SECTION 4

SECTION 4

The information in this section will assist the design engineer in the selection of the ball bearing products that best suit critical application requirements for performance, life and cost. Early involvement by NMB Sales and Application Engineers is recommended. Engineering support services available from the company's engineering laboratories are described together with special testing capabilities.

Size, materials, component parts and lubrication alternatives are discussed in this section. These are followed by a detailed analysis of the important considerations which should be evaluated simultaneously when choosing the proper bearing for a particular design. Emphasis is also placed on the operations and aftermarket services available to the designer for installation and use of the bearings after delivery.

Engineering Information

Authorized Distributor:

PIB Pacific International Bearing, Inc.
800.228.8895 • www.pibsales.com

Designing To Lower Total Cost

The majority of applications can be effectively handled using a "standard bearing". A "standard bearing", in this case, refers to bearing that is in such worldwide demand that large volumes are produced. This virtually guarantees continuity of supply while assuring pricing benefits for the O.E.M. Selection of a "standard bearing" at the design stage cannot be over emphasized. The considerations necessary to design for lower cost include:

- Dimensional size
- Material type
- Lubrication
- Enclosures
- Cage style (retainer)
- Manufacturability
- Assembly and fits
- Packaging
- Quality requirements

Although different designers may vary in their approach to bearing selection, the following is one method that works well.

- Establish operating, environmental and performance requirements such as load, speed, noise, etc.
- Select a bearing configuration to meet the above requirements.

  Some examples of configuration types are:

  1. Flanged or unflanged
  2. With or without a snap ring
  3. Ball complement/size

- Determine bearing envelope to accommodate shaft and housing requirements. This step is critical to cost. It is quite often more cost-effective to design the housing and shaft around a popular bearing size than vice versa.
- Specify enclosures as necessary. Be careful not to specify a more expensive enclosure than necessary to perform properly in the application.
- Specify required cage type. For the majority of cases, the standard cage for a particular chassis size will be adequate.
- Determine the bearing noise rating that is required for the application. For most cases, our standard "No Code" noise rating will provide quieter operation than most other components in the system. For extremely noise sensitive applications, a quieter noise rating can be specified.
- Determine degree of precision needed to achieve the performance requirements (ABEC Level). Do not over estimate what is truly necessary to achieve the desired performance.
- Determine the radial play specification. The standard radial play specification for a chassis size will be adequate to handle normal press fits, moderate temperature differentials and normal speeds.
- Determine lubrication requirements. This should include lubrication characteristics and the amount of lubricant needed. This is a critical step in the performance and reliability of the bearing in the application.

Care should be taken throughout this process with respect to both cost and performance. The key in designing for the lowest total cost is to involve the Sales and Application Engineering staff early in the selection process. Costs will be impacted greatly if the envelope dimensions are not given consideration at the time of bearing selection. NMB offers an experienced Sales and Engineering staff to help in the design and selection process insuring your success.

NMB Technical Center

The NMB Technical Center, a newly built, cutting edge testing facility located in Wixom, MI is designed to advance the function and performance of NMB products with customer applications. This facility supplies customer with application specific validation, as well as market leading technical information and services. The NMB Technical Center provides testing and analysis capabilities in the following areas:

- Equipment simulates a customer specific application to determine the expected performance of the component in real world conditions. Provides optimized component design and selection to maximize the performance and benefit under different conditions.
- Environmental testing analyzes product and application performance under varying environmental conditions such as temperature, humidity, altitude, contamination, corrosion and vibration. Provides baseline comparative tests of bearing and motor components such as lubricants, fits and sealing mechanisms to provide a database of performance characteristics. This capability can shorten the design time to reach an optimal bearing or motor selection.
- Chemistry testing analyzes the chemical makeup and condition of components and the interactions of various materials. Provides detailed analysis of lubricants and non-metallic parts to improve product performance.
- Metrology equipment measures NMB products and related components to determine their effect on system performance, including physical dimension, form, surface finish and roundness.
- Noise and vibration test equipment determines the audible noise and vibration of components and systems to improve noise characteristics of the application.
- Metallurgy tests determine the hardness, microhardness and microstructure of NMB and customer components.

Engineering Test Laboratory

NMB maintains a fully equipped Engineering Test Laboratory where we can confirm the performance characteristics of our ball bearing designs. NMB has a full complement of commercially available equipment such as Talysurfs, Talyronds and Anderonmeters, running and starting torque testers, and real time analyzers. In addition, we have developed our own specialized state-of-the-art equipment, precisely tailored to our own requirements.

Typical of this equipment is a specially designed anechoic chamber, that includes a spindle for rotating ball bearings under loaded conditions. This can be used with a sonic analyzer to measure and record airborne noise, vibration and structureborne vibration.

Materials Laboratory

Our Materials Laboratory has been specifically designed and equipped to perform complex chemical, metallurgical, and visual analysis of the many component parts in ball bearings. Besides internal projects, this laboratory can also perform wear and failure studies on a customer's bearings.

Modern chemical analysis of organic compounds is usually carried out on a dual-beam infrared spectrophotometer. Likewise, alloy composition is determined with x-ray defraction spectrography and non-destructive test methods.

Metallurgical studies can be done with metallograph and micro-hardness testers. The metallograph will perform microstructure photography at magnification from 25 to 2000 times. Micro-hardness testers investigate surface effects and alloy homogeneity using diamond indentation under loads from 1 to 10,000 grams.

During bearing inspection and failure analysis, ball bearings are disassembled and examined under a laminar flow hood. Many findings can be recorded permanently with a photo-microscope for analysis and future reference.

Functional Tests For Ball Bearings

We have devised a series of rigidly monitored tests to insure that every bearing we manufacture will meet our commitment to quality and reliability. Our testing procedures measure dimensional characteristics, radial play and noise performance.

A bearing envelope and internal tolerance will not always reveal how the bearing will perform under dynamic conditions. NMB has developed "noise ratings" to assure exact bearing performance.

Every motor quality bearing produced is evaluated. During the assembly process, anderonmeters test for bearing noise and vibration. The bearings are tested under a controlled load and speed to meet their particular noise specification. This procedure allows the user to know how the bearings will perform under dynamic conditions.

Starting torque defines the effort required to initiate bearing rotation. This is a prime concern to ball bearing users. It can be a critical factor in applications requiring multiple low speed, start/stop movements.

Running torque is a measure of effort required to maintain rotation, under a certain load, after rotation has been initiated. NMB has the capability to perform running torque tests under a variety of conditions, ranging from 1-7,000 rpm with various applied thrust loads. NMB can customize tests based on specific application requirements. Tests may be fully monitored and analyzed for various ball bearing characteristics.

Accurate testing of ball bearings requires the tester to closely simulate the actual operating conditions of the intended application. Please consult an NMB Sales Engineer or a member of the Applications Engineering staff for their recommendations on the many specialized tests we can perform.

INTERNAL BEARING GEOMETRY

When designing ball bearings for optimum performance, internal bearing geometry is a critical factor. For any given bearing load, internal stresses can be either high or low, depending on the geometric relationship between the balls and raceways inside the ball bearing structure.

When a ball bearing is running under a load, force is transmitted from one bearing ring to the other through the ball set. Since the contact area between each ball and the rings is relatively small, even moderate loads can produce stresses of tens or even hundreds of thousands of pounds per square inch. Because internal stress levels have such an important effect on bearing life and performance, internal geometry must be carefully chosen for each application so bearing loads can be distributed properly.

Definitions

Raceway, Track Diameter, and Track Radius

The raceway in a ball bearing is the circular groove formed in the outside surface of the inner ring and in the inside surface of the outer ring. When the rings are aligned, these grooves form a circular track that contains the ball set.

The track diameter and track radius are two dimensions that define the configuration of each raceway. Track diameter is the measurement of the diameter of the imaginary circle running around the deepest portion of the raceway, whether it be an inner or outer ring. This measurement is made along a line perpendicular to, and intersecting, the axis of rotation. Track radius describes the cross section of the arc formed by the raceway groove. It is measured when viewed in a direction perpendicular to the axis of the ring. In the context of ball bearing terminology, track radius has no mathematical relationship to track diameter.
The distinction between the two is shown in Figure 1.

Radial and Axial Play

Most ball bearings are assembled in such a way that a slight amount of looseness exists between balls and raceways. This looseness is referred to as radial play and axial play. Specifically, radial play is the maximum distance that one bearing ring can be displaced with respect to the other, in a direction perpendicular to the bearing axis, when the bearing is in an unmounted state. Axial play, or end play, is the maximum relative displacement between the two rings of an unmounted ball bearing in the direction parallel to the bearing axis. Figure 2 illustrates these concepts.

