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INTRODUCTION

During more than 18 years as a machine designer, I have observed that experienced practitioners usually have assembled a collection of reference materials that they rely on constantly. This valuable cache usually includes standards, data tables, procedures, symbol definitions, component sizing guides, articles, and manufacturer's publications. Experienced designers also tend to have selected a few really great reference books and other resources like technical website links. This book attempts to summarize some of that useful information and present it in a practical way that maximizes on-the-job efficiency.

Meant for machine designers at all stages of their careers, this book contains materials that should be useful to all practitioners of machine design. However, it will be especially useful to designers in the automation, assembly equipment, and light industrial machine industries. For those at the start of their careers or in need of a refresher, I have attempted to include enough background information on each topic to allow the reader to use the procedures and formulas, and communicate effectively with suppliers. For the experienced practitioner, *Machine Designers Reference* is constructed to quickly and concisely deliver the tips, formulas, and data that are needed most often. Most formulas and data are tabulated to make them easy to find and use. All tables are indexed at the start of each chapter to quickly direct the reader to information that is frequently referenced. Bullet lists of best practices and critical considerations are provided for each topic.

The value of *Machine Designers Reference* will be enhanced when used in conjunction with the standard handbooks and textbooks of our time. It is expected that the reader will exercise good engineering judgment, perform detailed analysis when needed, and work closely with component manufacturers and shops. I have included a recommended resources list at the beginning of each section to assist the reader in locating excellent in-depth treatment of most subjects. The companies and websites mentioned within are resources that I have used in the course of my machine design work. The books are mostly those I own and use regularly.

When providing fastener or component information, I consulted the catalogs of several major distributors I have had experience with. I have included only the items and sizes most commonly used in light industrial machinery. To include all the data would make for a truly gigantic book. For more sizes and types of machine components, refer to a manufacturer's catalog or the applicable standards. I have also chosen to represent fractional inch sizes as decimal values. I have found that in the age of CAD, decimals are preferred.

Machine design is a global business, and many companies have converted to metric in the United States. This book is directed at U.S. designers using either the Imperial (English) or SI (metric) system of units. When presenting metric components, I have favored the ISO standards because in my experience ISO-designated components are more frequently encountered in light industry. I have chosen to use the decimal point rather than the comma in metric numbering to avoid confusion for anyone unfamiliar with metric convention. (A comma is the proper metric decimal indicator.)

The tabulated formulas contained in this book are designed to be convenient and quick to use for those experienced in the subject at hand. I have also found that they help me understand the interplay of variables and make better design decisions. For those following along with the written procedures in this book, I hope the tabulated formulas will be instructive, if not always convenient.

It is my intention that this book be easily used as a quick reference. The data tables for materials and fasteners are grouped together to assist in quickly locating and using the information when performing selection activities. I hope that readers will find that their appreciation for the groupings and ordering of the data tables grows as they use this book on the job.

New information and updates regarding this book and its contents are expected to be posted at www.machinedesignersreference.com as they become available.

Any errata, comments, or suggestions that you can send to me at *info@industrialpress.com* would be greatly appreciated.

Jennifer Marrs
September, 2011

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BEARINGS

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Section 10.1

PLAIN BEARINGS

Plain bearings provide sliding contact between two surfaces. The most common type of plain bearing is the sleeve bearing or bushing. Plain bearings are often chosen over rolling element bearings due to cost or space limitations. They are also more rigid and quieter in operation than rolling element bearings. The main disadvantages of plain bearings are their higher potential to wear (as compared to rolling element bearings) as well as their relative vulnerability to contaminants. This section serves as a general introduction to plain bearings, with a focus on sleeve bearings used in rotary motion applications with boundary lubrication conditions. Please consult the recommended resources for more information and calculation methods for other types of lubrication. Plain bearings used in linear motion applications are discussed in Section 10.3.

RECOMMENDED RESOURCES

- R. Mott, ***Machine Elements in Mechanical Design***, 5th Ed., Pearson/Prentice Hall, Inc., Upper Saddle River, NJ, 2012
- R. L. Norton, ***Machine Design: An Integrated Approach***, 4th Ed., Prentice Hall, Upper Saddle River, NJ, 2011
- Oberg, Jones, Horton, Ryffel, ***Machinery's Handbook***, 28th Ed., Industrial Press, New York, NY, 2008

LUBRICATION OF PLAIN BEARINGS

Plain bearings must be lubricated in order to have long life and low friction. There are four types of lubrication conditions under which plain bearings are run: hydrostatic lubrication (full film), hydrodynamic lubrication (full film), mixed film lubrication, and boundary lubrication (thin film). Full film lubrication occurs when the lubricant layer between surfaces is thick enough to prevent any surface contact. Boundary lubrication occurs when the lubricant layer is present but not thick enough to prevent contact between surfaces. A graph showing relative coefficients of friction for the different types of lubrication are shown in Figure 10-1. The horizontal axis is a function of lubricant viscosity (Z), journal speed (N), and bearing pressure (P).

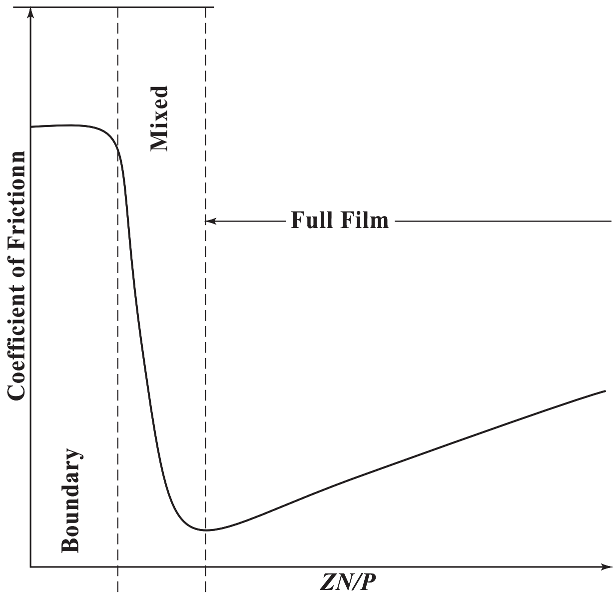


Figure 10-1: Plain Bearing Lubrication vs. Coefficient of Friction

Hydrostatic lubrication is full film lubrication, and occurs when high pressure lubricant is used to force the sliding surfaces apart. Plain bearings with hydrostatic lubrication can accommodate heavy loads at low speeds. Hydrostatic lubrication is normally used in planar or linear bearings rather than in sleeves. Design of an assembly using hydrostatic lubrication is extremely complex and must focus on lubricant feeding and containment. Hydrostatic lubrication is uncommon in light industry and is beyond the scope of this text.

Hydrodynamic lubrication is full film lubrication, and is most commonly employed with high-speed rotating shafts in plain sleeve bearings. Bearings with hydrodynamic lubrication are often called journal bearings. A wedge of lubricant is caught between the rotating shaft and bearing surface, providing sufficient pressure to carry the applied load. The shaft rides on a film of oil and does not contact the bearing except during periods of low speed or stasis. Typical coefficients of friction range from 0.002 to 0.010. Oil is typically used as the lubricant, and it must be supplied from a reservoir to maintain hydrodynamic lubrication. The lubricant also cools the bearing, and lubricant leakage and circulation enhances the cooling effect. Lubricant viscosity and temperature are important parameters in hydrodynamic lubrication performance, and temperature control is recommended. Hydrodynamically lubricated bearings go through periods of boundary lubrication during startup and shutdown periods. In light industrial machinery, relatively low speeds and/or intermittent movements mean that boundary lubrication of plain bearings is more common than hydrodynamic lubrication. Hydrodynamically lubricated journal bearings are beyond the scope of this text, but are detailed in the recommended resources.

Mixed film lubrication occurs when the lubricant film thickness is slightly greater than the surface roughness of the surfaces. This is not hydrodynamic lubrication because the surfaces are close enough together to create drag and some surface contact. The conditions for mixed film lubrication are between those of hydrodynamic and boundary lubrication.

Boundary lubrication occurs at low speeds and/or high pressures. Surfaces in contact are partially separated by a lubricant film and partially in direct rubbing contact. Bearings with boundary lubrication have higher coefficients of friction and shorter life expectancy than hydrostatically or hydrodynamically lubricated bearings. To reduce friction and reduce wear, lubricant is usually embedded in the bearing and/or added to the shaft. Values of friction coefficients vary depending on bearing material and construction, but typically range from 0.05 to 0.20. Sleeve bearings designed to run well without added lubricants are often called “oil free” bearings, and generally have lubricant embedded in them, or lubricity is a property of the material. Many plastics, for instance, are inherently lubricious and can provide low coefficients of friction without additional lubrication. Plain bearings with boundary lubrication are common in the machinery industry and will be examined further in this section.

Lubricant is often applied to boundary lubricated bearings to extend life and reduce drag. Check the manufacturer’s recommendations for recommended or prohibited lubricants. Grease is often used instead of oil in boundary lubrication applications because it’s convenient to use, stays in place, and allows for simpler design. Grease and other fixed lubricants do nothing to cool the bearing, so relatively slow speeds are a must with boundary lubrication.

When the intention is to add lubricant to a plain bearing, provide for lubricant proliferation through the assembly, even when designing for boundary lubrication. In a fixed shaft arrangement, the shaft is often cross drilled and has an axial or a radial groove to allow lubricant to enter the bearing. A grease fitting or oil supply is often placed at one end of the shaft. In a rotating or sliding shaft arrangement, the lubricant supply is usually routed through the bearing housing and relies on a hole or a hole and groove to allow lubricant to enter the bearing. A grease fitting or oil supply is then placed on the bearing housing. When planning for hydrodynamic or hydrostatic lubrication, a thorough analysis of the lubricant circulation must be undertaken.

BEARING MATERIALS FOR BOUNDARY LUBRICATION

Plain bearings designed to run with boundary lubrication are available in a variety of materials. There are many materials available, but the most common commercial bearing materials are tin, lead, bronze, copper, and plastic. Their general characteristics are as follows:

Bronzes and copper alloys: Bearings made of these materials are extremely common. They resist wear and temperature variation well. These bearings have a limited capacity to embed contaminants, so shaft scoring is a risk and requires shaft hardening. Alloys with higher lead content reduce shaft scoring and reduce friction during boundary lubrication, but will be more sensitive to elevated temperatures and require lower speeds. Copper alloy bearings often are available with dry lubricant embedded in the bearing surface. Sintered bronze bearings are porous and usually saturated with lubricant. Material properties vary, so consult the manufacturer's catalog for more information.

