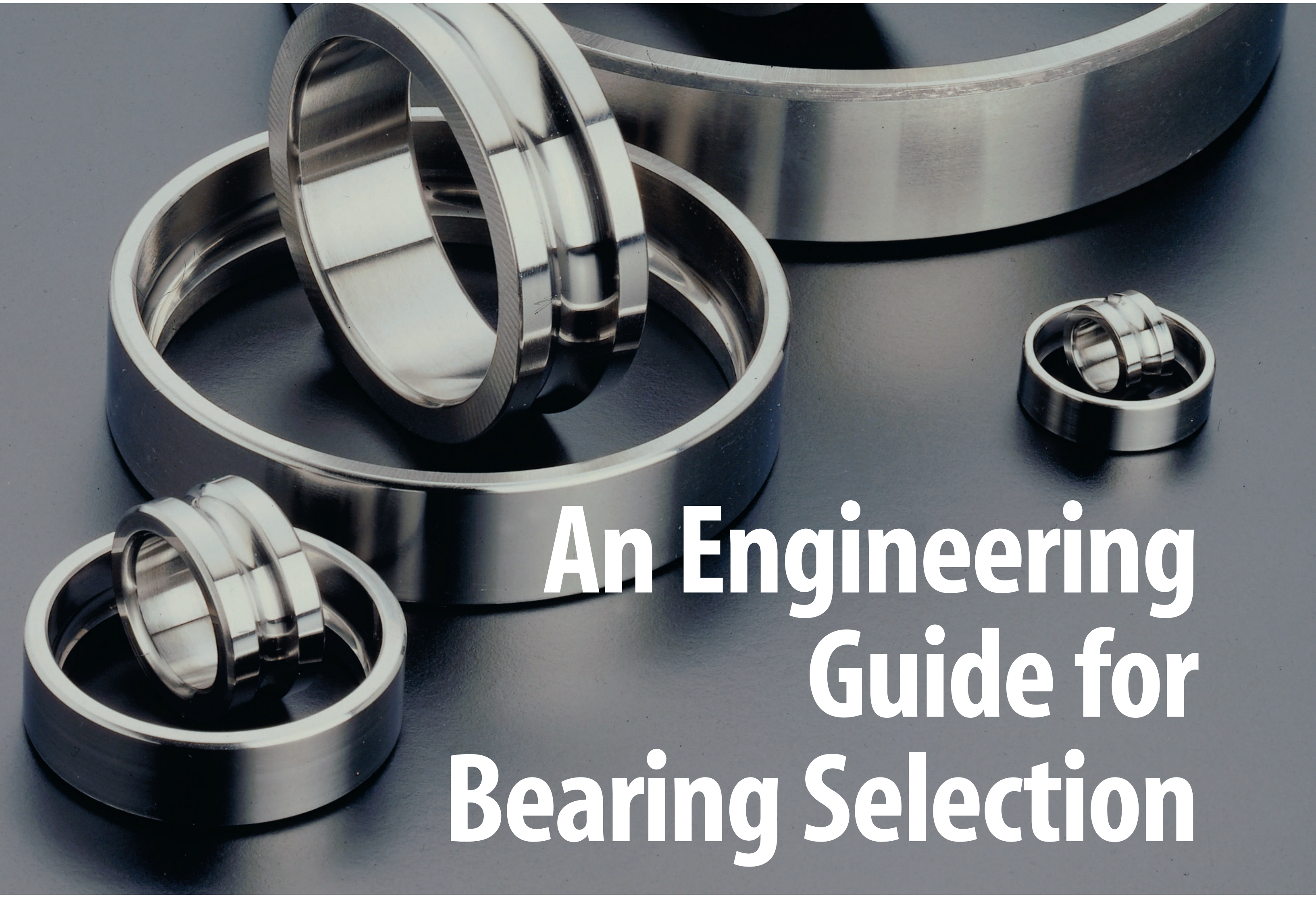

An Engineering Guide for Bearing Selection

By Michael M. Khonsari
(Fellow, STLE)
Louisiana State University,
Dept. of Mechanical Engineering,
Baton Rouge, Lo.

and

E. Richard Booser
(Fellow, STLE)
Niskayuna, N.Y.

Bearings are considered to be the most critical components of machinery. Today, billions of dollars are spent on the production of bearings. While the price of a single bearing is minimal, there are tremendous expenses associated with a bearing failure that may cause a forced shut down. Hence, the selection of the appropriate bearing type that can be performed as intended is a daunting engineering task.

