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Tribology – Friction, Wear, and Lubrication

Tribology is a relatively new term derived from the Greek word *tribos* for 'rubbing'. It is now universally applied to the emerging science of friction, wear, and lubrication involved at moving contacts. In its broad scope, it involves mechanical, chemical, and material technology. Usual tasks in tribology are reduction of friction and wear to conserve energy, enabling faster and more precise motions, increased productivity, and reduced maintenance.

Tribology was formally identified as an important and unified technical field in a report issued by a committee of the British Ministry of State for Education and Science chaired by Peter Jost (1966). The report concluded that huge savings would be possible in the United Kingdom by fully utilizing improved design and lubrication procedures. The unified approach to this field filled an existing void, and the American Society of Mechanical Engineers (ASME) then adopted the term for its Tribology Division in 1983 and the American Society of Lubrication Engineers revised its name to the Society of Tribologists and Lubrication Engineers in 1985.

Fundamental interest in tribology now exists for lubricant formulation, industrial operations, aerospace and transportation equipment, material shaping and machining, computers and electronic devices, power generation, and almost all phases of life where motion is encountered. This text focuses primarily on tribology of bearings and a number of related applications. Fundamentals are applied in such a way as to allow application of the principles of tribology to a variety of other machine elements.

1.1 History of tribology

Many of the advances in tribology and bearing technology evolved over years, decades, or even centuries to meet the needs of new machinery. The Industrial Revolution, with its increase in rotational speeds beyond those of the windmill and cart axle, brought full hydrodynamic lubrication into normal use. Theory and technical understanding commonly

Applied Tribology 2e: Bearing Design and Lubrication M. Khonsari and E.R. Booser

followed actual machinery applications. In many cases, this understanding of technical details played a vital role in continued improvements in bearing design, lubricants, and surface treatments for industrial machinery, aerospace units, transportation equipment, magnetic storage, and microelectromechanical devices.

A few historical stepping stones related to the primary subjects covered in this textbook will now be reviewed. Many of these events are covered in more detail in the references presented at the end of this chapter. The interested reader is referred to Dowson (1998) for an excellent review of the history of tribology.

FRICTION

Notebooks of the famed engineer and artist Leonardo da Vinci revealed his postulation in 1508 of the concept of a characteristic coefficient of friction as the ratio of friction force to normal load. The French physicist Guillaume Amontons (1699) again established the significance of a coefficient of friction, which was independent of the apparent area of contact. The French physicist C.A. Coulomb (1785) further distinguished between static friction and kinetic friction, which was independent of velocity. Mechanisms for reduction of friction and wear with soft coatings and adherent molecular and lubricant surface layers were elucidated by Bowden and Tabor (1950).

WEAR

This subject has proven to be quite complex, and generalizations are still elusive. Hundreds of empirical equations have been generated for specific materials and conditions. The most useful one appears to be that of Archard (1953), which enables a generalized dimensionless wear coefficient $k = VH/Wd$ to relate wear volume V to sliding distance d, normal load W, and indentation hardness H of the wearing material (see Chapter 4).

BEARING MATERIALS

For many centuries wood, stone, leather, iron, and copper were common bearing materials. Almost all engineering materials have now been employed in the continuing search for the best bearing material (see Chapter 4). In an early consideration of improved materials, Robert Hooke (1684) suggested steel shafts and bell-metal bushes as preferable to wood shod with iron for wheel bearings (Bhushan, 1999). High-lead and high-tin alloys patented in the United States in 1839 by Isaac Babbitt are unsurpassed for a wide range of industrial, automotive, and railway applications. An early German study of railway journal bearings by F.A. von Pauli (1849) established a composition similar to that of Babbitt (91% tin, 6% copper, and 3% zinc) as the best of 13 bearing metals (Cameron, 1966; 1976).

Suitably hard, fatigue-resistant rolling element bearing materials were achieved with modification of tool steel in Europe about 1900. This led to the development of AISI 52100 steel and its derivatives, which have since been used for all types of commercial and automotive rolling bearings.

Porous metal bearings were introduced in the 1920s. Plastics and composites involving polymers compounded with a wide variety of solid filler materials have found wide use, gaining their greatest impetus with the invention of nylon and polytetrafluoroethylene (PTFE) during World War II. The search continues, with ceramics among the materials being developed for high-temperature capability in aircraft engines and for high-speed rolling element bearings.

LUBRICANTS

Tallow was used to lubricate chariot wheels before 1400 BC. Although vegetable oils and greases were used later, significant advances in the development of lubricants occurred only after the founding of the modern petroleum industry with the opening of the Drake well in Titusville, Pennsylvania, in 1859. Lubricant production reached 9500 m^3/yr (2 500 000 gal/yr) in the following 20 years; worldwide production now exceeds 1000 times that volume at 3 billion gal/yr. Petroleum lubricants still constitute well over 95% of total oil and grease production volume.

Polymerized olefins were the first synthetic oils to be produced. This occurred in 1929 in an effort to improve on properties of petroleum oils. Interest in ester lubricants dates from 1937 in Germany, and their production and use expanded rapidly during and following World War II to meet military needs and for use in the newly developed jet aircraft engines. A broad range of other synthetic lubricants came into production during that same period for wide-temperature use, fire resistance, and other uses geared to a range of unique properties (see Chapter 2). Current production of synthetics approaches 100 million gal/yr, with nearly half being polyalphaolefin synthetic hydrocarbons.

Development of chemical additives to upgrade the properties and extend the lives of lubricating oils began about 1920. Commercial use has proceeded since about 1930 in step with increasing demands in automobiles, jet engines and other aerospace units, and high-speed and high-pressure hydraulic equipment (see Chapter 2).

