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# Ball Bearing Fundamentals

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# **Ball Bearing Fundamentals**

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## Introduction

One of the biggest challenges facing a ball bearing Application Engineer is to determine the forces (loads) that are acting on his product. The loads on ball bearings are either radial or thrust. The sketch at the top of Figure 1 shows that radial loads act perpendicular to the bearing axis of rotation while thrust loads act parallel to the axis of rotation. In some applications, there are two radial loads acting 90 degrees apart as shown on the second sketch of Figure 1. The Pythagorean Theorem is then used to calculate the resultant radial load.

In most applications, there are two bearings supporting a rotating shaft on a piece of mechanical equipment. The third sketch on Figure 1 shows an applied load straddle mounted between two shaft supporting bearings. The radial loads on bearings I and II are calculated as follows:

$$L_I = \text{Load } (b) / (a+b) \quad L_{II} = \text{Load } (a) / (a+b)$$

L is the radial load on bearings I and II. a and b are bearing locating dimensions shown on the third sketch of Figure 1. It can be seen that because the load is closer to bearing II, it supports the greater portion of the load. The fourth sketch on Figure 1 shows an overhung applied load acting on a shaft supported by two bearings. Overhung loads put a heavy force on the adjacent bearing. The following equations are used to calculate the radial load on bearings III and IV:

$$L_{III} = \text{Load } (d / c) \quad L_{IV} = \text{Load } (c+d) / c$$

It can be seen that the load on adjacent bearing IV is greater than the applied load itself. The loads acting on a bearing in pounds and its speed of rotation (rpm) are used to calculate bearing B10 life. Bearing B10 life predicts how many hours of operation 90% of the bearings will endure. The equation follows:

$$L_{B10} = 3000 (C/P)^{10/3} (500 / S)$$

$L_{B10}$  is the bearing B10 life. C is the bearing capacity in pounds found in industry catalogs. P is the equivalent radial load in pounds which takes into account both radial and thrust loads also found in industry catalogs. S is the speed in revolutions per minute (rpm). Should a bearing operate under a number of different loads and speeds, the following equation is used to calculate B10 life:

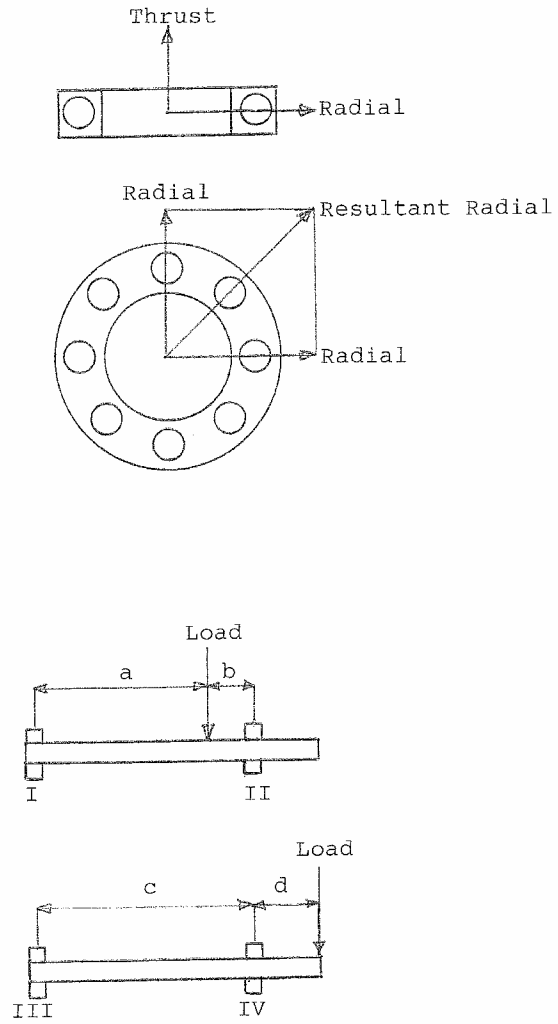
$$L_{B10} = 1 / [(t_1 / L_1) + (t_2 / L_2) + (t_3 / L_3) + \text{etc.}]$$

$L_{B10}$  is the bearing B10 life in hours.  $t$  is the percent 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 life:

<u>% Survival</u>	<u>B - Life</u>	<u>% of B10 Life</u>
99	B-1	21
98	B-2	33
95	B-5	62
90	B-10	100
50	B-50	400
40	B-60	500

The following will be examples of how to calculate bearing loads for three commonly used gear drives for various mechanical devices using the information given above. The loads, and the speed equations which will also be given, can then be used to calculate the life of the bearing in the application.

**Figure 1**  
**Bearing Loads**



## Spur Gear Bearing Loads

Spur gears have straight teeth that are parallel to the axis of rotation (as opposed to helical gear teeth that are at an angle to the axis of rotation). Spur gear teeth are shaped in the form of an involute curve which enables them to operate at efficiencies of well over 90%. They are used to transmit torque between parallel shafts and impose only radial loads on support bearings. Figure 2 has two spur gears in mesh. In order to calculate bearing loads, the input torque  $Q$  which is usually produced by a motor or an engine is calculated as follows:

$$Q = (\text{HP}) 63025 / N$$

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

$$P = Q / r$$

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

$$S = (P) \tan a$$

$S$  is the separating force in pounds.  $P$  is calculated above.  $a$  is the tooth pressure angle in degrees. The total radial load on bearing I, due to the tangential force  $P$  and the separating force  $S$  which act 90 degrees apart and can be seen on Figure 2, is as follows:

