ENGINEERING

The following topics are covered within this section:

This engineering section is not intended to be comprehensive, but does serve as a useful guide in bearing selection.

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BEARING SELECTION PROCESS

TIMKEN® SUPER PRECISION BEARINGS

Manufacturers require machine tools that are extremely accurate, reliable and capable of high levels of productivity. A major contribution to the performance of any machine tool is supplied by the rolling bearings used to support the spindles, rotating tables, ball screws and other critical precision positions. A manufactured bearing's precision level has a major influence on the ability to perform in high-speed applications commonly seen in factory machining environments.

WHICH TYPE OF TIMKEN BEARING IS MOST APPROPRIATE FOR YOUR MACHINE TOOL APPLICATION?

To achieve the highest possible performance precision level, the majority of machine tool-related bearing applications must address four primary requirements: speed, stiffness, accuracy and load capacity.

Speed

Today's industrial machining environments stress maximum production rates. To reach these high metal-removal goals, machines are operating at maximum speeds with working spindles tuned to provide premium running accuracy.

Achievable spindle rotating speeds require management of heat generation within the bearing assembly. The bearing's ability to not only minimize heat buildup, but also expel excess heat, is a crucial consideration in the bearing selection process. Because of the differences in rolling element contact geometry, ball bearings are superior in minimizing heat generation, especially where higher speeds are desired.

Fig. 1 compares the relative maximum speed of similar cross section ball and tapered roller bearings (both using synthetic grease as a baseline lubricant). Therefore, in applications where higher RPM levels are the primary concern, ball bearings have a distinct advantage.

Bearings must be carefully designed to minimize heat generation and vibration to enable high speeds. Specific strategies include overall bearing configuration, precision internal geometry and material selection. Optimizing bearing ring and shoulder construction with the ball complement supports higher speed performance. Engineering raceway and ball geometry helps to minimize friction, while ceramic rolling elements generate less heat with reduced skidding.

To prevent vibration at high speeds, close bearing tolerances are required. Timken® machine tool bearings are designed to meet or exceed industry tolerance standards and deliver smooth running performance.

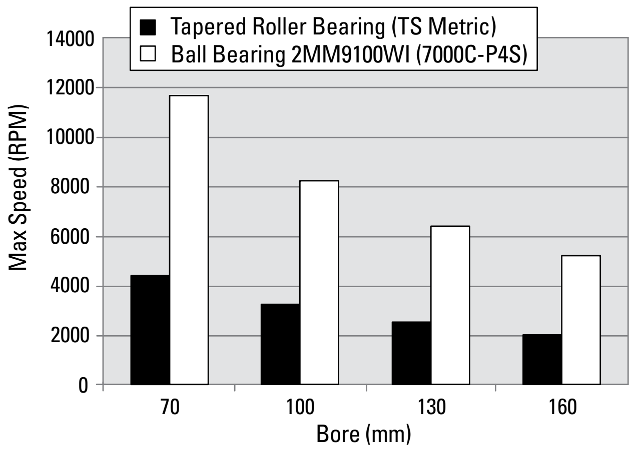


Fig. 1. Tapered roller bearing vs. ball bearing maximum permissible speed (with synthetic high-speed grease).

In addition to the use of hybrid ceramic technology, further enhancements in speed, as well as control of bearing noise and temperature, can be achieved through cage design or material, and choice of lubrication.

Many of the factors that allow for maximum speed have been designed into Timken's HX series of super precision ball bearings. Engineered for the reduction of friction and minimum heat buildup, this series features unique ball complements and precision-engineered surface geometries. Options for further enhancing speed include the use of low weight ceramic rolling elements, lubrication designed for high speed, and lighter preload levels. These are discussed in further detail later in this catalog.

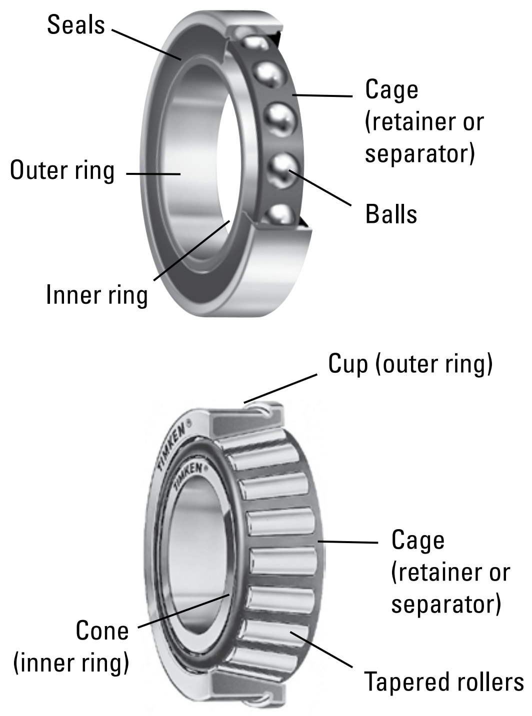


Fig. 2. Bearing components.

Stiffness

The ability to minimize tool deflection experienced under cutting loads is vital to achieving the accuracy needed to produce finished parts within specified tolerances. Less variance produces better quality and helps keep product scrap levels at a minimum. Bearings have a significant effect on spindle stiffness, due to their deflection under applied load. Because of their internal geometry and rolling element type, tapered roller bearings provide considerably higher stiffness levels as shown in Fig. 3 and 4.

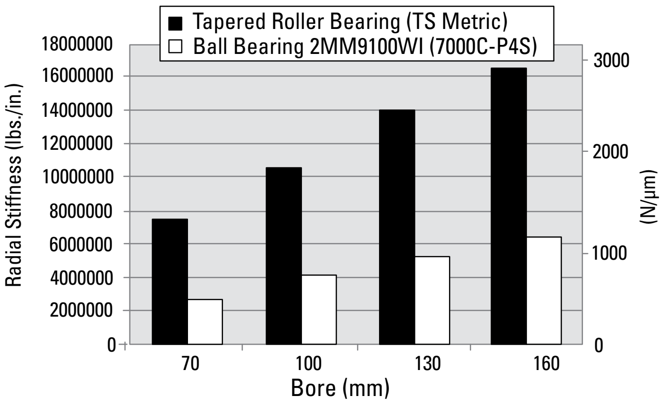


Fig. 3. Tapered roller bearing vs. ball bearing radial stiffness.

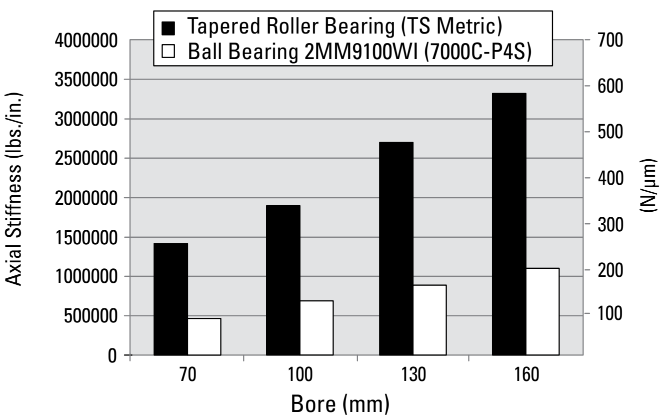


Fig. 4. Tapered roller bearing vs. ball bearing axial stiffness.

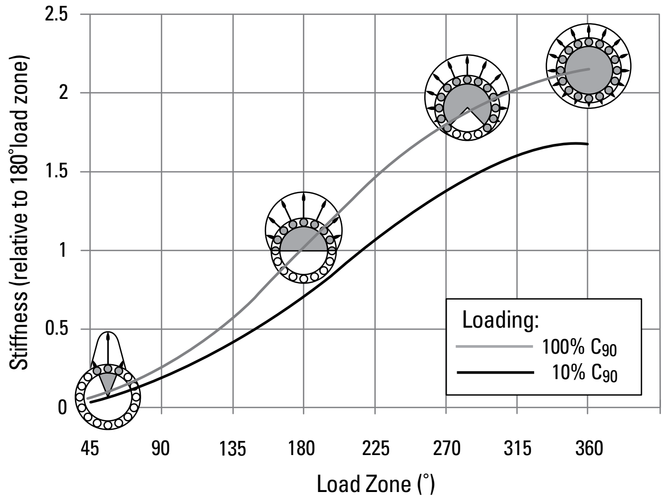


Fig. 5. Effect of load zone on bearing stiffness.

Bearing stiffness also depends on design load zone, which is directly related to bearing setting, clearances and applied loads. (Setting is defined as a specific amount of either end play or preload.)(1) A bearing under radial load with zero end play/zero preload has a load zone close to 180 degrees, while a bearing with preload can reach 360 degrees load zone. Fig. 5 shows the effect of load zone on tapered roller bearing stiffness. The curves demonstrate that while the effect of external loads on stiffness is important, the impact from setting is more significant.

Since thermal expansion can dramatically affect preload or setting, it can also play a very important role in the resulting static and dynamic stiffness of a spindle system. This applies to ball bearings as well.

An inherent advantage of the tapered roller bearing is that it can be adjusted after mounting. This means that the optimum stiffness can be obtained either by determining the proper setting during the mounting phase for a simple bearing arrangement, or during running by the use of a "variable preload" bearing design such as Timken® Hydra-Rib™ and Spring-Rib™ bearings.

To better manage the load sharing of the set of rolling elements, Timken offers a variety of designed-in preload levels for ball bearings. Be conservative with the addition of preloading as these forces will contribute to heat generation, reducing the maximum permissible speed of either ball or tapered roller bearing designs.

(1)For additional information, see guidelines on page 94.

Accuracy

Another key factor in the machine's precision is the runout (rotational accuracy) of the bearing. This affects the geometry and surface finish of the target workpiece. For the ultimate accuracy and repeatability of machine tool motion, Timken® super precision ball bearings offer the best control, with quietness of operation and reduced vibration. To achieve the highest level of precision with increased stiffness and load capacity, multiple ball bearings may be used in sets specifically designed for this purpose.

The most widely recognized definition of quality is contained within the ABEC/ISO classes; however some factors affecting performance of a bearing are not completely defined within these standards. This allows for a significant range of variability in product performance among bearing manufacturers. To provide premium performance, all Timken ball bearing MV(P4), MM/MMV(P4S) and MMX(P2) precision grades comply with strict controls over these non-specified parameters – all of which can have a direct impact on the service life and performance of a bearing.

**TABLE 1.
PRECISION BEARING CLASSES**

Tapered Roller Bearings - Precision Class				
Timken Metric	C	B	A	AA
Timken Inch	3	0	00	000
ISO/DIN	P5	P4	P2	-
ABMA Metric	C	B	A	-
ABMA Inch	3	0	00	-

Ball Bearings - Precision Class				
Timken FAFNIR	V	MV	MM/MMV	MMX
Timken ISO	P5	P4	P4S	P2
ISO/DIN	P5	P4	-	P2
ABMA	ABEC 5	ABEC 7	-	ABEC 9

Crossed Tapered - Precision Class		
Timken Metric	S	P
Timken Inch	3	0

High Precision Ball Bearings (ABEC 7; ISO P4)

Timken high precision ball bearings manufactured to the MV (P4) tolerance class operate with running accuracy, performance levels and dimensional controls conforming to ABEC 7 (ISO P4) tolerances. These bearings are the right choice for applications where both precision and cost effectiveness drive the selection.

Super Precision Ball Bearings, Super High Precision (ABEC 7/9; ISO P4/P2)

Timken super precision ball bearings manufactured to the MM/MMV(P4S) tolerance class operate with running accuracy and performance levels meeting ABEC 9 (ISO P2) standards yet maintain other features at ABEC 7 (ISO P4) level for cost-effectiveness. Bore and O.D. surfaces are coded in micron units for the convenience of the discriminating machine tool builder striving for optimum fitting of crucial spindle components.

Ultra-Precision Ball Bearings (ABEC 9, ISO P2)

Timken MMX(P2) super precision ball bearings, with closer tolerances and running accuracies than ABEC 7 (ISO P4) bearings, are made to ABEC 9 (ISO P2) tolerances. Bearings produced to these tolerances are generally used on ultra-high-speed grinding spindles designed for tight dimensional tolerances and superfine surface finishes. Contact your Timken representative for availability of product range.

Precision Tapered Roller Bearings (Class C/S/3, B/P/0 and A/00)

The more demanding the precision objective, the more accurate the bearing must be. Timken provides three tapered roller bearing classes, in both metric and inch systems, that cover the full range of precision application requirements. In ascending order of accuracy, they are identified in the metric system as Class C/S, B/P, and A, while in the inch system as Class 3, 0, and 00.

Precision Plus™ Tapered Roller Bearings (Class 000/AA)

To further minimize the influence of variations, Timken offers a fourth level of precision tapered roller bearing manufacture so tightly controlled that it goes beyond the grade levels of both ISO and ABMA standards. Timken's Precision PlusTM line offers (metric-nominal) AA level and (inch-nominal) 000 tapered roller bearings in various sizes and styles.

Timken's MV (P4) and MMV (P4S) bearings are micron coded for the best possible fit up with the shaft and housing in the application. Even though these bearings conform to MV (P4) precision dimensionally, allowing for a lenient tolerance spread, micron coding ensures that the bearing combinations in a set are made so that the bores and outer diameters do not vary by more than 2µm from bearing to bearing.

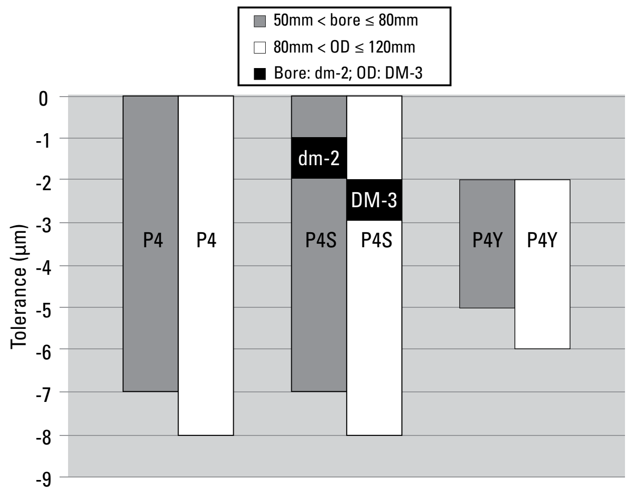


Fig. 6. Tolerance spreads of a bearing with a bore micron code of dm-2 and an O.D. micron code of DM-3.

TABLE 2.
TYPICAL PRECISION CLASSES FOR COMMON MACHINE TOOL BEARING APPLICATIONS

Class MV(P4)	• Live tools • Slab milling machines • Textile spindles
Class MM/MMV(P4S)	• High-speed motorized routers • Precision milling/boring machines • Super precision lathes • Precision surface grinding machines
Class C or 3	• Low precision machines • Drilling machines • Conventional lathes • Milling machines • Precision gear drives
Class B or 0	• NC lathes • Milling/boring machines • Machining centers
Class MMX(P2)	• Ultra-precision grinding machines • Ball screws
Class A or 00	• Grinding machines • Jig boring machines • Workpiece spindles (of cylindrical grinders)
Class AA or 000	• High accuracy machines • Precision measuring instruments • Special applications

Timken engineers have at their disposal vast resources of engineering data and application information to select the right bearing class and tune the critical components so that the machine tool achieves its performance objectives. The adjacent table can be considered as a general guideline for common machine tool bearing applications.

LOAD CAPACITY

Some machining centers, such as rough grinding operations, are strategically designed for higher material removal rates. The need for aggressive feed rates requires higher load-carrying capacities. These loads can be properly distributed among the rolling elements by providing a permanent force called "preload" or "setting." Preload is the strategic removal of radial play within the bearing to ensure proper rolling element contact on both the inner and outer race. While Timken posts its load capacities in the product tables within this catalog, many applications often approach only a fraction of those limits. For example, workpiece finish may determine the feed rates needed in an application, thereby decreasing the importance of bearing capacity.

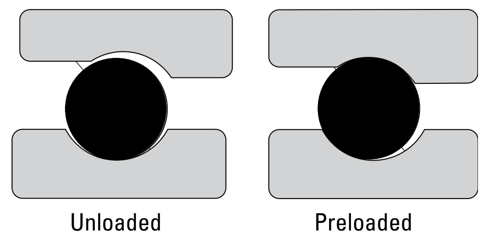


Fig. 7. Effect of preloading on ball bearing raceway contact.

Fig. 8 compares the levels of static capacity of ball vs. tapered roller bearings for the benefit of contrasting basic load capability of both bearing types.

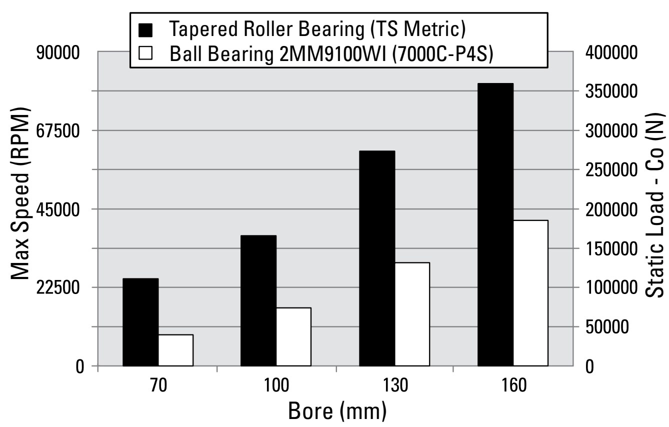


Fig. 8. Tapered roller bearing vs. ball bearing static capacities.

Consult the topics in this section addressing static and dynamic load capacity for more detailed information regarding ball bearings and tapered roller bearings to refine your choice.

Timken can assist in the final bearing selection to help you achieve your precision machining production goals. Timken's staff of application engineers is ready to put its vast experience to any test for assisting our customers with the challenging bearing applications commonly found in the machine tool industry. To refine your search, please turn to the sections covering tapered roller bearings or ball bearings for more information needed to obtain a complete Timken part number specification.

TIMKEN® MACHINE TOOL BEARING DESIGN

From this introductory discussion and the additional technical content within this catalog, one can obtain an indication of which rolling bearing type should be further investigated to meet the given set of boundary conditions and performance expectations.

TIMKEN® PRECISION TAPERED ROLLER BEARINGS

The fundamental design principles of the tapered roller bearing make it an ideal solution for low-speed/high-load or low-speed/high-stiffness requirements of machine tool applications.

True Rolling Motion

The angled raceways of a tapered roller bearing enable it to carry combinations of radial and axial loads. True rolling motion of the rollers and line contact on the race allow the bearing to run cooler and improve spindle stiffness and accuracy as compared to other roller bearing types. The true rolling motion is the result of two design features: the taper of the roller and the contact between the spherical surface ground on the large end of the rollers and the race rib. The rollers are designed so extensions of the lines along the roller body converge toward the centerline of the bearing and meet at an apex on this centerline (Fig. 9). As a result, there is no relative slip between the rollers and races.

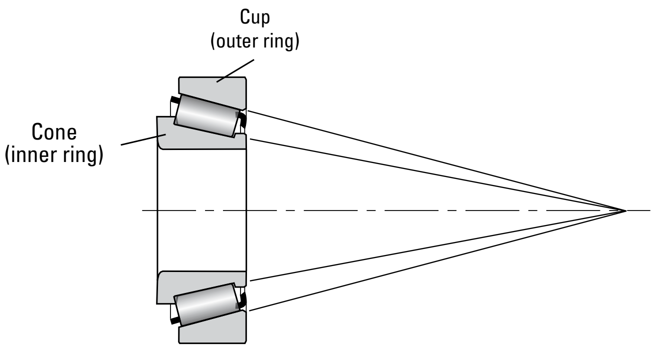


Fig. 9. On-apex design results in true rolling motion at all points along the roller body.

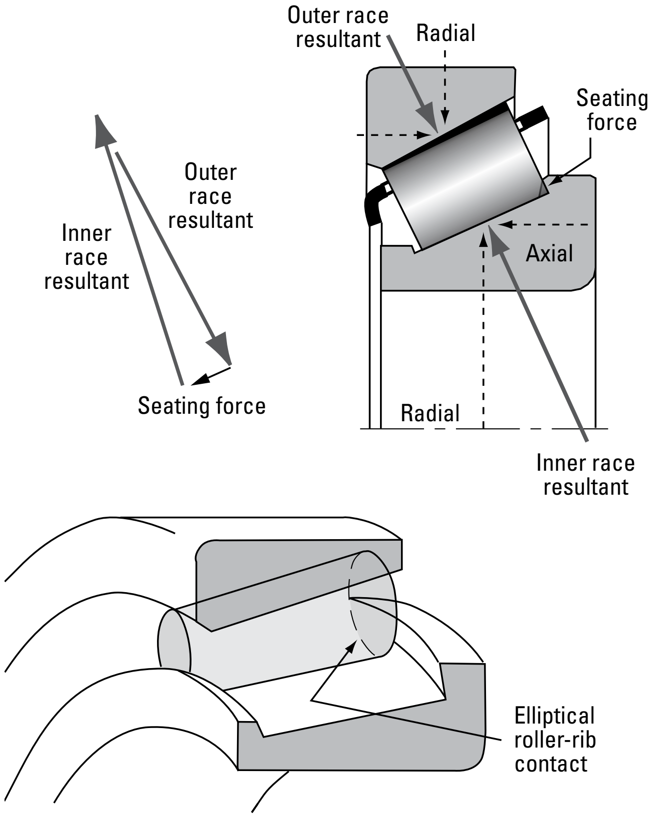


Fig. 10. Small seating force from the inner race rib keeps rollers aligned on the raceway.

The tapered configuration of the roller not only ensures that the surface speeds of the rollers and races match at every point along the roller body, but also generates a seating force that pushes the rollers' spherical ends against the race rib. This desirable seating force is a function of the different angles of the outer and inner races (Fig. 10) and prevents rollers from skewing off apex. No skew means positive roller alignment, thereby enhancing bearing life, stiffness and accuracy.