Air, water, gasoline, solvents, refrigerant gases, air, and various fluids being processed in individual machines began to find use as 'lubricants' on a broadening scale in fluid-film bearings in the second half of the 1900s as improved designs and mating materials were developed on a customized basis.

FLUID-FILM BEARINGS

The first studies of a shaft and bearing running under full hydrodynamic conditions were performed by F.A. von Pauli in 1849 and by G.A. Hirn in 1854 (see Cameron, 1966). In 1883 the celebrated Russian Nikilay Petroff concluded that friction in fluid-film bearings was a *hydrodynamic* phenomenon. His resulting power loss equation has continued to provide a foundation in this field.

Beauchamp Tower in 1883 found experimentally and reported the generation of pressure in the oil film of a journal bearing. In considering Tower's findings, Osborne Reynolds (both working under stimulus of the British Institution of Mechanical Engineers) in 1886 developed a mathematical expression for this pressure buildup that has become the foundation of hydrodynamic analysis of bearing performance (see especially Chapters 5 and 6).

Solution of the Reynolds equation was difficult, and Arnold Sommerfeld in 1904 developed a direct integration that enabled 'infinite length' analyses. Alastair Cameron and

Mrs W.L. Wood in 1949 made an extremely useful extension of Reynolds equation for finite-length journal bearings by a relaxation procedure carried out with a mechanical desktop calculator. Initiated in 1958 by Oscar Pinkus and by Albert Raimondi and John Boyd, digital computer analysis of journal bearing performance has come into widespread use for obtaining numerical solutions of the Reynolds equation. Recent advances have focused on solving the Reynolds equation with consideration of cavitation. Particularly fruitful has been evaluation of dynamics effects in bearings that take into account the orbit of the shaft, along with simultaneous solutions of the energy equation taking lubricant property variations into consideration.

ROLLING ELEMENT BEARINGS

Several isolated examples have been recorded of application of rollers in 484–424 BC for transporting vessels on land and for launching military missiles. While historical sources increasingly mention the use of balls and rollers for bearing purposes following about AD 1500, widespread application of rolling contact bearings occurred during the twentieth century (see especially Allan, 1945).

1780 Perhaps the earliest ball bearing: a 610 mm bore bearing was developed for a rotating mill in Norwich, UK.
1868 General use of ball bearings in bicycles began.
1899 Timken began the manufacture of tapered roller bearings.
1902 Conrad obtained a British patent for the present design of the deep-groove ball bearing with its cage.
1904 Ball and roller bearings were used in electric automobiles.
1907 SKF founded.
1949 Grubin established the elastohydrodynamic principles involved in the lubrication of rolling contacts.

Recent research has been geared toward understanding the relationship between surface topography (Chapter 3), film thickness, and pressure distribution in rolling element bearings. Under highly loaded conditions, when surface asperities tend to interact, a mixed-film lubrication regime may have to be considered. Prediction of subsurface stress fields and their relationship to fatigue are also subjects of current research.

NANOTRIBOLOGY AND SURFACE EFFECTS

Atomic-scale studies have been expanding rapidly to develop new materials with improved tribological properties. With an atom being approximately 1/3 nanometer (nm) in diameter, Krim in 1991 initiated naming this field ‘nanotribology’ to parallel the broad field of ‘nanotechnology’ (Krim, 1991; Cantor, 2004). The nanometer (10^{-12} m, or 10 ångstroms) then provides a convenient scale for atomic and molecular sizes such as those in Table 1.1.

An early definition of atomic surface compositions that produces satisfactory sliding characteristics was provided in General Motors rubbing tests of all available metallic elements

Table 1.1 Representative nanometer dimensions in tribology

	Size range (nm)
Atomic scale[a]	
C–C bond length	0.15
C–H bond length	0.11
Lubricant molecules	
Lube oil	1–5
Absorbed rust inhibitors[a]	3
Viscosity index improvers (VIs)	100–250
Lithium soap grease fibers, diameter × length[b]	0.2 × 2
Computer magnetic heads[a]	
Air film	0.15–0.3
Liquid film	1–2

[a] *Source*: Bhushan (1997, 1999).
[b] *Source*: Klamann (1984).

on steel (Roach et al., 1956). As detailed in Chapter 4, good scoring resistance was found only with elements that had atomic diameters at least 15% greater than iron and that were in the B subgroup of the periodic table, implying lack of tenacious metallic bonds at any localized junctions with steel.

Bushan (1997, 1999) updated consideration of atomic structure and related surface and lubricant details for avoiding wear under demanding operation with 0.15–0.3 nm thick air films in magnetic storage and micromechanical devices such as those involved in computer systems. Analysis of related air film bearings is provided in Chapter 11.

1.2 Tribology principles

Several distinct regimes are commonly employed to describe the fundamental principles of tribology. These range from *dry sliding* to complete separation of two moving surfaces by *fluid-film lubrication*, with an intermediate range involving partial separation in *boundary* or *mixed* lubrication. When elastic surface deformation exerts a strong influence on fluid-film behavior, as in ball and roller bearings, *elastohydrodynamic lubrication (EHL)* introduces its distinctive characteristics.

DRY SLIDING

In the absence of a fluid film, actual contact between two rubbing surfaces involves only sufficient high spots, or asperities, of the softer material so that their yield pressure balances the total load (Rabinowicz, 1995). Under load W in Figure 1.1, the real contact area is a relatively minute portion of the apparent total area and tends to increase proportionately with the load (see Chapter 4).