Moving parts of machines must be located and supported, invariably by the appropriate bearing design and selection. Bearings capable of supporting a radial load are often called journal or sleeve bearings, while those carrying an axial load are termed thrust bearings. This bear-

Factor	Fluid film	Dry	Semilubricated	Rolling element
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	n/a	Good	Lubricant	Lubricant
Damping capacity	High	Low	Low	Low
Noise	Low	Medium	Medium	High
Dirt/dust	Need seals	Good	Fair	Need seals
Radial space required	Small	Small	Small	Large
Cost	High	Low	Low	Medium

Table 1. Characteristics of common classes of bearings. *(Source: Kennedy, F.E., Booser, E.R. and Wilcock, D.F., in The CRC Handbook of Mechanical Engineering, 1998)*

ing selection involves a wide variety of choices that may be broadly placed in four general classifications: dry, semi-lubricated, fluid-film, or rolling element bearings.

Selection of the bearing type starts with listing the functions involved together with other requirements for such factors as life, reliability, ambient conditions and vibration. Cost, customer preference and previous experience in similar applications provide further guidelines. In general, the typical performance items (*as shown in Table* 1) and the following review of capabilities and limitations make possible a proper selection from among the four classifications.

Figure 1. Plastic composite bushings and thrust washers for dry operation. (Fig. 1.7 of our referenced book. (Fig. 1.7 of our Reference book) *(Photo courtesy of SKF)*

BEARING Classification

Dry bearings. These are the simplest and lowest cost, commonly closely conforming to the shape of their mating shaft or thrust surface (*See Fig.* 1). Primary limiting factors are generally low load and low surface speed in the range shown in Table 2, generally far lower for dry bearings than for the types involving at least some lubrication. Basically these bearings are suitable for relatively low values of PV (bearing load on projected area P [psi] x surface velocity V [ft/minute]).

Dry bearings are usually made of polymers, often blended for lower friction and added strength with solids such as molybdenum disulfide, graphite, other inorganic powders, PTFE (polytetrafluoroethylene) polymer, or nylon. Despite their limitation to low speeds and loads, dry bearings are commonly used for the mild rubbing conditions found in household appliances, toys, office machines and related applications.

Semilubricated bearings. Typical of this class are porous metal bearings consisting of sintered powders of bronze, iron and aluminum with self-contained lubricating oil in their pores. In some instances, minor additions of silicone or other oils are also incorporated in polymer bearings. The most attractive features of semilubricated bearings are their simplicity and low cost. While limited numbers of these bearings range in bore size up to 6 inches, porous metal bearings are produced by the millions for small shaft sizes ranging from 1/16 to 1-inch in small electric motors, automotive accessories, household appliances, machine tools and business machines within the load and speed limits (*see Table* 2).

Bearings of conventional bearing metals such as bronze, babbitt and cast iron may also operate under oil-starved conditions while using a very limited supply of oil from wicks, oil mist, or individual oil or grease applicators. This limited supply of lubricant greatly improves load and speed limits over that with dry operation but still limits operation to moderate speeds and loads.

Wear is the usual failure mechanism with both dry and semilubricated bearings. Bearing suppliers commonly provide expected wear rates (*see Table* 2) and limiting temperature rise as related to the PV load-speed factor. For applications involving oscillatory motion, the porous bearings are better suited since the lubricant tends to flow out of the pores upon the application of pressure, thus providing the necessary lubrication in each direction.

CONTINUED ON PAGE 28

Table 2. Typical operating limits and wear factors for dry and semi-lubricated bearings. Captions: a. Psi = (MN/m2) x 145; b. Ft/min = (m/s) x 197; c. psi x (ft/min) = MN/(m.s) x 28551 d. Cubic meters of material worn away in sliding 1 meter on a ground steel surface under 1 Newton load. Wear volume is proportional to load and sliding distance for other conditions.