Plastics: In recent years plastic plain bearings have met or exceeded many characteristics of their metal counterparts. Plastic bearings can be formulated to meet a wide variety of requirements, including sanitary applications. When specifying a plastic bearing, it is important to consider that these bearings tend to run hotter than their thermally conductive metal counterparts. Elevated ambient temperatures can be of particular concern with plastic bearings. Confirm the temperature ratings of any plastic bearings with the manufacturer. It is important to note that some lubricants can attack plastic bearings or cause them to swell, so be sure to check material compatibility. Material properties vary greatly, so consult the manufacturer's catalog for more information.

BOUNDARY LUBRICATED SLEEVE BEARINGS

Sleeve bearings used with boundary lubrication are very common in modern machine design. They are ideal for bearing applications requiring small radial clearances, quiet running, low speeds, and oscillating or intermittent movements. They are often available in straight and single flanged types (Figure 10-2). The flange can be used as a thrust bearing, and is also subject to boundary lubrication conditions. Sleeve bearings are most commonly pressed into a housing bore, but some have features for retaining the bearing with screws. Follow the manufacturer's recommendations for fits and tolerances of

Figure 10-2: Plain Sleeve Bearings
Source: igus Inc. (www.igus.com)

the bearing housing and shaft because proper fit is required to produce proper bore clearances, ensure even wear, and minimize drag.

Commercial catalogs often rate the load capacity of sleeve bearings intended for boundary and mixed film lubrication in terms of maximum allowable load, maximum allowable surface velocity, and PV factor. These values are used as maximum values to determine bearing size and materials. A safety factor of 2 is typical when examining PV factor. PV factor ratings will be highly variable depending on material, method of manufacture, and whether any embedded lubricants are present. Some manufacturers offer online software to select bearings and calculate life expectancy. One excellent example of such software can be found at www.igus.com.

Bearing pressure (P) is the radial force on the bearing divided by the projected area of contact. Projected area is equal to shaft diameter multiplied by length. Because PV is a function of bearing length, the use of a longer bearing is one method of getting the PV of the application lower than the PV rating for the bearing. In general, the length to bore ratio of plain sleeve bearings should never exceed 4:1.

PV factor is proportional to the heat of friction generated per unit of bearing area; it is used to measure the performance capability of a bearing. PV factor alone should not be used to size sleeve bearings. For low speed or high load applications, maximum allowable load will be the primary consideration. PV factor is calculated for rotating sleeve bearings according to the formulas in Table 10-1. Some common bearing material properties are listed in Table 10-2.

Selection of proper shaft material and hardness is critical for plain bearing and assembly performance. A smooth shaft surface makes the most of thin film lubrication and reduces friction. Sufficient hardness is essential to ensure that the shaft does not wear before the bearing does. A rule of thumb says that

Table 10-1: PV Formulas for Rotary Sleeve Bearings

PV FACTOR	PV factor = P x V
INCH UNITS P = Bearing pressure (psi) F = Load on the bearing (lbf) A = Bearing area (in^2) D = Bearing internal diameter (in) or shaft external diameter L = Bearing length (in) V = Bearing surface speed (ft/min) N = Revolutions per minute θ = Swept angle of oscillation (radians) f = Frequency of oscillation (oscillations per minute)	$P = \frac{F}{A}$ $A = DL$ Continuous rotation: $V = \frac{D\pi(N)}{12}$ Oscillating rotation: $V = \frac{\theta f D}{24}$
METRIC UNITS P = bearing pressure (MPa) F = load on the bearing (N) A = bearing area (m^2) D = bearing internal diameter (m) or shaft external diameter L = bearing length (m) V = bearing surface speed (m/min) N = revolutions per minute θ = Swept angle of oscillation (radians) f = Frequency of oscillation (oscillations per minute)	$P = \frac{F}{A(10^6)}$ $A = DL$ Continuous rotation: $V = D\pi(N)$ Oscillating rotation: $V = \frac{\theta f D}{2}$

Table 10-2: Typical Plain Bearing Material Properties

Bearing Material	Type	Temperature Range	Shaft Hardness	Dynamic P Max.	V Max.	PV Max
SAE 841 Bronze	Porous	10°F to 220°F	> 35 Rc	2,000 psi	1,200 ft/min	50,000 psi-ft/min
		-12°C to 104°C		14 MPa	6 m/s	1.8 MPa-m/s
SAE 660 Bronze	Non-porous	10°F to 450°F	> 35 Rc	4,000 psi	750 ft/min	75,000 psi-ft/min
		-12°C to 232°C		27 MPa	3.8 m/s	2.7 MPa-m/s
Nylon Plastic	Non-porous	10°F to 20°F	< 35 Rc	2,000 psi	600 ft/min	3,000 psi-ft/min
		-12°C to 93°C		14 MPa	3 m/s	0.11 MPa-m/s
PTFE Plastic	Non-porous	-350°F to 500°F	< 35 Rc	500 psi	50 ft/min	1,000 psi-ft/min
		-212°C to 260°C		3.5 Mpa	0.25 m/s	0.035 Mpa-m/s

the shaft should be 3 times harder than the bearing material. This difference in hardness ensures that any contaminant particles that enter the bearing will embed in the bearing material rather than score the shaft.

Hardened and ground steel shafts are generally recommended for use with bronze bearings. SAE 841 bronze bearings require steel shafts that have greater than 0.4% carbon content, such as SAE 1137 – 1141 steels. When corrosion protection is needed, hard chrome plating is a good choice. If stainless steel shafting must be used, choose a 400-series steel. When specifying stainless steel, a 416-series steel that is heat treated to maximum hardness is best. Shaft surface roughness should be 16 microinches (0.4 micrometers) or better.

Plastic bearings have different requirements depending on their makeup. Chrome plating may cause stick-slip behavior in some plastic bearings because the surface finish is too fine. Stainless steel in the 300 series may be used with some plastic bearings. For plastic bearings, always follow the manufacturer's recommendations for shaft material, hardness, and surface roughness.

SLEEVE BEARING SELECTION PROCEDURE FOR BOUNDARY LUBRICATION

When selecting a sleeve bearing, it can be beneficial to contact the manufacturers for application assistance. They often have specialists available to analyze the application and make recommendations to maximize bearing life and performance. The following procedure can be useful when selecting and sizing a sleeve bearing. This procedure assumes that the shaft is rotating in the bearing and there is boundary lubrication. This scenario is a very common one in light industrial machine design. Linear bearings are selected similarly, and are discussed in Section 10.3.

1. Determine the design parameters. These include the following:
   - Desired shaft diameter
   - Revolutions per minute of the shaft
2. Select a bearing length. A typical sleeve bearing should have a length of 1.5 to 2 times its inner diameter. An L/D as low as 1 is acceptable for some sintered bearings.
3. Select a bearing material. Compute the PV factor using the formulas in Table 10-1. Select a bearing in the desired size with a rated PV factor of at least twice (factor of safety) the computed value. Choosing a longer bearing will lower the PV factor for the application. Increasing the shaft size will also lower the PV factor.
4. Check the bearing maximum pressure rating. Verify that the chosen bearing maximum pressure rating is well above your calculated bearing pressure from Table 10-1. This includes checking the thrust load on the bearing flange if applicable.
5. Check the bearing maximum velocity rating. Verify that the chosen bearing maximum velocity rating is well above the calculated velocity of the shaft (journal) surface from Table 10-1.
6. Choose a bearing outer diameter. The bearing OD may already be set based on the results of previous steps. Once an OD is selected, verify that there is plenty of space for the OD of the chosen bearing in the bearing housing. Because the OD is generally pressed into the housing, avoid thin walls around the bearing.

7. Dimension and tolerance your shaft and bearing housing per the bearing manufacturer's recommendations. Specify the recommended shaft material and hardness. Add any required chamfers or radii, and specify any required surface finishes.

CRITICAL CONSIDERATIONS: Plain Bearings

- Plain bearing material selection is governed by the speed and loading conditions of the assembly.
- All plain bearings run under boundary lubrication conditions and will wear over time. Life can be maximized by staying well within the PV rating for the bearing.
- Shaft hardness and surface finish are critical for proper function and life expectancy.
- Heat is often the limiting factor with plastic bearings. This includes not only ambient temperature, but also the heat generated by friction during operation. Higher speeds generate more heat.
- Never store plain bearings with embedded lubricant in or on absorbent material, because the lubricant could wick out.

BEST PRACTICES: Plain Bearings

- Use a factor of safety of at least 2 when evaluating PV factors. This will ensure long bearing life.
- Add grease to plain bearings to prolong bearing life.
- Avoid ultra-soft bearing materials like PTFE unless absolutely needed. Soft materials are vulnerable to damage during installation and running, and tend to have shorter life spans than harder bearings.

ROLLING ELEMENT BEARINGS

Section 10.2

Rolling element bearings are extremely common in machinery. They create rolling contact between two parts to minimize the effects of friction and enable high relative speeds with very little wear. The most common types of rolling element bearings are ring shaped and provide rolling contact between a shaft and a housing bore. Another common rolling element bearing is a thrust bearing that takes an axial load while rotating about a shaft or pin. Both types use either balls or rollers to carry the applied loads. Design of bearings is beyond the scope of this text. This section will focus on the types, selection, and sizing of commercially available steel ball and roller bearings.

RECOMMENDED RESOURCES

- Juvinall, Marshek, ***Fundamentals of Machine Component Design***, 2nd Ed., John Wiley & Sons, Inc., New York, NY, 1991
- R. Mott, ***Machine Elements in Mechanical Design***, 5th Ed., Pearson/Prentice Hall, Inc., Upper Saddle River, NJ, 2012
- R. L. Norton, ***Machine Design: An Integrated Approach***, 4th Ed., Prentice Hall, Upper Saddle River, NJ, 2011
- Oberg, Jones, Horton, Ryffel, ***Machinery's Handbook***, 28th Ed., Industrial Press, New York, NY, 2008
- ***SKF General Catalogue***, SKF Group, June 2008
- ***ISO 281:*** "Rolling bearings — Dynamic load ratings and rating life"
- ***American Bearing Manufacturers Association Website:*** www.americanbearings.org

LUBRICATION, SEALS, AND SHIELDS

Rolling element bearings must be lubricated to ensure long life and low drag. Bearings are normally lubricated with grease for speeds below 500 rpm at normal ambient temperatures. High speeds and high temperatures both require a circulating oil lubrication system to keep the bearing cool. Manufacturers will normally specify the appropriate lubricant for each bearing.