A measurable force is required to slide a surface along on the contacting asperities (surface peaks) of another surface. This force must overcome the friction force associated with

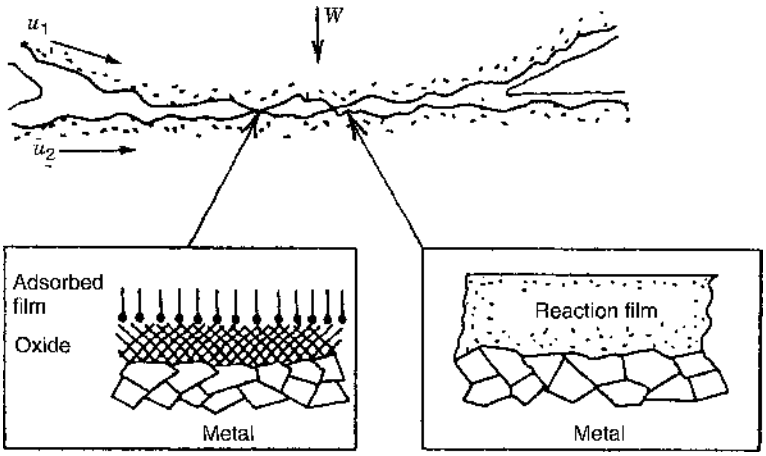

Figure 1.1 Surface films at asperity contacts between rubbing surfaces. (Harris, 1991. Copyright © 1991 by John Wiley & Sons, Inc. Reprinted by permission of John Wiley & Sons, Inc.)

asperity yield strength. As suggested in Figure 1.1, friction can be significantly reduced by a thin, soft coating of a solid film lubricant such as graphite, molybdenum disulfide, or PTFE plastic, or by a sulfur- or phosphorus-rich layer formed by adsorption of additives from a lubricating oil. Wear volume can be related to the generation of wear fragments from a portion of these asperity contacts during sliding (see Chapters 2–4).

FLUID-FILM LUBRICATION

In this regime, the moving surfaces are completely separated by a film of liquid or gaseous lubricant. A load-supporting pressure is commonly generated by one of the following types of action:

1. *Hydrodynamic lubrication (HL)* results from a film of separating fluid being drawn into a converging, wedge-shaped zone (as in Figure 1.2a) by the self-acting pumping action of a moving surface (Booser, 1995). Both the pressure and the frictional power loss in this film are a function of the lubricant's viscosity in combination with the geometry and shear rate imposed by bearing operating conditions (see Chapter 5). Hydrodynamic bearings are commonly in the form of either (a) a *sleeve bearing* (also called a *journal bearing*) surrounding its mating journal surface on the shaft of a machine as a radial bearing or (b) a *thrust bearing* to provide an oil film at the face of a shaft shoulder or collar for location and axial support of a rotor.
2. *The squeeze-film* action shown in Figure 1.2(b) is encountered in dynamically loaded bearings in reciprocating engines and under shock loads (see Chapter 9). Because time is required to squeeze the lubricant film out of a bearing, much higher loads can be carried in sleeve bearings for automotive engines and metal rolling mills than with a steady, unidirectional load, as reflected in the typical values presented in Table 1.2. The much

