Ball Bearing Design and Applications

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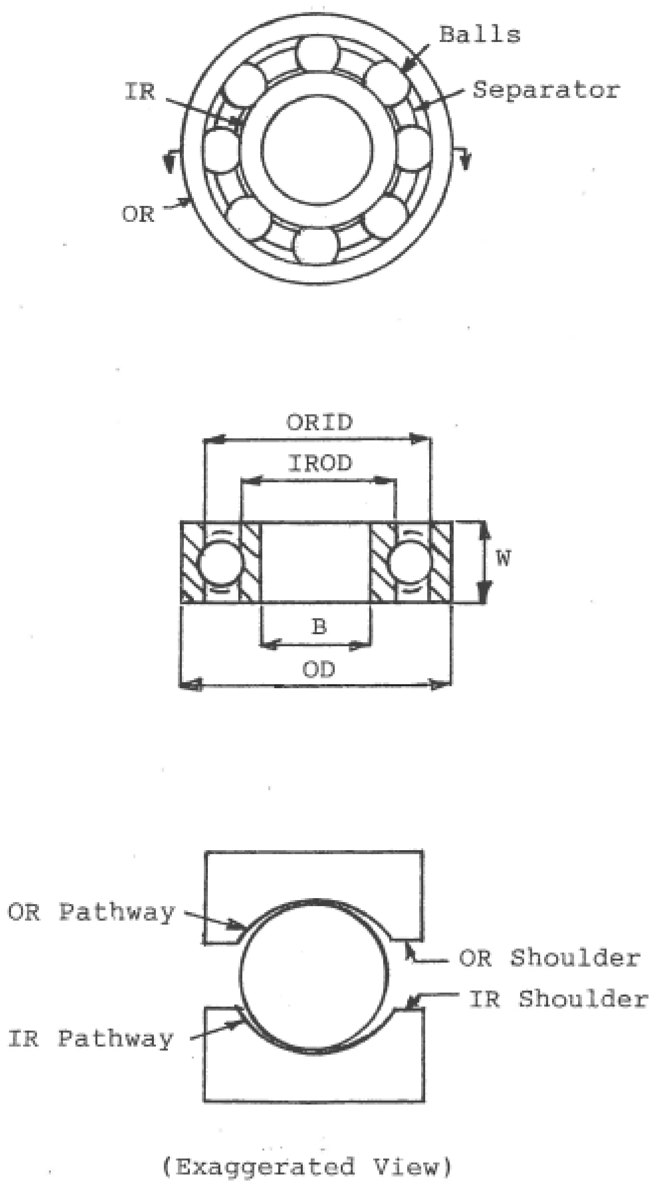
Background

Ball bearings are used primarily to support rotating shafts in mechanical equipment. They can be found in everything from personal computers to passenger cars. They are of simple design and can be precision made in mass production quantities. They can support heavy loads over a wide speed range and do it virtually friction free. They come in many different sizes and shapes, are relatively inexpensive, and require little or no maintenance. They have predictable design lives and operating characteristics and are truly a valuable asset to the rotating equipment industry of today.

Description

A ball bearing consists of an inner ring (IR), an outer ring (OR), a complement of balls, and a separator to contain the balls. (See Figure 1.) The outer diameter of the inner ring (IROD) and the inner diameter of the outer ring (ORID) have a groove in which the balls roll on. This groove is commonly called the *pathway*. The raised surfaces on each side of the pathway are called the *shoulders*. The balls are held equally spaced around the annulus of the bearing by the separator. The basic dimensions of the bearing are the bore (B), outside diameter (OD), and the width (W).

Figure 1
Ball Bearing Description



Design

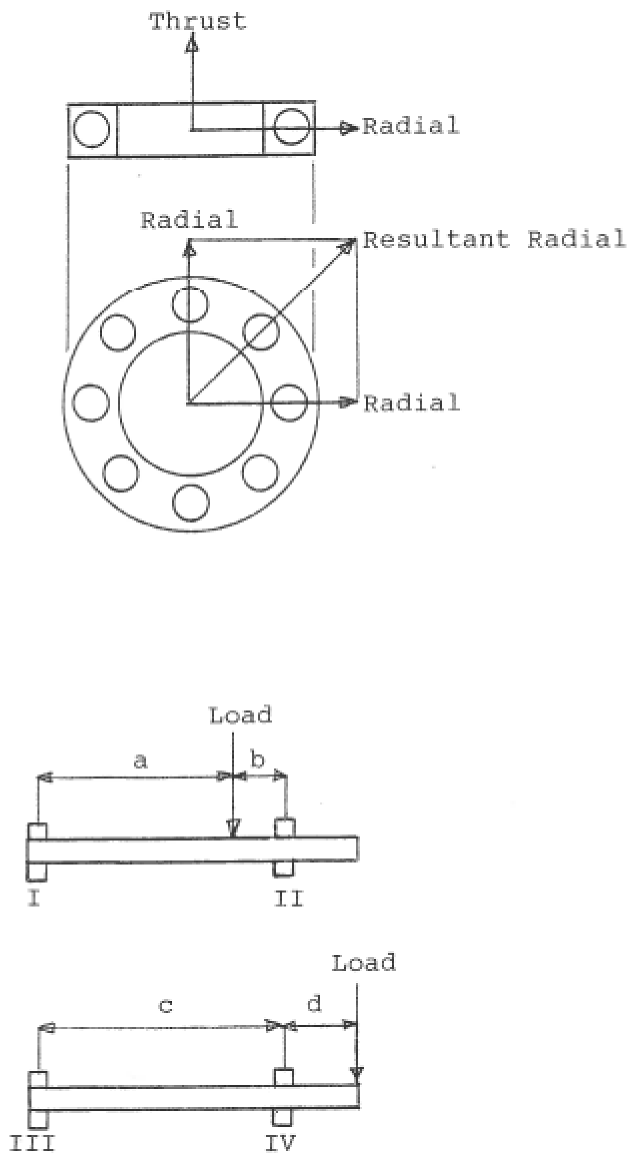
For most bearings, the radius of curvature across the pathway of an inner ring is held to 51-52% of the ball diameter while the radius of curvature across the pathway of the outer ring is held to 53-54% of the ball diameter. As the pathway radius of curvature approaches 50% of the ball diameter (100% of the radius), the stress between the ball and pathway decreases; however, it also moves the contact of the ball higher up the pathway wall producing more friction as the balls revolve around the bearing. For ball bearings, the best balance between stress and friction is attained with the pathway curvature slightly above 50% as described above for inner and outer rings. The number is higher for outer rings because, in the rotational plain, outer rings present a concave surface to the balls lowering the contact stress compared to the inner rings which, in the rotational plane, present a convex surface to the balls raising the stress.

As previously described, the raised surface on each side of the pathway is called the shoulder. The radial difference between the deepest part of the pathway to the OD of the inner ring and to the ID of the outer ring is called the shoulder height. In most bearings it is held to 22-30% of the ball diameter for inner rings and 18-22% for outer rings, with the difference being due to the same reason that the pathway curvatures are not the same for inner and outer rings. The design height of the shoulder is a balance between being high enough to support proportionately high thrust (side) loads and low enough to be able to assemble an adequately sized separator into the bearing. Another design consideration is the minimum wall thickness at the bottom of the pathway of both the inner and outer rings. This section must be thick enough to simultaneously support compressive stresses from the balls on the inside and hoop stresses from press fit assembly on the outside.

Theory of Operation

In most applications, there are two ball bearings supporting a rotating shaft. The ball bearing inner ring is a press fit on the shaft so there is no relative movement between the two while the shaft is rotating. The outer ring is a close push fit in the housing for assembly reasons and also to allow slight axial movement to accommodate manufacturing tolerances and differential thermal expansion between the shaft and housing. The separator and ball complement rotate together at approximately half the speed of the inner ring. The balls rotate around their own axis approximately twice the speed of the inner ring. Forces are imposed on the bearings by the member that is driving the shaft. The force can be separated into a radial load and a thrust load. The sketch at the top of Figure 2 shows the radial loads acting perpendicular to the bearing axis of rotation while thrust loads act parallel to the axis of rotation. In many instances there are two radial loads acting 90^o apart as shown on the second sketch of Figure 2. The Pythagorean Theorem is used to calculate the resultant load on the bearing. The radial load can sometimes be straddle mounted between two bearings as shown on the third sketch of Figure 2. Simple beam equations show that the bearing nearest the load supports the greater portion of the load. The fourth sketch of Figure 2 shows the load overhanging the two bearings. Beam calculations will show that the bearing closer to the overhung load supports a force that is actually greater than the load itself. Bearing loads, along with speed of rotation, are then inserted into an equation that predicts operating life in an application. Normally the radial load is reacted by just a few balls in the bearing while the thrust load is reacted by all the balls in the bearing. Assume that the radial load is acting downward on a shaft that is supported by two bearings. The balls at the top of the bearing are under little or no load. As they rotate to the bottom of the bearing they are compressed between the rings. As they rotate back to the top, the compressed metal expands to its original state. This constant compression and expansion of metal after many revolutions of the bearing leads to fatigue failure. The failure generally occurs as a spall (pit) in the inner ring. The inner ring is under more stress than the outer ring because, as previously explained, it presents a convex surface to the balls as opposed to the outer ring presenting a concave surface. The balls are not a high failure item because, besides being heat treated to a higher hardness and lapped to a finer finish than the rings, each ball rotates through the load zone once for every time that a point on the inner ring rotates through the load zone approximately twice.

Figure 2
Bearing Loads



Ball Bearing Manufacture

Ball bearings are manufactured to a very high precision level in high volume quantities. In some cases, lines are completely automated starting from the raw material phase to the finished product. Great care has to be taken to keep all parts clean and free of rust. The specification for ball bearing steel is very demanding. In normal service the steel must withstand compressive stresses of 200,000 to 300,000 pounds per square inch (psi) and, in extreme service, compressive stresses of 500,000 psi.

The standard grade steel for ball bearings is high carbon, high chromium, vacuumed degassed AIS/SAE 52100. The high carbon content of l% gives the steel excellent response to heat treatment resulting in very high strength and hardness. The high chromium content of 1.35% further increases responsiveness to heat treatment and adds depth of hardness penetration. Vacuum degassing removes impurities making the steel extra clean. For extremely critical applications, consumable electrode vacuum melted steel is available for an even higher degree of cleanliness and uniformity.

Rings and balls are heat treated to the RC60 level for optimum toughness and strength at operating temperatures up to 300°F. For operating temperatures over 300°F, the steel softens and loses dimensional stability. A special stabilization heat treat procedure is available for continuous operation at temperatures up to 400°F. Stabilization tempers the steel at a temperature above what is encountered in service resulting in a slight decrease in hardness from the RC60 level.

Stainless steel is used for rings and balls for corrosion resistance and high temperature operation up to 550°F. For even higher operating temperatures up to 1100°F, special tool steels and cobalt alloys are used.

Separator steel for most bearings is low carbon steel. Most angular contact bearings (explained later) operating at high speed use a non-metallic separator material. Non-metallic separator material combines low friction, light weight, and strength up to 275°F. With higher temperatures and speeds, iron silicon bronze and phosphor bronze provide low friction and a high strength-to-weight ratio. For temperatures up to 1000°F, S-Monel, special tool steel, and alloy steel are available. Figure 3 gives temperature limitations of the various bearing and separator materials.

Ball bearings inner and outer rings are processed as follows:

- All rings are machined from special sized steel tubing.
- They are thru-hardened in heat treat furnaces.
- Every surface is fine ground to exacting tolerances.
- Pathways are honed to even finer surface finishes.

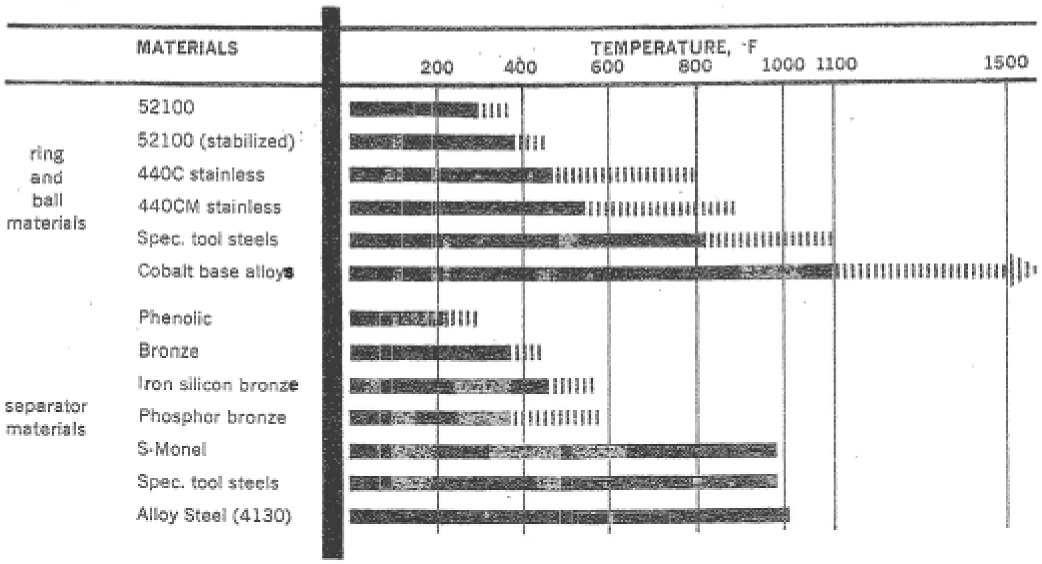
Balls are processed as follows:

- They are cold-headed to a spherical shape from drawn bar.
- They are soft ground.
- They are heat treated in high temperature furnaces.
- They are hard ground.
- They are lapped to a mirror-like finish.