Some applications require a level of precision that cannot be achieved with standard tapered roller bearings. Timken® precision tapered roller bearings promote and maintain the operating accuracy required of today's machine tool industry and various related, specialized markets. Precision class tapered roller bearings offer machine tool builders an economical design solution that exceeds most application needs for rotational accuracy and rigidity.

PRECISION CLASSES

Timken's high-precision tapered roller bearings consist of carefully matched components that offer an added degree of fine-tuning in the bearing setting and adjustment procedure to maximize customer machine productivity. Timken manufactures high-speed designs with a variable preload capability for optimum machining and Precision Plus bearings – having an overall radial runout less than a single micron.

The application of precision tapered roller bearings is not limited to machine tools. Wherever spindles turn and rotational accuracy is essential to the machine's performance, precision tapered roller bearings can be an excellent choice. Other typical applications are printing presses, optical grinders, profile cutters, indexing tables, precision drives, measuring gauges and ball screw drive applications.

To better serve the global machine tool market, Timken has manufacturing resources around the world focused exclusively on premium precision bearings. With these dedicated resources, precision quality is built into the bearing during manufacturing. To further increase service reliability, Timken precision tapered roller bearings are manufactured from high-quality steel alloys.

Precision Tapered Roller Bearing Types

The size range of Timken precision tapered roller bearings starts from less than 20.000 mm (0.7874 in.) bore and extends to more than 2000.000 mm (78.7402 in.) O.D., depending on bearing type. The most popular types made in precision classes are the single-row TS and flanged TSF. Comprised of two main separable parts, they are usually fitted as one of an opposing pair. These bearing types are supported by a range of special bearings which have been designed for machine tool applications, such as the variable preload Hydra-Rib bearing, the high-speed TSMA bearing, and the compact TXR crossed roller bearing, which is available only in precision classes. Timken also offers a selection of two-row precision tapered roller bearings types such as the double outer ring type TDO.

Crossed Roller Bearings

A crossed roller (TXR) bearing is comprised of two sets of bearing races and rollers brought together at right angles to each other, with alternate rollers facing opposite directions, within a section height not much greater than that of a single-row bearing. Also, the steep-angle, tapered geometry of the bearing causes the load-carrying center of each of the races to be projected along the axis, resulting in a total effective bearing spread many times greater than the width of the bearing itself.

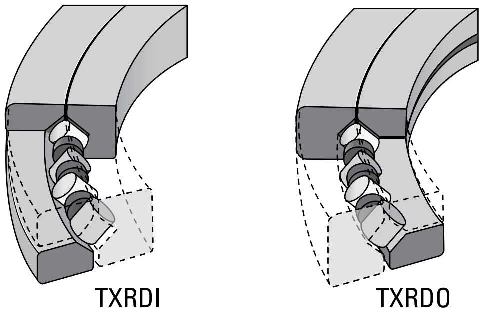


Fig. 11. TXRDI and TXRDO.

Because of the ability of the crossed roller bearing to withstand high overturning moments, it is ideal for the table bearing of machine tools such as vertical boring and grinding machines. This bearing also is well-suited for other pivot and pedestal applications where space is limited or the lowest possible center of gravity of a rotating mass is required.

Crossed roller bearings are available in two precision classes:

- Metric system Class S and P.
- Inch system Class 3 and 0.

The most common form of the bearing is type TXRDO, which has a double outer race and two inner races, with rollers spaced by separators.

Other mounting configurations and sizes of crossed roller bearings can be supplied to meet particular assembly or setting requirements. Please contact your Timken representative for further information. Also, refer to Section B for more details.

Hydra-Rib™ Bearings

Experience has demonstrated that by optimizing the design parameters of bearing geometry, spindle diameter, bearing spread, lubrication system and mounting, the two single-row bearing layout provides good results over a range of speeds and power. However, for very wide variations of speed and load, the variable preload Timken Hydra-Rib bearing concept is an excellent solution.

The Hydra-Rib bearing (Fig. 12 on the following page) has a floating outer ring rib in contact with the large roller ends instead of the usual fixed inner ring rib. This floating rib operates within a sealed cavity at a given pressure controlled by an appropriate hydraulic or pneumatic pressure system. Changing the pressure consequently changes the preload in the bearing system.

Fig. 12. Exploded view of a typical Hydra-Rib™ bearing.

The controlled pressure enables the floating rib to maintain constant spindle preload even though differential thermal expansion occurs in the spindle system during the working cycle. By changing the pressure, a variable preload setting can readily be achieved. This unique bearing concept allows the operator to control any machining condition by simply changing the pressure to optimize the dynamic stiffness and damping characteristics of the spindle. Furthermore, the hydraulic or pneumatic pressure control system can easily be monitored by the numerical control of the machine. In the case of oil pressure control, the hydraulic circuit of the machine can be used.

Your Timken representative should be consulted to determine the optimum bearing selection as well as the pressure figures, as a function of the given running conditions.

TIMKEN® SUPER PRECISION BALL BEARINGS

The Timken line of super precision machine tool bearings includes bearings designed to meet ABEC7 (ISO P4) and ABEC 7/9 (ISO P4/P2) tolerance levels. However, Timken manufactures all super precision ball bearings to surpass ISO/ABMA criteria to ensure that the end users receive only the highest quality product to maximize machine performance.

Spindle bearings are the most popular type of super precision ball bearing used within the machine tool industry. These angular contact bearings are used primarily in precision, high-speed machine tool spindles. Timken manufactures super precision machine tool bearings in four metric ISO dimensional series. In addition, because of specialized variations of bearing design and geometry, Timken offers a total of seven angular contact bearing types within these four basic series:

- ISO 19 (9300WI/71900, 9300HX/HX71900 series)
- ISO 10 (9100WI/7000, 9100HX/HX7000, 99100WN/WN7000 series)
- ISO 02 (200WI/7200 series)
- ISO 03 (300WI/7300 series)

Multiple internal geometries are available to optimize either load-carrying capacity or speed capability: WI, WN, HX or K. WI-type bearings are designed to maximize capacity of the various bearing cross sections and are used in low to moderate speeds. The HX is Timken's proven high-speed design. It has a significant advantage at higher speeds, generating less heat and less centrifugal loading forces. The WN-type is generally a compromise between the WI and HX as it offers higher speed capability than the WI, but lower capacity and higher stiffness than the HX design, with lower speed capability.

Most of the bearing types are available in either 15 degree (2/C) or 25 degree (3/E) contact angles. In addition, Timken now stocks more ceramic ball sizes than ever for the highest speed requirements.

The K-type deep-groove (Conrad) super precision radial bearing is generally used in applications where capacity and stiffness do not require sets containing multiple bearings. By virtue of the single-row, radial deep-groove construction, and super precision level tolerances, these are capable of carrying thrust loads in either direction, and have a relatively high-speed capability – especially if a light axial preload is applied. Timken offers deep-groove super precision radial machine tool bearings in the following ISO dimensional series:

- ISO 10 (9100K/6000 series)
- ISO 02 (200K/6200 series)
- ISO 03 (300K/6300 series)

Ball Screw Support Bearings

To meet the demands of the servo-controlled machinery field, Timken® ball screw support bearings are specially designed with steep contact angles and provide high levels of stiffness for ball screw application requirements. Timken's series of double-row, sealed, flanged (or cartridge) units use an integral double-row outer ring to help simplify installation procedures. Timken offers the following ball screw support bearing products:

- Inch series bearings (MM9300)
- Metric series bearings (MMBS)
- Flanged cylindrical cartridge housings (BSBU)
- Pillow block housings (BSPB)
- Integral double-row units (MMN, MMF)

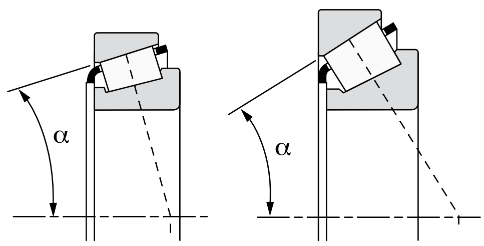
SELECTING THE APPROPRIATE MACHINE TOOL BEARING

PRECISION TAPERED ROLLER BEARINGS

Angularity (K-factor)

The angled raceways allow the tapered roller bearing to carry combinations of radial and axial loads. Since load capacities are intrinsically linked to the bearing stiffness, the selection of the most appropriate tapered roller bearing cup angle can help optimize the bearing selection for a given application.

The angularity of the bearing is often described by a factor called "K." This factor is the ratio of basic dynamic radial load rating (C90) to basic dynamic axial load rating (Ca90) in a single-row bearing. For a bearing with a ribbed cone (the most common design), it is a function of the half-included cup angle (α) and can be found listed with the geometry factors in the catalog appendix. The smaller the K factor, the steeper the bearing angle. (See Fig. 13).



Shallow angle for heavy radial load

Steep angle for heavy axial load

$$K = \frac{C_{90}}{C_{a90}} = \frac{0.39}{\tan(\alpha)}$$

Fig. 13. Designs to support radial and axial loads in any combination.

Size

Optimizing stiffness is often a customer's primary design goal when choosing a tapered roller bearing. This usually results in the determination of a desired spindle diameter. Therefore, meeting a given envelope narrows the choices for the tapered roller bearing size selection.

Speed

The next most common criteria are the speed capability/limitations of the remaining potential candidates. This can be challenging, since the speed rating of a tapered roller bearing is a function of its internal geometry, the axial setting under operation conditions, the lubricant used and method of delivery. There is a speed guideline matrix on page 43 that will aid in determining the speed rating and suggested lubricant/delivery method for your tapered roller bearing application. Included in the appendix is a table listing the G1 and G2 factors that can be utilized to compare the relative speed capability and heat generation between the various tapered roller bearing selections. Please refer to the topics on permissible operating speeds and heat generation for further discussion.

Construction

Tapered bearings are uniquely designed to manage both axial and radial loads on rotating shafts and in housings. The steeper the cup angle, the greater the ability of the bearing to handle axial loads. Customized geometries and engineered surfaces can be applied to these bearings to further enhance performance in demanding applications.

Timken has designed a variety of tapered roller bearing types to specifically address various machine tool requirements. Each of these designs is best suited to a specific set of application needs. The key features of each type are highlighted below:

TS or TSF Bearing

- Most widely used type of tapered roller bearing
- Minimum precision grade Level 3 or C (ISO P5)
- TSF has a flanged outer ring to facilitate axial location
- Available in most bearing series
- Used in rotating shaft applications

TSMA Bearing

- Axial oil manifold with axial holes through rib
- Suitable with circulating oil or oil mist lubrication
- Centrifugal force distributes oil to critical rib/roller end contact
- Available in most precision grades
- Available in most bearing series
- Used in rotating shaft applications

Hydra-Rib Bearing

- Designed to maintain optimum spindle system preload
- Floating outer ring rib is positioned by a “pressure” system
- Rib in contact with the large roller ends instead of the usual fixed inner ring rib
- Variable preload setting adaptable to manual, tape, or computer control
- Wide speed range with optimum preload setting
- Improved spindle accuracy
- Improved static and dynamic stiffness
- Lower operating temperatures
- Heavier cuts with better tool life

Spring-Rib Bearing

- Designed to maintain optimum spindle system preload
- Floating outer ring rib is positioned by a “spring” system – pressurizing system not required
- Rib in contact with the large roller ends instead of the usual fixed inner ring rib
- Improved spindle accuracy
- Improved static and dynamic stiffness
- Heavier cuts with better tool life

Crossed Roller Bearing

- Designed to resist overturning loads
- Steep-angled geometry provides wider effective spread
- High tilting stiffness
- Adjustable design for optimum preload
- Compact design reduces space requirement
- Reduced application machining costs

End Play (Preload)

The end play of a tapered roller bearing during installation affects:

- Load zone control, impacting bearing life
- System rigidity, impacting deflection
- Housing and shaft diameter tolerances

End play/preload setting is determined based on desired stiffness, reduction in heat generation, and optimal rated life. For information on setting, please see page 93.

Precision Class

Typically, once the most appropriate bearing part number is identified for a particular application, the final parameter is the desired precision level. Standard class tapered roller bearings have crowned or enhanced profiles for races and rollers. Timken precision tapered roller bearings have straight profiles with running accuracy and performance meeting ISO P5, ISO P4, and ISO P2 levels. The Precision Plus series offers total radial runout of less than a single micron, exceeding the ISO P2 precision level and allowing for improved accuracy. The suggested assembly and/or inspection code (precision class and performance code) can be applied to the chosen part number to obtain the necessary precision level.

Other

Consult Timken for suggestions related to appropriate bearing enhancements that can improve the performance of your application.

Such enhancements might include unique precision levels, conversion of a TS-style design to a (flanged) TSF or (multi-row TDO, or possibly ceramic rolling elements for better stiffness and speedability.

SUPER PRECISION BALL BEARINGS

As previously noted, optimizing speed, stiffness, accuracy and load capacity is often a customer's primary design goal. This usually results in the identification of several characteristics that will determine the final bearing selection. The following design variables influence bearing performance as noted.

Contact Angle

A contact angle is created between the rolling element and raceway to support a combination of radial and axial (thrust) loads. Deep- groove (Conrad) bearings designed with 0-degree contact angle are best for supporting radial loads or small axial loads in two directions. To support high axial or combination loads, angular contact bearings are often preferred. For additional support of radial and axial load in either direction, sets of bearings in opposing directions are utilized.

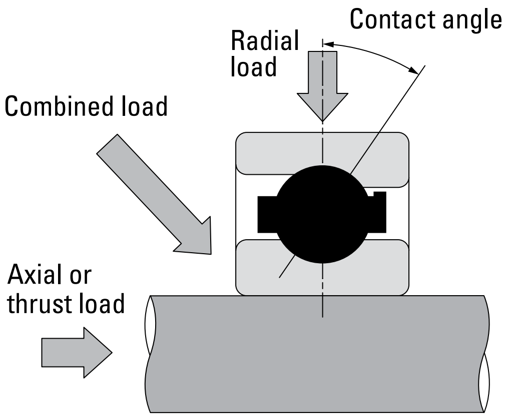


Fig 14. Example of ball bearing loading.

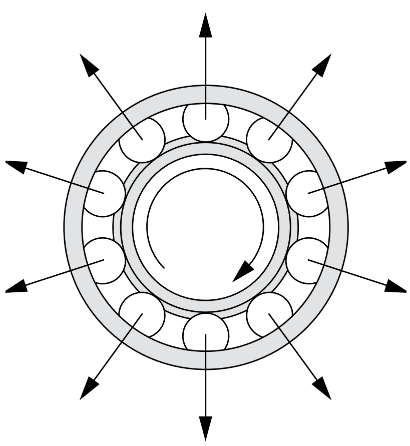


Fig 15. Centrifugal forces of a rotating ball bearing.

The majority of Timken angular contact super precision ball bearings are available with standard high (25 degree) and low (15 degree) contact angles, with additional options upon request. Each type has inherent characteristics that are desirable for machine tool spindles. Contact angle is chosen primarily based on the predominant load direction.

- 15°- used when loading is primarily radial; for very high speed applications
- 25°- used when loading is primarily axial
- 60°- highest axial stiffness; used in ball screw support bearings

High speeds cause centrifugal forces on the ball. A lower contact angle handles the centrifugal force more favorably as higher speeds lead to higher internal radial loads.

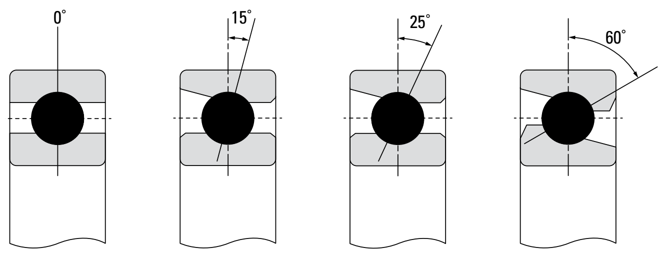


Fig 16. Examples of common machine tool ball bearing contact angles.

Spindle rotating speeds are limited by heat generated within the bearing assembly. As viewed in Fig. 17, a lower contact angle minimizes heat buildup yielding lower operating temperatures at higher speeds.

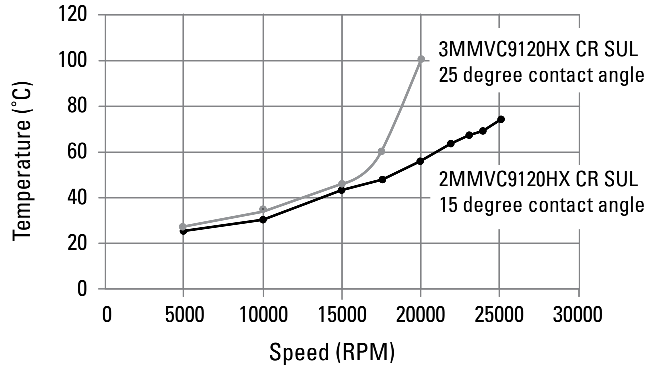


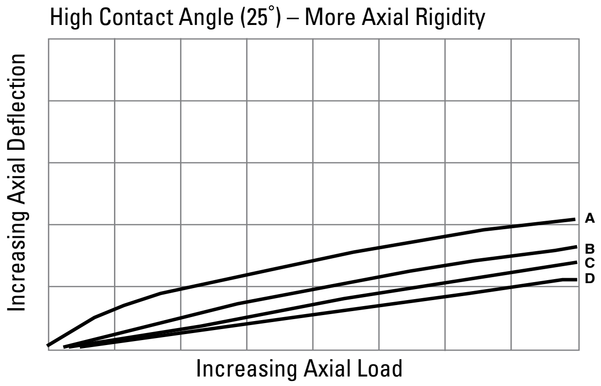
Fig 17. Effect of contact angle on temperature.

Axial deflection curves for two like-series, equal bore-sized bearings with standard preloads and contact angles are shown below.

A comparison of the curves in Fig. 18 shows the 25-degree contact angle bearing to be more rigid axially under axial loads than the 15-degree contact angle bearing. Similar comparisons of the radial deflection characteristics of the same two types of angular contact ball bearings can be made from the two graphs shown in Fig. 19. These curves show that decreased radial deflections result when bearings having a lower, 15-degree angle are used.

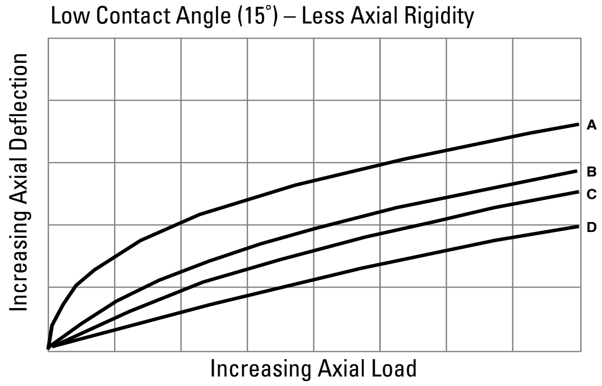
Generally, the force for the equivalent preload level for a 15-degree contact angle bearing is about one-half that of the preload level for a 25-degree contact angle bearing. Preload values for all Timken machine tool grade angular contact bearings are calculated to give optimum performance over a wide range of applications.

High Contact Angle (25˚) – More Axial Rigidity



A No Preload
B Light Preload
C Medium Preload
D Heavy Preload

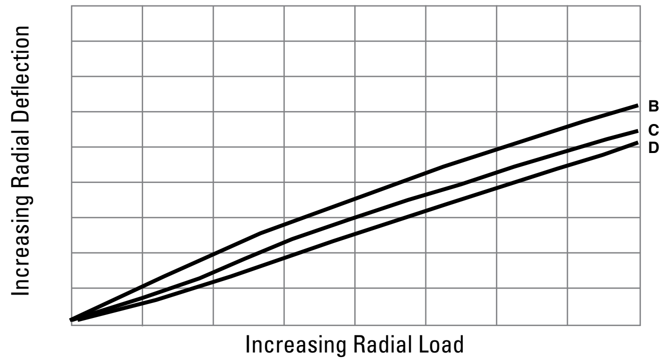
Low Contact Angle (15˚) – Less Axial Rigidity



A No Preload
B Light Preload
C Medium Preload
D Heavy Preload

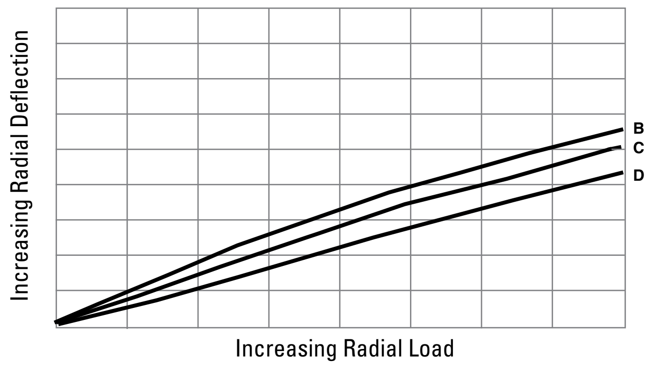
Fig. 18. Effect of contact angle on axial deflection.

High Contact Angle (25˚) – Less Radial Rigidity



B Light Preload
C Medium Preload
D Heavy Preload

Low Contact Angle (15˚) – More Radial Rigidity



B Light Preload
C Medium Preload
D Heavy Preload

Fig. 19. Effect of contact angle on radial deflection.

Timken ball screw support bearings are designed with a 60-degree contact angle. They are used for low-speed operations and offer the highest axial stiffness in the standard super precision ball bearing machine tool line. As contact angle increases, axial stiffness increases, radial stiffness decreases, and maximum operating speed decreases.

Spin To Roll Ratio

Though ball bearings are classified as rolling bearings, there is always some sliding action between the balls and the raceways in any bearing. No bearing can achieve 100 percent rolling action. This is because of relative rotation speeds between the points of primary contact on the inner and outer raceways. This is especially true in the case of angular contact ball bearings. An increase in spin is induced by excessive preloading, contact angle shift and increase in the operation speed. The measure of the roller spin is given by the spin to roll ratio. It describes the amount of spin a ball undergoes about an axis perpendicular to the bearing axis per unit rotation of the roller on the raceways. Higher the spin to roll ratio, higher is the heat generated in operation and lower is the life expectancy of the bearing.