Since radial play and axial play are both consequences of the same degree of looseness between the components in a ball bearing, they bear a mutual dependence. While this is true, both values are usually quite different in magnitude.

In most ball bearing applications, radial play is functionally more critical than axial play. If axial play is determined to be an essential requirement, control can be obtained through manipulation of the radial play specification. Please consult with Application Engineering if axial play ranges for a particular chassis size are required.

Some general statements about Radial Play:

1. The initial contact angle of the bearing is directly related to radial play- the higher the radial play, the higher the contact angle. The chart on the following page gives nominal values under no load.
2. For support of pure radial loads, a low level of radial play is desirable; where thrust loading is predominant, higher radial play levels are recommended.
3. Radial play is affected by any interference fit between the shaft and bearing I.D. or between the housing and bearing O.D. See the Assembly and Fitting Procedure section on page 4-15 for more details.

Also, since the actual play remaining after assembly of the complete device is the important condition, the radial play specification for the bearing itself must be modified in accordance with the discussion on page 4-15. If the system spring rate is critical, or if extremes of temperature or thermal gradient will be encountered, consult with our Engineering Department prior to design finalization.

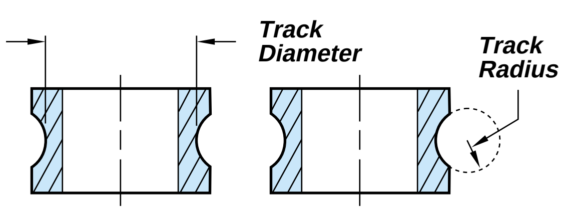


Figure 1. The distinction between track radius and track diameter (inner ring).

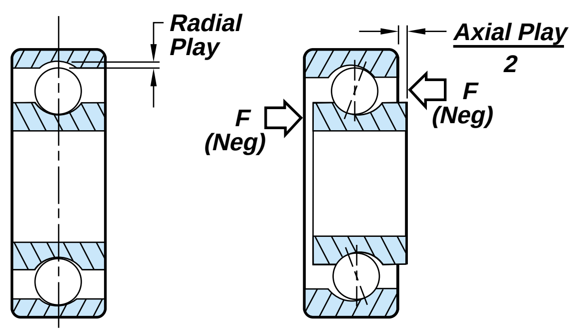


Figure 2. The distinction between radial play and axial play.

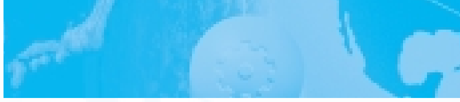


Table Of Contact Angles α_o

Ball Size D_W	RADIAL PLAY CODE P25	RADIAL PLAY CODE P58
.025	18°	$24\frac{1}{2}$°
1/32 & 0.8 mm	$16\frac{1}{2}$°	22°
1mm	$14\frac{1}{2}$°	20°
3/64	14°	18°
1/16	12°	16°
3/32	$9\frac{1}{2}$°	13°
1/8	$12\frac{1}{2}$°	17°
9/64	12°	16°
5/32	11°	15°
3/16	10°	14°

The contact angle is given for the mean radial play of the range shown i.e., for P25 (.0002" to .0005") - contact angle is given for .00035". Contact angle is affected by race curvature. For your specific application, contact NMB Engineering.

Typical radial play ranges are:

Description	Radial Play Range	NMB Code
Tight	.0001" to .0003"	P13
Normal	.0002" to .0005"	P25
Loose	.0005" to .0008"	P58

Raceway Curvature

Raceway curvature is an expression that defines the relationship between the arc of the raceway's track radius and the arc formed by the slightly smaller ball that runs in the raceway. It is simply the track radius of the bearing raceway expressed as a percentage of the ball diameter. This number is a convenient index of "fit" between the raceway and ball. Figure 3 illustrates this relationship.

Track curvature values typically range from approximately 52 to 58 percent. The lower percentage, tight fitting curvatures are useful in applications where heavy loads are encountered. The higher percentage, loose curvatures are more suitable for torque sensitive applications. Curvatures less than 52 percent are generally avoided because of excessive rolling friction that is caused by the tight conformity between the ball and raceway. Values above 58 percent are also avoided because of the high stress levels that can result from the small ball-to-raceway conformity at the contact area.

Contact Angle

The contact angle is the angle between a plane perpendicular to the ball bearing axis and a line joining the two points where the ball makes contact with the inner and outer raceways. The contact angle of a ball bearing is determined by its free radial play value, as well as its inner and outer track curvatures.

The contact angle of thrust-loaded bearings provides an indication of ball position inside the raceways. When a thrust load is applied to a ball bearing, the balls will move away from the median planes of the raceways and assume positions somewhere between the deepest portions of the raceways and their edges. Figure 4 illustrates the concept of contact angle by showing a cross sectional view of a ball bearing that is loaded in pure thrust.

Free Angle and Angle of Misalignment

As a result of the previously described looseness, or play, which is purposely permitted to exist between the components of most ball bearings, the inner ring can be cocked or tilted a small amount with respect to the outer ring. This displacement is called the free angle of the bearing, and corresponds to the case of an unmounted bearing. The size of the free angle in a given ball bearing is determined by its radial play and track curvature values. Figure 5 illustrates this concept.

For the bearing mounted in an application, any misalignment present between the inner and outer rings (housing and shaft) is called the angle of misalignment. The misalignment capability of a bearing can have positive practical significance because it enables a ball bearing to accommodate small dimensional variations which may exist in associated shafts and housings. A maximum angle of misalignment of 1/4° is recommended before bearing life is reduced. Slightly larger angles can be accommodated, but bearing life will not be optimized.

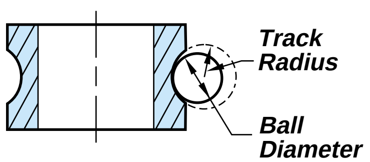


Figure 3. The relationship of track radius to ball diameter.

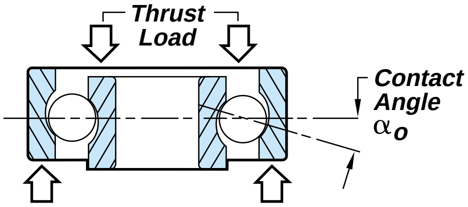


Figure 4. Contact angle for bearing loaded in pure thrust.

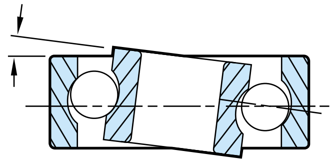


Figure 5. Free angle of the bearing.

MATERIALS/CAGES/RETAINERS

Bearing Materials

Chrome Steel

Bearing steel used for standard ball bearing applications in uses and in environments where corrosion resistance is not a critical factor.

52100 or Equivalent

The most commonly used ball bearing steel in such applications is SAE 52100 or its equivalent. Due to its structure, this is the material chosen for extreme noise sensitive applications.

MKJ3* Chrome Steel

Developed by NMB's parent company, MKJ3 is a high carbon chromium bearing steel combined with a heat treating process. This steel has a higher hardness and a more stable structure than standard chrome bearing steel. This allows the steel to retain its shape under adverse conditions. For bearings designated with the KJ part number, the bearing race material is MKJ3, while the balls are made of standard 52100 or equivalent. KJ bearings were developed for use in hard disk drive and other specialty applications where the running accuracy performance is crucial. The combination of materials used with the KJ designation results in a bearing that will have high shock load resistance, high load carrying capacity, and will resist increased sound levels with extended use.

Stainless Steel

DD400™ 0.7% C; 13% Cr

A 400 series Martensitic stainless steel combined with a heat treating process was exclusively developed by NMB's parent company. Miniature and instrument bearings manufactured from "DD™" Martensitic stainless steel, or "DD Bearings™", meet the performance specifications of such bearings using AISI 440C Martensitic stainless steel, and it is equal to or superior in hardness, superior in low noise characteristics, and is at least equivalent in corrosion resistance. These material characteristic advantages make for lower torque, smoother running, and longer life bearings.

*US and foreign patents pending

The retainer, also referred to as the cage or separator, is the component part of a ball bearing that separates and positions the balls at approximately equal intervals around the bearing's raceway. There are two basic types that we supply: the crown or open end design and the ribbon or closed ball pocket design. The most common retainer is the two-piece closed pocket metal ribbon retainer.

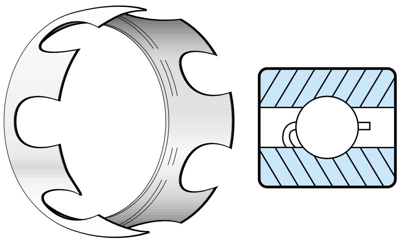


Figure 1. Standard one-piece crown retainer.