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

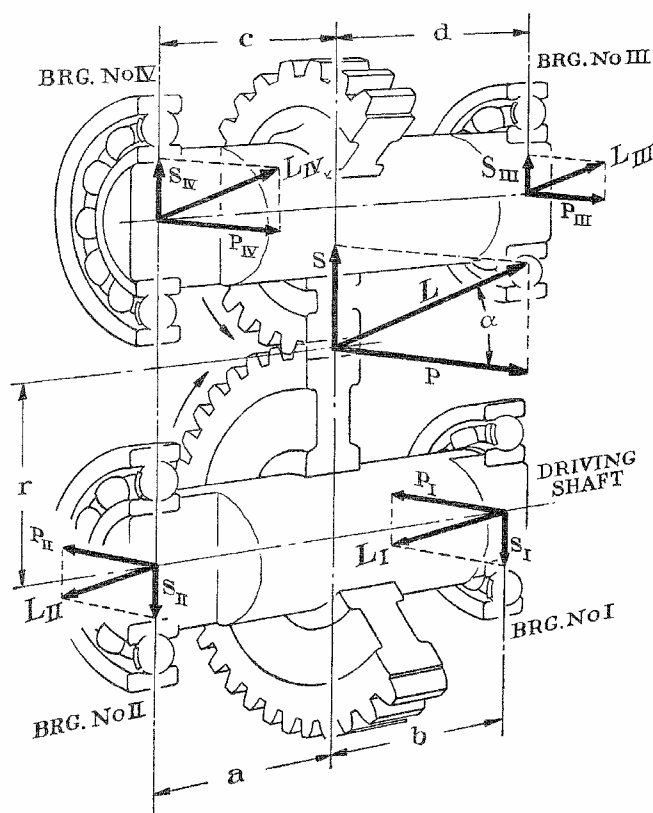
$L_I$  is the total radial load on bearing I in pounds.  $P$  is the tangential force calculated above.  $a$  and  $b$  are bearing locating distances shown on Figure 2.  $S$  is the separating force calculated above in pounds. Since the radial load due to the tangential force  $P$  and the radial load due to the separating force  $S$  act 90 degrees to each other, 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 forces, is as follows:

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

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 above, 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 ratio of the number of teeth in the driving gear to the number of teeth in the driven gear. Enough information is now available to calculate ball bearing life which the Application Engineer may have to do a number of times before the bearings that meet all design objectives are found.



**Figure 2**  
**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 as they are in spur gears. 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 3. Since the thrust force is applied at a distance from the gear center, it not only produces a straight thrust load on support bearings I and IV, but it also produces a moment load that acts on all four bearings as a radial load. Figure 3 shows the thrust force at the mesh as vector T and the 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 torque input 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:

$$T = P \tan y$$

T is the thrust load in pounds. P is the tangential force calculated the same way as for spur gears. y is the helical gear helix angle in degrees. (One commonly used helix angle is 15 degrees). The radial loads on bearing I due to the tangential force P, separating force S, and thrust force couple U are as follows:

$$P_I = P a / (a + b) \quad S_I = S a / (a + b) \quad U_I = T r / (a + b)$$

$P_I$ ,  $S_I$ , and  $U_I$  are radial loads on bearing I in pounds as shown on Figure 3. a and b are support bearing I locating dimensions in inches. r is the pitch radius of the driving gear in inches. The total radial load on bearing I is calculated as follows:

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

The radial loads on bearing II due to the tangential force P, separating force S, and thrust couple U are as calculated as follows:

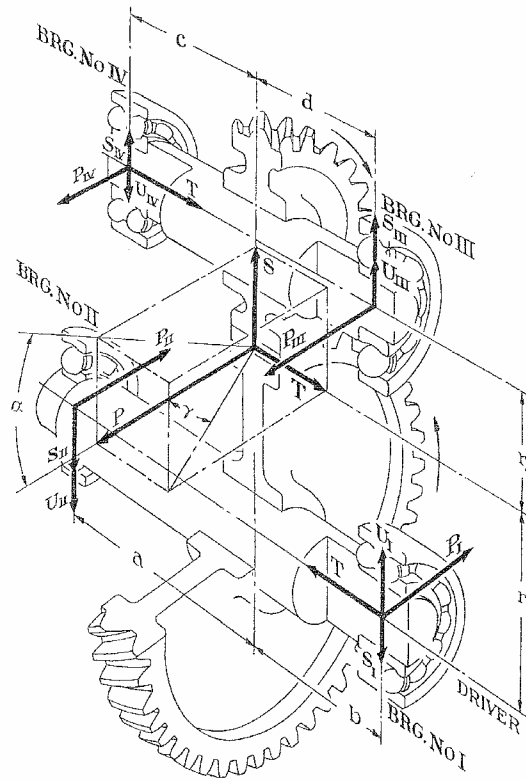
$$P_{II} = P b / (a + b) \quad S_{II} = S b / (a + b) \quad U_{II} = T r / (a + b)$$

The total radial load on bearing II is as follows:

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

The loads on bearings III and IV are calculated in a similar manner. The rpm of the driven gear = the rpm of the driving gear x (the number of teeth in the driving gear / the number of teeth in the driven gear). Now that the total radial load and speed of both bearings and the thrust load on bearing I are known, the life can be calculated according to the equation previously given.

**Figure 3**  
**Helical Gear Bearing Loads**



## Bevel Gear Bearing Load

Bevel gears have the normal force due to the driving gear tooth contact (E) divided into three components: tangential force P, thrust force  $T_G$  on the larger gear, and thrust force  $T_P$  on the smaller gear (pinion). See Figure 4. The bevel gears shown have the applied load overhung on the shafts as opposed to the spur gears that had the applied load straddle mounted on the shaft. The tangential force is calculated as follows:

$$P = Q / r$$

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

$$T_G = P \tan a \cos b$$

$T_G$  is the gear thrust in pounds. a is the tooth pressure angle in degrees, b is one-half the pinion pitch cone angle in degrees. The pinion thrust is calculated as follows:

$$T_P = P \tan a \sin b$$

The radial loads on bearing I due to the tangential force P, the gear thrust  $T_G$ , and pinion thrust  $T_P$  are as follows;