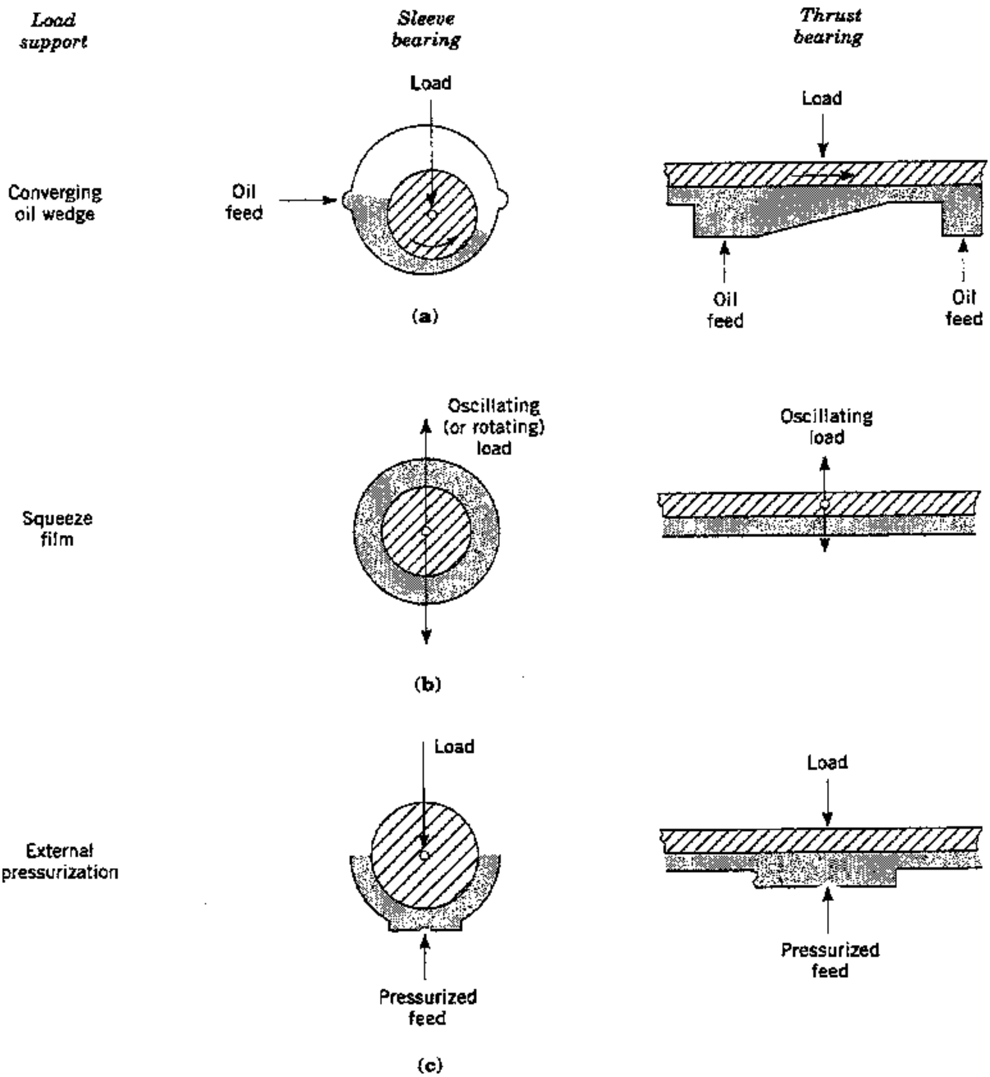

Figure 1.2 Principles of fluid-film action in bearings. (Booser, 1995. Copyright © 1995 by John Wiley & Sons, Inc. Reprinted by permission of John Wiley & Sons, Inc.)

lower load capacity of bearings lubricated with low-viscosity fluids such as water and air is also indicated in Table 1.2.

3. *Externally pressurized* feed by an external pumping source may be used to generate pressure in the fluid before its introduction into the bearing film, as shown in Figure 1.2(c). Such a procedure is common in cases where limited hydrodynamic pumping action is available within the bearing itself, as during starting and slow-speed running with heavy machines or with low-viscosity fluids (see Chapter 10).

Table 1.2 Typical design loads for hydrodynamic bearings

Bearing type	Load on projected area MPa (psi)
Oil lubricated	
Steady load	
Electric motors	1.4 (200)
Turbines	2.1 (300)
Railroad car axles	2.4 (350)
Dynamic loads	
Automobile engine main bearings	24 (3 500)
Automobile connecting-rod bearings	34 (5 000)
Steel mill roll necks	35 (5 000)
Water lubricated	0.2 (30)
Air bearings	0.2 (30)

Source: Khonsari and Booser (2004).

ELASTOHYDRODYNAMIC LUBRICATION (EHL)

This is a form of hydrodynamic lubrication in which pressures are large enough to cause significant elastic deformation of the lubricated surfaces. As with HL, converging film thickness, sliding motion, and fluid viscosity play an essential role. Temperature effects, inadequate lubricant supply to the EHL film, and boundary lubrication play roles of varying importance, much as with fluid films in HL.

Significant differences between HL and EHL involve the added importance of material hardness, viscosity increase under high pressure, and degree of geometric conformation of the contacting surfaces. *Conformal surfaces* match snugly, such as the journal in a sleeve bearing with hydrodynamic lubrication shown in Figure 1.2, so that the load is carried on a relatively large area. With *nonconformal surfaces*, as with the two contacting rollers in Figure 1.3, the load must be carried on a small area: typically of the order of 1000-fold smaller than with a conformal conjunction. The following two distinct regimes exist in EHL:

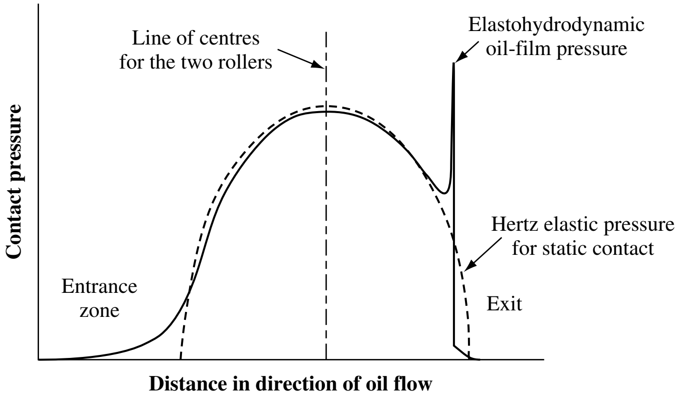

Figure 1.3 Pressure distribution between two rollers under load

1. *Hard EHL.* This involves materials of high elastic modulus in *nonconformal* contacts, commonly contacts involving primarily rolling action or combined rolling and sliding in ball and roller bearings, gear teeth, cams, and some friction drives. Since the surfaces do not conform well, the load is concentrated on small, elastically deformed areas, and hydrodynamic lubrication of these EHL contacts is commonly characterized by a very thin separating oil film that supports local stresses that tax the fatigue strength of the strongest steels.

   Oil-film pressures in these EHL contacts commonly range up to 0.5–3 GPa, 1000 times that in most hydrodynamic bearings, with a pattern which closely follows the Hertz elastic contact stress for a static contact, as illustrated in Figure 1.3. The overall oil-film thickness (often about 0.1–0.5 μm) is set primarily by oil viscosity, film shape, and velocity at the entry of the contact zone (see Chapter 15).
2. *Soft EHL.* Elastic deformation also plays an important role in film formation in rubber seals, tires, human joints, water-lubricated rubber bearings, and babbitt journal bearings under heavy loads and low speeds, as well as in similar applications of hydrodynamic bearings using soft bearing materials having low elastic modulus. Since the low contact pressures involved have a negligible effect on fluid viscosity in the conjunction, analytical relations are simpler for *soft EHL* in contrast to the *hard EHL* encountered with rolling element bearings. Hamrock (1994) and Khonsari and Hua (1997) give analytical tools for performance analysis within the various realms of EHL.