The Open End design, or crown retainer, as shown in Figure 1 is of metal material. The metal retainer, constructed of hardened stainless steel, is very light-weight and has coined ball pockets which present a hard, smooth, low-friction contact surface to the balls. A feature of this assembly is its smooth running characteristic. Crown retainers manufactured from molded plastics are available for some sizes. Plastic molded nylon retainers are advantageous when application speeds are high relative to the particular bearing used. For special retainer requirements, please consult a member of our Sales Engineering or Applications Engineering Department.

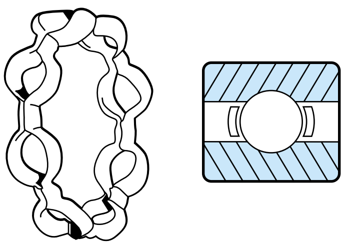


Figure 2. Two-piece closed pocket metal ribbon retainer.

Closed Pocket Design (two-piece construction). The two-piece closed pocket design, as outlined in Figure 2 with clinching tabs, is our standard design for most miniature and instrument size ball bearings. The use of loosely clinched tabs is favorable for starting torque, and the closed pocket design provides good durability required for various applications.

For special retainer requirements, please consult a member of our Sales Engineering or Applications Engineering Department.

SHIELD AND SEAL TYPES

Shields and seals are necessary to provide optimum ball bearing life by retaining lubricants and preventing contaminants from reaching central work surfaces. NMB can manufacture ball bearings with several types of protective closures that have been designed to satisfy the requirements of most applications. Different types of closures can be supplied on the same bearing and nearly all are removable and replaceable. They are manufactured with the same care and precision that goes into our ball bearings. The following are descriptions of the most common types of shields and seals we can supply. Please consult a member of the company's Sales Engineering or Applications Engineering staff for information on the availability of special designs that may be suited to your specific applications.

Z & H Type Shields

"Z" and "H" type shields designate non-contact metal shields. "Z" type shields are the simplest form of closure and, for most bearings, are removable. "H" type shields are similar to "Z" types but are not removable.

It is advantageous to use shields rather than seals in some applications because there are no interacting surfaces to create drag. This results in no appreciable increase in torque or speed limitations and operation can be compared to that of open ball bearings.

Contact Seals

"D" type seals consist of a molded Buna-N lip seal with an integral steel insert. While this closure type provides excellent sealing characteristics, several factors must be considered for its application. The material normally used on this seal has a maximum continuous operating temperature limit of 250°F. Although it is impervious to many oils and greases, consideration must be given to lubrication selection. It is also capable of providing a better seal than most other types by increasing the seal lip pressure against the inner ring O.D. This can result in a higher bearing torque than with other type seals and may cause undesirable seal lip heat build-up in high speed applications.

The DSD64 and the DSD21 type seals have the same operating characteristics as the "D" type seal, resulting in the same considerations of temperature limitations and lubricant selection. The DSD64 seal is comprised of a double-lip contact rubber seal with a stepped inner ring similar to the "D" type seal. The double-lip contact design configuration offers additional protection from extreme environments such as liquid contamination or high-pressure drops accross the bearing. The DSD21 type seal is comprised of a contact rubber seal combined with a labyrinth designed inner ring. The labyrinth design configuration creates an extended path to the raceway, combined with a contact seal, minimizes the tendency for contaminates to enter the bearing.

Non-Contact Seals

"S" type seals are constructed in the same fashion as the "D" type seals. This closure type has the same temperature limitation of 250°F. It also is impervious to many oils and greases, but the same considerations should be noted on lubrication selection. The "S" type seal is uniquely designed to avoid contact on the inner ring land, significantly reducing torque over the "D" type configuration.

"L" type seals are fabricated from glass re-inforced teflon. When assembled, a very small gap exists between the seal lip and the inner ring O.D. It is common for some contact to occur between these components, resulting in an operating torque increase. The nature of the seal material serves to keep this torque increase to a minimum. In addition, the use of this material allows high operating temperatures with this configuration.

The SSD21 type seals have the same operating characteristics as the "D" and "S" type seals, resulting in the same considerations of temperature limitation and lubricant selection. The SSD21 type seal is comprised of a non-contact rubber seal combined with a labyrinth designed inner ring, while the DSD21 type seal is the contact seal version with the labyrinth inner ring. The labyrinth design configuration creates an extended path to the raceway minimizing the tendency for contaminants to creep into the ball bearing.

If you have any questions concerning the performance of NMB Technologies Corporation seals in special environments or high speed applications, please contact a member of our Sales Engineering or Applications Engineering staff.

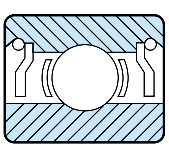


Figure 1. Two "Z" Shields (removable)

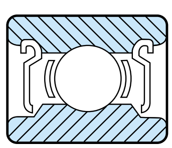


Figure 2. Two "H" Shields (non-removable)

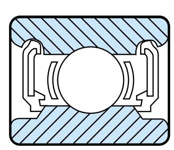


Figure 3. Two "D" Seals (contact rubber)

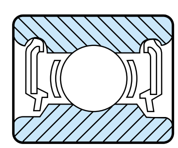


Figure 4. Two "S" Seals (non-contact rubber)

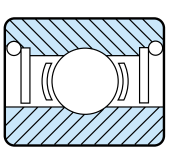


Figure 5. Two "L" Seals (non-flexed teflon)

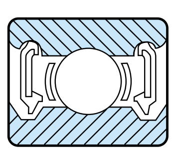


Figure 6. Two "SSD21" Seals (labyrinth design seal)

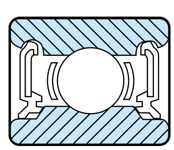


Figure 7. Two "DSD64" Double-lip Seals (contact seal)

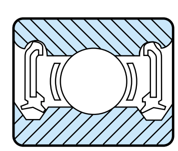


Figure 8. Two "DSD21" Labyrinth Seals (light contact)

Lubricant Types

Oil

Oil is the basic lubricant for ball bearings. Previously most lubricating oil was refined from petroleum. Today, however, synthetic oils such as diesters, silicone polymers, and fluorinated compounds have found acceptance because of improvements in properties. Compared to petroleum base oils, diesters in general have better low temperature properties, lower volatility, and better temperature/viscosity characteristics. Silicones and fluorinated compounds possess even lower volatility and wider temperature/viscosity properties.

Virtually all petroleum and diester oils contain additives that limit chemical changes, protect the metal from corrosion, and improve physical properties.

Grease

Grease is an oil to which a thickener has been added to prevent oil migration from the lubrication site. It is used in situations where frequent replenishment of the lubricant is undesirable or impossible. All of the oil types mentioned in the next section can be used as grease bases to which are added metallic soaps, synthetic fillers and thickeners. The operative properties of grease depend almost wholly on the base oil. Other factors being equal, the use of grease rather than oil results in higher starting and running torque and can limit the bearing to lower speeds.

Oils and Base Fluids

Petroleum Mineral Lubricants

Petroleum lubricants have excellent load carrying abilities and are naturally good against corrosion, but are useable only at moderate temperature ranges (-25° to 250°F). Greases that use petroleum oils for bases have a high dN ($\frac{OD+bore}{2}$ in mm X speed in rpm) capability. Greases of this type would be recommended for use at moderate temperatures, light to heavy loads and moderate to high speeds.

Super-Refined Petroleum Lubricants

While these lubricants are usable at higher temperatures than petroleum oils (-65° to 350°F), they still exhibit the same excellent load carrying capacity. This further refinement eliminates unwanted properties, leaving only the desired chemical chains. Additives are introduced to increase the oxidation resistance, etc.

Synthetic Lubricants

The esters, diesters and poly-α-olefins are probably the most common synthetic lubricants. They do not have the film strength capacity of a petroleum product, but do have a wide temperature range (-65° to 350°F) and are oxidation resistant.

Synthetic hydrocarbons are finding a greater use in the miniature and instrument ball bearing industry because they have proved to be a superior general purpose lubricant for a variety of speeds, temperatures and environments.

Silicone Lubricants

Silicone products are useful over a much wider temperature range (-100° to 400°F), but do not have the load carrying ability of petroleum types and other synthetics. It has become customary in the instrument and miniature bearing industry, in recent years, to derate the dynamic load rating (C_r) of a bearing to 1/3 of the value shown in this catalog if a silicone product is used.

Perfluorinated Polyether (PFPE)

Oils and greases of this type have found wide use where stability at extremely high temperatures and/or chemical inertness are required. This specialty lubricant has excellent load carrying capabilities but its inertness makes it less compatible to additives, and less corrosion resistant.