Seals are available in many formats, but in general all are capable of fully sealing the bearing from contaminants and retaining lubricant in the bearing. "Lubricated for life" or "maintenance free" are phrases commonly used to describe rolling element bearings that have seals and are pre-lubricated at the manufacturer. If seals are not available for a bearing, provisions must be made in the assembly to direct and retain lubricant at the bearing. Some bearing manufacturers also sell a line of externally mounted seals for bearing arrangements.

Shields are intended to protect the bearing from large contaminants and accidental contact damage. They are generally metal and do not form a complete seal around the bearing. Bearings can have just seals, just shields, or both seals and shields. Not all bearings are available with seals or shields. Separable bearings are not sealed. In general, seals are available for deep groove ball bearings, angular contact ball bearings, self-aligning ball bearings, cylindrical roller bearings, needle roller bearings, and spherical roller bearings.

BEARING CHARACTERISTICS

Rolling element bearings are generally made up of two rings, called "races," separated by a set of rolling elements. Radial bearings are composed of an inner and outer race, whereas thrust bearings are composed of two (or more, in the case of double direction bearings) races. Rolling elements can be balls, cylindrical rollers, tapered rollers, or other specialized shapes.

A bearing cage is an internal component that captivates each rolling element in the bearing and prevents contact between individual rolling elements. Not all bearings have cages. Whether a bearing does or not will be clearly stated by the manufacturer. Cages help reduce internal friction in the bearing and ensure that the load bearing elements are evenly distributed for smooth running. "Full complement" bearings normally forego a cage so that more rolling elements can be fit into the bearing for greater load capacity. Cages are not always metal, so material and thermal compatibility must be verified if using caged bearings.

Some types of bearings, such as cylindrical roller bearings, are capable of axial displacement between the inner and outer races. When axial displacement is a function of the bearing, the maximum allowable displacement will be provided in the catalog information. Axial displacement can occur during normal running conditions as the shaft warms and elongates. This effect must be accommodated in the bearing arrangement through the use of either a bearing that allows axial displacement, or by designing the bearing arrangement and fits so that one bearing can easily move axially in its housing.

Most radial bearings are made with a cylindrical bore which is meant to be mounted on a cylindrical shaft. Some bearings are available with a tapered bore that is designed to be mounted on an adapter sleeve or tapered shaft. A tapered bore allows the bearing to be adjusted to remove internal clearance from the bearing. Cylindrical bores are much more commonly used and will be assumed for the purposes of this text unless otherwise specified. See the recommended resources for more information on the use of tapered bores.

Some bearings are separable, which means the inner and outer races can be separated axially from each other. Tapered roller bearings are a common example of separable bearings. Separability eases installation, but generally precludes integral sealing.

Temperature rating is not always stated in catalogs. In general, catalog values for life, speed, and loading assume an ambient temperature of 68°F (20°C). Most standard steel bearings have a maximum temperature of 250°F (121°C). The maximum temperature must include the elevated temperature of the bearing due to running conditions and friction.

Limiting speed is usually provided in catalog data for bearings. In all cases, the limiting speed should never be exceeded. The bearing can run up to limiting speed, but only for short periods of time. The reference or operating speed rating normally represents the speed at which the heat generated by the bearing is in equilibrium with the heat dissipated. It is assumed that operating above the rated speed for any length of time will cause the bearing to overheat. Reference or operating speed ratings are usually given for bearings in catalogs. Both the limiting and operating speed ratings assume a given set of operating and lubricant conditions. The standard ISO 15312: 2003 provides a set of reference conditions for the operating speed rating, which include an ambient temperature of 20°C (68°F), a bearing temperature of 70°C (158°F), and a constant load. These conditions also assume that the inner ring of the bearing is rotating. When the outer ring rotates, the speed ratings should be lowered. Note that when a bearing is operated at a very low speed or undergoes oscillating movements rather than continuous running, hydrodynamic lubrication does not occur in the bearing. In those cases, lubricants should be used that form a film on the bearings. Lubricants with EP additives form such a film.

Internal clearance is defined as the total distance that one race can be moved relative to the other in the stated direction. Bearings have radial and axial internal clearances. Normal clearance bearings are typically used and must be mounted with the recommended standard fits on the shaft and housing. If different mounting fits are used, the bearing must have a different internal clearance in order to work properly. Bearings with less internal clearance are available commercially for ultra-precision applications.

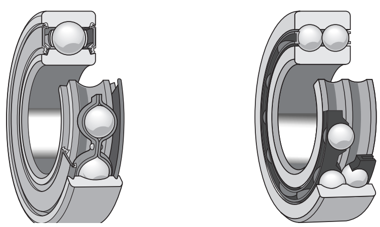
Basic dynamic load rating is the load for which each bearing is expected to meet its life expectancy. Static load rating is the maximum load the bearing can withstand without internal damage. These load ratings are discussed in more detail later in this section.

Starting torque is the minimum torque required to start a bearing rotating from rest. For small instrument bearings, starting torque and bearing friction is of concern. The addition of seals will increase starting torque as well as running friction. Values for starting torques and friction calculations can be found in manufacturer's data.

BEARING TYPES

The most common bearings in light industrial machine design are deep groove, angular contact, cylindrical roller, and spherical roller bearings. Deep groove ball bearings are by far the most commonly used type of rolling element bearing. The following are descriptions of some of the bearings found in machinery.

Deep groove ball bearings (Figure 10-3) are extremely versatile and are available fully lubricated and sealed. They can accommodate some axial loading in both directions and are capable of high speeds. They can be used as the locating bearing in bearing arrangements. Deep groove ball bearings typically can accommodate angular misalignment between 2 and 10 minutes of arc. Verify this with the manufacturer for every bearing. Double row deep groove ball bearings are used when the load capacity of single row deep groove ball bearings is insufficient. Their ability to handle axial loads and misalignment is about the same as that of single row bearings.



Single Row Double Row

Figure 10-3: Deep Groove Ball Bearings

Source: SKF USA Inc. (www.skf.com)

Self-aligning ball bearings (Figure 10-4) were invented by SKF USA Inc. These bearings have a double row of balls that ride in a spherical raceway. This type of bearing is capable of high speeds and is particularly suited for applications with significant angular misalignment. Misalignment up to 3 degrees is allow-

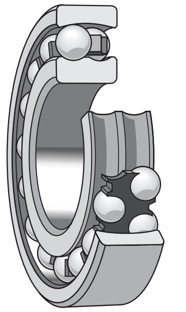


Figure 10-4: Self-Aligning Ball Bearing

Source: SKF USA Inc. (www.skf.com)

able with some bearings. Verify this with the manufacturer for every bearing. Self-aligning ball bearings are available with seals.

Angular contact ball bearings (Figure 10-5) are designed to handle combined loading. They are available in single and double row configurations. A single row angular contact ball bearing can handle axial loading in only one direction. These bearings are normally applied in a pair, arranged back to back, and adjusted against each other to provide a stiff arrangement that can handle axial loads in both directions, as well as radial loads and tilting moments. Angular contact ball bearings have a very limited ability to accommodate angular misalignment.

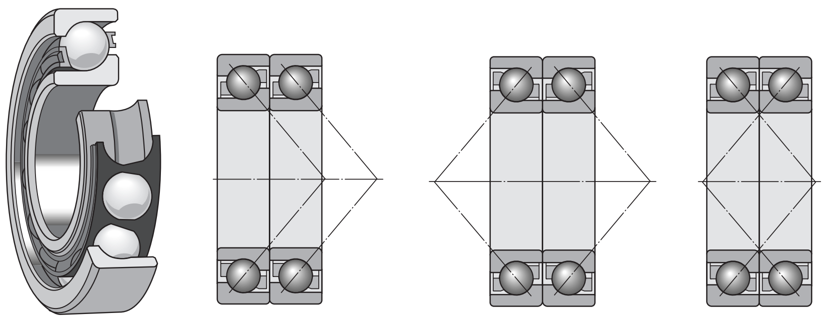


Figure 10-5: Angular Contact Ball Bearings

Source: SKF USA Inc. (www.skf.com)

Four-point contact ball bearings (Figure 10-6) are single-row, angular contact ball bearings that have special raceways designed to allow the bearing to support axial loads in both directions. These bearings are meant for applications with predominantly axial loads and can support only limited radial loading.

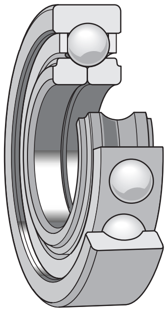


Figure 10-6: Four-Point Contact Ball Bearing

Source: SKF USA Inc. (www.skf.com)

Cylindrical roller bearings (Figure 10-7) are available in many different configurations. The most common configuration consists of a single row of cylindrical rollers, caged, with an inner race that is axially displaceable. These bearings are suitable for very heavy radial loads and can operate at high speeds. Cylindrical roller bearings are not suitable for axial loads and can accommodate angular misalignment up to only a few minutes of arc. These bearings are generally not used to locate a shaft axially, but special designs are available that allow the bearing to be used as a locating bearing.

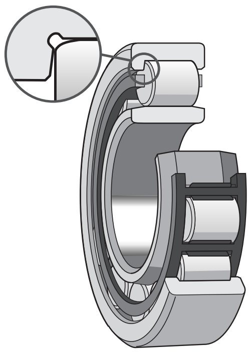


Figure 10-7: Cylindrical Roller Bearing

Source: SKF USA Inc. (www.skf.com)

Spherical roller bearings (Figure 10-8) are comprised of a double row of rollers within a spherical raceway. They are capable of accommodating significant angular misalignment up to about 3 degrees. Verify this with the manufacturer for every bearing. These bearings can handle heavy radial and axial loads acting in both directions. Sealed versions are available.

Tapered roller bearings (Figure 10-9) are available in a variety of configurations. These bearings have tapered rollers that are arranged at an angle to the

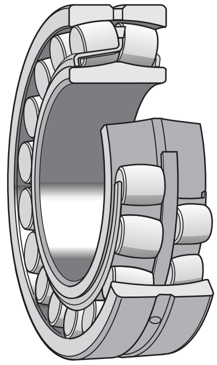


Figure 10-8: Spherical Roller Bearing

Source: SKF USA Inc. (www.skf.com)

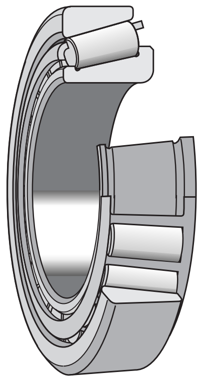


Figure 10-9: Tapered Roller Bearing

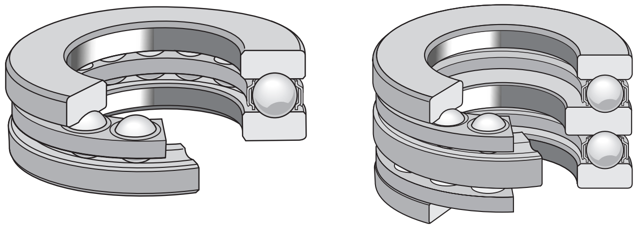
Source: SKF USA Inc. (www.skf.com)

bore. Tapered roller bearings are particularly suitable for significant combined loading, but can accommodate axial loads in only one direction. Single-row tapered roller bearings are generally separable and are most often used in pairs arranged such that axial loads in both directions can be borne. Single-row tapered roller bearings can accommodate angular misalignment only up to a few minutes of arc. These bearings are usually preloaded at installation and require special care to install and run in.