Operating limits and wear factors for dry and semilubricated bearings

Material	Max. temp., °C	Max. pressure, P, MN/m²	Max. speed, V, m/sec	PV limit, MN/(m-sec)	(a)Wear factor, 10^{-15}m³/(N-m)
Thermoplastics					
Nylon	90	5	3	0.90	4.0
Filled	150	10	—	0.46	0.24
Acetal	100	5	3	0.10	1.3
Filled	—	—	—	0.28	0.49
PTFE	250	3.4	0.3	0.04	400
Filled	250	17	5	0.53	0.14
Fabric	—	400	0.8	0.88	—
Polycarbonate	105	7	5	0.03	50
Thermosetting					
Phenolics	120	41	13	0.18	—
Filled	160	—	—	0.53	—
Polyimides	260	—	8	4	1.7
Filled	260	—	8	5	0.4
Porous metals					
Bronze	100	28	6.1	1.8	—
Iron	100	25	2.0	1.3	—
Aluminum	100	14	6.1	1.8	—
Others					
Carbon-graphite	400	4.1	13	0.53	0.8
Wood	70	14	10	0.42	—
Rubber	65	0.3	20	—	—

Conversion factors:
psi = MN/m² × 145; ft/min = m/sec × 197; psi × ft/min = MN/m-sec × 28,551

(a) Cubic meters of material worn away in sliding one meter on a ground steel surface under one Newton load. Wear volume is proportional to load and sliding distance for other conditions.

Figure 2. Oil-film bearing combining babbitted journal and thrust surfaces. (Fig. 1.8 of our book.) *(Photo courtesy of Kingsbury, Inc.)*

Fluid-film bearings. Full separation of the moving surfaces in a bearing is provided in fluid-film (*hydrodynamic*) bearings by a thin film of a liquid such as oil (*see Figure* 2) or water, or even by a gas such as air. The type of fluid-film bearing is commonly further classified by the method of fluid feeding: (1) *self-acting* in which the relative motion between a shaft or thrust runner pumps up the fluid internally within the bearing to a pressure for load support, or (2) *externally pressurized* (*hydrostatic*) with a fluid already under pressure being fed to generate load support within the bearing clearance where self-pumping action would be inadequate at low speeds or with gases.

Fluid film bearings are used in a wide range of machinery where their usually higher cost becomes secondary to the long life they provide in medium to large-size machinery such as steam and gas turbines, large electric motors and generators, pumps and compressors, large fans, steel rolling mills, and reciprocating gasoline and diesel engines. Table 3 gives some typical design loads.

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)

Table 3. Typical design loads for hydrodynamic bearings.

Ball and roller bearings. These transform sliding motion with the previous types to rolling action with its low starting and running friction and easier lubrication. Their low lubrication needs make grease the primary choice for many industrial applications. With either grease or oil a very thin lubricant film forms to avoid wear in the very heavily stressed contact area between the balls or rollers and their mating rings. They are also well suited for applications that involve transient interruptions in oil supply, which might be experienced in aircraft jet engines during start-up and maneuvering.

With their worldwide standardization to exacting geometric standards, ball and roller bearings are now used in huge quantities in 3-mm to 150-mm and larger bore size for automotive, railroad, aircraft, electric motors, fans, and a wide variety of other industrial, farm machinery, and transportation applications (S*ee Fig*. 3).

OPERATING RANGE

Unless a bearing matches all operating requirements in a machine, cost and environmental considerations are of no importance. Primary consideration as to the capability of the bearing to match speed and load requirements is followed by further review of space needs, available life and lubrication requirements.

Speed. Each of the four classes of bearings in Table 1 has a practical speed limit. Usual practice for non-aerospace application limits rolling bearings using oil lubrication to a DN value (mm bore ¥ rpm) of 500,000 to 1,000,000 corresponding to a shaft surface speed of 5,000 to 10,000 ft/min. Recent designs in aerospace applications, such as aircraft gas turbine engines, incorporate ball bearings that operate at DN values exceeding 3,000,000. On the other hand, fluid film bearings have a much higher limit and are used at speeds limited only by the rotor bursting strength. Shaft surface speeds in power plant turbine bearings range up to about 30,000 ft/min. Field experience with these large bearings revealed satisfactory operation for many years with no signs of distress. Much lower speed limits in the range of 300 to 1,500 ft/min (3 to 10 meters/s) are imposed by localized surface heating effects at surface asperities with dry and semi-lubricated bearings. Figure 4 (*see page* 30) is a useful guide for an initial bearing selection.