Assembly of most ball bearings is as follows:

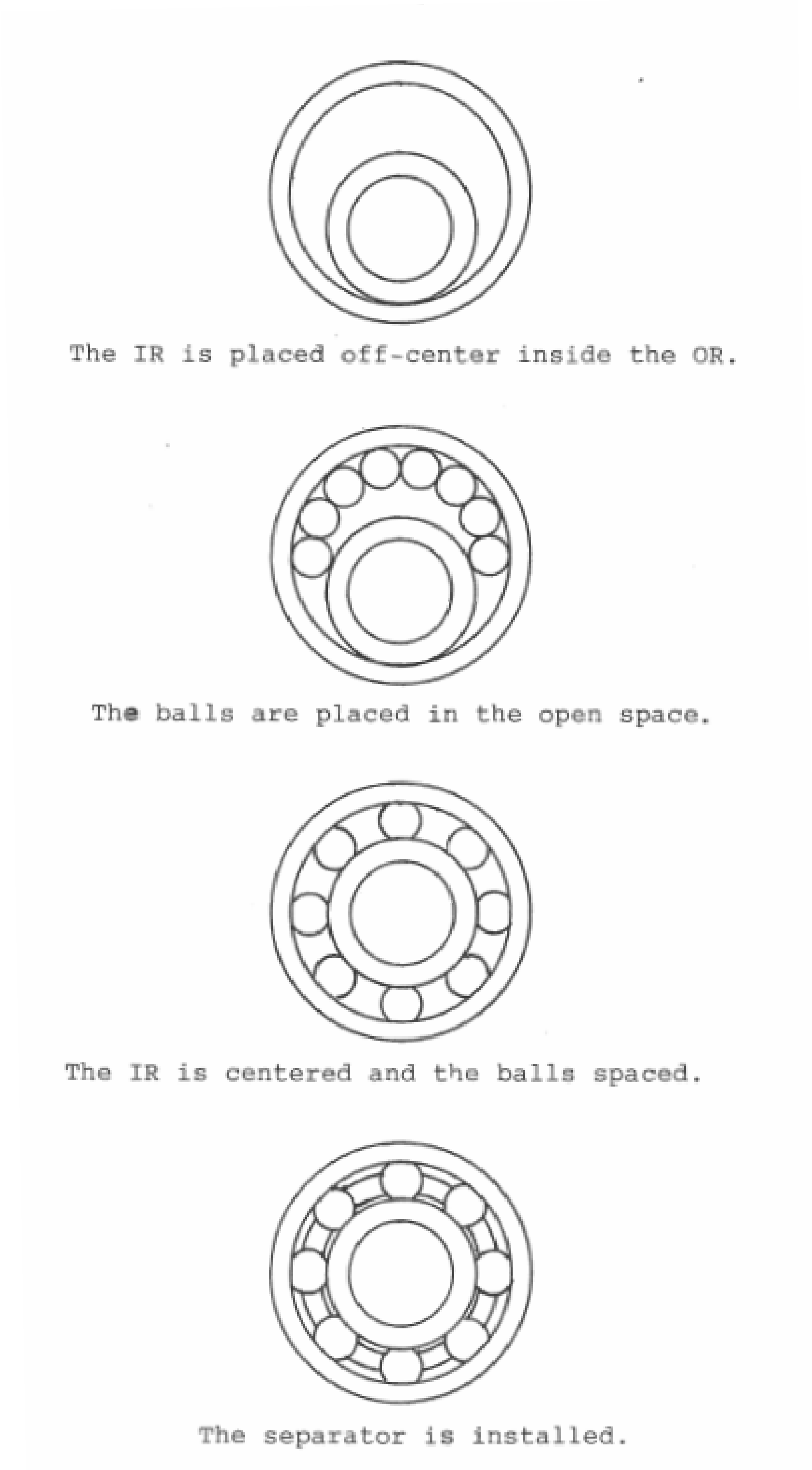
- On a flat surface the inner ring is placed off-center inside the outer ring.
- The balls are loaded inside the crescent shape that is formed.
- The inner ring is centered and the balls evenly spaced.
- The separator is installed. See Figure 4.

Figure 3
Ball Bearing Material



Ball Bearing Ring, Ball, and Separator
Material Temperature Limitation

Figure 4
Conrad Assembly



Ball Bearing Types

The three most commonly used types of ball bearings are the radial bearing, the angular contact bearing, and the double row ball bearing. (See Figure 5.) The radial ball bearing is designed to accommodate primarily radial loads but the deep groove type will support bidirectional thrust loads up to 35% of the radial load before bearing life becomes progressively shorter. The assembled radial bearing is inseparable and may be equipped with seals, shields, and/or snap rings (discussed later).

Angular contact ball bearings are single row bearings designed so that the line of contact between the balls and inner and outer ring pathways is at an angle to a line 90^o to the bearing axis of rotation. The angle between the two lines is called the contact angle as indicated by the letter “a” on the top sketch of Figure 6. In angular contact ball bearing design, one of the pathway shoulders is removed to allow assembly of a maximum complement of balls for increased load carrying capacity. Angular contact ball bearings support both radial and high one-direction thrust loads.

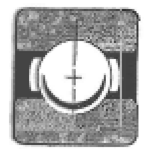
The second sketch on Figure 6 has two angular contact ball bearings mounted *back-to-back*. This type of mounting has good axial and radial rigidity and provides resistance to overturning moments and angular deflection of the shaft.

The third sketch of Figure 6 has two angular contact ball bearings mounted *face-to-face*. This type of mounting has the same axial and radial rigidity as back-to-back mounting but less resistance to overturning moments and more compliance to misalignment or bending of the shaft.

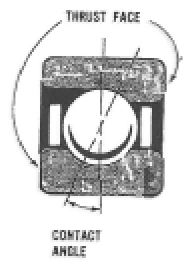
The fourth sketch of Figure 6 depicts two angular contact ball bearings mounted in *tandem* (face-to-back). This mounting arrangement provides resistance to high one-direction thrust loading. The total thrust capacity of the pair is 1.62 times the thrust capacity of one bearing. For even higher thrust loading, three or more angular contact bearings can be mounted in tandem as shown on the fourth sketch of Figure 6.

Double row ball bearings support heavy radial loads, thrust loads from either direction, or combined radial and thrust loads. They are normally used in positions where radial loads exceed the capacity of a single row bearing with a comparable bore and OD. Double row bearings are designed with the bore and outside diameter the same as single row bearings but are narrower than two single row bearings. Figure 7 has two double row angular contact ball bearings. The first sketch has contact angles similar to two single row angular contact bearings mounted back-to-back (contact lines internally diverging). The second (third sketch) has contact angles similar to two angular contact bearings mounted face-to-face (internally converging). Double row ball bearings with contact lines internally diverging can be designed with preload giving it rigidity high enough to resist axial, radial, and overturning loads making it ideal for single bearing mounting such as inside pulleys, gears, and wheels. Double row ball bearings with contact lines internally converging have the same resistance to axial and radial deflection as their counterparts but lack the resistance to angular loading. Larger sizes as shown on the second sketch of Figure 7 have a loading groove in order to assembly a maximum number of balls.

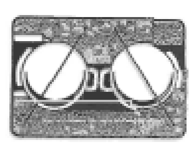
Figure 5
Ball Bearing Types



Radial



Angular Contact



Double Row

Figure 6
Angular Contact Ball Bearings

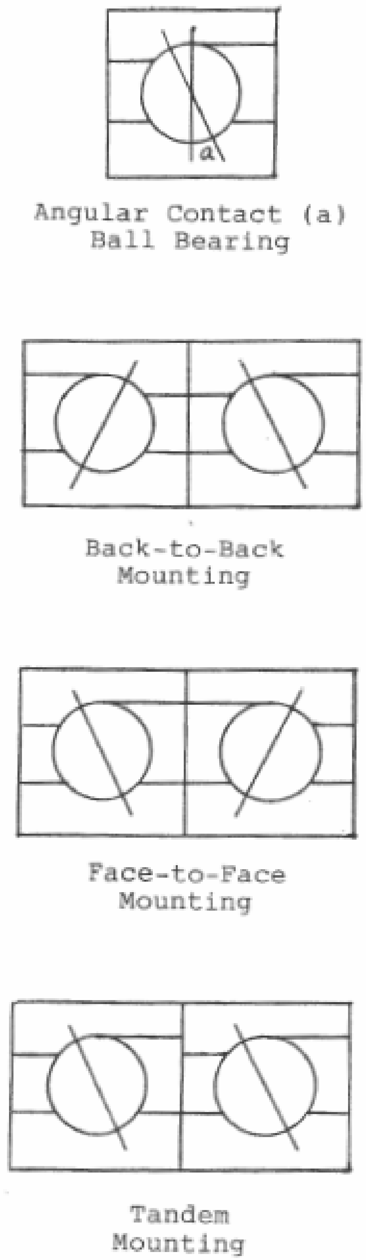
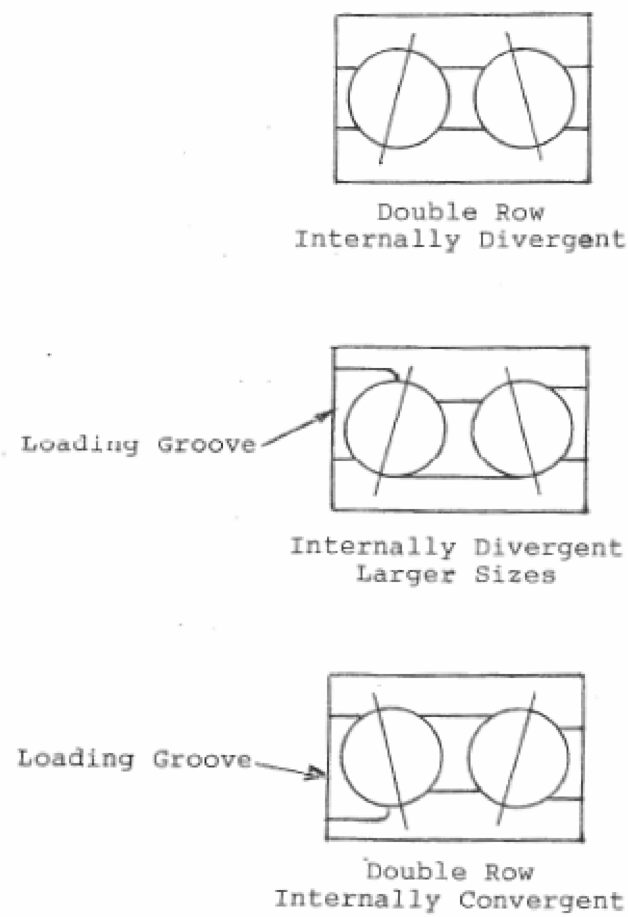


Figure 7
Double Row Ball Bearings



Ball Bearing Sizes

Most ball bearings fall into three series or categories based on their cross section area with the larger sizes being able to support heavier loads. The series are extra-light, light, and medium. (See Figures 8 and 9.) Ball bearing envelope surfaces are generally dimensioned in millimeters, although inch sizes are also available. A common range of sizes offered by industry is from a 4 mm bore, 16 mm OD, 130 lb radial capacity bearing to a 180 mm bore, 280 mm OD, 35,500 lb radial capacity bearing. Figure 10 illustrates the comparison between various ball bearing bore sizes.

Ball bearings and ball bearing mounting surfaces are established by an industry wide standard for their boundary dimensions so that product from various manufacturers can be used interchangeably. As mentioned, it is standard practice to have the bearing rotating ring be a press fit on its mounting surface and the non-rotating ring be a loose fit on its mounting surface. The following table lists the *rotating* inner ring fit and *stationary* outer ring fit (in inches) for a commonly used light series 40 mm ball bearing for the five industry-wide classifications established by the Annular Bearing Engineers Committee (ABEC):

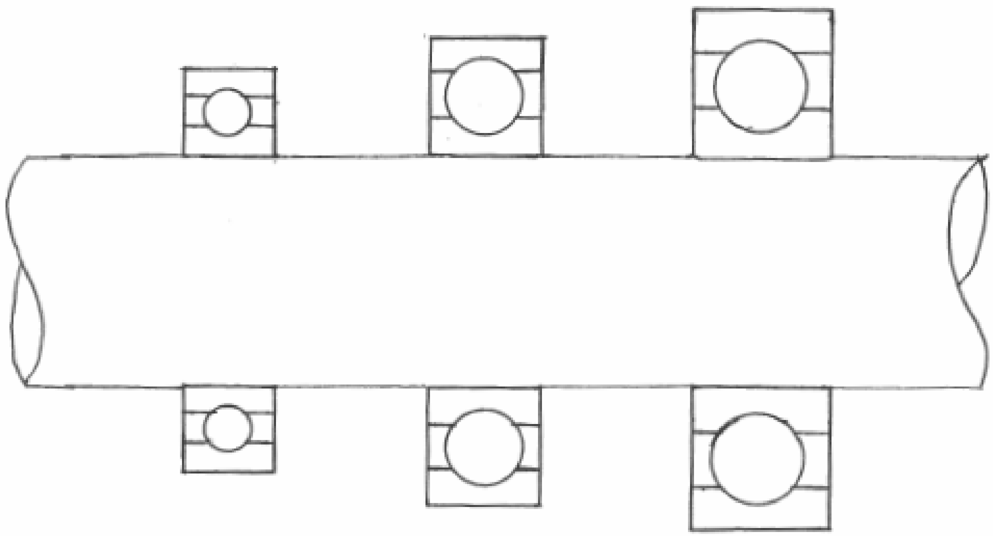
ABEC Number	Rotating Inner Ring	Stationary Outer Ring
1	.0009T-.0001L	.0001T-.0010L
3	.0006T-.0001L	.0001T-.0008L
5	.0004T-.0001L	.0000 -.0006L
7	.0003T-.0001L	.0000 -.0004L
9	.0001T-.0001L	.0000 -.0003L

The following lists the *stationary* inner ring fit and the *rotating* outer ring fit (in inches) for the same 40mm bore ball bearing for all five ABEC fit classifications:

ABEC Number	Stationary Inner Ring	Rotating Outer Ring
1	.0001T-.0009L	.00060T-.00050L
3	.0001T-.0006L	.00050T-.00040L
5	.0001T-.0004L	.00030T-.00030L
7	.0001T-.0003L	.00025T-.00015L
9	.0001T-.0001L	.00020T-.00010L

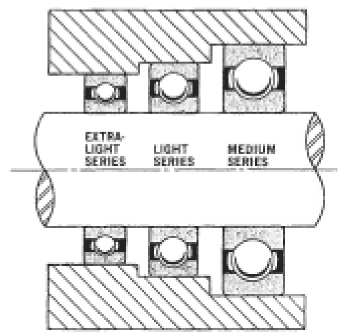
It can be seen from the table that as the ABEC number increases, the tolerances get tighter and the precision gets greater resulting in closer control of the fit of the rings on their mounting surfaces. ABEC 1 is the standard grade of mounting fit precision and is generally adequate for most ball bearing applications. ABEC 3&5 grades are specified where narrower tolerances are required to give desired refinements in mounting and running accuracy. ABEC 7&9 are available for applications where extreme accuracy and true running are necessary. These super precision grades are used in aircraft, machine tool, and fine instrument service.