Needle roller bearings are ideal for applications where radial space is limited. They have rollers that are very long and thin. These bearings have high radial load capacities, but cannot tolerate axial loads. These bearings are very intolerant of angular misalignment. Needle roller bearings are sometimes used without an inner race. This provides an even more compact bearing solution, but special care must be taken in the manufacture of the shaft that will become the inner race.

Figure 10-10: Ball Thrust Bearings

Source: SKF USA Inc. (www.skf.com)



Ball thrust bearings (Figure 10-10) are designed for axial loads only. Single-direction thrust bearings have a single row of balls and can handle axial loading in only one direction. Double-direction thrust bearings, as the name suggests, can handle axial loading in both directions and have two rows of balls. These bearings are generally separable.

Cylindrical roller thrust bearings have cylindrical rollers and are designed to handle axial loads acting in one direction only. These bearings are ideal for heavy axial loading and are fairly compact.

Spherical roller thrust bearings use sphered rollers arranged at an angle to the shaft axis to transmit loading. This enables the bearing to handle both axial and radial loading. These bearings are self-aligning and can handle angular misalignments up to 3 degrees. Verify this with the manufacturer for every bearing.

Needle roller thrust bearings are ideal for supporting heavy axial loads in a minimum of axial space. These bearings are similar to cylindrical roller thrust bearings in that they are intolerant of misalignment, cannot support radial loads, and are available in single-direction or double-direction versions. When space is at an absolute premium, the needles and cage can be used without upper and lower races. In this situation, the abutting parts must be carefully manufactured to serve as races.

Some common application conditions are listed in Table 10-3 with one or more suitable bearing types. Table 10-4 lists some of the commonly available bearing types and a few key characteristics. There are many more bearing types and options available. It is important to note that rolling element bearings are less appropriate than plain bearings for applications where the angle of rotation is small; for example, a shaft that undergoes a small oscillating motion. Rolling element bearings in such an application would lubricate and wear unevenly, causing premature failure.

Table 10-3: Applications and Suitable Bearings

Application Condition	Typical Bearing Types
High Misaligment	Self-Aligning Ball Bearings, Spherical Roller Bearings
Bearing Used To Locate Shaft Axially	Angular Contact Bearings, Deep Groove Ball Bearings (Light axial load only)
High Speeds (Radial Loads Only)	Deep Groove Ball Bearings, Self-Aligning Ball Bearings
High Speeds (Combined Loads)	Angular Contact Bearings
Light to Moderate Radial Loads	Deep Groove Ball Bearings
Heavy Radial Loads	Cylindrical Roller Bearings, Needle Roller Bearings
Combined Loads, Radial Dominant	Angular Contact Bearings, Tapered Roller Bearings
Combined Loads, Axial Dominant	Spherical Roller Thrust Bearings
Light to Moderate Axial Loads	Thrust Ball Bearings, Four-Point Contact Bearings
Heavy Axial Loads	Roller Thrust Bearings
Moment Loads	Paired Angular Contact Ball Bearings, Face-To-Face Tapered Roller Bearing Pair
Axial Displacement Expected	Cylindrical Roller Bearings, Needle Roller Bearings
Low Profile	Needle Roller Bearings
Maximum Stiffness	Cylindrical Roller Bearings, Tapered Roller Bearings, Needle Roller Bearings

Table 10-4: Selected Bearing Types and Characteristics

Key:
● **Characteristic of Bearing**
• **Lesser Characteristic of Bearing**
○ **Characteristic of Some Configurations**

		Characteristics of Bearing									
Bearing Type	**Typical Application**	Axial Load Capability	Radial Load Capability	Moment Load Capability	Accommodates Misalignment	Internal Axial Displacement	High Stiffness	Seals Available	Low Profile	Separable	Tapered Bore
Deep Groove Ball Bearing	General Use. Moderate Radial, Light Axial Loads	•	•		•			●			
Deep Groove Ball Bearing (Double Row)	Heavy Radial Loads	•	●								
Self-Aligning Ball Bearing	High Misalignment	•	•		●			●			●
Angular Contact Ball Bearings	Unidirectional Combined Loads	•	•		•						
Angular Contact Pair (Back To Back)	Bidirectional Combined Loads	•	•	•				○			
Four-Point Contact Ball Bearings	Light Axial Loads	•								●	
Cylindrical Roller Bearings	Heavy Radial Loads		●		•	●	●			●	
Full Complement Cylindrical Roller Bearings	Very Heavy Radial Loads		●		•	●	●			○	
Needle Roller Bearings	Low Profile		●			●	●	●	●	●	
Spherical Roller Bearings	High Misalignment, Heavy Combined Loads	●	●		●			●			●
Taper Roller Bearings	Unidirectional Combined Loads	•	●		•		●			●	
Taper Bearing Pair (Face to Face)	Bidirectional Combined Loads	•	●	●	•		●			●	
Thrust Ball Bearings	General Use, Thrust	•								●	
Cylindrical Roller Thrust Bearings	Heavy Loads, Thrust	●								●	
Needle Roller Thrust Bearings	Low Profile, Thrust	●							●	●	
Spherical Roller Thrust Bearings	High Misalignment, Thrust	●	•		●					●	

RADIAL BALL AND ROLLER BEARING DIMENSIONS AND TOLERANCES

Rolling element bearing dimensions and tolerances are heavily standardized. The Anti-Friction Bearing Manufacturers Association (AFBMA) and ISO have standards that are in agreement. Bearings from different manufacturers are generally interchangeable as a result. The Annular Bearing Engineers Committee (ABEC) defines the standard tolerances for bearings in the United States, and ISO standards are in agreement for metric bearings. Radial ball and roller bearings have five standard classes of tolerances, which are given in Table 10-5. Higher precision levels yield a longer life expectancy, but come

Table 10-5: Select Radial Bearing Tolerance Classes

Application	ANSI/ABMA Standard 20 Tolerance Class		ISO 492 Tolerance Class
	Ball Bearings	Roller Bearings	
Standard	ABEC 1	RBEC 1	Normal
Semi-Precision	ABEC 3	RBEC 3	Class 6
Precision	ABEC 5	RBEC 5	Class 5
High Precision	ABEC 7	RBEC 7	Class 4
Ultra Precision	ABEC 9	RBEC 9	Class 2

at a higher cost. Most machinery applications will use ABEC 1 class bearings. Applications that require extra smooth and accurate shaft running may require ABEC 5 or even ABEC 7 bearings. For tolerance values in each class, consult the recommended resources.

Machine designers will primarily be concerned with the dimensions they have control over: shaft diameter and housing bore diameter. In general, a light press fit on the rotating race is sought for bearing mounting. The rotating race is generally the inner race, but not always. It is essential to apply the appropriate fits to the shaft and housing because the bearing internal clearance in operation relies on the proper mounting fits. For bearings with an ABEC 1 (standard) tolerance class, Tables 10-6 and 10-7 are useful for determining the appropriate tolerances for the shaft and housing bore. For other types of bearings, such as needle roller bearings, consult with the manufacturer on tolerance information. It is typical for manufacturers to specify tolerances on bearing features and recommend tolerances for mating parts. These tolerances are typically very small (a few ten-thousandths of an inch) and are the responsibility of the machine designer.

Bearing seats should generally be ground and have a surface roughness of 16 to 32 microinches (0.4 to 0.8 micrometers). Maximum fillet dimensions for shoulders and steps are normally provided for each bearing in the manufacturer's catalog. Larger fillet radii reduce stress concentrations, so be sure to take advantage of the fillet allowance.

Proper geometric tolerancing of the shaft and housing bearing seats is essential for proper bearing fit and function. Both the diameters and the perpendicular abutments must be tightly controlled. Recommended geometric tolerances for metric bearing seats and abutments are shown in Table 10-8. IT (International Tolerance) grades are provided; they can be used to determine the numerical tolerances. The numerical values for tolerance grades are provided in Section 3.1 of this book. For cylindricity and total radial runout, the numerical tolerance represented by the tolerance grade is divided by 2 as shown.

Table 10-6: Shaft Fits for Radial Bearings (ABEC/RBEC 1)

Numerical values for tolerance designations are given in Section 3.1

Operating Conditions			Ball Bearings		Cylindrical Roller Bearings		Spherical Roller Bearings		ISO Tolerance Designation (a)
			mm	inch	mm	inch	mm	inch	
Inner Ring Stationary w/Respect to Load	**All Loads**	Inner ring has to be easily displaceable	All Diameters	All Diameters	All Diameters	All Diameters	All Diameters	All Diameters	g6
		Inner ring does not have to be easily displaceable	All Diameters	All Diameters	All Diameters	All Diameters	All Diameters	All Diameters	h6
Inner Ring Rotating w/Respect to Load or Direction of Load Indeterminate	**RADIAL LOAD**		**NOMINAL SHAFT DIAMETER**						
	LIGHT: Loads up to 0.075C (C = dynamic load rating)		≤ 18	≤ 0.71					h5
			> 18	> 0.71	≤ 40	≤ 1.57	≤ 40	≤ 1.57	j6 (b)
					(40) -140	(1.57) - 5.51	(40) - 100	(1.57) - 3.94	k6 (b)
					(140) - 320	(5.51) - 12.6	(100) - 320	(3.94) - 12.6	m6 (b)
	NORMAL: Ball: 0.075C to 0.15C Cylindrical Roller: 0.075C to 0.2C Spherical Roller: 0.075C to 0.25C (C = dynamic load rating)		≤ 18	≤ 0.71					j5
			> 18	> 0.71	≤ 40	≤ 1.57	≤ 40	≤ 1.57	k5
					(40) - 100	(1.57) - 3.94	(40) - 65	(1.57) - 2.56	m5
					(100) - 140	(3.94) - 5.51	(65) - 100	(2.56) - 3.94	m6 (b)
					(140) - 320	(5.51) - 12.6	(100) - 140	(3.94) - 5.51	n6
					(320) - 500	(12.6) - 19.7	(140) - 280	(5.51) - 11.0	p6
	HEAVY: Loads over 0.15C (C = dynamic load rating)		(18) - 100	(0.71) - 3.94					k5
			>100	> 3.94	≤ 40	≤ 1.57	≤ 40	≤ 1.57	m5
					(40) - 65	(1.57) - 2.56	(40) - 65	(1.57) - 2.56	m6 (b)
					(65) - 140	(2.56) - 5.51	(65) - 100	(2.56) - 3.94	n6 (b)
					(140) - 200	(5.51) - 7.87	(100) - 140	(3.94) - 5.51	p6 (b)
					(200) - 500	(7.87) - 19.7	(140) - 200	(5.51) - 7.87	r6 (b)
					> 500	> 19.7	> 200	> 7.87	r7 (b)
Pure Thrust Load			All Diameters	All Diameters	Consult Manufacturer				j6 (b)

a) Tolerances are shown for solid steel shafts. For hollow or nonferrous shafts, tighter fits may be needed.

b) When greater accuracy is required, use j5, k5, and m5 instead of j6, k6, and m6, respectively.