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

$P_I$ ,  $T_{GI}$ , and  $U_I$  are the tangential force, gear thrust, and pinion thrust calculated above. a and b are bearing locating distances shown on Figure 4. r is the mean pinion pitch radius. The total radial load on bearing I is as follows:

$$L_I = [(P_I + (T_{GI} - U_I))^2]^{1/2}$$

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

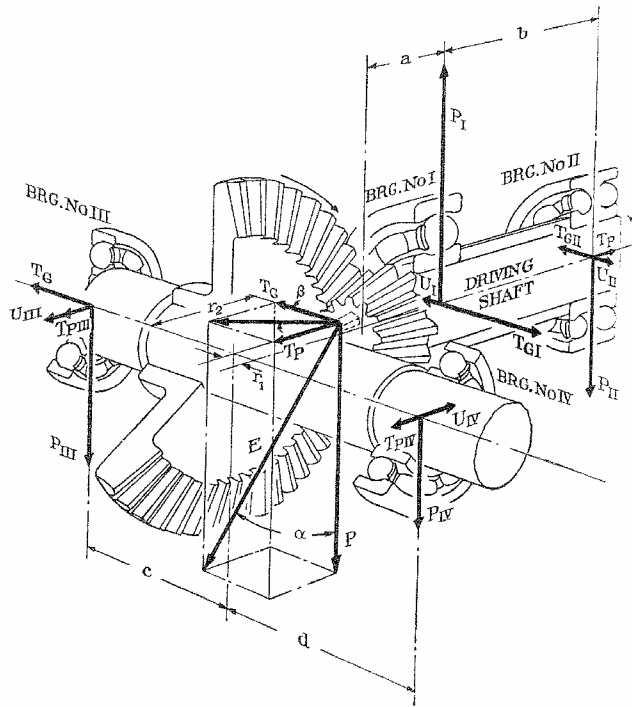
$$P_{II} = P (a / b) \quad T_{GII} = T_G (a / b) \quad U_{II} = T_P (r / b)$$

The total radial load on bearing II is as follows:

$$L_{II} = [(P_{II} + (T_{GII} - U_{II}) 2)]^{1/2}$$

There is a thrust on bearing II which is the pinion thrust calculated above. Bearings III and IV are calculated in a similar manner. The rpm of the gear = the rpm of the pinion x (the number of teeth in the pinion / the number of teeth in the gear).

**Figure 4**  
**Bevel Gear Bearing Loads**



## Selection

The selection of a ball bearing for an application involves the three S's: series, sort, and size:

1) Series: The bearing must have the height-to-width cross section that best fits the space provided. Ball bearings are manufactured in predominately three different cross-sectional configurations: extra-light, light, and medium. Extra-light and light are used when loads are light to moderate and housing and shaft space require the smallest bearing available. The medium series provides a capacity increase of 30% to 40% over the light series but requires more space on the shaft and in the housing. See Figure 5.

2) Sort: The bearing must be of the sort that best supports the nature of the load that the application provides whether it is radial, thrust, or a combination of both. The three basic types available are single row radial, angular contact, and double row. Single row radial are the most common types and, as the name implies, are designed to support predominately radial loads although accommodation of thrust loads of up to 60% of radial load is not uncommon. See Figure 6.

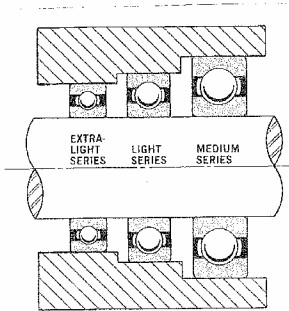
Angular contact ball bearings are designed to support a higher percentage of one direction thrust load than radial ball bearings. This is accomplished by designing the bearing to have a line of contact between the balls and rings to be at an angle to the vertical (line of contact for radial ball bearings). This angle is the contact angle and can be 15, 25, or 35 degrees. Angular contact bearings have one outer ring or one outer ring and one inner ring shoulder removed which allows a full complement of balls to be assembled for added capacity. See Figure 7. Angular contact bearings are sometimes mounted in pairs. See Figure 8. "Back-to-back" mounting provides high resistance to moment or overhung loading. "Face-to-face" mounting allows for some shaft misalignment. Tandem (face-to-back) mounting supports high one-direction thrust load. Double row ball bearings have the same contact line patterns as pairs of angular contact bearings and therefore have the same characteristics of angular contact bearings (as will be shown later), but fit in a narrower space.

3) Size: The bearing must be large enough to have the capacity to support the load imposed on it so that the required service life can be achieved without premature failure. Bearings are manufactured in a wide variety of sizes in each of the series

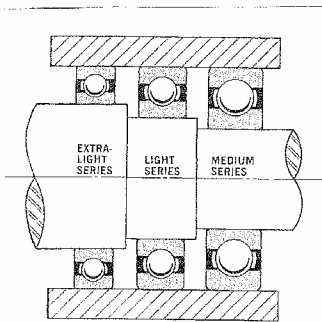


mentioned above. Standard internal diameters range from 10 mm (.3937 inches) to 110 mm (4.3307 inches) in each of the series. Corresponding external diameters range from 26 mm (1.0236 inches) to 240 mm (9.4488 inches). Capacities for the same sizes range from 230 pounds to 13,800 pounds. The capacity, as mentioned, is the load that the 90% of the bearings will support at 500 revolutions per minute (rpm) for 3,000 hours without failure.