Contact Angle Shift

Bearings are generally assembled onto a shaft or into a housing using interference fits to avoid creep (slippage). This causes a reduction in the bearing internal clearance. Additionally, angular contact ball bearings are preloaded in order to eliminate any remaining internal clearances and achieve the desired axial and radial rigidity. Preloading of a bearing causes deflections in the internal geometry and results in a shift in the contact points between the rollers and the raceways. This in turn causes a shift in the contact angle. While preloading is recommended to achieve the optimum life and performance of the bearing, the selection of the catalog preload must be done carefully, taking into consideration the fits, operation temperature and the operational preload to minimize the contact angle shift.

Precision Class

Super high precision MM/MMV (P4S) bearings are manufactured with running accuracy and performance meeting ABEC 9/ISO P2 while maintaining other features at ABEC 7/ISO P4 levels for cost-effectiveness.

Ultraprecision MMX (P2) ABEC 9/ISO P2 have closer tolerances on bore and O.D. Bearings produced to these tolerances are used on ultra-high-speed grinding spindles designed for tight dimensional tolerances and superfine surface finishes.

Hybrid Ceramic

Timken has designed an advanced bearing that combines ceramic rolling elements with premium steel rings and state-of-the-art bearing technology to achieve maximum speed capability and greater stiffness. Compared with an all-steel bearing, the hybrid ceramic bearing's lower friction characteristics, even under marginal lubrication, result in less ball skidding, lower heat generation, higher speeds and greater overall system reliability.

Ceramic balls are 60 percent lighter than steel with extremely fine surface finishes equal to or less than 0.5 microinch Ra. This helps to reduce centrifugal forces and allows for a 20 percent higher speed factor than steel balls. Oil-lubricated ceramic hybrid bearings can operate up to three million dN, while grease-lubricated ceramic hybrids can run up to one million dN.

$$dN = \text{bore size (mm)} \times \text{speed (RPM)}$$

The ceramic material has a modulus of elasticity 50 percent greater than steel, increasing bearing rigidity. This higher stiffness may result in higher contact stress levels in the bearings. Ceramics are generally not appropriate for higher loads and/or low speeds. As seen in Fig. 20, below 750.000 dN, ceramic balls experience a reduction in overall rated bearing life.

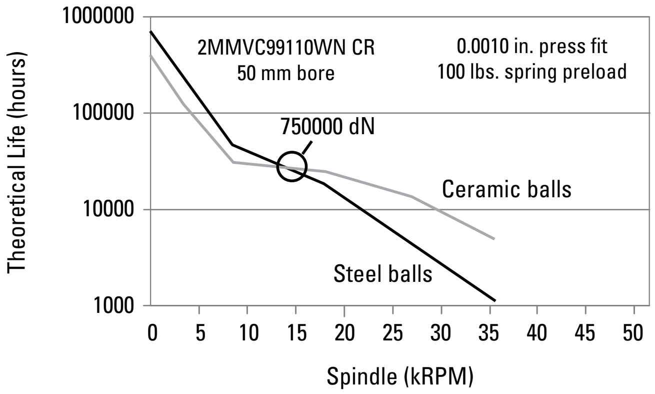


Fig. 20. L_{10} life vs. speed comparison of steel and ceramic balls.

Size/Series

Increasing bearing size causes an increase in centrifugal forces which, as discussed earlier, reduces the speedability of the bearing. An increase in ISO cross-section series improves the ability of a bearing to carry higher loads as shown in Fig. 21.

- 9300 (71900) series: ultra-light loads, high speed
- 9100 (7000) series: extra-light loads, high speed
- 200 (7200) series: light loads
- 300 (7300) series: medium loads

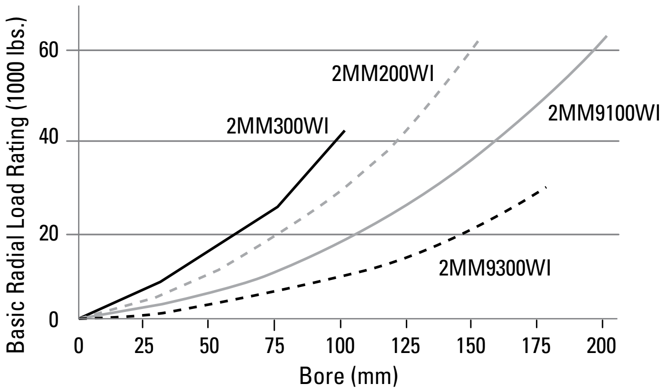


Fig 21. Angular contact ball bearing load ratings.

When stiffness is a factor of the design, this usually results in the determination of a desired spindle diameter and corresponding bearing bore size. Speedability and load capacity then determine the series selected.

Construction

WI is the standard construction for an angular contact ball bearing. The WI-type has full shoulders on both sides of the inner ring and a low shoulder on the outer ring. This simplifies bearing assembly during manufacturing.

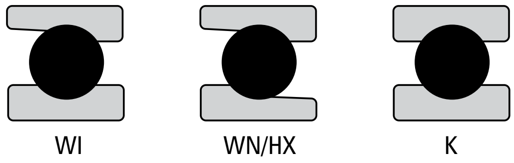


Fig. 22. Angular contact spindle bearing types.

Timken has developed an HX bearing design to enhance two key factors contributing to metalworking throughput: spindle speed and radial stiffness. This design enables spindle heads to remove more material in less time while maintaining superior finished product tolerances by minimizing tool "wander." This efficient combination translates into faster turnaround of finished product. These improvements are imparted by subtle changes to ball complements and internal geometries. The Timken HX Series is dimensionally interchangeable with the 9300 and 9100 (71900 and 7000) series ball bearings.

The HX and WN-types also are designed to meet the needs of machine manufacturers who require optimum lubrication through the bearings. These designs incorporate a low shoulder on the non-thrust side of both the inner and outer rings to facilitate oil flow.

K-type, deep groove (Conrad), 0-degree radial ball bearings are generally chosen in applications where capacity and stiffness do not require a duplex set of bearings. Axial load applied to the bearing will increase the contact angle.

Cage (Retainer or Separator)

A cage's function is to separate the rolling elements within the bearing. It first affects bearing performance by adding weight, which will increase the centrifugal forces and resulting radial load at high speeds. This limits the ability of the bearing to perform at high temperatures. Timken's range of cage types includes:

CR/T	Phenolic (composition) - standard
PRL	Molded polymer cage - new
PRC	Molded reinforced nylon - former standard
MBR	Machined bronze
PRJ	High-performance polymer
SR	Silver-plated machined steel

Timken's phenolic composition cage is standard for high-speed applications up to 93° C (200° F). For high-heat applications, machined bronze is often used. However, the increase in weight reduces the speedability of the bearing assembly.

Seals

Seals are used to exclude contaminants and retain lubricant for reliable bearing operation and extended service life. The HX bearing has an optional non-contacting seal, available as single or pair (V, VV). The WI 9100/7000 and 9300/71900 series have an optional light-drag contacting seal (P, PP). Ball screw support bearing cartridges (MMN/MMF) include integral, low-torque contact seals.

Bearing Set Quantity

Timken super precision ball bearings are available as single, duplex, triplex, and quadruplex matched sets:

SU	(X, L, M, H) - single bearing/preload level
DU	(X, L, M, H) - duplex pairs/preload level
TU	(X, L, M, H) - triplex set/preload level
QU	(X, L, M, H) - quadruplex set/preload level

Each additional bearing increases system rigidity and load-carrying capabilities. The quantity of bearings in a set is specified in the part number as shown.

Timken super precision ball bearings are universal flush ground, which allows for DB (back-to-back), DF (face-to-face), or DT (tandem) mounting of all matched duplex sets. This does not need to be specified during order.

Preload Level

The internal condition of a preloaded ball bearing is similar to that of one in operation under axial load. This initial axial load serves to decrease markedly the axial and radial deflections when subsequent operational loads are imposed on the bearing assembly. Preload levels limit change in contact angle at very high speeds, and prevent ball skidding under very high acceleration and speed.

In many cases, the amount of bearing preload is a trade-off between having the desired degree of rigidity and reducing any adverse effect preloading has on the equipment. If the operating speed is high, a heavy preload can lead to excessively high operating temperatures, which may result in bearing damage. To match general performance requirements, four classes of ball bearing preloads are used – extra-light, light, medium and heavy. To maximize performance, Timken can design specially preloaded super precision ball bearings.

Bearings come from the factory with a dimensional preload ground into the bearing ring faces. The preload changes as the raceway diameters change. Once the operational preload is determined, it is important that the correct factory (catalog) preload is selected because when the bearings are mounted on the shaft with an interference fit, the dimensional setting (or preload) increases. When the bearings are in operation and there is a temperature difference between the inner and outer rings, the dimensional setting (or preload) again increases.

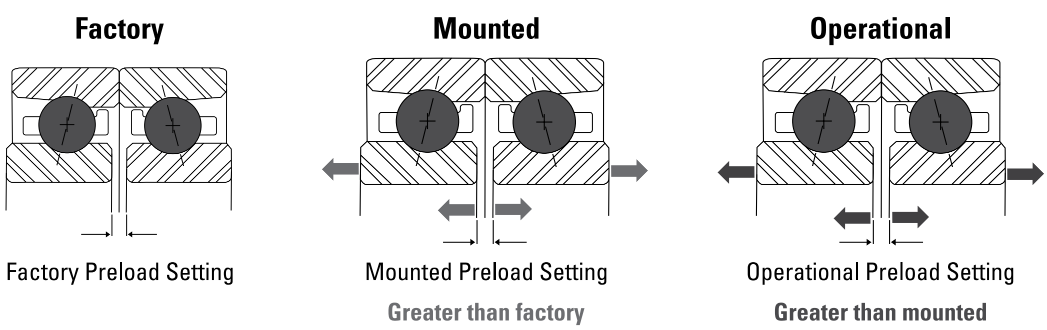


Fig. 23. Preload settings.

Sealed Ball Screw Support Bearings

Available in flanged (MMF) and non-flanged (MMN) series, these designs simplify installation for both standard and rotating nut mountings. The flanged version eliminates the need for external clamping of the outer ring. The standard version is the sealed duplex configuration, but both series may be ordered with seals or shields in either a duplex or quad set arrangement.

Wear/Debris-Resistant Material

Advanced bearings can resist debris wear for improved machine tool performance and service life. Timken super precision bearings with WearEver technology combine a special high-alloy steel with ceramic balls to provide superior performance and cutting quality for high-speed applications. Extended capabilities help improve machine efficiency and utilization while reducing secondary finishing operations and downtime.

This unique, wear/debris-resistant steel was specifically incorporated into the bearing design to withstand damage from hard contaminants, a primary cause for shortened bearing life. Studies have shown that bearings with this advanced material can deliver up to ten times the standard service life in the presence of hard contamination. This represents a significant advancement in high-speed machining productivity and improved product quality. Consult your Timken representative for availability.

Other

Bearings may be customized to specify lubrication type, special coding, special part marking, etc. Contact your Timken representative for more information.

2 MM C 91 04 WI CR DUL $\boxed{\text{A4437}}$

(C 70 04 C - T - P4S - DUL) $\boxed{\text{FS637}}$

A4437 = Special coding requirement

FS637 = Kluber isoflex NBU 15 Grease

Fig. 24. Examples of non-standard specification numbers.

SPINDLE SYSTEM CHARACTERISTICS

A machine tool designer's goal is to build a precise spindle with the least possible vibration and with the optimum heat generation and dissipation characteristics. This will then produce the best surface finish, dimensional accuracy and optimum production rates.

Due to increasing cutting speeds and forces, machine tool builders are developing spindle designs to improve dynamic stiffness.

Dynamic stiffness depends upon:

- Static stiffness
- Damping
- Mass

From a design standpoint, the bearing selection has little effect on mass, but static stiffness and damping can be altered by bearing and application design criteria. The natural frequency of a system can be radically altered by any change in the static stiffness. On the other hand, damping will determine the magnitude of displacement of a system in the chatter mode. Tests have shown that the damping varies with the type of rolling bearing used.

SPINDLE SYSTEM STATIC STIFFNESS

The static stiffness, or "spring rate," of a system is defined as the ratio of the amount of load to the deflection of the spindle at the point of load, and is expressed in N/mm (lbs./in.).

In conventional spindle designs, the load is usually applied at the end of the spindle nose.

In a spindle system, a few factors contribute to the total static stiffness:

- Bare spindle stiffness
- Bearing stiffness
- Housing stiffness

Bare Spindle Stiffness

Fig. 25 illustrates the important elements that need to be considered to determine the bare spindle stiffness:

- Diameter of the spindle
- Overhung distance from the nose bearing to the load
- Bearing spread

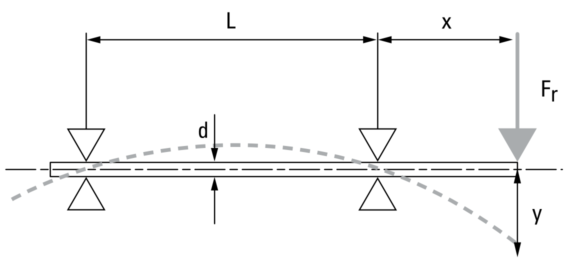


Fig. 25. Deflection of the bare spindle on two supports.

The maximum value of the spindle deflection at the point of load is:

$$y = \frac{F_r x^2 (x + L)}{3EI} \text{ (mm)} \qquad \text{with } I = \frac{\pi \times d^4}{64} \text{ (mm}^4\text{)}$$

Where:

F_r	=	radial load applied at spindle nose	(N)
L	=	bearing spread	(mm)
x	=	overhung distance	(mm)
I	=	moment of inertia	(mm^4)
y	=	deflection at point of load	(mm)
d	=	diameter of spindle	(mm)
E	=	modulus of elasticity	(N/mm^2)

Therefore, the static stiffness of the bare spindle at this point is

$$K = \frac{F_r}{y} = \frac{3EI}{x^2 (x + L)} = \frac{3E\pi d^4}{64x^2 (x + L)}$$

The previous formula shows that the diameter of a shaft is considered to the fourth power. Thus, any increase in spindle diameter will significantly increase stiffness. From a design standpoint, this means that the selected bearings should have as large a bore diameter as practical for a given outside diameter (Fig. 26 on the following page).

The overhung distance from the nose bearing to the applied loads is generally fixed by design constraints (or load cycles). However, the stiffness of the bare spindle can be increased by determining the optimum spread between the two supports. For a given overhung distance "x ," the bearing spread has an optimum value for minimum deflection at the cutting point (Fig. 27).

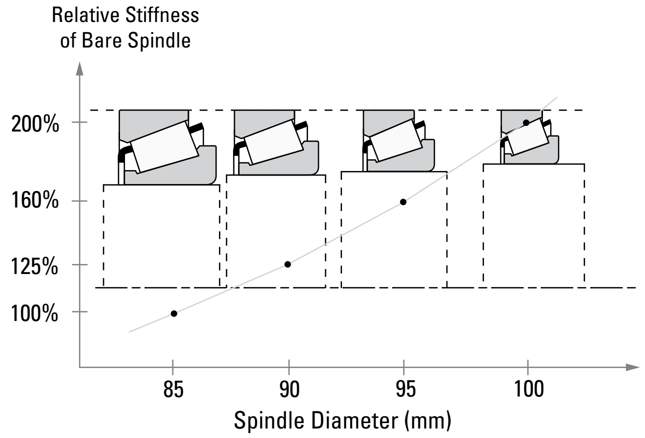


Fig. 26. Influence of spindle diameter on its stiffness for different tapered roller bearings sections within same envelope (85 mm bore taken as reference).

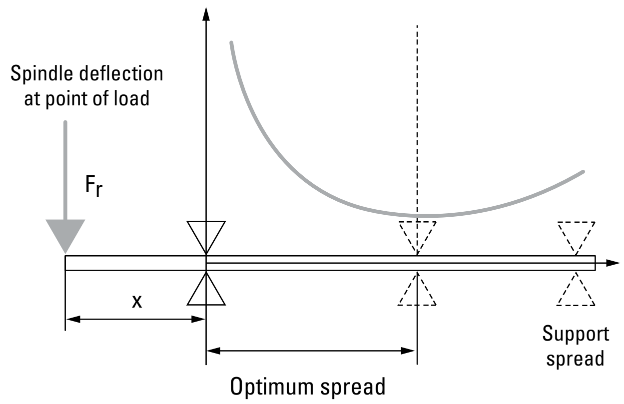


Fig. 27. Influence of spread on bare spindle deflection at point of load.

Bearing Stiffness

Stiffness is significant in precision machining applications. This impacts machine repeatability and running accuracy. Bearing stiffness is determined by movement or microscopic deflection under load within the bearing assembly.

Because of the contribution of bearing stiffness to the global system, it is of prime importance to consider the effect of the selection of the bearing and its geometrical characteristics. A tapered roller bearing is a line contact bearing with a high number of rolling elements. Compared to other popular bearings in spindle applications, such as angular contact ball bearings (point contact) or cylindrical roller bearings (line contact), the preloaded tapered roller bearing (line contact) has a significantly higher radial stiffness in the same given envelope.

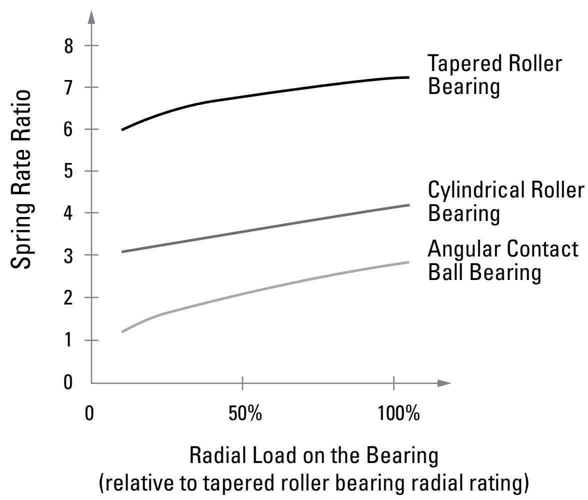


Fig. 28. Radial spring rate comparison between popular machine tool bearings of a comparable size under zero internal clearance.

Comparisons (Fig. 28) show that a tapered roller bearing has as much as six times more radial stiffness than a comparable size angular contact ball bearing, and twice as much as a comparable size cylindrical roller bearing, for a zero clearance condition Therefore, for most spindle applications only two tapered roller bearings are required, which can result in a more economical solution.

Housing Stiffness

Experience and basic calculations show that good axial and radial housing stiffness are required to support the loads that are transmitted through the bearings. In most machine tool designs, the housing is normally adequate. However, when light sections or nonferrous housings are used, the axial and radial housing stiffness should be verified.

SPINDLE SYSTEM DYNAMIC STIFFNESS

Dynamic stiffness is influenced to a large degree by the damping characteristics and the static stiffness of the system.

Fig. 29 demonstrates that bearing setting plays a major role in the static stiffness of a spindle-bearing-housing system. As the preload is increased, the static stiffness increases.

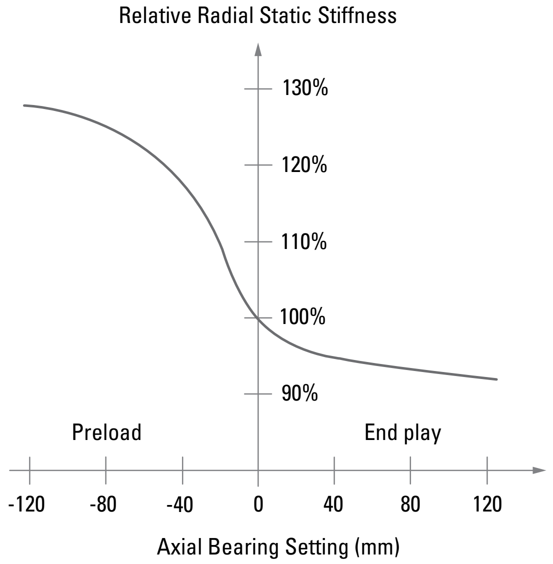


Fig. 29. Effect of bearing setting on spindle system static stiffness.

A load that would cause very little static deflection can cause very high dynamic deflections if the frequency of the dynamic load is the same as the natural frequency of the spindle. To control the dynamic stiffness, the damping characteristics of the system are very important.

Damping can be visualized as resistance to vibration. It can be seen in Fig. 30 that the damping ratio of a spindle system is higher when bearings are preloaded. The optimum value is, however, obtained around the zero clearance condition.

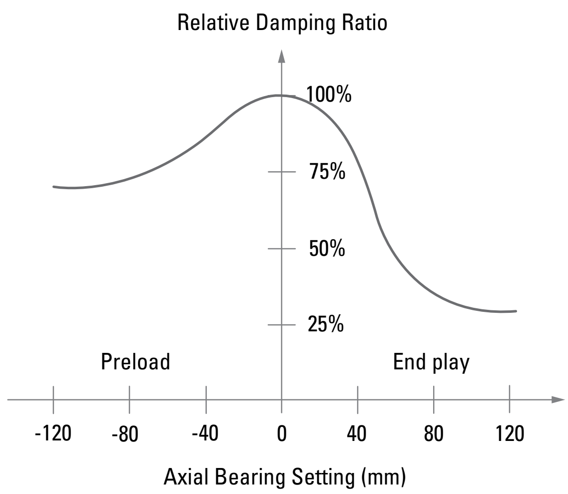


Fig. 30. Effect of bearing setting on spindle system damping ratio.

Finally, the resulting dynamic stiffness characteristics of a spindle system are directly affected by the bearing setting. The curve plotted in Fig. 31 shows an optimum setting slightly in the preload region. This gives the least compliance, or maximum dynamic stiffness, of a spindle system since the damping decreases as preload increases. As previously explained, any preload increase beyond the optimum setting will reduce the dynamic spindle characteristics.