Lubrication Methods

Grease packing to approximately one quarter to one third of a ball bearing's free volume is one of the most common methods of lubrication. Volumes can be controlled to a fraction of a percent for precision applications by special lubricators. In some instances, customers have requested that bearings be lubricated 100% full of grease. Excessive grease, however, is as detrimental to a bearing as insufficient grease. It causes shearing, heat buildup, unnecessarily high torque, and deterioration through constant churning which can ultimately result in bearing failure.

Centrifuging an oil lubricated bearing removes excess oil and leaves only a very thin film on all surfaces. This method is used on very low torque bearings and can be specified by the customer for critical applications.

Operating Speed

When petroleum or synthetic ester oils are used, the maximum speed N_{max} is dictated by the ball cage material and design or the centrifugal ball loads rather than by the lubricant.

For speed limit values N_{max}, the N_{max}/f_n values shown on the product listing pages must be multiplied by f_n values found on the following table.

fn vs. Cage, Lubricant Types and Ring Rotation

	Metal Cage		Acetal			
	2-Piece or Crown Type		Crown Type		Full Section Type	
Lubricant / Ring Rotation	Inner	Outer	Inner	Outer	Inner	Outer
Petroleum Oil	1.0	0.8	2.0	1.2	4.0	2.4
Synthetic Oil	1.0	0.8	2.0	1.2	4.0	2.4
Silicone Oil	0.8	0.7	0.8	0.7	0.8	0.7
Non-Channeling Grease	1.0	0.6	1.6	1.0	1.6	1.0
Channeling Grease	1.0	0.8	2.0	1.2	2.4	1.6
Silicone Grease	0.8	0.7	0.8	0.7	0.8	0.7

It must be noted that the N_{max} speed limiting values shown in this catalog are theoretical. It is possible to exceed these values, but the desired life of the bearing in a particular application must be considered. In applications where rotation speeds will exceed those shown in the product listing pages, please consult our Applications Engineering Department for additional information.

Lubricant Selection

Through years of experience, NMB has simplified the potentially confusing task of selecting the proper lubricant for your ball bearing applications. Although there are hundreds of lubricants available, the following step by step process will assist in clearly defining the needs of the application, and will help to determine what particular type of lubricant is required. An NMB Sales Engineer or Applications Engineer can assist you in this process and can offer the best lubricant choice for any ball bearing application.

NMB successfully uses the selection method below.

Step 1.

Define the temperature range of the application, including the environmental temperature plus any heat rise from motors, etc. Refer to Figure 1 (on the next page) and select the proper lubricant base for the maximum and minimum operating temperature.

When selecting a base fluid type, the fluid with the greatest film support is the preferred choice. Refer to the description of lubricant types for individual capabilities.

Step 2.

Determine the speed of the bearing and calculate the dN value (see Speed Factor on next page). Select the lubricant type that will operate within the dN speed factor, refer to Figure 1.

Step 3.

Knowing the dN value, determine the proper viscosity of the lubricating oil, or the base oil of the grease, see Figure 2. Since grease consists of approximately 80% oil, it is necessary to determine the viscosity of the oil for any high speed application. Improper selection can result in rapid deterioration of the base oil and failure of the unit.

Step 4.

Once you have determined these factors, the lubricant selection has been narrowed to the type of base oil, the operating temperature, and the oil viscosity range for a particular dN value. Next, determine whether a grease or oil is needed for the application. Then, individual lubricants should be examined to determine their suitability for the application.

In cases of extreme loads, adverse environments, or special requirements, please consult an NMB Applications Engineer for assistance. Some examples of adverse environments would be high humidity, excessive contamination, and water or other fluid exposure. An example of a special requirement would be the need for an FDA approved grease or oil for direct human contact.

TYPE	dN	Temperature range F° (C°)
Silicone	200,000	–100 to +400 (–73 to +204)
Diester	400,000	–65 to +350 (–54 to +177)
Petroleum	600,000	–25 to +250 (–32 to +121)

Figure 1. Relationship between lubricants, dN values, and temperature ranges.

Speed Factor

The maximum usable operating speed of a grease lubricant is dependent on the type of oil. The speed factor is a function of the bore (d) and O.D. (D) of the bearing in millimeters (mm) and the speed of the bearing (N) in revolutions per minute (RPM) where:

$$dN = \left(\frac{d + OD}{2} \text{ mm} \right) \times N \text{ (RPM)}$$

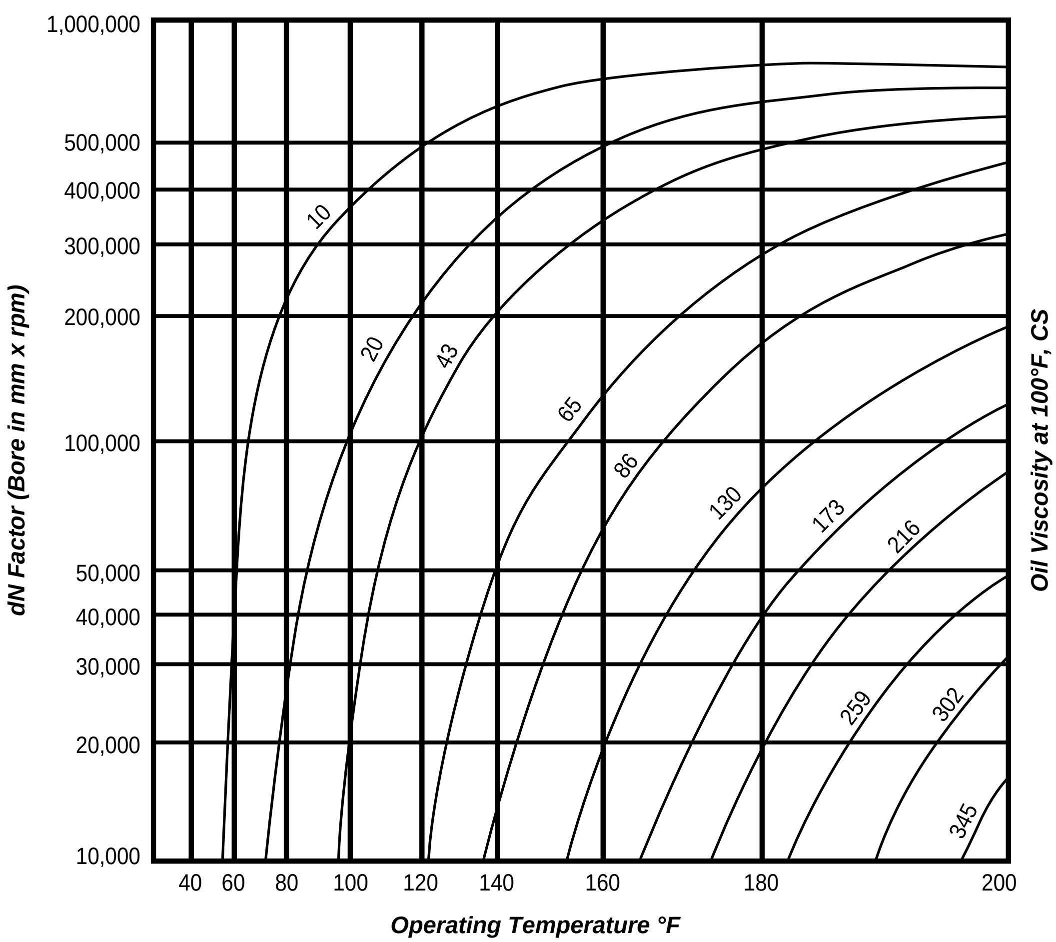


Figure 2. Courtesy of Exxon Corp.

There are many lubricants available for ball bearings. The chart below lists a variety of types, one of which should work well for most operating conditions.

Table of Commonly Used Lubricant Types

Code	Brand Name	Basic Type Oil	*Operating Temp. °F	Uses
LO1	Fuchs Windsor L245X (MIL-L-6085A)	Ester oil	-60° to +250°	Low speed instrument oil Rust preventative. Low torque.
Code	**Brand Name Grease**	**Basic Type**	***Operating Temp. °F**	**Uses**
LG20	Exxon Beacon 325	Ester oil + lithium soap thickener	-60° to +250°	General purpose grease for bearings and small gears. High and low temperatures. Low torque.
LY48	Mobil 28 (MIL-G-81322)	Synthetic oil + clay thickener	-65° to +350°	Developed for aircraft bearings and mechanisms. OK for low-speed oscillation. Low torque. Considered noisy in bearings.
LY121	Kyodo Multemp SRL	Ester oil + lithium soap thickener	-40° to +300°	Very quiet, widely-used motor grease. HDD spindle motor applications. OK for low speed oscillation.
LY255	Nippon Oil NIG Ace WS	Synthetic hydrocarbon and refined mineral oil + diurea soap thickener	-50° to +300°	Encoders, HDD actuators applications. OK for high speed oscillation.
LY532	Kluber Asonic HQ72-102	Ester oil + urea soap thickener	-40° to +350°	Suitable for automotive radiator cooling fan applications and other high temperature motor bearings.
LY551	Proprietary	Poly-alpha-olefin oil + urea soap thickener	-40° to +300°	Vacuum cleaner and power tool applications. Low noise and high speed.

