication regime, and almost all the bearing load is carried by direct asperity contact. The friction in this regime is due to adhesive and abrasive asperity contact. It is well known that increasing temperature tends to promote adhesion between materials. Hence the propensity for asperities to weld together increases rapidly in the boundary lubricated region.

Depending on the materials, a sufficiently high temperature may be reached to cause scuffing damage. Scuffing damage may be catastrophic to a bearing operation in the form of galling and seizure.

Key factors that tend to keep the peak bearing temperatures low are low friction (low adhesion), early transition, good heat conduction and low, local normal stress. As far as material selection is concerned, we should select soft materials with low friction and high conductivity. Copper based bearing materials possess several of these key requirements.

From the foregoing discussion of lubrication regimes we should note that a very important consideration of the bearing design is the bearing material itself. This is where the various bronze materials are providing us with an excellent choice.

Readers interested in a more in-depth explanation of asperity interaction and the impact of materials may wish to consult Rabinowicz[3]. An excellent treatise on the different wear modes in bearings is given by Lansdown[4].

## Bearing Material Selection Criteria

Alloy selection is obviously very important in the design of bearings and especially when operating in the boundary lubricated regime. While the choice of alloy is less critical for full-film bearings, it does influence performance during the non-hydrodynamic conditions at start-up or upset operation brought about by shock loads, dirt intrusion, temperature surges and intermittent lubricant flow. Certainly, the more precisely a bearing's operating conditions can be defined, the simpler and more obvious will be the choice of bearing materials.

Selection of the specific bearing alloy requires a very careful consideration of the application and the characteristics of the material itself. There is no single best bearing material. The following are considered to be the attributes of a "good" bearing material:

- ***A low coefficient of friction against steel.*** The need for this property is self-evident, however good inherent frictional properties are particularly important in cases of unreliable lubrication, *i.e.,* boundary- or mixed-film conditions. In such cases, the metal itself must be relied upon to provide adequate antifriction behavior until lubrication is restored.
- ***High wear resistance.*** This property is different from, but related to strength, hardness, friction and a number of other properties. As with other properties, wear resistance is most important when lubrication momentarily fails. Wear resistance is also essential when bearings are run against hardened steel shafts.
- ***The ability to conform to non-uniformities in the shaft surface and to shaft misalignment.*** Low strength or relative softness are sometimes advantageous in a bearing alloy. Not all shafts are turned to precision tolerances or finishes, and service conditions can provoke misalignment; a softer alloy can tolerate such defects.
- ***Good corrosion resistance.*** Pump bearings are routinely exposed to sea water, sour mine waters, acids and any number of corrosive industrial process streams. All bearings occasionally get wet, and even some constituents of lubricants are or can become aggressive. It is therefore important that bearing materials exhibit sufficient corrosion resistance, both in service and during storage.
- ***High compressive strength combined, where needed, with good impact strength.*** Strength obviously determines the bearing's load-carrying capacity, while impact behavior frequently dictates the permissible types of service. Alloys with low impact strength must be adequately supported in suitable housings.
- ***Low shear strength.*** This property is characteristic of certain leaded alloys, whose ability to smear across the shaft surface provides a form of "internal lubrication." This is highly beneficial when normal lubrication is intermittent or unreliable.
- ***High creep strength.*** Some bearings must withstand elevated temperatures. In such cases, creep and stress-rupture properties have to be taken into account.
- ***High fatigue strength.*** Some bearings are subject to repeated pounding, others to pulsating or vibratory stresses. High fatigue strength generally accompanies high tensile strength, but there are exceptions to this rule.
- ***High thermal conductivity.*** Heat is damaging to most bearings and lubricants, therefore the ability to conduct frictional heat away from the bearing surface results in lower operating temperatures.
- ***A uniform metallurgical structure.*** Sound, non-porous surfaces are needed to promote and maintain a stable lubricant film.
- ***Compatible thermal expansion.*** The thermal expansion of the bearing must be known in order to establish proper clearances at the bearing's operating temperature.

Some of these attributes are obviously more important than others, and only a few may be critical in particular installations. All, however, influence bearing performance to some degree, and none should be ignored.

## Cast Bronze Bearing Materials

Literally hundreds of different bearing alloys are available for use with steel mating surfaces. Along with sintered and wrought bronzes, aluminum and zinc alloys and a variety of polymers and composites, these materials make up many of the sliding bearings in use today. For additional information, the reader is referred to an excellent detailed review by Pratt[5] and a more general description by Glaeser[6].

Many bearing materials offer promising individual properties (light weight, low friction) to meet particular service requirements, but the cast bearing bronzes offer the broadest range of applicability. Their favorable combinations of mechanical and physical properties allow the designer to optimize a bearing design without compromising needed characteristics unnecessarily. **Table 1.2.** from CDA[7] compares many of the tribologically important criteria for a range of commonly used materials.

| Bearing Alloy Families | Load Capacity and Fatigue Resistance | Maximum Operating Temperature | Conformability and Embeddability | Resistance to Seizure | Hardness and Wear Resistance |
|---|---|---|---|---|---|
| **Leaded Red Brasses** | Moderate | Moderate | Low | Low | Good |
| **Tin Bronzes** | High | High | Moderate | Moderate | High |
| **High-Leaded Tin Bronzes** | Moderate / High | High | Good | Good | High |
| **Aluminum Bronzes** | Very High | Very High | Poor | Moderate | Very High |
| **High-Strength Brasses** | Moderate | Moderate | Poor | Moderate | High |
| **Copper Beryllium Alloys** | High | High | Poor | Poor | Very High |
| **Leaded Coppers** | Moderate | High | Very Good | Very Good | Moderate |
| **Tin-Based Babbitts** | Moderate | Moderate | Excellent | Excellent | Low |
| **Lead-Based Babbitts** | Moderate | Moderate | Excellent | Excellent | Low |
| **Aluminum Low-Tin or Lead Alloys** | High | High | Good | Low | Moderate |
| **Aluminum High-Tin or Lead Alloys** | Moderate | High | Good | Moderate | Moderate |

**Table 1.2. Comparison of Important Bearing Selection Criteria for Commonly Used Materials.**

Note the broad range of bearing bronzes to choose from, and the superior load and wear performance of the copper based materials. These latter two properties become extremely important when less than ideal lubrication conditions exist. **Chapter 2** contains descriptions of the various bearing bronzes in greater detail.

## Other Benefits of Bearing Bronzes

Besides the superior strength and wear properties of bearing bronzes, there are additional benefits when using these materials. Some of these are:

- bearing bronze materials are readily available,
- all grades can be produced in almost any size in large or small quantities,
- normal machine shop facilities are adequate for preparation,
- substantial structural strength permits the use of bearing bronzes as an integral part of the structure. This inherent strength also permits their use without backing,
- no need to rely upon specialist manufacturers and suppliers, and manufacture can usually be in-house,
- easy to fine tune bearings. If a characteristic of a specific bearing is found deficient, a review can be made of the important material properties needed and a selection made from available materials to provide improved properties. The large selection of cast copper-based bearing alloys provides a wide choice of engineering properties, so that incremental adjustments can be made.

# Properties and Selection of Cast Bearing Bronzes

# SELECTING CAST BEARING BRONZES

A very important aspect of sliding bearing design is the selection of the actual bearing materials. Literally hundreds of different bearing alloys are available for use with steel mating surfaces. Selection of a specific bearing alloy requires very careful consideration of the application and the material characteristics themselves. There is no single best bearing material. Technically, an assessment must be made of component and system requirements to allow placing a priority on the material properties for a given application.

## Bronzes and Copper Alloys

Many bearing materials offer promising individual properties (light weight, low friction) to meet particular service requirements, but the cast bearing bronzes offer the broadest range of applicability. Their favorable combinations of mechanical and physical properties allow the designer to optimize a bearing design without compromising needed characteristics unnecessarily. The large selection of cast copper-based bearing alloys provides a wide choice of engineering properties. **Table 2.1,** compiled from data presented in CDA[8], gives the composition of the most important cast bearing bronzes in use today.

## Economics and Availability

As with all engineering, economic factors play heavily in the selection of materials. Therefore, the selection of a material also requires a review of the economic factors. Overall costs are frequently the deciding factor in selection of a bearing material. Some materials cost more than others. Bearing costs are, to a certain extent, subjective. The economic assessment should also include factors such as availability, ease of manufacture, replacement cost, the degree of precision required in its manufacture and installation, spare parts cost, and ability of making incremental adjustments to the bearing.

While bronze bearing alloys are certainly not the least expensive engineering materials, they are easily competitive in terms of performance and life-cycle costs. Finally, and not incidentally, bronze sleeve bearings offer the distinct advantage that virtually all grades can be produced in almost any size in large or small quantities.

Cast bronze bearing alloys are available in a large variety of standard and custom shapes and sizes. Bearing blanks are cast as cylindrical shapes using the one of several techniques available. They may be sand cast, chill cast, continuous cast or centrifugally cast. The rate of cooling of the molten alloy influences the grain size of the solidified material, and slow cooling usually gives a coarser structure with reduced mechanical properties. Chill casting can be used instead of sand casting to get improved mechanical properties.

A further variation is centrifugal casting in which the mold is rotated during casting. Any impurities which are present are usually less dense and are therefore separated towards the center of the mold where they can be removed, and as a consequence a cleaner alloy is produced. With centrifugal casting methods, however, segregation of high lead content alloys can occur. Continuously cast alloys are also of high quality, with properties comparable to alloys cast by centrifugal casting.

The cast bearing bronzes are available in a great variety of sizes and shapes. Particularly suitable for the manufacture of bearings are the following:

**Tin and leaded tin bronzes:**

- As finished bearings in sizes ranging from 3 mm to 152 mm ($^5/_{16}$ in to 6 in ) in diameter, up to 230 mm (9 in) long;
- As solid bars ranging from 10 mm to 260 mm ($^3/_8$ in to 10 in) in diameter, up to 2700 mm (105 in) long; and,
- As cored bars 25 mm to 300 mm (1 in to 12 in) in diameter, in various bore sizes, also up to 2700 mm (105 in) long.

**Aluminum bronzes:**

- As solid bars 13mm to 200 mm (0.5 in to 8.0 in) in diameter, from 3,000 mm to 3,700 mm (120 in to 144 in) long;
- As cored bars 30 mm to 260 mm (1.25 in to 10 in) in diameter, in

| Designation UNS | SAE | Cu | Sn | Pb | Zn | Fe | Ni | P | Al | Other |
|---|---|---|---|---|---|---|---|---|---|---|
| | | Nominal Composition, Wt% | | | | | | | | |
| **Leaded Red Brass** | | | | | | | | | | |
| C83600 | 40 | 85 | 5 | 5 | 5 | – | – | – | – | – |
| **Tin Bronzes** | | | | | | | | | | |
| C90300 | 620 | 88 | 8 | – | 4 | – | – | – | – | – |
| C90500 | 62 | 88 | 10 | - | 2 | - | - | – | - | - |
| C90700 | 65 | 88 | 11 | - | - | - | - | – | - | – |
| **Leaded Tin Bronzes** | | | | | | | | | | |
| C92200 | 622 | 88 | 6 | 1.5 | 4.5 | – | – | – | – | – |
| C92300 | - | 87.3 | 8 | 0.7 | 4 | - | - | – | - | - |
| C92700 | 63 | 88 | 10 | 2 | - | - | - | – | - | - |
| **High-Leaded Tin Bronzes** | | | | | | | | | | |
| C93200 | 660 | 83 | 7 | 7 | 3 | – | – | – | – | – |
| C93400 | - | 84 | 8 | 8 | - | - | - | – | - | - |
| C93500 | 66 | 85 | 5 | 9 | 1 | - | - | – | - | - |
| C93600 | M-64 | 80 | 7 | 12 | 1 | - | - | – | – | - |
| C93700 | 64 | 80 | 10 | 10 | - | - | - | – | - | - |
| C93800 | 67 | 78 | 7 | 15 | - | - | - | – | - | - |
| C94100 | - | 73 | 5.5 | 20 | 1 | - | - | – | - | - |
| C94300 | (AMS-4840) | 68 | 5 | 25 | .8 | - | 1 | – | - | - |
| C94500 | - | 72 | 7 | 19 | 1.2 | - | 1 | – | - | - |
| **High-Strength Brasses** | | | | | | | | | | |
| C86300 | 430B | 63 | – | – | 25 | 3 | – | – | 6 | 3 Mn |
| C86400 | – | 59 | - | 1 | 40 | - | - | – | - | - |
| **Aluminum Bronzes** | | | | | | | | | | |
| C95300 | 68b | 89 | – | – | – | 1 | – | – | 10 | – |
| C95300HT | - | 89 | - | - | - | 1 | - | – | 10 | - |
| C95400 | - | 85 | - | - | - | 4 | - | – | 11 | - |
| C95400 HT | - | 85 | - | - | - | 4 | - | – | 11 | - |
| C95500 | - | 81 | - | - | - | 4 | 4 | – | 11 | - |
| C95500 HT | - | 81 | - | - | | 4 | 4 | – | 11 | - |
| C95520 | - | 75 | - | - | - | 5 | 5 | – | 11 | 3 Mn |
| C95800 | | 79 | - | - | – | 4 | 4.5 | – | 9 | 1 Mn |
| **Silicon Brasses** | | | | | | | | | | |
| C87600 | – | 90 | – | – | 5.5 | – | – | – | – | 4.5 Si |
| C87900 | - | 65 | - | - | 34 | - | - | – | - | 1 Si |
| **Copper Beryllium Alloys** | | | | | | | | | | |
| **C82800** | – | 96.6 | - | - | - | - | - | – | - | 2.6Be/ .5Co/ .25Si |
| **Leaded Copper** | | | | | | | | | | |
| C98820 | 484 | Rem | – | 42 | – | – | – | – | – | – |
| **Lead-Free Bronze** | | | | | | | | | | |
| C89320 | - | 89 | 6 | - | - | - | - | – | - | 5 Bi |

**Table 2.1. Nominal Compositions of the Most Common Cast Bearing Bronzes.**

various bore sizes, up to 3,700 mm (144 in) long; and,

- As rectangles, 6 mm x 26 mm (0.25 in x 1.0 in) to 75 mm (3 in) square, up to 3,700 mm (144 in) long.

Custom castings are available in most bearing materials. They range in size from miniatures only 6 mm (0.25 in) in diameter to large rings up to 3 m (10 ft) across.

## Detailed Discussion of Specific Alloys

The broad classification of cast bearing bronzes in **Table 2.1** are along composition lines. Different levels of alloying elements impart characteristics to the individual materials that may be used in the materials selection. The following is a fairly detailed discussion of the specific alloys.

### Tin Bronzes

Tin increases the strength of copper alloys. Unless it is present in high concentrations, it has only a small effect on thermal conductivity, compared with most other alloying elements. Copper-tin bearing alloys, traditionally called tin bronzes, also exhibit low friction coefficients against steel. The combination of these properties causes the surface temperatures of the tin bronzes to tend to remain lower than other bearing alloys. Since heat adversely affects all bearings, this is seen as a distinct advantage.

All of the alloys in this series contain more than 5% tin and contain the hard intermetallic compound $Cu_{31}Sn_8$ in their microstructure. The more tin (C90300 < C90500 < C90700), the greater the proportion of $Cu_{31}Sn_8$ will be present. The hard intermetallic compound imparts high wear resistance, but it also causes the alloys to be somewhat abrasive, and as a result, the tin bronzes should be used against hardened steel shafts (minimum hardness 300-400 HB). The addition of zinc to tin bronzes displaces some of the tin and increases the presence of a strong delta phase. Zinc is therefore added as a strengthening agent, as in alloys C90300 and C90500.

The tin bronzes are hard and strong, and have good corrosion resistance, especially against seawater. They are wear resistant and withstand pounding well. The tin bronzes can be classified as being only moderately machinable. They can be turned, bored and reamed, but are difficult to broach.

The alloys work well with grease lubricants. They can function as boundary-type bearings due to their ability to form polar compounds with small traces of lubricants, thus stabilizing (holding on to) extremely thin organic layers and thereby reducing the chance for metal-to-metal contact. Lacking lead, however, the tin bronzes require adequate and reliable lubrication.

### Leaded Tin Bronzes

This group of tin bronzes contains between one and two percent lead, chiefly to improve their machinability. Their lead content is too small to materially affect their bearing properties, and they can be thought of as free-cutting versions of the tin bronzes described above. It therefore follows that they are used in applications similar to their lead-free counterparts and are especially advantageous where high-speed volume production and/or complex shapes demand a relatively large amount of machining.

### High-Leaded Tin Bronzes

This class of alloys consists of the most commonly used bearing materials. They rank somewhat lower than the unleaded or leaded copper-tin alloys described above in load-carrying capacity, however they adequately satisfy requirements for bearings operating under moderate loads and medium-to-high speeds—the bulk of bearing applications.

Lead has very low solubility in liquid copper and is almost entirely insoluble in solid copper or its alloys. Further, lead melts at a relatively low temperature (327 C, 621 F), and when bearings are cast, the lead in the alloy freezes last, forming irregular globules between boundaries of the copper-tin alloy. The lead content in these alloys, nominally between 7% and 15%, is sufficiently high to improve their bearing properties. The islands of free lead tend to smear over the bearing and shaft surfaces acting, in effect, like a built-in lubricant; this can prevent seizing in the event of an interruption in the lubricant supply. These alloys are therefore especially recommended for boundary-lubricated and mixed-film applications. The alloys can be run against unhardened shafts. Like the lower-leaded bronzes described above, these alloys are free-cutting.

Since lead is not in solid solution, its effect on thermal conductivity is much lower than its high proportion of the microstructure would suggest. High-leaded tin bronzes therefore retain much of the favorable thermal conductivity of their unleaded counterparts. Lead does, however, reduce the alloys' strength and ductility, therefore alloys with very high lead contents (C93800 and C94300 at nominally 15% and 25% lead, respectively) should be used at lower operating stresses and in situations in which there is little chance of impact or shock loading.

In a boundary-type bearing, one rule of thumb advises that designers choose the softest (lowest strength) alloy able to support the applied load. The advantage offered by this method of alloy selection lies in the fact that soft alloys such as high-leaded tin bronzes are better able to conform to shaft misalignment or deflection, and can accommodate (embed) dirt particles carried in the lubricant stream, be it oil or grease. Some designers follow this rule to the point of lengthening the bearing to reduce the applied stress, thereby enabling the use of softer materials.

Alloy C93200 (SAE Alloy 660) is generally considered to be the workhorse alloy of this series. Originally developed as a low-tin composition during World War II, the alloy remains in wide use today. Major areas of application include light to moderate duty general purpose bearings.

Alloy C93600 can be thought of as a modified version of Alloy C93200; it contains about one-third as much zinc and almost twice as much lead as the SAE 660 alloy. As a result of the alloy's high lead content, its machinability and anti-seizing properties are considerably improved. Reduced zinc content improves corrosion resistance somewhat. High hardness and strength impart the alloy with good pounding resistance.

Alloy C93700 exhibits good corrosion resistance against sulfuric acid (in limited concentrations), acid mine waters, mineral waters and paper-mill sulfite liquors. It has excellent wear resistance at high speed and heavy pressure and, among alloys in this family, tolerates shock and vibration well.

Alloy C93800 can be characterized as having fair strength, good corrosion resistance to sulfuric acid and sour mine waters. It has fair wear resistance but excellent antifriction properties. It is non-seizing, readily machinable and can be used where lubrication is doubtful.

Alloy C94100 has moderate load carrying capacity but, like the other members of this family, it exhibits excellent antifriction properties. It is especially good for use under boundary and mixed-film conditions.

## High-Strength Brasses

The high-strength brasses (sometimes improperly referred to as manganese bronzes) are modifications of the familiar 60% Cu-40% Zn yellow brass known as Muntz Metal. Alloy C86300 contains manganese, aluminum and iron, which raise its tensile strength to well over 800 MPa (115 ksi). Alloy C86400 is somewhat lower in strength, but it contains one percent lead to improve its machinability. Despite their high tensile strength, the alloys' fatigue resistance can only be rated as moderate. These high-tensile brasses can operate under very high loads and at moderately high speeds; however, they require hardened, well aligned shafts and reliable lubrication. Being relatively hard, they have little capacity to embed trapped particles and therefore do not tolerate dirty lubricants.

## Aluminum Bronzes

Aluminum bronzes are the highest-strength standard copper-based bearing alloys. Aluminum is a potent strengthener in these alloys; in general, the higher the alloys' aluminum content, the higher their strength. Alloys with aluminum contents higher than about 9.5% can be heat treated, and this is noted in the designation "HT" following the alloys' UNS numbers. Alloy C95500HT, for example, can attain tensile strengths in excess of 800 MPa (115 ksi).

One interesting feature of the aluminum bronzes is their high elevated temperature strength. The compressive strength of C95400 at 260 C (500 F) is the same as that of tin bronzes at room temperature. The aluminum bronzes have the highest fatigue strength of all bronze bearing materials, and they can resist repeated and severe impact loads very well. One important disadvantage is their relatively poor machinability. They can be turned, bored and reamed, but like the tin bronzes, they are difficult to broach.

Like the high tensile brasses, most of the aluminum bronzes must be used against steel shafts hardened to greater than 500 HB. Alloy C95200 is somewhat more forgiving in this respect. As a class, aluminum bronzes require reliable full-film lubrication to prevent metal-to-metal contact and possible scoring. Both bearing and shaft should be machined to surface finishes finer than 0.4-0.5 μm (15-20 μinch) rms. The alloys do not embed dirt well, and generally require good shaft alignment. Their thermal conductivity is only slightly lower than some of the lower strength alloys, and as a result, they can be used at moderate speeds if shaft hardness, shaft alignment and lubrication are all well controlled.

## Silicon Brasses

Silicon imparts good castability to copper-zinc alloys, and also adds significantly to the alloys' strength. Silicon brasses are not among the most common bearing alloys, but they do have good bearing characteristics at moderately high speeds. They are quite readily machinable, considering they contain no lead and are higher in strength than standard tin bearing alloys. Silicon brasses require reliable, clean lubrication (dirt embedding ability in these relatively hard alloys is only rated as fair). The alloys should be used against hardened shafts.

## Copper Beryllium Alloys

Copper beryllium alloys can be heat treated to attain higher strengths than any other copper alloy. Heat treatment is fairly straightforward: solution annealing at 780-800 C (1450-1500 F) is followed by water quenching and aging at 340 C (650 F). This heat treatment, in alloy C82800, produces tensile strengths at or higher than 1,100 MPa (165 ksi). Other copper beryllium alloys have similar or slightly lower strengths.

Copper beryllium alloys have good bearing properties. When fully hardened, they withstand extremely high stresses. Heat treatment can be adjusted to produce a range of mechanical properties, and the alloys can be used in less demanding applications. Copper beryllium alloys are, however, quite expensive and their use is therefore usually reserved for applications in which their strength can be fully exploited.

The bearings require very hard shafts, machined to close tolerances and very precise alignment. They require reliable lubrication. Dirt embedding properties are poor. Although very strong, the alloys have fair to poor impact resistance, therefore bearings should be supported as required by suitable housings.

## Lead-Free Bearing Bronze

Bismuth may be substituted for lead in the cast bronze bearing materials. Lead-free bronzes are formulated to mimic the properties of the leaded alloys but without the safety concerns of lead. Friction, yield and tensile strength and corrosion resistance are all comparable to leaded bearing bronzes. Alloy C89320 is thus very similar in performance to C93200.