Extensive research by Timken has resulted in a better knowledge of machine tool spindle behavior. It was identified that higher accuracy and improved surface finish can be achieved at an optimum preload setting (Fig. 32).

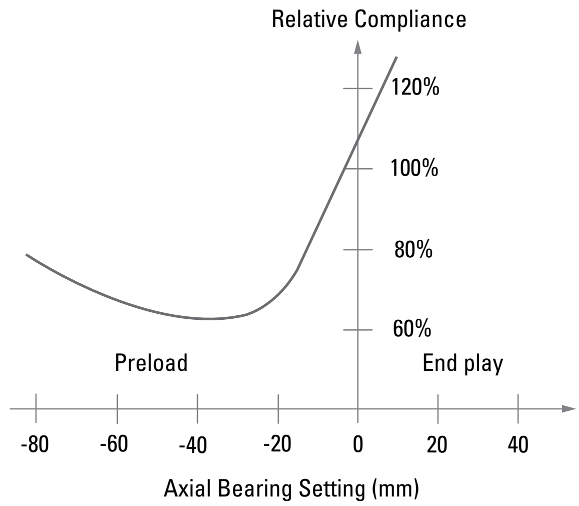


Fig. 31. Effect of bearing setting on spindle system dynamic stiffness.

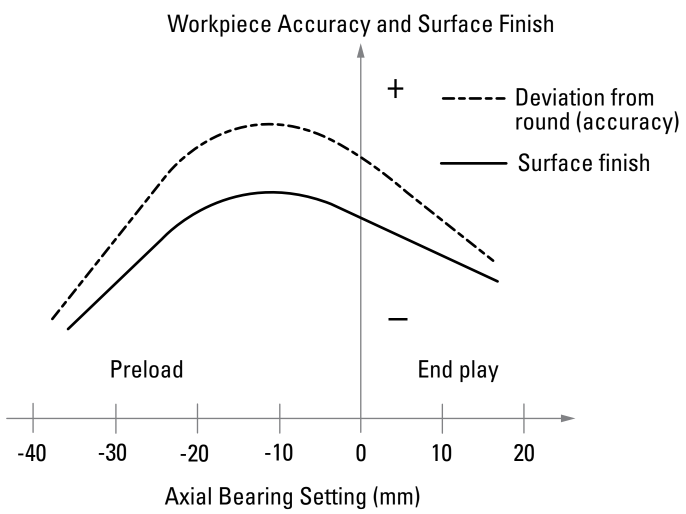


Fig. 32. Effect of bearing setting on surface finish and accuracy of the workpieces.

The unique design of a tapered roller bearing with its line contact produces a damping characteristic that is not necessarily inherent to other bearing designs (Fig. 33). This is due to the bending mode of the spindle and bearing centerline caused by dynamic deflection which is resisted inside the bearing through a shearing action of the viscous lubricant between the rollers and the cup and cone races.

It is the combination of the tapered roller bearing construction and proper bearing setting that results in improved damping characteristics.

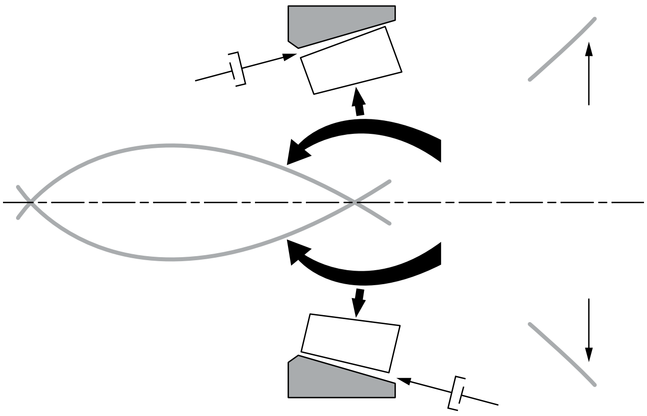


Fig. 33. Damping in a tapered roller bearing.

An extension of this insight culminated in the development of a bearing system called the Hydra-Rib. It is, specifically designed to provide the optimum bearing preload and thus the ultimate dynamic stability for the spindle system under any operating conditions.

Proper selection of the preload for a given application must not focus only on stiffness and damping characteristics. Also, the lubrication method, operating speeds and loads must be reviewed to determine the optimum setting/preload to maximize performance. Consult the appropriate topic in this engineering section for more details.

OTHER FACTORS AFFECTING BEARING SELECTION

Some additional controllable factors having a significant impact on bearing performance include mounting fits, internal clearances, lubricant type and integrity. For example, when using radial ball bearings, appropriate internal clearance is needed to ensure proper operation.

THERMAL EXPANSION

Issues such as axial displacement must be addressed where shaft length differentials must be tolerated when thermal expansion occurs. For these situations, the rotating component supports include a fixed (locating) and floating (non-locating) bearing arrangement.

The fixed bearing is subject to combination loads and is usually placed nearest the working end of the shaft to minimize motion and thereby maintain workpiece accuracy. Installation considerations for the typical fixed bearing positions should note the fitting suggestions listed in this catalog. These are compiled from a wealth of experience in a wide range of operating conditions.

Where floating bearings are necessary, the design must allow for axial displacement of the shaft. This can be accomplished by allowing the bearing to "slide" laterally along the shaft or housing respectively. A looser fit during the machining of the shaft or housing cavity is required. This not only alleviates the axial stresses on this end of the assembly, but will also facilitate bearing and shaft installation.

MISALIGNMENT

When the size of the machine increases, so do concerns about alignment. Shaft bending or additional loading can impart moment loads that need to be considered. Bearing selection must further consider installation practices with distant machined bearing housing cavities. Manufacturing limitations to position housing bores might encourage the choice of a self-aligning bearing mounting (DF, face-to-face).

This can help compensate for machining variations and assist in managing dynamic forces by featuring a spherical outside diameter or thrust face.

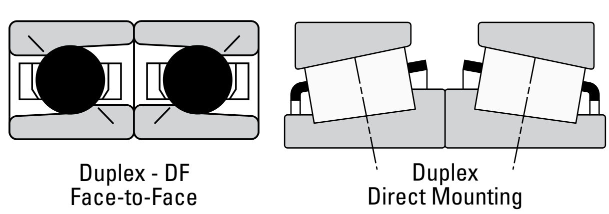


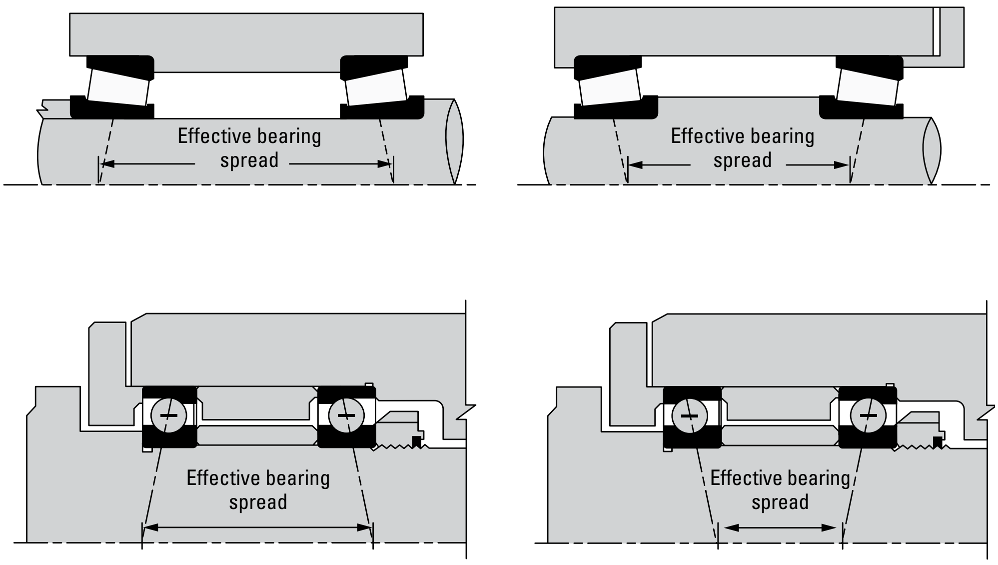
Fig. 34. Self-aligning bearing mountings.

EFFECTIVE SPREAD

When a load is applied to a tapered roller or angular contact ball bearing, the internal forces at each rolling element-to-outer raceway contact act normal to the raceway. These forces have radial and axial components. With the exception of the special case of pure axial loads, the inner ring and the shaft will experience moments imposed by the asymmetrical axial components of the forces on the rolling elements.

It can be demonstrated mathematically that, if the shaft is modeled as being supported at its effective bearing center rather than at its geometric bearing center, the bearing moment may be ignored when calculating radial loads on the bearing. Only externally applied loads need to be considered, and moments are taken about the effective centers of the bearings to determine loads or reactions. Fig. 35 shows single-row bearings in a direct and indirect mounting configuration. The choice of whether to use direct or indirect mounting depends upon the application.

With so many factors to consider for the successful operation of any device incorporating rolling bearings, Timken brings to its customers more than a century of talent and experience to assist with these choices. Though the content in this catalog is an excellent start in the rolling bearing selection process, it should not be considered the final word. Timken bearing expertise is only a phone call away.



Indirect Mounting – Tapered Roller Bearings
Back-to-Back/DB – Angular Contact Ball Bearings

Direct Mounting – Tapered Roller Bearings
Face-to-Face/DF – Angular Contact Ball Bearings

Fig. 35. Choice of mounting configuration for single-row bearings, showing position of effective load-carrying centers.

APPLIED LOADS AND BEARING ANALYSIS

This key is not intended to be comprehensive, but serves as a useful reference for symbols found in this catalog.

SUMMARY OF SYMBOLS USED IN THIS CATALOG

Symbol	Description	Units
A_1	Reliability Life Factor	
a_2	Material Life Factor	
a_3	Operating Condition Life Factor	
a_{3d}	Debris Life Factor	
a_{3h}	Hardness Life Factor	
a_{3k}	Load Zone Life Factor	
a_{3l}	Lubrication Life Factor	
a_{3m}	Misalignment Life Factor	
a_{3p}	Low Load Life Factor	
a_e	Effective Bearing Spread	mm, in.
b	Tooth Length	mm, in.
c_1, c_2	Linear Distance (positive or negative)	mm, in.
C	Dynamic Radial Load Rating`	N, lbf
C_0	Static Load Rating	N, lbf
C_{90}	Single-Row Basic Dynamic Load Rating	N, lbf
C_e	Extended Dynamic Capacity	N, lbf
C_p	Specific Heat of Lubricant	J/(kg -˚C), BTU/(lb x ˚F)
d	Bearing Bore Diameter	mm, in.
d_c	Distance Between Gear Centers	mm, in.
d_m	Mean Bearing Diameter	mm, in.
dN	Bore in mm • RPM	
D	Bearing Outside Diameter	mm, in.
D_m	Mean Diameter or Effective Working Diameter of a Sprocket, Pulley, Wheel or Tire Also, Tapered Roller Bearing Mean Large Rib Diameter	mm, in.
D_{mG}	Mean or Effective Working Diameter of the Gear	mm, in.
D_{mP}	Effective Working Diameter of the Pinion	mm, in.
D_{mW}	Effective Working Diameter of the Worm	mm, in.
D_{pG}	Pitch Diameter of the Gear	mm, in.
D_{pP}	Pitch Diameter of the Pinion	mm, in.
D_{pW}	Pitch Diameter of the Worm	mm, in.
e	Life Exponent	
f	Lubricant Flow Rate	L/min, U.S. pt/min
f_B	Belt or Chain Pull Factor	
F	General Term for Force	N, lbf
F_a	Applied Axial Load	N, lbf
F_{ae}	External Axial Load	N, lbf
F_{aG}	Axial Force on Gear	N, lbf
F_{aP}	Axial Force on Pinion	N, lbf
F_{aW}	Axial Force on Worm	N, lbf
F_c	Centrifugal Force	N, lbf
F_r	Applied Radial Load	N, lbf
F_{sG}	Separating Force on Gear	N, lbf
F_{sP}	Separating Force on Pinion	N, lbf
F_{sW}	Separating Force on Worm	N, lbf
F_{tG}	Tangential Force on Gear	N, lbf
F_{tP}	Tangential Force on Pinion	N, lbf
F_{tW}	Tangential Force on Worm	N, lbf
F_W	Force of Unbalance	N, lbf
$F_{\Phi P}$	Axial Force on Pinion	N, lbf
H	Power	kW, HP
HF_s	Static Load Rating Adjustment Factor for Raceway Hardness	
k	Centrifugal Force Constant	lbf/RPM^2
k_1	Bearing Torque Constant	N-m, lbf-in.
k_4, k_5, k_6	Dimensional Factor to Calculate Heat Generation	

Symbol	Description	Units
K	Tapered Roller Bearing Radial-to-Axial Dynamic Load Rating Factor	
K_T	Relative Axial Load Factor (Ball Bearings)	
L_H	Lead Axial Advance of a Helix for One Complete Revolution	mm, in.
L	Distance Between Bearing Geometric Center Lines	mm, in.
m	Gearing Ratio	
M	Bearing Operating Torque or Moment	N-m, N-mm, lbf-in.
n	Bearing Operating Speed or General Term for Speed	rot/min, RPM
n_G	Gear Operating Speed (RPM)	rot/min, RPM
n_P	Pinion Operating Speed (RPM)	rot/min, RPM
n_W	Worm Operating Speed (RPM)	rot/min, RPM
N_G	Number of Teeth in the Gear	
N_P	Number of Teeth in the Pinion	
N_S	Number of Teeth in the Sprocket	
P_a	Dynamic Equivalent Axial Load	N, lbf
P_0	Static Equivalent Load	N, lbf
P_{0a}	Static Equivalent Axial Load	N, lbf
P_{0r}	Static Equivalent Radial Load	N, lbf
P_r	Dynamic Equivalent Radial Load	N, lbf
Q	Generated Heat or Heat Dissipation Rate	W, BTU/min
r	Radius	
R	Percent Reliability Used in the Calculation of the a_1 Factor	
T	Torque	N-m, lbf-in.
v	Vertical (used as subscript)	
V	Linear Velocity or Speed	km/h, mph
V_r	Rubbing, Surface or Tapered Roller Bearing Rib Velocity	m/s, fpm
α X	Dynamic Radial Load Factor	
Y	Dynamic Axial Load Factor	
Y_1, Y_2, Y_3	Axial Load Factors	
Υ_G	Bevel Gearing – Gear Pitch Angle	deg.
	Hypoid Gearing – Gear Root Angle	deg.
Υ_P	Bevel Gearing – Pinion Pitch Angle	deg.
	Hypoid Gearing – Pinion Face Angle	deg.
	Half Included Cup Angle	deg.
A, B...	Bearing Position (used as a subscript)	cSt
η	Efficiency, Decimal Fraction	
θ_{ambt}	Ambient Temperature	˚C, ˚F
θ_1, θ_2, θ_3	Gear Mesh Angles Relative to the Reference Plane	deg.
θ_i, θ_o	Oil Inlet or Outlet Temperature	˚C, ˚F
λ	Worm Gear Lead Angle	deg.
μ	Coefficient of Friction	
ν	Lubricant Kinematic Viscosity	cSt
σ_0	Approximate Maximum Contact Stress	MPa, psi
Φ	Normal Tooth Pressure for the Worm (Gearing)	deg.
Φ_G	Normal Tooth Pressure Angle for the Gear	deg.
Φ_P	Normal Tooth Pressure Angle for the Pinion	deg.
ψ_G	Helix (Helical) or Spiral Angle for the Gear	deg.
ψ_P	Helix (Helical) or Spiral Angle for the Pinion	deg.
ρ	Lubricant Density	kg/m^3, lb/ft^3
1, 2, ...n	Conditions (used as subscript)	

The following equations are used to determine the forces developed by machine elements commonly encountered in bearing applications.

SPUR GEARING

TANGENTIAL FORCE

$$F_{tG} = \frac{(1.91 \times 10^7)\ H}{D_{pG}\ n_G} \quad \text{(newtons)}$$

$$= \frac{(1.26 \times 10^5)\ H}{D_{pG}\ n_G} \quad \text{(pounds-force)}$$

SEPARATING FORCE

$F_{sG} = F_{tG} \tan \phi_G$

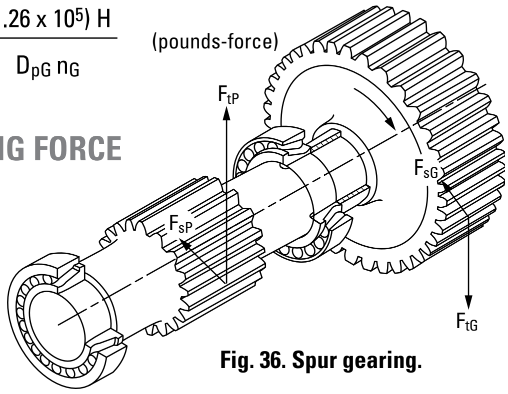


Fig. 36. Spur gearing.

SINGLE HELICAL GEARING

TANGENTIAL FORCE

$$F_{tG} = \frac{(1.91 \times 10^7)\ H}{D_{pG}\ n_G} \quad \text{(newtons)}$$

$$= \frac{(1.26 \times 10^5)\ H}{D_{pG}\ n_G} \quad \text{(pounds-force)}$$

THRUST FORCE

$F_{aG} = F_{tG} \tan \psi_G$

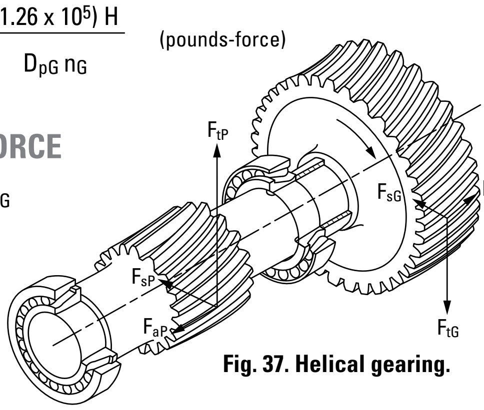


Fig. 37. Helical gearing.

SEPARATING FORCE

$$F_{sG} = \frac{F_{tG} \tan \phi_G}{\cos \psi_G}$$

STRAIGHT BEVEL AND ZEROL GEARING WITH ZERO DEGREES SPIRAL

In straight bevel and zerol gearing, the gear forces tend to push the pinion and gear out of mesh, such that the direction of the thrust and separating forces is always the same regardless of direction of rotation (Fig. 38). In calculating the tangential force (F_{tP} or F_{tG}) for bevel gearing, the pinion or gear mean diameter (D_{mP} or D_{mG}) is used instead of the pitch diameter (D_{pP} or D_{pG}). The mean diameter is calculated as follows:

$$D_{mG} = D_{pG} - b \sin \Upsilon_G \qquad \text{or} \qquad D_{mP} = D_{pP} - b \sin \Upsilon_P$$

In straight bevel and zerol gearing:

$F_{tP} = F_{tG}$

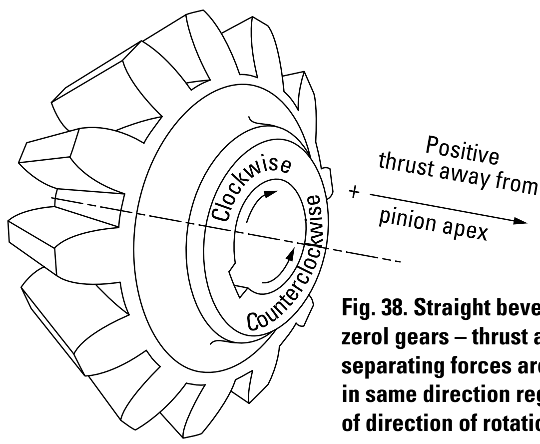


Fig. 38. Straight bevel and zerol gears – thrust and separating forces are always in same direction regardless of direction of rotation.

PINION

TANGENTIAL FORCE

$$F_{tP} = \frac{(1.91 \times 10^7)\ H}{D_{mP}\ n_P} \quad \text{(newtons)}$$

$$= \frac{(1.26 \times 10^5)\ H}{D_{mP}\ n_P} \quad \text{(pounds-force)}$$

THRUST FORCE

$F_{aP} = F_{tP} \tan \phi_P \sin \Upsilon_P$

SEPARATING FORCE

$F_{sP} = F_{tP} \tan \phi_P \cos \Upsilon_P$

STRAIGHT BEVEL GEAR

TANGENTIAL FORCE

$$F_{tG} = \frac{(1.91 \times 10^7)\,H}{D_{mG}\,n_G} \quad \text{(newtons)}$$

$$= \frac{(1.26 \times 10^5)\,H}{D_{mG}\,n_G} \quad \text{(pounds-force)}$$

THRUST FORCE

$F_{aG} = F_{tG} \tan \phi_G \sin \gamma_G$

SEPARATING FORCE

$F_{sG} = F_{tG} \tan \phi_G \cos \gamma_G$

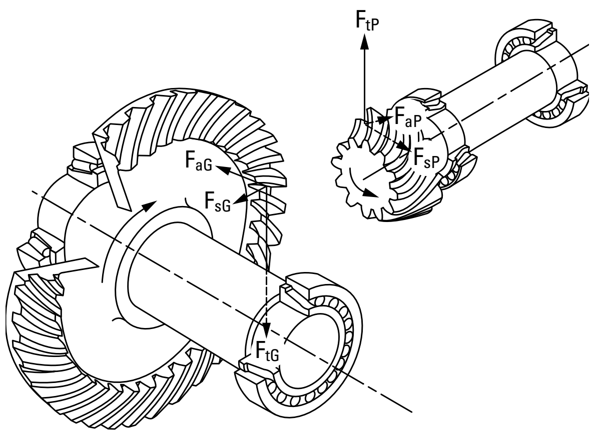


Fig. 39. Straight bevel gearing.

