Applied Tribology

Bearing Design and Lubrication

Second Edition

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Applied Tribology

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Dedicated to:
Karen, Maxwell, Milton and Mason Khonsari and Katherine Booser

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Series Preface

The first edition of this book was published in 2001. The book has quickly become a highly successful and popular text amongst the students, academics as well as amongst the practicing engineers. The readers valued this book as a good teaching material and a useful reference. It is thus of no surprise that it enjoys a five star ranking on the Amazon book list. The second edition is an extended and updated version of the original. The new edition follows the same format as the first edition and covers topics such as tribology in bearings followed by the demonstrations of the applications of the same basic principles to other machine components such as piston pins, piston rings, seals, viscous pumps, viscous dampers, hydraulic lifts, wet clutches and brakes. New additions include two new chapters on dynamic seals and bearing failure modes. The chapter on bearing failure modes is an especially welcomed addition as it completes the bearing's 'life cycle' from design to its failure. The practical aspects of bearing's failure are rather rarely discussed in the textbooks. Information provided in this chapter would be equally valuable for practicing engineers, designers, academics and the students. The text in the remaining chapters has been updated and is supported by numerous numerical examples. A manual with solutions adds to the attractiveness of this book as a teaching material in the classroom. Therefore this book is highly recommended as both the textbook to be used in teaching undergraduate and postgraduate courses in tribology and machine design as well as the preferred reference book for practicing engineers and designers.

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Preface

Tribology is a diverse field of science involving lubrication, friction, and wear. In addition to covering tribology as involved in bearings, the same basic principles are also demonstrated here for other machine elements such as piston rings, magnetic disk drives, viscous pumps, seals, hydraulic lifts, and wet clutches.

In this second edition of *Applied Tribology* all the chapters were updated to reflect recent developments in the field. In addition, this edition contains two new chapters: one on the fundamentals of seals and the other on monitoring machine behavior and lubricants, as well as bearing failure analysis. These topics are of considerable interest to the industry practitioners as well as students and should satisfy the needs of the tribology community at large.

Computer solutions for basic fluid film and energy relations from the first edition have been extended to seal performance. New developments in foil bearings are reviewed. For ball and roller bearings, conditions enabling infinite fatigue life are covered along with new ASME and bearing company life and friction factors.

Properties of both mineral and synthetic oils and greases are supplemented by an update on the greatly extended service life with new Group II and Group III severely hydrocracked mineral oils. Similar property and performance factors are given for full-film bearing alloys, for dry and partially lubricated bearings, and for fatigue-resistant materials for rolling element bearings. Gas properties and performance relations are also covered in a chapter on gas bearing applications for high-speed machines and for flying heads in computer read–write units operating in the nanotribology range.

Problems at the close of each chapter aid in adapting the book as a text for university and industrial courses. Many of these problems provide guidelines for solution of current design and application questions. Comprehensive lists of references have been brought up to date with 145 new entries for use in pursuing subjects to greater depth.

Both SI and traditional British inch-pound-second units are employed. Units in most common use are generally chosen for each section: international SI units for ball and roller bearing dimensions and for scientific and aerospace illustrations, traditional British units for oil-film bearings in industrial machinery. Many analyses are cast in dimensionless terms, enabling use of either system of units. Conversion factors are tabulated in two appendices, and guidelines for applying either system of units are given throughout the book.

The authors are indebted to past coworkers and students for their participation in developing topics and concepts presented in this book. Comprehensive background information has been assembled by the authors from their combined 80 years of laboratory, industrial, teaching, and consulting experience. Much of this background has been drawn from well over 250 technical publications, most of them in archival literature. Theses and dissertation projects at Ohio State University, the University of Pittsburgh, Southern Illinois University, Louisiana State University; and the Pennsylvania State University; numerous industrial projects at the Center for Rotating Machinery at Louisiana State University; four tribology handbooks organized and edited for the Society of Tribologists and Lubrication Engineers; and a 1957 book *Bearing Design and Application* coauthored with D. F. Wilcock.

This book is intended (1) for academic use in a one-semester senior engineering elective course, (2) for a graduate-level course in engineering tribology and (3) as a reference book for practicing engineers and machine designers. It is hoped that these uses will provide paths for effectively designing, applying, and lubricating bearings and other machine elements while taking advantage of concepts in tribology—the developing science of friction, wear, and lubrication.