## Effect of Other Alloying Elements

The compositions of the bearing alloys described above have been standardized and can be ordered according to their UNS designation or applicable standard specifications. As in most cast alloys, allowable compositions are given in ranges to accommodate foundry practice. Except in the case of copper beryllium alloys and aluminum bronzes, minor deviations from set compositions have only minor effects, if any, on mechanical property.

Modifications to standard compositions are always possible, of course. These may be called for to accommodate special or severe service conditions. For example, manganese, aluminum and phosphorus can be added in small amounts to enhance the properties imparted by one or more of the major alloying elements; tin, zinc, and in some cases, silicon. Manganese increases strength but moderately decreases ductility. It does, however improve an alloy's high-temperature working properties. Aluminum, a potent strengthener in its own right, tends to raise the strength of copper-zinc and copper-tin alloys when added in small amounts. Phosphorus is a deoxidizer and potent strengthener. It also improves fluidity during casting. When used to excess, phosphorus can give rise to localized hard spots, a condition which obviously should be avoided.

# Summary of Cast Bronze Characteristics and Usage

A distillation of the major bearing characteristics and usage of the different cast bronze materials is shown in **Table 2.2.** This table may be used for an initial scan for potential materials or when searching for an improved material.

# Selection Procedure for Bearing Materials

The selection of a suitable bearing material for a given application is a rather difficult task because, unlike the theoretical calculations used for film thickness determinations, a large number of variables are involved. In practice though, we seldom design a completely new bearing system. Often there is a similar design, or an extension of a given design that we can glean some information from for the start of a material selection.

| Alloy | Characteristics and Uses |
|---|---|
| **Leaded Red Brass** | Reasonable strength, excellent thermal conductivity, reasonable corrosion resistance to sea-water and brine, and good machining and casting properties. Lead content ensures pressure tightness. Leaded Red Brass is used as a low-cost bearing material when low loads and low speeds are encountered. Requires good, reliable lubrication and a moderately hard shaft. |
| **Tin Bronzes** | Hard, strong alloys with good corrosion resistance, especially against seawater. Moderately machinable. As bearing materials, they are wear resistant and resist pounding well. Best for high loads, low speeds. Require good, reliable lubrication and a moderately hard shaft. Higher tin concentrations improve strength, but at the expense of conformability and embeddability. |
| **Leaded Tin Bronzes** | Lead improves machinability in these tin bronzes but does not materially affect mechanical properties. The alloys are essentially free-cutting versions of the tin bronzes, above, and have similar properties and uses. |
| **High-Leaded Tin Bronzes** | Most commonly used bearing alloys, found in bearings operating at moderate loads and moderate-to-high speeds. Excellent for boundary-lubricated situations or where lubrication is uncertain. Since lead is insoluble in the solid phases, it is dispersed in the matrix as small isolated globules, acting as small reservoirs of lubricant. Increasing lead concentrations increase conformability and scoring resistance, but at the expense of reduced strength and pounding resistance.<br>Alloy C93200 is considered the workhorse alloy of the series. Alloy C93600 has improved machining and anti-seizing properties. C93800 is noted for its good corrosion resistance against concentrations of sulfuric acid below 78%. Alloy C94100 is especially good under boundary-lubricated conditions. |
| **High-Strength Brasses** | Alloys with high mechanical strength, good corrosion resistance and favorable castability, and of low relative cost. Can be machined but, with the exception of C86400 and C86700, are less readily machined than leaded compositions.<br>Brasses have typically poor tribological properties, and hence as a bearing material tend to be used in non-critical applications. When used for high-strength bearings, alloys C86300 and C86400 require hardened shafts and reliable lubricant supplies. |
| **Aluminum Bronzes** | The aluminum bronzes are characterized by high strength and excellent corrosion resistance. Some can be heat treated ("HT"). Physical properties remain good at elevated temperatures.<br>Excellent heavy duty bearing alloys with very good abrasion resistance and excellent resistance to repeated, severe impacts.<br>Poor anti-seizure properties, relatively poor conformability and embeddability, hence require good reliable lubrication, hard shafts and proper shaft alignment, with both shaft and bearing machined to fine surface finishes. |
| **Silicon Brasses** | Moderate-to-high strength alloys with good corrosion resistance and favorable casting properties. Used for mechanical products and pump components where combination of strength and corrosion resistance is important. |
| **Copper Beryllium Alloys** | Relatively high-strength materials with good electrical and thermal conductivity. Used where bearings and bushings with a good combination of strength and conductivity are needed. Excellent heavy duty bearing alloys, but do not tolerate misalignment or dirty lubricants and generally should be used against hardened steel shafts, with both shaft and bearing machined to fine surface finish. |
| **Leaded Copper** | Ultrahigh-lead alloys for special purpose bearings. Alloys have relatively low strength and poor impact properties and generally require reinforcement. Excellent for boundary-lubricated situations, or where lubrication is uncertain. |
| **Lead-Free Bronzes** | Formulated to mimic the properties of the leaded alloys but without the safety concerns of lead. Friction, yield and tensile strength and corrosion resistance are all comparable to leaded bearing bronzes. |

Table 2.2. Summary of Cast Bronze Characteristics.

## Initial Selection of Candidate Materials Based on Past Usage

To know what has been used for a specific application is a very powerful way to start with a given material selection. Even if the application is not quite the same as what you are working on, it will most likely form a good starting point. **Table 2.3** is a compilation of different bearing applications using cast bronze alloys. See Rippel[9]. It is suggested that you select several materials from this list; do not be too restrictive in the type of application. Again, this should not be taken as the last word on materials selection, but rather the starting point. Knowledge of how well previous designs fared can be an invaluable help in such cases.

Another way to make an initial materials selection is by considering that a leaded-tin bronze such as C93200 is the most commonly used material. We may then examine this choice in light of some other considerations given next.

## Additional Materials Selection Considerations

After we have made some initial materials selection we can apply some additional criteria. For all the different choices of bearing materials there are two critical conditions that must be met. These are:

- the material must be strong enough to withstand the peak applied bearing stresses, and
- it must have sufficient temperature capacity.

| | |
|---|---|
| agricultural machinery | C94100 |
| air compressor wrist pin bushings | C90300 |
| aircraft accessory drives | C90700 |
| aircraft control bushings | C93700 |
| aircraft landing gear bushings | C90300 |
| aircraft-carburetor bearings | C94300 |
| automotive spindle bushings | C92700 |
| automotive transmission thrust washers | C83600 |
| backs for lined bearings | C93800, C93700, C93600 |
| bearings for earth moving machines | C92700 |
| boring bar guide bushings | Al Bronzes |
| brass and copper rolling-mill neck bearings | C93800 |
| bridge bearings | C90700, C90300 |
| cam bearings | Al bronzes |
| cam bushings for diesel engines | C93700, C93200, C93600 |
| cam bushings for farm equipment | C93700 |
| cam bushings for mechanical devices | C93700 |
| clutch pilot bushings | C90500 |
| connecting-rod bushings for farm equipment | C93700, C92700 |
| cornpicker snapping roll bushings | C83600 |
| cranes and hoist bearings | C94100 |
| crankshaft bushings for farm equipment | C93700 |
| crankshaft main bearings | C93600, C93700 |
| deep-well pump-line shaft bearings | C83600, C89320 |
| deep-well pump-bowl bushings | C93800, C93600 |
| drum bushings for cranes | C93800 |
| drum bushings on earthmoving machinery | C93800 |
| earthmoving equipment bushings | Al bronzes |
| electric-motor bushings | C93700, C93200, C94100, C93600 |
| elevators | C94100 |
| fire-pump bushings | C94300 |
| freight and streetcar bearings | C93800 |
| fuel and water-pump bushings | C93700 |
| garden tractors | C93700 |
| gas, gasoline and diesel engine bearings | C93800 |
| gasoline pump bearings | C93800 |
| gear bushings for farm equipment reducers | C93700, C92700 |
| gear bushings for motorcycles | C93700 |
| generator and distributor bushings | C93200 |
| guide bushings for piston rods | C93200, C89320, C93600 |
| guide bushings for rams | C93200, C89320, C93600 |
| guide bushings for valves | C93200, C89320, C93600 |
| guide post bushings | Al bronzes |
| heavy earthmoving equipment bushings | Al bronzes |

**Table 2.3. Candidate Bearing Bronzes Based on Past Usage.** *(cont. p. 24)*

High levels of applied bearing stress may cause plastic flow of the bearing material. This should be avoided because it will otherwise cause severe dimensional changes. Similarly, high operating temperatures will cause a reduction in the bearing strength and higher potential for wear and seizing.

To obtain exact values of maximum recommended stress and temperatures for the different alloys is very difficult and depends a lot on the nature of the bearing design. Typical values may be found in the literature of, for example: Rippel[9], CDA[7], and DIN[10]. Table 2.4 lists typical maximum bearing pressures and operating temperatures that can be used with careful bearing design practice. The range of temperatures and pressures within each alloy class results from the various levels of soft phase present. Materials with higher soft phase content (more lead) tend towards the lower temperature and pressure levels.

These numbers should be used as guidelines only. Also, we typically select a material that is just strong enough for the job. Allowance should be made for possible overload conditions.

Other tribological characteristics to consider are the propensity to seizure and the resistance to wear. These are indicated in **Table 1.1** on a relative basis. This table should be consulted for further refinement of the materials selection. The relative wear resistance of a bearing material may also be judged from the identation hardness. It is well known that hardness plays a major role in preventing wear. It should be realized however that increasing hardness levels will

| | |
|---|---|
| hydraulic glands seals | C93800, C89320 |
| hydraulic pump bushings | C94300 |
| king pin bushings for off-highway equipment | C93800 |
| lathe bearings | C93700 |
| linkage bushings for machine tools and presses | C90500, C93700 |
| locomotive bearing parts | C94300, C92700, C93600 |
| low-pressure valve bearings | C83600, C89320 |
| machine tool bearings | C94100,C90700, Al bronzes, C90300 |
| main bearings for presses | C93200 |
| main bearings for refrigeration compressors | C94300 |
| manifold bushings for earthmoving machinery | C83600 |
| marine equipment | C90700 |
| mechanical linkage bushings for farm and material handling equipment | C93200, C93600, C89320 |
| mechanical linkage for farm equipment and packaging machinery | C92700 |
| motorcycle-engine bearings | C93200 |
| outboard motor crankshaft bearings | C93700 |
| piston pin bushings | C90500, C93600, C93700 (diesels) |
| piston pins for packaging equipment | C93700 |
| power lawnmower bushings | C93700 |
| power shovel bushings | Al bronzes |
| propeller bushings | C83600 |
| pump sleeves | C83600, C89320 |
| rail and heavy equipment trunnion bearings | C93700 |
| railroad car wheel bearings | C93700, C93800, C94300 |
| reduction-gear pinion bearings | C94100 |

**Table 2.3.** ***(cont.)*** **Candidate Bearing Bronzes Based on Past Usage.** ***(cont. p. 25)***

| Bearing Alloy | Max. Recommended Operating Temp. °C, | °F | Range of Max. Recommended Operating Pressure, MPa | psi |
|---|---|---|---|---|
| Leaded Red Brass | 180-230 | 356-446 | 20-30 | 2900-4350 |
| Tin Bronze and Leaded Tin Bronze | 170 | 338 | 25-35 | 3625-5075 |
| High-Leaded Tin Bronzes | 170 | 338 | 15-25 | 2175-3625 |
| Aluminum Bronze | 300 | 572 | 50-70 | 7250-10155 |
| High Strength Brass | 200 | 392 | 30-50 | 4350-7250 |
| Copper Beryllium Alloys | >200 | >392 | 50-200 | 7250-29010 |
| Leaded Copper | 160 | 320 | 10-15 | 1450-2175 |

**Table 2.4. Maximum Recommended Operating Temperatures and Pressures.**

| | |
|---|---|
| rod bushings | C93800, C93600 |
| rod bushings for refrigeration compressors | C94300 |
| roll bushings | C90500 |
| roll neck bearings in rolling mills | C93700, Al bronzes |
| roller bushings for conveyors | C93200 |
| rolling mill bearings | C90500, C93800, C93700 |
| seals | C93800, C89320 |
| sleeve bushings for cranes and draglines | C93200, C93600 |
| spacer bushings and bearings for pumps. | C93800, C89320, C93600 |
| speed reducers bearings | C93700 |
| spindle bushings for farm equipment and trucks | C93700, C93200 |
| spindles and connecting rods in farm equipment | C93700 |
| spring bushings for farm and automotive equipment | C83600 |
| spring-shackle bushings | C93200 |
| starter motor bushings | C93200 |
| steel mill equipment | C90700, C93600 |
| steering-knuckle bushings | C93700 |
| supercharger bushings | C94100 |
| textile machinery bushings | C94100 |
| thrust washers and components for chemical process equipment | C93600 |
| torque-tube bushings | C93200 |
| track-roller bushings for crawler tractors | C93700, C93200 |
| trolley wheel bushings | C90300 |
| trunnion bearings | C90300, C92700, C93700 |
| turbocharger spindle bushings (floating type) | C94100,C89320 |
| turntable bushings | Al bronzes |
| valve guides | C90500 |
| valve rocker arm bushings | C93700 |
| valve stem nuts | Al bronzes |
| water-lubricated bushings | C94300 |
| wristpin bushings | C93700, C93600, C89320 |

**Table 2.3.** ***(cont.)*** **Candidate Bearing Bronzes Based on Past Usage.**

reduce the resistance to seizure, and also reduce the conformability and embeddability. Typical hardness values are indicated in **Table 2.4.** The values are indicated as Brinnell hardness. This is a hardness test using a fairly large indenter, and thus, averages the hardness reading over a fairly large domain. The indicated hardness values do not necessarily reflect the hardest possible phase in the alloy.

## Physical Properties of Bearing Materials

Other properties such as thermal conductivity, thermal expansion coefficients and density are also important tribological parameters, especially so when high bearing temperatures are expected. **Table 2.5** list values for the most commonly used physical properties.

This data is from DIN[10]; CDA[7]; and Rippel[9]. The values listed in this table should only be used as a guideline. More specific information should be obtained once a more definitive material selection has been made. A range of values suggests that slightly different results were reported by the references. The range for the hardness values may also reflect the different heat treatment conditions for the alloy.

## Compatibility with Working Fluids

While for the most part we design sliding bearings using grease or oil as the lubricant, we do not need to restrict ourselves to these lubricants. In fact the use of sliding bearings can be turned to enormous advantage when using a given process fluid as the lubricant. The use of seals, always a weak link in the design, can then be avoided altogether. Typical process fluids such as water, gasoline, kerosene, creosote, paints, and even food products such as ketchup and molasses are used as lubricants. Even highly corrosive fluids such as hydrochloric and muratic acids can be used as liquid lubricants for sliding bearings.

It is also important that the bearing and shafting materials to

| UNS | Name/ISO Designation | Hardness H(HB) | Yield Strength $\sigma_y$ (0.2%) | | Elastic Modulus | | Therm. Exp. $\alpha$ | | Therm. Cond k | | Density $\rho$ | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | MPa | ksi | GPa | ksi | µm/m•°K | µin/in•°F | W/m•°K | Btu/ft²/h/°F | gm/cm³ | lb/in³ |
| **Leaded Red Brass** | | | | | | | | | | | | |
| C83600 | CuSn5Pb5Zn5 | 55-60 | 80-100 | 11.6-14.5 | 95 | 14 | 19 | 11 | 72 | 42 | 8.90 | 0.322 |
| **Tin Bronzes** | | | | | | | | | | | | |
| C90300 | | 70 | 145 | 21.0 | 96 | 14 | 18 | 10 | 75 | 43 | 8.80 | 0.318 |
| C90500 | | 75 | 150 | 21.8 | 103 | 15 | 20 | 11 | 75 | 43 | 8.72 | 0.315 |
| C90700 | | 80 | 150 | 21.8 | 103 | 15 | 18 | 10 | 71 | 41 | 8.77 | 0.317 |
| **Leaded Tin Bronzes** | | | | | | | | | | | | |
| C92200 | | 65 | 130 | 18.9 | 96 | 14 | 18 | 10 | 70 | 40 | 8.64 | 0.312 |
| C92300 | | 70 | 140 | 20.3 | 96 | 14 | 18 | 10 | 75 | 43 | 8.77 | 0.317 |
| C92700 | | 77 | 145 | 21.0 | 110 | 16 | 18 | 10 | 47 | 27 | 8.78 | 0.317 |
| **High-Leaded Tin Bronzes** | | | | | | | | | | | | |
| C93200 | CuSn7Pb7Zn3 | 65-70 | 100-120 | 14.5-17.4 | 100 | 14.5 | 18 | 10 | 59 | 34 | 8.80 | 0.318 |
| C93400 | | 60 | 110 | 16.0 | 76 | 11 | 18 | 10 | 58 | 34 | 8.87 | 0.320 |
| C93500 | | 60 | 110 | 16.0 | 100 | 14.5 | 18 | 10 | 70 | 40 | 8.87 | 0.320 |
| C93600 | | 65-70 | 135(0.5%) | 19.6 | 77 | 11 | 18.5 | 10 | 49 | 28 | 9.05 | 0.327 |
| C93700 | CuPb10Sn10 | 60-70 | 80-110 | 11.6-14.5 | 76-90 | 11-13 | 18 | 10 | 47 | 27 | 9.00 | 0.325 |
| C93800 | | 55 | 110-140 | 11.6-20.3 | 72 | 10 | 18.5 | 10 | 52 | 30 | 9.25 | 0.334 |
| C94300 | AMS 4840 | 48 | 105 | 15.2 | 72 | 10 | 18 | 10 | 63 | 36 | 9.30 | 0.336 |
| C94100 approx. | CuPb20Sn5 | 45-50 | 60-80 | 8.7-11.6 | 75 | 10.9 | 19 | 11 | 59 | 34 | 9.30 | 0.336 |
| **High-Strength Brasses** | | | | | | | | | | | | |
| C86300 | | 215 | 400-450 | 58.0-65.3 | 98 | 14 | 22 | 12 | 35 | 20 | 7.83 | 0.283 |
| C86400 | | 90 | 170 | 24.7 | 96 | 14 | 20 | 11 | 28 | 16 | 8.33 | 0.301 |
| C86800 approx. | CuZn37Mn2A12Si | 150-170 | 280-350 | 40.6-50.8 | 100 | 14.5 | 19 | 11 | 65 | 38 | 8.10 | 0.293 |
| **Aluminum Bronzes** | | | | | | | | | | | | |
| C95300 | | 140-170 | — | — | 110 | 16 | 16 | 9 | 63 | 36 | 7.53 | 0.272 |
| C95400 | | 170-195 | — | — | 107 | 16 | 16 | 9 | 59 | 34 | 7.45 | 0.269 |
| C95500 | | 195-230 | — | — | 110 | 16 | 16 | 9 | 42 | 24 | 7.53 | 0.272 |
| C95800 | | 160 | 240(0.5%) | 34.8 | 114 | 17 | 16 | 9 | 36 | 21 | 7.64 | 0.276 |
| C95520 approx. | CuA110Fe5Ni5 | 140-150 | 250-280 | 36.3-40.6 | 120 | 17 | 16 | 9 | 60 | 38 | 7.60 | 0.275 |
| C95800 approx. | CuA19Fe4Ni4 | 180-220 | 480-530 | 69.6-76.9 | 118 | 17 | 16 | 9 | 27 | 16 | 7.60 | 0.275 |
| **Silicon Brasses** | | | | | | | | | | | | |
| C87200 | | 80 | 150-200 | 21.8-29.0 | 100 | 14.5 | 17 | 9 | 28 | 16 | 8.36 | 0.302 |
| C87600 | | 135 | 220 | 31.9 | — | — | — | — | 28 | 16 | 8.30 | 0.300 |
| **Copper Beryllium Alloys** | | | | | | | | | | | | |
| C82800 | | 180-380 | 500 | 72.5 | 130 | 18.9 | 22 | 12 | 123 | 71 | 8.30 | 0.300 |
| **Leaded Copper** | | | | | | | | | | | | |
| C98820 | | 30-45 | 65 | 9.4 | 75 | 10.9 | 16 | 9 | 80 | 46 | — | — |
| **Lead-Free Bronze** | | | | | | | | | | | | |
| C89320 | | 70 | 120(0.5%) | 17.4 | 98 | 14.2 | 18 | 10 | 56 | 32 | 8.80 | 0.318 |

**Table 2.5. Physical Properties of Bearing Materials.**

be used are compatible with the process fluid. Extensive corrosion resistance ratings for various cast bronzes as a function of different process media is given in CDA's **Copper Casting Alloys**[8].

**NOTES:**

# Design of Boundary Lubricated Bronze Bearings

# DESIGN OF BOUNDARY LUBRICATED BRONZE BEARINGS

Boundary lubrication occurs with sliding interfaces where the transmitted load is carried completely by the contacting asperities. While a liquid lubricant may be present, an insignificant portion of the load is supported by it. Wear occurs with boundary lubrication. The wear rate is dependent upon the design parameters (materials, load, sliding distance) and the ability of the lubricant to form protective, sacrificial chemical compounds to avoid direct metal-to-metal contact. When significant chemical action takes place, the role of temperature is very important. System temperatures may result from some ambient condition, or they may be self-generated due to sliding under friction conditions. The level of friction will of course depend on the nature of the asperity boundary interaction and the films that may form here due to the chemical kinetics.

The whole aspect of boundary lubrication is thus a complex phenomena of chemical, thermal and mechanical interactions. It is very hard to predict what may happen for a given situation, but we can say a few things about what brings on the boundary lubrication mode.

The accompanying program, **Bound,** is based on the experimental data and the modeling presented in this chapter.

## Conditions Leading to Boundary Lubrication

While the nature of the interactions that take place between asperities during sliding conditions are impossible to predict (as of today), there are several macro parameters that force a lubricated tribological system to operate in the asperity or boundary mode of load bearing. These conditions are;

- High load
- Low speed
- Low lubricant viscosity
- Starting and stopping
- Rough surfaces
- Irregular surfaces
- Lack of suitable or adequate film formation mechanisms
- Inadequate clearance
- Misalignment

The above list indicates that almost every tribo-system will operate at one point or another in the lubricated wear regime. Therefore, correct design considerations and criteria must be kept in mind in the design stage of every system.

## Boundary Lubricated Wear Data for Cast Bronze Bearings

As mentioned under the materials selection of the various cast bronzes, the superior boundary lubrication aspects of copper based alloys are second to none. These characteristics are even further enhanced when lubricants such as greases are introduced into the interface. Friction and wear now decrease dramatically, and very long lives may be obtained from grease-lubricated bronze bearings.

To determine the boundary lubrication performance of several commonly used cast bronze bearing alloys, the International Copper Research Association sponsored extensive research that produced invaluable data for use in the design of such bearings. This research also furthered the understanding of the underlying phenomena that set the limits of operation. Glaeser *et al*[11], conducted an extensive set of lubricated journal bearing wear experiments on the following bearing bronzes;

C93200 Leaded Bronze
C90500 Tin Bronze
C95400 Aluminum Bronze
C94500 High-Leaded Bronze.