Load. Rolling element bearings are generally more versatile in being able to carry their fatigue-rated load at all speeds from zero up

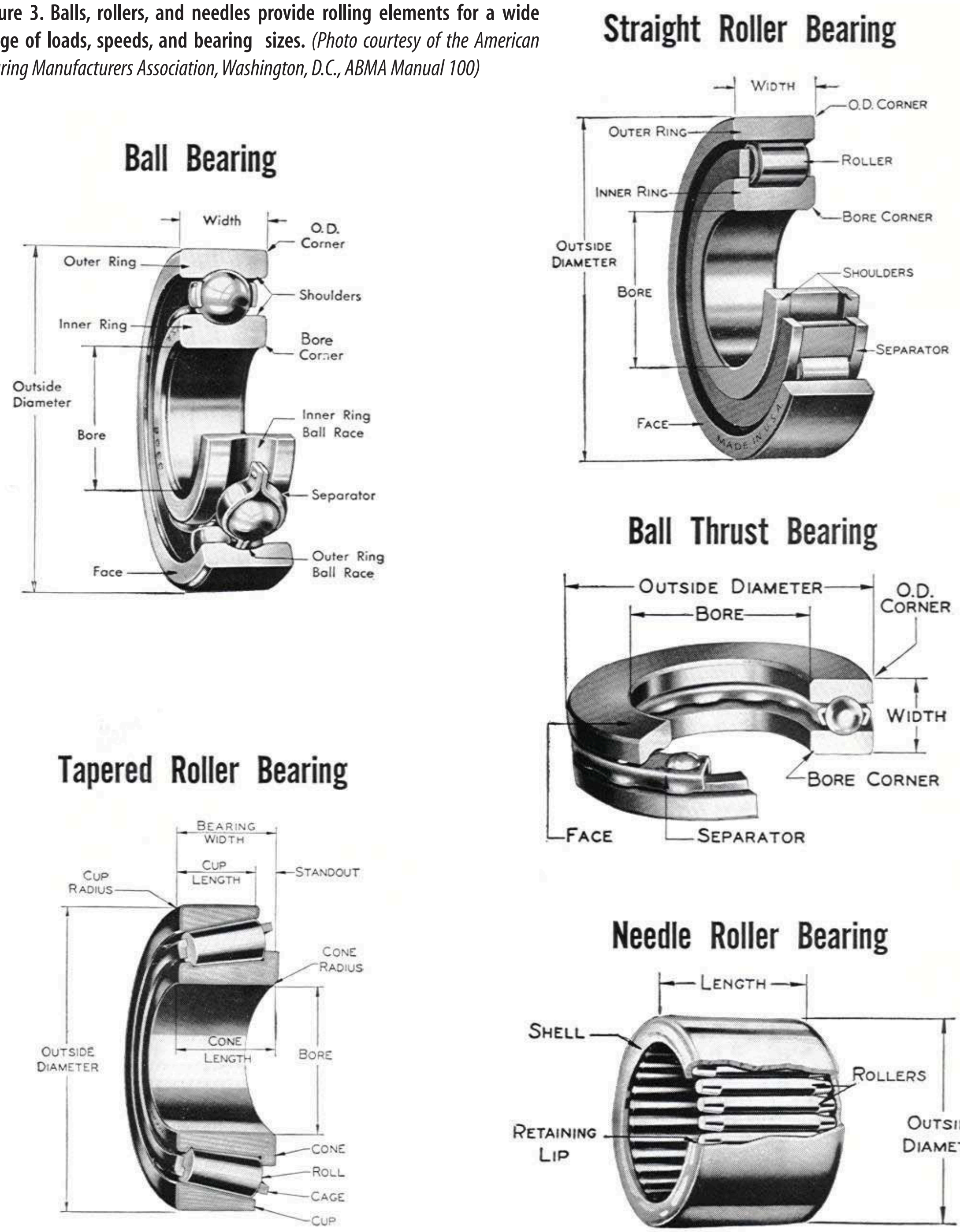

Figure 3. Balls, rollers, and needles provide rolling elements for a wide range of loads, speeds, and bearing sizes. *(Photo courtesy of the American Bearing Manufacturers Association, Washington, D.C., ABMA Manual 100)*

and in all directions. 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 x surface velocity) which gives a much lower load limit with rising speed. The high load limits in Table 2 can approach the material yield strength with appropriate material combinations in bearings operating at low speeds.