**Figure 5**  
**Ball Bearing Series**

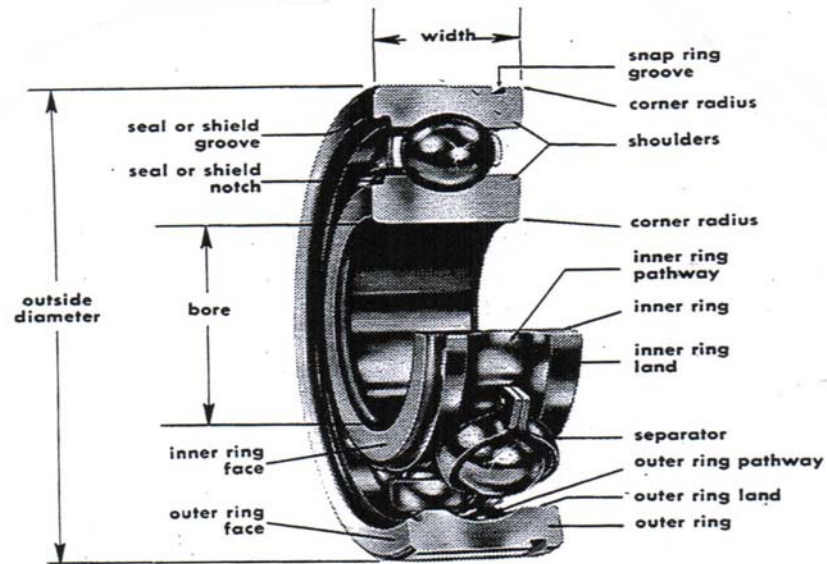


Relative Proportions of Bearings  
With Same Inside Diameter



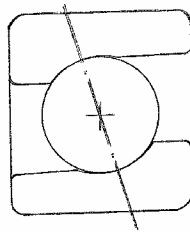
Relative Proportions of Bearings  
With Same Outside Diameter

**Figure 6**  
**Radial Ball Bearing**

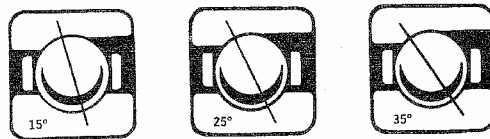


**Ball/Pathway Contact**

**Figure 7**  
**Angular Contact Ball Bearing**

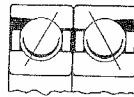


Angular Contact Bearing  
Cross Section

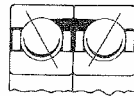


Angular Contact Bearing Types

**Figure 8**  
**Angular Contact Bearing Pairs**



Back-To-Back Mounting  
Internally Divergent



Face-To-Face Mounting  
Internally Convergent



Face-To-Back Mounting  
Tandem Pairs

## **Bearing Mounting**

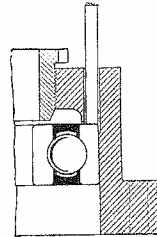
The bearing mounting surfaces on shafts and in housings should be machined as accurately as the bearing rings themselves. If not, mounted bearing rings can take the shape of inaccurately machined seats and fail from excessive stress levels. Housing and shaft wall thicknesses should be heavy enough to be able to support bearing press fit rings without undue stress or deflection which can cause the rings to lose their press fit. Corner radii on shafts and in housings must be machined smaller than corner radii on bearing rings to provide for proper seating. The Society of Automobile Engineers has provided standards for corner radii.

Shafts should be designed to be strong enough to resist excessive bending. Excessive bending causes high bearing misalignment and premature failure. Designs should not incorporate over two bearings on one shaft. Over two bearings on one shaft can cause extremely high load and failure of any one bearing. Care should be taken not to mount two double row bearings with internally divergent contact lines on one shaft. Two such bearings mounted on one shaft will fail from "fighting eccentricities" because of resistance to the slightest shaft irregularity.

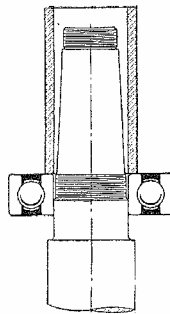
Bearings are usually mounted with the rotating ring a press fit and the non-rotating ring a close or push fit on its seating surface. The rotating ring press fit must be heavy enough to prevent turning on its seat during operation. The non-rotating ring with its close or push fit experiences slight rotational movement called "creep" which allows ball forces to be distributed over 360 degrees of the pathway. Care should be taken not to make the non-rotating ring fit too loose or a phenomenon called "journaling" takes place causing excessive seat wear and eventual failure. Also, one ring of a bearing assembly being a push fit greatly aids assembly. Five different classes of shaft and housing fits have been established by the American Annular Bearing Engineers Committee (ABEC). The classes of fit range from ABEC 1 which is the standard used by most applications, to ABEC 9 which is for high precision mounting. The use of non-ferrous material (aluminum) requires special mounting fits because of differences in thermal expansion and material properties.

Bearings should be installed and removed by applying pressure to the press fit ring only. Pressing on the other ring puts excessive loading on the balls brinelling (indenting) the pathways which causes noisy operation. See Figure 9.

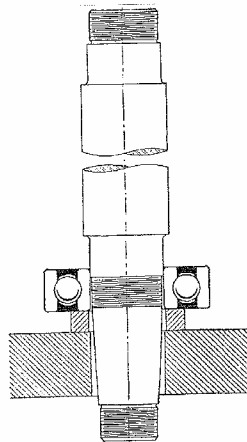
**Figure 9**  
**Bearing Removal and Assembly**



Bearing Removal



Bearing Assembly



Bearing Assembly

## **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 10. Most bearings are built to radial play. The definition of radial play is the outer ring pathway diameter minus the inner ring pathway diameter minus twice the ball diameter. This is done in production by gaging the inner and outer ring pathway diameters and selecting a class of balls with a diameter that will result in the specified radial play. The purpose of radial play is as follows:

- 1) It permits interference fits with inner rings on shafts and outer rings in housings. Interference fits on shafts or in housings causes the bearing inner ring pathway diameter to expand in the case of press fit inner rings and the outer ring pathway to contract in the case of press fit outer rings. The amount of the expansion or contraction is 80% of the press fit. Having no radial play would cause the bearing to become internally preloaded which is not normally a good condition for radial ball bearings to be in for optimum performance.
- 2) It allows 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 preloading the bearing if it had no internal clearance. Standard ball bearing life equations do not account for radial preloading of ball bearings.
- 3) Radial play allows the inner ring to misalign slightly with the outer ring without preloading the bearing thus accommodating shaft and housing manufacturing tolerances and shaft deflections under load.