It was found that the bearing performance could be presented on a plot of sliding velocity versus the mean bearing stress, or a so-called lubricated wear map, **Figure 3.1.**

This lubricated wear map for a journal bearing shows three areas of wear identified as follows:

Region of Moderate Wear, ($K = 3 x 10^{-7}$)
Region of High Wear, ($K = 1 x 10^{-6}$)
Region of High Temperature.

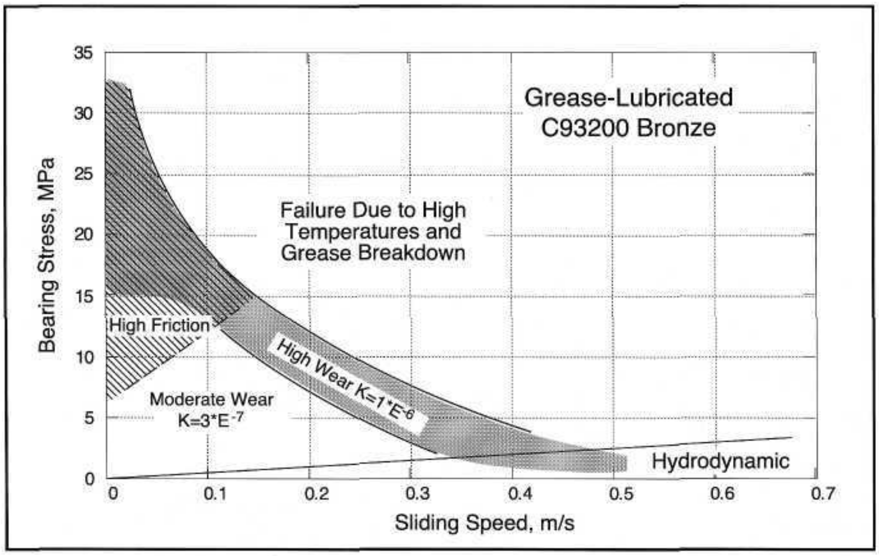


**Figure 3.1. Lubricated Wear Map for a Journal Bearing.**

The high-wear limit occurs at a temperature of about 150 C (302 F). At temperatures above this, severe wear develops and failure takes place. This temperature limit will depend on the nature of the materials and lubricant and is thus very system specific.

The experimental results show that wear and bearing distress becomes unacceptable at bearing stress levels from 24-28 MPa (3500-4000 psi). In many grease-lubricated continuous rotation bearing installations, 3.5 MPa (500 psi) is accepted as a maximum. Therefore, boundary lubricated bearings are seldom operated at bearing stress levels much above 28 MPa (4000 psi). Some high-strength bronzes, such as aluminum bronze, are operated at bearing pressures on the order of 70 MPa (10,000 psi) in oscillating motion applications (airframe bearings, for instance).

## Lubricated Wear Limits for Bearing Bronzes

The maximum load-speed limit at which moderate lubricated wear might be expected to take place is critical in the design of bearings. This limit was determined by Glaeser[11] and is summarized in **Figure 3.2** for the four alloys. Unlike the original publication by Glaeser[11], which averaged data from four separate tests to obtain the wear transition limit, here the lowest pressure limit at which a transition occurred for any one speed is taken as the upper stress limit. Original comments by Glaeser on the behavior of the materials follow:

*Alloy C95400 (aluminum bronze)* shows the most consistent and highest load-capacity results over the speed range studied. At speeds of 0.10 m/s (0.33 ft/s) and 0.15 m/s (0.49 ft/s), the maximum allowable operating stress is sharply limited because of excessive temperatures from frictional heating. Up to the maximum allowable stresses at these speeds, the coefficient of friction is typically 0.01 or less. Exceeding the maximum results in an abrupt increase in the coefficient of friction to values around 0.1. This, coupled with the higher applied load, causes a thermal runaway with rapidly increasing temperatures. Such behavior is indicative of at least partial hydrodynamic films separating the sliding surfaces at the lower stresses. However, electrical continuity measurements always showed metal-to-metal contact or the absence of a complete lubricant film. At the lower speeds of 0.025 m/s (0.08 ft/s) and 0.05 m/s (0.16 ft/s), the coefficient of friction is typically 0.1 at all bearing stresses. The maximum allowable stress was determined by frictional heating, but no abrupt transition to runaway temperatures occurred.

*Alloy C90500 (tin bronze)* shows similar behavior to Alloy C95400, but has a distinctly lower load capacity at 0.10 m/s (0.33 ft/s). The reduction in load capacity is directly related to the stress at which the transition from low friction coefficients (0.01) to high coefficients (0.1) occurs.

*Alloy C93200 (leaded bronze)* is similar to alloy C90500 at speeds of 0.10 m/s (0.33 ft/s) and 0.15 m/s (0.49 ft/s), but shows the lowest load capacity of all four alloys at 0.025 m/s (0.08 ft/s) and 0.05 m/s (0.16 ft/s). Reduced load capacity at the lower speeds results from higher coefficients of friction of up to 0.18, which cause a correspondingly higher thermal input from friction.

*Alloy C94500 (high-leaded bronze)* shows the most inconsistent performance of the four alloys, which results from variability in the transition from low friction to high friction as load is increased. This alloy also produces low coefficients of friction (0.01 to 0.02) at the lower speeds as well. As a result, the transition from acceptable to unacceptable stress levels was fairly sharply defined at 0.05 m/s (0.16 ft/s).

## Lubricated Wear Rates for Bearing Bronzes

The wear rates of four bearing alloys were measured by Glaeser at 7 MPa bearing pressure and 0.10 m/s sliding speed. The results are plotted in **Figure 3.3** as the average of three bearings for each material. The four alloys are clearly divided in their extent of wear under identical running conditions.

Alloy C95400 was the most wear resistant, followed closely by alloy C90500. Alloy C93200 had an intermediate wear resistance, while alloy C94500 experienced a large amount of wear. All four alloys showed the classic high initial wear rate ("break-in") followed by a leveling off to a more constant, lower value.

Predictions of the amount of diametral wear encountered over a given time period for the different materials may be made by fitting the data to a wear model. For the lubricated wear data, an equation that follows the typical Archard-Holm[12] works well, see **Appendix A** for further details on this model. The predicted amount of diametral wear is given by:

$$d = K\frac{P\,U\,t_s}{H}$$

where;

- d = diametral wear
- K = wear coefficient
- P = nominal bearing stress $F_n/B\bullet D$
- $F_n$ = applied radial load
- D, B = bearing diameter and width
- H = hardness
- $t_s$ = total sliding time
- U = sliding velocity

To account for the high wear rate in the initial break-in, two different wear coefficients are needed for this model. These are incorporated in the **Bound** program.

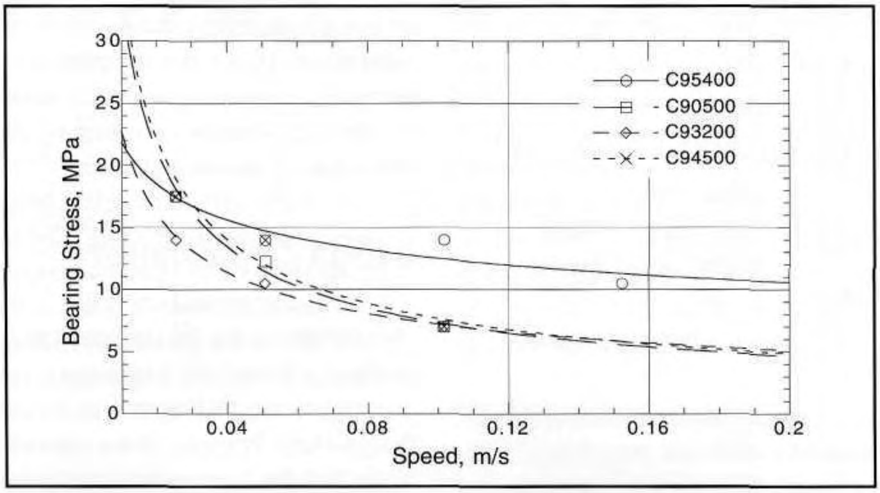


**Figure 3.2. Load-Speed Limits for the Different Bronze Materials.**

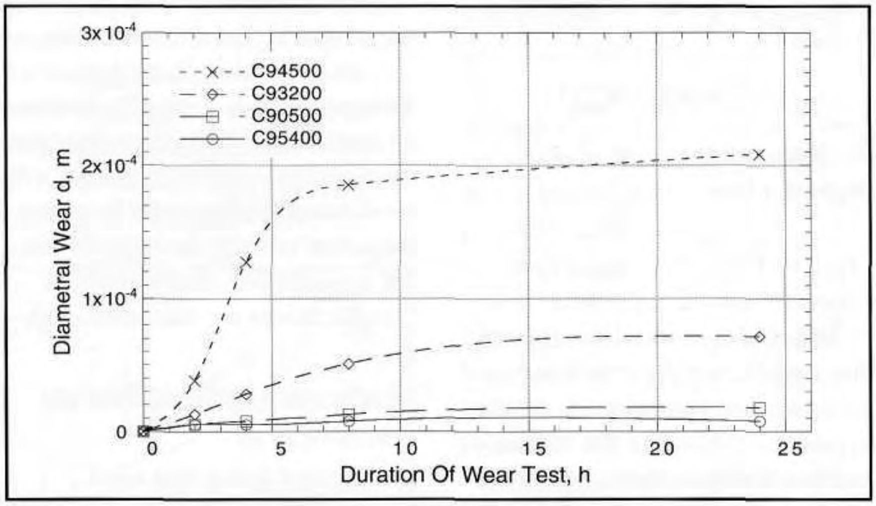


**Figure 3.3. Lubricated Wear Behavior for Four Bronze Materials.**

## Bearing Temperatures

The energy loss, W, in the bearing may be calculated from the product of frictional force and sliding velocity, *viz:*

$$W = fF_nU$$

The coefficient of friction, f, here should be chosen carefully. It depends very much on the lubrication conditions. For the grease-lubricated bearings we will assume that the friction coefficient is 0.1. This will be an overestimate in most cases, but it will give us a conservative estimate for the bearing temperature, and it is in keeping with the experimental findings.

The dissipated energy will give rise to a bearing temperature increase. For boundary lubricated bearings, the heat generated is removed by air cooling of some form. This may be natural convection or, in some cases, forced convection. It is a very complex mechanism that involves many difficult to estimate variables. To make the model workable from an engineering point of view, we simplify it to the following:

$$P_{ta} = k_a A_B(\theta_B - \theta_{amb})$$

**where:**

$P_{ta}$ = heat flow to ambient
$k_a$ = heat transfer coefficient to ambient
$A_B$ = outer surface area of bearing housing
$\theta_B$ = bearing temperature
$\theta_{amb}$ = ambient temperature

Typical values for the heat transfer coefficient are calculated by a method detailed in **Appendix B.** By equating the heat generated to that convected away to the environment, we get a heat balance as follows:

$$fFU = k_a A_B(\theta_B - \theta_{amb})$$

Hence at the point of equilibrium we obtain:

$$\theta_B = \theta_{amb} + \frac{fFU}{k_a A_B}$$

Note, this expression suggests that constant temperature lines on the lubricated wear map should be hyperbolas, assuming the friction and heat transfer coefficients stay constant. The limit of operation region where high wear takes place has the general shape of a hyperbola. This limit on the bearing temperature was found to be about 150 C (302 F) for the four materials evaluated.

## Bearing PV Factors

Examination of the wear equation and the bearing temperature equation shows that both use the product of pressure and velocity. This is sometimes referred to as the "PV" factor of a bearing. Some design manuals use this as a general rule for boundary lubricated bearing design. As we see from the above two expressions, this could either reflect a temperature limit, or a wear limit. Also, strict reciprocity between pressure and speed cannot be used, as is apparent from the bearing temperature equation.

## Design Considerations

There are several critical considerations for the design of a successful boundary lubricated bronze bearing. Adherence to these design considerations make it more likely that the bearing performance will be satisfactory.

Bearing diameter is usually dictated by shaft stiffness considerations and is thus not something we can truly select at random. Bearing width is normally chosen so as to keep the width-to-diameter ratio close to unity, though this will be dictated somewhat by the mean projected bearing stress levels for the application. Other important considerations are discussed next.

## Maximum Allowable Bearing Temperatures

Glaeser found that rapid increases in bearing wear occur when the bearing temperatures exceed about 150 C (302 F). This is often accompanied by sudden increases in friction coefficients from low numbers like 0.01-0.03 to high values like 0.10-0.15. We should thus calculate the theoretical temperature rise based on the higher friction coefficient.

The 150 C (302 F) temperature limit is well below the capability of the various bronzes used in the investigation. Glaeser suggests it is more indicative of the grease limit, rather than the bearing materials *per se.* Accepting the fact that the experimental limit is in fact more a grease limit, we may make some recommendations for the maximum safe temperatures in a bearing from the known temperature limits for different types of greases, see **Table 3.2.**

Besides the type of filler used for the grease, the base lubricant will also significantly affect performance. Overall however, the temperatures shown in **Table 3.2** may be used as guidelines.

Additional information on the selection and application of greases may be found in NLGI[13].

## Maximum Allowable Bearing Stress

The maximum allowable bearing stress depends on the type of material used for the bearing and the form of lubrication used. For grease-lubricated bearings operating in the boundary lubrication mode, the maximum bearing stress is normally kept below 3.5-5 MPa (508-725 psi). From the test data by Glaeser[11], it is clear that some of the materials are capable of much higher stress levels. However, it is recommended to use only these higher levels under special design circumstances when space is critical.

| Grease Type | Maximum Temperature, °C | °F | Comments |
|---|---|---|---|
| Lithium soap based | 140 | 284 | General purpose |
| Calcium soap | 60 | 140 | Relatively cheap |
| Clay-based | 160-200 | 320-392 | Wide temperature range |

**Table 3.1. Recommendation for Grease/Bearing Temperature Limits.**

Normally the projected bearing area is selected such that the recommended stress levels are not exceeded.

## Clearance

Clearance values for boundary lubricated sleeve bearings should be larger than for hydrodynamic lubrication. This is because the bearing has to operate in the presence of wear debris and any other adventitious debris that would normally be flushed out by a flowing lubrication system. In addition, sufficient clearance space is needed to allow injection of fresh grease into the bearing periodically.

Recommended clearance values range from 0.2% to 0.5% of the journal diameter. Sufficient clearance should also be allowed for possible reduction in clearance due to different thermal expansion coefficients. **Figure 3.4** shows the recommended diametral bearing clearances as suggested by Bartz[14].

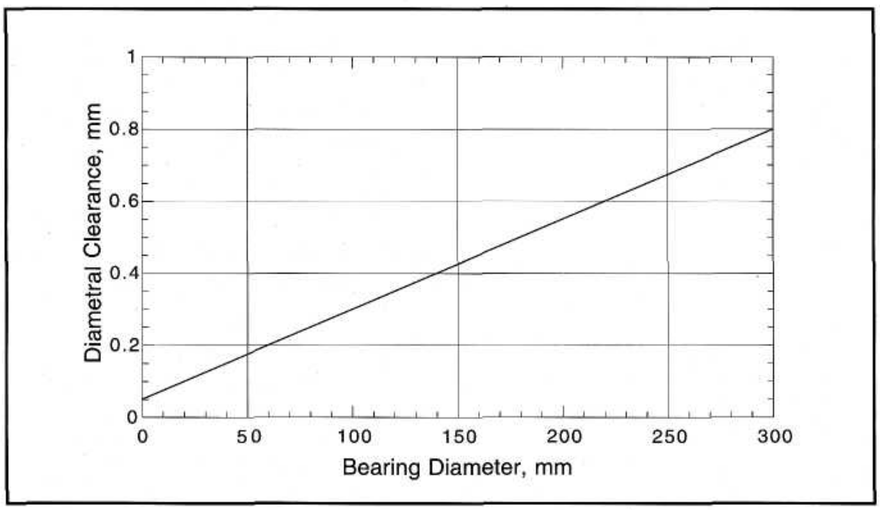


**Figure 3.4. Recommended Clearance for Grease-Lubricated Bearings.**

## Shafting

The hardness and surface finish of steel shafting is an important consideration in the design of boundary lubricated bearings. The shaft plays a major role in the wear-in process and subsequent performance is directly influenced by wear-in. For the leaded bronzes, cold-rolled steel, finished to 0.5-0.7 µm (20-28 µin.) Ra (arithmetical average of roughness), can be used. However, steel shafting, AISI 1045, hardened to HRC 30 minimum, ground and polished (or comparable steel), is a better choice and certainly should be used with tin bronzes containing no lead.

Harder shafting is appropriate as long as it has the fracture toughness and fatigue strength required for the application. When using harder shafting, it is important to keep the surface roughness levels low to prevent abrasive wear of the bearing by the shaft.

## Surface Finish

In boundary lubricated bronze bearings, the shaft surface finish is important. Since the shaft should always be harder than the bearing material, steel asperities will tend to plow the bronze bearing surface, producing an abrasive type wear. The original surface finish of the bearing material will change during the "wear-in" period early in the life of the bearing. Some wear of the shaft will also occur owing to the hard intermetallics in the bronze microstructure. A shaft surface finish of between 0.25-0.7 µm (10-28 µin.) Ra is recommended.

## Wall Thickness and Flange Dimensions

Bronze sleeve bearings are normally contained within a housing or shell. A reasonable 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. 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 difficult to maintain. Varying clearances within the bearing can result.

To locate a bearing axially in a housing bore, a shoulder flange as shown in **Figure 3.5** may be provided. This can be used as a marginal thrust surface as well.

When a bearing is pressed or shrunk in its 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 is that wall thickness should be at least three times the depth of any grooving.

Solid bushing minimum and maximum wall thicknesses and recommended flange sizes according to ISO-4379 are shown in **Figure 3.6.** For flanged bushings, it is recommended that the flange height, $h_{flange}$, is at least equal to the bushing wall thickness, $t_{wall}$. Also the flange thickness, $t_{flange}$, is made the same as the wall thickness. Maximum wall thickness and flange dimensions may be desirable for high load applications where a lot of interference fit is required. This will reduce the possibility of buckling.

## Methods of Retaining Bearings

Many different techniques are used to ensure that a bronze bearing stays put within its housing. The method used depends upon the particular application, but it is important to ensure that the unit lends 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

The most 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 125 mm (4.92 in.) bore size. Stock bushings are commonly provided with slightly over nominal OD sizes. Since these tolerances are built into standard bushings, the amount of press fit is controlled by housing bore. Recommended fits for general applications with cast bronze bushings in cast iron or steel housings are H8/p7 according to ISO-1101. For aluminum housings, the interference may have to be increased by as much as 0.001 D to provide satisfactory tightness.

For high-temperature work, the recommended interference fits may need to be adjusted to allow for expansion differences between bearing and housing and to avoid yielding of the bearing material.

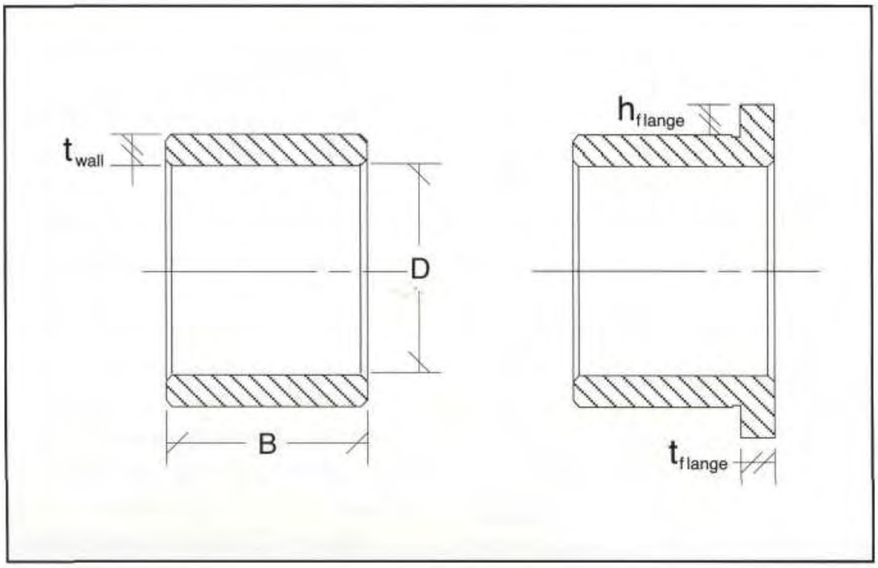


**Figure 3.5. Dimensions for Straight and Flanged Bushings.**

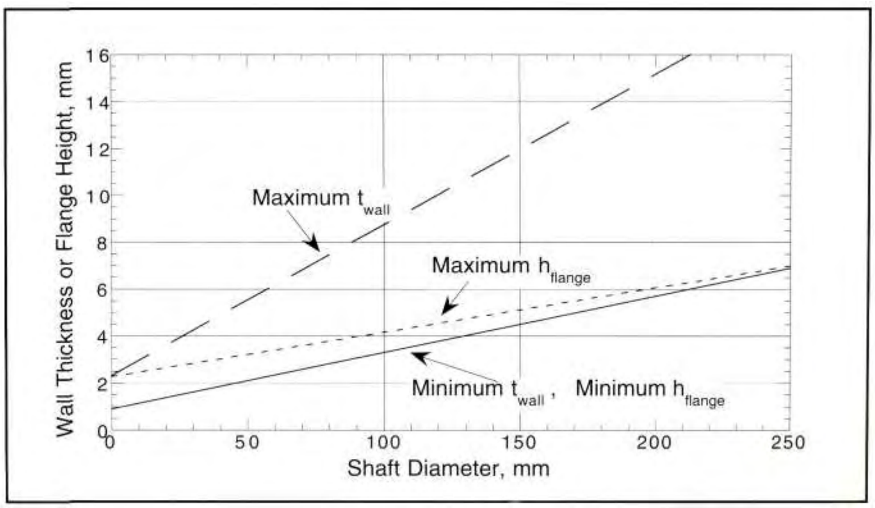


**Figure 3.6. Recommended Wall Thickness and Flange Dimensions for Solid Cast Bronze Bushings.**

Thin walled housings 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 105% 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 nitrogen. These methods are easier than heating the housing and are preferred. Dry ice in alcohol has a temperature of -78 C (-108 F) and liquid nitrogen boils at -196 C (-321 F). The use of a hollow mandrill filled with liquid nitrogen will greatly extend the useful working time to install bushings into machine bores. Lubricants should not be used when employing shrink-fits as this may lead to hydraulic blister formation between the bearing and the housing bore.

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. A steady pressing force is preferred. An insertion arbor should always be used. 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 are:

- Set screws
- Woodruff keys
- Bolted bearing flanges
- Threaded bearing OD
- Dowel pins
- Housing caps

*These methods may be easier than a press or shrink fit, but they are much less satisfactory as they do not result in a very solid connection.* They should only be used for light duty applications and certainly not when dynamic loads are applied to the bearing.

## Grease Grooving Patterns

Because grease will only go where it is pushed or dragged, it will not distribute itself in a bearing like a liquid lubricant will. Therefore, grooving is generally cut into the inner surface of a bushing. There are many grooving designs used. Some of the more common designs are shown in **Figure 3.7,** where the bushing has been unfolded along an imaginary line opposite the grease feed hole. Most of these configurations are easily cut with special tools.