BOUNDARY LUBRICATION

As the severity of operation increases, the speed N and viscosity μ eventually become incapable of generating sufficient oil-film pressure P to support the entire load. Asperities of the mating surfaces then penetrate with increasing contact area, plastic deformation, higher asperity temperatures, and finally surface tearing and seizure on a broad scale. The region of lubrication in Figure 1.4 shifts from full film with a friction coefficient (ratio of friction

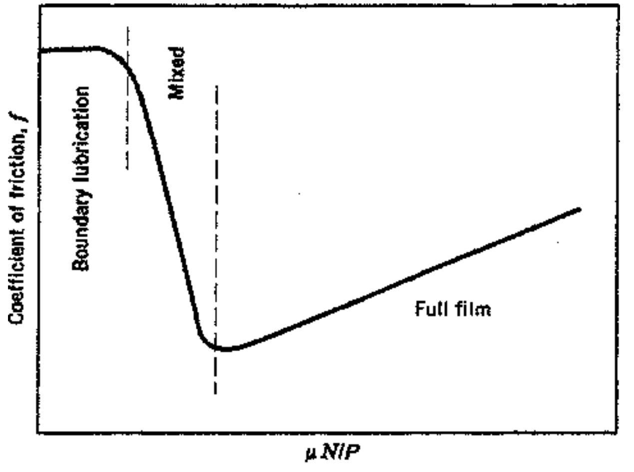

Figure 1.4 Stribeck dimensionless $\mu N/P$ curve relating lubrication regime and friction coefficient to absolute viscosity μ, rotational speed N, and unit load P. (Booser, 1995. Copyright © 1995 by John Wiley & Sons, Inc. Reprinted by permission of John Wiley & Sons, Inc.)

force to normal force) of the order of 0.001 to *boundary* (or *mixed-film*) lubrication, where the friction coefficient rises to 0.03–0.1, and finally to complete loss of film support, where the friction coefficient may reach 0.2–0.4, typical for dry sliding.

When operating with an adsorbed surface layer or a chemical reaction coating in boundary lubrication, chemical additives in the oil and chemical, metallurgical, and mechanical factors involving the two rubbing surfaces determine the extent of wear and the degree of friction.

1.3 Principles for selection of bearing types

Most bearings can be classified as either fluid-film bearings, dry or semilubricated sliding bearings, or rolling element bearings (Khonsari and Booser, 2004). Their relative advantages are listed in Table 1.3. A general preliminary guide to selection of each of these types for different load, speed, and size relations is provided in Figure 1.5.

Example 1.1 A radial load of 500 N (112 lb) is to be carried by a bearing on a 25 mm (0.98 in.) diameter shaft. Which bearing types could be considered for speeds of 100, 1000, 10 000, and 100 000 rpm?

Table 1.3 Characteristics of common classes of bearings

	Fluid film bearings	Dry bearings	Semilubricated	Rolling element bearings
Start-up friction coefficient	0.25	0.15	0.10	0.002
Running friction coefficient	0.001	0.10	0.05	0.001
Velocity limit	High	Low	Low	Medium
Load limit	High	Low	Low	High
Life limit	Unlimited	Wear	Wear	Fatigue
Lubrication requirements	High	None	Low/none	Low
High-temperature limit	Lubricant	Material	Lubricant	Lubricant
Low-temperature limit	Lubricant	None	None	Lubricant
Vacuum	Not applicable	Good	Lubricant	Lubricant
Damping capacity	High	Low	Low	Low
Noise	Low	Medium	Medium	High
Dirt/dust	Need seals	Good	Fair	Need seals
Radial space requirement	Small	Small	Small	Large
Cost	High	Low	Low	Medium

Source: Kennedy *et al*. (1998).