*Based on manufacturer's published operating temperatures.

DYNAMIC LOAD RATINGS AND FATIGUE LIFE

Dynamic Radial Load Rating

The dynamic radial load rating (C_r) for a radial ball bearing is a calculated, constant radial load which a group of identical bearings can theoretically endure for a rating life of one million revolutions. The dynamic radial load rating is a reference value only. The base value of one million revolutions Rating Life has been chosen for ease of calculation. Since applied loading equal to the basic load rating tends to cause permanent deformation of the rolling surfaces, such excessive loading is not normally applied. Typically, a radial load that corresponds to 15 percent, or more, of the dynamic radial load rating is considered heavy loading for a ball bearing. In cases where loading of this degree is required, please consult an NMB Application Engineer for information regarding bearing life and lubricant recommendations.

Rating Life

The "rating life" (L_{10}) of a group of apparently identical ball bearings is the life in millions of revolutions, or number of hours, that 90 percent of the group will complete or exceed. For a single bearing, L_{10} also refers to the life associated with 90 percent reliability. The life which 50 percent of the group of ball bearings will complete or exceed ("median life" L_{50}) is usually not greater than five times the rating life.

Calculation of Rating Life:
The magnitude of the rating life, L_{10}, in millions of revolutions for a ball bearing application is

$$L_{10} = \left(\frac{C_r}{P_r}\right)^3$$

Where

L_{10} = Rating life as described above
C_r = Dynamic radial load rating (Kgf)
P_r = Dynamic equivalent radial load (Kgf)

The dynamic radial load rating (C_r) can be found on the product listing pages. The dynamic equivalent load must be calculated according to the following procedure:

$$P_r = XF_r + YF_a$$

Where

P_r = Dynamic equivalent radial load (Kgf)
X, Y = Obtained from the following X and Y table
F_r = Radial load on the bearing during operation (Kgf)
F_a = Axial load on the bearing during operation (Kgf)

The L_{10} life can be converted from millions of revolutions to hours using the rotation speed. This can be done as follows:

$$L_{10}\text{ (millions of revolutions)} \times \frac{1{,}000{,}000}{\text{RPM} * 60} = L_{10}\text{ (hours)}$$

Relative Axial Load		$F_a/F_r \le e$		$F_a/F_r \ge e$	
$\frac{F_a}{Z * D_w^2}$	e	X	Y	X	Y
0.0175	0.19	1	0	0.56	2.30
0.0352	0.22				1.99
0.0703	0.26				1.71
0.105	0.28				1.55
0.143	0.30				1.45
0.211	0.34				1.31
0.352	0.38				1.15
0.527	0.42				1.04
0.703	0.44				1.00

Z = Number of balls D_W = Ball size (mm)

Step 1: Calculate F_a/ZD_W^2 and cross reference value "e".
Step 2: Determine F_a/F_r relationship to find X and Y values.
NOTE: Pounds to Kilograms Force Conversion:
Multiply pounds by .45359 to get Kgf (Lbs*.45359 = Kgf)

Life Modifiers

For most cases, the L_{10} life obtained from the equation discussed previously will be satisfactory as a bearing performance criterion. However, for particular applications, it might be desirable to consider life calculations for different reliabilities and/or special bearing properties and operating conditions. Reliability adjustment factors, bearing material adjustment, and special operating conditions are discussed below. For assistance with questions regarding bearing life, please consult an NMB Applications Engineer.

Bearing Material

NMB recommends that radial load ratings published for chrome steel be reduced by 20% for stainless steel. This is a conservative approach to insure that bearing capacity is not exceeded under the most adverse conditions. This is incorporated in the a_2 modifier as shown in the table below.

Reliability Modifier

Where a more conservative approach than conventional rating life (L_{10}) is desired, the ABMA offers a means for such estimates. The table below provides selected modifiers (a_2) for calculating failure rates down to 1% (L_1).

Table of Reliability/Material Life Modifier a_2

Required Reliability –%	L_n	Value of a_2 Chrome	DD
90	L_{10}	1.00	0.50
95	L_5	0.62	0.31
96	L_4	0.53	0.27
97	L_3	0.44	0.22
98	L_2	0.33	0.17
99	L_1	0.21	0.11

Other Life Adjustments

The conventional rating life often has to be modified as a consequence of application abnormalities, whether they be intentional or unknown. Seldom are loads ideally applied. The following conditions all have the practical effect of modifying the ideal, theoretical rating life (L_{10}).

a. Vibration and/or shock-impact loads
b. Angular misalignment
c. High-speed effects
d. Operation at elevated temperatures
e. Fits
f. Internal design

NMB can assist in gauging the impact of these conditions when they are of a major concern to the application. Please consult an NMB Sales Engineer or a member of the Applications Engineering staff.

Oscillatory Service Life

Frequently, ball bearings do not operate with one ring rotating unidirectionally. Instead, they execute a partial revolution, reverse motion, and then repeat this cycle, most often in a uniform manner. Efforts to forecast a reliable fatigue life by simply relating oscillation rate to an "equivalent" rotational speed are invalid. The actual fatigue life of bearings operating in the oscillatory mode is governed by four factors; these factors are: applied load, angle of oscillation, rate of oscillation, and lubricant. NMB can provide guidance on practical life of ball bearings in oscillatory applications.

Lubricant Life

In many instances a bearing's effective life is governed by the lubricant's life. This is usually the case where applications involve very light loads and/or very slow speeds.

With light loads and/or slow speeds the conventional fatigue life forecast will be unrealistically high. The lubricant's ability to provide sufficient film strength is sustained only for a limited time. This is governed by:

a. Quality and quantity of the lubricant in the bearing
b. Environmental conditions
   (i.e., ambient temperature, area cleanliness)
c. The load-speed cycle

Specialized oils and greases are available that exhibit favorable properties over an extended period. Please consult an NMB Sales Engineer or a member of the Applications Engineering staff for guidance on practical lubricant life.

Static Radial Load Rating

The static radial load rating (C_{or}) given on the product listing pages is the radial load which a non-rotating ball bearing will support without damage, and will continue to provide satisfactory performance and life.

The static radial load rating is dependent on the maximum contact stress between the balls and either of the two raceways. The load ratings shown were calculated in accordance with the ABMA standard. The ABMA has established the maximum acceptable stress level resulting from a pure radial load, in a static condition, to be 4.2 GPa (609,000 psi).

Static Axial Load Capacity

The static axial load capacity is axial load which a non-rotating ball bearing will support without damage. The axial static load capacity varies with bearing size, bearing material, and radial play. Due to the multiple combinations possible for each bearing, the axial static load capacities are not listed in this catalog. For information regarding axial load capacities, please consult an NMB Sales Engineer or Applications Engineer.

High Static Loads

Radial static load ratings and thrust static load ratings in excess of the C_{or} value shown in this text have practical applications where smoothness of operation and/or low noise are **not** of concern. Properly manufactured ball bearings, when used under controlled shaft and housing fitting practice, can sustain significantly greater permanent deformation (i.e., brinells) than the deformations associated with the static load ratings listed in this catalog.

NMB can provide specific recommendations for extraordinary high static load applications. Please consult a member of our Sales Engineering or Applications Engineering staff.

Ball bearing systems are preloaded:

1. To eliminate radial and axial looseness.
2. To reduce operating noise by stabilizing the rotating mass.
3. To control the axial and radial location of the rotating mass and to control movement of this mass due to external force influences.
4. To reduce the repetitive and non-repetitive runout of the rotational axis.
5. To reduce the possibility of damage due to vibratory loading.
6. To increase stiffness.

Spindle motors and tape guides are examples of applications where preloaded bearings are used to accurately control shaft position when external loads are applied. As the name implies, a preloaded assembly is one in which a bearing load (normally a thrust load) is applied to the system so the bearings are already carrying a load before any external load is applied.

There are essentially two ways to preload a ball bearing system - by using a spring and by a solid stack of parts.

Spring Preloading

For many applications, one of the simplest and most effective methods of applying a preload is by means of a spring. This can consist of a coil spring or perhaps a wavy washer which applies a force against the inner or outer ring of one of the bearings in an assembly.