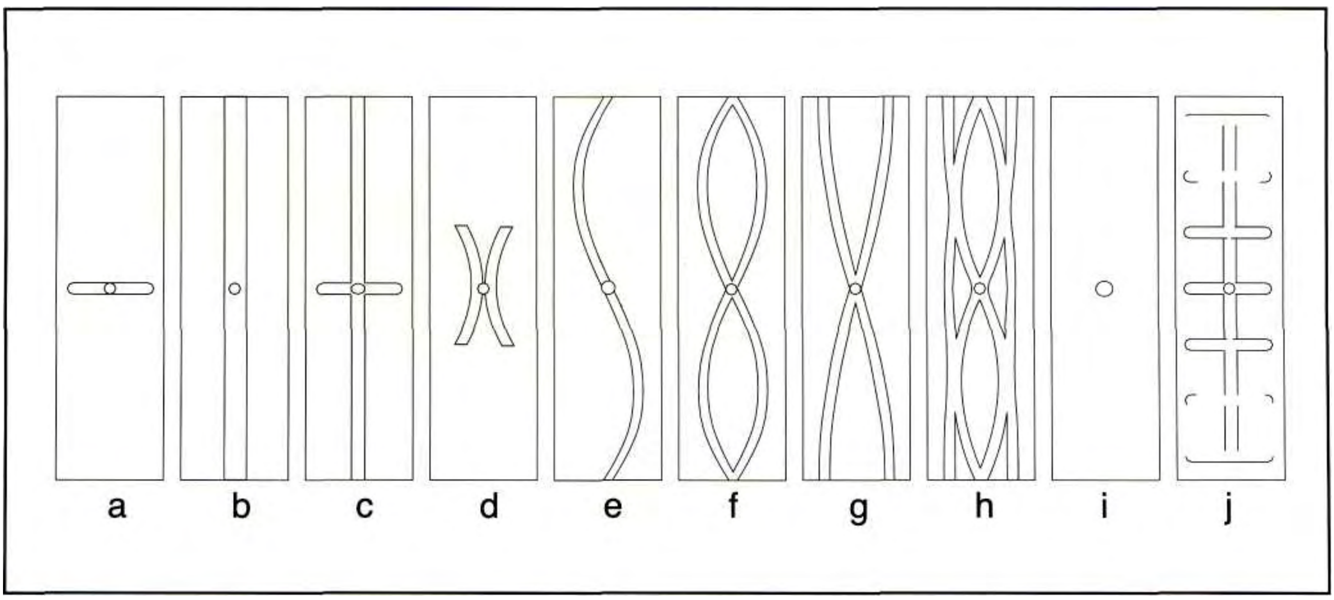


**Figure 3.7. Various Grease Redistribution and Supply Groove Patterns: (a) straight axial, (b) circular, (c) axial and circular, (d) butterfly groove, (e) single loop, (f) double loop, (g) figure eight, (h) double figure eight, (i) chain-link, (j) ladder.**

## Straight Axial Supply Groove

Where the applied load is predominantly in one direction and continuous rotation exists, then the optimum design is to use one or more axial feed grooves, see **Figure 3.7a.** The axial slot should be oriented so it is in the unloaded area of the bearing. In addition to distributing the grease, the groove also provides a reservoir for the grease and a trap to collect wear debris. Axial lubricant grooves may have an angular extent of as much as 30°. If one groove is specified this should be positioned between 90° and 120° upstream of the direction of applied load. If two grooves are used, they should be diametrically opposite at 90° to the applied load. Axial groove length should be approximately 0.8 times the bearing width.

## Circumferential Grooves

If the applied load varies considerably in direction, then the choice of a central circumferential groove is preferred, see **Figures 3.7b** and **c.** Such a design has, however, a lower load capacity than an axial groove bearing of equivalent size. Typically the width of the circumferential groove is approximately 20% of the total bearing width. When using this groove pattern for oscillating bearings, it is preferred that the axial groove spacing is about the same as the sweep during a given oscillatory move. This is to ensure that the entire shaft is relubricated for each cycle. If rotational motion is very limited, then the ladder-groove as shown in **Figure 3.7j** should be used.

## Recirculating Configurations

One difficulty with grease lubrication is to keep the grease in the bearing without affecting the load capacity greatly. Configurations such as those shown **in Figures 3.7f,** g, **h** and **i,** and to lesser extent in **e** achieve this. These configurations also tend to be preferred for longer bearings. The side grooves recapture the lubricant and bring it back to the grease supply location for redistribution. Dirt and debris are however also brought back into the load zone and this may lead to abrasive wear in the bearing. The grease application hole is typically placed opposite the point of load application with these configurations.

Configurations such as shown in **Figures 3.7f** and **h** reduce the load capacity by virtue of interrupting the load bearing area. This reduction is less for **g** and i. Both **g** and **i** are also very efficient in recapturing the grease. Tests conducted by Bethmann and reported by Spiegel[15] show the benefit in recirculation of grease for these types of groove patterns, see **Figure 3.8.** At a bearing stress of 3 MPa (435 psi), the chain-link type groove consumes about 20 times less grease than the straight axial groove.

## Groove Geometry

The grooves should not extend out to the ends of the bearing and should not take up too much of bearing area providing load support. Conventional design calls for the grooves to be machined to within about 10% of the ends of the bearing so the lubricant is kept within the bearing area. However, under heavy load conditions where bearing temperature is high and debris accumulation becomes a problem, it is better to have the grooves exit at the ends of the bearing so periodic flushing of the bearing with fresh grease will remove the debris and lubricant decomposition products collected in the grooves. Experiments with "non-recirculating" grooves indicate lower wear than with recirculating grooves.

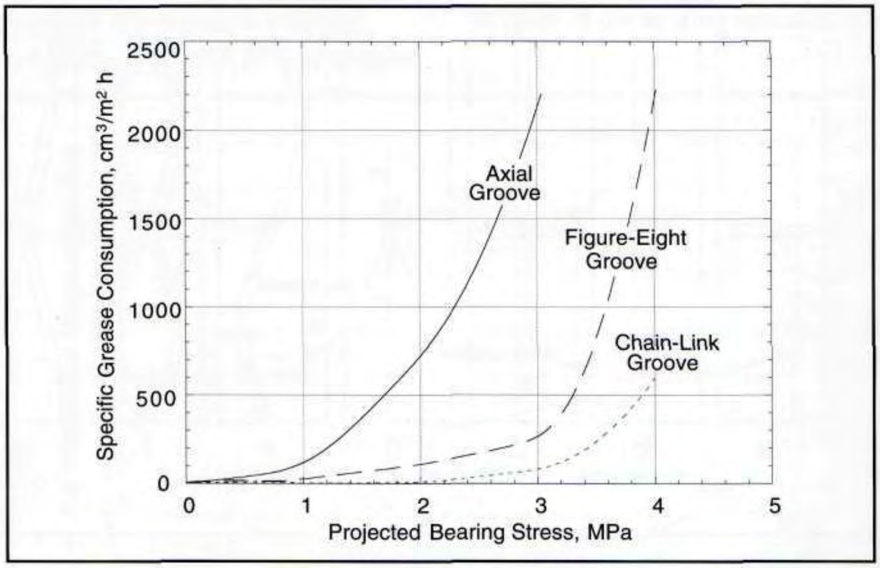


**Figure 3.8. Experimental Grease Consumption for Different Groove Patterns (Sliding speed is 1 m/s, bearing diameter is 80mm).**

There are many different cross-sectional groove geometries. The most commonly used are shown in **Figure 3.9.**

It is very important that the groove edges are all properly rounded. Sharp edges tend to act as a grease scraper and thus remove grease from the shaft rather than apply it.

Recommendations for groove depth and width for grease lubrication are given by Rippel[9], and are shown in **Figure 3.10.**

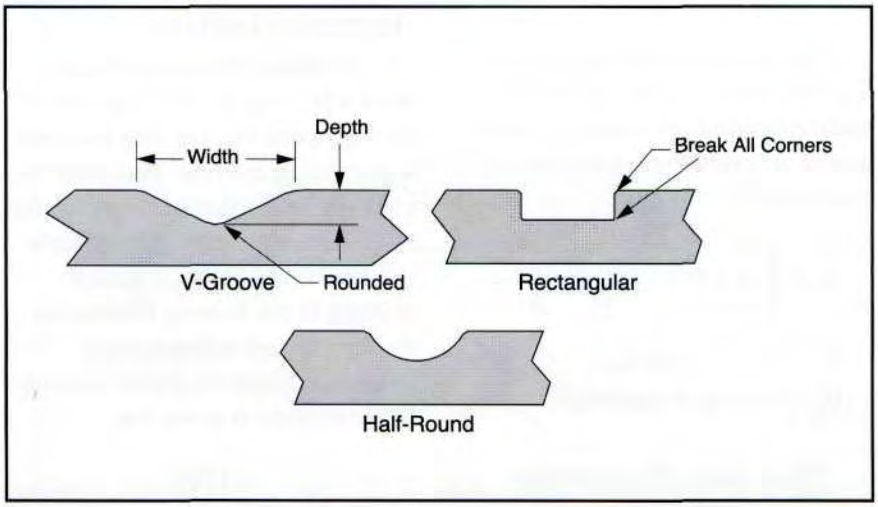


**Figure 3.9. Common Forms of Grease Grooving. From Rippel[9].**

## Grease Consumption

Low wear rates can be maintained only over long periods of time when grease is present in the bearing and can perform its intended function. Because grease is lost from the bearing due to side leakage, and also somewhat consumed due to chemical action, we have to make provisions for its replacement. This may be done on a continual basis or as is more common, on a periodic basis. Certainly a major advantage of grease-lubricated cast bronze bearings is that they are not overly critical for a continual grease supply, and can thus be used for periodic relubrication.

Grease consumption for grease-lubricated bearings is a very complex phenomenon, and thus far has been treated on an empirical basis only. It clearly must be a function of the bearing size, and practical experience indicates that it lies somewhere between:

$V_f = (40\text{-}1000)$ Aj, where Aj is the inside bearing area ($\pi BD$).

This expression shows a very wide range of possible grease consumption, and can only be used as a starting point.

Besides bearing size, we must also expect that it is a function of bearing speed, bearing loading, operating temperature and groove patterns. These effects were incorporated in an empirical expression by Henningsmeyer as reported by Bartz[14] that may be used for a more refined estimate. Henningsmeyer gives the following expression for the grease consumption rate:

$$V_f = 3600\ BU$$

when: $0.01\ \Lambda = G$

and where;

B = bearing width,
U = sliding velocity,
$\Lambda$ = loading factor, and
G= environmental factor.

This expression does not take the groove pattern into account, and we will thus need experimental refinement to get better estimates. This can easily be done through very careful monitoring of the bearing temperature. Rapidly increasing temperatures suggest more grease is needed.

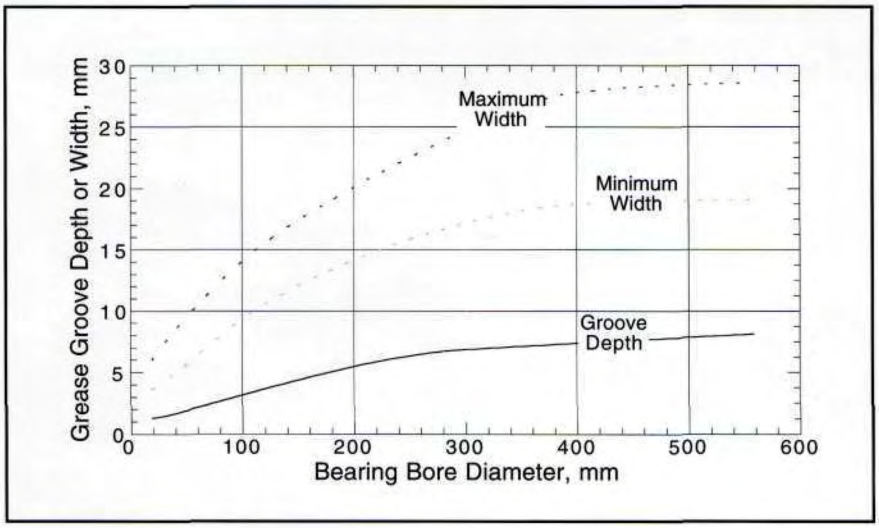


**Figure 3.10. Recommended Groove Dimensions for Grease Lubrication.**

The bearing loading factor, $\Lambda$, is a function of the bearing load and the bearing operating temperature. The following expression is suggested:

$$\Lambda = \left[ 0.5\,P + \frac{(\theta_B - 60)}{20} \right]$$

P = bearing stress,
$2_B$ = bearing temperature.

When using this expression care should be taken that $\Lambda \geq 1$. The environmental factor may be estimated from the conditions surrounding the bearing. Values are suggested in **Table 3.2.**

## Regreasing Intervals

Assuming that grease leaks from a bearing at a constant rate during operation, we may estimate a regreasing interval. This interval is based on the assumption that the maximum allowable grease depletion is 25% of the total grease volume in the bearing (excluding the supply and redistribution grooves). The total grease volume in the bearing is given by;

$$V_L = \frac{\pi D B C_D}{2}$$

where:
B = bearing width,
D = journal diameter,
$C_D$ = diametral clearance.

Thus, the regreasing interval may be estimated from;

$$T_{regrease} = \frac{0.25 V_L}{V_f} = \frac{\pi D B C_D}{8 V_f}$$

When using this estimate, it should be realized that the estimate is for the actual operating time and not clock time. Also grease quantities smaller than about 0.04 $cm^3$ (.002 $in.^3$) are difficult to deliver.

| | |
|---|---|
| Likely ingression of dirt or dust | G = 0.01 |
| Likely ingression of water | G = 0.02 |
| Likely ingression of dirt and water | G = 0.03 |

**Table 3.2. Values for Estimating the Environmental Factor of Conditions Surrounding the Bearing.**

NOTES:

# Hydrodynamic Journal Bearing Design

# HYDRODYNAMIC JOURNAL BEARING DESIGN

When the load on a bearing is fully supported by hydraulic pressure in a lubricant film, there will be a finite separation between the bearing and shaft. The two components do not contact each other and wear is nonexistent or very low. If this separating lubricant film is solely caused by the motion of the journal surface relative to the bearing, we speak of hydrodynamic lubrication.

In a hydrodynamic journal bearing, the part that moves is generally the journal (as with the shaft of a motor), sometimes the bearing (for example, the wheel of a Conestoga wagon), and sometimes both parts (such as a connecting rod that joins a piston and the crankshaft in an automobile engine).

Since many motors, engines and other machines incorporate hydrodynamic journal bearings, the annual production of hydrodynamic journal bearings is in the billions. Not only is the hydrodynamic journal bearing very common, it is superior to rolling-element bearings for many purposes because it can carry heavier loads, operate at higher speeds, is less expensive, is more reliable, and can last indefinitely when designed well.

A typical hydrodynamic journal bearing is shown in **Figure 4.1.** Because the hydrodynamic operation is dependent on the presence of lubricant, we typically require a constant supply. Several ways of achieving this are commonly used. The lubricant to a journal bearing may be supplied by a non-pressurized lubricant feed such as an oil ring, a wick or other means. It may also be supplied by pressure. Pressure-fed systems may include supply and return lines for the oil, a storage tank, filter, and temperature and pressure regulators. Similar to grease-lubricated bearings, grooves may be used to spread the oil along the bearing.

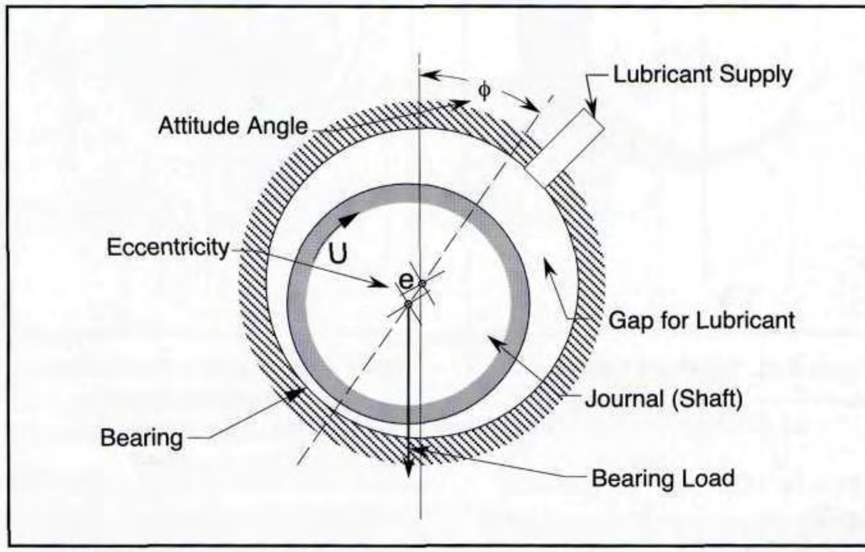


**Figure 4.1. Basic Journal Bearing Component Designation.**

## Load Capacity of Hydrodynamic Journal Bearings

The key to hydrodynamic lubrication is the formation of a lubricant film that separates the shaft from the bearing. Pressure in this film, when integrated over the bearing area, will support the load applied to the journal. Because this film consists of lubricant, it may be sheared as much as necessary without any permanent damage to the bearing.

Pressure in the lubricant film develops when we drag lubricant into a converging space, A, shown in **Figure 4.2a.** This drag is imparted on the fluid because fluids like to adhere to surfaces. As the lubricant is drawn around into the converging space, a certain amount of the fluid is forced out against the viscous action, B. This results in a local pressure. As the gap converges, the sideways squeezing becomes more and more difficult and higher pressures develop. In the end, only a small but finite amount goes through the smallest gap in the bearing. After the smallest gap is passed, C, the space

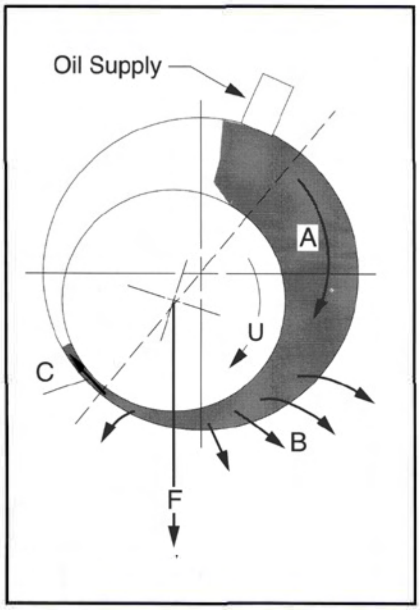


**Figure 4.2a. Lubricant Flow in a Journal Bearing.**

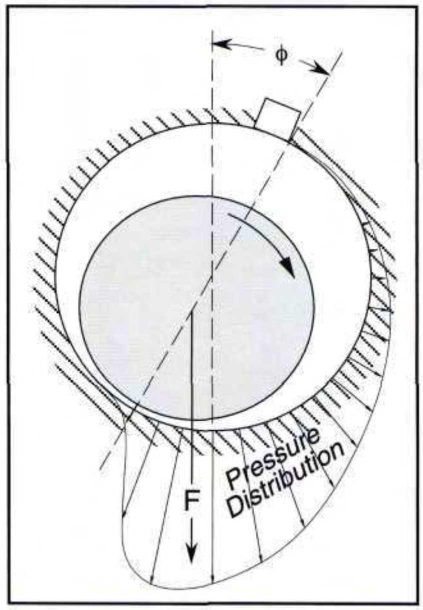


**Figure 4.2b. Pressure Distribution Resulting from the Lubricant Flow.**

opens up again, and the pressure rapidly drops to ambient.

The resulting pressure distribution around the bearing looks thus something like that shown in **Figure 4.2b.** Note, this pressure is generated by the action of the lubricant film in the bearing, and not by the supply pressure. Integrating this generated pressure over the bearing area results in the applied load.

The beauty of a journal bearing is that we can get this hydrodynamic action almost for free. The only special provisions we need to make are to ensure a suitable clearance exists between the shaft and the bearing surface and sufficient lubricant is supplied at all times. The clearance is easy to achieve, simply by making the shaft somewhat smaller than the bearing bore. The rest of the action takes place by itself. The thickness of the gap, for example, adjusts itself to the load that is applied. If this load is small, then a fairly large gap at C is possible because not much pressure is needed to support the load. For high loads, this gap will be small because higher pressures are needed.

Full-film formation occurs as long as the smallest gap in the bearing is greater than the combined roughness on the journal and bearing. The smallest gap in the bearing occurs along the line of centers in the vicinity of the load. The line of centers makes an angle, $\phi$, with this load line. The actual value of the minimum film thickness is given as $h_{min}$. From the displacement of the journal center relative to the bearing center, see **Figure 4.1,** and the radial clearance between the journal and the bearing, we calculate this gap as;

$$h_{min} = C - e$$

where C = radial clearance $(D_b\text{-}D_j)/2$, and
e = bearing eccentricity.

From the foregoing discussion of action in a hydrodynamic journal we might expect that the load supported by the film would be a function of the viscosity, speed, journal area, and radial clearance. These variables are often combined into a convenient dimensionless variable called the Sommerfeld number, in honor of the work done by Sommerfeld in finding a closed form solution to load capacity. Hence the Sommerfeld number is given as:

$$S \equiv \frac{\eta NDB}{F}\left(\frac{R}{C}\right)^2 = \frac{\eta N}{\bar{P}}\left(\frac{R}{C}\right)^2$$

This number contains all the essentials required to find the load a given bearing can support. It is a function of the bearing eccentricity and the bearing width-to-diameter ratio, as shown in **Figure 4.3. In Figure 4.3** the independent variable is given as the eccentricity ratio. This ratio is defined as:

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

and forms yet another convenient dimensionless group to use. We may use the eccentricity ratio to calculate the minimum film thickness in the bearing by;

$$h_{min} = C - e = C(1 - \varepsilon)$$

The calculation of Sommerfeld numbers has been studied and researched at great length ever since Osborne Reynolds formulated the differential equations to solve for this problem. For in depth details of the mathematics involved, the reader is referred to

Cameron[16], Constantinescu[17], Szeri[18], Hamrock[19], and others. Tables prepared by CD A[20] contain some of the earliest calculated data by Raimondi[21].

Besides the load capacity we also need data on the required oil flow and the friction generated by the bearing. These data may be found in CDA[20]. The data from these tables were incorporated into the **Hydro** program for bearing calculation purposes.

## Thermal Effects

Heat generated by lubricant shear in a bearing is of major importance in practical applications, mostly because it causes an increase in lubricant temperature. For most lubricants, the viscosity decreases with increasing temperature. This directly affects the load capacity of a bearing as may be seen from the Sommerfeld number. Lower viscosities mean lower load capacity for a fixed Sommerfeld number.

A second effect of increasing bearing temperatures is the increase or decrease in bearing clearance. If materials with different thermal expansion coefficients are used, the resulting clearance increase can result in a significant loss of load capacity. Both the viscosity-temperature and thermal expansion effects are incorporated into **Hydro.**

Frictional heat can be removed by conduction through the bearing components or by oil flow. In some cases, sources external to the bearing bring additional heat into the system so that the lubricant and bearing then act as a cooling system as well.