Figure 8
Ball Bearing Sizes

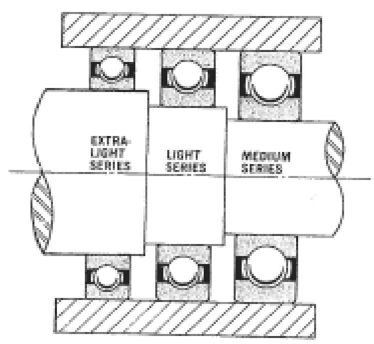


	Extra Light Series	Light Series	Medium Series
Bore	35 mm	35 mm	35 mm
OD	62 mm	72 mm	80 mm
Width	14 mm	17 mm	21 mm
Balls	11-5/16 in	9-15/32 in	8-9/16 in
Capacity	950 lb	1900 lb	2400 lb

Figure 9
Ball Bearing Sizes

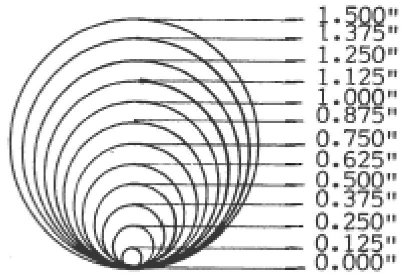


Relative Proportions of Bearings
With Same Inside Diameter

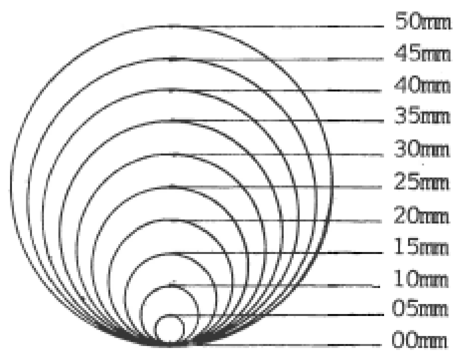


Relative Proportions of Bearings
With Same Outside Diameter

Figure 10
Relative Bore Sizes



Relative Bore Sizes
Inch Series
Full Scale



Relative Bore Sizes
Metric Series
Full Scale

Internal Clearance

All radial ball bearings are normally built with a slight looseness called "internal clearance". If measured in the radial direction, it is called *radial play*, and if measured in the axial direction, it is called *end play*. (See Figure 11.) Most ball bearings are built to radial play. Radial play equals the outer ring pathway diameter minus the inner ring pathway diameter minus twice the ball diameter. This is done in production by gauging the inner and outer ring pathway diameters and selecting a class of balls with a diameter that results in the specified radial play. The purpose of radial play follows:

- It allows the inner ring to misalign slightly with the outer ring without preloading the bearing thus accommodating shaft and housing manufacturing tolerances and shaft deflection under load.

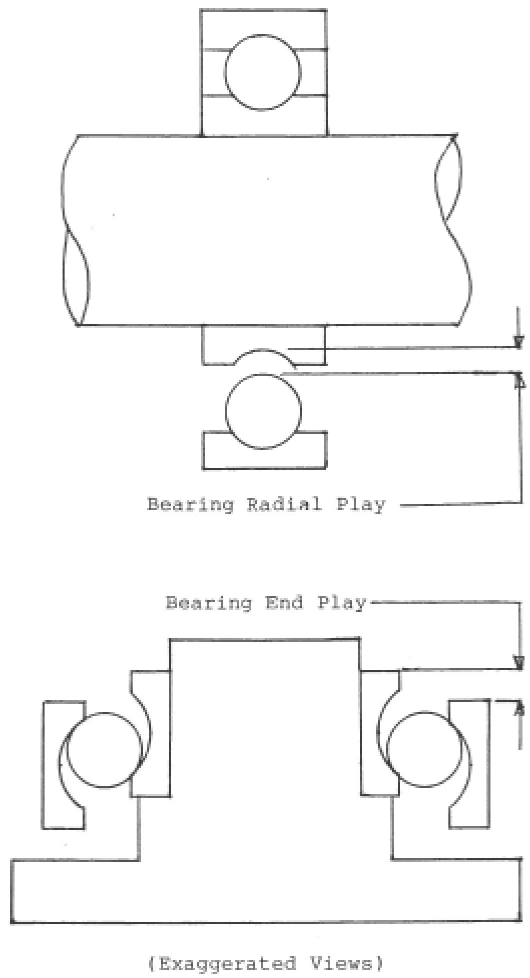
- It permits interference fits with inner rings on shafts and outer rings in housings. Interference fits on shafts and in housings causes the pathway diameter to expand when press fitting inner rings and the pathway diameter to contract when press fitting outer rings. The amount of expansion or contraction is approximately 80% of the press fit.

- It accommodates unequal thermal expansion of the shaft and the housing. Shafts normally run hotter than housings which will expand the inner ring more than the outer ring. This results in producing unwanted preload in the bearing if it had no internal clearance.

As an example, the standard mean radial play for the same light series 40mm bore radial ball bearing is .00085 inches. The ABEC 1 mean press fit for the same bearing is .0004 inches. Eighty percent of .0004 equals .00032 inches. Subtracting .00032 from .00085 equals .00053 inches which is the average running radial play for the subject bearing with a rotating inner ring.

Standard radial play satisfies the requirements of most applications. For unusual applications, special radial play may be required. Lower than standard radial play (.0002 inches average for the above bearing) may be required for bearings operating at low to moderate speeds where accurate radial and axial location is critical. Higher than standard radial play (.00145 inches average) is specified for high speed and high thrust loads; however, high, no load acceleration can cause ball skidding and bearing damage with higher than standard radial play.

Figure 11
Ball Bearing Clearance



Ball Bearing Life

The steel in ball bearings is a special clean grade; however, in rare occasions, a random inclusion (impurity) in the steel will find its way into the highly stressed load zone of a bearing and cause a failure. This is one of the reasons why bearing life is expressed as a probability number. Great strides have been made in refining a cleaner grade of steel; however, to date, science has not found a way to totally eliminate inclusions from appearing in the load zone. B10 life is a calculated number of hours that 90% of bearings are expected to achieve in their lifetime under a specified load and speed without failure. The B10 ball bearing life formula follows:

$$L_{10}=3000(C/P)^{10/3}(500/S)$$

L_{10} is the bearing B10 life in hours. C is the capacity of the bearing and is found in industry catalogs; it is the number of pounds that the bearing can support for 3,000 hours of operation at 500 rpm. The factors in determining bearing capacity include steel cleanliness and quality, ball diameter, number of balls, and inner ring pathway curvature. Ball diameter is the biggest single contributor to bearing capacity. P is the equivalent radial load in pounds which takes into account both radial and thrust loads and is also found in industry catalogs. S is the application speed in revolutions per minute (rpm). It can be seen that because of the 10/3 power exponent, bearing life is especially dependent on load and not as dependent as speed which has no exponent. Should a bearing operate under a number of different loads and speeds, the following equation is used:

$$L_{10}=1/[(t_1/L_1)+(t_2/L_2)+(t_3/L_3)+etc]$$

L_{10} is the bearing B10 life in hours. t is the percent of time spent at each different life (L) level. Bearing life calculations are necessary to determine if predicted values meet actual design requirements. The following table gives approximate bearing life levels for other survival rates should the application require something other than B10 levels:

% Survival	B Life	% B10 Life
99	B1	21
98	B2	33
95	B5	62
90	B10	100
50	B50	400
40	B60	500

Ball Bearing Retention

A commonly used method of retaining bearings is shown on the top sketch of Figure 12. Here a lock washer mounted between the nut and the bearing is keyed to the shaft to prevent rotation. The outside diameter of the lock washer has tabs which are designed so that one aligns with one of the slots on the outside diameter of the nut after the nut is torqued. The tab is then bent into the slot preventing the nut from turning. This system is used to retain wheel bearings on some automotive vehicles. It can be used to lock the nut when torqued tight or when the nut is loose allowing some specified end play.

In some circumstances, there is insufficient space to lengthen the shaft enough to provide for the thread used on the device above. In this instance, a thick washer is secured against the bearing by means of screws which are threaded into the end of the shaft and lock wired to prevent loosening. (See second sketch of Figure 12.)

The simplest method is shown on the third sketch of Figure 12 where a snap ring is inserted into a groove on the shaft. Some snap rings are made tapered so that after installation pressure is always put on the face of the bearing inner ring to prevent axial movement.

A method of clamping outer rings is shown on the top sketch of Figure 13. Here the retainer is fastened to the housing with screws. An important feature is that the clamp is piloted into the same diameter of the housing as the bearing. This accurately locates the clamp in the radial direction which is important for proper functioning of the seal incorporated in the clamp.

The second sketch of Figure 13 has a two piece clamp that provides the shoulder for bearing outer ring retainment. This design allows for through boring the housing for better bearing alignment.

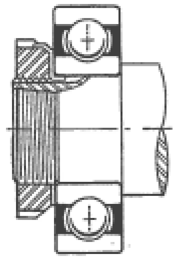
The third sketch of Figure 13 has the bearing outer ring clamped between an outer cap which is fastened to the housing with screws (not shown) and an inner member retained by a wire ring. This design is not recommended where high thrust loads are present in the direction of the wire ring.

There are instances in design when assembly cannot be accomplished because the opening in the housing is too small to allow components to pass through. When this occurs, the opening in the housing is made slightly larger as seen on the upper sketch of Figure 14 where an adapter sleeve is used. This sleeve should be a light tap fit in the housing and should have the bore and OD machined as parallel and concentric as possible.

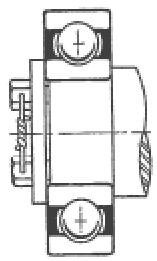
The second sketch on Figure 14 shows a bearing and sleeve arrangement used on a precision spindle (short shaft). The bearings are a press fit in their respective sleeves. Before this is done, the high points of eccentricity of the bearings are positioned diametrically opposite the high points of eccentricity of the sleeves thus compensating for the eccentricity of both parts and improving spindle runout.

It is sometimes necessary to mount ball bearings to allow a certain amount of eccentricity between the shaft and housing for various reasons, one of which is for belt tightening. This is accomplished with a sleeve whose outside diameter is sufficiently eccentric with the inside diameter to produce the desired amount of shaft movement. See the third sketch on Figure 14.