Table 10-7: Housing Bore Fits for Radial Bearings (ABEC/RBEC 1)

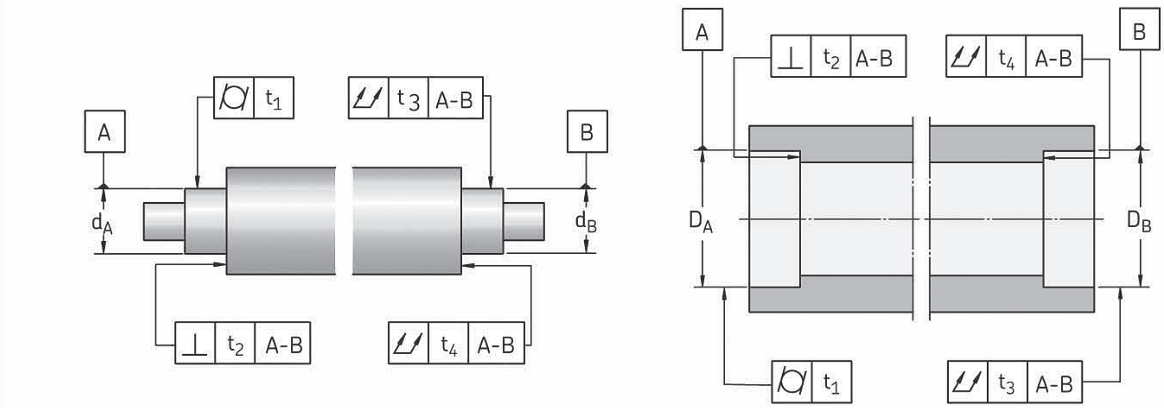
Numerical values for tolerance designations are given in Section 3.1

<table>
<tr><th colspan="4">Design and Operating Conditions</th><th rowspan="2">ISO Tolerance Designation (a)</th></tr>
<tr><th>Rotational Conditions</th><th>Loading</th><th>Outer Ring Axial Displacement Limitations</th><th>Other Conditions</th></tr>
<tr><td rowspan="4">Outer Ring Stationary w/Respect to Load</td><td rowspan="3">Light, normal, and heavy</td><td rowspan="3">Outer ring must be easily displaceable axially</td><td>Heat input through shaft</td><td>G7</td></tr>
<tr><td>Housing split axially</td><td>H7 (b)</td></tr>
<tr><td rowspan="4">Housing not split axially</td><td>H6 (b)</td></tr>
<tr><td>Shock with temporary complete unloading</td><td rowspan="5">Transitional range (c)</td><td rowspan="3">J6 (b)</td></tr>
<tr><td rowspan="3">Load Direction is Indeterminate</td><td>Light and normal</td></tr>
<tr><td>Normal and heavy</td></tr>
<tr><td>Heavy shock</td><td rowspan="3">Split housing not recommended</td><td>K6 (b)</td></tr>
<tr><td rowspan="3">Outer Ring Rotating w/Respect to Load</td><td>Light</td><td>M6 (b)</td></tr>
<tr><td>Normal and heavy</td><td rowspan="2">Outer ring need not be axially displaceable</td><td>N6 (b)</td></tr>
<tr><td>Heavy</td><td>Thin wall housing not split</td><td>P6 (b)</td></tr>
</table>

a) For cast iron or steel housings. For nonferrous alloy housings tighter fits may be needed.
b) Where wider tolerances are permissible, use tolerances P7, N7, M7, K7, J7, and H7
c) The tolerance zones are such that the outer ring may be either tight or loose in the housing.

Table 10-8: Bearing Seat Geometric Tolerances

Accuracy of form and position for bearing seatings on shafts and in housings



Surface Characteristic	Symbol for characteristic	tolerance zone	**Permissible deviations** Bearings of tolerance class[1)] Normal CLN	P6	P5	
Cylindrical seating						
Cylindricity	⌭	t_1	IT5/2	IT4/2	IT3/2	IT2/2
Total radial runout	⌰	t_3	IT5/2	IT4/2	IT3/2	IT2/2
Flat abutment						
Rectangularity	⊥	t_2	IT5	IT4	IT3	IT2
Total axial runout	⌰	t_4	IT5	IT4	IT3	IT2

Explanation

For normal demands

For special demands with respect to running accuracy or even support

[1)]For bearings of higher accuracy (tolerance class P4, etc.), please consult with the manufacturer.

Source: SKF USA Inc. (www.skf.com)

BEARING ARRANGEMENTS

Bearing arrangements can be broken into two basic types: preloaded and non-preloaded. Most bearing arrangements found in light industrial machinery will be of the non-preloaded variety. Preloaded bearing arrangements are often found in machine tools and other extremely high accuracy applications. Preloading is also called for when a bearing is run under extremely light loading conditions at high speeds. Preloading may be accomplished using springs or by adjusting two bearings against each other, using spacers, until zero operational clearance is achieved. Preloading is beyond the scope of this text, but is detailed in the recommended resources.

A non-preloaded bearing arrangement generally includes two bearings and associated hardware that support and locate a shaft, both radially and axially. Normally just one bearing, called the locating bearing, is tasked with axial location of the shaft in both directions. The locating bearing (or bearing combination) necessarily must be able to handle combined axial and radial loading. Suitable locating bearings include deep groove ball bearings when axial loading is light, angular contact bearings, spherical roller bearings, or a matched pair of tapered roller bearings. The second (non-locating) bearing usually constrains the shaft only radially. This type of arrangement requires that the locating bearing be axially and radially fixed on both the shaft and housing, while the other is allowed to float in the axial direction. Allowing one bearing to float axially prevents overloading of the bearings due to shaft expansion. If the non-locating bearing is of a type that allows axial displacement within the bearing, as do some types of cylindrical roller bearings, the bearing races may be axially constrained without fear of overloading.

Figure 10-11a shows a bearing arrangement consisting of a deep groove ball bearing and a cylindrical roller bearing capable of internal axial displacement. In this situation, both bearings may be fully captivated. Figure 10-11b shows an arrangement using two deep groove ball bearings. In this case, the non-locating bearing is free to move in the housing and allow axial displacement. Figure 10-11c illustrates an arrangement using two opposed angular contact bearings to locate the shaft axially, while a deep groove bearing is allowed to move axially in its housing. Figure 10-11d shows a pair of tapered roller bearings locating and supporting a shaft. In this type of arrangement, the bearing housings must be axially adjusted relative to one another to achieve proper preload on the bearings.

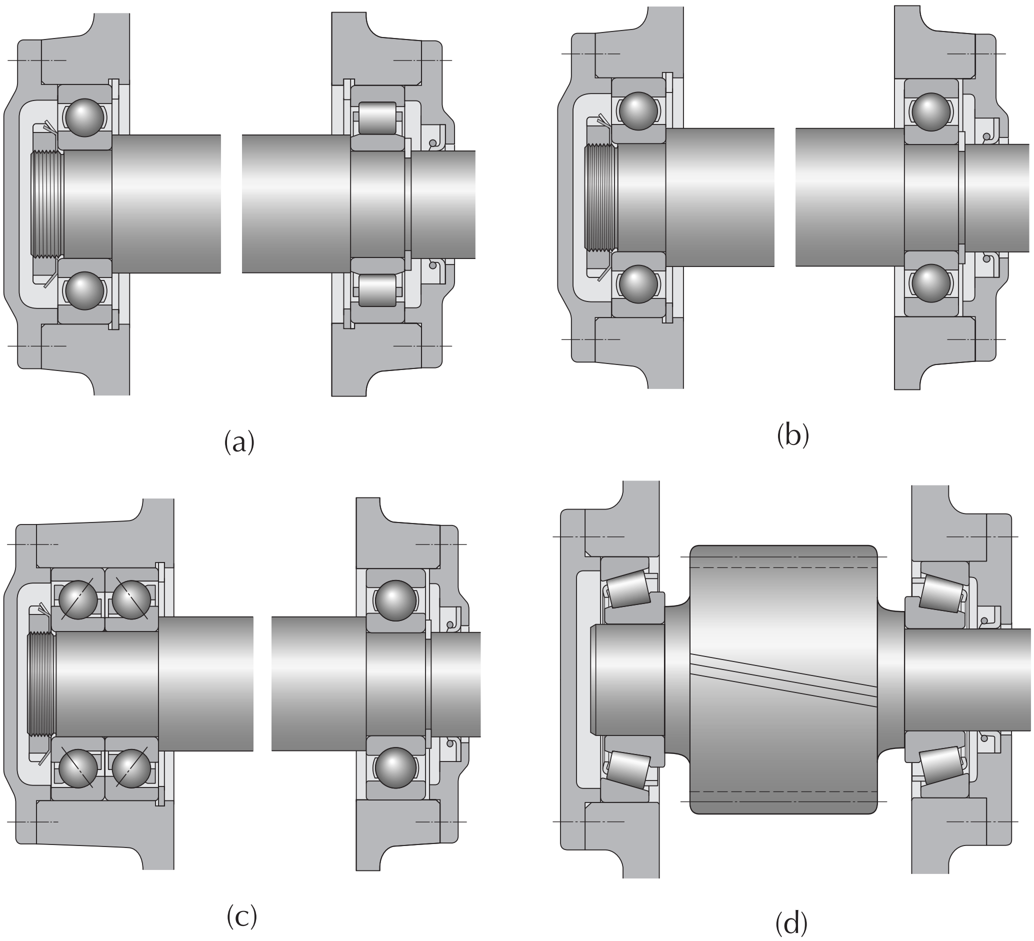
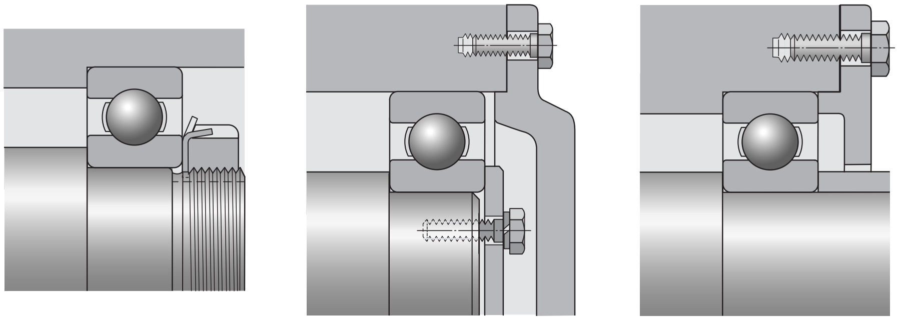