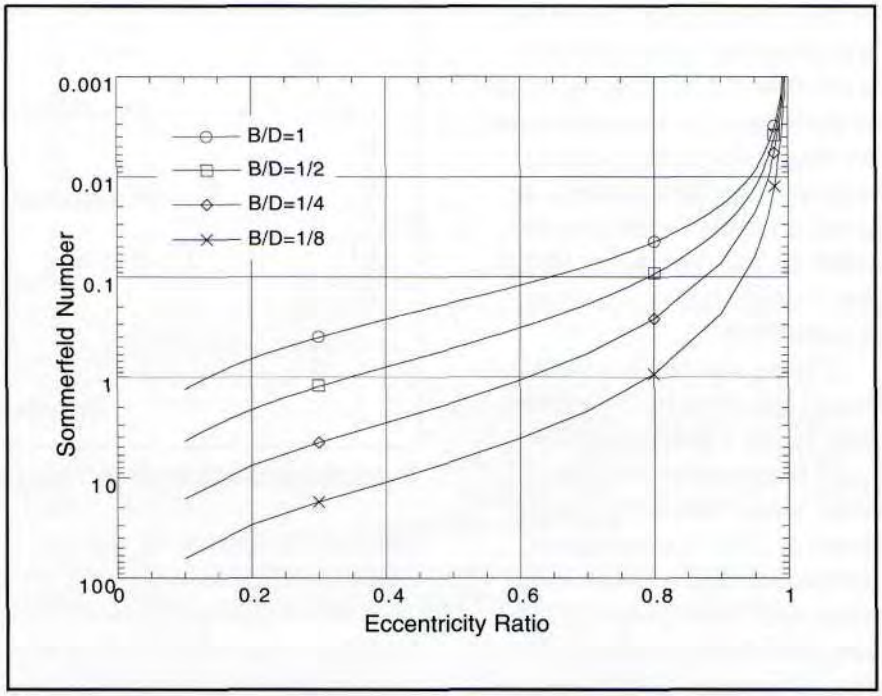


**Figure 4.3. Sommerfeld Number as a Function of the Eccentricity Ratio for Bearings of Different Width.**

Cast bronzes are excellent choices as bearing materials because of their very high heat transfer coefficients. Hence structural cooling of the bearing is quite feasible for a moderately loaded hydrodynamic bearing.

Heat removal by a continuous supply of cooler lubricant is also a very effective way to cool the bearing. Both structural and lubricant cooling of the bearing are incorporated in the **Hydro** program.

Additional cooling of the bearing may be obtained by supplying lubricant at higher than required flow rates through the bearing. In fact automotive engine bearings are cooled this way. The current version of **Hydro** does not have this feature. It is expected that a future release will have this option, together with some additional lubricant supply methods.

## Transition to Hydrodynamic Lubrication

For a given journal bearing with a fixed R/C ratio, the coefficient of friction may be plotted as a function of the hydrodynamic parameter, $\eta N/P$. This friction is due to the viscous shear of the lubricant only. In real bearings however the surfaces have a definite roughness, and it may be expected that at high values of the eccentricity, *i.e.*, when the film thickness is small, some interaction of asperities on opposing surfaces occurs. When the eccentricity goes to unity, the friction coefficient plot needs to be modified to include the boundary layer friction, see **Figure 4.4.**

Minimum friction occurs at an eccentricity of $\varepsilon \approx 1 - h_c/C$. The transition takes place where the film thickness is equal to the convection heat transfer coefficient, $h_c$.

For any practical application, we always operate to the right of this point. This region is known as the hydrodynamic friction region, and the slope reflects the operating viscosity. The point at which the action is mostly hydrodynamic is called the transition point. Notice that viscosity has an impact on this transition.

During start-up from cold conditions, the viscosity is typically high. Hence a lower transition speed is required to get hydrodynamic action. When the bearing comes to a halt after extensive operation at higher temperatures, viscosity is much lower, and the danger of scuffing is higher. If the transition occurs at a higher speed, more heat is produced at this point. Critical in the prevention of scuffing is the choice of bearing materials. Here the cast bearing bronzes excel because of their very good boundary friction characteristics. (See **Chapter III.**)

The transition journal speed, $N_T$ may be calculated from the minimum recommended safe film thickness and the Sommerfeld number. Consider the critical eccentricity ratio as follows;

$$\varepsilon_T \approx 1 - h_c / C \text{ and thus:}$$

$$S(\varepsilon_T) = \frac{\eta_{eff} N_T D B}{F} \left( \frac{R}{C_{eff}} \right)^2$$

We extract the critical speed from this as:

$$N_T = S(\varepsilon_T) \left( \frac{C_{eff}}{R} \right)^2 \frac{F}{\eta_{eff} D L}$$

This speed can now be used in the boundary lubrication calculations

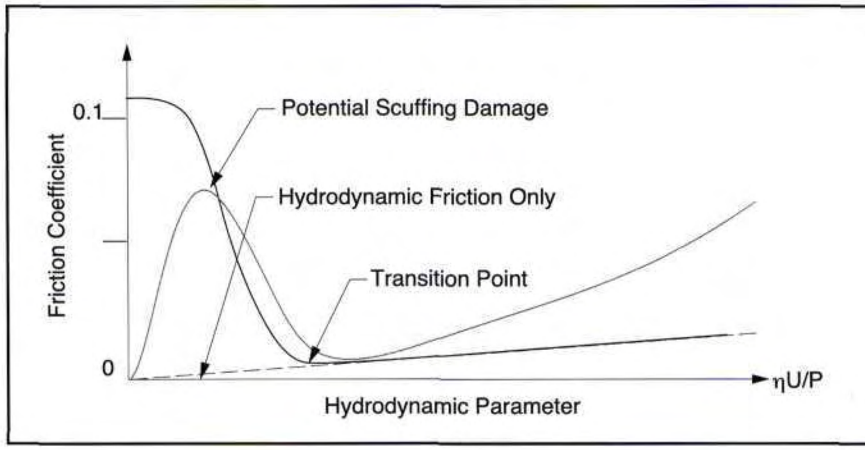


**Figure 4.4. Modified Bearing Friction due to Boundary Lubrication.**

to make sure the bearing will not suffer a scuffing failure. It may be required to adjust the surface roughness, viscosity or bearing size to make possible a reliable transition-to-boundary lubrication.

## Design Criteria for Journal Bearings

Having considered some of the theoretical aspects of hydrodynamic lubrication, we now have come to the hardest part of all. By what criteria do we judge a bearing design to be satisfactory?

We need to define several design criteria. For example: what is the minimum safe film thickness for operation? What are the factors for safety? What are the allowable temperatures? In some cases, you may have internal guidelines to follow based on previous experience. It may also be very worthwhile to use **Hydro** and reverse engineer a couple of designs that you know to give satisfactory performance.

### Bearing Size

The design of a hydrodynamic bearing is aimed at avoiding extensive metal-to-metal contact when components are under load. Based on this aim, it would seem that we should try to make the separation between the two bearing members as large as possible. This however would take a lot of power, since a hydrodynamic bearing system is nothing more than a viscous pump, and a very leaky one at that.

To arrive at some reasonable choices about the separation, we tend to make the bearing size as small as necessary, and at the same time have a reasonably safe separation. Because bearings of large length tend to suffer more from edge contact due to shaft deflections and misalignment, we seldomly use B/D ratios larger than 1. Also we will try to maximize the safe temperature increase in the bearing, within the limits as specified. A design that follows these guidelines will be safe and, at the same time, not waste unnecessary energy.

### Minimum Recommended Film Thickness

From the above discussion, it is quite clear we need to establish some criteria for the separation of the two surfaces. This suggested separation will depend on a number of judgmental variables that should be carefully considered before making a final choice. Consideration should be given to the following:

- size of the bearing (larger bearings require larger separation just due to manufacturing tolerances),
- applied loads during running and during starting periods,
- bearing deflections and frame distortions,
- consequences of a bearing failure, loss of life, *etc.*

With these considerations in mind, we now must choose a minimum acceptable film thickness for the bearing. Below are some guidelines based on empirical information and with a solid experimental basis. They should help to minimize the probability of any trouble.

- Minimum recommended film thickness $h_{2min}$. In most cases, this is a function of machine size and surface roughness. In the absence of any internal company guidelines, **Figure 4.5** may be used. These data are based on marine bearing design practice, and usually have some form of load sharing, see Hill[22].

The DIN[10] guidelines for the minimum allowable film thickness take the operating speed into account. This is a more refined method.

Comparison between the values in **Figure 4.5,** and those in **Table 4.1** indicates they are roughly in line with each other.

- Eccentricity ratios of less than 0.7 should be avoided because of possible dynamic instabilities.

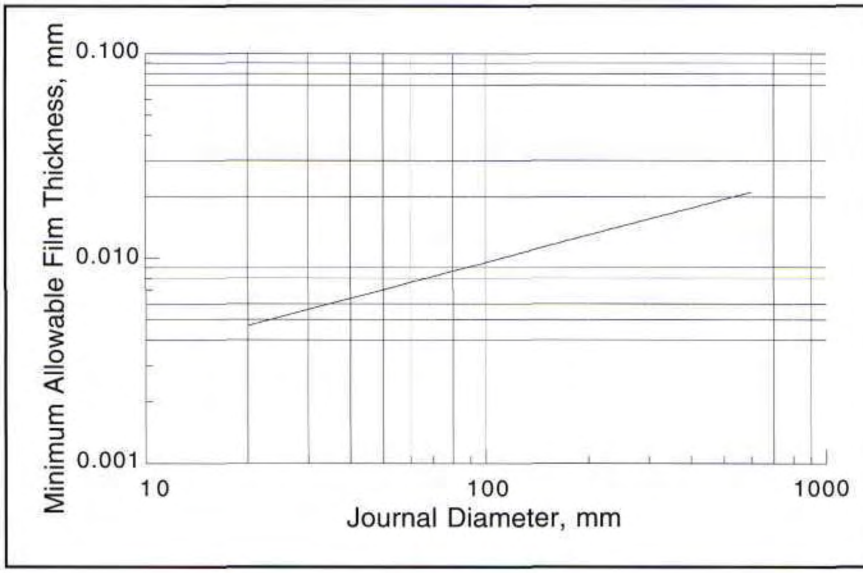


**Figure 4.5. Minimum Recommended Film Thickness for Journal Bearings.**

| Journal Diameter D, mm | | Sliding Velocity U, m/s | | | | |
|---|---|---|---|---|---|---|
| from | | - | >1 | >3 | >10 | >30 |
| | to | 1 | 3 | 10 | 30 | - |
| 24 | 62 | 3 | 4 | 5 | 7 | 10 |
| 63 | 159 | 4 | 5 | 7 | 9 | 12 |
| 160 | 399 | 6 | 7 | 9 | 11 | 14 |
| 400 | 999 | 8 | 9 | 11 | 13 | 16 |
| 1000 | 2500 | 10 | 12 | 14 | 16 | 18 |

**Table 4.1. DIN[x] Guidelines for Minimum Allowable Film Thickness for Journal Bearings, µm.**

## Design Clearances

Equally important in journal bearings is the correct choice of bearing clearance. We have seen that small bearing clearances result in high operating temperatures. On the other hand, large clearances result in loss of performance. The right clearance is thus very important.

- Clearance ratios: for small bearings use a clearance ratio $500 \leq R/C \leq 1000$, with the larger clearances for high speed application. For larger bearings, use the values as given in **Figure 4.6.**
- Seizure concerns: When the bearing and journal materials have different thermal expansion coefficients we must make sure this effect is considered. Also we need to check and see if the chosen clearances can result in bearing seizure at temperature extremes. This may be the case at either end of the temperature spectrum. If at all possible, the thermal

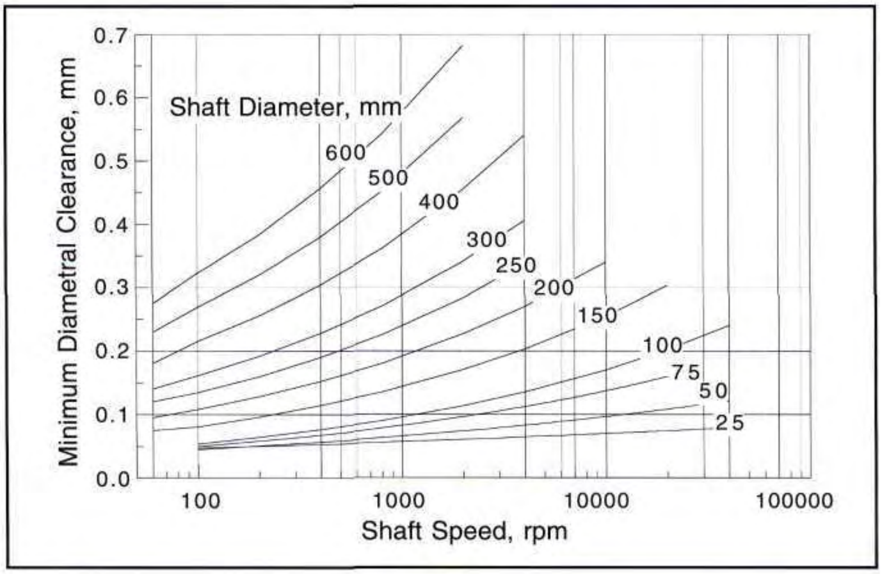


**Figure 4.6. Minimum Recommended Diametral Clearances for Journal Bearings.**

expansion coefficients of the bearing should always be equal to or greater than the journal material. This requirement is automatically satisfied for most common configurations of steel journals and cast bronze bearings. It is also possible to get start-up seizures when very quick heating of the shaft allows it to grow more rapidly than the bearing. Seizures of this type typically occur very fast (within 10 seconds of start-up).

To check for possible seizure conditions, perform a layout calculation as shown in **Figure 4.7.** The tolerance range on the nominal bearing and shaft size is taken into consideration.

## Maximum Temperatures and Temperature Rises

- Maximum permissible lubricant temperature $\theta_{max}$ This is calculated at the cavitation point in the bearing. This criteria is mostly for oil oxidation life. This gets rapidly shorter as the operating temperature exceeds 80 C (176 F). There are two different cases that are considered;
  - Bearings with oil circulation: $\theta_{max}$ is 100-125 C (212-257 F).
  - Self-contained bearings (air cooling): $\theta_{max}$ is 90 C (194 F).

  When synthetic lubricants are used we add 30-40 C (86-104 F).
- Permissible lubricant temperature rise, $\Delta\theta$ (from lubricant inlet to exit), is from 30-50 C (86-122 F), depending on the materials used. It is restricted to keep thermal gradients and subsequent distortions to reasonable levels. Because of the high thermal conductivity of cast bronze materials we can use the upper limit.

## Bearing Stresses and Materials

- Satisfactory boundary lubrication must be provided for start-up purposes. This translates into a maximum bearing pressure of about F/LB < 2.5-3 MPa (362-435 psi) at start-up. When running, this may be considerably higher. If the start-up load results in pressures higher than those recommended we should provide a hydrostatic assist.
- Maximum recommended bearing pressure calculated over the projected area depends on the bearing materials chosen. For cast bronze bearings, we will follow the guidelines as indicated in **Table 2.3,** page 24.

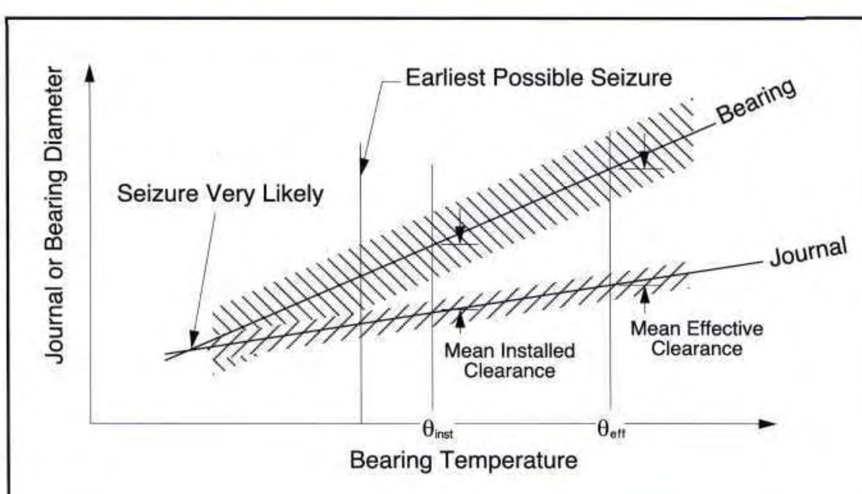


**Figure 4.7. Clearance Variation with Temperature for a Bearing with a Higher Thermal Expansion Coefficient than the Journal**

## Safety Factors

One has to be very careful with the use of safety factors in hydrodynamic bearing design. Liberal use of large safety factors will lead to very inefficient bearing design, and in some cases, may even lead to design failures.

- Safety factor on load. In practice, a safety factor of 1.2 to 2.0 on load is used depending on how well the load is known. Under no circumstances should a factor greater than 2.0 be used.
- Safety factors on oil supply should be about 1.5 to 2.0.

## Lubricants and Filtration

- In modern designs the viscosity grade does not exceed ISO VG100 (lighter oils are often used).
- Adequate filtration must be provided. Typical filtration levels should be for particles greater than 50% of the minimum operating film thickness. Filtration below 10 µm may require special filters. The filter must be sized for full-flow filtration, and should consider the effects of loading-up.

## Geometry Considerations

- Oil inlet may be anywhere in the low pressure region, but is safest in the region between the angle of eccentricity and opposite the line of the normal load (as in **Figure 4.2b).**
- Bearing wall thickness should be about 30-40% of the largest surface dimension to avoid load based deflections that might affect performance.
- Shaft and housing deflections should be kept to levels less that half of the minimum film thickness in the bearing. In some cases it may be necessary to provide special constructional modifications to reduce effects of distortions.

## Lubricant Grooving

Grooves may be machined into the bearing surfaces to promote the distribution of lubricant. The application and precautions are similar to grease grooving, see page 35. For oil-lubricated bearing grooves, use only about 60% of the suggested grease groove width.

## Machining Considerations

- Cylindricality and waviness errors should be less than 50% of the smallest expected film thickness for the operation of the bearing, thus:
  waviness error $< 0.5 \times h_{2min}$.
- The maximum surface roughness should be less that 25 to 40 times the minimum separation, or:
  rms surface roughness $< h_{2min}/(25 \text{ to } 40)$

This is for the harder of the two bearing components, normally the shaft.

A word of wisdom about surface roughness. Experience has taught us that *bearing surfaces should be as smooth as possible, yet as rough as necessary.*

A certain amount of roughness acts in a beneficial way. Small pockets that retain lubricant can significantly improve load capacity during start-up times.

## Other Journal Bearing Details

The analysis in **Hydro** is for the journal bearing with the journal rotating and the bearing stationary. What if the bearing rotates and the journal is fixed? What if the load rotates relative to the bearing? We will provide some simple rules for this in the next section. We will also provide some simple guidelines on easy ways to avoid severe edge-loading problems.

## Bearing and Load Rotation

So far the load capacity and friction analysis for journal bearings has dealt with the simple case of journal rotation. In many practical applications we also have bearing rotation, load rotation, or combinations of all three. It is quite simple to extend the foregoing analysis to include these additional possible rotations by using an equivalent journal speed. Note that all rotations are referenced to the load vector. The general case of journal, bearing and load rotations may be analyzed by considering journal and bearing rotations independent from load rotations, see **Figure 4.8.**

In the first case, the equivalent speed results from the addition of journal and bearing speeds. The second case derives from arresting the load rotation through the backward rotation of the entire bearing system. This imparts a speed of $-N_F$ on both the journal and bearing, hence the equivalent journal speed is $-2\,N_F$. The final case is the most general and derives from the addition of the two cases above. Thus we convert journal, load and bearing rotations into a single journal rotation by:

$$N_e = N_j + N_b - 2\,N_F$$

where:

$N_e$ = equivalent journal rotation
$N_j$ = original journal rotation
$N_b$ = bearing rotation
$N_F$ = load vector rotation.

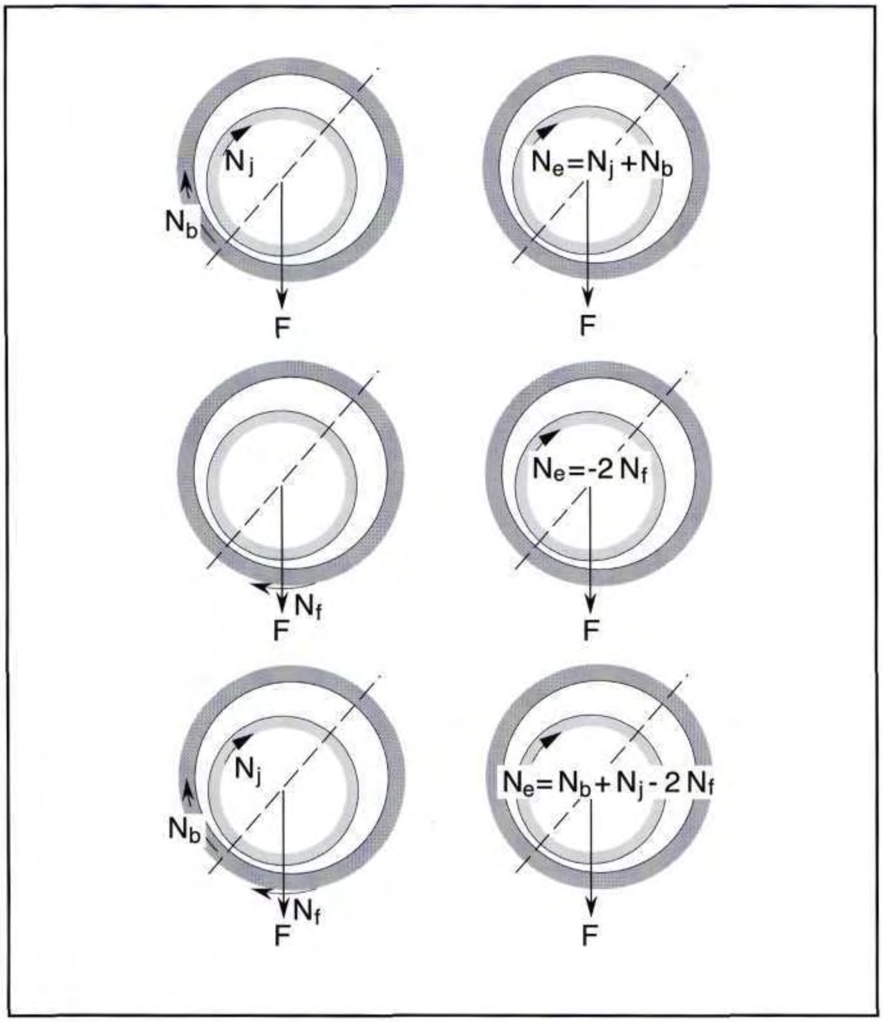


**Figure 4.8. Equivalent Journal Rotations for Different Journal, Load and Bearing Rotations.**

As might be apparent by now, the sense of the rotations should be kept in mind. We have used clockwise here as positive. (Note: Some of these bearings will require a ring groove supply in order to function correctly.)

## Geometries to Mitigate Edge Loading due to Shaft Bending

Edge loading due to shaft bending is a serious problem, and can significantly reduce load capacity. **In Figure 4.9** are several simple ideas that can reduce the severity of the problem (from Bartz[14]).

The last column in **Figure 4.9** shows some very simple constructions that often can easily be incorporated in a new design. Often these 'silent' features introduce a significant robustness into the design.