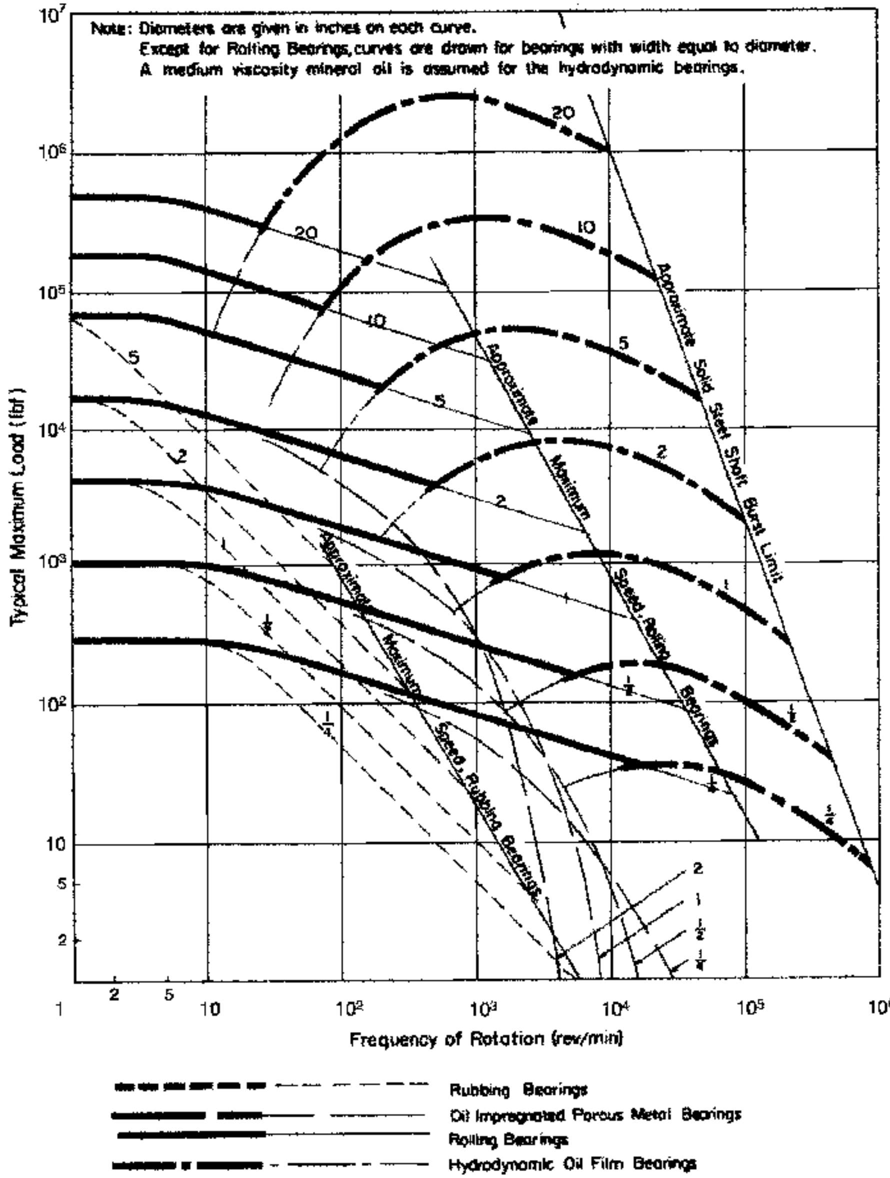

Figure 1.5 General guide for selecting the journal bearing type. Except for rolling element bearings, the length/diameter ratio = 1. Medium-viscosity mineral oil is assumed for hydrodynamic bearings. (From ESDU Data Item No. 65007, 1965)

Follow horizontally across Figure 1.5 for a ‘Typical maximum load’ of 500 N. Speed and load limits are then found to be adequate for the following bearing types:

Speed (rpm)	Possible bearing types
100	Rubbing, porous metal, rolling element, oil film
1 000	Porous metal, rolling element, oil film
10 000	Rolling element, oil film
100 000	Oil film

While the Engineering Sciences Data Unit (ESDU) also has a companion chart for thrust bearings, the same general type used for the journal bearing should be considered first.

Final selection of the most suitable basic type is then related to the following factors:

1. Mechanical requirements;
2. Environmental conditions;
3. Economics.

Representative considerations follow for each of these factors, and later chapters give further guidelines for performance to be expected with individual bearing types. In many cases, preliminary pursuit of design details for several possibilities is helpful in making a final selection.

MECHANICAL REQUIREMENTS

Unless the bearing fulfills all mechanical requirements imposed by the machine of which it is a part, other considerations such as environment and cost are of no importance. Brief comparisons of these items are included in Table 1.3.

Friction and power loss

Low starting friction, especially under a load, is a prime advantage of ball and roller bearings. While involving added complexity, externally pressurized oil lift pockets provide oil-film bearings with zero starting friction in a variety of large, heavy machines such as electric generators at hydroelectric dams and in utility power plants. For running machines, a coefficient of friction of the order of 0.001–0.002 is typical for both rolling-element and oil-film bearings. Start-up coefficients at breakaway of 0.15–0.25 are typical for oil-film bearings and for dry and semilubricated plastic and porous metal bearings. With dry and semilubricated surfaces, friction then drops by about half as motion gets underway.

The localized temperature increase generated by friction in rubbing asperities was analyzed by Blok (1937) to place a limit on the maximum speed and load which can be tolerated in bearing and gear contacts.

Speed

Each of the four classes of bearings presented in Table 1.3 has a practical speed limit. Common practice limits rolling bearings using oil lubrication to a (*DN*) value (mm bore × rpm) of 500 000 to 1 000 000, corresponding to a surface speed of 1600–3100 m/min (5000–10 000 ft/min). Limits with fluid-film bearings are much higher and less well defined, but surface speeds in turbine bearings range up to about 8700 m/min (30 000 ft/min). A typical dental drill uses air for journal bearing lubrication while spinning smoothly at speeds of 300 000 rpm. Much lower speed limits in the range of 100–500 m/min are imposed by surface heating effects with dry and semilubricated bearings, for which the appropriate operating zone is illustrated in Figure 1.6. (Note: Figure 1.6 does not show scales on its axes and is only meant to show the trend. See Chapter 12 for typical numerical values.)

Load

Rolling element bearings are generally more versatile in being able to carry their fatigue-rated load at all speeds from zero up and in all directions with normal lubrication. The load capacity of oil-film bearings, on the other hand, is very much a function of speed and oil viscosity, with their influence on oil-film formation. Dry and semilubricated porous metal and plastic bearings encounter a surface heating limit in their *PV* factor (contact pressure × surface velocity) which gives a much lower load limit with rising speed, but high loads are possible with appropriate material combinations at low speeds.

Hydrostatic bearings using externally pressurized oil have been used to support enormous structures such as observatory domes, telescopes, and large radio antennas where weight requirements range from 250 000 to over 1 000 000 pounds (see Chapter 10).

Momentary shock loads can be reasonably tolerated by both fluid film and rolling element bearings. Rotor unbalance loads and cyclic loads in internal combustion engines are well carried by oil-film bearings. Combined radial and thrust load capacity is a useful attribute of conventional deep-groove, single-row ball bearings.

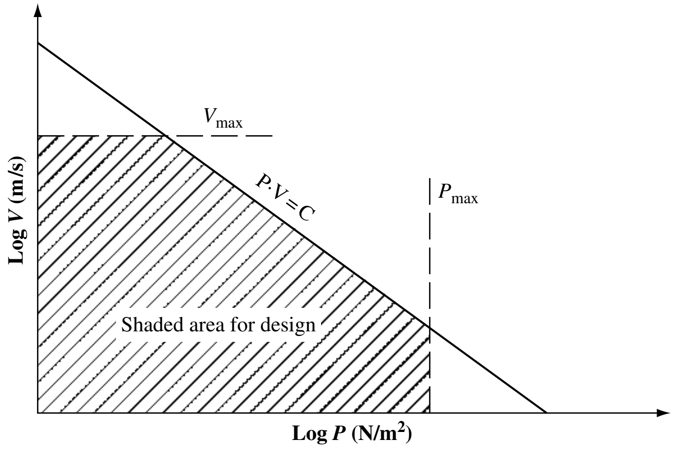

Figure 1.6 Operating zone for the design of dry and semilubricated bearings. (Fuller, 1984. Copyright © 1984 by John Wiley & Sons, Inc. Reprinted by permission of John Wiley & Sons, Inc.)