Figure 10-11: Typical Bearing Arrangements

Source: SKF USA Inc. (www.skf.com)

Axial constraint of bearings cannot be accomplished merely through the bearing fit on the shaft and in the housing. When a bearing with a cylindrical bore must be located axially on a shaft, this is usually accomplished using a shoulder on the shaft. Another component, such as a nut, collar, or retaining ring, is used to captivate the bearing race against the shaft shoulder. This location method is illustrated in Figure 10-12a and b. The shoulder and other hardware must be properly sized to fully support the inner race without interfering with bearing rotation. The outer race of a bearing is normally captured between a shoulder in the housing bore and a clamping ring attached to the housing (Figure 10-12c). Bearings with tapered bores are less commonly used and require the use of adapter sleeves and spacers to mount and locate properly. Consult the recommended resources for information on installation of tapered bore bearings.



(a) (b) (c)

Figure 10-12: Typical Axial Bearing Location Methods
Source: SKF USA Inc. (www.skf.com)

LOADS ON BEARINGS

Axial and radial load carrying capacities are highly variable among different types of bearings. Many radial bearings have very limited axial load carrying capability. Some bearings can carry axial loads in only one direction. The size and direction of applied loads is of primary concern when selecting a bearing type.

When selecting and sizing a bearing, one must first quantify the axial and radial loads seen by the bearing during operation. Section 11.1 of this text includes some formulas for calculating loads on a shaft for common machine elements. Two types of loading on each bearing must be evaluated: static loading and dynamic loading. Static loading must take into account any maximum loads seen by the bearing, even if only for a short length of time. Dynamic loading considers normal operation loads. In both cases, an equivalent load must be calculated for the bearing. The equivalent static and dynamic loads are then compared with catalog static and dynamic load ratings to select the appropriate bearing.

Equivalent load is a function of both the loading on the bearing and the bearing configuration. Pure thrust bearings have an equivalent axial load equal to the applied axial load, and the radial load must be zero. Purely radial bearings have an equivalent radial load equal to the applied radial load and the axial load must be zero. Bearings designed for combined loading, like angular contact bearings, are configured such that applied axial loads induce radial loads, and applied radial loads induce axial loads. This means that even a purely axial or radial load condition will result in combined loading on the bearing. In many applications, a combined axial and radial applied load is present. Based on the bearing type and configuration, an equivalent load must

Table 10-9: Loads On Rolling Element Bearings

F_R = applied radial load	F_A = applied axial (thrust) load	
FLUCTUATING LOADS	Speed, load direction constant: $F_R = \dfrac{F_{R\min} + 2F_{R\max}}{3}$ $F_A = \dfrac{F_{A\min} + 2F_{A\max}}{3}$	Load direction rotating: $F_R = f_m(F_c + F_r)$ F_c = Constant direction load F_r = Rotating direction load $0.75 \le f_m \le 1$ (See manufacturer)
RADIAL BEARINGS AND SPHERICAL ROLLER BEARINGS	**STATIC EQUIVALENT LOAD** W = Consult manufacturer's catalog Z = Consult manufacturer's catalog	$F_{MAX} = WF_R + ZF_A$ If $F_{MAX} < F_R$ then $F_{MAX} = F_R$
	DYNAMIC EQUIVALENT LOAD $V = 1$ if inner bearing race rotates $V = 1.2$ if outer bearing race rotates e = Consult manufacturer's catalog X = Consult manufacturer's catalog Y = Consult manufacturer's catalog	Radial Load Only: $F = VF_R$ Combined Radial And Thrust Loads: $F = VXF_R + YF_A$ Test For Relatively Small Thrust: If $\dfrac{F_A}{VF_R} \le e$ Then $X = 1$, $Y = 0$
ANGULAR CONTACT BEARINGS AND TAPERED ROLLER BEARINGS	**EQUIVALENT LOAD** Note: Radial loads on angular contact and tapered roller bearings induce axial loads. Consult with the manufacturer for axial load calculation methods.	Depends on bearing geometry. Consult Manufacturer
THRUST BEARINGS	**EQUIVALENT LOAD**	Axial (Thrust) Load Only: $F = F_A$ Radial or Combined Loads: Consult manufacturer

be calculated from the applied loads. The formulas in Table 10-9, in combination with the manufacturer's catalog data, can be used to accomplish this.

Most bearings must be subjected to a minimum load in order to function properly (have the rolling elements roll inside the bearing rather than slide) and reach their expected lifespans. In most cases, the weight of the components is enough to meet this minimum load requirement. When dealing with very light components and loads, consult the manufacturer for calculation methods for minimum load requirements. A rule of thumb is to apply a minimum load of 0.01C to ball bearings and 0.02C to roller bearings. C is the dynamic load rating for the bearing, which is provided in catalogs and calculated in Table 10-10.

The radial loads used in Table 10-9 are assumed to be in a constant direction. When a load varies, its mean value must be determined and used as the load value in life and dynamic load calculations. When a load fluctuates while its direction is fairly constant, its mean value can be estimated using the equation in Table 10-9. If the direction of the load or part of the load rotates, as with an unbalance condition, the mean load can be calculated using the equation in Table 10-9.

BEARING LOAD RATINGS AND LIFE EXPECTANCY

Life expectancy is a statistical phenomenon. Basic dynamic load rating is typically defined as the load at which 90% of a population of identical bearings will complete 1 million revolutions without failure. This may vary among manufacturers, so it is essential to verify the assumptions behind any catalog information. Most manufacturers follow the standard ISO 281: 1990 which contains bearing life and dynamic load calculation methods. Life calculations based on the standard can be found in Table 10-10. Part of load calculation involves applying a service factor (Table 10-12) which accounts for shock loading in service.

Table 10-10: Life and Load Ratings for Radial and Angular Contact Bearings

$L =$ Life expectancy in revolutions $F =$ Equivalent load on bearing	Ball Bearings: $k = 3$ Roller Bearings: $k = 3.33$
BASIC STATIC LOAD C_0 = Required basic static load rating SF_0 = Factor of safety (See Table 10-11) F_{MAX} = Maximum equivalent load on bearing	$C_0 = (SF_0)F_{MAX}$
BEARING LOAD VS. LIFE	$\frac{L_2}{L_1} = \left(\frac{F_1}{F_2}\right)^k$
LIFE EXPECTANCY h = Lifetime hours of operation N = Speed in revolutions per minute For Variable Loads R_1 = Number of revolutions under Load 1 L_1 = Lifetime (revolutions) under Load 1 R_T = Total revolutions expected from bearing	Conversion from lifetime In hours : $L = 60hN$ Variable loading: $L = \dfrac{1}{\frac{R_1}{R_T L_1} + \frac{R_2}{R_T L_2} + \ldots}$
BASIC DYNAMIC LOAD $L_{rated} = 10^6$ revolutions typically Always verify rated life value K_S = Shock service factor (Table 10-12) These factors can be applied if the load rating is based on 90% reliability: $K_R = 0.62$ for 95% reliability $K_R = 0.33$ for 98% reliability $K_R = 0.21$ for 99% reliability	Dynamic load using rated (usually 90%) reliability: $C = FK_S \sqrt[k]{\frac{L}{L_{rated}}}$ (If $L = L_{rated}$, $C = FK_S$) Dynamic load using adjusted (>90%) reliability: $C = FK_S \sqrt[k]{\frac{L}{K_R L_{rated}}}$

Table 10-11: Static Load Safety Factors for Bearings

	Rotating Bearings						Non-Rotating Bearings	
	Quiet Running Importance							
	Low		Normal		High			
Application Conditions	Ball Bearings	Roller Bearings	Ball Bearings	Roller Bearings	Ball Bearings	Roller Bearings	Ball Bearings	Roller Bearings
Smooth, No Vibration	0.5	1	1	1.5	2	3	0.4	0.8
Normal	0.5	1	1	1.5	2	3.5	0.5	1
Significant Shock Loads	≥ 1.5	≥ 2.5	≥ 1.5	≥ 3	≥ 2	≥ 4	≥ 1	≥ 2

Table 10-12: Shock Service Factors for Bearings

Application Conditions	Typical Range of K_S	
	Ball Bearing	Roller Bearing
Smooth, Uniform Load	1	1
Gearing	1.0 - 1.3	1
Light Impact	1.2 - 1.5	1.0 - 1.1
Moderate Impact	1.5 - 2.0	1.1 - 1.5
Heavy Impact	2.0 - 3.0	1.5 - 2.0

Loads used in life calculations are assumed to be constant and radial to the rolling elements. When calculating the required dynamic load rating for an application, the equivalent load (Table 10-9) on the bearing must be used rather than the applied loading. Load ratings in catalogs are usually based on an assumed bearing material and hardness. Typical assumptions are chromium steel bearing material and a hardness of 58 Rc. If alternative materials are required, load ratings may be affected.

The effects of friction and lubricant viscosity affect the operating temperature and life of the bearing. Bearing failure in the field is usually due to contamination, misalignment, and lubrication failure. As a result, actual life expectancy will be shorter than the rated value. A modified life calculation is available in standard ISO 281:1990/Amd 2:2000 to account for contamination and lubrication conditions. This method is quite complicated and is not presented here. Drag caused by seals can also be important in critical applications. The manufacturer can assist in calculating lubrication or seal drag.