When a spring is used, it is normally located on the non-rotating component; i.e., with shaft rotation, the spring should be located in the housing against the outer rings. Springs can be very effective where differential thermal expansion is a problem. In the spindle assembly (Figure 1), when the shaft becomes very hot and expands in length, the spring will move the outer ring of the left bearing and thus maintain system preload. Care must be taken to allow for enough spring movement to accommodate the potential shaft expansion.

Since, in a spring, the load is fairly consistent over a wide range of compressed length, the use of a spring for preloading negates the necessity for holding tight location tolerances on machined parts. For example, retaining rings can be used in the spindle assembly, thus saving the cost of locating shoulders, shims, or threaded members.

Normally, a spring preload would **not** be used where the assembly is required to withstand reversing thrust loads.

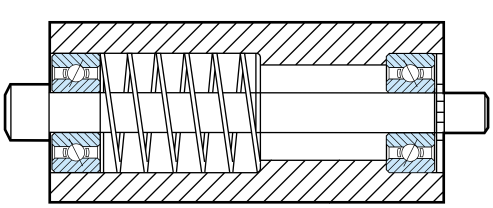


Figure 1. Spindle Assembly using compression coil spring — shaft rotation

Solid Stack Preloading

Where precise location control is required, as in a precision motor (Figure 2) or a flanged tape guide (Figure 3), a solid preload system is indicated.

A solid stack, "hard" or "rigid" preload, can be achieved in a variety of ways. Theoretically, it is possible to preload an assembly by tightening a screw as shown (Figure 3) or inserting shims (Figure 4) to obtain the desired rigidity. It should be noted that care must be taken when using a solid stack preloading system with miniature and instrument bearings. Overload of the bearings must be avoided so that the bearings are not damaged during this process.

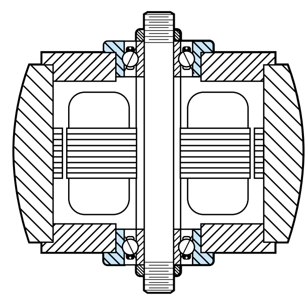


Figure 2. Typical Motor design using rotor as outer ring spacer and stator mount as inner ring spacer – outer ring rotation – Solid preload

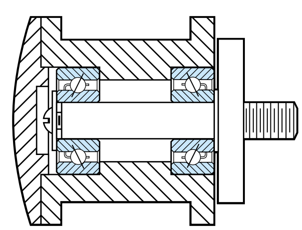


Figure 3. Typical Tape Guide design using screw and washer to solidly preload by clamping inner rings – Outer Ring Rotation

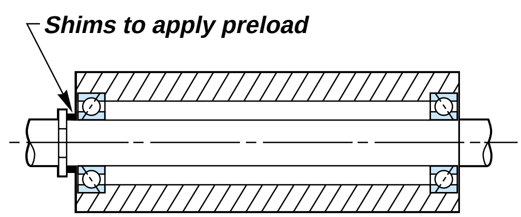


Figure 4.

Preload Levels

Preloading is an effective means of positioning and controlling stiffness because of the nature of the ball/raceway contact. Under light loads, the ball/raceway contact area is very small and so the amount of "yield" or "definition" is substantial with respect to the amount of load. As the load is increased, the ball/raceway contact area increases in size (the contact is in the shape of an ellipse) and so provides increased stiffness or reduced "yield" per unit of applied load.

This is illustrated in the single-bearing deflection curve shown in Figure 5. When two bearings are preloaded together and subjected to an external thrust load, the axial yield rate for the pair is drastically reduced because of the preload and the interaction of the forces exerted by the external load and the reactions of the two bearings. As can be seen by the lower curve in Figure 5, the yield rate for the preloaded pair is essentially linear.

Miniature and instrument bearings are typically built to accept light preloads ranging from 0.25 lbs. to normally not more than 10 lbs.

Application Engineers at NMB can provide assistance in selecting appropriate preload specifications and in providing specific information on bearing yield rates where that is required.

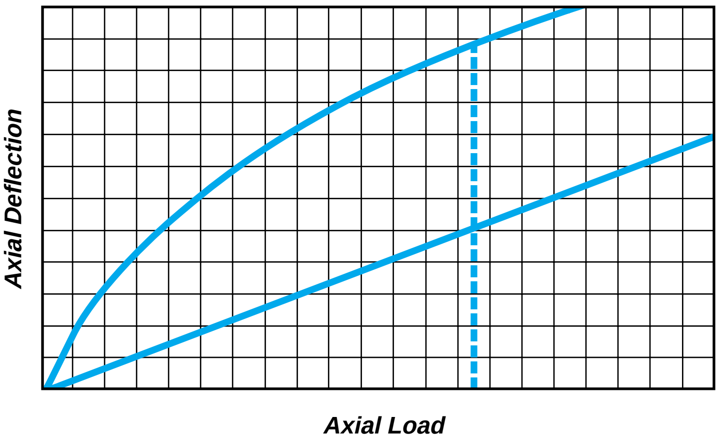


Figure 5.

The operating characteristics of a system can be drastically affected by the way in which the ball bearings are handled and mounted. A bearing which has been damaged due to excessive force or shock loading during assembly, or which is fitted too tight or too loose, may cause the device to perform in a substandard manner.

By following a few general guidelines during the design of mating parts and by observing some basic cautions in the assembly process, the possibility of producing malfunctioning devices will be considerably reduced.

The chart on the following page lists recommended fits for most normal situations. There are four cautions which must be observed:

1. When establishing shaft or housing sizes, the effect of differential thermal expansion must be accounted for. The Table of Recommended Fits assumes stable operating conditions, so if thermal gradients are known to be present or dissimilar materials are being used, the room temperature fits must be adjusted so that the proper fit is attained at operating temperature. Approximate thermal coefficients for common material are available from NMB Applications Engineering staff.

2. When miniature and instrument ball bearings are interference fitted (either intentionally or as a result of thermal gradients) the bearing radial play can be estimated to be reduced by an amount equal to 80% of the actual diametrical interference fit. This 80% figure is conservative, but is of good use for design purposes. Depending on the materials involved, this factor will typically range from 50% to 80%. The following is an example of calculating loss of radial play:

Radial Play of Bearing:	.0002"
Total Interference Fit:	.0003" Tight
80% of Interference Fit (.0003" x 80%)	.00024"
Theoretical Resultant Radial Play of Bearing	.00004" Tight

Theoretically, this bearing could be operating with negative radial play. A bearing operated in an excessive negative radial play condition will perform with reduced life. However, the above calculation is for design only, and does not take into account housing material, shaft material, or surface finish of the housing or shaft surfaces. As an example, if the finish of the shaft surface is rough, a part of the interference between the inner ring and shaft will be absorbed by the deformation of the shaft surface. This will serve to reduce the overall interference fit, and thus, the radial play of the bearing will not be reduced as much as is shown in the calculation above. If assistance on fits and their effect on bearing performance is required, please consult a member of NMB Applications Engineering staff.

The table of recommended fits is based on the use of bearings of ABEC 5 or better tolerance level.

3. If the outer or inner ring face is to be clamped or abutted against a shoulder, care must be taken to make sure that this shoulder configuration provides a good mounting surface:
   - The shoulder face must be perpendicular to the bearing mounting seat. The maximum permissible angle of mis-alignment is recommended to be 1/4°.
   - The corner between the mounting diameter and the face must have an undercut or a fillet radius no larger than that shown on the listing page under the column "Fillet Radius r".
   - The shoulder diameter must meet the requirements shown on the table of recommended shoulder diameters.

4. Assembly technique is extremely critical. After the design is finalized and assembly procedures are being formulated, the bearing Static Capacity - C_{or} - becomes extremely important. It is easy, for instance, to exceed the 3 pound capacity of a DDRI-2 during assembly. After assembly to the shaft, damage can be done either by direct pressure or by moment load while the bearing-and-shaft subassembly is being forced into a tight housing. A few simple calculations will underscore this point.

   Adequate fixturing should always be provided for handling and assembling precision bearings. This fixturing must be designed so that, when assembling the bearing to the shaft, force is applied only to the inner ring, and, when assembling into the housing, force is applied only to the outer ring. Further, the fixturing must preclude the application of any moment or shock loads which would be transmitted through the bearing. Careful attention to this assembly phase of the total design effort can prevent many problems and provide savings when production starts. You will find our engineers eager to help in this, one of the most important phases of taking a product from design conception to the marketplace.