## Mating Surfaces

The choice and properties of **the** mating surface in a bearing system are just as important as the selection of the bearing material. Specifically, the material type, the surface roughness and preparation methods, and the hardness of the mating bearing surface are extremely important. In the following discussion of each of these topics we will refer to the mating surface as the shaft. This is only out of convenience, and the details of the discussion apply equally well to the mating surface of a thrust bearing, a spherical bearing or any other form of bearing surface.

## Shafting Materials

The predominant number of bearing applications involve steel as one of the members. The predominant shafting material is steel, either a carbon steel, tool steel, gray or nodular cast iron. Stainless steel shafting may be used, but care must be taken in selecting grades that have high seizure resistance when used with certain bearing alloys. Fatigue strength of the shaft is important if cyclic loading is anticipated on the bearing.

The use of the case-hardened shafts in many machine elements is very common. Hardened steel is very resistant to both adhesive and abrasive wear. This wear resistance increases with hardness. From a practical standpoint, the maximum obtainable hardness on carbon steel components is about 65 HRC. At this level of hardness, the ductility of the steel is very low, and failure by brittle fracture can occur very rapidly if the shafts are overloaded. To avoid this problem we use case-hardening techniques.

In case-hardening, the surface of the material is hardened to maximize wear resistance, while the core is kept soft to avoid brittle fracture of the part. This hardened case may be achieved on shafts by suitable induction heat treating of moderate to high carbon steels or by carburizing of a low-carbon based metal.

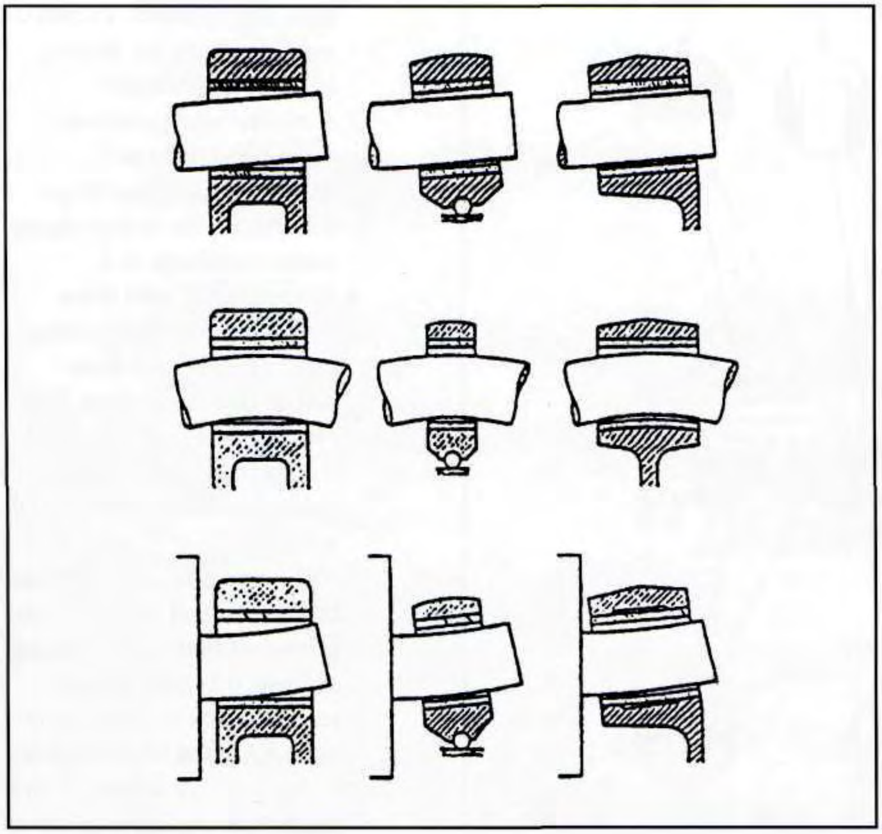

**Figure 4.9. Methods to Avoid Edge-Loading Problems.**

The latter method is used most commonly for mass production techniques and is less subject to error.

## Surface Preparation

The most common method of surface preparation is grinding. This may be done by centerless grinding or by conventional methods. The control of surface roughness and waviness levels, surface lay and fuzz generation are very important in the satisfactory operation of a bearing.

To keep circumferential waviness, out-of-round and grinding chatter under control, it is suggested that roughness and waviness traces be made periodically. Chatter should be kept to a minimum, and lobing effects of grinding must be within the circumferential waviness limits. The presence of 3 to 7 definite lobe patterns on the shaft is highly undesirable.

## Surface Roughness Levels

The best levels of roughness on the bearings and shaft are often a compromise between the cost to produce these levels of roughness, and the amount of run-in wear that can be tolerated. Wear of the bearing and shaft will lead to larger clearances, and thus part of the wear life of a bearing system may be consumed simply by not producing the optimum roughness on the new components.

Wear in a bearing system can take the form of adhesive wear, abrasive wear or a combination of both. We control adhesive wear through a proper materials selection, while abrasive wear can be controlled through roughness. To avoid two-body abrasion of the softer bearing material by the shaft, its roughness should be low. Typical nominal Ra values are around 0.2-0.3 μm (8-12μin.) for the shaft and double this value for bearings.

If the differential hardness between bearing and shafting materials exceeds 150-200 HV, the shaft roughness should be made correspondingly lower. Also, as the hardness of the bearing material increases, its roughness and the shaft roughness should be made correspondingly lower to avoid two-body abrasion by the bearing material on the shaft and by the shaft on the bearing.

## Hardness of Shafts

The surface hardness of the shaft should be high to resist wear by the bearing and to resist wear by the ever-present abrasives such as silica. In most cases, the shaft is the expensive, difficult-to-change member which increases the importance of minimizing wear on the shaft.

Tin bronzes require harder steel shafting and a good finish. The hard constituents in tin bronzes will tend to wear in the steel surface and improve its finish. Aluminum bronzes require a hardened steel shaft or chromium plating on the shafting. Copper beryllium alloys also require a hardened shaft material—preferably a tool steel.

As a good starting point, we should aim to make the shaft at least 150-200 HV points harder than the hardest constituent in a bearing material. This will ensure that almost all the wear in a bearing system will take place on the softer component. If the hardness differential is greater than this, care should be taken to lower the levels of roughness for the shaft. We tend

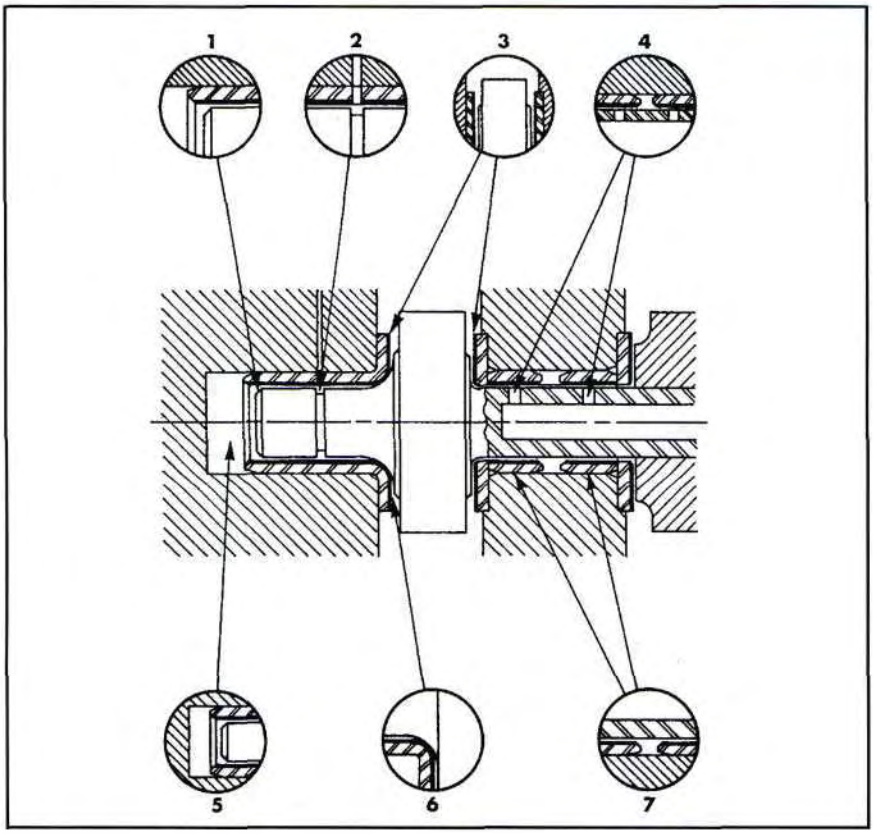


**Figure 4.10. Common Design Problems with Bearings.**

to rely on a certain amount of polishing of the shaft by the bearing material. This becomes more and more difficult as the shaft gets harder.

## Other Issues

Designing a good bearing can be helped immeasurably by knowing the pitfalls of bad design. **Figure 4.10** shows some of the common design errors presented by Pesek[23] to be avoided in order that satisfactory bearing life may be realized.

The index numbers shown on the figure are explained as follows:

1. Shaft ending within the bushing allows journal to wear a matching ridge in the relatively soft bearing material. This restricts free flow of the lubricant across the heavily loaded area, causes edge loading and results in high temperatures, excess wear debris in the bearing and eventual seizure.
2. Grooved shaft produces same conditions as 1.
3. Bearing thrust face larger than mating surface produces same conditions as 1.
4. Opposing oil inlet holes within one bearing bushing prevent proper oil flow across bearing surface. The pressure gradient produced results in high temperature, excessive wear and eventual seizure.
5. Dead end acts as a reservoir until filled and then prevents proper oil flow across bearing surface, resulting in high temperatures and excessive wear. (A point to remember is that it is just as critical in a bushing application to get the oil out as it is to get the oil in.)
6. Fillet ride also prevents oil flow.
7. Two precision bushings in one short hole can result in misalignment due to accumulated manufacturing tolerances. This forces the shaft out of line and results in edge loading. Do not use coaxial flats on shaft if they can be avoided.

NOTES:

# Computer Programs

# COMPUTER PROGRAMS

The CDA computer programs **Bound** and **Hydro** provide a means for the design of either boundary or hydrodynamic lubricated journal bearings. These programs can be used on either IBM PC compatible computers or Macintosh. They will accept units in both English (IPS) or Metric (SI).

The hydrodynamic program **Hydro** develops a table of film thickness, eccentricity ratio, power absorption, bearing oil temperature and oil flow rate for a given range of radial clearances and at a fixed load. The designer inputs the load, speed (rpm), bearing size, inlet temperature and selects a lubricant. The program stores a database of properties for various types of oils and inserts these data as needed.

The boundary lubricated bearing design program **Bound** is for heavily loaded, slow speed grease-lubricated bearings. It develops a table that includes estimated wear, maximum operating bearing temperature (with or without air cooling) and adsorbed horsepower for four classes of bronze bearing alloys. The program also sets bounds for acceptable operating temperatures.

Both programs cover a wide range of bearing operating conditions, from high bearing pressures and low speed (25 MPa, 5 cm/s (3500 psi, 10 fpm)) to fully hydrodynamic conditions (14 MPa, 5 m/s (2000 psi, 1000 fpm)). Both programs overlap in the range of operating conditions as shown in **Figure 5.1.**

**In Figure 5.1** maximum allowable bearing pressure (load divided by the projected bearing area) is plotted versus journal speed in rpm, for a 25 mm (1 in.) diameter bearing. In the hydrodynamic portion, the maximum operating bearing pressures are shown for two different grades of oil. In the boundary lubrication zone, maximum bearing pressures are shown for two different friction coefficients

The boundary lubrication program is based on grease lubrication only. The criterion for maximum bearing pressure is grease thermal instability. Boundary lubricated bearings wear continuously during service, and bronze bearing alloys have maximum allowable compressive loads. Thus it is possible to consider grease-lubricated operation for a hydrodynamic designed journal bearing that is clearly overloaded at, say, 10 MPa, 10 cm/s (1500 psi, 20 fpm) or to investigate low friction hydrodynamic design for a relatively slow speed, lightly loaded bearing, say, 3.5 MPa, 15 cm/s (500 psi, 30 fpm) that had been originally considered a boundary lubricated bearing.

## Background to the Programs

The hydrodynamic bearing program is based on the material presented in the previous chapter. The bearing performance charts of Raimondi and Boyd[21], developed from solutions to Reynold's equations have been computerized. The program automatically iterates the converging estimates for oil temperature. This is essentially a greatly expanded computerized version of the Rippel[9] manual.

The boundary lubrication program is based on empirical data

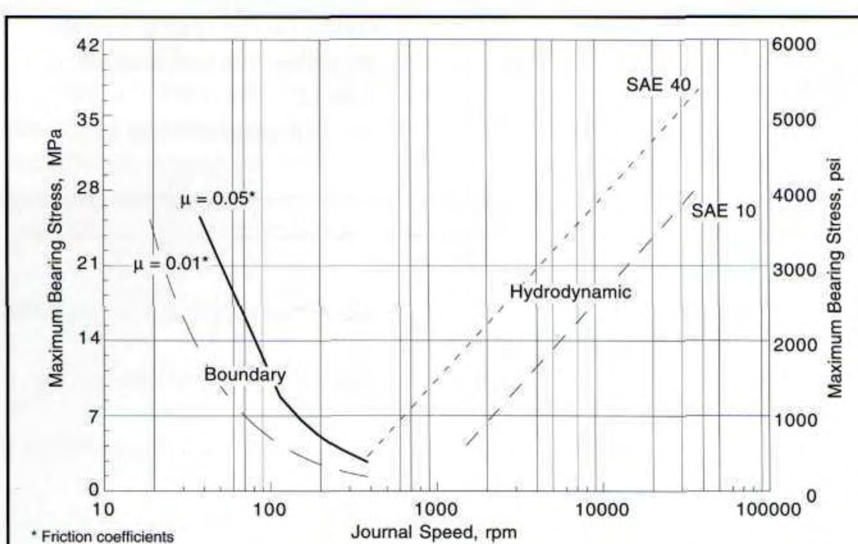


**Figure 5.1. Maximum Bearing Stress Operating Range for a 25 mm (1 in.) Journal Bearing for Both Boundary and Hydrodynamic Lubrication.**

for grease-lubricated bearings and heat-transfer equations with the maximum operating temperature as a criterion for design, see **Chapter III.**

The user should be aware that both programs have certain limitations. Specific assumptions and limitations are:

- Programs are for steady state loading only,
- Program are for full 360° bearings only,
- Oil is supplied at a single oil supply hole located opposite the applied load,
- The effect of oil oversupply is not considered in the cooling or performance of the bearing,
- The designer should ensure suitable alignment to avoid edge loading of the bearing,
- The heat transfer model for both **Hydro** and **Bound** assume certain boundary conditions (See **Appendix B** for more details.),
- Installed clearances are at room temperature, 20 °C (68 F),
- Clearances indicated are *radial* clearances,
- The included viscosity data on SAE oils is for high viscosity index oils.(See **Appendix C** for details.)

The two programs are intended to provide quick estimates for bearing performance for the design engineer. Special effects such as the influence of grooving, edge loading or non-circularity of the components is not taken into account by these programs. Nevertheless, they do provide a good first cut estimate of bearing performance.

For such exceptional running conditions as turbulent flow or bearing instability (whirl), other analytical approaches should be used. In any instance, it is best to run bearing tests to verify the design before freezing the design.

Graphical presentation of the hydrodynamic program output is an option. For IBM DOS compatible systems this requires a VGA or higher level monitor. Hard-copies of the graphs can be obtained on dot matrix printers that are Epson compatible, or on laser printers that are HP Laserjet compatible.

Graphics can be viewed, saved, and printed on all Macintosh systems.

## Using the Hydrodynamic Program

Insert the floppy disk and, depending on the system you are using, follow the instructions in **Figure 5.2.**

NOTE: The programs may also be moved to a directory on your hard-drive. Make sure you also copy the LUBE.TRU file to the same directory. All the input and output files will then show up on this directory.

**Macintosh**
**On the desktop:**
Double click on floppy disk icon
Double click on file **"HYDRO"**

**IBM PC DOS**
**At the DOS prompt:**
Type **"HYDRO"**
Press RETURN/ENTER

**Figure 5.2. Opening the Program.**

Something like the message in **Figure 5.3** should appear.

Answer the query. If you know of a file on the current disk that relates to this program enter it. If you want to start afresh, answer N, or simply press **RETURN** because the default value is (N)o.

The screen display shown in **Figure 5.4** will appear.

This table allows you to input bearing load, speed, size, ambient temperature, bearing thermal properties, lubricant and lubricant inlet temperature.

The table appears with the values inserted from a default set or the last run. Some of the values are default values such as thermal conductivity and expansion (values assumed for steel and bronze).

```
*************************************************************************

            CDA HYDRODYNAMIC LUBRICATED BEARING CALCULATION

Rev= 16.2  Date 930722  Time 12:13:43  Project: Journall  Units: IPS

*************************************************************************

Do you want to get an input file from disc? Y/N (N)
```

**Figure 5.3. Opening Screen.**

*************************************************************************

CDA HYDRODYNAMIC LUBRICATED BEARING CALCULATION

Rev= 16.2 Date 930722 Time 12:13:43 Project: Journal 1 Units: IPS

*************************************************************************

| ITEM # | NAME | UNITS | VALUE |
|---|---|---|---|
| A | Bearing Radial Load | ($lb_f$) | 500 |
| B | Shaft Speed | (rpm) | 3500 |
| C | Bearing Inside Diameter | (inch) | 1.0000 |
| D | Effective Bearing Width | (inch) | 1.0000 |
| E | Total Shaft Length | (inch) | 4.00 |
| F | Housing Outside Diameter | (inch) | 3.00 |
| G | Housing Width | (inch) | 1.00 |
| H | Shaft Thermal Conductivity | (Btu/h ft F) | 29.00 |
| I | Housing Thermal Conductivity | (Btu/h ft F) | 28.00 |
| J | Ambient Air Temperature | (F) | 83 |
| K | Air Velocity | (fpm) | .0 |
| L | Shaft Therm. Exp. Coeff. | (mu-in/in F) | 6.30 |
| M | Bearing Therm Exp. Coeff. | (mu-in/in F) | 10.20 |
| N | Lubricant Name | SAE 30 | |
| O | Lubricant Supply Temperature | (F) | 83 |
| P | Heat Remvl by lube (On/Off) | On | |
| Q | Heat Remvl by amb.air (On/Off) | Off | |
| U | Toggle the Unit system | | |

Enter Letter for Item # or Y(es) to accept ? ()

**Figure 5.4. Input Table.**

Initially, the program assumes your system uses an oil supply that the bearing pumps through its clearance, removing the bearing heat. NOTE: Clearance values in this program are *radial.*

## Entering Bearing Data

The input values shown above can be changed by selecting their item number (such as **B** or **b** for shaft speed) and then entering the appropriate values. If the system uses a drip feed or oil vapor system, air cooling can be assumed by selecting line **Q** and toggling line **P** (heat removal by air) on. In addition the air velocity may be supplied (line **K**) if forced air cooling is anticipated. **Appendix A** has more details on the assumptions made in the thermal modeling and some of the different ways that control can be exercised over the heat transfer mechanisms.

The bearing and shaft material thermal properties (thermal expansion and thermal conductivity) are parameters used in the program. These values in the default table (File Journal 1) are for C93200 bronze and AISI 1040 carbon steel. Since thermal expansion is a significant process in bearing performance, it is essential that the correct conductivity and expansion values are entered in the program for materials selected for a given design. Therefore items **H**,**I**, **L** and **M** should be adjusted for materials.

## Lubricant Selection

To select a different lubricant, press **N** and a table of lubricant names will appear. When you select a lubricant, its temperature-viscosity properties are entered into the program. Note that when you select the lubricant, the program displays the lubricant name and asks if it is correct. If you answer **Y**(es), you are returned to the input table.

The default data in the data file is based upon high viscosity index (HVI) lubricants. It should be realized that there may be a significant variation in viscosity-temperature behavior from one nominally same lubricant to another. Hence it is always best to enter your own measured viscosity data for the specific lubricant you intend to use.

Additional lubricant data may be added to the data file as shown in **Appendix** C.

### Entering Data in Alternate Unit Systems

You may use IPS (inch, lb-force, seconds) units or SI (meters, Newtons, seconds) units in your design. The unit system can be changed at any time during data input by toggling between the two when pressing U. You can mix your units as they are entered. For instance, if you have entered load, shaft speed and bearing dimensions in IPS units ($lb_f$, rpm, inches) and you have the shaft and bearing thermal expansion in SI units (µm/m °C), you can toggle to SI units by pressing **U** and **Return.** The table converts to all SI units. You can then enter your thermal expansion values. If you wish to have results in IPS, press **U** and **Return.** The table is then in IPS units.

## Calculating Bearing Performance

When the correct values are entered into the table, the program can be asked to compute the bearing performance values by answering Y(es) to the inquiry. The input can be saved in a file as instructed by the program.

However, if you do not save as a file, the same input values will come up again when you activate the input table. While the program is computing, a series of messages show the percentage of completion and information related to the bearing design. The results then appear as a table. See **Figure 5.5.**
NOTE: On some systems you may be asked to press **Return** half way through the output.

The output reiterates the input values, including the lubricant selected and its properties. The table of calculated data below the input table shows a number of performance values for 10 different radial clearances.

*(NOTE: If you are using diametral clearances in your design, rather than radial clearance, multiply the clearance values in the table by 2.)*

The output table provides power loss in the bearing, minimum oil flow rate, mean oil film temperature, minimum oil film thickness, eccentricity ratio and a rating for the reliability of the calculations (confidence). It also prints out the minimum recommended film thickness and clearance in two statements below the table.

After the bearing calculation results are displayed, the program gives choices for G(raphic) display of performance characteristics or sending the output to a F(ile). Graphics provides a valuable charting of bearing behavior over a range of clearances and show limitations such as recommended minimum film thickness and bearing instability threshold. If a D(ot) matrix or a L(aser) printer is on line, you may choose to print out the graphics. Otherwise it may be displayed on the screen. (NOTE: For IBM PC systems this requires VGA graphics and EPSON dot matrix or HP Laserjet compatibility of your hardware.)

Two plots are available: minimum film thickness versus radial load for all the clearance values displayed in the output table and minimum film thickness versus mean film temperature for all the clearances. Both plots show the selected operating load, recommended minimum film thickness and whirl instability threshold. Examples of the graphics are shown in the **Figures 5.6** and **5.7.**

**Figure 5.6** shows minimum film thickness versus radial load for 10 different clearances. The design load is shown as the applied load line. A vertical dot-dash line shows recommended minimum film thickness limit. A dashed line shows the limit for reduced load or "whirl limit." This limit represents a condition where the journal position has come close to the bearing axis. This can lead to instability (vibration) or shaft whirl.

**Figure 5.6** shows another operating characteristic for a hydrodynamic bearing: shearing of the lubricant film in the bearing generates heat. The program estimates the bearing temperature expected for the input parameters. It is wise to operate at as low a bearing temperature as possible to minimize thermal breakdown of the lubricant. Note that temperatures decrease with increasing clearance and increase with increasing film thickness. Thus with a given clearance, a fluctuating load will result in variable bearing temperature as the film thickness varies with load.