BEARING SELECTION PROCEDURE FOR A ROTATING SHAFT APPLICATION

Rotating shaft applications are the most common type of bearing selection situations. The best way to properly select and size a rolling element bearing is to speak with the manufacturer. Most bearing manufacturers have engineers on their staff ready to analyze your application thoroughly and make recommendations. When selecting a bearing for this type of application, the following procedure can be used:

1. Determine the design parameters. These include:
   - Expected lifetime of the bearing in hours or revolutions
   - Speed of the bearing
   - Bearing arrangement, defining the locating bearing(s)
   - Running load on the bearing, broken into axial and radial components
   - Expected shock load
   - Mounting and sealing requirements
   - Required precision
2. Select a bearing type based on performance or characteristics. Tables 10-3 and 10-4 can be useful when selecting a bearing type. One of the primary considerations will be whether radial, axial, or combined loading must be accommodated.
3. Select a bearing size from a catalog based on dynamic load rating. Calculate the equivalent load on the bearing using the equations in Table 10-9 and the manufacturer's catalog. Then calculate the required basic dynamic load needed for the application and required reliability level using the formulas in Table 10-10. Basic dynamic load rating (C) is normally given in catalogs. Select a bearing with a dynamic load rating that exceeds the calculated dynamic load for the application by a factor of safety. A factor of safety for dynamic load as high as 5 is not out of the question.
4. Check the basic static load rating for the chosen bearing. Using an appropriate factor of safety (Table 10-11) and the equations in Tables10-9 and 10-10, calculate the basic static load for the application and compare it to the basic static load rating for the chosen bearing. Basic static load rating (C_0) is normally given in catalogs. Account for any peak loads including impact loads.

5. Check the limiting and rated operating speeds for the bearing. Speed in operation should never exceed the limiting speed of the bearing. Normal running speed should never exceed the rated operating speed.
6. Select bearing options such as seals, shields, lubricant, etc.
7. Select bearing accessories such as nuts, locking washers, spacers, wrenches, etc. The geometry for the shaft and housing should be finalized and all hardware accounted for on the bill of materials.

CRITICAL CONSIDERATIONS: Rolling Element Bearings

- An appropriate factor of safety must be used when evaluating static and dynamic loading of bearings.
- Equivalent load, not applied load, must be used when selecting bearings based on static and dynamic load ratings.
- Most bearings have some “play” or internal clearance. If this is unacceptable, consider preloaded bearings or stiffer bearings (like needle roller bearings).
- Be sure not to axially constrain any bearing that cannot tolerate axial loading. This could cause the bearing to become overloaded if expansion occurs in the assembly due to the heat of running.
- Lubrication is essential to bearing life. Bearings used in oscillating applications may have reduced life due to inadequate lubricant distribution.
- Over-lubrication causes bearings to fail. Observe the manufacturer’s lubrication instructions. Calibrated grease guns, automatic lubrication systems, or ultrasonic monitoring can be used to prevent improper lubrication.
- Bearings should have a light press fit on the rotating race. If the fit is too loose, the bearing may rotate in/on its mount and cause wear and particulate contamination. If the fit is too tight, the bearing may be damaged, or internal clearance could be compromised. Tables 10-6 and 10-7 can be used to select proper fits on standard bearings.
- Tight geometric tolerancing is required on bearing support surfaces like bores, shafts, and seating surfaces. If the bearing is not fully and evenly supported, premature bearing wear and failure can occur. It is typical for manufacturers to specify these tolerances.
- Fine surface finishes are required on all bearing support surfaces. Rough surfaces can break down and cause fits to loosen.
- Care must always be used when installing bearings. Contamination and shock loading during installation must be avoided. Press bearings on with steady, even pressure around the entire race that is being fitted. Pressing forces should never be applied across the rolling elements in the bearing (i.e. never press on the inner race by applying force to the outer race, etc.).

BEST PRACTICES: Rolling Element Bearings

- Consult the manufacturer for best results when designing a bearing application.
- Whenever possible, axial loading should be carried by thrust bearings. Use one thrust bearing per direction of axial load.
- The use of lubricated, sealed bearings simplifies the assembly design.
- Use standard precision class bearings (ABEC 1) for most applications.
- Design provisions for bearing dismounting if possible. Slots for bearing removal tools, grease fittings and channels for pressure dismounting, and holes for jacking screws are all useful.
- Use a reputable bearing distributor or manufacturer to avoid counterfeit components. Counterfeit bearings are a big business.
- Avoid using rolling element bearings in locations where heavy vibration occurs while the bearing is not rotating. This condition can cause deformation and wear in the bearing.
- Wave springs can be used to lightly bias a bearing in a housing or on a shaft with axial clearance.

LINEAR BEARINGS

Section 10.3

There are many types of linear bearings. Some examples are plain linear bushings, linear ball bushings, linear rails, and ball splines. Plain bearings are simple and cost effective, but are limited in speed and life by friction between the sliding elements. Ball or roller bearings replace sliding with rolling contact for higher speeds, load ratings, and life expectancies. Rollers tend to have higher load ratings than balls and are used in high load applications. For all linear bearings, life expectancy is measured in linear travel distance and is determined by component materials, hardness, lubrication, and load conditions.

RECOMMENDED RESOURCES

- Consult a manufacturer for more information on linear bearings.

LUBRICATION

Plain linear bearings are generally used in a state of boundary lubrication. Grease or other thick lubricants are applied to the shaft and must be reapplied periodically to reduce friction and extend bearing life. Contaminants have easy access to linear plain bearings, so proper precautions or soft bearing materials must be used. For more information on boundary lubrication and plain bearing materials, see Section 10.1.

Rolling element linear bearings usually incorporate internal lubrication pathways into the bearing housing. Grease fittings are usually present on the bearing housing for lubrication purposes. These bearings often have contact seals at the shaft to reduce lubricant loss and prevent contaminants from entering the housing. Linear bearings should be lubricated periodically since some lubricant loss is to be expected.

PLAIN LINEAR BEARINGS

Plain linear bearings are essentially the same as rotary plain bearings in design, materials, and static pressure ratings. See Section 10.1 for more information on these aspects of plain bearings. Linear plain bearings can take a variety of forms, including round sleeves, flat plates, and other shapes. In many cases, a linear plain bearing arrangement may be that of a slide and slideway; this arrangement allows movement along only one axis. It is not uncommon for both slide and slideway to be made out of hardened tool steel for long life and coated with dry film lubricants as well as thoroughly greased. Heavy loads may be carried with some configurations, and high precision achieved as long as wear is kept to a minimum. Heavy duty slide and slideways are found on machining centers, and light duty versions are commonly found on assembly machines and fixtures. Many versions are commercially available, but these are also easily designed in-house if needed. Some common slide and slideway configurations are shown in Figure 10-13, and some commercial plain bearing slide assemblies are shown in Figure 10-14.

Plain linear sleeve bearings are very common in industry. These bearings will allow rotation of the bearing on the shaft if torque is applied. It is important to prevent moment loading of the bearing because that will result in uneven wear and reduced life. Shafts for plain linear bearings are generally hardened and ground steel, and must be precisely aligned if used in parallel.

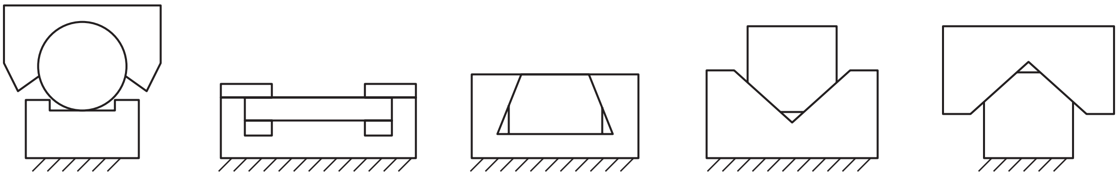


Figure 10-13: Slide and Slideway Configurations

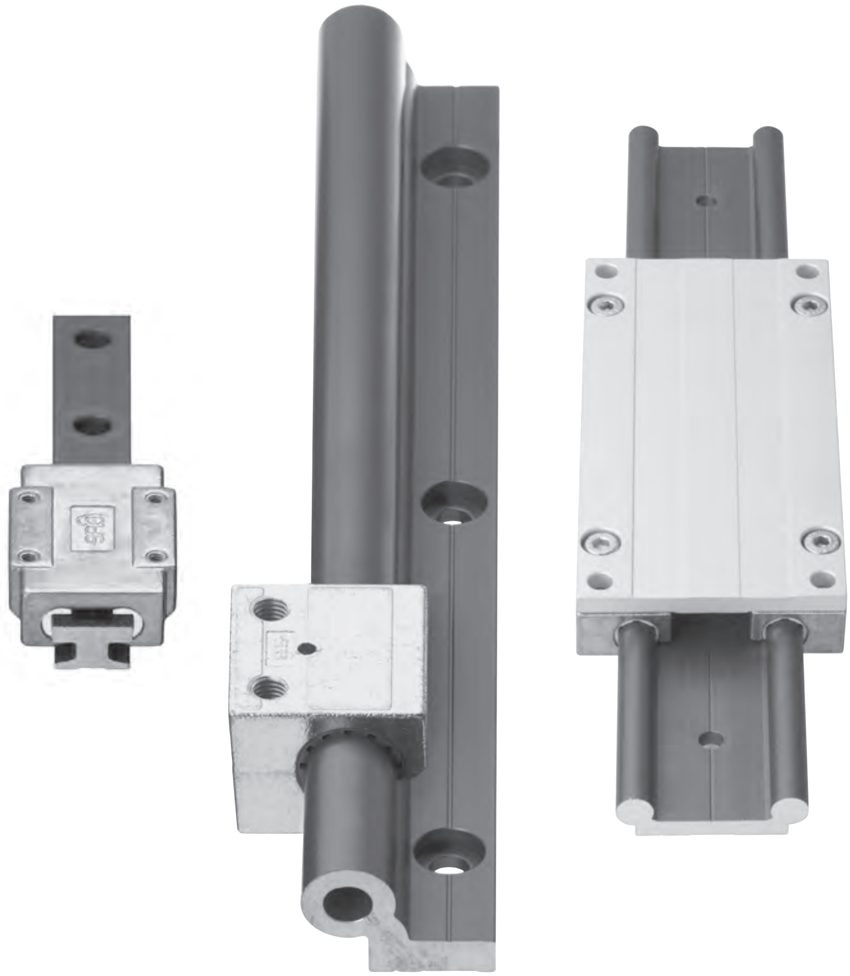


Figure 10-14: Commercial Plain Bearing Slide Assemblies

Source: igus Inc. (www.igus.com)

Commercial linear plain bearings are normally selected based on PV factor, similarly to rotary plain bearings. However, linear plain bearings dissipate heat much more quickly than rotating bearings do. As a result, the PV factor rating for a linear plain bearing will normally be much higher than its rotary counterpart. A plain bearing that is meant for linear movement should be clearly labeled as such in the manufacturer's catalog and should have a linear PV factor.