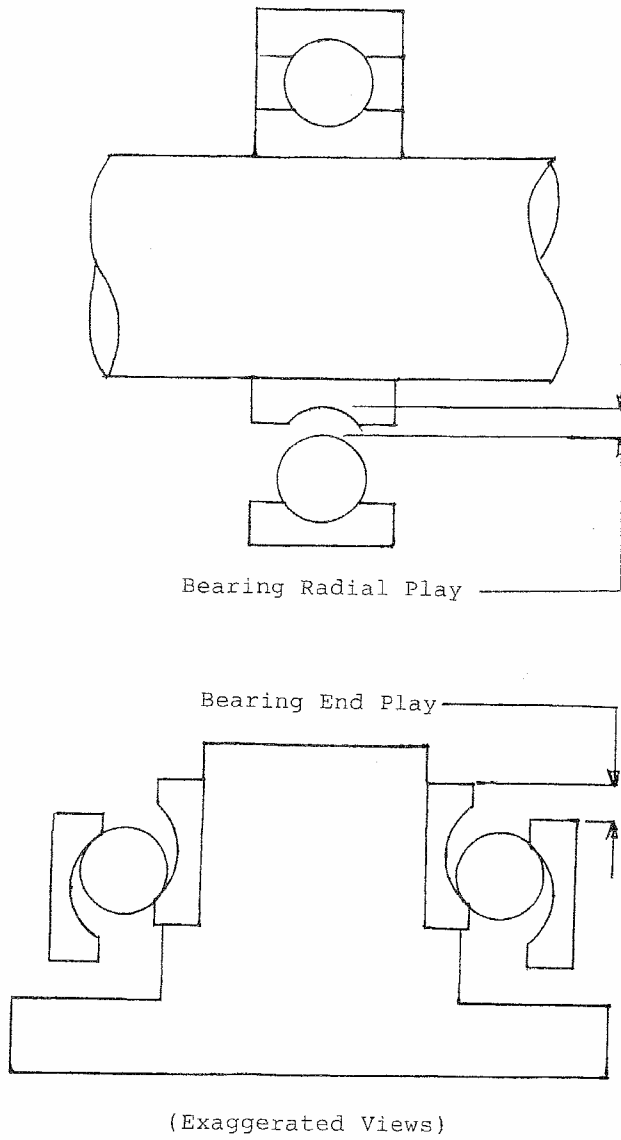
The average standard radial play for a light series 40mm bore radial bearing is .00085 inch. The average ABEC 1 press fit for the same bearing rotating inner ring is .0004 inch. Eighty percent of .0004 is .00032 inch. Subtracting .00032 from .00085 yields .00053 inch which is the average running radial play for a light series 40 mm bore bearing.

Standard radial play satisfies the requirements of most applications. For unusual applications, special radial play may be required. Less than standard radial play (.0002 inch average) may be required for bearings operating at low to moderate



speeds where accurate radial and axial locations are critical. Higher than standard radial play (.00145 inch 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 10**  
**Internal Clearance**



## **Bearing Retainers**

A commonly used method of retaining bearings is shown at the top of Figure 11. Here a lock washer mounted between the nut and bearing is keyed to the shaft to prevent rotation. The outside diameter of the lock washer has tabs which are designed so that one of them 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 the nut is torqued tight or when the nut is loose allowing some specified bearing end play.

In some circumstances, there is not sufficient 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 two screws which are threaded into the end of the shaft and lock wired to prevent loosening as shown in middle of Figure 11.

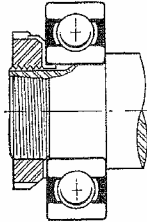
The simplest method is shown at the bottom of Figure 11 where a snap ring is inserted into a groove on the shaft. Some snap rings are made tapered so that, after installation, pressure is always put on the face of the bearing inner ring to prevent movement.

A method of clamping outer rings is shown at the top of Figure 12. 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.

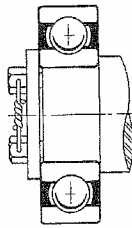
The middle of Figure 12 has a two-piece clamp that provides the shoulder for bearing outer ring retainment. This design allows for through boring the housing for better bearing alignment.

The bottom of Figure 12 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 snap wire. This design is not recommended where high thrust loads are present.

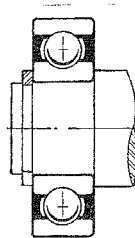
## Figure 11 Bearing Clamping



Slotted Nut and Tabbed Washer

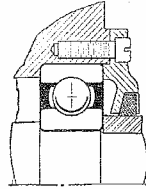


Thick Washer and Lock Wire

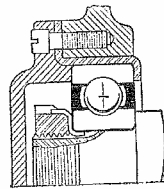


Snap Ring

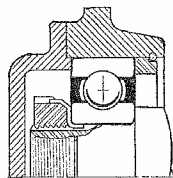
## Figure 12 Bearing Clamping



Piloted Outer Ring With Clamp  
With Seal



Two Piece Clamp  
Thru-Bored Housing



Wire Ring Retainer  
Thru-Bored Housing

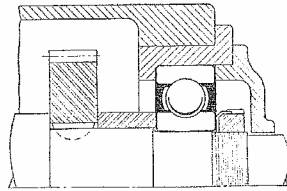
## **Bearing Sleeves**

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 13) and an adapter sleeve is used. The sleeve is a press fit in the housing and the bearing installed inside the sleeve. When establishing the fit of the bearing in the sleeve, the reduction of the internal diameter of the sleeve due to the press fit in the housing must be considered. For precision bearing mounting, the internal and external diameters of the sleeve must be concentric and parallel.