MMK
ERB

Part I

General Considerations

1

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 desk-top 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

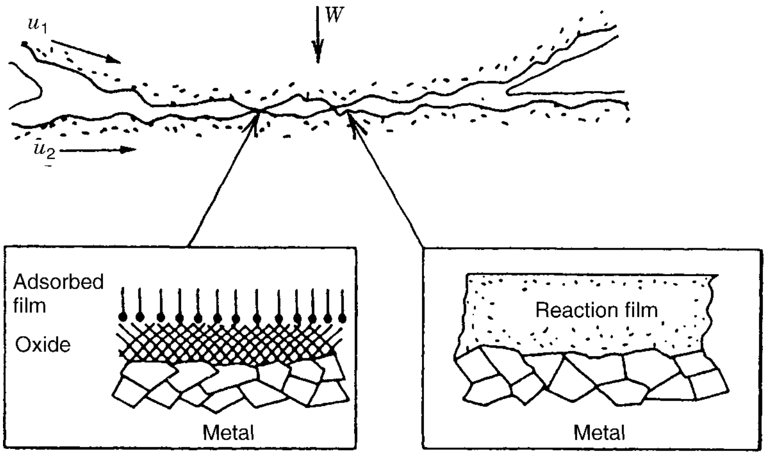

Figure 1.1 Surface films at asperity contacts between rubbing surfaces. (Harris, 1991. Copyright © 1991 by John Wiley & Sons, Inc. Reprinted by permission of John Wiley & Sons, Inc.)