Boundary lubrication is the norm with linear plain bearings. Selection and sizing of linear plain bearings is performed similarly to selection and sizing of plain sleeve bearings for rotating service. See Section 10.1 for that procedure, using the formulas in Table 10-13 for PV calculation.

Table 10-13: PV Formulas for Plain Linear Bearings

PV FACTOR	PV factor = P x V
INCH UNITS $P =$ bearing pressure (psi) $F =$ load on the bearing (lbf) $A =$ bearing area (in^2) $D =$ bearing internal diameter (in) or shaft external diameter $L =$ bearing length (in)	$P = \frac{F}{A}$ $A = DL$ $V =$ bearing surface speed (ft/min)
METRIC UNITS $P =$ bearing pressure (MPa) $F =$ load on the bearing (N) $A =$ bearing area (m^2) $D =$ bearing internal diameter (m) or shaft external diameter $L =$ bearing length (m)	$P = \frac{F}{A(10^6)}$ $A = DL$ $V =$ bearing surface speed (m/min)

ROLLING ELEMENT LINEAR BEARINGS

For rolling element linear bearings, rolling element bearing concepts and calculations discussed in Section 10.2 apply. When specifying linear bearings, the main criteria are bearing precision, internal clearance, static load rating, dynamic load rating, speed, and life expectancy. Equivalent load must be used when calculating dynamic load, and manufacturer's catalogs must be consulted in doing so. The life of linear bearings is often given in linear distance traveled. The ISO standard, which is commonly used by most manufacturers, is a dynamic loading life expectancy of 50 km. The formulas in Table 10-14 can be used to calculate dynamic load rating for linear bearings.

There are many types of rolling element linear bearings available commercially. The following are some common types and their descriptions.

Table 10-14: Life and Load Ratings for Linear Bearings

$L =$ life expectancy in linear distance $F =$ equivalent load on bearing	Ball Bearings: $k = 3$ Roller Bearings: $k = 3.33$
BASIC STATIC LOAD $C_0 =$ required basic static load rating $SF_0 =$ factor of safety (See Table 10-11) $F_{MAX} =$ maximum equivalent load on bearing	$C_0 = (SF_0)F_{MAX}$
BEARING LOAD VS. LIFE	$\frac{L_2}{L_1} = \left(\frac{F_1}{F_2}\right)^k$
LIFE EXPECTANCY $h =$ lifetime hours of operation $S =$ Stroke length $n =$ number of strokes per minute For Variable Loads $R_1 =$ distance under Load 1 $L_1 =$ lifetime (distance) under Load 1 $R_T =$ total distance expected from bearing	Conversion from lifetime In hours $L = 60h(2Sn)$ Variable loading: $L = \dfrac{1}{\dfrac{R_1}{R_T L_1} + \dfrac{R_2}{R_T L_2} + \ldots}$
BASIC DYNAMIC LOAD $L_{rated} = 50$ km typically Always verify rated life value $K_S =$ shock service factor (Table 10-12) These factors can be applied if the load rating is based on 90% reliability: $K_R = 0.62$ for 95% reliability $K_R = 0.33$ for 98% reliability $K_R = 0.21$ for 99% reliability	Dynamic load using rated reliability: $C = FK_S \sqrt[k]{\dfrac{L}{L_{rated}}}$ (If $L = L_{rated}$, $C = FK_S$) Dynamic load using adjusted reliability: $C = FK_S \sqrt[k]{\dfrac{L}{K_R L_{rated}}}$

Ball bushings are very commonly used and consist of a sleeve containing ball bearings that ride on a smooth shaft. They are generally meant for light loading conditions. Some designs allow rotation of the bearing on the shaft (Figure 10-15), while most are not intended to rotate. In general, torque loading must be avoided. Ball bushings are often used in pairs, running on parallel shafts. Ball bearings in the sleeve are usually re-circulated to provide continuous rolling contact with the shaft. Ball bushings are available in both closed and open configurations, with and without lubrication-preserving end caps. Some examples of these are shown in Figures 10-16 and 10-17. A closed sleeve is used with end supported shafts while an open sleeve is used with continuously supported shafts. Preloaded ball bushings are commercially available. Ball bushings have a limited capacity to carry moment loads (about an axis perpendicular to the bushing/shaft axis). Moment load ratings are often given for ball bushings, and using them in pairs is a common solution to eliminate moment loading. Longer bushings generally have greater moment load capacities.

Figure 10-15: Rotary Linear Ball Bushings

Source: Misumi USA Inc. (www.misumiusa.com)

Figure 10-16: Ball Bushings

Source: Misumi USA Inc. (www.misumiusa.com)

Figure 10-17: Open Ball Bushings

Source: Misumi USA Inc. (www.misumiusa.com)

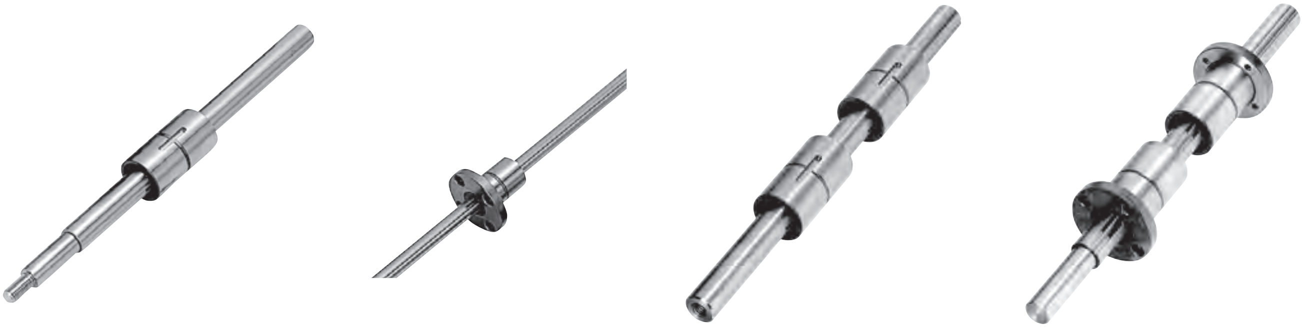


Figure 10-18: Ball Splines
Source: Misumi USA Inc. (www.misumiusa.com)

Ball splines (Figure 10-18) are recommended when the assembly must resist rotation of the bearing on the shaft under torsional loading. Engagement of the ball bearings within the sleeve with axial splines on the shaft prevents rotation. Ball splines are often rated in terms of allowable torque load, and are generally designed to transmit torque while allowing linear movement. Ball splines have a limited capacity to carry moment loads (about an axis perpendicular to the bushing/shaft axis). Moment load ratings are often given for ball splines, and using them in pairs is a common solution to eliminate moment loading. In general, longer spline nuts can carry greater moment loads.

Linear rails and guides (Figure 10-19) are a compact solution to the problem of preventing rotation while allowing linear movement. These assemblies normally consist of a rectangular rail and a bearing carriage that rides along the rail. In general, the carriage or guide is captive to the rail and can support loads in every

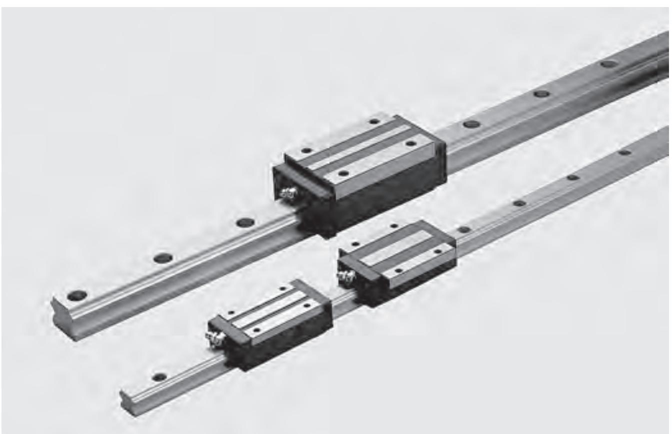


Figure 10-19: Linear Rail and Guide Assemblies
Source: SBC Linear Co., Ltd. (sbclinear.co.kr/eng/index.php) Courtesy of TPA (www.tpa-us.com)

direction. Consult the manufacturer for specific load carrying capabilities because these vary between models. These assemblies can generally carry moment loads as well, though this must be verified for each model. If large moment loads are present, the use of two guides on one rail, or two rails side by side should be considered. Both ball and roller versions are available commercially.

Crossed roller ways (Figures 10-20 and 10-21) are highly compact and rigid assemblies consisting of two angled rows of rollers between two linear races. They are capable of carrying very heavy loads. These assemblies can withstand significant moment loads and are often used as table bearings in machine tools. The rollers do not re-circulate, so movement is generally smoother than that of recirculating ball bushings. Due to their configuration, crossed roller ways must be twice as long as their expected stroke.

Extended Load Capacity

Needle Rollers

Curved Crossed Roller Way

Figure 10-20: Crossed Roller Ways

Source: PM Bearings (www.pmbearings.nl) Courtesy of TPA (www.tpa-us.com)

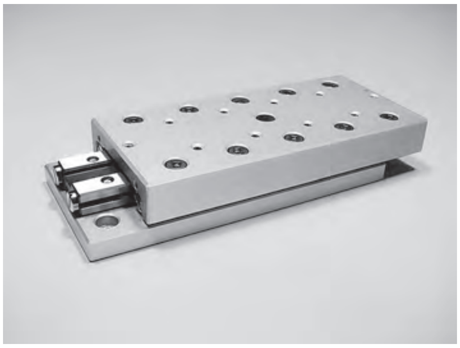



Figure 10-21: Crossed Roller Slide Assemblies
Source: PM Bearings (www.pmbearings.nl) Courtesy of TPA (www.tpa-us.com)

Selection and sizing of linear rolling element bearings is performed in a similar way to the sizing and selection of rotating bearings. See Section 10.2 for that procedure, and use the formulas in Table 10-14 to calculate dynamic load ratings for linear bearings.

CRITICAL CONSIDERATIONS: Linear Bearings

- Use an appropriate factor of safety when performing bearing calculations.
- Proper alignment is critical to linear bearing function and life. When two linear bearings are used in parallel, overconstraint must be avoided.
- Lubrication is essential to long bearing life, and contamination of bearings must be avoided.

BEST PRACTICES: Linear Bearings

- Consult the manufacturer for best results when selecting and sizing linear bearings.
- Avoid applying moment loads to linear bearings.
- Prevent contamination of linear bearings as much as possible.

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