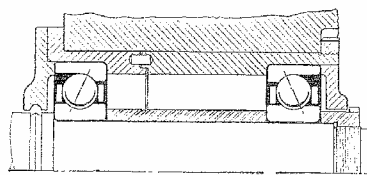
Two bearing outer ring sleeves, an inner ring sleeve, and clamping arrangement are used to ease assembly and to improve the runout and rigidity of the spindle shown in the middle of Figure 13. To ease assembly, a sub-assembly is made of the spindle and attaching components and installed into the housing as a unit. During the sub-assembly of the spindle and attaching components, the bearing outer rings high point of eccentricity are aligned diametrically opposite the high points of eccentricity of the sleeves and then press fitted to improve spindle run out. Notice that the two sleeves are pinned together to avoid relative rotation. The spindle nut clamps together the bearing inner rings and the spindle sleeve. The spindle sub-assembly is secured in the housing with the housing nut. The housing nut is torqued to preload the right hand bearing which, in turn, preloads the left hand bearing through the clamped inner rings putting the spindle in tension. The spindle now operates with improved run out and rigidity.

It is sometimes necessary to mount ball bearings to allow a certain amount of eccentricity of the shaft with respect to the housings 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 lower sketch on Figure 13. This should not be done with a two-piece housing with separate sleeves because very careful adjustment of one sleeve with respect to the other would have to be made in order to prevent misalignment of the shaft.

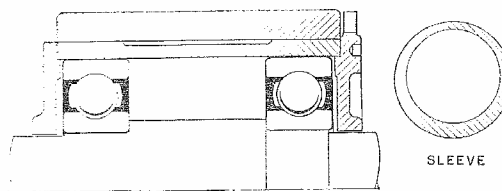
**Figure 13**  
**Bearing Sleeves**



Outer Ring Sleeve  
Allows Gear Assembly



Precision Sleeve  
Improves Spindle Runout



Eccentric Sleeve  
For Belt Tightening

## **Ball 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. Good EHD films have been found to provide better than calculated bearing lives while poor EHD films have been found to result in less than calculated lives. EHD films have been found to be dependent on the type and bearing size, the type and viscosity of the oil used, the bearing speed, and the bearing load. A number is calculated for each factor based on test results and the numbers multiplied. The resulting number called the "lubrication factor" is located on a graph where it can be determined whether the bearing is operating in a "region of marginal lubrication" or a "region of good lubrication". See Figure 14. Because of the steepness of the curve on Figure 14, it can be seen that a small change in the lubrication factor can produce a large change in the oil film thickness.

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 at 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 wide 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. The oil is then splashed throughout the gearbox during operation. 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. This system is used in electric motors.

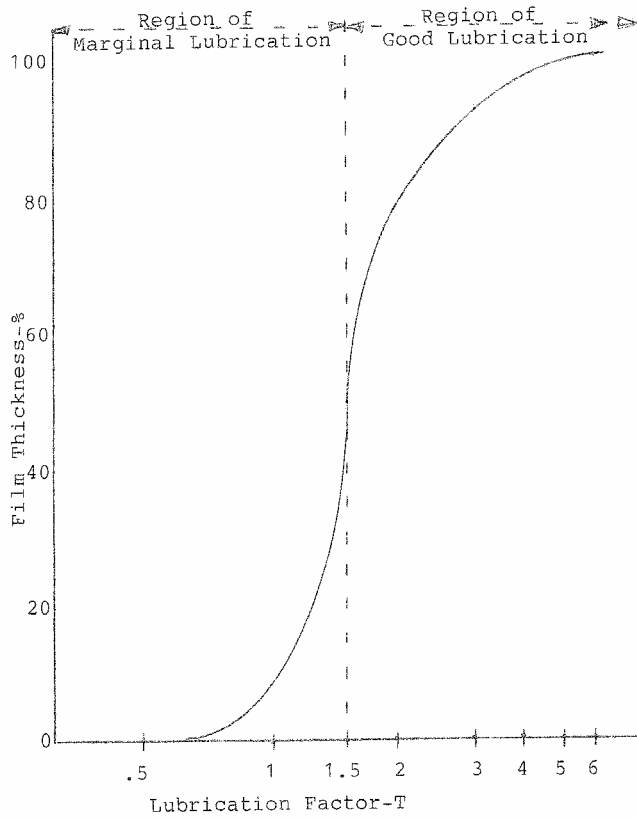
A simple method of lubricating bearings is by using grease. A carefully measured quantity of grease is evenly distributed throughout the bearing where it is contained by seals or shields. This configuration can run for the life of the bearing. Grease consistency is important. Greases too soft will cause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



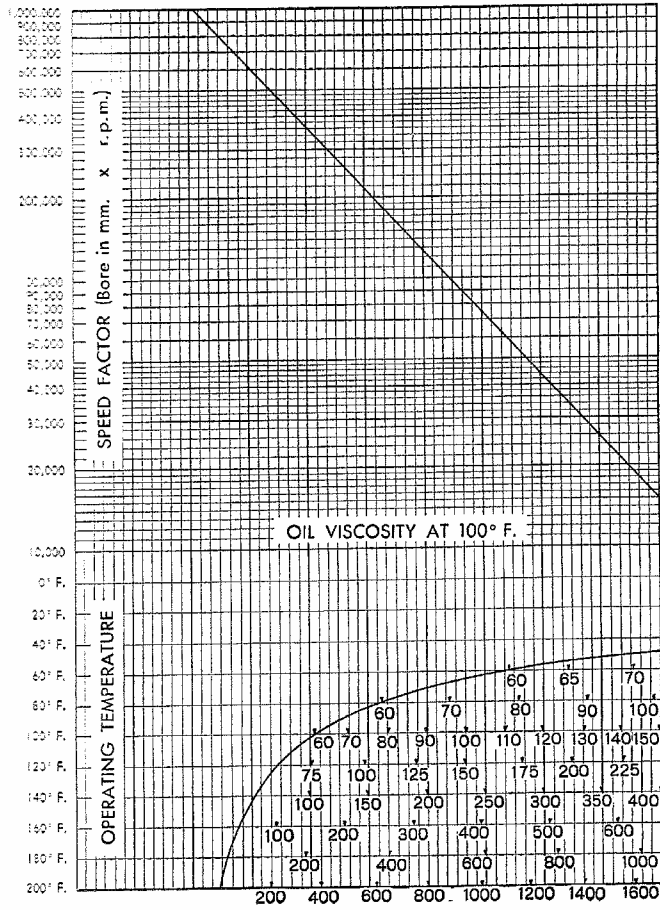
- 1) Mineral oil grease for general purpose operation from -30 degrees F to +300 degrees F.
- 2) Ester based greases operate from -100 degrees F to +350 degrees F.
- 3) Silicone greases operate from -100 degrees F to +450 degrees F but lack good load carrying characteristics.