SPIRAL BEVEL AND HYPOID GEARING

In spiral bevel and hypoid gearing, the direction of the thrust and separating forces depends upon spiral angle, hand of spiral, direction of rotation, and whether the gear is driving or driven (see table 4). The hand of the spiral is determined by noting whether the tooth curvature on the near face of the gear (Fig. 40) inclines to the left or right from the shaft axis. Direction of rotation is determined by viewing toward the gear or pinion apex.

In spiral bevel gearing:

$F_{tP} = F_{tG}$

In hypoid gearing:

$$F_{tP} = \frac{F_{tG} \cos \psi_P}{\cos \psi_G}$$

Hypoid pinion effective working diameter:

$$D_{mP} = D_{mG} \left(\frac{N_p}{N_G}\right)\left(\frac{\cos \psi_G}{\cos \psi_P}\right)$$

TANGENTIAL FORCE

$$F_{tG} = \frac{(1.91 \times 10^7)\,H}{D_{mG}\,n_G} \quad \text{(newtons)}$$

$$= \frac{(1.26 \times 10^5)\,H}{D_{mG}\,n_G} \quad \text{(pounds-force)}$$

Hypoid gear effective working diameter:

$D_{mG} = D_{pG} - b \sin \gamma_G$

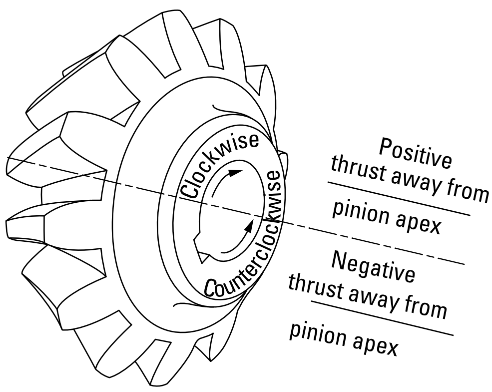


Fig. 40 Spiral bevel and hypoid gears – the direction of thrust and separating forces depends upon spiral angle, hand of spiral, direction of rotation, and whether the gear is driving or driven.

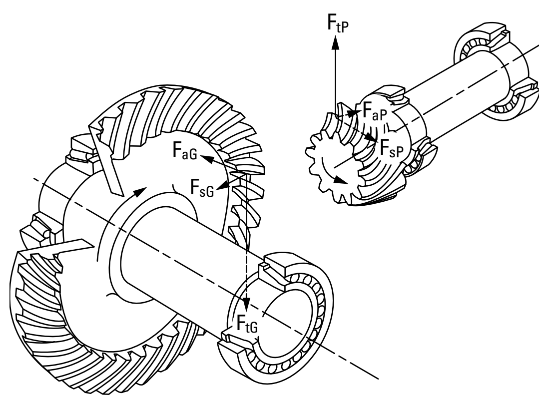


Fig. 41. Spiral bevel and hypoid gearing.

TABLE 3.
SPIRAL BEVEL AND HYPOID BEARING EQUATIONS

Driving Member Rotation	Thrust Force	Separating Force
Right hand spiral clockwise or Left hand spiral counterclockwise	Driving member: $F_{aP} = \frac{F_{tP}}{\cos\psi_P}(\tan\phi_P \sin\Upsilon_P - \sin\psi_P \cos\Upsilon_P)$	Driving member: $F_{sP} = \frac{F_{tP}}{\cos\psi_P}(\tan\phi_P \cos\Upsilon_P - \sin\psi_P \sin\Upsilon_P)$
	Driven member: $F_{aG} = \frac{F_{tG}}{\cos\psi_G}(\tan\phi_G \sin\Upsilon_G + \sin\psi_G \cos\Upsilon_G)$	Driven member: $F_{sG} = \frac{F_{tG}}{\cos\psi_G}(\tan\phi_G \cos\Upsilon_G - \sin\psi_G \sin\Upsilon_G)$
Right hand spiral counterclockwise or Left hand spiral clockwise	Driving member: $F_{aP} = \frac{F_{tP}}{\cos\psi_P}(\tan\phi_P \sin\Upsilon_P + \sin\psi_P \cos\Upsilon_P)$	Driving member: $F_{sP} = \frac{F_{tP}}{\cos\psi_P}(\tan\phi_P \cos\Upsilon_P + \sin\psi_P \sin\Upsilon_P)$
	Driven member: $F_{aG} = \frac{F_{tG}}{\cos\psi_G}(\tan\phi_G \sin\Upsilon_G - \sin\psi_G \cos\Upsilon_G)$	Driving member: $F_{sG} = \frac{F_{tG}}{\cos\psi_G}(\tan\phi_G \cos\Upsilon_G + \sin\psi_G \sin\Upsilon_G)$

STRAIGHT WORM GEARING

WORM

Tangential force

$$F_{tW} = \frac{(1.91 \times 10^7)\,H}{D_{PW}\,n_W} \quad \text{(newtons)}$$

$$= \frac{(1.26 \times 10^5)\,H}{D_{pW}\,n_W} \quad \text{(pounds-force)}$$

Thrust force

$$F_{aW} = \frac{(1.91 \times 10^7)\,H\,\eta}{D_{pG}\,n_G} \quad \text{(newtons)}$$

$$= \frac{(1.26 \times 10^5)\,H\,\eta}{D_{pG}\,n_G} \quad \text{(pounds-force)}$$

or

$$F_{aW} = \frac{F_{tW}\,\eta}{\tan\lambda}$$

Separating force

$$F_{sW} = \frac{F_{tW}\sin\phi}{\cos\phi\,\sin\lambda + \mu\cos\lambda}$$

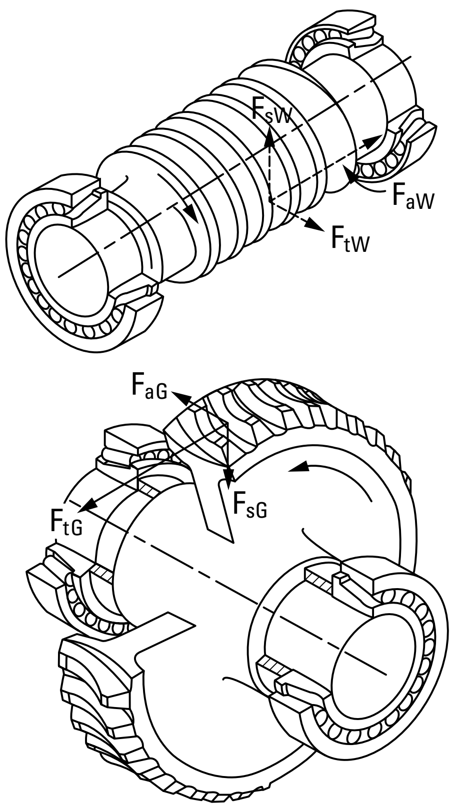


Fig. 42. Straight worm gearing.

WORM GEAR

TANGENTIAL FORCE

$$F_{tG} = \frac{(1.91 \times 10^7)\, H\, \eta}{D_{pG}\, n_G} \quad \text{(newtons)}$$

$$= \frac{(1.26 \times 10^5)\, H\, \eta}{D_{pG}\, n_G} \quad \text{(pounds-force)}$$

or

$$F_{tG} = \frac{F_{tW}\, \eta}{\tan \lambda}$$

THRUST FORCE

$$F_{aG} = \frac{(1.91 \times 10^7)\, H}{D_{pW}\, n_W} \quad \text{(newtons)}$$

$$= \frac{(1.26 \times 10^5)\, H}{D_{pW}\, n_W} \quad \text{(pounds-force)}$$

SEPARATING FORCE

$$F_{sG} = \frac{F_{tW} \sin \Phi}{\cos \Phi \sin \lambda + \mu \cos \lambda}$$

Where:

$$\lambda = \tan^{-1} \left(\frac{D_{pG}}{m\, D_{pW}} \right)$$

or

$$\lambda = \tan^{-1} \left(\frac{L_H}{\pi\, D_{pW}} \right)$$

and

$$\eta = \frac{\cos \Phi - \mu \tan \lambda}{\cos \Phi + \mu \cot \lambda}$$

METRIC SYSTEM

$$\mu^{(1)} = (5.34 \times 10^{-7})\, V_r^3 + \frac{0.146}{V_r^{0.09}} - 0.103$$

$$V_r = \frac{D_{pW}\, n_W}{(1.91 \times 10^4) \cos \lambda} \quad \text{(meters per second)}$$

INCH SYSTEM

$$\mu^{(1)} = (7 \times 10^{-14})\, V_r^3 + \frac{0.235}{V_r^{0.09}} - 0.103$$

$$V_r = \frac{D_{pW}\, n_W}{3.82 \cos \lambda} \quad \text{(feet per minute)}$$

(1)Approximate coefficient of friction for the 0.015 to 15 m/s (3 to 3000 ft/min) rubbing velocity range.

DOUBLE ENVELOPING WORM GEARING
WORM

TANGENTIAL FORCE

$$F_{tW} = \frac{(1.91 \times 10^7)\, H}{D_{mW}\, n_W} \quad \text{(newtons)}$$

$$= \frac{(1.26 \times 10^5)\, H}{D_{mW}\, n_W} \quad \text{(pounds-force)}$$

THRUST FORCE

$F_{aW} = 0.98\, F_{tG}$

Use this value for F_{tG} for bearing loading calculations on worm gear shaft. For torque calculations, use the following F_{tG} equations.

SEPARATING FORCE

$$F_{sW} = \frac{0.98\, F_{tG} \tan \Phi}{\cos \lambda}$$

WORM GEAR

TANGENTIAL FORCE

$$F_{tG} = \frac{(1.91 \times 10^7)\, H\, m\, \eta}{D_{pG}\, n_W} \quad \text{(newtons)}$$

$$= \frac{(1.26 \times 10^5)\, H\, m\, \eta}{D_{pG}\, n_W} \quad \text{(pounds-force)}$$

Use this value for calculating torque in subsequent gears and shafts. For bearing loading calculations, use the equation for F_{aW}.

or

$$F_{tG} = \frac{(1.91 \times 10^7)\, H\, \eta}{D_{pG}\, n_G} \quad \text{(newtons)}$$

$$= \frac{(1.26 \times 10^5)\, H\, \eta}{D_{pG}\, n_G} \quad \text{(pounds-force)}$$

THRUST FORCE

$$F_{aG} = \frac{(1.91 \times 10^7)\, H}{D_{mW}\, n_W} \quad \text{(newtons)}$$

$$= \frac{(1.26 \times 10^5)\, H}{D_{mW}\, n_W} \quad \text{(pounds-force)}$$

SEPARATING FORCE

$$F_{sG} = \frac{0.98\, F_{tG} \tan \Phi}{\cos \lambda}$$

Where:

η = efficiency (refer to manufacturer's catalog)

$D_{mW} = 2d_c - 0.98\, D_{pG}$

Lead angle at center of worm:

$$\lambda = \tan^{-1} \left(\frac{D_{pG}}{m\, D_{pW}} \right) = \tan^{-1} \left(\frac{L}{\pi\, D_{pW}} \right)$$

BELT AND CHAIN DRIVE FACTORS

Due to the variations of belt tightness as set by various operators, an exact equation relating total belt pull to tension F_1 on the tight side and tension F_2 on the slack side (Fig. 43) is difficult to establish. The following equation and Table 5 may be used to estimate the total pull from various types of belt and pulley, and chain and sprocket designs:

$$Fb = \frac{(1.91 \times 107)\ H\ f_B}{D_m\ n} \quad \text{(newtons)}$$

$$= \frac{(1.26 \times 10^5)\ H\ f_B}{D_m\ n} \quad \text{(pounds-force)}$$

Standard roller chain sprocket mean diameter.

$$D_m = \frac{P}{\sin\left(\frac{180}{N_s}\right)}$$

Where:

P = chain pitch

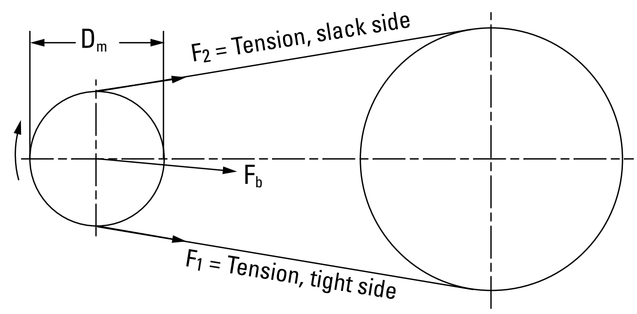


Fig. 43. Belt or chain drive.

TABLE 4.
BELT OR CHAIN PULL FACTOR
BASED ON 180 DEGREES ANGLE OF WRAP

Type	f_B
Chains, single	1.00
Chains, double	1.25
"V" belts	1.50

SHAFT ON TWO SUPPORTS

Simple beam equations are used to translate the externally applied forces on a shaft into bearing reactions acting at the bearing effective centers.

With two-row tapered roller and angular contact ball bearings, the geometric center of the bearing is considered to be the support point except where the thrust force is large enough to unload one row. Then the effective center of the loaded row is used as the point from which bearing load reactions are calculated. These approaches approximate the load distribution within a two-row bearing, assuming rigid shaft and housing. These are statically indeterminate problems in which shaft and support rigidity can significantly influence bearing loading and require the use of computer programs to solve.

SHAFT ON THREE OR MORE SUPPORTS

The equations of static equilibrium are insufficient to solve bearing reactions on a shaft having more than two supports. Such cases can be solved using computer programs if adequate information is available.

In such problems, the deflections of the shaft, bearings and housings affect the distribution of loads. Any variance in these parameters can significantly affect bearing reactions.

Bearing radial loads are determined by:

- Resolving forces applied to the shaft into horizontal and vertical components, relative to a convenient reference plane.
- Taking moments about the opposite support.
- Combining the horizontal and vertical reactions at each support into one resultant load.

Shown (on the next page) are equations for the case of a shaft on two supports with gear forces F_t (tangential), F_s (separating), and F_a (thrust), an external radial load F, and an external moment M. The loads are applied at arbitrary angles (1, 2, and 3) relative to the reference plane indicated in Fig. 44 on the following page. Using the principle of superposition, the equations for vertical and horizontal reactions (F_{rv} and F_{rh}) can be expanded to include any number of gears, external forces or moments. Use signs as determined from gear force equation.

Care should be used when doing this to ensure proper supporting degrees of freedom are used. That is, tapered roller bearings and ball bearings support radial loads, moment loads and axial loads in both directions.

BEARING REACTIONS

TABLE 5.
CALCULATION EQUATIONS

Symbols Used		
ae	Effective bearing spread	mm, in.
A, B, ...	Bearing position, used as subscripts	
$C_1, C_2, ...$	Linear distance (positive or negative)	mm, in.
DpG	Pitch diameter of the gear	mm, in.
F	Applied force	N, lbf
F_r	Radial bearing load	N, lbf
h	Horizontal (used as subscript)	
M	Moment	N-mm, lbf-in.
v	Vertical (used as subscript)	
1	Gear mesh angle relative to plane of reference defined in the figure below	deg, rad
2	Angle of applied force relative to plane of reference defined in the figure below	deg, rad
3	Angle of applied moment relative to plane of reference defined in the figure below	deg, rad

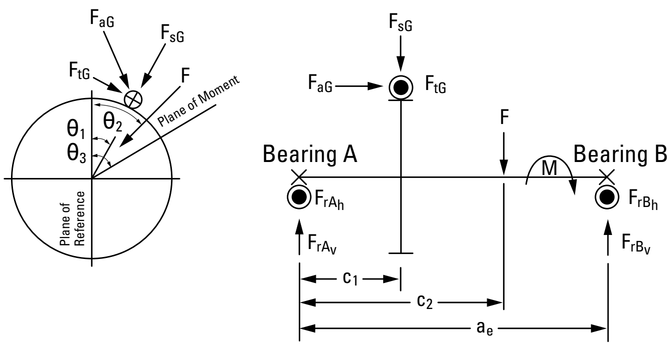


Fig. 44. Bearing radial reactions.

CENTRIFUGAL FORCE

Centrifugal force resulting from imbalance in a rotating member:

$$F_c = \frac{F_w\, r\, n^2}{8.94 \times 10^5} \quad \text{(newtons)}$$

$$= \frac{F_w\, r\, n^2}{3.52 \times 10^4} \quad \text{(pounds-force)}$$

SHOCK LOADS

It is difficult to determine the exact effect that shock loading has on bearing life. The magnitude of the shock load depends on the masses of the colliding bodies, their velocities, and deformations at impact.

The effect on the bearing depends on how much of the shock is absorbed between the point of impact and the bearings, as well as whether the shock load is great enough to cause bearing damage. It also is dependent on frequency and duration of shock loads.

At a minimum, a suddenly applied load is equivalent to twice its static value. It may be considerably more than this, depending on the velocity of impact.

Shock involves a number of variables that generally are not known or easily determined. Therefore, it is good practice to rely on experience. The Timken Company has years of experience with many types of equipment under the most severe loading conditions. Your Timken representative should be consulted on any application involving unusual loading or service requirements.

Vertical reaction component at bearing position B:

$$F_{rB_v} = \frac{1}{a_e}\left(c_1 (F_{sG} \cos \theta_1 + F_{tG} \sin \theta_1) + \frac{1}{2}(D_{pG} - b \sin \gamma_G)\, F_{aG} \cos \theta_1 + c_2\, F \cos \theta_2 + M \cos \theta_3 \right)$$

Horizontal reaction component at bearing position B:

$$F_{rB_h} = \frac{1}{a_e}\left(c_1 (F_{sG} \sin \theta_1 - F_{tG} \cos \theta_1) + \frac{1}{2}(D_{pG} - b \sin \gamma_G)\, F_{aG} \sin \theta_1 + c_2\, F \sin \theta_2 + M \sin \theta_3 \right)$$

Vertical reaction component at bearing position A:

$$F_{rA_v} = F_{sG} \cos \theta_1 + F_{tG} \sin \theta_1 + F \cos \theta_2 - F_{rB_v}$$

Horizontal reaction component at bearing position A:

$$F_{rA_h} = F_{sG} \sin \theta_1 - F_{tG} \cos \theta_1 + F \sin \theta_2 - F_{rB_h}$$

Resultant radial reaction: $F_{rA} = \left[(F_{rA_v})^2 + (F_{rA_h})^2\right]^{1/2}$ $F_{rB} = \left[(F_{rB_v})^2 + (F_{rB_h})^2\right]^{1/2}$

Resultant axial reaction: $F_{aA} = F_{aG}$ (fixed position) $F_{aB} = 0$ (float position)

LOAD RATINGS

The basic dynamic load rating and the static load rating are commonly used for bearing selection. The basic dynamic load rating is used to estimate life of a rotating bearing. Static load ratings are used to determine the maximum permissible load that can be applied to a non-rotating bearing.

The basic philosophy of The Timken Company is to provide the most realistic bearing rating to assist our customers in the bearing selection process. Published ratings for Timken bearings include the basic dynamic radial load ratings, C1, for tapered roller bearings, and Ce for ball bearings. These values are based on a basic rating life of one million revolutions. Timken tapered roller bearings also include the basic dynamic load rating C90, which is based on a basic rating life of ninety million revolutions. The basic static radial load rating is Co.

STATIC LOAD RATING

The basic static radial load ratings for Timken bearings are based on a maximum contact stress within a non-rotating bearing of 4000 MPa (580 ksi) for roller bearings and 4200 MPa (609 ksi) for ball bearings, at the center of contact on the most heavily loaded rolling element.

The 4000 MPa (580 ksi) or 4200 MPa (609 ksi) stress levels may cause visible light Brinell marks on the bearing raceways. This degree of marking will not have a measurable effect on fatigue life when the bearing is subsequently rotating under a lower-application load. If sound, vibration or torque are critical, or if a pronounced shock load is present, a lower load limit should be applied.

TABLE 6.
HARDNESS FACTORS TO MODIFY BASIC STATIC LOAD RATING

Raceway Hardness HRC	Hardness Factor HFs
58	1.00
57	1.06
56	1.13
55	1.21
54	1.29
53	1.37
52	1.46
51	1.55
50	1.65
49	1.76
48	1.88
47	2.00
46	2.13
45	2.27
44	2.41
43	2.57
42	2.74
41	2.92
40	3.10

When the loading is static, it is usually suggested that the applied load be no greater than the basic static load rating divided by the appropriate hardness factor (HFs) as shown in Table 6.

For more information on selecting a bearing for static load conditions, consult your Timken representative.

STATIC RADIAL AND/OR AXIAL EQUIVALENT LOADS

The static equivalent radial and/or axial loading is dependent on the bearing type selected. For bearings designed to accommodate only radial or axial loading, the static equivalent load is equivalent to the applied load.

For all bearings, the maximum contact stress can be approximated using the static equivalent load and the static rating.

For roller bearings:

$$\sigma_0 = 4000 \times \left(\frac{P_0}{C_0}\right)^{1/2} \text{ MPa}$$

$$\sigma_0 = 580 \times \left(\frac{P_0}{C_0}\right)^{1/2} \text{ ksi}$$

For ball bearings:

$$\sigma_0 = 4200 \times \left(\frac{P_0}{C_0}\right)^{1/3} \text{ MPa}$$

$$\sigma_0 = 609 \times \left(\frac{P_0}{C_0}\right)^{1/3} \text{ ksi}$$

Radial ball bearings

The dynamic equivalent radial load is used for comparison with the static load rating.

$$P_{0r} = 1/2\ C_0$$

Thrust ball bearings

Similar to radial ball bearings, thrust ball bearings use the same equation for equivalent static and dynamic loading.

$$P_{0a} = XF_r + YF_a$$

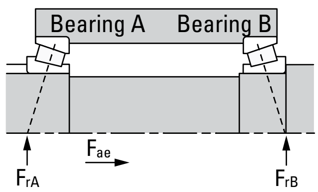
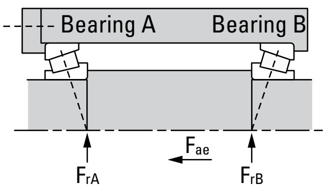
The X and Y factors are listed later in this section along with the minimum required axial load-to-radial load ratio for maintaining proper operation.