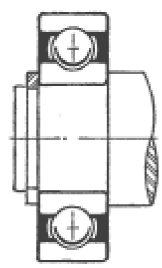
Figure 12
Ball Bearing Retention



Slotted Nut and Tabbed Washer

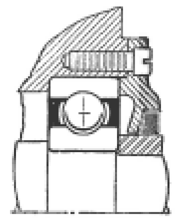


Thick Washer and Lock Wire

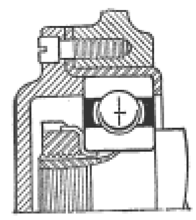


Snap Ring

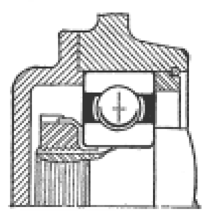
Figure 13
Ball Bearing Retention



Piloted Outer Ring With Clamp
With Seal

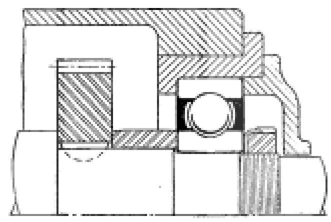


Two Piece Clamp
Thru-Bored Housing

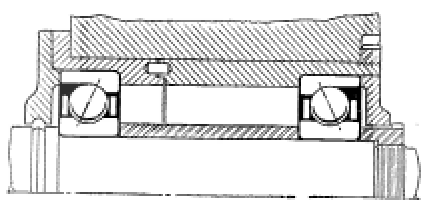


Wire Ring Retainer
Thru-Bored Housing

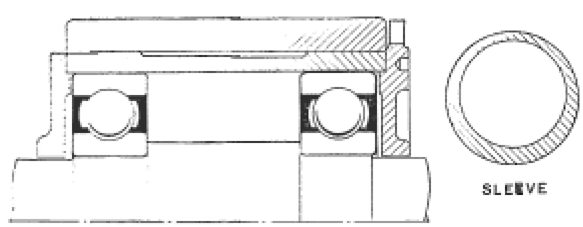
Figure 14
Ball Bearing Retention



Outer Ring Sleeve
Allows Gear Assembly



Precision Sleeve
Improves Spindle Runout



Eccentric Sleeve
For Belt Tightening

Spur Gear Bearing Loads

A common use of ball bearings is in power transmission devices where they are used to support shafts that have gears mounted on them. Spur gears have straight teeth that are aligned parallel to their axis of rotation. They are used to transmit power between two parallel shafts. Spur gears impose only radial loads on shafts and their support bearings. Figure 15 has two spur gears in mesh. In order to calculate bearing loads, the input torque Q, which is usually produced by an engine or electric motor, is calculated as follows:

$$Q = HP \cdot 63{,}025/N$$

Q is the input torque in inch-pounds. HP is the input horsepower. N is the speed of rotation of the driving gear in revolutions per minute (rpm). The input torque is divided into a tangential force and a separating force. The tangential force P is calculated as follows:

$$P = Q/r$$

P is the tangential force of the driving gear in pounds. Q is the input torque calculated above. The letter r is the pitch radius of the driving gear in inches. The separating force between the two gears is calculated as follows:

$$S = P \cdot \tan \alpha$$

S is the gear separating force in pounds. P is the tangential force calculated above. The Greek letter α (alpha) is the tooth pressure angle in degrees. The total radial load on bearing I due to P and S which act 90° apart and can be seen on Figure 15, is calculated as follows:

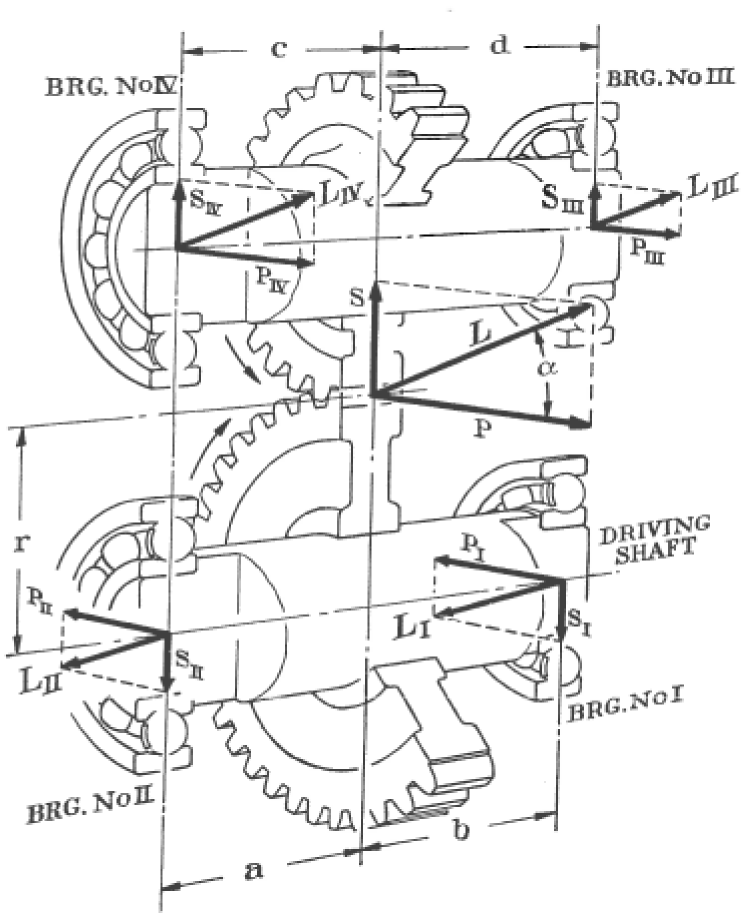
$$R_I = \{[P \cdot a/(a+b)]^2 + [S \cdot a/(a+b)]^2\}^{1/2}$$

R_I is the total radial load on bearing I in pounds. P is the tangential force calculated above. The letters a and b are bearing locating dimensions shown on Figure 15 in inches. S is the separating force calculated above. Since the radial loads due to the tangential force P and the separating force S act 90° apart, the Pythagorean Theorem is used to calculate the resultant radial load on bearing I. Similarly, the total radial load on bearing II due to the tangential and separating forced is as follows:

$$R_{II} = \{[P\cdot b/(a+b)]^2+[S\cdot b/(a+b)]^2\}^{1/2}$$

The total radial load on driven shaft bearings III and IV is calculated using the same equations that were used for drive shaft bearings I and II except substitute bearing locating dimensions c and d for a and b. The rpm of the driven gear equals the rpm of the driving gear times the number of teeth in the driving gear divided by the number of teeth in the driven gear. Enough information is now available to calculate ball bearing life which the bearing Application Engineer may have to do a number of times before the bearing that meets all design objectives can be found.

Figure 15
Spur Gear Bearing Loads



Helical Gear Bearing Loads

Helical gears are similar to spur gears except that the teeth are at an angle to the gear centerline as opposed to being parallel to the gear centerline. Because of the helix angle, the transmitted torque is divided into three vectors instead of two as is the case for spur gears. The three forces are tangential, separating, and thrust. (See Figure 16.) Since the thrust force is applied at a distance from the gear center, it not only produces a thrust load on support bearings I and IV, but it also produces a moment load that is reacted by a radial load on all four bearings. Figure 16 shows the thrust force at the mesh as vector T and the thrust load it produces as T on bearings I and IV. The radial load from the thrust couple is shown as U_I, $U_{II,}$ U_{III}, and U_{IV} on support bearings I, II, III, and IV respectively. The input torque Q, the tangential force P, and the separating force S are calculated as was done previously. The equation for the thrust load is as follows:

$$T=P\cdot\tan\gamma$$

T is the gear thrust in pounds. P is the tangential force in pounds. The Greek letter γ (gamma) is the helix angle of the gears in degrees. (One commonly used helix angle is 15^o.) The radial loads on bearing I due to the tangential force P, the separating force S, and thrust force couple U are as follows:

$$P_I=P\cdot a/(a+b) \qquad S_I=S\cdot a/(a+b) \qquad U_I=T\cdot r_1/(a+b)$$

P_I, S_I, and U_I are radial loads on bearing I as shown on Figure 16. The letters a and b are bearing I locating dimensions in inches. The letter r_1 is the pitch radius of the driving gear in inches. The total radial load on bearing I is calculated as follows: (There is a thrust load T on bearing I.)

$$R_I=[P_I^{\,2}+(S_I-U_I)^2]^{1/2}$$

The radial loads on bearing II are calculated as follows:

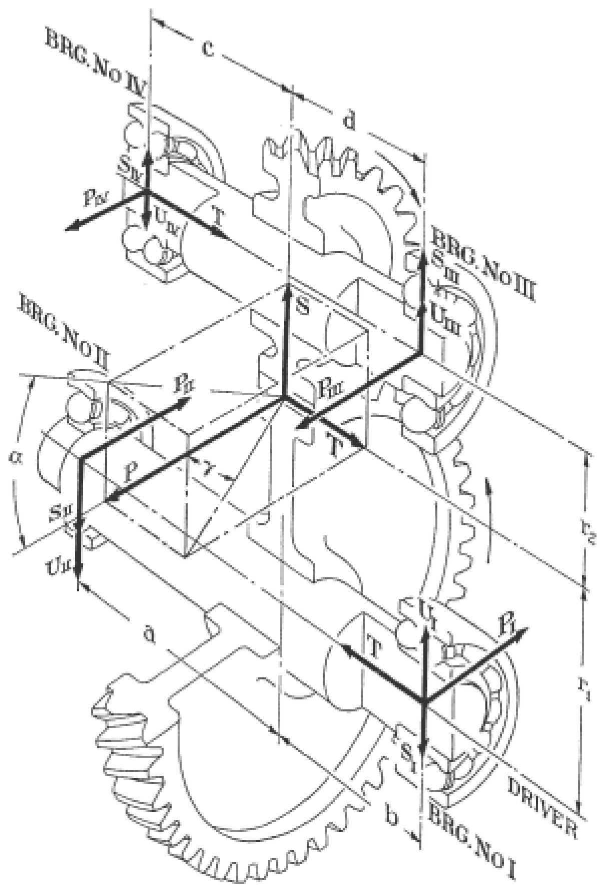
$$P_{II}=P\cdot b/(a+b) \qquad S_{II}=S\cdot b/(a+b) \qquad U_{II}=T\cdot r_1/(a+b)$$

The total radial load on bearing II is as follows: (There is no thrust load on bearing II.)

$$R_{II}=[P_{II}^{\,2}+(S_{II}+U_{II})^2]^{1/2}$$

The load on bearings III and IV are calculated in a similar manner. The rpm is calculated as was done for spur gears. Now that the total radial loads of all four bearings and the thrust load of bearings I and IV are known, each bearing life can be calculated.

Figure 16
Helical Gear Bearing Loads



Bevel Gear Bearing Loads

Bevel gears are used when the driving and driven shafts are not parallel but are at an angle to each other such as 90^o as shown in Figure 17. Bevel gear teeth are tapered. The vector E is the input force drawn normal to the driving tooth contact and can be resolved into three forces. The first, P, is directed vertically down from the driving gear (pinion) axis; the second, T_G, is along the gear axis; the third, T_P, is along the pinion axis. The tangential force P is calculated as follows:

$$P=Q/r_1$$

P is the tangential force in pounds. Q is the input torque in inch-pounds. r_1 is the pinion mean pitch radius in inches. The gear thrust force T_G is calculated as follows:

$$T_G=P \cdot \tan\alpha \cdot \cos\beta$$

T_G is the gear thrust force in pounds. P is the tangential force calculated above. The Greek letter α (alpha) is the tooth pressure angle in degrees. The Greek letter β (beta) is ½ pinion pitch angle in degrees. The pinion thrust T_P is calculated as follows:

$$T_P=P \cdot \tan\alpha \cdot \sin\beta$$

T_P is the pinion thrust in pounds. The Greek letters α and β are as above. The radial loads on bearing I due to the tangential force P, the gear thrust T_G, and the pinion thrust T_P are as follows:

$$P_I=P(a+b)/b \qquad T_{GI}=T_G(a+b)/b \qquad U_I=T_P(r_1/b)$$

P_I, T_{GI}, and U_I are the three radial loads on bearing I. The letters a and b are bearing locating distances in inches as shown on Figure 17. The letter r_1 is the pinion mean pitch radius in inches. The total load on bearing I is a radial load and is as follows:

$$R_I=[(P_I)^2+(T_{GI}-U_I)^2]^{½}$$

The tangential, gear thrust, and pinion thrust loads on bearing II are as follows:

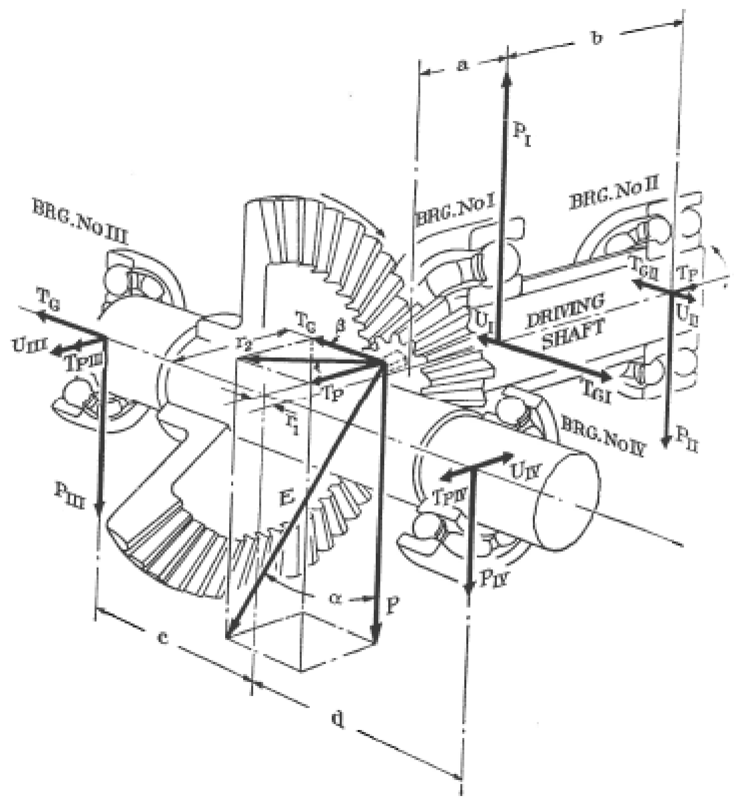
$$P_{II}=P \cdot (a/b) \qquad T_{GII}=T_G \cdot (a/b) \qquad U_{II}=T_P \cdot (r_1/b)$$

The total radial load on bearing II is as follows:

$$R_{II}=[(P_{II})^2+(T_{GII}-U_{II})^2]^{½}$$

There is a thrust load on bearing II equal to T_P. The speeds of bearings I and II are given. Now that the loads and speed are known, bearings I and II lives can be calculated. The lives of bearings III and IV can be calculated in a similar manner. The speed of rotation of bearings III and IV equals the rpm of the pinion times the rpm of the pinion divided by the rpm of the gear.