Life

Rolling element bearings have a distinct fatigue life limit which results from the repeated contact stressing of the balls and rollers on their raceways, while fluid-film bearings in the usual rotating equipment provide essentially unlimited life. The same life pattern has been found with rolling element and oil-film bearings, however, in industrial electric motors, where maintenance needs and contamination commonly dictate life. Fatigue life is also a limiting factor for oil-film bearings under cyclic loading in internal combustion engines.

In dry and semilubricated bearings, life is estimated from a *wear coefficient*, which relates wear rate to unit loading and peripheral sliding distance. In the rapidly expanding use of this class of bearings, life is also related to temperature, contamination, and other environmental conditions.

Lubrication

In general, a rolling element bearing requires only enough lubricant to provide a film coating over the surface roughness of working surfaces. Less than one drop supplies this need for many small and medium-sized ball and roller bearings. Under conditions of heavy load and high speed, additional lubricant must be supplied, not for lubrication needs but to remove heat and maintain a reasonable limit on temperature. Many rolling element bearings depend only on an initial grease fill for years of operation when DN values are less than 300 000 (mm bore $\times$ rpm).

While no oil is usually fed to small sliding-type bearings which are to operate dry or semilubricated, oil-film bearings generally require relatively large quantities of oil to maintain the separating film between the bearing and its mating surface. This feed rate is proportional to the bearing length, width, clearance, and surface velocity and ranges up to $4\,\mathrm{m}^3/\mathrm{min}$ (1000 gal/min) for the oil-film bearings in a steam turbine generator at an electric power station. Specially designed fluid-film bearings are unique in being able to operate with gas or ambient liquids such as water or gasoline as their hydrodynamic fluid.

Space requirement

Dry and semilubricated bearings require a minimum of space. A porous metal, plastic, or plastic-lined bearing is commonly a bushing or thrust washer, such as that illustrated in Figure 1.7. These bearings have just enough wall thickness to provide the needed strength for insertion in a supporting housing. The bearing can even consist of no more than a formed or machined hole in a suitable plastic housing of an appliance or instrument.

In the case of ball and roller bearings, the outside diameter commonly ranges from about one and a half to three times that of the bore, and the axial dimension ranges from one-fifth to one-half of the shaft diameter. Oil-film journal bearings are more compact in their radial dimension, and range in axial length from about one-third to equal to the shaft diameter. Considerable additional volume is commonly required with oil-film bearings to accommodate seals plus feed and drain passages. Figure 1.8 illustrates the general proportions for a babbitt journal bearing combined with an integral thrust face with its oil-distributing grooves.

Figure 1.7 Representative plastic composite bushings and thrust washers. (Courtesy of SKF)

Figure 1.8 Oil-film bearing combining babbitted journal and thrust surfaces. (Courtesy of Kingsbury, Inc.)

ENVIRONMENTAL CONDITIONS

Temperature range, moisture, dirt, and corrosive atmospheres are among the important environmental factors that require consideration. Within each bearing type, established design methodology is now available for meeting most of the usual environmental demands.

Fluid-film bearings generally are most limited in their temperature range. Common soft metal bearing surfaces of tin and lead babbitt limit their upper temperature to the 125–150 °C (260–300 °F) range. Resistance to flow from the high viscosity of cold mineral oil in feed and drain passages, at the other extreme, limits the low temperature range of many oil-film bearings to -20 to $+10\,°C$ (-5 to $50\,°F$).

Ball and roller bearings can be adapted to meet wide-ranging environmental demands. For general applications, multipurpose greases and double-sealed and double-shielded enclosures provide water and contamination resistance, rust protection, and sufficiently long life to eliminate the need for routine maintenance. Synthetic oils and greases used with tool steel and ceramic bearing materials allow applications in an ever-broadening range of temperatures and other environmental conditions.

ECONOMICS

Overall economic considerations in selecting a bearing type include not only the first cost of the bearing itself, but also later costs in maintenance and eventually in replacement at the end of the useful life.

Cost

Where other factors do not dictate the choice, initial cost commonly is a dominant selection factor. For many mass-produced items such as small electric motors, household appliances, and automotive auxiliaries with low surface speeds and light loads, dry and semilubricated bushings and thrust washers have by far the lowest cost and are finding continually broadening use in bore sizes up to about 25 mm. Mass-produced oil-film bearings of smaller sizes also fall in this low-cost range for automobile engines, appliances, and fractional-horsepower motors. In general, they avoid the speed limits for porous metals and polymer compositions.

If these sliding element bearings appear marginal or impractical, ball and roller bearings should be considered as the next higher-priced alternative. While available from very small diameters in the 3–5 mm range, their use is common for 25–100 mm shaft diameters, and their use in larger diameters is growing. Chapters 13–15 provide application, lubrication, and performance guidelines for their use over a broad range of conditions.

More expensive oil-film bearings are common in larger sizes for high loads and high speeds, such as those used with auxiliary oil feed systems involving cooling and filtration of contaminants.

Maintenance and life

Fluid-film bearings operating with parts separated by an oil film require little or no maintenance and their life is virtually unlimited, except under cyclic loading, where the bearing metal may suffer fatigue. In contrast, the repeated cyclic loading on the contacts in rolling element bearings results in limited life due to fatigue.

Maintenance of greased rolling bearings may at most require occasional purging or grease change. With oil lubrication of either sliding or rolling bearings, the supply system requires occasional attention for a change of oil, a new filter, or general cleaning.