Tapered roller bearings

To determine the static equivalent radial load for a single-row mounting, first determine the axial load (F_a), then use the following equations, depending on the appropriate axial load condition.

Where: F_r = applied radial load
F_a = net bearing axial load. F_{aA} and F_{aB} calculated from equations

TABLE 7.
STATIC EQUIVALENT LOAD EQUATIONS, SINGLE-ROW TAPERED ROLLER BEARING

Design	Thrust Condition	Axial Load	Static Equivalent Radial Load
	$\frac{0.47\ F_{rA}}{K_A} \le \frac{0.47\ F_{rB}}{K_B} + F_{ae}$	$F_{aA} = \frac{0.47\ F_{rB}}{K_B} + F_{ae}$ $F_{aB} = \frac{0.47\ F_{rB}}{K_B}$	$P_{0B} = F_{rB}$ for $F_{aA} < 0.6\ F_{rA} / K_A$ $P_{0A} = 1.6\ F_{rA} - 1.269\ K_A\ F_{aA}$ for $F_{aA} < 0.6\ F_{rA} / K_A$ $P_{0A} = 0.5\ F_{rA} + 0.564\ K_A\ F_{aA}$
 Design (external axial load, F_{ae}, onto bearing A)	$\frac{0.47\ F_{rA}}{K_A} > \frac{0.47\ F_{rB}}{K_B} + F_{ae}$	$F_{aA} = \frac{0.47\ F_{rA}}{K_A}$ $F_{aB} = \frac{0.47\ F_{rA}}{K_A} - F_{ae}$	for $F_{aB} < 0.6\ F_{rB} / K_B$ $P_{0B} = 0.5\ F_{rB} + 0.564\ K_B\ F_{aB}$ for $F_{aB} < 0.6\ F_{rB} / K_B$ $P_{0B} = 1.6\ F_{rB} - 1.269\ K_B\ F_{aB}$ $P_{0A} = F_{rA}$

Use the values of PO calculated for comparison with the static rating (CO), even if PO is less than the radial applied Fr.

Static Equivalent Radial Load (Two-Row Bearings)

The bearing data tables do not include static rating for two-row bearings. The two-row static radial rating can be estimated as:

$$C_{o(2)} = 2C_o$$

Where:

$C_{o(2)}$ = two-row static radial rating

C_o = static radial load rating of a single-row bearing, type TS, from the same series.

Dynamic Equivalent Radial Bearing Loads (P_r)

To calculate the L10 life, it is necessary to calculate a dynamic equivalent radial load, designated by P_r. The dynamic equivalent radial load is defined as a single radial load that, if applied to the bearing, will result in the same life as the combined loading under which the bearing operates.

$$P_r = XF_r + YF_a$$

Where:

P_r = dynamic equivalent radial load
F_r = applied radial load
F_a = applied axial load
X = radial load factor
Y = axial load factor

Tapered roller bearings use the equations based on the number of rows and type of mounting utilized. For single-row bearings in direct or indirect mounting, the figure on page 31 can be used based on the direction of the externally applied axial load. Once the appropriate design is chosen, review the table and check the thrust condition to determine which axial load and dynamic equivalent radial load calculations apply. For ball bearings, the dynamic equivalent radial load can be found in the table below.

TABLE 8.
DYNAMIC EQUIVALENT LOAD EQUATIONS

Bearing Description (ref.)	Contact Angle	Single-Row and Tandem Mountings	Double-Row and Preload Pair Mountings
Bearing type and or series		$K_T = \frac{F_a}{(\#\ of\ bearings) \times C_o}$	$K_T = \frac{F_a}{C_o}$
Radial type ball bearings – use larger of resulting "P_r" value[1]			
MM9300K MM9100K MM200K MM300K	0°	$P_r = F_r$ or $P_r = 0.56F_r + Y_1F_a$	$P_r = F_r + 1.20Y_1F_a$ or $P_r = 0.78F_r + 1.625Y_1F_a$
Angular contact ball bearings – use larger of resulting "P_r" value			
2MMV9300WI 2MMV9300HX 2MM9300WI 2MV9300WI 2MMV9100WI 2MMV9100HX 2MM99100WI 2MM9100WI 2MV9100WI 2MV200WI 2MMV200WI 2MMV300WI 2MM200WI 2MM300WI 2MV300WI	15°	$P_r = F_r$ or $P_r = 0.44F_r + Y_2F_a$	$P_r = F_r + 1.124Y_2F_a$ or $P_r = 0.72F_r + 1.625Y_2F_a$
2MM9100WO	15°	$P_r = F_r$ or $P_r = 0.44F_r + Y_3F_a$	$P_r = F_r + 1.124Y_3F_a$ or $P_r = 0.72F_r + 1.625Y_3F_a$
3MMV9300WI 3MMV9300HX 3MM9300WI 3MV9300WI 3MMV9100WI 3MMV9100HX 3MM9100WI 3MV9100WI 3MM99100WI 3MMV200WI 3MV200WI 3MM200WI 3MM300WI 3MV300WI 3MV300WI	25°	$P_r = F_r$ or $P_r = 0.41F_r + 0.87F_a$	$P_r = F_r + 0.92F_a$ or $P_r = 0.67F_r + 1.41F_a$

(1)If $P_r > C_o$ or $P_r > 1/2$ Ce consult with your Timken representative on Life Calculations.

TABLE 9.
THE REQUIRED Y FACTORS FOR BALL BEARINGS

K_T	Y_1	Y_2	Y_3
0.015	2.30	1.47	1.60
0.020	2.22	1.44	1.59
0.025	2.10	1.41	1.57
0.030	2.00	1.39	1.56
0.040	1.86	1.35	1.55
0.050	1.76	1.32	1.53
0.060	1.68	1.29	1.51
0.080	1.57	1.25	1.49
0.100	1.48	1.21	1.47
0.120	1.42	1.19	1.45
0.150	1.34	1.14	1.42
0.200	1.25	1.09	1.39
0.250	1.18	1.05	1.35
0.300	1.13	1.02	1.33
0.400	1.05	1.00	1.29
0.500	1.00	1.00	1.25
0.600	—	—	1.22
0.800	—	—	1.17
1.000	—	—	1.13
1.200	—	—	1.10

Dynamic Equivalent Axial Bearing Loads (P_a)

For thrust ball and thrust tapered roller bearings, the existence of radial loads introduces complex load calculations that must be carefully considered. If radial load is zero, the dynamic equivalent axial load (P_a) will be equal to the applied axial load (F_a). If any radial load is expected in the application, consult your Timken representative for advice on bearing selection.

For thrust angular contact ball bearings, the dynamic equivalent axial load is determined by:

$$P_a = XF_r + YF_a$$

When: $\frac{F_a}{F_r} < 2.17$ $X = 1.90$ $Y = 0.54$

When: $\frac{F_a}{F_r} > 2.17$ $X = 0.92$ $Y = 1.00$

If: $\frac{F_a}{F_r} \leq 2.17$ then bearings with a contact angle $< 60^\circ$ should be considered.

BEARING EQUIVALENT LOADS AND REQUIRED RATINGS FOR TAPERED ROLLER BEARINGS

Tapered roller bearings are ideally suited to carry all types of loads – radial, axial, and any combination of both. Due to the tapered design of the bearing, a radial load will induce an axial reaction within the bearing that must be opposed by an equal or greater axial load to keep the bearing cone and cup from separating. The ratio of the radial to the axial load and the bearing included cup angle determine the load zone in a given bearing. The number of rollers in contact as a result of this ratio determines the load zone in the bearing. If all the rollers are in contact, the load zone is referred to as being 360 degrees.

When only radial load is applied to a tapered roller bearing, for convenience it is assumed in using the traditional calculation method that half the rollers support the load – the load zone is 180 degrees. In this case, induced bearing axial load is:

$$F_{a(180)} = \frac{0.47}{K}$$

The equations for determining bearing axial reactions and equivalent radial loads in a system of two single-row bearings are based on the assumption of a 180-degree load zone in one of the bearings and 180 degrees or more in the opposite bearing.

Dynamic Equivalent Radial Load

The basic dynamic radial load rating, C_{90}, is assumed to be the radial load-carrying capacity with a 180-degree load zone in the bearing. When the axial load on a bearing exceeds the induced thrust, $F_{a(180)}$, a dynamic equivalent radial load must be used to calculate bearing life.

The dynamic equivalent radial load is that radial load which, if applied to a bearing, will give the same life as the bearing will attain under the actual loading.

The equations presented give close approximations of the dynamic equivalent radial load assuming a 180-degree load zone in one bearing and 180 degrees or more in the opposite bearing.

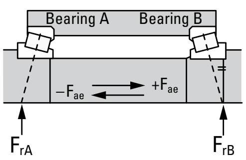
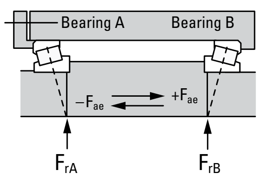
Tapered roller bearings use the equations based on the number of rows and type of mounting utilized. For single-row bearings in direct or indirect mounting, the following table can be used based on the direction of the externally applied axial load. Once the appropriate design is chosen, review the table and check the thrust condition to determine which axial load and dynamic equivalent radial load calculations apply.

ALTERNATE APPROACH FOR DETERMINING DYNAMIC EQUIVALENT RADIAL LOADS

The following is a general approach to determining the dynamic equivalent radial loads. Here, a factor "m" has to be defined as +1 for direct-mounted single-row or two-row bearings, or -1 for indirect mounted bearings. Also, a sign convention is necessary for the external axial load F_{ae} as follows:

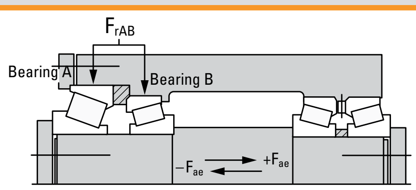
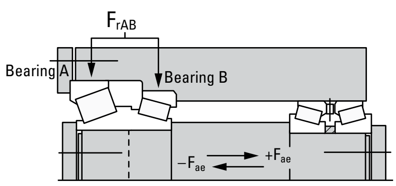
- In case of external axial load applied to the shaft (typical rotating cone application), F_{ae} to the right is positive; to the left is negative.
- When external axial load is applied to the housing (typical rotating cup application), F_{ae} to the right is negative; to the left is positive.

TABLE 10.
DYNAMIC EQUIVALENT RADIAL LOAD EQUATIONS, SINGLE-ROW TAPERED ROLLER BEARING MOUNTING

Design	Thrust Condition	Axial Load	Dynamic Equivalent Radial Load
Indirect Mounting (m = -1) 	$\frac{0.47 \times F_{rA}}{K_A} \leq \frac{0.47 \times F_{rB}}{K_B} - m\ F_{ae}$	$F_{aA} = \frac{0.47 \times F_{rB}}{K_B} - m\ F_{ae}$ $F_{aB} = \frac{0.47 \times F_{rB}}{K_B}$	$P_A = 0.4\ F_{rA} + K_A\ F_{aA}$ $P_B = F_{rB}$
Direct Mounting (m = +1) 	$\frac{0.47 \times F_{rA}}{K_A} > \frac{0.47 \times F_{rB}}{K_B} - m\ F_{ae}$	$F_{aA} = \frac{0.47 \times F_{rA}}{K_A}$ $F_{aB} = \frac{0.47 \times F_{rA}}{K_A} + m\ F_{ae}$	$P_A = F_{rA}$ $P_B = 0.4\ F_{rB} + K_B\ F_{aB}$

If $P_A < F_{rA}$, use $P_A = F_{rA}$ or if $P_B < F_{rB}$, use $P_B = F_{rB}$.

TABLE 11.
DYNAMIC EQUIVALENT RADIAL LOAD EQUATIONS, TWO-ROW TAPERED ROLLER BEARING MOUNTING – FIXED BEARING WITH EXTERNAL AXIAL LOAD, F_{ae} (SIMILAR OR DISSIMILAR SERIES)

Design	Thrust Condition	Dynamic Equivalent Radial Load
 Fixed Bearing Indirect Mounting (m = -1)	$F_{ae} \leq \frac{0.6 \times F_{rAB}}{K^{(1)}}$	$P_A = \frac{K_A}{K_A + K_B}(F_{rAB} - 1.67\ m\ K_B\ F_{ae})$ $P_B = \frac{K_B}{K_A + K_B}(F_{rAB} + 1.67\ m\ K_A\ F_{ae})$
 Fixed Bearing Direct Mounting (m = +1)	$F_{ae} > \frac{0.6 \times F_{rAB}}{K^{(1)}}$	$P_A = 0.4\ F_{rAB} - m\ K_A\ F_{ae}$ $P_B = 0.4\ F_{rAB} + m\ K_B\ F_{ae}$

(1)If "m F_{ae}" is positive, $K = K_A$; If "m F_{ae}" is negative, $K = K_B$.

F_{rAB} is the radial load on the two-row assembly. The single-row basic dynamic radial load rating, C_{90}, is to be applied when calculating life based on the above equations.

BEARING LIFE AND SYSTEM LIFE

Many different performance criteria exist that dictate how a bearing should be selected. These include bearing fatigue life, rotational precision, power requirements, temperature limits, speed capabilities, sound, etc. This publication deals primarily with bearing life as related to material-associated fatigue. Bearing life is defined here as the length of time, or number of revolutions, until a fatigue spall of 6.0 mm^2 (0.01 in.^{2}) develops. Since metal fatigue is a statistical phenomenon, the life of an individual bearing is impossible to predetermine precisely. Bearings that may appear to be identical can exhibit considerable life scatter when tested under identical conditions. Thus it is necessary to base life predictions on a statistical evaluation of a large number of bearings operating under similar conditions. The Weibull distribution function is commonly used to predict the life of a population of bearings at any given reliability level.

RATING LIFE

Rating life, (L_{10}), is the life that 90 percent of a group of apparently identical bearings will complete or exceed before a fatigue spall develops. The L_{10} life also is associated with 90 percent reliability for a single bearing under a certain load.

BEARING LIFE EQUATIONS

Traditionally, the L_{10} life has been calculated as follows for bearings under radial or combined loading where the dynamic equivalent radial load, (P_r), has been determined:

$$L_{10} = \left(\frac{C}{P_r}\right)^e \times (1\text{x}10^6) \qquad \text{revolutions}$$

or,

$$L_{10} = \left(\frac{C}{P_r}\right)^e \times \left(\frac{1\text{x}10^6}{60n}\right) \qquad \text{hours}$$

For thrust bearings, the above equations change to the following.

$$L_{10} = \left(\frac{C_a}{P_a}\right)^e \times (1\text{x}10^6) \qquad \text{revolutions}$$

or,

$$L_{10} = \left(\frac{C_a}{P_a}\right)^e \times \left(\frac{1\text{x}10^6}{60n}\right) \qquad \text{hours}$$

e = 3 for ball bearings
= $^{10}/_3$ for roller bearings

Tapered roller bearings often use a dynamic load rating based on ninety million cycles, as opposed to one million cycles, changing the equations as follows.

$$L_{10} = \left(\frac{C_{90}}{P_r}\right)^{10/3} \times (90\text{x}10^6) \qquad \text{revolutions}$$

or,

$$L_{10} = \left(\frac{C_{90}}{P_r}\right)^{10/3} \times \left(\frac{90\text{x}10^6}{60n}\right) \qquad \text{hours}$$

and

$$L_{10} = \left(\frac{C_{a90}}{P_a}\right)^{10/3} \times (90\text{x}10^6) \qquad \text{revolutions}$$

or,

$$L_{10} = \left(\frac{C_{a90}}{P_a}\right)^{10/3} \times \left(\frac{90\text{x}10^6}{60n}\right) \qquad \text{hours}$$

As the first set of equations for radial bearings with dynamic ratings based on one million revolutions is the most common form of the equations, this will be used through the rest of this section. The dynamic equivalent load equations and the life adjustment factors are applicable to all forms of the life equation.

With increased emphasis on the relationship between the reference conditions and the actual environment in which the bearing operates in the machine, the traditional life equations have been expanded to include certain additional variables that affect bearing performance. The approach whereby these factors, including a factor for useful life, are considered in the bearing analysis and selection has been termed Bearing Systems Analysis (BSA).

The ISO/ABMA adjusted bearing life equation is:

$$L_{na} = a_1 a_2 a_3 L_{10}$$

Where:

- a_1 = reliability life factor
- a_2 = material life factor
- a_3 = operating condition life factor (to be specified by the manufacturer)

The Timken adjusted bearing life equation is:

$$L_{na} = a_1 a_2 a_{3d} a_{3h} a_{3k} a_{3l} a_{3m} a_{3p} \left(\frac{C}{P_r}\right)^e (1\text{x}10^6) \qquad \text{revolutions}$$

Where:

- a_1 = reliability life factor
- a_2 = material life factor
- a_{3d} = debris life factor
- a_{3h} = hardness life factor
- a_{3k} = load zone life factor
- a_{3l} = lubrication life factor
- a_{3m} = misalignment life factor
- a_{3p} = low load life factor
- C = dynamic radial load rating
- P_r = dynamic equivalent radial load
- e = 3 for ball bearings
  = 10/3 for roller bearings

RELIABILITY LIFE FACTOR (a_1)

The equation for the life adjustment factor for reliability is:

$$a_1 = 4.26 \times \left(\ln \frac{100}{R} \right)^{2/3} + 0.05$$

ln = natural logarithm (base e)

To adjust the calculated L_{10} life for reliability, multiply by the a_1 factor. If 90 (90 percent reliability) is substituted for R in the above equation, a_1 = 1. For R = 99 (99 percent reliability), a_1 = 0.25. The following table lists the reliability factor for commonly used reliability values.

TABLE 12.
RELIABILITY FACTORS

R (percent)	L_n	a_1
90	L_{10}	1.00
95	L_5	0.64
96	L_4	0.55
97	L_3	0.47
98	L_2	0.37
99	L_1	0.25
99.5	$L_{0.5}$	0.175
99.9	$L_{0.1}$	0.093

Note that the equation for reliability adjustment assumes there is a short minimum life below which the probability of bearing damage is minimal (e.g., zero probability of bearing damage producing a short life). Extensive bearing fatigue life testing has shown the minimum life, below which the probability of bearing damage is negligible, to be larger than shown above. For a more accurate prediction of bearing lives at high levels of reliability, consult your Timken representative.

MATERIAL LIFE FACTOR (a_2)

The life adjustment factor for bearing material (a_2) for standard Timken bearings manufactured from bearing-quality steel is 1.0. Bearings also are manufactured from premium steels, containing fewer and smaller inclusion impurities than standard steels and providing the benefit of extending bearing fatigue life (e.g., DuraSpexx™). Application of the material life factor requires that fatigue life is limited by nonmetallic inclusions, that contact stresses are approximately less than 2400 MPa (350 ksi), and adequate lubrication is provided. It is important to note that improvements in material cannot offset poor lubrication in an operating bearing system. Consult your Timken representative for applicability of the material factor.

DEBRIS LIFE FACTOR (a_{3d})

Debris within a lubrication system reduces the life of a rolling bearing by creating indentations on the contacting surfaces, leading to stress risers. The Timken life rating equations were developed based on test data obtained with 0.040 mm (0.00157 in.) oil filtration and measured ISO cleanness levels of approximately 15/12, which is typical of cleanness levels found in normal industrial machinery. When more or less debris is present within the system, the fatigue life predictions can be adjusted according to the measured or expected ISO lubricant cleanness level to more accurately reflect the expected bearing performance.

As opposed to determining the debris life factor based on filtration and ISO cleanness levels, a Debris Signature Analysis™ can be performed for more accurate bearing performance predictions. The Debris Signature Analysis is a process for determining the effects of the actual debris present in your system on the bearing performance. The typical way in which this occurs is through measurements of dented/bruised surfaces on actual bearings run in a given application. This type of analysis can be beneficial because different types of debris cause differing levels of performance, even when they are of the same size and amount in the lubricant. Soft, ductile particles can cause less performance degradation than hard, brittle particles. Hard, ductile particles are typically most detrimental to bearing life. Brittle particles can break down, thus not affecting performance to as large a degree as hard ductile particles. For more information on Debris Signature Analysis or the availability of Debris Resistant bearings for your application, consult your Timken representative.

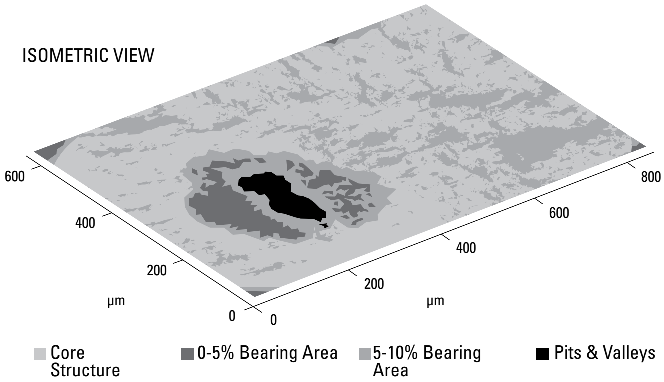


Fig. 45. Surface map of a bearing raceway with debris denting.

HARDNESS LIFE FACTOR (a_{3h})

For Timken bearings in this catalog, supplied as a full assembly, the hardness life factor will be one [1].

LOAD ZONE LIFE FACTOR (a_{3k})

The fatigue life of a bearing is a function of the stresses in rolling elements and raceways and the number of stress cycles that the loaded bearing surfaces experience in one bearing revolution. The stresses depend on applied load and on how many rolling elements support that load. The number of stress cycles depends on bearing geometry and, again, on how many rolling elements support the load. Therefore, life for a given external load is related to the loaded arc or load zone of the bearing.

The load zone in a bearing is dominated by the internal clearance, either radial or axial depending on the bearing type. Without considering preload, less clearance in a bearing results in a larger load zone and subsequently longer bearing life.