Figure 17
Bevel Gear Bearing Loads



Preloading

It is important in machine design that the products manufactured are made as accurately as possible. One way to do this is to ensure that the shafts and spindles on these machines are rigidly supported and run true. The graph at the top of Figure 18 has a load vs. deflection plot for a typical angular contact bearing. It can be seen that the slope is the greatest at the beginning of the curve and lessens as the curve progresses to the right. If something could be done to make bearings run higher on the curve, the spindles that the bearings support would be more rigidly supported and run truer. The process that is used to do this is called “preloading”.

Figure 19 has a sketch of a spindle supported by two angular contact ball bearings. The inner rings of the bearings are clamped tight against the shaft shoulder using a locking nut. Each outer ring is mounted in its own sleeve. Torquing nut N puts an axial load on the right (back) bearing through sleeve B. This axial load is then transferred through the clamped inner rings to the left (front) bearing putting the shaft in tension and preloading the two bearings.

Assume that nut N is torqued so that a preload of 3000 pounds tension is put on the bearings and then a work force of 2500 pounds is applied to the right on the front bearing end of the shaft. This additional force increases the load on the front bearing while decreasing the tensile preload on the spindle thus decreasing the load on the rear bearing. The front bearing is now supporting less than the preload and work load (3000+2500=5500 lb) and the rear bearing is supporting less than the 3000 lb preload. Both bearings are operating above the steepest part of the curve and are giving the shaft greater support and accuracy.

The second and third plots on Figure 18 are used to determine the deflection and final load on each of the two spindle bearings. The second plot shows that the part of the curve from 0 to 3000 lb is rotated up. The load of 3000 lb was chosen because it is the initial preload put on the bearings. This section of the plot was rotated up because this is the path taken when preload is being relieved. The third plot shows the rotated section moved to the right to 5500 lb which is the momentary load on the front bearing. Moving up the transplanted curve from 5500 lb to the original curve shows that the final load on the front bearing is 4500 lb. Applying the distance R to the 3000 lb line of the original curve shows that the rear bearing final load is 2000 lb.

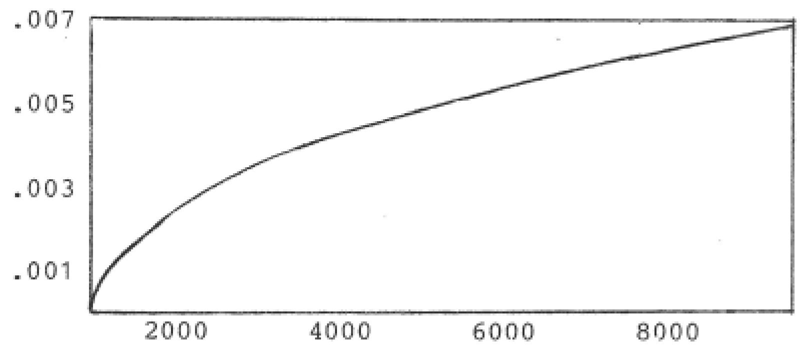
Without preload the 2500 lb work load would have produced a shaft deflection of .003 inch, but with preload, the 2500 lb work force shaft deflection is down to .001 inch. The reduction in shaft deflection from .003 inch to .001 inch is a big improvement when the machine spindle is used to support a wheel that is grinding bearing parts that have tolerances as small as .00005 inch.

With the preload set at 3000 lb, it can be seen on the graph that each bearing deflects .0035 inch. In order to reduce the preload down to zero, the spindle would have to be deflected twice that amount or .007 inch. The force to eliminate the preload can be seen on the graph to be 10,000 lb or 3.33 times the amount of the preload itself.

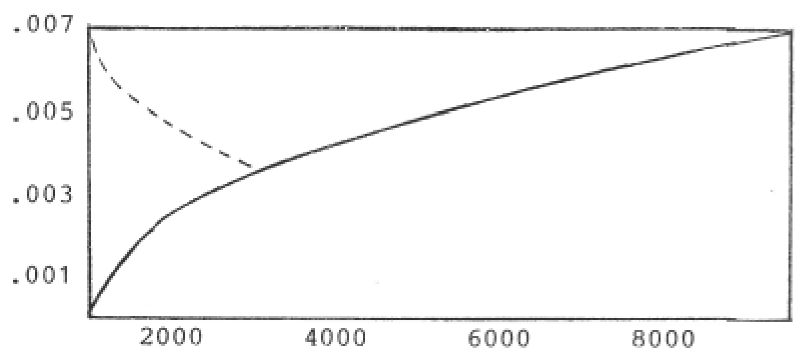
A double row bearing can be manufactured preloaded. The graph on Figure 19 compares the load vs. deflection curve of a double row ball bearing to a similar sized non-preloaded single row ball bearing. It can be seen how much more stiffness the preloaded double row ball bearing has compared to the non-preloaded single row bearing. The double row bearing preload is relieved at 5000 lb and after that the two plots are parallel.

Figure 18
Bearing Preloading

Deflection in inches vs axial load in pounds



Typical Load vs Deflection Curve



0 to 3000# Part of Curve Rotated Upward

Rotated Part of Curve Moved to 5500#

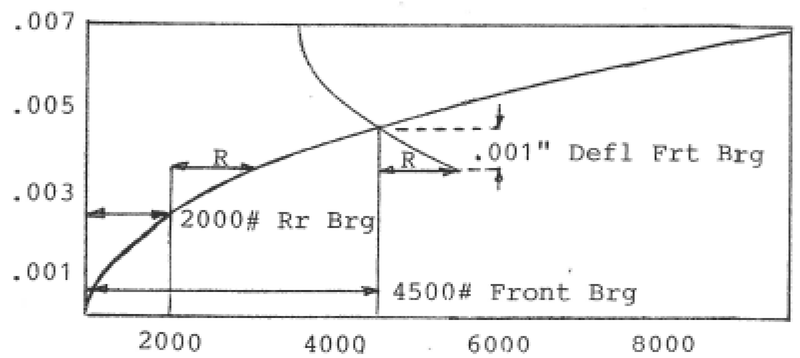
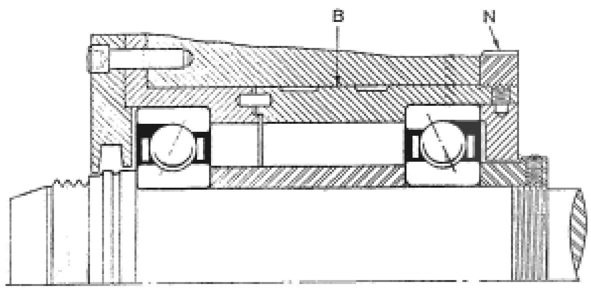
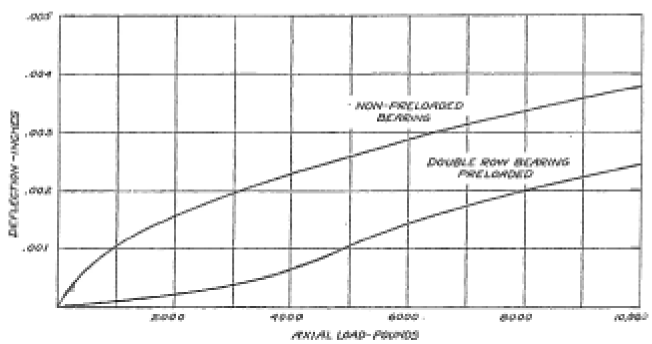


Figure 19
Bearing Preloading



Preloaded Angular Contact Bearings



Non-Preloaded vs Preloaded

Double Row Bearing

Lubrication

Highly refined mineral oils are among the best lubricants for ball bearings. Synthetics have been developed that are good but some do not form *elastohydrodynamic* (EHD) films as well as mineral oils. EHD refers to what happens to the oil between the ball and pathway when the bearing is rotating. Research has shown that a film of oil builds up and, under some circumstances, becomes thick enough to completely separate the balls from the pathways. Thicker films result in longer than what the life equation predicts while films that are too thin result in excessive metal-to-metal contact and shorter lives than what the equation predicts. Empirical equations have been developed based on laboratory testing of the various factors that affect oil film development. One such equation follows:

$$T=B(OS)^{x}L^{-y}$$

T is a measure of oil film thickness. Figure 20 shows that values of T below 1.5 can expect marginal lubrication for bearings while values of T above 1.5 can expect good lubrication. B is a bearing factor which takes into account physical characteristics of bearings that influence oil film thickness. B is largely dependent on bearing size with larger diameter bearings developing thicker oil films. The kind of bearing used has a smaller effect with radial and angular contact ball bearings falling into the middle of the category. O is an oil factor which is influenced primarily on oil viscosity at bearing operating temperature. The type of oil used has a smaller effect with naphthenic being the best, paraffin lying in the middle, and synthetic being the worst. S is a speed factor which shows that higher speeds generate thicker oil films because of the wedging effect of oil into the contact zone. L is a load factor which demonstrates that higher loads result in thinner films. Graphs of all the above factors have been developed which makes it easy to calculate oil film thickness and its affect on bearing life. Use of the graphs simplifies the equation as follows:

$$T=BOSL$$

The above equation can now be rearranged as follows to determine the oil viscosity needed for a bearing to operate successfully in an application:

$$O=T/BSL$$

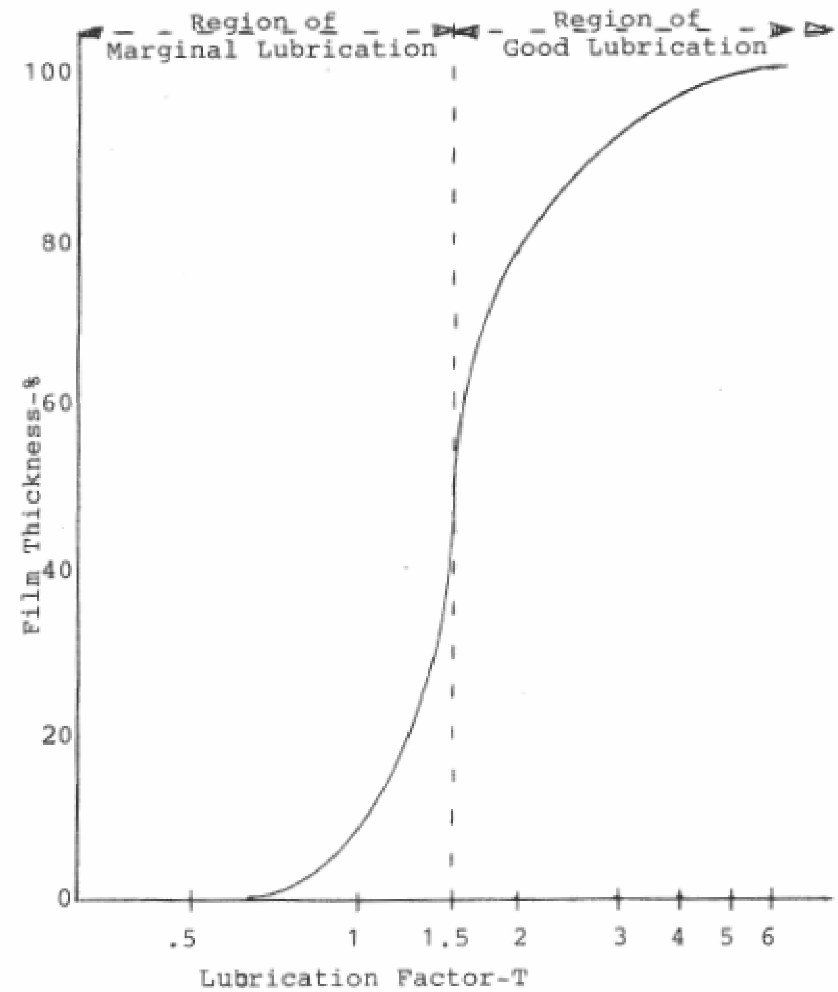
As a sample problem the above equation along with graphs for each factor (not available) are used to determine the oil viscosity needed for a 40mm bore medium sized ball bearing operating at 1000 rpm and with an equivalent load of 1000 lbs. The exercise shows that an oil viscosity of 215 SSU (Saybolt Seconds) is required at the bearing operating temperature for good lubrication and bearing life.

Commonly used means for delivering oil to bearings include jet, bath, mist and wick feed. The best overall system is oil jet combined with a recirculating system. This method directs a pressurized stream of oil into the bearing load zone. The oil is then drained back to a sump where it is filtered, cooled, and returned. This system is good for a variety of loads and speeds. The oil bath method is commonly used in gear boxes. The housing is filled with oil until it just touches the lowest rotating component distributing the oil throughout the gearbox internals. Mist systems use pressurized air to atomize oil. The mixture is then sprayed on the bearing where it lubricates and cools. Air-oil mist systems are used primarily for high speed applications. Wick systems use an absorbent material to store oil and slowly deliver it to a bearing in a controlled manner. Wick systems are used in electric motors.

A simple method of lubricating bearings is by using grease. A carefully measured quantity of grease is evenly distributed throughout the bearing where it is contained by seals or shields. This configuration can run for the life of the bearing. Grease consistency is important. Greases that are too soft will cause excessive churning losses in a bearing while greases that are too hard will not lubricate properly. The following is a list of important greases:

- Mineral oil grease for general purpose operation from -30^{o}F to +300^{o}F
- Ester based greases for operation from -100^{o}F to +350^{o}F
- Silicone based greases for operation from -100^{o}F to +350^{o}F but lack good load carrying capacity

Figure 20
EHD Lubrication



Ball Bearing Closures

Bearing closures are sealing devices that are installed on one or both sides of a ball bearing to contain grease lubricant, to protect against moisture or foreign substance entry, or to control the flow of lubricant into the bearing when exposed to an oil sump. Grease lubricated and double sealed ball bearings can offer maintenance free operation for the life of the bearings.