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

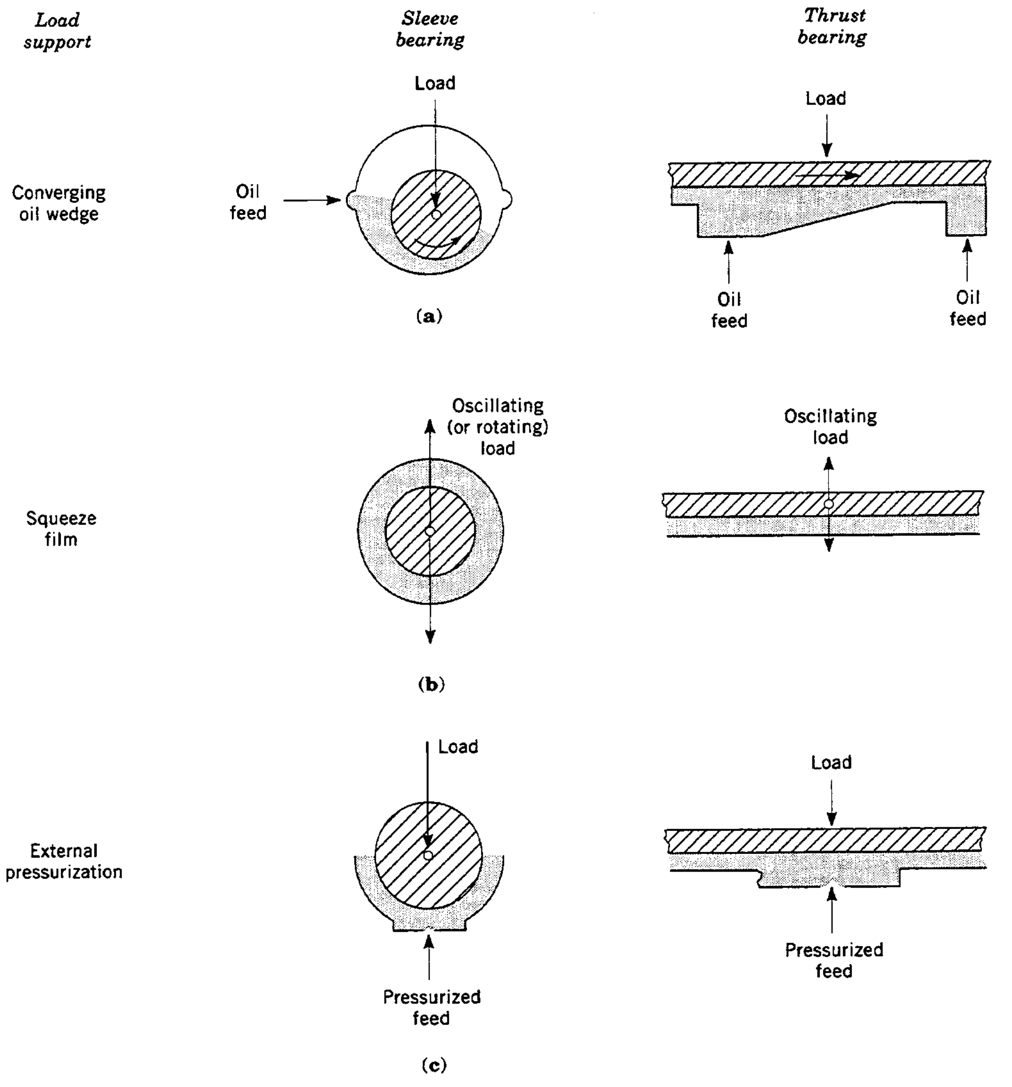

Figure 1.2 Principles of fluid-film action in bearings. (Booser, 1995. Copyright © 1995 by John Wiley & Sons, Inc. Reprinted by permission of John Wiley & Sons, Inc.)

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:

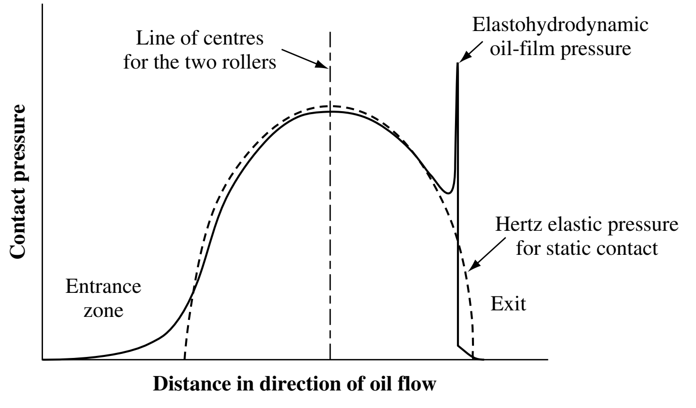

Figure 1.3 Pressure distribution between two rollers under load

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

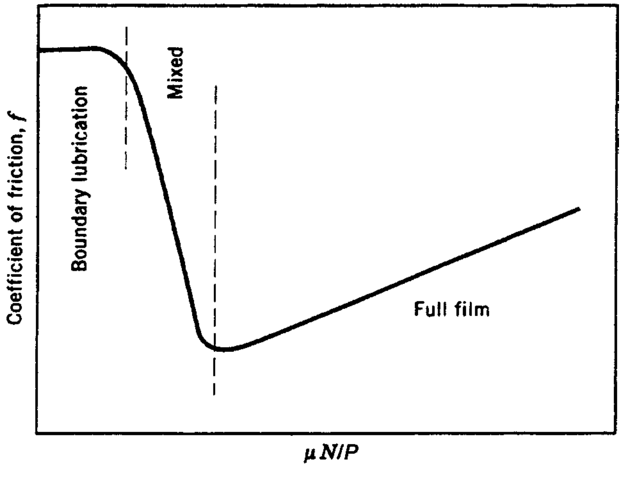

Figure 1.4 Stribeck dimensionless $\mu N/P$ curve relating lubrication regime and friction coefficient to absolute viscosity μ, rotational speed N, and unit load P. (Booser, 1995. Copyright © 1995 by John Wiley & Sons, Inc. Reprinted by permission of John Wiley & Sons, Inc.)

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).