Recommended Shoulder Diameter

*Basic Size	Minimum Shaft Shoulder Diameter Inches	Maximum Housing Shoulder Diameter Inches
DDRI-2	.060	.105
DDRI-21/2	.071	.132
DDRI-3	.079	.164
DDRI-4	.102	.226
DDRI-3332	.114	.168
DDRI-5	.122	.284
DDRI-418	.148	.226
DDRI-518	.153	.284
DDRI-618	.153	.347
DDR-2	.179	.325
DDRI-5532	.180	.288
DDR-1640	.210	.580
DDRI-5632	.210	.288
DDRI-6632	.216	.347
DDR-3	.244	.446
DDR-1650	.250	.580
DDR-1950	.250	.700
DDR-1960	.290	.700
DDRI-614	.272	.352
DDRI-814	.284	.466
DDR-4	.310	.565
DDRI-1214	.322	.678
DDR-2270	.325	.810
DDR-2280	.370	.810
DDRI-8516	.347	.466
DDRI-1038	.435	.565
DDRI-1438	.451	.799
DDRI-1212	.560	.690
DDRI-1458	.665	.835
DDRI-1634	.790	.960

* "DD" is a trademark of NMB

Table of Recommended Fits

Typical Applications	Shaft Fit	Shaft Diameter	Housing Fit Diameter	Housing
General Application-Inner ring rotation (Inner ring press fit, outer ring loose fit)	.0000 - .0004T	d+.0000 d+.0002	.0000 - .0004L	D+.0002 D+.0000
General Application-Outer ring rotation (Inner ring loose fit, outer ring press fit)	.0000 - .0004L	d–.0002 d–.0004	.0000 - .0004T	D–.0002 D–.0004
Tape guide roller	.0000 - .0004L	d–.0002 d–.0004	.0001L-.0003T	D–.0001 D–.0003
Drive motor (spring preload)	.0001T - .0003L	d–.0001 d–.0003	.0000 - .0004L	D+.0002 D–.0000
Precision synchro or servo	.0001T - .0003L	d–.0001 d–.0003	.0001T - .0003L	D+.0001 D–.0001
Potentiometer	.0001T - .0003L	d–.0001 d–.0003	.0000 - .0004L	D+.0002 D–.0000
Encoder spindle	.0001T - .0003L	d–.0001 d–.0003	.0001L - .0003T	D–.0001 D–.0003
Gear reducer	.0000 - .0004L	d–.0002 d–.0004	.0000 - .0004L	D+.0002 D–.0000
Light duty mechanism	.0000 - .0004L	d–.0002 d–.0004	.0000 - .0004L	D+.0002 D–.0000
Clutches, brakes - inner race floats	.0000 - .0004L	d–.0002 d–.0004	.0001T - .0003L	D+.0001 D–.0001
Pulleys, rollers, cam followers (outer race rotates)	.0000 - .0004L	d–.0002 d–.0004	.0000 - .0004T	D–.0002 D–.0004

L = Loose Fit   d = Nominal Bearing Bore
T = Tight Fit   D =Nominal Bearing O.D.

Recommended Fits are for ABEC 5 Bearings.
Table dimensions are given in inches.

Example: To use *DDR-2 bearing in potentiometer, the shaft O.D. should be .1250 - .0001 to .1250 -.0003 or .1249 to .1247. The housing should be .3750 +.0002 to .3750 +.0000 or .3752 to .3750
* “DD” is a trademark of NMB

PACKAGING/ POST SERVICE ANALYSIS

Our bearings are normally packaged in plastic vials, a quantity of 10 or more per vial. For chrome bearings, if prelubrication or protective coating is not specified by the applicable drawing or order, a preservative oil will be used to prevent corrosion.

Other special types of packaging to suit specific needs will be considered. Check with our Engineering Department when questions or special requirements arise.

Our Engineering staff stands ready to perform post service analysis on any bearings that have been in actual use. If bearings have failed in service, it is frequently possible to determine the cause of failure by examining parts and debris, even though the failure was catastrophic. All of the bearing components and as much as possible of the assembly in which they ran, should be made available for examination by our engineers. For example, if a small motor fails on life test, send the complete motor, assembled just as it came off the test bench, to us. A complete detailed examination will be made and a written report submitted. The report will contain details of the condition of bearings and mating parts, including actual measurements where applicable, and specific recommendations for overall improvement of the bearing performance in this particular application. Even if no failure occurs and particularly when units have been in actual field service for a long period of time, a wealth of valuable information and data can frequently be accumulated from post service analysis. This information can be very useful in product improvement and cost reduction programs.

The keys to gaining useful information from post service analysis are:

- Availability of the undisturbed assembled device, or as many components as possible, and
- Availability of as much historical information as possible describing the conditions under which the device operated. Speeds, loads, temperature, atmospheric conditions, any unusual shock, vibration or handling situations, etc., should be noted for consideration when the parts are examined.

When a failure occurs, or better yet, when a significantly successful test or field unit is obtained, contact us prior to tear-down to make arrangements for a post service analysis that may help you in your product improvement efforts.

QUALITY ASSURANCE/ DIMENSIONAL CONTROL

NMB Quality Control Systems meet ISO 9002 Standards.

In addition to the normal incoming material, first article, lot and in-process component inspections, the QC Department maintains process surveillance on all production operations particularly heat treat, deburring, grinding, and race finishing. This is to ensure that these operations, which generate the characteristics of the finished product, remain in control at all times.

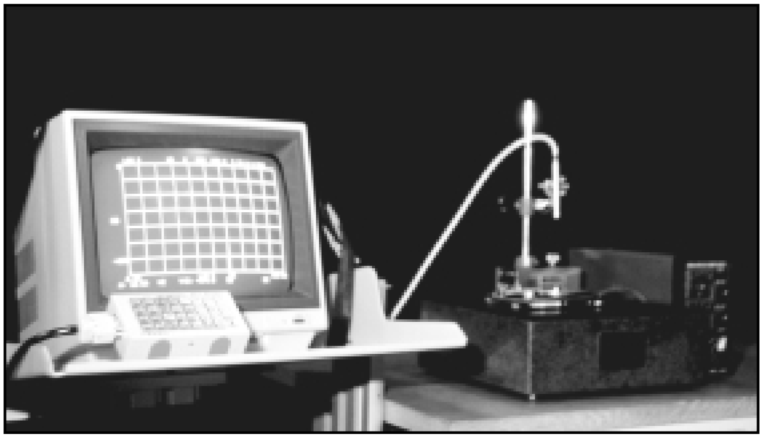
Much effort is also expended in correlating gages and gaging methods with user QC Departments. This is always mutually beneficial, and any user, or potential user, of NMB bearings is urged to contact the Quality Assurance Department to discuss correlation. This is particularly important where low runouts, low torque, or low noise is required.

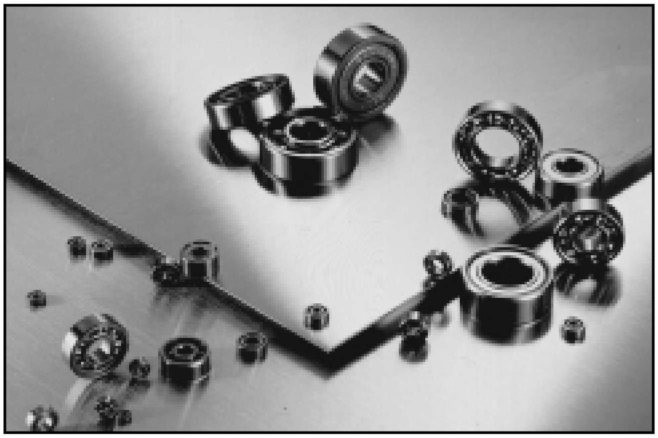
The company has equipped the Quality Assurance Department with the latest and finest test and measurement equipment available. Roundness, concentricity and squareness are measured.

Every bearing is guaranteed to be free of defects in workmanship and materials for twelve (12) months from invoice date. Any bearing found defective within the warranty period may be repaired, replaced or the purchase price repaid, provided that it is returned to the company and, upon inspection, is found to have been property mounted, lubricated, protected and not subjected to any mishandling.

NMB follows the specifications of the American Bearing Manufacturers Association (ABMA) and its associated ball bearing technical committee, the Annular Bearing Engineer's Committee (ABEC).

The ABEC tolerances on the next page are current at this catalog's printing. These tolerances are reviewed regularly and updated as required. The ABMA Standards may be obtained by contacting: ABMA, 1200 19th Street, NW, Suite 300, Washington, DC 20036. All dimensions are in inches.





Tolerances: Miniature and Instrument Ball Bearings

Inner Ring

Characteristic	ABEC 1	ABEC 3	ABEC 5	ABEC 7
Mean Bore Tolerance Limits	+.0000 -.0003	+.0000 -.0002	+.0000 -.0002	+.0000 -.0002
Radial Runout Width Variation	.0003 (1) —	.0002 (1) —	.00015 .00020	.0001 .0001
Bore Runout with Face	—	—	.00030	.0001
Race Runout with Face	—	—	.00030	.0001

(1) Add .0001 to the tolerance if bore size is over 10mm (.3937 inch).