At the top of Figure 21 is a sketch of a single rubber lip seal installed on a standard width ball bearing. The ability to design and assemble effective sealing elements on both sides of a standard width ball bearing without going outside the bearing envelope and to grease lubricate the bearing for a lifetime of operation, offers a distinct advantage to the designer in packaging mechanical devices over having to provide alternative means of lubrication for the bearing or having to provide extra space to accommodate an extra wide sealed bearing.

The seal design shown at the top of Figure 21 has rubber molded around a flat steel ring insert which imparts rigidity and strength to the construction and helps to control lip sealing pressure which is needed to accommodate relative movement of the bearing rings. The seal is snapped into a groove in the outer ring where the rubber provides a leak proof joint. A standard design synthetic rubber seal has an operating temperature range of -65^{o}F to +225^{o}F. There are other elastomeric materials available for higher temperature operation. The limiting speed of operation is 2000 rpm for a large 70 mm bore bearing to 13,000 rpm for a small 10mm bore bearing.

Another version of the single lip seal is shown as the second sketch of Figure 21. It consists of a steel shield on the outside with a rubber lip seal molded on the lower inside. The metal is positioned on the outside to protect the bearing internals from hard foreign object entry. The metal is crimped into a groove on the outer ring and becomes a permanent part of the bearing. The operating temperature range and limiting speed of operation are the same as the standard snap-in lip seal.

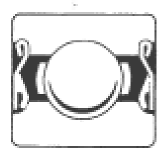
The third sketch of Figure 21 is of a triple lip seal with the outer steel shield protection as discussed above. The seal is called “land riding” because the three lips ride on the inner ring outer diameter rather than on a notch as do the previous lip seals. Besides having three lips for triple sealing, grease can be packed between the lips to further impede contaminant and moisture entry and to lubricate the lips. This concept is the ultimate in lip seal design for heavy duty applications. The drag of the three lips causes an increase in torque level of the bearing; consequently, speeds are limited to 30 rpm for the large bearing described above and 2500 rpm for the small bearing. It is commonly used on farm machinery, construction equipment, and automotive engine coolant pump bearings.

The fourth sketch on Figure 21 is of a felt seal. It is held between two steel pieces which are crimped into a notch on the outer ring. Felt seals are good for lubricant retention and light particle exclusion and their low friction allows for higher speeds of operation than all other seal designs. The limiting speed is 3000 rpm for the large bearing and 19,000 rpm for the small bearing. The limiting temperature is 275^{o}F which is the charring temperature of the felt element.

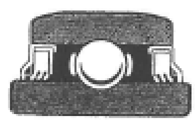
The bottom closure on Figure 21 is a one-piece all metal design called a shield. It is crimped permanently into a groove in the outer ring. It does not contact the inner ring so it does not limit the bearing speed. It is used to contain grease or control the amount of oil ingested by the bearing when exposed to an oil sump. Excessive oil in a bearing can cause high running torque and subsequent failure.

Figure 21
Ball Bearing Closures

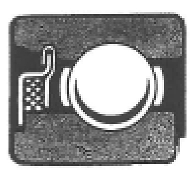
Standard Rubber Lip Seal



Lip Seal With Outer Guard



Triple Lip Seal With Guard



Felt Seal

Double Shields

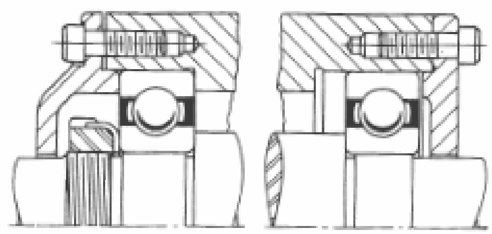
Radial Ball Bearing Application

The application of ball bearings involves more than load and speed calculations. Of equal importance is how bearings are to be mounted and retained in order to minimize machining and aid assembly. Fitting one bearing loose in the housing to accommodate machining tolerances and differential thermal expansion between the shaft and housing is a major design consideration. The sketch at the top of Figure 22 describes how the bearing on one end of the shaft is clamped tight and the other is allowed to move axially in the housing. The second sketch illustrates how both bearings can be made to float if shaft end play is not critical. The third sketch shows how loading grooves put in bearing rings enabling the assembly of extra balls can be used to support heavy radial loads when high thrust loads are not present.

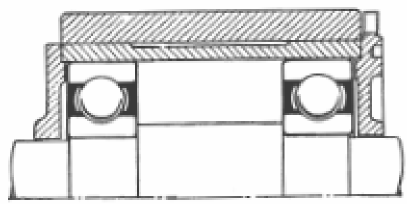
The top sketch of Figure 23 has a flexible shaft handpiece supported by two single row radial ball bearings. The flexible shaft connects on the right and the appropriate tool attaches to the left. The assembly must be small enough to conveniently fit inside a clasped hand. Single row radial ball bearings are well suited for this application where moderate thrust loads are resisted by a single bearing. The larger bearing on the right is axially clamped both on the shaft and in the housing to support the two-direction thrust loading. The long, tightly clamped sleeve helps to stiffen the shaft. The smaller bearing on the left is free to float to accommodate manufacturing tolerances and differential thermal expansion between the shaft and housing.

The sketch at the bottom is of the flexible shaft multi-speed drive. The flexible shaft attaches at the upper left with the pulley tensioning device at the top and the drive motor and pulley at the bottom. The drive motor bearings have felt seals on the exposed end to contain grease lubrication with a minimum of friction. Both bearings are mounted to float in the housing to accommodate manufacturing tolerances and differential thermal expansion. The upper multi-grove pulley is supported with a larger radial ball bearing on the right which supports moderate bi-directional thrust loads and incorporates a felt seal on the exposed end to contain the lubricant. The left hand bearing is free to float.

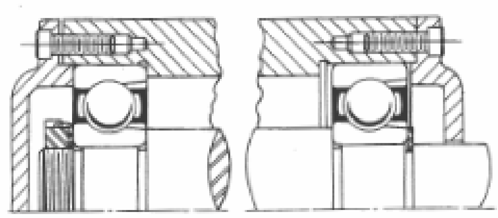
Figure 22
Radial Ball Bearing Application



The bearing on the left is clamped to the housing and the shaft. The bearing on the right is free to accomodate shaft thermal expansion and tolerance build-up.

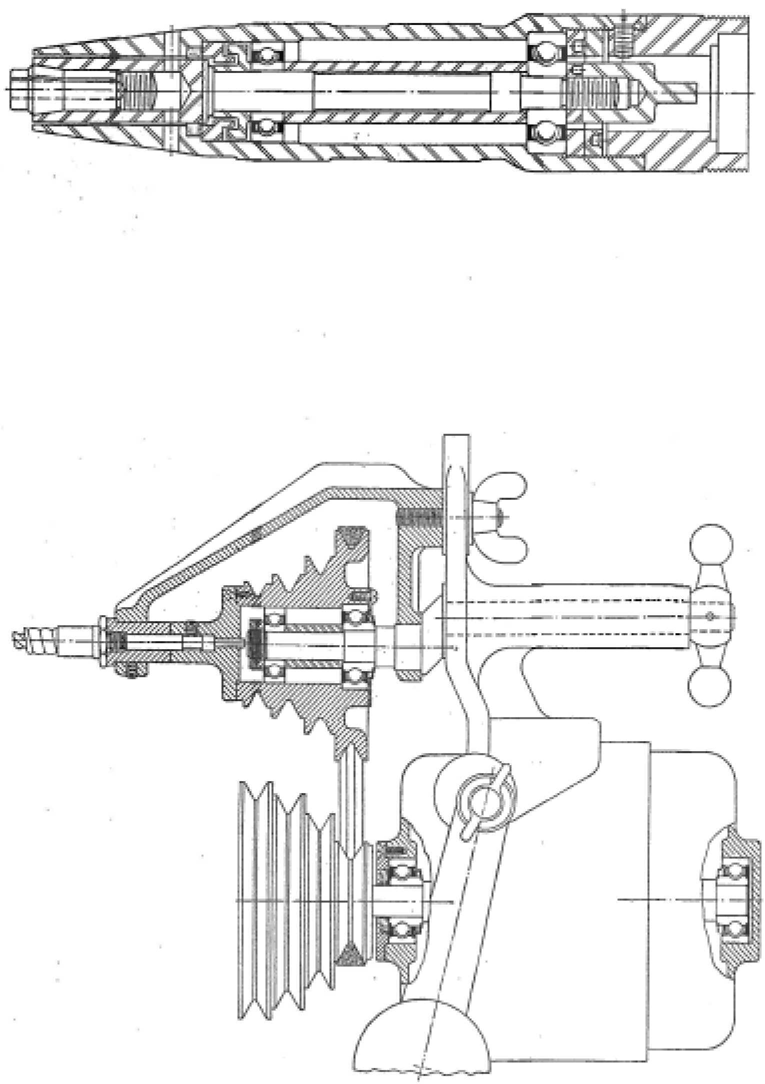


Both bearings can be made to float in the housing if shaft end play is not critical.



When thrust loads are low, loading groove bearings can be used to take heavy radial loads.

Figure 23
Radial Ball Bearing Application



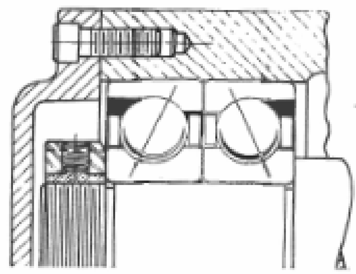
Angular Contact Ball Bearing Application

The top sketch of Figure 24 shows how two angular contact ball bearings mounted back-to-back with inner rings clamped and outer rings free to float can be used to resist high moment loads. The second sketch on Figure 24 shows how two angular contact ball bearings mounted face-to-face with both inner and outer rings clamped can be used to accommodate high shaft misalignment. The third sketch on Figure 24 illustrates how two angular contact ball bearings mounted in tandem can be used to support high one-direction thrust loads.

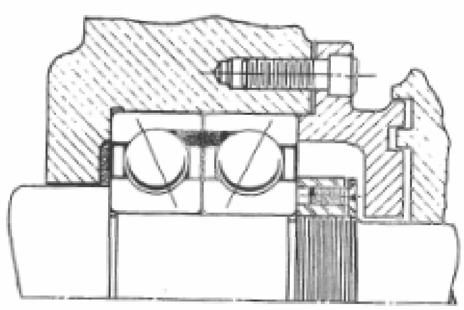
The two precision bench lathe spindles shown on Figure 25 employ a light series radial ball bearing on the left side and two extra light series angular contact ball bearings on the right side. This is done so that the housing can be thru-bored for better bearing and shaft alignment. All three bearing inner rings on both spindles are clamped tight. The two angular contact bearings on the right are back-to-back mounted and are precision off-set ground so that clamping the inner rings results in the correct preload. The back-to-back mounting adds rigidity to the shaft for better alignment under load. The clamped flange on the right bearing pair outer ring fixes the two bearings while the extreme left hand bearing is free to float to accommodate manufacturing tolerances and differential thermal expansion.

The boring machine spindle at the top of Figure 26 is supported by a single radial ball bearing on the right and two angular contact ball bearings on the left. The two angular contact ball bearings are fixed while the single row radial ball bearing is free to float axially to accommodate above-mentioned length changes. The two angular contact ball bearings have the ring faces precision ground so that when the inner rings are clamped tightly together, the prescribed internal preload is automatically established. The two angular contact bearings are mounted back-to-back (DB) to provide the maximum amount of resistance to shaft bending caused by the nearby gear. The sketch at the bottom of Figure 26 has a massive gear cutting spindle supported by two angular contact bearings mounted face-to-face (DF) at the left and two spaced apart angular contact bearings at the right. The right hand bearings are mounted with the contact angles internally divergent to provide the smaller diameter section of the shaft with added rigidity.

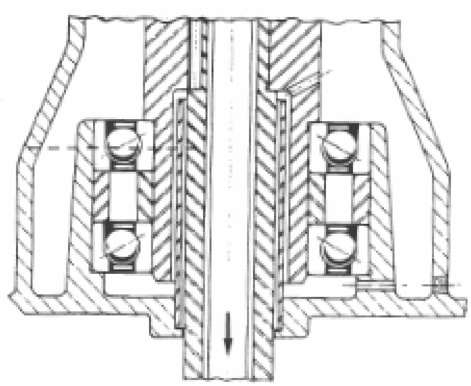
Figure 24
Angular Contact Ball Bearing Application



Maximum resistance to high moment loading is obtained by using two angular contact ball bearings mounted back-to-back.



Compliance to high shaft misalignment is accomodated by using two angular contact ball bearings mounted face-to-face.



Support of high one-direction thrust loading is accomplished by using two angular contact ball bearings mounted in tandem. The thrust is downward on the shaft.