With dry and semilubricated bearings, calculated bearing wear life is commonly selected to at least match the expected life of the overall unit. Required maintenance is usually limited to the replacement of worn bearings.

1.4 Modernization of existing applications

Reassessing the bearing class and bearing design is a common need for existing machines or products. This need may arise from problems with the present bearings, from extension of the product line, or in updating a machine design.

Industrial electric motors provide an example of modernization over an extended time. The earliest electric motors around 1900 used fluid-film bearings with an oil ring for a lubricant supply (see Chapter 12). With increasing availability of ball bearings by the time of World War I, their small size and reduced maintenance needs allowed their use in fractional horsepower motors and in motors for electric automobiles and small mine locomotives. In the 1920s, ball bearing use was extended to motor sizes up to about 200 horsepower (hp), but only at a premium price and with concerns over reliability. With lower bearing costs and the availability of long-life greases, ball bearings became standard for industrial motors up to 25–50 hp in the 1950s. Today, following subsequent redesigns, very few motors using oil-film bearings are available in sizes below about 1000 hp.

On the other hand, bearings continue to be of the oil-film type in stationary steam and gas turbines, particularly those in electric power service. The value of their essentially unlimited life far surpasses the higher cost of these bearings and their extended lubrication system compared with rolling element bearings. The trend toward lightweight and high-speed construction also depends to a large extent on the inherent damping and higher speed limit available with oil-film bearings.

For jet aircraft turbine engines, however, rolling element bearing use is firmly established. Dominant factors are a small lubricant supply, easy low-temperature starting, high thrust load capacity, and light weight. Moreover, ball bearings are better suited to these applications, where there may be an instantaneous interruption in the oil supply. This is an important consideration since aircraft experience a variety of operating conditions including takeoff, landing, and maneuvering, which can be conveniently handled by ball bearings.

With turbines, as with many other machines, selection of the bearing type will continue to be a question. Oil-film bearings may be considered for land-based derivatives of aircraft turbines, and ball and roller bearings will probably continue to be evaluated for small land-based turbines, compressors, and their accessory units, where oil-film bearings have been traditional.

Automotive chassis bearings have undergone a wide range of modifications to eliminate the need for relubrication. Automobile engine bearings will likely also undergo detailed evaluation of their type and materials with the advent of new electric propulsion and fuel cell components in the drive system. Such factors and innovations such as ceramic rolling elements, new plastic and composite sliding bearing materials, and new high-performance lubricants will call for continuing reevaluation of almost all bearing design and performance details.

1.5 A look ahead

Coming trends for bearing applications will likely be set by the needs for lower maintenance while operating at higher speeds and higher temperatures in more compact designs. The following are likely trends for the future of classification of bearings covered in this chapter (Khonsari and Booser, 2004).

DRY AND SEMILUBRICATED BEARINGS

Plastics and their composites will continue to find broadening use for mild operating conditions as small bushings in household appliances, machine tools, instruments, business machines, automobile chassis and construction equipment. Many of these applications will involve direct integration of the bearings with housing and structural elements.

BALL AND ROLLER BEARINGS

While continuing their dominance in jet engines and aerospace, rolling element bearing use will broaden in small electric motors, automobile accessories, machine tools, construction and agriculture machines, and railroad equipment. Their very small lubrication needs will likely bring innovations in greases, self-contained lubricant impregnation in ball and roller bearing cages, and solid lubricant films. Further developments of ceramics, tool steels, and special lubricants will enable continually higher speeds and temperatures.

FLUID-FILM BEARINGS

Conventional oil-lubricated bearings will continue to fill their traditional role in reciprocating automotive and diesel engines, electrical turbine generators, and other large machinery. Use of gases and low-viscosity liquids will likely escalate. Air-film bearings now used in flying heads on computer discs and in airliner cabin compressors will gradually expand in their use for aerospace, industrial, and instrument units. Water, gasoline, liquid ammonia, and a wide variety of liquid chemical and food process streams will likely be employed in bearings to avoid the complexity and contamination with conventional mineral oils and greases.

These are but a sampling of challenges coming in the twenty-first century: new bearing materials, new lubricants, advanced analysis and designs techniques, and improved surface profiles to match extremely thin fluid films.

Problems

1.1 Select a bearing type for applications with a 2.0 in. diameter shaft, medium-viscosity mineral oil lubricant, and the following specifications:

   (a) 10 rpm, 500 lbf load;
   (b) 1000 rpm, 500 lbf load;
   (c) 10 000 rpm, 500 lbf load;
   (d) 1000 rpm, 10 000 lbf load.

1.2 List three general advantages and disadvantages to be expected in applying water-lubricated bearings rather than oil-film or rolling element bearings. Give five applications in which you conclude that water bearings would be advantageous.

1.3 Repeat Problem 1.2 for air bearings.

1.4 A 30 mm bore tapered roller bearing is being considered for use in supporting the impeller in a household food-disposal unit for kitchen sinks. High-shock loading is expected, as solids and possibly tableware enter the unit. Suggest two other possible types of bearings and compare their major attributes, as selected from items in Table 1.3. Which type of bearing is your primary recommendation?

1.5 A 5000 rpm gas turbine with a 6 in. shaft diameter at the bearing locations is to be used in driving a natural gas line pumping unit. List the major factors that might be expected to dictate the selection of rolling or fluid-film bearings. Should magnetic bearings be considered that would suspend and center the shaft while using an electronic control for the magnet elements?

1.6 State two types of bearings or bearing systems that you would consider for a conveyor carrying 10 lb castings in 20 min passage at a speed of 2 m/min through a 350 °C heat-treating oven. List five selection factors to be considered. Which factor appears to be most critical?

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