Instinctively, one would like to choose design parameters that result in the largest value of minimum film thickness. However, other factors must be considered.

```
************************************************************************
            CDA HYDRODYNAMIC LUBRICATED BEARING CALCULATION
Rev= 16.2  Date 930722  Time 12:13:43  Project: Journall   Units: IPS
************************************************************************

INPUT DATA:
  Bearing Radial Load                  (lbf)          500
  Shaft Speed                          (Rpm)         3500
  Bearing Inside Diameter              (inch)           1.0000
  Effective Bearing Width              (inch)           1.0000
  Total Shaft Length                   (inch)           4.00
  Housing Outside Diameter             (inch)           3.00
  Housing Width                        (inch)           1.00
  Shaft Thermal Conductivity        (Btu/h ft F)       29.00
  Housing Thermal Conductivity      (Btu/h ft F)       28.00
  Ambient Air Temperature               (F)            83
  Air Velocity                         (fpm)             .0
  Shaft Therm.  Exp.  Coeff.     (mu-in/in F)       6.30
  Bearing Therm Exp.  Coeff.     (mu-in/in F)      10.20
  Lubricant Name                        SAE 30
  Lubricant Supply Temperature          (F)            83
  Heat Remvl by lube (On/Off)             On
  Heat Remvl by amb.air (On/Off)         Off

LUBRICANT DATA : Name = SAE 30
                 Dynamic Viscosity = 1.200e+l Pa.s at - 18 C
                 Dynamic Viscosity = 9.000e-2 Pa.s at  40 C
                 Dynamic Viscosity = 9.500e-3 Pa.s at  99 C
                 Specific Heat     = 1.850e+3 Joules/kg C  at 20 C
                 Lubricant Density = 8.850e+2 kg/mA3  at 20 C

CALCULATED DATA:
PREDICTED HYDRODYNAMIC PERFORMANCE FOR DIFFERENT CLEARANCES:
```

| Rad Clear. (mu-in) | Load (Ibf) | Pwr Loss (hp) | Oil Flow (Gpm) | Oil Temp (F) | hmin (mu-in) | Ecc. (-) | Confdnc (-) |
|---|---|---|---|---|---|---|---|
| 2666.67 | 500.0 | .181 | .077 | 98 | 687.67 | .748 | Good |
| 2133.33 | 500.0 | .181 | .058 | 103 | 671.69 | .695 | Good |
| 1706.67 | 500.0 | .178 | .044 | 109 | 635.74 | .644 | Good |
| 1365.33 | 500.0 | .172 | .033 | 117 | 586.85 | .598 | Good |
| 1092.27 | 500.0 | .165 | .026 | 125 | 532.61 | .557 | Good |
| 873.81 | 500.0 | .156 | .020 | 134 | 478.89 | .522 | Good |
| 699.05 | 500.0 | .147 | .016 | 143 | 429.45 | .492 | Good |
| 559.24 | 500.0 | .138 | .013 | 153 | 385.95 | .467 | Good |
| 447.39 | 500.0 | .130 | .011 | 162 | 349.27 | .447 | Good |
| 357.91 | 500.0 | .123 | .009 | 171 | 318.88 | .429 | Good |

```
Minimum recommended Film thickness   :  210.0     (mu-in)
Minimum recommended Radial Clearance : 1051.5     (mu-in)
```

Figure 5.5. Calculation Screen.

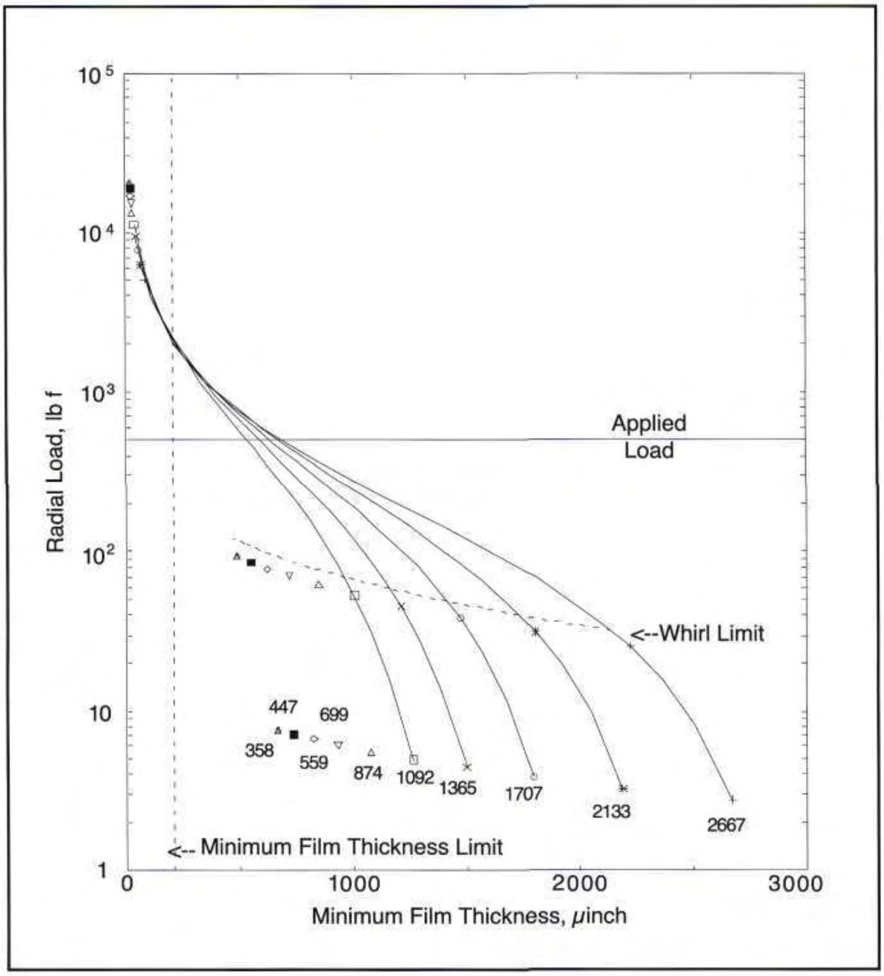


**Figure 5.6. Diagram of Bearing Operating Characteristics Showing Minimum Film Thickness Limit. (Data points are clearances, µinches)**

## Clearance

The dotted lines in **Figure 5.7** are radial clearances that may be too small for safe operation. Selecting the optimum clearance for a journal bearing requires consideration of several options. Referring to **Figure 5.7,** it can be seen that bearing stiffness (spring rate) increases with decreasing clearance.

Keep in mind, manufacturing practices limit the precision possible in producing a bearing. This includes bearing I.D. and journal O.D. tolerances, geometry (out of round condition), misalignment, and thermal gradients existing during start-up.

The latter condition, thermal gradient, becomes quite significant as bearing sizes and speeds increase. During start-up, the frictional heating causes the journal to expand while the bearing may expand inward until the bearing housing equilibrates in temperature. Thus the clearance closes down during this initial phase of operation. If the clearance is too small, the bearing clamps down on the journal and seizure occurs.

Therefore clearance must be large enough to get through this initial critical start-up period. In addition, one can determine the effect of manufacturing tolerances on bearing performance by determining minimum film thicknesses for the maximum and minimum clearance resulting from dimensional tolerances.

A good approach to clearance selection involves picking the smallest clearance on the basis of the above considerations so that as the bearing wears during its life (during start-up and shutdown), the increase in clearance will stay at the lower end of the recommended values. So one might select a radial clearance of 1092 (µinch (28 µm) from **Figures 5.6** and **5.7.**

If one should choose to use the lower clearance values included as dotted lines in the figure, one must make sure to prescribe the needed precision in manufacture and the fine surface finish values for reliable operation. The recommended clearance range in the program output is based on several investigators findings, from journal bearing tests. See Fuller[24].

## Air-Cooled Design

If the bearing under consideration is expected to operate under limited lubrication flow (such as drip feed, vapor lubrication, or hydrodynamic grease lubrication), the design process should be modified for air-cooling only. To change the program to air-cooling mode, press P and answer the instruction by typing "off, then press Q and enter "on". When the input table shows "Heat removal by lube... off" and "Heat removal by amb. air...on", press K to input the cooling air velocity selected for your design.

The output shown below will have different values from the output for oil-cooled design. Generally, when air-cooling is used, bearing load capacity will be reduced as compared with oil-cooling mode. To illustrate the difference in bearing operating characteristics for air-cooled and oil-cooled operation, two bearing designs will be examined. The graphic results for the two conditions are shown in **Figures 5.4** and **.7.**

## Applying the Hydrodynamic Program Results

The bearing design and lubricant selection is aimed at long life and reliable operation. These concerns are of great importance when reviewing the program output. Let us refer to the graphics, **Figures 5.6** and **5.7. Figure 5.2,** showing minimum film thickness curves for ten clearances and a large load range, is probably the most useful chart.

First of all, where the load line crosses the recommended minimum film thickness line, it can be seen that the design is within safe operating limits for all clearances shown.

Looking at the family of curves, it can be seen that as clearance increases, the slope of the load-film thickness curve decreases. This means that the smaller the clearance, the stiffer the bearing, *i.e.,* if the load increases, the journal center will move a smaller amount away from the bearing center. In addition, the smaller the clearance, the closer to the recommended minimum film thickness the bearing will operate. Owing to precision limits in manufacturing, a minimum clearance should be specified. Also, a maximum clearance should also be specified to insure a reasonably stiff bearing.

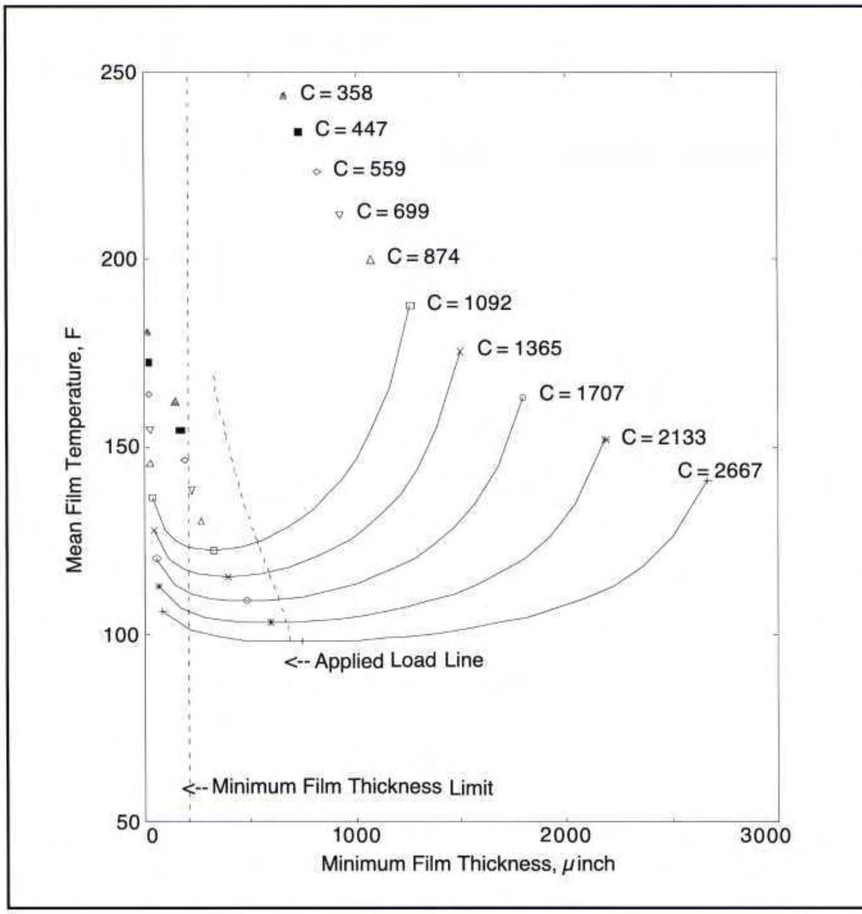


**Figure 5.7. Mean Film Temperature versus Minimum Film Thickness. (Data points are clearances, μinches)**

Referring to the output table, it is also evident that the larger clearances lead to higher oil flow and higher power losses in the bearing.

As bearings wear in service, the original clearance will open up. Therefore it is best to base the design on the minimum clearance or the left side of the "recommended design envelope" shown in **Figure 5.6.**

For the usual surface finish achieved in manufacturing, the clearance should be set at between 0.10% and 0.15% of the journal diameter. This limit is shown in **Figure 5.6.** Thus the "safe" operating limits of the bearing are contained within the box outlined in **Figure 5.6.** An operating bearing may be subjected to periods of load changes, increased temperatures and speed changes. The design must be able to operate within that set of operating conditions and still be within the safe operating limits.

Referring to **Figure 5.7,** note that with increasing clearance, the mean film temperature of the bearing decreases. Heat is detrimental to oil since extended exposure to elevated temperatures causes deterioration. Therefore, it is wise to try to set the design for minimum operating temperature. Higher clearance limits might thus be advisable in a design that appears to involve elevated temperatures.

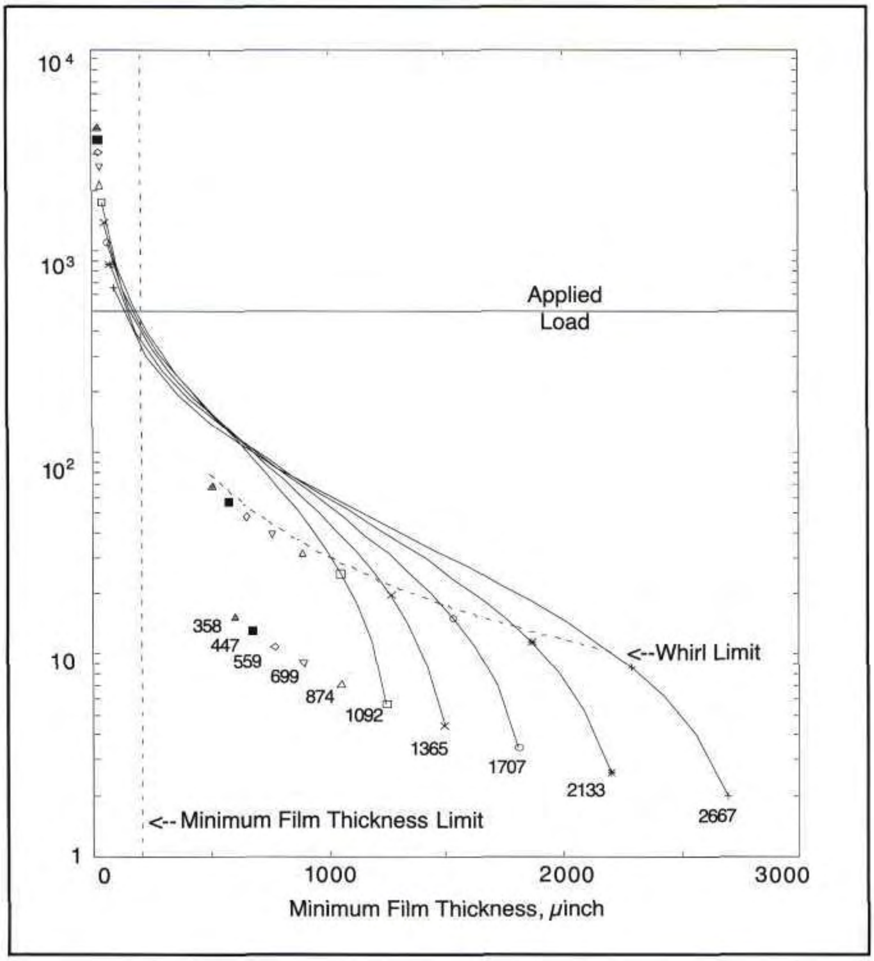


**Figure 5.8. Operating Characteristics for Air-Cooled Conditions. (Data points are clearances, μinches)**

## Using the Boundary Lubrication Program

NOTE: This program is based on the use of grease as the lubricant.

The **Bound** program is basically for slow moving, heavily loaded grease-lubricated bronze bearings. To initiate the program, insert the CDA floppy disk and, depending on the computer system you are using, follow the instructions in **Figure 5.10.**

**Macintosh**
**On the desktop:**
Double click on floppy disk icon
Double click on file **"BOUND"**

**IBM PC DOS**
**At the DOS prompt:**
Type **"BOUND"**
Press RETURN/ENTER

**Figure 5.10. Opening the Program.**

The message shown in **Figure 5.11** should appear:

Answer the query. If you know of a file on the current disk that relates to this program enter the file name. If you wish to start afresh, answer **N**(o) and press return. The input table in **Figure 5.12** of the appears on the opposite page.

## Entering Bearing Data

The above table shown in **Figure 5.12** allows you to input bearing load, speed, size, ambient temperature, bearing thermal properties and bearing friction coefficient. The table appears with the values inserted from a default set of the last run. The program assumes that frictional heat is dissipated by convection, conduction and radiation. Other cooling accepted is forced air moved over the bearing housing. The program allows one to input cooling air velocity.

## Friction

The default friction coefficient is 0.10. This is a maximum level determined in the bearing performance experiments. It is also considered a likely starting value for newly installed bearings. As the bearings wear in, the friction level can decrease. The lower end of the friction range is assumed to be 0.05. Friction coefficient levels tend toward the maximum at high-loads, low-velocity conditions.

## Units

The input can be in IPS or SI units. Inputting **U** will toggle from one units system to the other.

```
*********************************************************************
         CDA BOUNDARY LUBRICATED BEARING PERFORMANCE PROGRAM
Rev=  6.2  Date 930725  Time 16:06:22  Project:   None     Units: IPS
*********************************************************************

Do you want to get an input file from disc? Y/N (N)
```

**Figure 5.11. Opening Screen.**

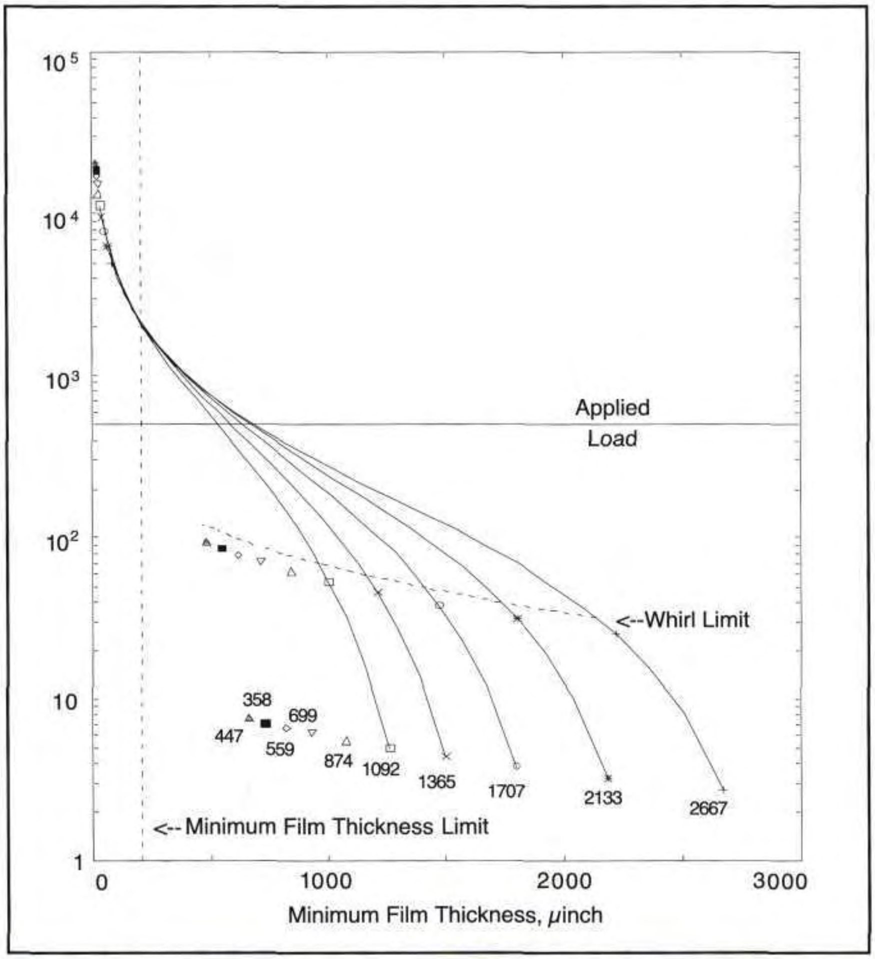


**Figure 5.9. Operating Characteristics for Oil-Cooled Operation. (Data points are clearances, μinches)**

The unit system can be changed during data input. For instance if you have input load in $lb_f$, speed in rpm and bearing size in inches and you have thermal conductivity data in W/m °C, press **U** and **RETURN** and all entries in the input table will be changed to SI units. You may then enter your conductivity value. If you want your results in IPS, toggle your table back to IPS by pressing **U** and **Return.**

## Calculating Bearing Performance

When the correct values have been entered into the table, the program can be asked to compute the estimated operating temperature and wear by answering Y(es) to the inquiry. In boundary lubricated bearings, some metal-to-metal contact occurs, and the bearing will wear continuously during its operating life. Heat developed from friction in the bearing is not dissipated by lubricant flow. Therefore, bearing temperature is the critical

```
*********************************************************************
         CDA BOUNDARY LUBRICATED BEARING PERFORMANCE PROGRAM
Rev= 6.2  Date 930725  Time 16:06:22  Project:  None   Units: IPS
*********************************************************************
ITEM #     NAME                              UNITS           VALUE
   A       Bearing Radial Load               (lbf)            225
   B       Shaft Speed                       (rpm)            300
   C       Required Operational Life         (hr)            1000
   D       Bearing Inside Diameter           (inch)             1.000
   E       Shaft Length                      (inch)             4.00
   F       Effective Bearing Width           (inch)             1.000
   G       Shaft Thermal Conductivity        (Btu/h ft F)      29.00
   H       Housing Outside Diameter          (inch)             3 .00
   I       Housing Width                     (inch)             1.00
   J       Housing Conductivity              (Btu/h ft F)      28.00
   K       Ambient Air Temperature           (F)               75
   L       Air Velocity                      (fpm)            300.0
   M       Bearing Friction Coefficient      (-)                 .10
   U       Change the Unit system
Enter Letter for Item # or Y(es) to accept ? ()
```

**Figure 5.12. Input Table.**

parameter for judging performance. The program assumes that a bearing temperature above 300 F (150 C) will cause unacceptable lubricant deterioration for conventional mineral oil-based greases.

The results shown in **Figure 5.13** will appear when the query is answered and **Return** is pressed. The output provides values for sliding velocity, bearing stress (load/projected area) and horsepower absorbed by the bearing. It also shows estimated bearing temperature with or without forced-air convection cooling. Finally it shows a table of estimated increase in diametral clearance caused by wear for four different grades of bronze bearing materials. The wear is related to the total number of hours of operational life. Note that the wear values differ among the four bronzes. This may influence your selection of material. For instance, the leaded bronze materials (C93200 and C94500) tend to operate at a lower friction coefficient than the non-leaded bronzes. Thus one might want to compromise on wear life for a lower operating temperature.

The default coefficient of friction (0.10) is conservative for leaded bronzes. Thus, if one selects one of the leaded bronzes, it would be appropriate to select a lower coefficient, say, 0.08. With that change in input and the program recalculated, the results shown in **Figure 5.13** are obtained.