Figure 25
Angular Contact Ball Bearing Application

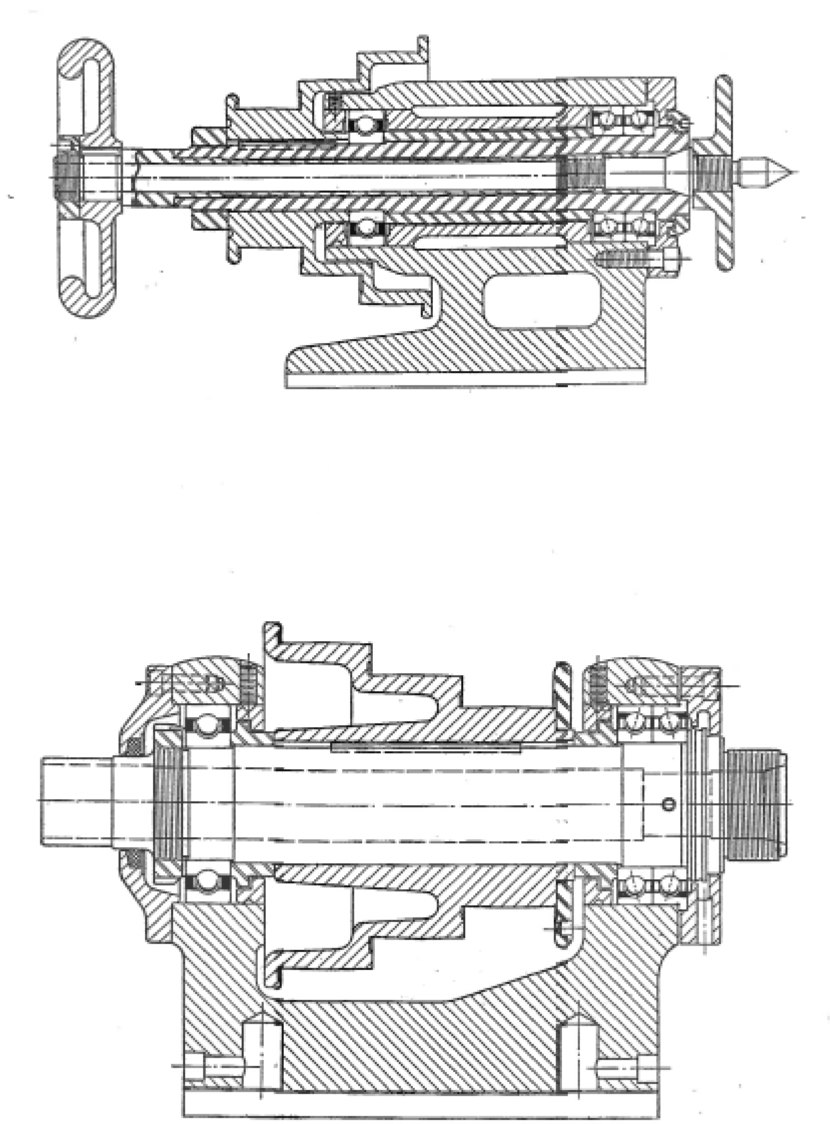
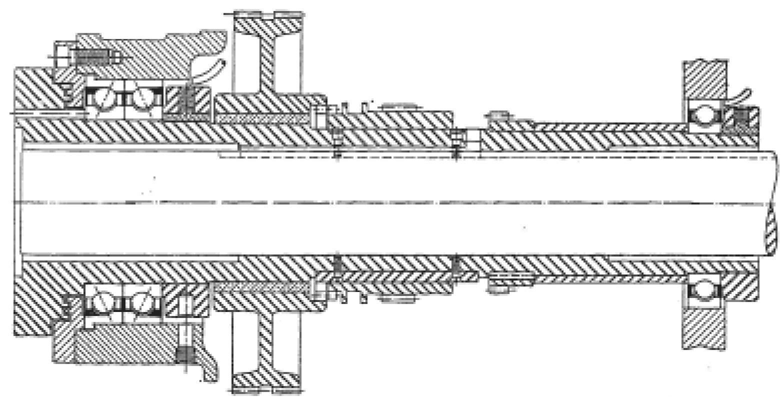
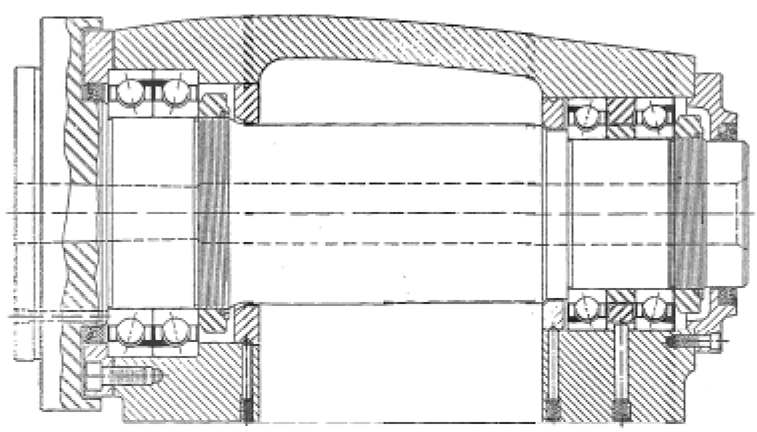


Figure 26
Angular Contact Ball Bearing Application



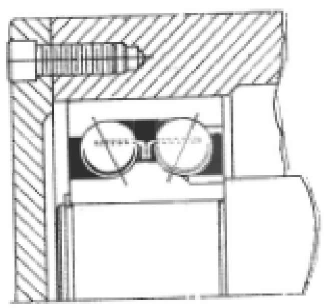


Double Row Ball Bearing Application

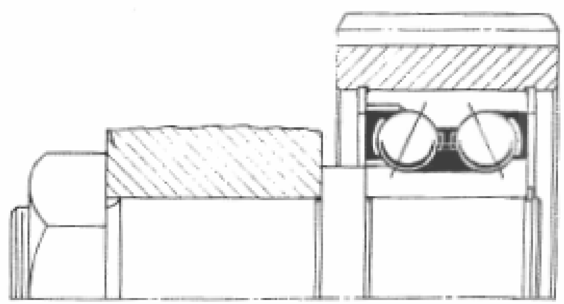
The advantage of using double row ball bearings is that they fit in a narrower space than two single row ball bearings with the same ID and OD. The top sketch of Figure 27 has a double row ball bearing with internally converging contact angle lines. This design can support heavy radial loads and is compliant to shaft misalignment. The lower sketch on Figure 27 has a double row ball bearing mounted with internally diverging contact angle lines supporting a gear. The advantage of this configuration is that it can be used in a single bearing mounting and support heavy radial and overturning moment loading. The upper sketch on Figure 28 has a double row ball bearing retained with a snap ring. This eliminates the need for a flange on the housing and allows thru-boring for better bearing alignment. The lower sketch on Figure 28 has a double row ball bearing also with internally diverging contact angle lines. The bearing is clamped tight on both the inner and outer rings and is able to support heavy reversing thrust loads.

The sketch on Figure 29 has a speed change mechanism. Both shafts are supported by a double row bearing on one end and a radial ball bearing on the other. (The third set of bearings support gears.) The DB mounted double row ball bearings mounted close to the gears gives shafts needed stiffness for proper gear teeth alignment. Notice that snap rings are used to mount the double row bearings on the left eliminating the need for housing shoulders and allowing thru-boring the housings for good bearing and gear alignment. A double row ball bearing is used to give rigid support to the smaller gear at the top. The larger gear at the bottom is supported by two wide-mounted single row radial ball bearings.

Figure 27
Double Row Ball Bearing Application

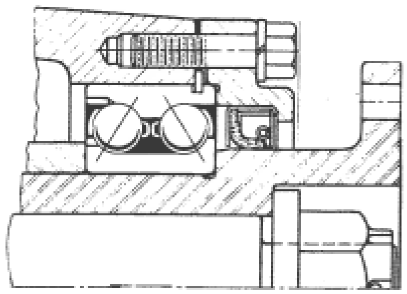


Double row ball bearings with contact angles internally convergent can take misalignment and heavy radial loads.

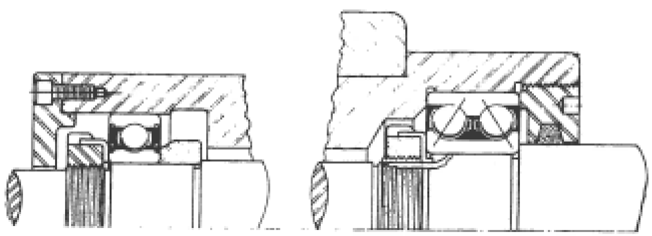


Double row ball bearings with contact angles internally divergent can take heavy overturning moment loading.

Figure 28
Double Row Ball Bearing Application

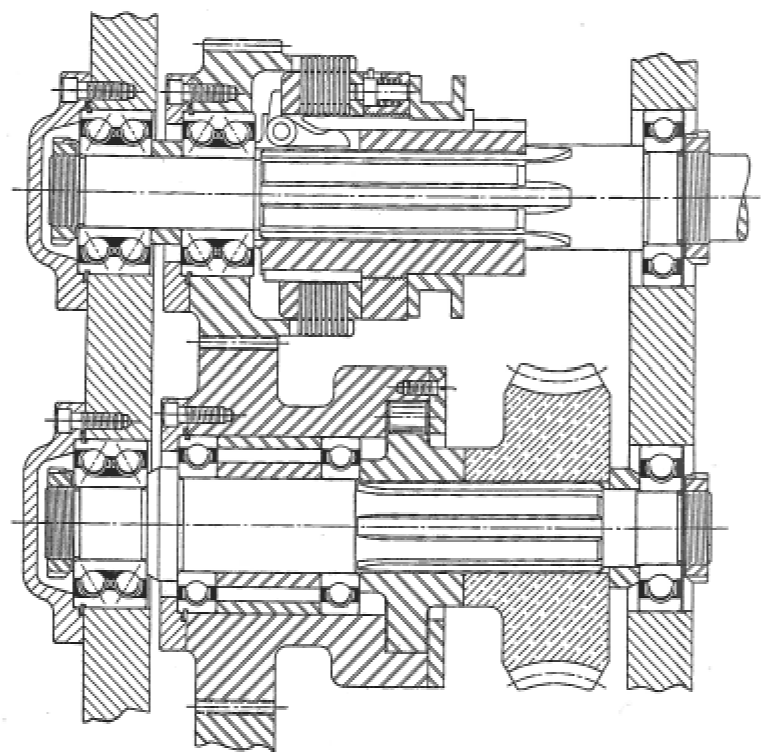


The housing can be thru-bored when a snap ring is used on the bearing outer ring.



A double row ball bearing is used when reversing thrust loads are present in an application.

Figure 29
Double Row Ball Bearing Application



Specialty Ball Bearing Product

Cam follower bearings have thick outer rings to provide for the strength and shock resistance required for an application where the outer rings are not supported by a housing. They are sealed, Conrad type bearings with spherical OD's for minimizing the effects of misalignment. They have bores dimensioned to receive standard machine bolts. A sketch of a cam follower bearing is shown at the top of Figure 30.

Adapter bearings have extended inner rings with eccentric locking collars for easy mounting on standard commercial grade size shafts. The collar is rotated and locked in place with a set screw for permanent bearing mounting. They are designed for installations where loads and speeds are moderate and concentricity requirements not critical. The bearings are lubed and sealed for life and have spherical outer rings for misalignment compensation. Two piece flange mounting units are available which clamp onto the outer ring as shown on the lower sketch of Figure 30.

Disc harrow bearings are for agriculture use. They are sealed, Conrad type bearings with extra wide inner rings. They have bores which fit standard machine bolts for economical mounting. There are a number of special seals and shields available for extreme duty use. Disc harrow bearings are shown on the upper sketch of Figure 31.

Hay rake tine bearings are standard Conrad type bearings with inch dimension bores to fit standard machine bolts for easy mounting to agriculture equipment. Special mounting studs can be shipped with the bearings. Heavy duty seals are also available. Hay rake tine bearings are shown as the second sketch on Figure 31.

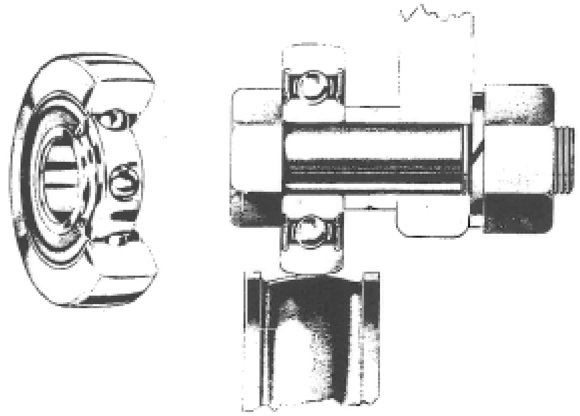
The third sketch on Figure 31 is a conveyor bearing. Conveyor bearings have heavy duty seals because of the contaminated conditions inside idler rolls where they are used. The bearings are lubed for life to avoid costly maintenance procedures. The bore fits standard round or hex shafts. The bearings can be supplied with special stub shafts with crowned teeth that can fit the hex bore of the bearing to compensate for misalignment.

Integral shaft ball bearings are an excellent design tool that is used in everything from table saws to lawn tractors. Inner pathways are ground directly on a hardened steel shaft permitting use of a larger shaft for increased strength and rigidity. The two ball rows are spread apart providing increased stability and resistance to moment loading. A variety of different seals can be used at each end to exclude contaminants and contain a large amount of lubricant that is placed between the spread apart ball rows. The shaft extensions can be furnished to various configurations for mounting a wide variety of mechanical components. (See Figure 32.)