Figure 15 is a chart which can be used to determine the proper oil viscosity for various size bearings running at various speeds. First multiply the bearing bore (inside diameter) by the rpm. Locate the number on the upper left hand scale of the chart. Draw a horizontal line to the diagonal line (upper right). At the intersection, draw a vertical line down to the horizontal line that represents the operating temperature of the bearing. Read the oil viscosity at this intersection.

**Figure 14**  
**Elastohydrodynamic Lubrication**



**Figure 15**  
**Bearing Oil Viscosity Chart**



## **Ball Bearing Closures**

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

At the top of Figure 16 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 of 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 an alternate 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 16 has rubber molded around a flat steel ring insert which imparts rigidity and strength to the construction and helps to control sealing lip pressure which is needed to accommodate small movements of the inner ring. 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 degrees F to 225 degrees F. There are other similar 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 10 mm bore bearing.

Another version of the single lip seal is shown as the second sketch on Figure 16. 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 objects 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 for the standard snap-in single lip seal design.

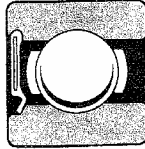
The third sketch on Figure 16 is of a triple lip seal with the outer steel shield protection the same as was discussed on the previous single lip seal. The seal is called "land riding" because the three lips ride on the inner ring outer diameter rather than on a notch as the previous two seals did. Besides having triple lips,

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 the torque level of the bearing; consequently, speeds are limited to 30 rpm for the large size bearing mentioned above to 2500 rpm for the small size bearing. It is commonly used on farm machinery, construction equipment, and automotive in-line engine water pump bearings.

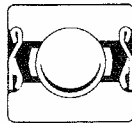
The fourth sketch on Figure 16 is a felt seal. It is held between two steel pieces which are crimped into a notch in the outer ring. Felt element seals are good for lubricant retention and light particle entrance and their low friction allows for higher speeds of operation than all other seal designs. Limiting speed is 3000 rpm for the large size bearing and 19,000 rpm for the small bearing. Limiting temperature is 275 degrees F which is the charring temperature of the felt element.

The bottom closure on Figure 16 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 have a limiting speed other than what the bearing has. It is used to contain grease or control the amount of oil flowing into the bearing when exposed to an oil sump. Excessive oil in a bearing can cause an unusually high running torque and premature failure.

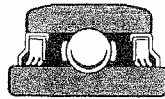
**Figure 16**  
**Ball Bearing Closures**



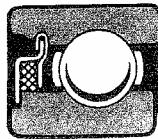
Standard Rubber Lip Seal



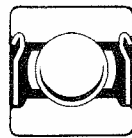
Lip Seal With Outer Guard



Triple Lip Seal With Guard



Felt Seal



Double Shields

## **Radial Ball Bearing Application**

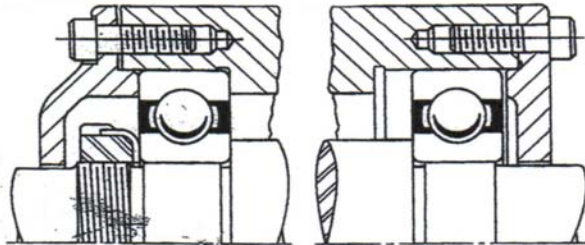
Radial ball bearings support high radial loads as well as thrust loads from either direction. Figure 17 upper left has a sketch of a radial bearing fixed mounted on the shaft and clamped on the outer ring. The small clearance between the end cap and housing assures that the bearing outer ring remains clamped. This bearing will support radial as well as thrust loads in either direction if the bearing on the other end of the shaft is allowed to float (move axially).

Figure 17 upper right shows the bearing on the other end of the shaft shown on the upper left. The bearing is allowed to move in either direction axially to account for differential thermal expansion between the shaft and housing. The shaft axial movement is restricted to the end play of the bearing at the left end of the shaft. On this installation, the end cap is seated tightly against the housing to act as a positive stop.

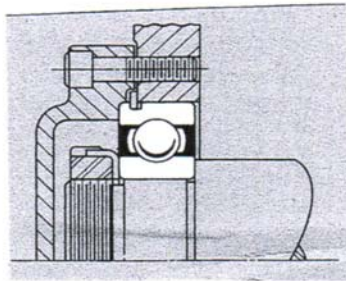
The radial bearing shown in the middle of Figure 17 is mounted to allow through boring of the housing. Through boring of housings results in accurate alignment of bearings at both ends of a shaft. The bearing is located axially by the use of a snap ring between the outer ring and housing. Notice that there is clearance between the end cap and housing and that the bearing is supported axially by pressure exerted by the inner shoulder of the end cap through the bearing outer ring through the snap ring to the housing. Snap rings support less two-directional thrust than if the bearing were to be clamped tight by more positive means.

When radial and thrust loads are moderate, both bearings may be allowed to float as seen on the lower sketch on Figure 17. There must be enough axial movement to allow for shaft and housing tolerances and differential thermal expansion. If this was a precision mounting, the bearing inner and outer ring high points of eccentricity would be mounted diametrically opposite the high points of eccentricity for the shaft and sleeve to minimize shaft runout. The same procedure holds true for the sleeve mounting in the housing if the sleeve is a press fit in the housing. If the sleeve is a press fit in the housing, the contraction of the inner diameter due to the press fit in the housing must be taken into account when calculating the press fit of the bearing outer ring in the sleeve.