Note that the estimated bearing temperature now is 43 F lower (289 F-246 F). This would insure longer lubricant life. The wear of leaded bronze remains the same. If the bearing temperature for natural convection exceeds 150 C (300 F) and the value for forced convection stays below that temperature, the following message will be added to the output:

```
  **********************************************************************
            CDA BOUNDARY LUBRICATED BEARING PERFORMANCE PROGRAM
  Rev=  6.2  Date 930725  Time 16:06:22  Project:   None    Units:  IPS
  **********************************************************************
  INPUT DATA:
    Bearing Radial Load                 (lbf)           225
    Shaft Speed                         (rpm)           300
    Required Operational Life           (hr)            1000
    Bearing Inside Diameter             (inch)             1.000
    Shaft Length                        (inch)             4.00
    Effective Bearing Width             (inch)             1.000
    Shaft Thermal Conductivity       (Btu/h ft F)         29.00
    Housing Outside Diameter            (inch)             3.00
    Housing Width                       (inch)             1.00
    Housing Conductivity             (Btu/h ft F)         28.00
    Ambient Air Temperature             (F)               75
    Air Velocity                        (fpm)           300.0
    Bearing Friction Coefficient        (-)                .10
  CALCULATED DATA:
      Sliding Speed  =     78.54 (fpm)
      Bearing Stress =    225 (psi)
      Power Loss     =       .054 (hp)
     Bearing Temperature due to Natural Convection     289       (F)
     Bearing Temperature due to Forced  Convection     199       (F)
 PREDICTED DIAMETRAL WEAR FOR DIFFERENT MATERIALS:
                                   Diametral
           MATERIAL              Wear (inch)
     C93200 (Bronze 660)            .0458
     C90500 (Bronze 62)             .0114
     C95400 (AL. Bronze)            .0034
     C94500 (Bronze 520)            .0567
```

**Figure 5.13. Results Screen.**

*"The bearing temperature exceeds the recommended value for use with simple natural convection cooling. You must use the forced convection cooling."*

If both the natural convection and forced convection cooling results in temperatures above 150 C (300 F), the following message will appear:

*"The bearing temperature exceeds the recommended value. It can be reduced by increasing the forced convection cooling."*

Of course, other means can be taken to reduce the operating temperature. One can increase the housing surface area (increase the housing diameter) and/or the bearing length. Bearing load or bearing rpm can be reduced also, if the design permits.

```
**********************************************************************
         CDA BOUNDARY LUBRICATED BEARING PERFORMANCE PROGRAM

Rev=  6.2  Date 930725  Time 17:30:41  Project:   None     Units: IPS
**********************************************************************

INPUT DATA:

  Bearing Radial Load                   (Ibf)          225
  Shaft Speed                           (rpm)          300
  Required Operational Life              (hr)         1000
  Bearing Inside Diameter               (inch)           1.000
  Shaft Length                          (inch)           4.00
  Effective Bearing Width               (inch)           1.000
  Shaft Thermal Conductivity         (Btu/h ft F)       29.00
  Housing Outside Diameter              (inch)           3.00
  Housing Width                         (inch)           1.00
  Housing Conductivity               (Btu/h ft F)       28.00
  Ambient Air Temperature                (F)           75
  Air Velocity                          (fpm)          300.0
  Bearing Friction Coefficient           (-)              .08

CALCULATED DATA:

    Sliding Speed  =     78.54 (fpm)
    Bearing Stress =    225 (psi)
    Power Loss     =       .043 (hp)

    Bearing Temperature due to Natural Convection     246       (F)
    Bearing Temperature due to Forced  Convection     174       (F)

PREDICTED DIAMETRAL WEAR FOR DIFFERENT MATERIALS:

                               Diametral
            MATERIAL          Wear (inch)

     C93200 (Bronze 660)         .0458
     C90500 (Bronze 62)          .0114
     C95400 (AL. Bronze)         .0034
     C94500 (Bronze 520)         .0567
```

**Figure 5.14. Results Screen.**

# References and Appendices

# REFERENCES AND APPENDICES

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# Appendix A. Quantifying Lubricated Wear

Wear in the presence of a lubricant occurs commonly in mechanisms and is a primary cause of deteriorated performance and failure. The extent of wear occurring depends upon the lubrication regime encountered. In some applications that are specifically designed to develop a hydrodynamic film, the lubricated wear phase is short and transitional during running-in and during start-stop cycles only. When conditions of full hydrodynamic film generation cannot be established, it is the main mode of lubrication. It should be pointed out here that even subtle surface features such as machining waviness can contribute significant hydrodynamic action in what appears to be an otherwise non-hydrodynamic application.

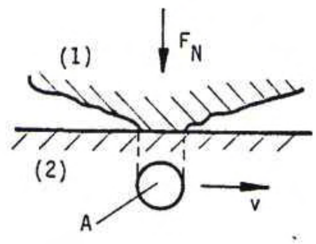
The extremely complicated nature of wear virtually precludes accurate quantitative relationships to predict wear. However, for engineering purposes, relationships exist for assessing the general wear regime and the likely effect of changing one of the variables on the resulting wear rate.

## Archard-Holm Equation

From very simple basic arguments, it may be expected that the removal of material between two surfaces in contact is proportional to their real area of contact. Also material removal may be expected to be proportional to the total distance through which the surfaces move. Thus it is perfectly reasonable to expect that wear is:

$$V = \alpha L A_r$$

where,
$V$ = volume of material worn
$A_r$ = real area of contact
$L$ = total sliding distance
$\alpha$ = thermal expansion coefficient




It is also well known that the real area of contact between two surfaces is directly proportional to the applied load. This is regardless of whether the asperity contact was elastic or plastic. If the contacts are predominantly plastic then we find that the real area of contact is inversely proportional to the hardness of the softer of the two bodies in contact. Material removal from the surface is thus expected to be proportional to the applied load, and inversely proportional to hardness. Hence the relationship between wear volume V and distance L, load $F_n$, and hardness H is expected to be:

$$V = \alpha \frac{L F_n}{H}$$

where,
$V$ = wear volume
$F_n$ = normal load
$L$ = sliding distance
$H$ = hardness

If the contact conditions at the asperities stays the same while material is removed, then we may introduce a constant of proportionality K, and the wear equation now reads:

$$V = K \frac{L F_n}{H}$$

where,
$K$ = wear coefficient

The above wear equation was developed by Archard[12], drawing on Holm's[25] theory of wear. The constant, K, is often called the wear coefficient. It is extremely important to realize the nature of the assumptions that have gone into the derivation of this equation. If we in anyway alter the contact conditions during asperity interactions, we generally expect that to alter the removal rate of material. Hence the wear coefficient is very dependent on the particular conditions that prevail at the asperities. Of particular importance are the following:

- Adhesion conditions of the materials. This in turn depends on:
  - Local temperatures (dependent upon ambient temperature, friction and sliding velocity)
  - Local cleanliness conditions
  - The solubilities of the two materials
- Asperity contact conditions:
  - Load and geometry result in elastic or plastic contact stress
- What portion of the applied load is carried by the asperities.

Hence the above wear equation with a constant, K, is only valid over a range of loads and distance when asperity conditions stay the same. Experience verifies this.

The lubricated wear data obtained by Glaeser" was observed to behave according to the Archard model. The experimental wear coefficients are as indicated in **Figure 3.1.** Because the bearings were grease-lubricated, the wear mechanism remained the same during the tests, with the exception of the initial wear due to running-in.

The experimental data fitted to this model is used in the **Bound** program.

## Other Forms of the Archard Equation

It is often convenient to write the wear equation in a somewhat different form. Sometimes we prefer to express wear as a linear dimension change. For example, we may be more interested in a change of bearing clearance over time. Such an expression may be derived from the above, if we consider the wear volume V to be the product of the apparent area of contact A and the depth of wear. This allows us to derive an expression for the depth of wear simply by dividing by the apparent area of contact, or:

$$\frac{V}{A} = d = K\frac{L}{H}\frac{F_n}{A} = K\frac{L\,P}{H}$$

where,
A = apparent area of contact
d = depth of wear
P = nominal contact pressure bearing stress

$$\left(\frac{F_n}{A}\right)$$

The nominal contact pressure in a bearing is thus simply the radial load divided by the projected bearing area. This is taken to be D*B.

To calculate the time t to reach a critical wear depth d*, when sliding takes place at constant velocity U, can be done by letting L = U t. Rearranging gives:

$$t = \frac{d^* H}{KPU}$$

where,
U = constant sliding velocity,
d* = critical depth of wear

We may also calculate the wear depth when the bearing operates at constant speed for a given time t as:

$$d = K\frac{P\,U\,t}{H}$$

## Lubricated Wear under Variable Loading Conditions.

In many applications, the loading on a bearing may be variable, and the speed may change, or the direction of rotation may even reverse. To predict wear in such instances, we need to apply cumulative damage models. Fortunately this is fairly simple for bearing wear calculations, provided that the wear mode stays the same over the life prediction period. With grease- and oil-lubricated cast bronze bearings, this is expected to be the case.

To apply cumulative damage methods for lubricated wear calculations, we simply make a summation of the product of load and sliding distance over the prediction lifetime. Hence the wear depth equation becomes:

$$d = \frac{K}{H}\int_0^L |P\,dx| = \frac{K}{H}\int_0^T |P\,U|dt$$

We have here used the absolute value of the product of pressure and velocity because wear is essentially path dependent. For situations where the load and sliding velocities are essentially constant over long periods of time, and where a distinct finite number of these constant usage periods exists, we may write:

$$d = \frac{K}{H}\left[|P_1U_1|t_1 + |P_2U_2|t_2 + |P_3U_3|t_3 + |P_4U_4|t_4\right] = \frac{K}{H}\sum_{i=1}^{n}|P_iU_i|t_i$$

The cumulative damage concept may also be used to calculate an equivalent wear period for an application with variable loading and speed. The time to reach the same wear depth may be found from manipulation of the above equation as:

$$\bar{P}\bar{U}t_e = \sum_{i=1}^{n}|P_iU_i|t_i$$

$$\text{or} \quad t_e = \frac{\sum_{i=1}^{n}|P_iU_i|t_i}{\bar{P}\bar{U}}$$

where,

$$\bar{P} = \frac{1}{T}\int_0^T |P|dt$$

is average pressure, and

$$\bar{U} = \frac{1}{T}\int_0^T |U|dt$$

is the average speed.

This simple model will be very useful when using the **Bound** program for applications with variable loading.

## Appendix B. Thermal Modeling of the Bearing

In the performance analysis of journal bearings, the operating temperature plays an important role. For the boundary lubrication mechanism, the peak temperature at the interface is the key to determine the life and suitability of the construction. For hydrodynamically lubricated bearings, it is the strong temperature dependence of viscosity for lubricating fluids that interacts with the load capacity. Hence it is essential that reasonably accurate prediction of the bearing temperature be performed. This prediction, however, has to be in keeping with the requirements of the solution.

Computer programs have been developed which are capable of obtaining a thermo-hydrodynamic solution to the energy and Reynold's equation for a given bearing configuration. The difficulty which often remains here is what boundary values to use for the various component temperatures. Thermal analysis of the entire bearing construction for different boundary conditions is then required.

An alternate solution, and the one used here, is one whereby a reasonably simple empirical model is used for the heat loss from a bearing configuration, and then to use the resulting temperatures as a way to calculate an effective viscosity in the bearing. This method essentially is a film energy balance, whereby the frictional heat generated is carried away by conductive and convective mechanisms. The simplicity of the model, however, necessitates some restrictions in the applicability of the model. These restrictions are explained in the next section.

### General Heat Transfer Model

A general heat transfer model of the bearing is shown in **Figure B.1.** This model allows for the generation of heat within the bearing, and for dissipation of this heat by convective, conductive and radiative heat transfer. This model is essentially implemented in both programs with the following restrictions:

1. Shaft diameters are the same on both sides and are the same as the bearing inside diameter.
2. Both sides of the shaft have an equal length of exposure.
3. The ambient air temperature is the same for the shaft and the bearing.
4. The effective length of the shaft is at such a location where the shaft temperature is ambient. This means there is no heat flux in the shaft at that location.
5. The radiative emmissivity coefficients for the shaft and bearing housing have been fixed at 0.8 and 0.7, respectively.
6. The model assumes that the shaft is solid.

With these assumptions and restrictions in mind, the program calculates the heat transfer from the bearing by conductive, convective and radiative mechanisms. All of this heat has to be absorbed by the ambient air. Additional heat transfer from the bearing is allowed for by the convective mechanism in the lubricant flow.

The important dimensions required by the program for the heat transfer calculations are shown in **Figure B.2.**

### Control of Heat Transfer Mechanisms

In designing a bearing application it is often required that the bearing performance be checked

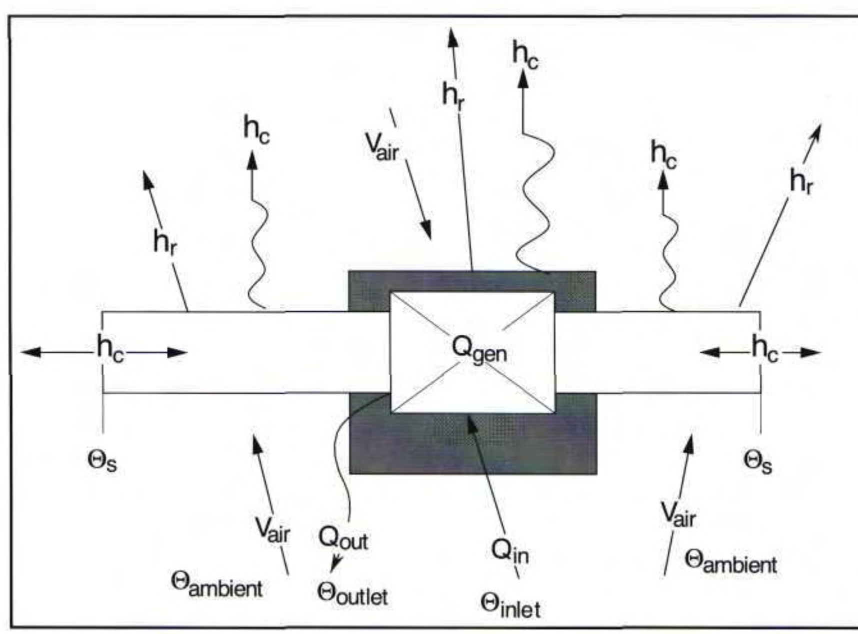



**Figure B.1. General Model of the Various Heat Transfer Modes from a Journal Bearing.**

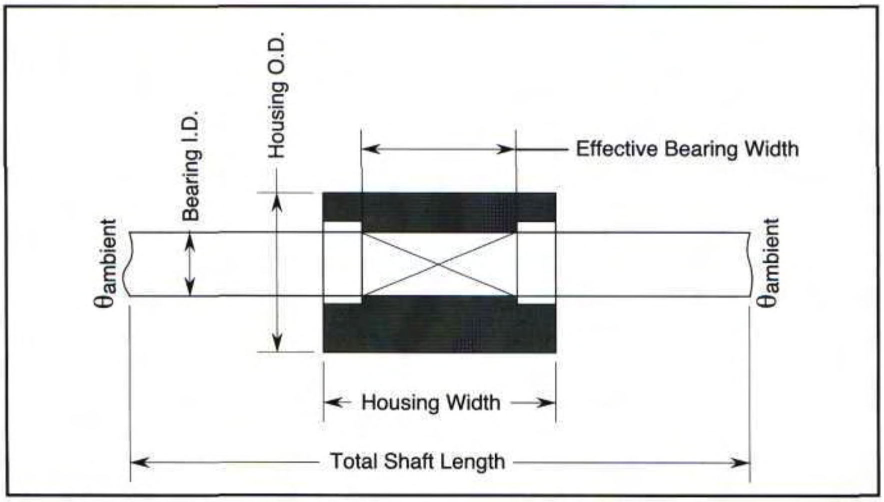



**Figure B.2. Definition of Bearing Dimensions for the Heat Transfer Model.**

under a number of different conditions. This is especially important for bearings where the heat transfer conditions are not well known. The bearing design programs **Hydro** and **Bound** have a number of controls that can be exercised by the user to determine their influence.
These are:

■ **Heat removed by the ambient air.** For **Hydro,** this may be turned "On" or "Off." When turned "Off," only convection by the lubricant is permitted. For **Bound,** it cannot be changed.

■ **Heat removed by the lubricant.** Can be switched "On" or "Off " in **Hydro.** When it is switched "Off," it simulates the heat transfer from a grease-lubricated bearing operating in the hydrodynamic mode.

■ **Control of the air velocity.** Heat transfer from the bearing and shafts is based upon both natural and forced-air convection. By changing the "air velocity," we can alter the degree of forced convection. When the "air velocity" is set to 0, only natural convection and radiative heat transfer from the housing take place. The heat transfer from the shaft is, however, based on the forced convection mode due to shaft rotation.

Because bearings must have a mechanism whereby viscous/frictional heat is removed, a number of safeguards have been implemented. Thus it is not possible to set both "heat removal by lubricant" and ambient air to "Off." The programs will reset the value for "heat removal by lubricant" to "On." For the **Bound** program, the heat transfer mechanism by the lubricant does not exist and is thus not provided as an option.

It is recommended that the user implement a cautious approach to the heat management of bearings, *i.e.,* always estimate on the conservative side.

## Appendix C. Fluid Viscosity Database

The **Hydro** program as supplied comes with a limited fluids database. The current fluids in the database are:

- ISO VG32
- SAE 10W
- ISO VG46
- SAE 10
- SAE 20
- ISO VG68
- SAE 30
- ISO VG100
- SAE 40
- ISO VG150
- SAE 50
- ISO VG220
- SAE 60
- Di-2 Ethylhexyl Sebacate
- TetraChloroDiphenyl
- Polydimethyl Siloxane
- Glycerol
- Tolulene
- Castor Oil
- Methanol
- Ethanol
- Water

The database contains the temperature-related properties of these fluids. Because the standards for both the ISO VG and the SAE grades **DO NOT** specify the temperature viscosity behavior of these specific grades, certain assumptions were made in selecting data for these lubricants. The user should be aware of and consider the following:

- The default data for mineral oil-based lubricant is based upon high viscosity index (HVI) lubricants;
- There may be a significant variation in viscosity-temperature behavior from one nominally-the-same lubricant to another;
- It is always best to enter your own measured viscosity data for the specific lubricant you intend to use;
- It is imperative that the user verifies the assumed temperature-viscosity and the actual temperature viscosity behavior of the fluid he selects.

### Adding Additional Fluids to the Database

Additional fluids may be added to the database by using any word processing program. The data are contained in an ASCII text file called LUBE.TRU. This file must be called up as an ASCII text file and saved as an ASCII text file. Be aware that some word processing programs add special characters to a file if you do not save it as ASCII. The data essential to the hydrodynamic program are the following:

- Fluid name,
- 3 temperature-viscosity points,
- Fluid density data point, and
- Fluid specific heat data point.

This information needs to be entered on a single line in comma separated value (CSV) format. A typical entry, for example, for Glycerol is given below. Details of each entry are given next.

Glycerol, 0,12.1, 100,0.013, 200, 0.00022, 1260, 2350

- Fluid name. Any name up to 15 characters long that describes the fluid. (Glycerol);
- Temperature-viscosity points: These points are used to interpolate the fluid viscosity at different temperature forms. The values must be given in the Temp 1, Vis 1, Temp 2, Vis 2, Temp 3, Vis 3 order. Try to use the maximum spread in temperatures for best results. It is also recommended that you enter the data in increasing temperature order. The temperatures need to be expressed in °C, and the viscosity needs to be expressed in the dynamic units of Pa·s (a section dealing with viscosity conversion factors is given later);

| Temperature, °C | Dynamic Viscosity, Pa·s |
|---|---|
| 0 | 12.1 |
| 100 | 0.013 |
| 200 | 0.00022 |

- Fluid density data point. One data point on the fluid density in kg/ms at 20°C is required. (1260kg/m$^3$);
- Specific heat capacity data point. One data point for the specific heat capacity at constant volume ($C_v$) in Joule/kg•°K at 293 K is required. (2350 Joule/kg•°K).

### Viscosity Conversions

Traditionally viscosity was measured by noting the time required for a given amount of fluid to be drained by gravity through a small orifice. The Saybolt Universal Seconds (SUS), the Redwood Seconds (RS), and the Degree Engler (E) are all based on this method. The information thus obtained is a measure of the kinematic viscosity. This measured time in seconds may be converted to proper kinematic viscosity by the following formula due to Herschel[26]:

$$\nu = At - \frac{B}{t}$$

where,

n = kinematic viscosity in centistokes (cSt)

t = measured runout time.

The constants A and B depend on the type of instrument used and are given below.

| Type of Viscometer | A | B |
|---|---|---|
| Saybolt (SUS) | 0.22 | 180 |
| Redwood (RS) | 0.26 | 171 |
| Engler (E) | 0.147 | 374 |

The units of kinematic viscosity in SI units are $m^2/s$. Formerly the kinematic viscosity was quoted in Stokes or Centistokes. Conversion factors from other units of kinematic viscosity are:

| Kinematic Viscosity Unit | | Conversion Factor |
|---|---|---|
| 1 Stoke (St) | = | $10^{-4}$ $m^2/s$ |
| 1 Centistoke (cSt) | = | $10^{-6}$ $m^2/s$ |
| 1 square foot per hour | = | $0.258 \times 10^{-4}$ $m^2/s$ |
| 1 square foot per second | = | $929.03 \times 10^{-4}$ $m^2/s$ |
| 1 square inch per hour | = | $0.179 \times 10^{-6}$ $m^2/s$ |
| 1 square inch per second | = | $6.452 \times 10^{4}$ $m^2/s$ |

Temperature changes affect the oil kinematic viscosity in two ways: by changing the flow resistance or dynamic viscosity and by changing the density. For hydrodynamic lubrication, we only require the change in flow resistance and therefore the dynamic viscosity. In the SI system of units, the dynamic viscosity has units of:

$$\eta = \text{dynamic viscosity} = \frac{\text{shear stress}}{\text{shear rate}}, \ [\text{N/m}^2 \text{ sec or Pa·s}]$$

The dynamic viscosity of a lubricant is obtained by multiplying the kinematic viscosity by the density of that fluid. Note that the density needs to be expressed in $kg/m^3$ and the kinematic viscosity in $m^2/s$ for this calculation. (Units are: $m^2\ s^{-1} \times kg\ m^{-3} = kg\ m^{-1}$ s-1 $= N\ s\ m^{-2} = Pa\cdot s$)

Conversion factors from other dynamic viscosity units are:

| Viscosity Unit | | Conversion Factor |
|---|---|---|
| 1 poise (P) | = | 0.1 Pa·s |
| 1 centipoise (cP) | = | 0.001 Pa·s |
| 1 $\text{pound}_{\text{force}}$ second per square inch (Reyn) | = | 6896 Pa·s |
| 1 $\text{pound}_{\text{force}}$ second per square foot | = | 47.88 Pa·s |
| 1 poundal second per square foot | = | 1.488 Pa·s |
| 1 pound per foot-second | = | 1.488 Pa·s |
| 1 slug per foot-second | = | 47.88 Pa·s |

**NOTES:**

CDA

Copper Development Association

Non-Ferrous Founders' Society

A1063-00/97