Outer Ring

Characteristic	Configuration	Size Range	ABEC 1	ABEC 3	ABEC 5	ABEC 7
Mean O.D. Tolerance Limits	All	0-18mm (0-.709)	+.0000 -.0003	+.0000 -.0003	+.0000 -.0002	+.0000 -.0002
	All	over 18-30mm (.709-1.1811)	+.00000 -.00035	+.0000 -.0003	+.0000 -.0002	+.0000 -.0002
Radial Runout	All	0-18mm	.0006	.0004	.0002	.00015
	All	over 18-30mm	.0006	.0004	.0002	.00015
Width Variation	All	0-30mm	—	—	.0002	.00010
O.D. Runout with Face	All	0-30mm	—	—	.0003	.00015
Race Runout with Face	Plain	0-18mm	—	—	.0003	.00020
	Plain	over 18-30mm	—	—	.0003	.00020
	Flanged	0-30mm	—	—	.0003	.00030
Flange Width Tolerance Limits	— —	— —	— —	+.0000 -.0020	+.0000 -.0020	+.0000 -.0020
Flange Diameter Tolerance Limits	— —	— —	— —	+.0050 -.0020	+.0000 -.0010	+.0000 -.0010

Ring Width

Characteristic	ABEC 1	ABEC 3	ABEC 5	ABEC 7
Width Tolerance	+.000 -.005	+.000 -.005	+.000 -.001	+.000 -.001

TEMPERATURE CONVERSION TABLE

The numbers in the center column refer to the temperatures either in Celsius or Fahrenheit which need conversion to the other scale. When converting from Fahrenheit to Celsius, the equivalent temperature will be found to the left of the center column. If converting from Celsius to Fahrenheit the answer will be found to the right.

°C	°C/°F	°F	°C	°C/°F	°F	°C	°C/°F	°F	°C	°C/°F	°F
-79	-110	-166	37.7	100	212	204	400	752	371	700	1292
-73	-100	-148	43	110	230	210	410	770	376	710	1310
-68	-90	-130	49	120	248	215	420	788	382	720	1328
-62	-80	-112	54	130	266	221	430	806	387	730	1346
-57	-70	-94	60	140	284	226	440	824	393	740	1364
-51	-60	-76	65	150	302	232	450	842	565	1050	1922
-46	-50	-58	71	160	320	238	460	860	571	1060	1940
-40	-40	-40	76	170	338	243	470	878	576	1070	1958
-34	-30	-22	83	180	356	249	480	896	582	1080	1976
-29	-20	-4	88	190	374	254	490	914	587	1090	1994
-23	-10	14	93	200	392	260	500	932	593	1100	2012
-17.7	0	32	99	210	410	265	510	950	598	1110	2030
-17.2	1	33.8	104	220	428	271	520	968	604	1120	2048
-16.6	2	35.6	110	230	446	276	530	986	609	1130	2066
-16.1	3	37.4	115	240	464	282	540	1004	615	1140	2084
-15.5	4	39.2	121	250	482	288	550	1022	620	1150	2102
-15.0	5	41.0	127	260	500	293	560	1040	626	1160	2120
-14.4	6	42.8	132	270	518	299	570	1058	631	1170	2138
-13.9	7	44.6	138	280	536	304	580	1076	637	1180	2156
-13.3	8	46.4	143	290	554	310	590	1094	642	1190	2174
-12.7	9	48.2	149	300	572	315	600	1112	648	1200	2192
-12.2	10	50.0	154	310	590	321	610	1130	653	1210	2210
-6.6	20	68.0	160	320	608	326	620	1148	659	1220	2228
-1.1	30	86.0	165	330	626	332	630	1166	664	1230	2246
4.4	40	104.0	171	340	644	338	640	1184	670	1240	2264
9.9	50	122.0	177	350	662	343	650	1202	675	1250	2282
15.6	60	140.0	182	360	680	349	660	1220	681	1260	2300
21.0	70	158.0	188	370	698	354	670	1238	686	1270	2318
26.8	80	176.0	193	380	716	360	680	1256	692	1280	2336
32.1	90	194.0	199	390	734	365	690	1274	697	1290	2354

METRIC CONVERSION TABLE

Fraction	Inch	mm	Fraction	Inch	mm	Fraction	Inch	mm
1/64	0.0156	0.3969		0.2883	7.3228	11/16	0.6875	17.4625
	0.0250	0.6350	19/64	0.2969	7.5406	45/64	0.7031	17.8594
1/32	0.0312	0.7937	5/16	0.3125	7.9375		0.7087	**18.0000**
	0.0394	**1.0000**		0.3150	**8.0000**	23/32	0.7187	18.2562
	0.0400	1.0160	21/64	0.3281	8.3344	47/64	0.7344	18.6532
3/64	0.0469	1.1906	11/32	0.3437	8.7312		0.7435	18.8849
	0.0472	1.2000		0.3543	**9.0000**		0.7480	**19.0000**
	0.0550	1.3970	23/64	0.3594	9.1281	**3/4**	0.7500	19.0500
	0.0591	**1.5000**	**3/8**	0.3750	9.5250	49/64	0.7656	19.4469
1/16	0.0625	1.5875	25/64	0.3906	9.9213		0.7717	19.6012
	0.0709	1.8000		0.3937	**10.0000**	25/32	0.7812	19.8433
5/64	0.0781	1.9844	13/32	0.4062	10.3187		0.7874	**20.0000**
	0.0787	**2.0000**		0.4100	10.4140	51/64	0.7969	20.2402
	0.0906	2.3012	27/64	0.4219	10.7156	13/16	0.8125	20.6375
3/32	0.0937	2.3812		0.4250	10.7950		0.8268	**21.0000**
	0.0984	**2.5000**		0.4331	**11.0000**	53/64	0.8281	21.0344
	0.1000	2.5400	7/16	0.4375	11.1125	27/32	0.8437	21.4312
	0.1024	2.6000	29/64	0.4531	11.5094	55/64	0.8594	21.8281
7/64	0.1094	2.7781		0.4600	11.6840		0.8661	**22.0000**
	0.1100	2.7940	15/32	0.4687	11.9062	**7/8**	0.8750	22.2250
	0.1102	2.8000		0.4724	**12.0000**	57/64	0.8906	22.6219
	0.1181	**3.0000**	31/64	0.4844	12.3031		0.9055	**23.0000**
1/8	0.1250	3.1750	**1/2**	0.5000	12.7000	29/32	0.9062	23.0187
	0.1256	3.1902		0.5118	**13.0000**	59/64	0.9219	23.4156
	0.1378	**3.5000**	33/64	0.5156	13.0968	15/16	0.9375	23.8125
9/64	0.1406	3.5719	17/32	0.5312	13.4937		0.9449	**24.0000**
5/32	0.1562	3.9687	35/64	0.5469	13.8906	61/64	0.9531	24.2094
	0.1575	**4.0000**		0.5512	**14.0000**	31/32	0.9687	24.6062
11/64	0.1719	4.3656	9/16	0.5625	14.2875		0.9843	**25.0000**
3/16	0.1875	4.7625	37/64	0.5781	14.6844	63/64	0.9844	25.0031
	0.1892	4.8057		0.5906	**15.0000**		**1.0000**	25.4000
	0.1969	**5.0000**	19/32	0.5937	15.0812		1.0236	**26.0000**
13/64	0.2031	5.1594	39/64	0.6094	15.4781		1.0415	26.4541
	0.2165	5.4991	**5/8**	0.6250	15.8750		1.0480	26.6192
7/32	0.2187	5.5562		0.6299	**16.0000**	1-1/16	1.0625	26.9875
15/64	0.2344	5.9531	41/64	0.6406	16.2719		1.0630	**27.0000**
	0.2362	**6.0000**		0.6500	16.5100		1.1025	**28.0000**
1/4	0.2500	6.3500	21/32	0.6562	16.6687	**1-1/8**	1.1250	28.5750
17/64	0.2656	6.7469		0.6620	16.8148		1.1417	**29.0000**
	0.2756	**7.0000**		0.6693	**17.0000**		1.1812	**30.0000**
9/32	0.2812	7.1437	43/64	0.6719	17.0656	1-3/16	1.1875	30.1625
						1-1/4	1.2500	31.7500
						1-1/2	1.5000	38.1000

ERRORS — All information, data and dimension tables in this catalog have been carefully compiled and thoroughly checked. However, no responsibility for possible errors or omissions can be assumed.

CHANGES — The company reserves the right to change specifications and other information included in this catalog without notice.