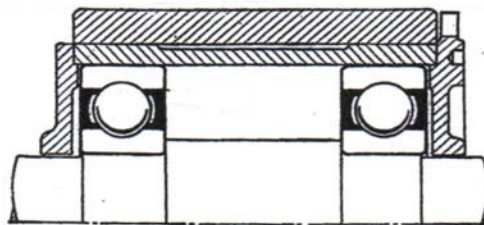
**Figure 17**  
**Radial Ball Bearing Application**



Fixed and Floating Radial Ball Bearings



Snap Ring Allows Thru-Bore Housing



With Light Loads Both Bearings Can Float



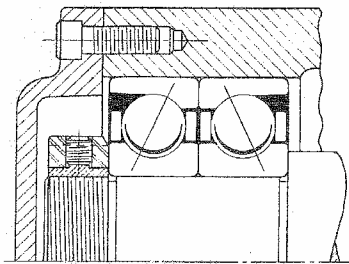
### **Angular Contact Ball Bearing Application**

At the top of Figure 18 is a pair of angular contact bearings mounted back-to-back. By clamping the inner rings together, the bearings undergo a predetermined preload which sets the contact angle lines at an internally divergent direction. This pattern gives the bearings maximum resistance to misalignment and/or moment loading. The angular rigidity of this type of mounting necessitates accurate machining of the bearing and shaft seats. This bearing set is mounted to float in the housing to accommodate shaft and housing tolerance buildup and differential thermal expansion between the shaft and housing.

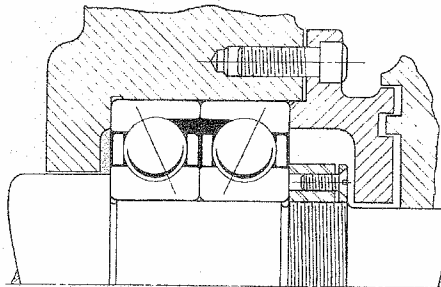
At the middle of Figure 18 is another pair of angular contact bearings mounted face-to-face. By clamping the outer rings together, the contact angle line aligns in an internally convergent pattern. This alignment allows the bearing to operate under more misalignment than the back-to-back mounting. Because the outer rings have to be clamped, this pair cannot be made to float in the housing using ordinary means.

At the bottom of Figure 18 are two angular contact bearings mounted in tandem. This arrangement accommodates radial load plus high one-direction thrust load. The thrust capacity of the pair, if accurately mounted, is 1.62 times the thrust capacity of one of the bearings. The tandem pair may or may not be preloaded against another angular contact bearing at the other end of the shaft.

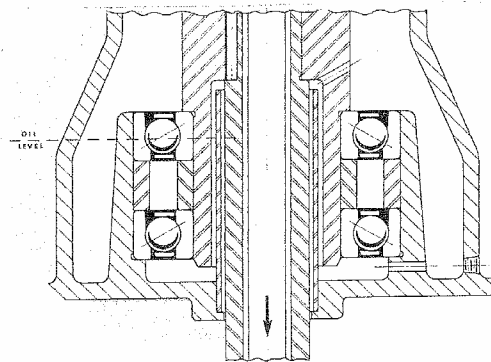
**Figure 18**  
**Angular Contact Ball Bearings**



Back-To-Back Mounting



Face-To-Face Mounting



Tandem Pairs

### **Double Row Ball Bearing Application**

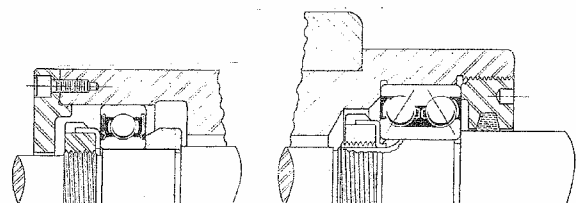
At the top of Figure 19 is a double row bearing with internally divergent contact angle lines similar to a back-to-back pair of angular contact bearings. This mounting is used because of the presence of reversing thrust loads. The bearing is clamped tight on the shaft and in the housing to maintain shaft position. An important aspect of this arrangement is that the heaviest thrust should be directed to the shaft shoulder and not to the thread which is the weaker element. A single radial bearing is at the other end and is allowed to float axially.

The second sketch on Figure 19 shows that through bored housings can be used for the fixed bearing position when a snap ring is used. The outer ring is clamped to resist reversing thrust loads. Clearance is left between the end cap and housing to ensure positive clamping pressure on the back side of the snap ring.

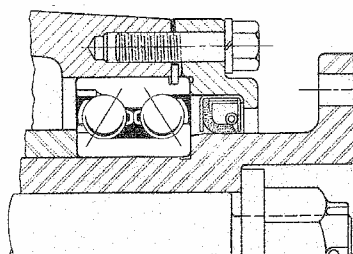
The third sketch on Figure 19 shows a double row bearing with face-to-face type mounting for rigid support of radial loads only. It can be made to float in the housing. This pair was also used because of slight misalignment of the shaft.

The fourth sketch on Figure 19 shows a double row bearing with back-to-back type mounting with internally divergent contact angle load lines. This type of bearing offers radial and axial rigidity and resistance to overturning moments making it good for a single bearing mounting such as the idler gear shown.

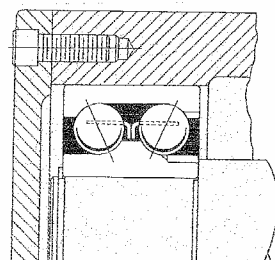
**Figure 19**  
**Double Row Ball Bearings**



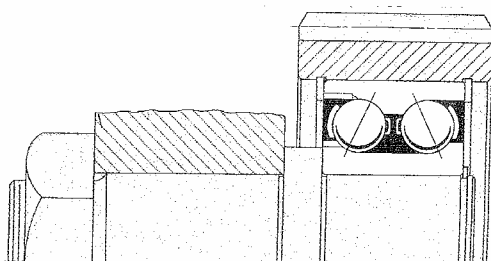
Double Row Internally Divergent



Snap Ring With Thru-Bored Housing



Double Row Internally Convergent



Single Bearing Mount