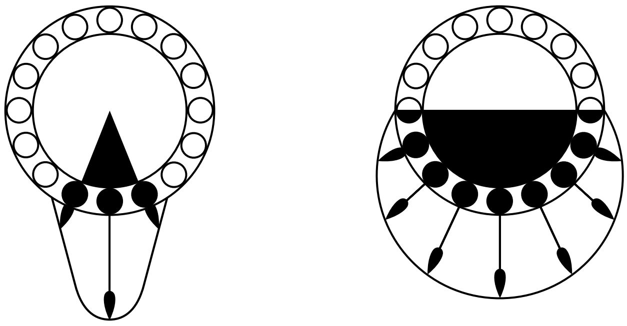


Fig. 46. Bearing load zones and rolling element-raceway contact loading.

Using the dynamic equivalent load (P_r) instead of the applied radial load (F_r) in the equation for L_{10a} roughly approximates the load zone factor for combined loading only. If a more accurate assessment of the load zone adjusted life is necessary (e.g., including the effects of internal clearance or fitting practice), consult your Timken representative.

LUBRICATION LIFE FACTOR (a_{3l})

The influence of lubrication film due to elastohydrodynamic (EHL) lubrication on bearing performance is related to the reduction or prevention of asperity (metal-metal) contact between the bearing surfaces. Extensive testing was done at the Timken Technology Center to quantify the effects of the lubrication-related parameters on bearing life. It was found that the rolling element and raceway surface finish, relative to lubricant film thickness, has the most notable effect on improving bearing performance. Factors such as bearing geometry, material, loads and load zones also play an important role in bearing performance.

The following equation provides a method to calculate the lubrication factor for a more accurate prediction of the influence of lubrication on tapered roller bearing life (L_{10a}). For more information on calculating this factor for ball bearings, consult your Timken representative.

$$a_{3l} = C_g \times C_l \times C_j \times C_s \times C_v \times C_{gr}$$

Where:

C_g = geometry factor
C_l = load factor
C_j = load zone factor
C_s = speed factor
C_v = viscosity factor
C_{gr} = grease lubrication factor

The a_{3l} maximum is 2.88 for all bearings. The a_{3l} minimum is 0.200 for case-hardened bearings and 0.126 for through-hardened bearings.

A lubricant contamination factor is not included in the lubrication factor because Timken endurance tests are typically run with a 0.040 mm (0.00157 in.) filter to provide a realistic level of lubricant cleanness for most applications.

Geometry Factor (C_g)

C_g is given for most tapered roller bearing part numbers in the appendix. The geometry factor also includes the material effects and load zone considerations. It should be noted that the primary effect of the load zone is on rolling element load distributions and contact stresses within the bearing, which are not quantified within the lubrication factor. Refer to the previous section, Load Zone Life Factor (a_{3k}), for more information.

LOAD FACTOR (C_l)

The C_l factor is obtained from the following figure. Note that the factor is different based on the type of bearing utilized. P_r is the equivalent load applied to the bearing in Newtons and is determined in the Equivalent Bearing Loads (P_r) section.

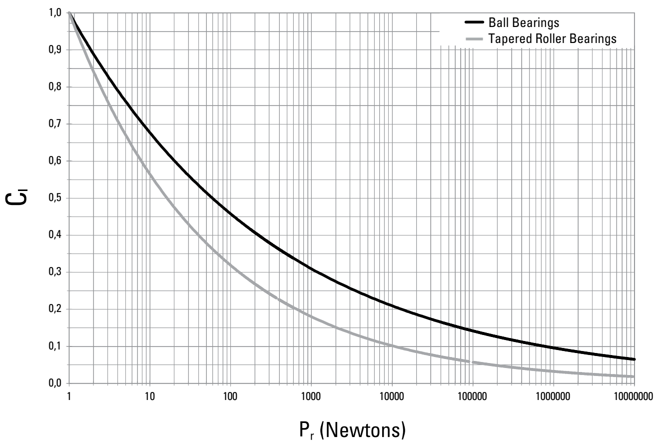


Fig. 47. Load factor (C_l) vs. equivalent bearing load (P_r).

LOAD ZONE FACTOR (C_j)

As mentioned previously, for all non-tapered roller bearings the load zone factor is one [1]. For tapered roller bearings, the load zone factor can be taken from the graph based on the axial load applied to that bearing.

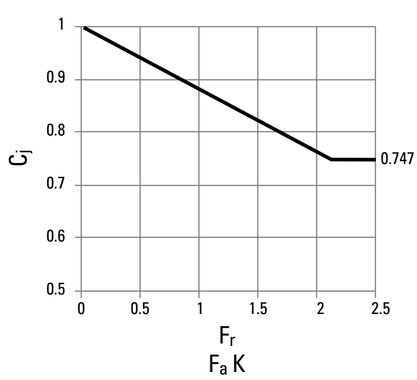


Fig. 48. Load factor (C_j) vs. tapered bearing axial load (F_a).

SPEED FACTOR (C_s)

C_s is determined from the following figure, where rev/min (RPM) is the rotational speed of the inner ring relative to the outer ring.

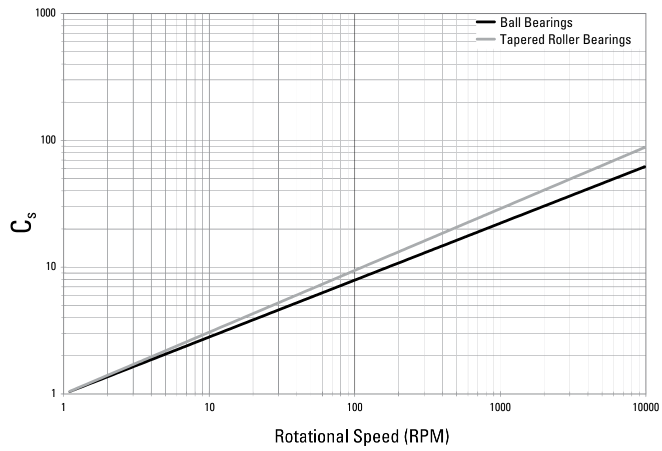


Fig. 49. Speed factor (C_s) vs. rotational speed.

VISCOSITY FACTOR (C_v)

The lubricant kinematic viscosity (centistokes [cSt]) is taken at the operating temperature of the bearing. The operating viscosity can be estimated by Fig. 50. The viscosity factor (C_v) can then be determined from Fig. 51 on the following page.

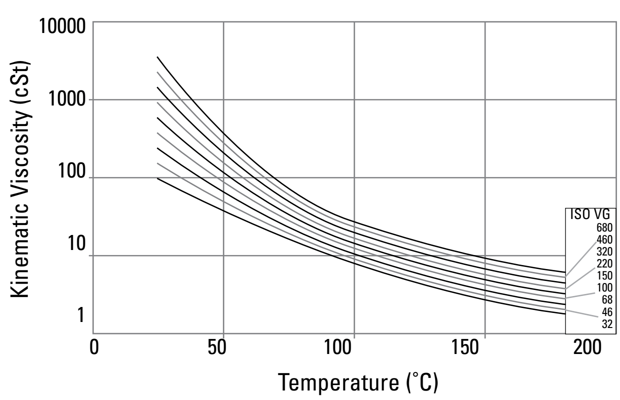


Fig. 50. Temperature vs. kinematic viscosity.

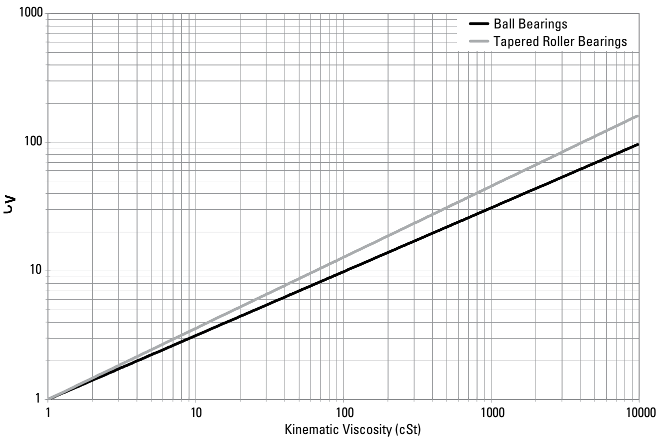


Fig. 51. Viscosity factor (C_v) vs. kinematic viscosity.

GREASE LUBRICATION FACTOR (C_{gr})

For grease lubrication, the EHL lubrication film becomes depleted of oil over time and is reduced in thickness. Consequently, a reduction factor (C_{gr}) should be used to adjust for this effect.

$$C_{gr} = 0.79$$

MISALIGNMENT LIFE FACTOR (a_{3ma})

The life of the bearing depends on the magnitude of the angle of misalignment, on the internal bearing geometry, and on the applied loads.

Accurate alignment of the shaft relative to the housing is critical for best performance. The life prediction using the method defined in this publication is relatively accurate up to the limits listed within, based on bearing type. The base condition, for which the load rating of the roller bearing is defined, is 0.0005 radians misalignment.

Performance of all Timken bearings under various levels of misalignment and radial and axial load can be predicted using sophisticated computer programs. Using these programs, Timken engineers can design special bearing-contact profiles to accommodate the conditions of radial load, axial load and/or bearing misalignment in your application. Consult your Timken representative for more information.

LOW LOAD LIFE FACTOR (a_{3p})

Bearing life tests at the Timken Technology Center have shown greatly extended bearing fatigue life performance is achievable when the bearing contact stresses are low and the lubricant film is sufficient to fully separate the micro-scale textures of the contacting surfaces. Mating the test data with sophisticated computer programs for predicting bearing performance, Timken engineers developed a low-load factor for use in the catalog to predict the life increase expected when operating under low-bearing loads. The following figure shows the low-load factor (a_{3p}) as a function of the lubricant life factor (a_{3l}) and the ratio of bearing dynamic rating to the bearing equivalent load.

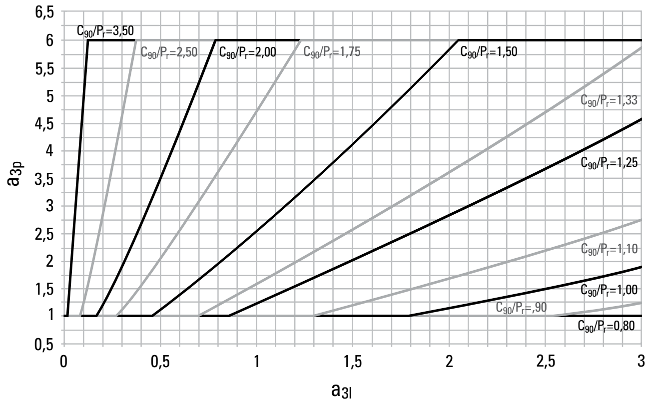


Fig. 52. Low load factor (a_{3P}) vs. lubricant life factor (a_{3l}) and C_{90}/P_r ratio.

SYSTEM LIFE AND WEIGHTED AVERAGE LOAD AND LIFE

SYSTEM LIFE

System reliability is the probability that all of the given bearings in a system will attain or exceed some required life. System reliability is the product of the individual bearing reliabilities in the system:

$$R_{(system)} = R_A R_B R_C \ldots R_n$$

In the application, the L_{10} system life for a number of bearings each having different L_{10} life is:

$$L_{10}\,(system) = [(1/L_{10A})^{3/2} + (1/L_{10B})^{3/2} + \ldots(1/L_{10n})^{3/2}]^{-2/3}$$

WEIGHTED AVERAGE LIFE AND LOAD EQUATIONS

In many applications, bearings are subjected to various conditions of loading and bearing selection is often made on the basis of maximum load and speed. However, under these conditions a more meaningful analysis may be made examining the loading cycle to determine the weighted average load.

Bearing selection based on weighted average loading will take into account variations in speed, load, and proportion of time during which the variable loads and speed occur. However, it is still necessary to consider extreme loading conditions to evaluate bearing contact stresses and alignment.

Weighted Average Load

Variable speed, load and proportion time:

$$F_{wt} = [(n_1 T_1 F_1^{\ 10/3} + \ldots n_n T_n F_n^{\ 10/3}) / n_a]^{0.3}$$

Where during each condition in a load cycle:

- T = proportion of total time
- F = load applied
- n = RPM
- n_a = reference speed of rotation for use in bearing life equations. For convenience, 500 RPM is normally used by Timken.

Uniformly increasing load, constant speed:

$$F_w = [(3/13)\,(F_{max}^{\ 13/3} - F_{min}^{\ 13/3}) / (F_{max} - F_{min})]^{0.3}$$

Where, during a load cycle:

- F_{max} = maximum applied load
- F_{min} = minimum applied load

The above formulae does not allow the use of the life-modifying factor for lubrication a3l, except in the case of constant speed. Therefore, when these equations are used in the bearing selection process, the design L10 bearing life should be based on a similar successful machine that operates in the same environment. Life calculations for both machines must be performing on the same basis. To allow for varying lubrication conditions in a load cycle, it is necessary to perform the weighted average life calculation.

Weighted Average Life

$$L_{nwt} = 1/\{[T_1 / (L_n)_1] + [T_2 / (L_n)_2] + \ldots [T_n / (L_n)_n]\}$$

Where, during a load cycle:

- T = proportion of total time
- L_n = calculated rating life for each condition

RATIOS OF BEARING LIFE TO LOADS, POWER AND SPEEDS

In applications subjected to variable conditions of loading, bearing life is calculated for one condition. Life for any other condition can easily be calculated by taking the ratio of certain variables. To use these ratios, the bearing load must vary proportionally with power, speed, or both. Nevertheless, this applies only to "catalog" lives or adjusted lives by any life adjustment factor(s). The following relationships hold under stated specific conditions:

**TABLE 13.
LIFE RATIO EQUATIONS**

Condition	Equation
Variable load Variable speed	$(L_{10})_2 = (L_{10})_1 (P_1 / P_2)^{10/3} (n_1 / n_2)$
Variable power Variable speed	$(L_{10})_2 = (L_{10})_1 (H_1 / H_2)^{10/3} (n_2 / n_1)^{7/3}$
Constant load Variable speed	$(L_{10})_2 = (L_{10})_1 (n_1 / n_2)$
Constant power Variable speed	$(L_{10})_2 = (L_{10})_1 (n_2 / n_1)^{7/3}$
Variable load Constant speed	$(L_{10})_2 = (L_{10})_1 (P_1 / P_2)^{10/3}$
Variable power Constant speed	$(L_{10})_2 = (L_{10})_1 (H_1 / H_2)^{10/3}$

[P = Load, torque or tangential gear force]

To calculate system weighted life Timken determines the weighted life for each bearing separately and then calculates a system weighted life.

PERMISSIBLE OPERATING SPEED AND LUBRICATION

When determining the permissible operating speeds corresponding to the bearing preloads used in machine tool spindles, many influencing factors are involved. Among those considered are spindle mass and construction, type of mounting, spindle rigidity and accuracy requirements, spindle loads, service life, type of service (intermittent or continuous), and method of lubrication.

Bearing temperatures, generally, vary directly with both speed and load. However, high-speed applications must have sufficient axial loading on the bearings to prevent heat generation due to rolling element skidding. The amount of bearing preload is determined primarily from these operating conditions. At lower speeds, the operating loads are heavier and the bearing deflections are greater. Therefore, the bearing preload must be high enough to provide adequate bearing rigidity under the heaviest loads and still maintain reasonable temperatures when the spindle is operated at high speeds.

TAPERED ROLLER BEARINGS

MEASURING RIB SPEED

The usual measure of the speed of a tapered roller bearing is the rib speed, which is the circumferential velocity at the midpoint of the inner ring large end rib (Fig. 53). This may be calculated as:

Rib speed:

$$V_r = \frac{\pi D_m n}{60000} \quad \text{(m/s)}$$

$$= \frac{\pi D_m n}{12} \quad \text{(ft/min)}$$

Where:

D_m	=	mean inner ring large rib diameter	mm, in.
n	=	bearing speed	rev/min

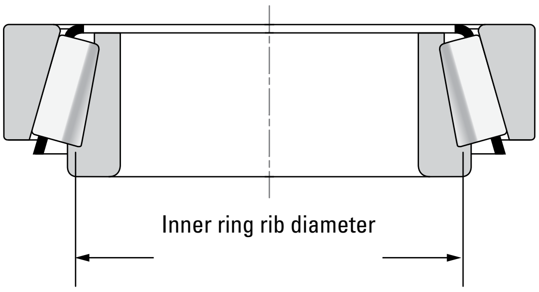


Fig. 53. Cone rib diameter. The inner ring rib diameter may be scaled from a print.

EFFECT OF LUBRICATION ON SPEED CAPABILITY

The design of the tapered roller bearing results in a natural pumping effect on the lubricant, where the lubricant is forced from the small end of the roller end, heading toward the wider end. As speed increases, the lubricant begins to move outward due to centrifugal effects. At excessive speed, the contact between the roller large ends and the cone's rib face can become a concern. This is the primary reason for suggestions on the use of oil jets at this large end, ribbed-cup designs, or high-speed TSMA designs as operating speeds increase. Refer to the following speed guidelines for more details.

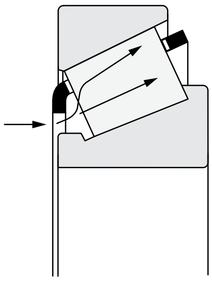


Fig. 54. Pumping effect of a tapered roller bearing.

There are no clear-cut speed limitations for tapered roller bearings since performance depends on the bearing design and lubrication system. The guidelines given in the table on page 39 are based on typical industrial experiences relating to speed and temperature for various types of lubrication systems, with bearings having low G_1 factor.

Timken suggests that testing be performed for all new high-speed applications.

SPEED GUIDELINES FOR TAPERED ROLLER BEARINGS IN MACHINE TOOL SPINDLES

TABLE 14.
SPEED GUIDELINES

Bearing Type And Position	Cone Rib Speed (Feet/Minute)					
	0 - 2500	2500 - 4000	4000 - 6000	6000 - 8000	8000 - 10000	10000 - 20000
TS bearing at nose position						
With standard stamped cage	Yes	Yes	Yes	No	No	No
With thermal compensating device	N/R [10]	N/R	Consider [1]	Yes [1]	No	No
With internal geometry modifications	N/R	N/R	Consider	Yes	No	No
With cage modifications	N/R	N/R	N/R	Yes	No	No
With silver-plated cage	N/R	N/R	N/R	Consider	No	No
With machined cage	N/R	N/R	N/R	N/R	No	No
With improved finish	N/R	N/R	N/R	Consider	No	No
TSMA bearing at nose position						
TSMA	N/R	N/R	N/R	Consider [1]	Yes [1]	Yes [1]
With internal geometry modifications	N/R	N/R	N/R	Yes	Yes	Yes
With cage modifications	N/R	N/R	N/R	Yes	Yes	Yes
With silver-plated cage	N/R	N/R	N/R	Consider	Yes	Yes
With machined cage	N/R	N/R	N/R	N/R	Consider	Yes
With improved finish	N/R	N/R	N/R	Consider	Consider	Yes
Ribbed cup bearing at nose position						
Ribbed cup	N/R	N/R	N/R	N/R	Yes	Yes
With internal geometry modifications	N/R	N/R	N/R	N/R	Yes	Yes
With silver-plated cage	N/R	N/R	N/R	N/R	Yes	Yes
With machined cage	N/R	N/R	N/R	N/R	Consider	Yes
With oil drainage holes in cup	N/R	N/R	N/R	N/R	Consider	Yes
With improved finish	N/R	N/R	N/R	N/R	Consider	Yes
Hydra-Rib bearing at rear position						
Standard Hydra-Rib	Consider	Consider	Consider	Yes	No	No
Modified Hydra-Rib	N/R	N/R	N/R	N/R	Yes	Yes
With internal geometry modifications	N/R	N/R	N/R	Yes	Yes	Yes
With silver-plated cage	N/R	N/R	N/R	Consider	Yes	Yes
With machined cage	N/R	N/R	N/R	N/R	Consider	Yes
With oil drainage holes in cup	N/R	N/R	N/R	N/R	Consider	Yes
With improved finish	N/R	N/R	N/R	Consider	Consider	Yes
Lubrication system						
Standard spindle grease	Yes	No	No	No	No	No
Special high-speed grease	N/R	Yes [2]	No	No	No	No
Oil level	Yes [4]	Yes [4]	No	No	No	No
Air/oil or mist	N/R	Consider	Yes [1]	Yes [1][8]	No	No
Circulating oil	N/R	N/R	Yes [5]	Yes [5]	Yes [6]	Yes [6]
Oil jets required under cage	N/R	N/R	N/R	Yes [7]	Yes [7]	Yes [7]
Oil jets required to backface rib	N/R	N/R	N/R	Yes [4][7]	Yes [3][7]	Yes [3][7]

[1]Requires use of Hydra-Rib, Spring-Rib, or spring loaded design at rear position.
[2]Kluber NBU15, Mobil 28, or equivalent.
[3]Only for TSMA bearings.
[4]Use ISO VG32 or equivalent for oil level.
[5]Do not use greater than ISO VG32 or equivalent for circulating oil. Preferred is ISO VG22 or equivalent.
[6]Same as (9) except water jackets in housing would also be required.
[7]3 Jets at 120 degrees.
[8]Not to be used with TSMA design.
[9]Normally used for operating speeds less than 12.7 m/s (2500 fpm).
[10]Not required (N/R).

TAPERED ROLLER BEARING DESIGNS
2TS mounting (standard design)
2TS + TDO at rear (box mounting) [9]
2TS mounting w/spring mounting
TS mounting + Hydra-Rib
2 TSMA mounting front and rear
TSMA mounting + Hydra-Rib
2TS ribbed cup mounting
Ribbed cup mounting + Hydra-Rib

SPINDLE BEARING DESIGN FACTORS
K-Factor of 1.00 to 1.80 preferred.

Look at G_1 Factor for indication of heat generation characteristics.

Thin section L and LL type bearings should be given primary consideration.

Consult with your Timken representative to ensure bearings selected have good high-speed characteristics.