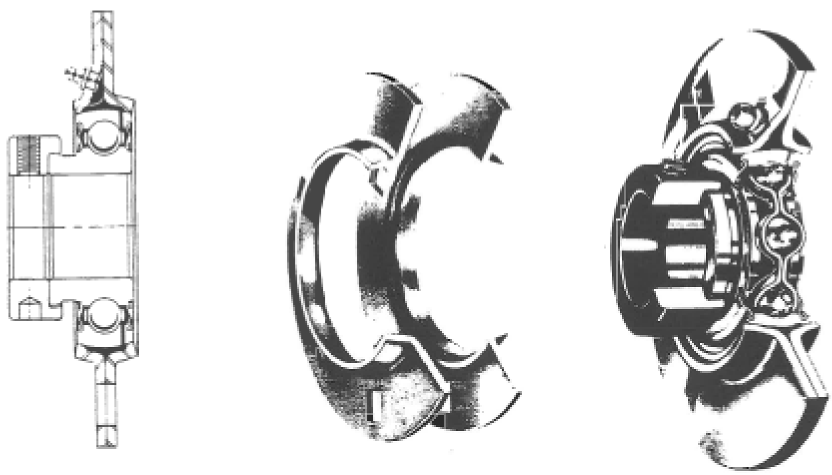
Millions of integral shaft ball bearings are being used as fan and waterpump bearings on automobile engines. The bearing is a press fit into a housing with the water impeller a press fit on the rear shaft extension and the engine cooling fan and accessory drive pulley a press fit on the front extension. Because of the very stringent demands placed on this bearing, state of the art seals and lubricants are being used. The top sketch of Figure 33 has a traditional integral shaft bearing in an automobile engine waterpump and the lower sketch has a more recent design utilizing a close coupled bearing with a stepped shaft.

A very high volume specialty ball bearing product now being used in the automotive industry is the integral spindle wheel bearing unit. It is comprised of two rows of balls riding directly on pathways on the spindle and in the hub. The units are assembled, grease lubricated, sealed, and tested on automatic equipment in the bearing plant. It fits the modular assembly concept well because it comes as a sealed for life package that bolts to the vehicle and the vehicle wheels bolt directly to it. There is a design for both drive and non-drive wheels. They are extensively used on front drive vehicles. (See Figure 34.)

Figure 30
Specialty Ball Bearing Product

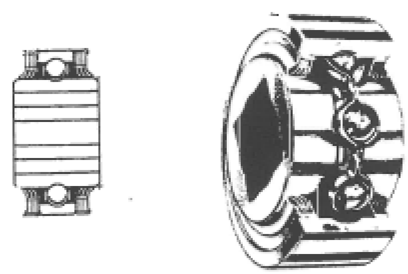


Cam Follower Bearing

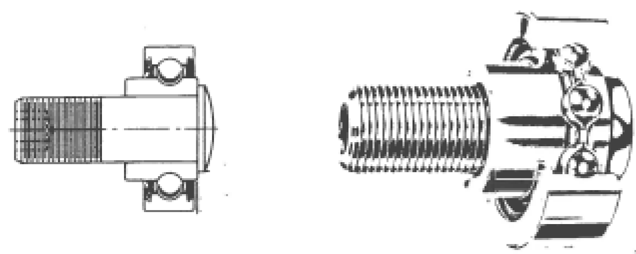


Adapter Bearing With Flange

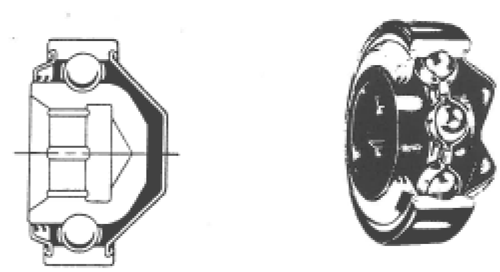
Figure 31
Specialty Ball Bearing Product



Disc Harrow Bearing

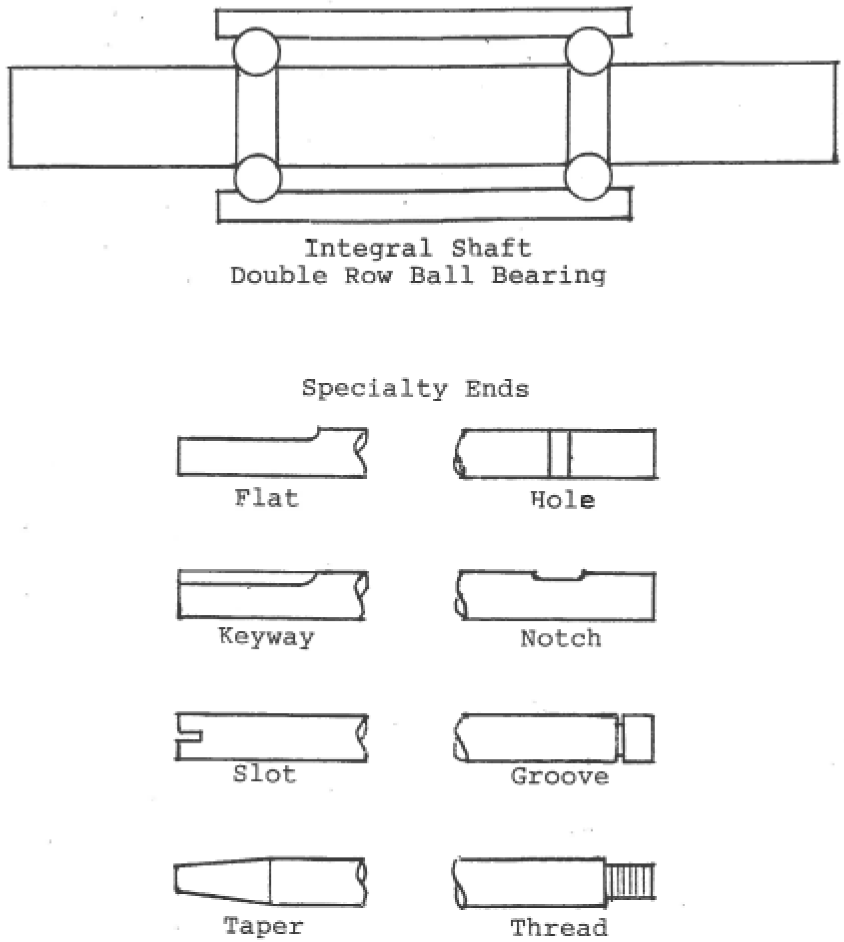


Hay Rake Tine Bar Bearing



Conveyor Bearing

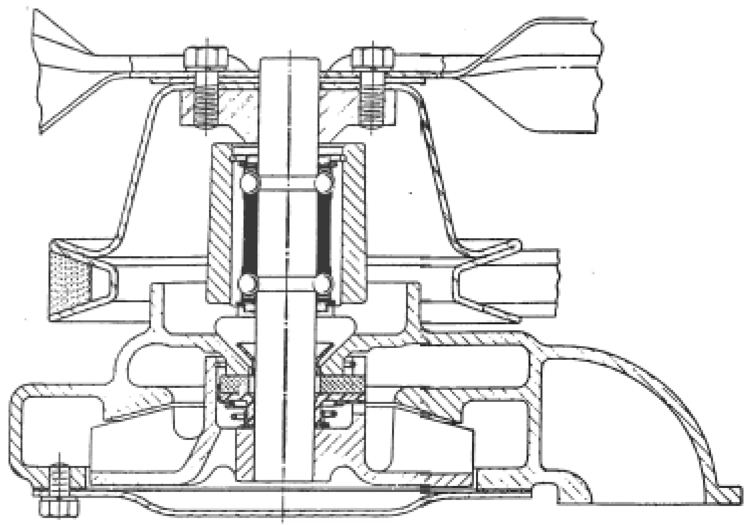
Figure 32
Specialty Ball Bearing Product



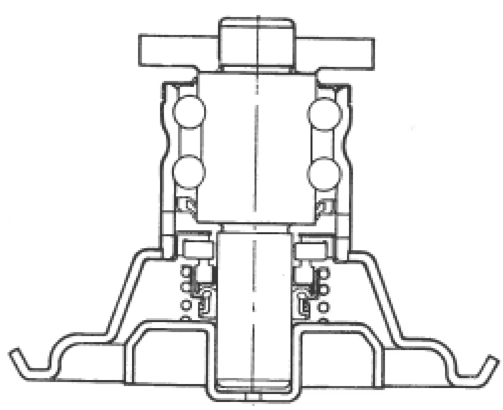
Applications

Bench Grinders	Fans & Blowers	Table Saws
Centrifugal Pumps	Lawn Tractors	Washing Machines
Fan & Waterpumps	Polishing Heads	Wheel Mounts

Figure 33
Specialty Ball Bearing Product

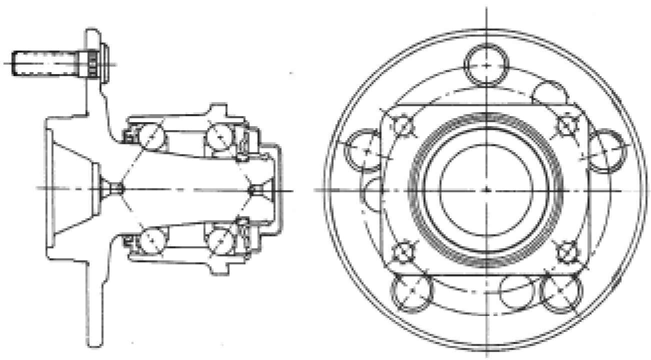


Traditional Automobile Waterpump
With Integral Shaft Ball Bearing

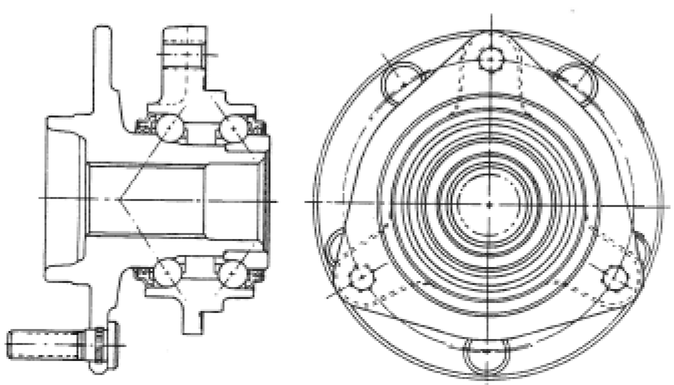


Later Version Waterpump
With Stepped Shaft Bearing

Figure 34
Specialty Ball Bearing Product



Integral Non-Drive Wheel Bearing



Integral Drive Wheel Bearing

Sample Problem

A shaft supported by two identical single row radial ball bearings has a gear at the center driving a tool at 2000 rpm. One bearing supports a 100 lb radial load while the other supports a 100 lb radial load and a 100 lb thrust load. The capacity of each bearing is 570 lb. The calculated life of each bearing using the previously mentioned equation is as follows:

$$L=3000(570/100)^{10/3}(500/2000) = 247{,}966 \text{ B10 hrs } (R=100\#,\ T=000\#)$$
$$L=3000(570/172)^{10/3}(500/2000) = 40{,}680 \text{ B10 hrs } (R=100\#,\ T=100\#)$$

The B10 life of the first bearing is very high. A smaller bearing could be used; however, it is common to use the same bearing at each end of a shaft for reasons of standardization and also to simplify shaft and housing design and manufacture. The B10 life of the second bearing represents 4.8 years of continuous operation.

Suppose the machine tool in the previous example experienced severe vibration and corrective action had to be taken. One approach in solving the problem is to preload the bearings. Preloading is accomplished by applying a thrust load to one bearing and having it reacted by the other on the same shaft. This has the effect of stiffening the bearings and making the shaft less prone to vibration. Assume that, after preloading, the thrust increases from 0 to 100 lb on the first bearing and from 100 to 200 lb on the second. The radial load remains the same. Using the existing radial bearings, the new calculated lives are as follows:

$$L=3000(570/172)^{10/3}(500/2000) = 40{,}480 \text{ B10 hr } (R=100\#,\ T=100\#)$$
$$L=3000(570/284)^{10/3}(500/2000) = 7{,}647 \text{ B10 hr } (R=100\#,\ T=200\#)$$

It can be seen that the lives of the two bearings have decreased significantly from the 247,966 and 40,680 B10 hr obtained before preloading. Now, let us repeat the above calculations using angular contact ball bearings with the same basic overall dimensions:

$$L=3000(475/103)^{10/3}(500/2000) = 122{,}376 \text{ B10 hr } (R=100\#,\ T=100\#)$$
$$L=3000(475/169)^{10/3}(500/2000) = 23{,}493 \text{ B10 hr } (R=100\#,\ T=200\#)$$

The use of angular contact bearings, as seen from the above calculations, has increased predicted life by more than threefold. This exercise has shown how angular contact bearings can improve design life when higher thrust loads are present.

Another possible solution to the tool vibration problem would be to use the original radial ball bearing in the first position and replace the radial bearing in the second position with a double row ball bearing. The double row ball bearing would be the same size as the radial ball bearing but would be 46% wider. The additional width could be accommodated by a minor rework to the shaft and housing. The double row ball bearing would be internally divergent which would add rigidity to the shaft to prevent vibration. It would also eliminate the need for preloading the two bearings as was investigated in the previous analysis. The life of the bearing in position 1 would remain at a very high level. The life of the double row ball bearing in position 2 would be as follows:

$$L=3000(800/208)^{10/3}(500/2000) = 66{,}828 \text{ B10 hr } (R=100\#, T=100\#)$$

The life of the double roll ball bearing in position 2 is almost a threefold increase over the angular contact bearing. In summary, the double row ball bearing, because of its inherent design, stiffened the shaft and increased the life of the bearing in position 2. The design no longer needs special devices to preload the two bearings which can be a more difficult task than fitting in a slightly wider bearing in the position.