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.

CONTINUED ON PAGE 30

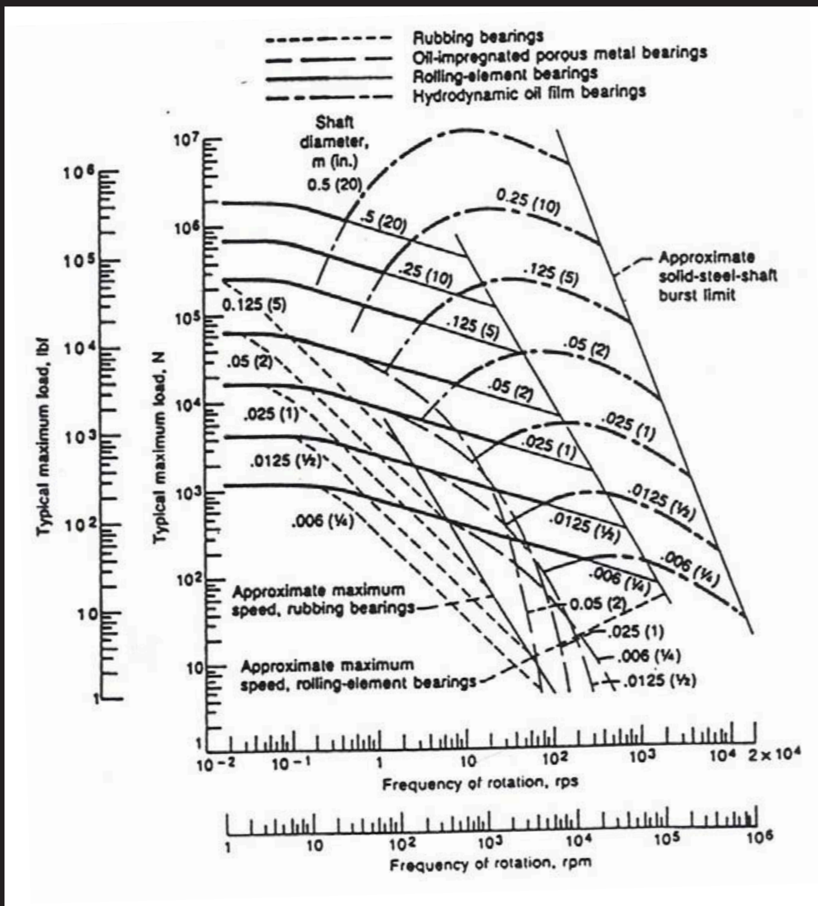

Figure 4. General guide for selecting bearings for radial loads. Bore diameters are given in inches on each curve. Except for rolling-element bearings, the length/diameter ratio = 1. Medium-viscosity mineral oil lubrication is assumed for hydrodynamic bearings. *(Photo courtesy of ESDU Item 65007, Engineering Sciences Data Unit, Institution of Mechanical Engineers, London, 1965)*

Example of bearing selection. Using Figure 4, possible types of bearings can be checked for possible use on a 2-inch diameter shaft for carrying 1,000 lb radial load at several speeds. Following across the horizontal line for this load gives the following:

Speed, rpm	Suitable bearing types
10	Rubbing, porous metal, rolling, oil film
100	Porous metal, rolling, oil film
1,000	Rolling, oil film
10,000	Oil film

The first type listed at each speed would generally be least expensive. At 10 rpm, surface speed would commonly be too low to build up a full oil film and may eliminate consideration of this type. The 10,000 rpm speed could also be accommodated with precision grade ball bearings using oil lubrication.

Friction and power loss. Low starting friction, especially under load, is a prime advantage of ball and roller bearings. While involving added complexity, externally pressurized oil lift pockets also 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 break away of 0.15 to 0.25 are typical for oil film bearings and for dry and semi-lubricated plastic and porous metal bearings. With dry and semi-lubricated surfaces, friction then drops about in half as motion gets underway.

Life. Rolling element bearings have a distinct fatigue life limit which results from repeated contact stresses by the balls and rollers on their raceways, while fluid film bearings in usual rotating equipment can provide essentially unlimited life. Fatigue life is also a limiting factor for oil film bearings under cyclic loading in internal combustion engines. In dry and semi-lubricated bearings, life is estimated from an approximate material *wear factor*, such as shown in Table 2, that relates bearing 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.