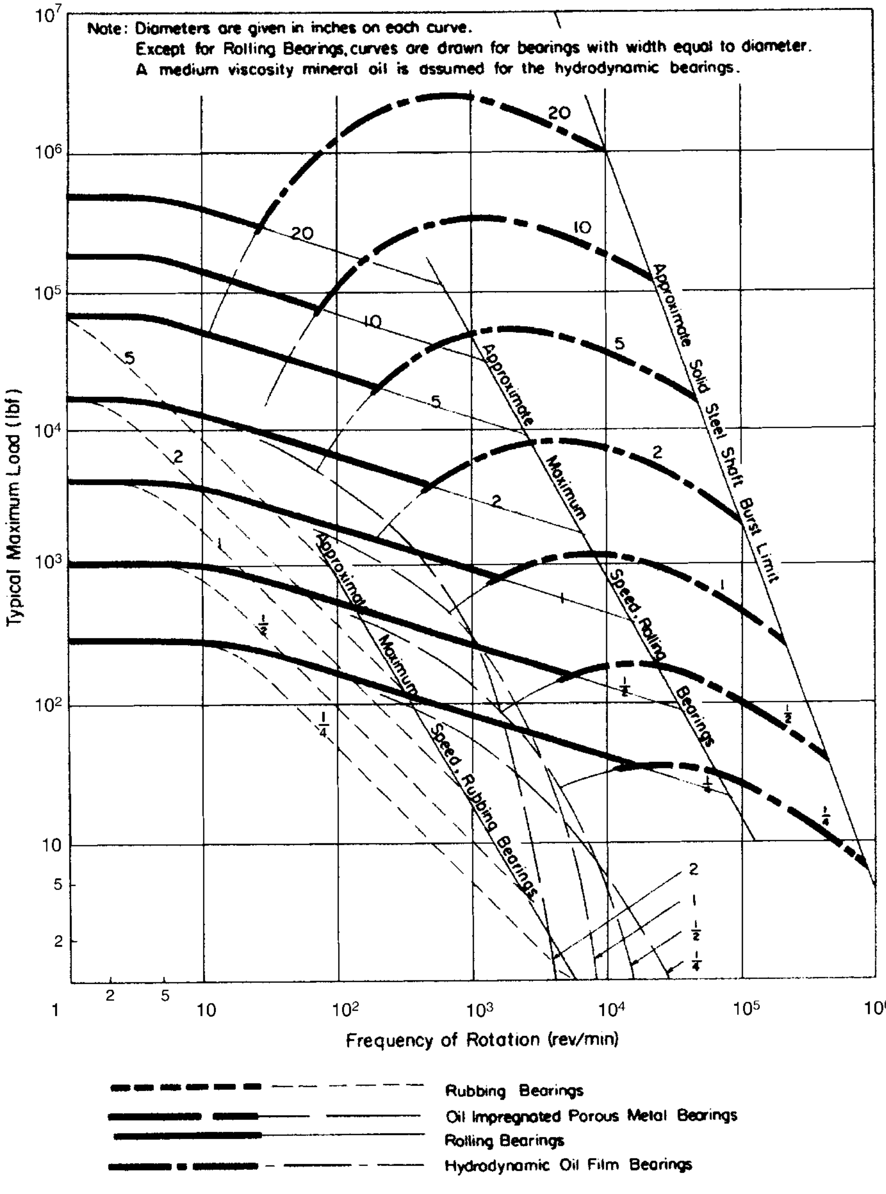

Figure 1.5 General guide for selecting the journal bearing type. Except for rolling element bearings, the length/diameter ratio = 1. Medium-viscosity mineral oil is assumed for hydrodynamic bearings. (From ESDU Data Item No. 65007, 1965)

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.

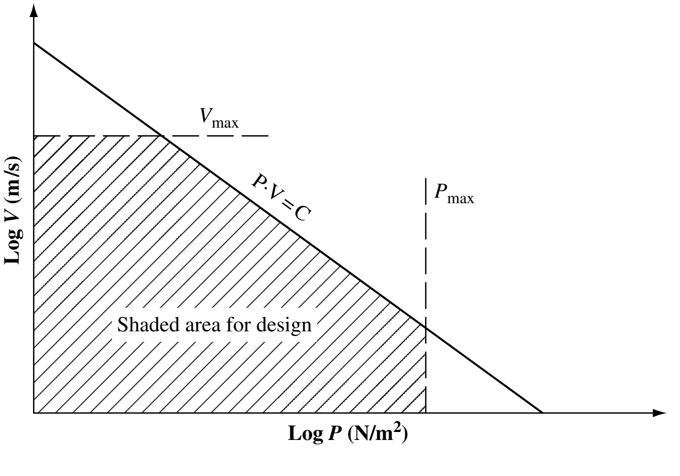

Figure 1.6 Operating zone for the design of dry and semilubricated bearings. (Fuller, 1984. Copyright © 1984 by John Wiley & Sons, Inc. Reprinted by permission of John Wiley & Sons, Inc.)

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 $4\,m^3/min$ (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.