Space requirement. Dry and semilubricated bearings require minimum space. A porous metal, plastic, or plastic-lined bearing is commonly a bushing or thrust washer such as illustrated in Fig. 1. These involve just sufficient wall thickness to provide the needed strength for insertion into 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.

The outside diameter of a ball and roller bearing commonly ranges from about 1.5 to 3 times the bore, and the axial dimension range from one-fifth to one-half the shaft diameter. Oil film journal bearings are more compact in their radial dimension, and range in axial length from about one-third up to being equal to the shaft diameter. Considerable additional volume is commonly required with oil film bearings to accommodate seals plus feed and drain passages. Fig. 2 illustrates the general proportions for a babbitt journal bearing combined with an integral thrust face and its oil distributing grooves.

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 in many small and medium size ball and roller bearings. Many rolling element bearings depend only on an initial grease fill for years of operation when DN values (mm bore ¥ rpm) are less than 300,000.

Under conditions of heavy load and high speed such as encountered by aircraft jet engine bearings, additional lubricant must be supplied to remove heat and maintain a reasonable limit on temperature. Caution is required: Too much added oil results in churning with an extraneous power loss which may heat up the bearing and cause loss of internal clearance. Oil circulation with external cooling should be added to the bearings with the highest operating demands. 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 1,000 gal/min. for the oil film bearings in a steam turbine-generator at an electric power station. While full oil feed is generally desirable for cooling and rotor vibrational stability, excess feed should be minimized at high surface speeds as it can induce up to 25-50% parasitic power loss from oil churning (Khonsari and Booser, 2001).

Elastic and damping response. Some damping capacity is desirable within a bearing to absorb vibrational energy of rotating parts. Ball and roller bearings have virtually no damping capacity on their own, but oil or friction damping can be introduced through specially designed mounts in the housing. External squeeze-film dampers often fulfill this need for jet engines and aerospace units where severe vibration would otherwise be encountered at critical speeds of the rotor. With fluid-film bearings, on the other hand, the oil film itself will often provide suitable elastic and damping response for a wide range of rotor system needs.

ENVIRONMENTAL CONDITIONS

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

Temperature. Fluid film bearings generally are the most limited in their temperature range. Common soft metal bearing surfaces of tin and lead babbitt limit upper temperatures to the 125-150C (260-300F) range. At the other extreme, the low temperature range with many oil film bearings is limited to 0 to 50 F by resistance to flow by the high viscosity of cold mineral oil in feed and drain passages. High start-up friction torque is also critical at low operating temperatures, particularly with hydraulics in mobile equipment.

Ball and roller bearings using conventional high carbon, low alloy steels such as AISI 52100 and case-hardening steels, employed for their high hardness and fatigue strength, are generally limited to an upper temperature of 125-150 C (260-300 F) and can be specially stabilized for operation up to 200 C (400 F). Tool steel bearing materials allow operation into the 315 C (600 F) range, and with vapor phase lubrication or solid lubricants 650C (1200F) can be reached with ceramic bearings.

Contamination. Within each of the four bearing classes, methods are available to accommodate most exposures to dirt and corrosive atmospheres. While common ball and roller bearing steels are subject to rusting, multipurpose greases and double-sealed and double-shielded enclosures generally provide sufficient water and contamination resistance to eliminate need for special maintenance. For high demand bearings, such as in an aircraft gas turbine engine, a recirculating oil system with a filter and oil cooler is a must. Similar filtration has also become an integral part of almost all circulating oil systems for electrical machinery, automotive and diesel engines, and general industrial machinery.

EXAMPLES OF SELECTIONS

Appliances. Minimum needs for lubrication and maintenance and low cost frequently guide selection of bearing design in household appliances. The evolution often involved is illustrated in the food-disposing units for kitchen sinks. Initially a tapered roller bearing was used in the impeller unit to absorb any shock loads during ingestion of items such as kitchen utensils. Later steps brought a double-sealed ball bearing and then a porous metal bearing with adequate performance and involving only a few percent of the tapered roller bearing cost.

Wide use in appliances of dry and semi-lubricated bearings has likely reached an ultimate step in some clothes driers. Here two small thin strips of a composite plastic are inserted in the frame to support the rotating drier drum. In instruments, timers, and small appliances, bearing surfaces are often formed as simple holes in structural plastic elements.

Gas turbines. Ball and roller bearing use is firmly established for aircraft turbines used in jet aircraft. Light weight, small size, high thrust loads, and wide temperature ranges have always led to use of rolling type bearings. Their small lubrication needs further meet the requirements both for a compact oil system and for any

CONTINUED ON PAGE 32

lack of lubricant feed at start-up or during transient flight conditions. Supplemental oil is also usually pumped through the bearings to match their cooling needs and to supply damping at the interface between the outer bearing ring and its housing.

Land-based steam and gas turbines for electric power generation, on the other hand, use a large, pressurized oil supply and oil film bearings both for absence of a fatigue life limit and for their inherent damping response to control rotor vibration.

Electric motors. Industrial electric motors illustrate the evolution of their bearing type over an extended period. Earliest motors in the 1900 era used available fluid-film bearings fed by oil rings hanging from their rotating shaft into a small self-contained oil reservoir. Then, early in the last century, newly available ball bearings with their small size and reduced maintenance needs led to their use in fractional horsepower motors and in motors for electric automobiles and small mine locomotives. With lowering costs and long life greases, ball bearings became standard for industrial motors up to 25-50 horsepower size in the 1950s. Today, ball bearings are almost exclusively used in motors ranging up to about 1,000-hp size.

Large celestial telescope. Hydrostatic bearings have been employed to support enormous structures such as telescopes, observatory domes, and large radio antennas where often the weight requirements range from 250 to over one million lbs. For example, the Magellan Telescopes for the Observatories of the Carnegie Institution of Washington is designed by L & F Industries to support 320,000 lbs. of rotating weight, which includes an enormous 6.5-meter diameter mirror. The system utilizes 18 hydrostatic bearing pads to support the weight of the azimuth and elevation axes, allowing them to rotate merely by gentle pressure from the operator's fingertip.

High-speed dental drills. A typical dental drill journal bearing uses air to lubricate its surface while it spins smoothly at speeds of 600,000 rpm. The practical limit for a gas bearing can range up to 700,000 rpm without auxiliary cooling requirements.

A LOOK AHEAD

Coming trends for bearing applications will likely be set by the needs of future machines for lower maintenance while operating at higher speeds, higher temperatures, and with more compact designs.

Dry and semilubricated bearings will continue ever-broadening use of plastics and their composites for mild conditions as small bushings in household appliances, machine tools, instruments, construction equipment, automobile chassies and business machines. These applications will involve more and more direct integration of the bearings with housings and structural elements.

Ball and roller bearings, continuing their dominance in jet engines and aerospace, will broaden their use in small and medium-sized electric motors; automobile accessories; machine tools; and railroad, construction, and agriculture equipment. Their very small lubrication needs may bring innovations in greases, self-contained lubricant impregnation in ball and roller cages, and surfaces augmented with solid lubricants. Further developments in ceramics, tool steels, and special lubricants will enable continually higher speeds and temperatures.

Fluid-film bearings will continue to fill their essential role with oil lubrication in reciprocating diesel and automotive engines; and in electrical turbine-generators, metal rolling mills, and other large machinery. Use of gases and low-viscosity liquids as "lubricants" will likely escalate. The air-film bearings in flying heads on computer discs and in airliner cabin compressors will gradually come into use for an increasing array of industrial, instrumentation, and aerospace units. Water, gasoline, liquid ammonia, and a wide variety of petroleum and chemical process streams are likely to be employed in bearings to avoid the complexity and contamination problems with conventional lubricants. These will involve development of new bearing materials, plus advancing analysis and design techniques for surface profiles to match the extremely thin fluid films.

Such are but a sampling of the challenges for bearing technology in the 21st century. <<

Reference:

Khonsari, M.M. and Booser, E.R. (2001), Applied Tribology: Bearing Design and Lubrication, John Wiley & Sons, New York, NY.

Mike Khonsari is Dow Chemical endowed chair and professor of mechanical engineering at Louisiana State University in Baton Rouge, La. You can reach him at **khonsari@me.lsu.edu**.

Dick Booser is a consulting engineer and is past vice-chairman of the STLE Editorial and Publications Committee and Editor of the STLE Lubrication and Tribology Handbooks. You can reach him at **erbooser@aol.com**.