LUBRICATION GUIDELINES FOR HIGHER SPEED BEARINGS

A precision tapered roller bearing can meet almost any level of speed required by the machine tool industry with the TSMA and Hydra-Rib high-speed bearing designs, providing circulating oil lubrication can be accommodated.

Both the lubricant and lubrication system have an effect on heat-generation and heat-dissipation rates and thus are important to the speed capabilities of a bearing.

The choice of lubrication will depend on:

- Maximum speed requirement
- Heat dissipation rate of the system
- Spindle layout
- Orientation of the spindle

INTERNAL BEARING DESIGN – G_1 FACTOR

Internal bearing geometry has a direct influence on torque and, therefore heat generation. In order to rate the torque/heat generation characteristics of its bearings and to assist designers in selecting appropriate bearings, Timken developed a factor called G_1: the lower the G_1 factor, the lower the heat generation. The G_1 factors are published in Timken catalogs and are listed in the appendix of this catalog for the precision tapered roller bearing part numbers listed in the tapered roller bearing section.

This G_1 factor is of prime importance to a designer because of the influence of operating temperature on spindle accuracy.

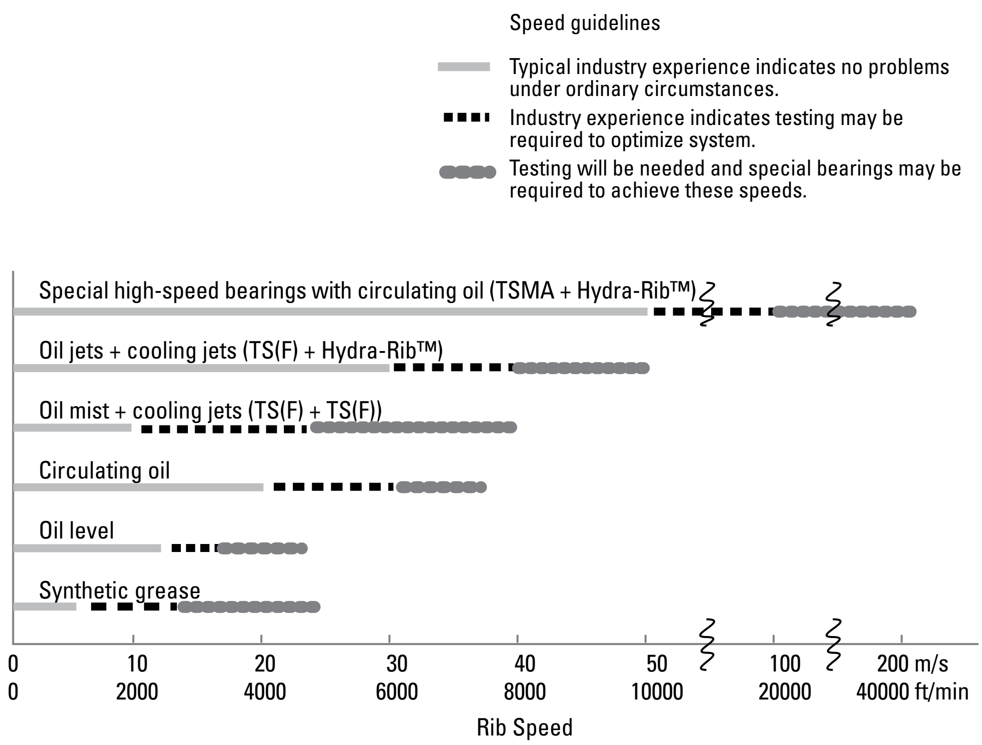


Fig. 55. Speed capability guidelines for various types of lubrication systems.

LUBRICATION

TAPERED ROLLER BEARINGS

The selection of the lubricant and lubricant delivery method is directly linked with the speedability of a bearing. It is strongly suggested that the section on permissible operating speed be reviewed by the customer in addition to this section on lubrication.

GREASE

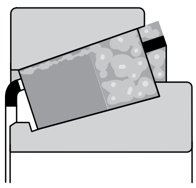


Fig. 56. Filling a bearing with synthetic grease.

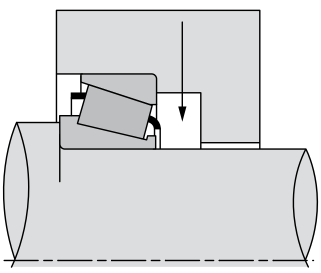


Fig. 57. Simple radial oil inlet hole with oil collector.

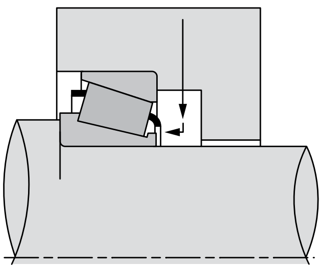


Fig. 58. Axial oil jet to direct oil at small end of the rollers.

Grease lubrication speed limits are lower than limits for oil lubrication because all the heat must be carried away by conduction through the shaft and housing.

Mineral Grease

When conventional (mineral) greases are used, the rib speed should be limited to 5 m/s (985 fpm). This limit can be increased under pure radial loads up to 13 m/s (2560 fpm) provided that the bearings remain in end play under all operating conditions. Generally, No. 2 consistency greases are used with medium- to low-viscosity base oils.

Lubricant quantity (V_{mg}) may be approximated by using the following equation:

$$V_{mg} = f_{mg} \times V = f_{mg} \times \left[\frac{\pi}{4} \times T \times (D^2 - d^2) \times 10^{-3} - \frac{M}{7.8 \times 10^{-3}} \right] \quad (cm^3)$$

Where:

f_{mg}	=	factor depending on speed: $0.3 < f_{mg} < 0.5$	
V	=	free volume of the bearing	(cm^3)
T	=	overall bearing width	(mm)
D	=	cup outer diameter	(mm)
d	=	cone bore	(mm)
M	=	bearing weight	(kg)
π	=	3.1416	

Synthetic Grease

The use of "low-torque" greases (or synthetic greases) can be considered for rib speeds over 13 m/s (2560 fpm), up to maximum of 25 m/s (4920 fpm). Experience has shown that stabilized temperatures, around 15° C to 20° C (27° F to 36° F) above ambient, can be obtained at the maximum permissible speed.

It is important to follow these procedures to help achieve the above performance:

- All corrosion protection is removed from the bearing surfaces by using an organic solvent.
- Very small initial quantity of grease is applied to prevent excessive churning.
- Initial run-in period to evacuate unnecessary grease from the bearing.
- Good spindle design to retain grease around the bearings.
- Efficient sealing to protect against external contamination.

Lubricant quantity (V_{sg}) may be approximated by using the following equation:

$$V_{sg} = f_{sg} \times V = f_{sg} \times \left[\frac{\pi}{4} \times T \times (D^2 - d^2) \times 10^{-3} - \frac{M}{7.8 \times 10^{-3}} \right] \quad (cm^3)$$

Where:

f_{sg} = factor depending on speed: 0.15 < fsg < 0.3

π = 3.1416

When using synthetic greases, the limiting factor is the "lubrication for life" concept (without re-greasing). Depending on load and speed conditions, the grease life will typically be limited to 5000 to 8000 hours.

The Timken Company's suggestions for the use and run-in of synthetic greases are illustrated later in this section.

A normal way to fill the bearing with grease is to do it by hand before heating and fitting the components. For the cone, the free volume corresponding to the first third of the rollers, starting from their large end, is filled with grease; an additional quantity is provided below the cage. For the cup, a thin film of grease is spread all around the race as shown in Fig. 56 on the previous page.

OIL

Grease lubrication of spindle bearings is generally preferred by machine tool builders over oil circulation lubrication due to its simplicity and low heat generation. For high loads or high speeds, however, circulating oil is probably the most widely used method because of its capability to remove heat from the spindle.

Oil Circulation

Many parameters have to be considered in designing an efficient oil circulation lubrication system:

- Oil characteristics.
- Oil flow rates.
- Oil feed and drain systems.
- Heat dissipation rate of the bearing system.

The latter is affected by factors such as conduction through the housing walls and convection by the circulating lubricant.

Oil Characteristics

A low-viscosity mineral oil in the range of ISO VG10 to ISO VG22 is generally specified for the bearings. This choice will minimize heat generation, particularly at high speeds, where the lowest practical viscosity is required. Care must be taken, however, if gears are used for the power transmission because the choice of the common lubricant will be systematically dictated by the needs of the gears. High-quality mineral oils having suitable additives for lubricating both the gears and bearings are available with a relatively low viscosity.

Oil Feed System

Forced-feed oil systems are generally used in the machine tool industry. In a typical system, oil is pumped from a central reservoir to each bearing separately. Oil is introduced at the small end of the rollers and drained away at the large end to take advantage of the natural pumping action of tapered roller bearings.

Circulating oil allows a continuous regulated oil flow. Apart from providing the advantages of maximum heat dissipation, it also has the added benefit of removing any contamination or debris that could possibly cause bearing wear.

Heat exchangers can be included in a circulating system to reduce oil inlet temperature and better regulate the running temperature of the system. Filters of 0.040 mm (0.00157 in.) size also are generally provided to remove debris.

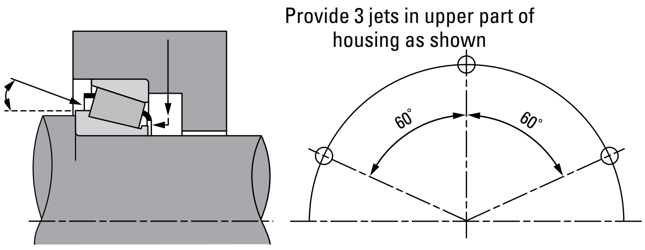


Fig. 59. Cooling jets in top part of the housing for speed above 25 m/s (4920 fpm).

Experience has shown that for speeds up to 20 m/s (3940 fpm), a simple radial oil inlet hole in the top part of the housing in conjunction with an oil collector is sufficient (Fig. 57 on the previous page). For speeds over 20 m/s (3940 fpm), an axial oil jet should be positioned to direct oil at the small end of the roller at the gap between the cage and the inner ring (Fig. 58 on the previous page). For high speeds or in case of large size bearings, additional oil jets can be arranged about the circumference to better distribute the oil within the bearings.

With increasing speeds (approx-imately 25 m/s [4920 fpm] and above), the effect of centrifugal force will throw the oil to the outside along the cup race. To prevent lubricant starvation at the inner ring rib, and consequent bearing burnup, additional oil jets have to be provided in the top part of the housing (Fig. 59 on the previous page).

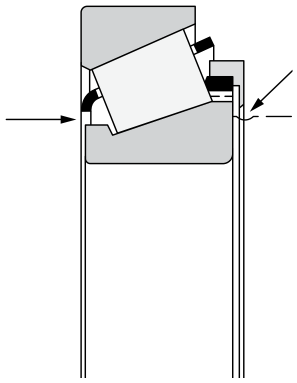


Fig. 60. The TSMA bearing.

For rib speeds over 40 m/s (7874 fpm), special high-speed TSMA bearings have been developed. A special provision for lubrication of the roller-rib contact area is provided to ensure adequate lubrication. The concept works by capturing oil in a manifold attached to the inner ring and directing it to the roller-rib contact area through holes drilled axially in the inner ring (Fig. 60).

Oil Drainage System

An effective circulating oil system requires adequate drainage to prevent an oil buildup that would cause excessive churning and unnecessary heat generation. Oil passing through a high-speed bearing will exit the bearing at a high velocity and also will swirl around the housing in the direction of rotation of the bearing. To effectively drain the oil away, the high velocity must be slowed and the swirling action stopped so that the oil will fall down into the drain area. A drain catch basin is required to break up the flow of oil and direct the oil to the drain hole (Fig. 61). Oil drain sections must be adequately dimensioned to ease the rapid evacuation of the oil.

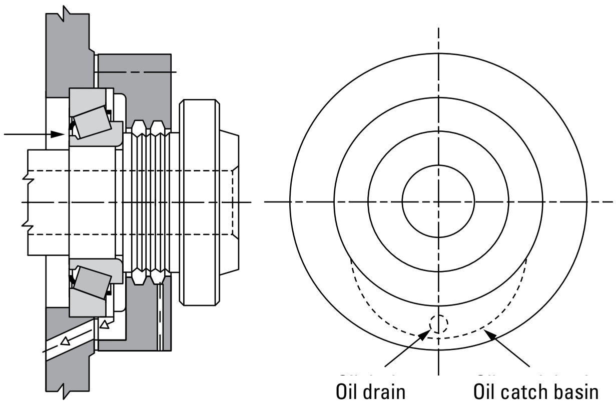


Fig. 61. Oil drain design.

BALL BEARINGS

Even though ball bearings have the least amount of friction of any of the common rolling bearings, lubrication is required to minimize rolling resistance due to deformation of the balls in the raceways under load, and to minimize any sliding friction that occurs between the balls, the raceway and the cage. Lubrication also serves to protect the accurately ground and polished surfaces from corrosion. In addition, lubrication in general dissipates generated heat and can help protect the bearing from the entry of foreign matter.

Regardless of the method of lubrication or type of lubricant, it is important that quality lubricants be used to minimize oxidation, gumming or sludging and that the lubricant be clean and free of moisture to minimize wear.

Only enough lubrication to accomplish these purposes should be used since another source of heat may become present, namely friction between the lubricant and the moving parts, in the form of churning or internal shear of the lubricant itself.

In the lubrication of ball bearings it is important to realize that a small quantity of oil or grease will, if constantly present in the bearing, suffice for its requirements. It should be noted that trouble can result from too much lubrication just as it can from too little. Both conditions should be avoided. Excessive oil or grease will result in high temperature and possible damage. When grease is used, it is necessary to take into consideration the maximum operating temperature. Particular attention must be given to the housing design relating to the proximity of the grease to the bearing, to assure adequate purge room and grease retention.

Depending upon operating speeds, loads and temperatures, machine tool ball bearings are lubricated with grease, oil or oil mist. In general, oils are required when bearings operate at high speeds as they provide greater cooling than is possible with grease.

The following relationship may be used to estimate the effect of preload and lubrication method on the Permissible Operating Speed (S_P).

$$S_P = F_L \times F_P \times F_B \times N_G$$

Where:

F_L = lubrication factor

F_P = preload factor

F_B = ball material factor

N_G is Permissible Speed for a single, grease-lubricated bearing with inner ring rotation. This value is found with the part number tables.

Factors are as follows:

TABLE 15.
LUBRICATION FACTOR = (F_L)

Lubrication Factor (F_L)	
Grease	$F_L = 1.00$
Oil bath	$F_L = 1.50$
Oil mist	$F_L = 1.70$
Oil jet or metered oil	$F_L = 2.00$

TABLE 16.
BEARING PRELOAD FACTOR = (F_p)

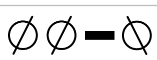
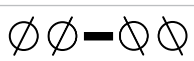
Bearing Mounting Arrangement	Bearing Preload L	M	H
	0,85	0,70	0,50
	0,80	0,60	0,40
	0,65	0,50	0,30
	0,65	0,50	0,30
	0,70	0,60	0,35
	0,60	0,40	0,20
	0,65	0,45	0,25

TABLE 17.
BALL MATERIAL FACTOR (F_B)

Ball Material Factor (F_B)	
Steel balls	$F_B = 1.00$
Ceramic balls[1]	$F_B = 1.20$

[1]Ceramic balls allow 20 percent increase to speed factor.

GREASE

The use of grease as a lubricant for Timken super precision ball bearings on various spindle applications is becoming more popular, due to the development of better ball bearing greases, simplification of design and elimination of the “human maintenance factor,” which is frequently responsible for too much lubrication, not enough lubrication, or the wrong kind of lubrication. Prelubricating the bearings at assembly with the correct amount of the correct grease, thus eliminating all grease fittings, has increased bearing life in many instances.

For successful lubrication, grease for ball bearings should have good mechanical and chemical stability with low-torque characteristics. Two different types of grease, one soft and the other firmer, have proved to be suitable lubricants for machine tool spindle bearings. The “soft” greases have a worked penetration factor corresponding to NLGI of two or less. The firmer grease has a worked penetration factor of three or more and is of the channeling type. All greases show a very slight change in consistency after operation in a bearing. As the softer grease has a tendency to churn, particular attention should be given to the quantity packed into the bearing. Because the firmer grease is of the channeling type, the amount used is not as critical.

For super precision ball bearings below a 500000 dN value, which is equivalent to a 50.000 mm (1.9685 in.) bore bearing rotating at 10000 RPM, either a light-consistency grease or the channeling grease may be used.

At continuous speeds above a dN value of 500000, the operating temperature is generally lower when the bearings are lubricated with a lower-consistency grease and after sufficient break-in.

However, the grease quantity in each bearing must be limited. At these higher speeds, an excessive amount of grease in the bearing may result in greatly increased operating temperatures due to churning action. This condition, if uncontrolled, may lead to premature bearing damage.

Grease and Speed Capability

Before selecting a grease, it is important to define a relative speed capability of the application. There is no precise method that can be applied to determine the operating speed of a bearing. Over the years, designers of machine tool systems have been guided by their own experiences from which many basic "rules of thumb" have been established. One such rule is the dN speed value.

dN = Bore in millimeters x RPM

The most common spindle greases that Timken suggests for Timken spindle ball bearings are:

Unirex N3	Vertical applications < 500000 dN
Mobil 28	Light loads < 600000 dN
Chevron SRI	Medium to heavy loads < 350000 dN
Kluber Isoflex NBU 15/ NCA 15	Light loads, vertical or horizontal applications > 500000 dN
Kluber BF-7222	Light loads, vertical or horizontal applications at higher speeds > 750000 dN

OIL

Although several grease products have been successful at dN values as high as one million, oils are generally suggested for bearings operating at high speeds or to provide more cooling and dissipation of heat than is possible with grease. High-grade spindle oil having a viscosity of 100 seconds Saybolt at 37° C (100° F) is suggested for use in drip-feed oilers, oil bath lubrication arrangements and oil mist systems. In heavily loaded applications, oil in relatively large quantities must be supplied, and where temperatures run higher than normal, oil coolers will be suggested. Churning of a large pool of oil is to be avoided if speed is significant.

Oil Bath

The conventional oil-bath system for lubricating the bearings is satisfactory for low and moderate speeds. The static oil level must never be higher than the center of the lowermost ball. When the shaft is rotating, the running level may drop considerably below the standstill level, depending on the speed of the revolving parts. A sight gauge or other suitable method should be provided to permit an easy check.

Drip-Feed Oil

Where the speeds are considered high for oil bath and the bearings are moderately loaded, oil introduced through a filter-type, sight-feed oiler is suggested. This assures a constant supply of lubricant. The feed in drops-per-minute is determined by closely observing the operating temperatures.

Oil Jet

In applications where the ball bearing is heavily loaded and operating at high speed and high temperatures, or where the operating conditions are severe with high ambient temperatures, oil jet lubrication may be required. In such cases, it is necessary to lubricate each bearing location individually and to provide adequately large drain openings to prevent excessive accumulation of oil after it has passed through the bearings.(1)

Oil Mist

Oil mist lubrication is often used for spindles running continuously at high speeds. With this method of lubrication, oil of the proper viscosity is atomized into finely divided particles, mixed with clean, filtered, dry compressed air and directed to pass through the bearings in a constant stream. This oil is metered into the air under pressure. This system not only lubricates the bearings, it affords some cooling due to the air flow. This continuous passage of air and oil through the bearings and the labyrinth seals also serves to prevent the entrance of contaminants into the bearings.

To ensure the "wetting" of the bearings and to prevent possible damage to the balls and raceways, it is imperative that the oil mist system be turned on for several minutes before the spindle is started. The importance of wetting the bearings before starting cannot be overstated and has particular significance for spindles that have been idle for extended periods of time. To avoid such effects, most oil mist systems have interlocks that make it impossible to start the spindle until the lubricating system is working properly and the bearings are thoroughly wetted.[1]

Metered Air/Oil

This method is similar to the oil mist; however, the oil is fed by periodic pulses to the lubrication line providing a higher air-to-oil ratio. Therefore, this method lowers the operating bearing temperature and lubricant shear effects, enabling higher operating speeds.[1]

[1]For further information, refer to the lubrication specification tables found on pages 240-241.

TABLE 18.
LUBRICATION SYSTEM COMPARISON

Lubrication Type	System Cost	Typical Speed (dN)[2]
Grease	Low	500000
High-speed grease	Low	750000
Oil bath	Low	400000
Oil drip	Low	600000
Oil mist	Medium	1000000
Metered air/oil	High	>1000000
Oil jet	High	>1000000

[2]Speed value is an approximation and assumes proper mounting and preload techniques along with average loading conditions. For more specific guidance contact your Timken representative.

The speed, "dN," value is obtained by multiplying the bearing bore size in millimeters by the shaft RPM.

RUN-IN PROCEDURES

TAPERED ROLLER BEARINGS WITH SYNTHETIC GREASE

The aim of run-in cycles is to correctly spread the grease inside the bearing, to avoid churning of the grease and excessive bearing temperature.

During run-in operations, the bearing temperature must be constantly monitored and immediately plotted on a graph so that any tendency of the curve toward a vertical asymptote can be averted. Temperature probes, placed closest to the bearings, will provide better control of the run-in operations.

The other advantage of the graph is to help determine the run-in time at a given speed. When the curve becomes horizontal, it shows that the temperature has stabilized. It is then possible to proceed to the next speed.

The indicated times may vary depending on the speeds and heat dissipation capacity of the spindles.

According to the results obtained on the prototype, it may be possible to reduce either the number or the length (or both) of the run-in steps for production spindles. In any event, temperature control should be retained for safety reasons.

When running-in multi-speed spindles, reduced speeds must be chosen at start-up of the cycles. The speed can be progressively increased until the bearings evacuate any excessive quantities of grease.

TABLE 19.
RUN-IN SUGGESTIONS FOR SYNTHETIC GREASE-LUBRICATED TAPERED ROLLER BEARINGS WITH SINGLE-SPEED SPINDLES

Time		Action	
10	sec	Run	
1	min		Stop
20	sec	Run	
1	min		Stop
30	sec	Run	
1	min		Stop
40	sec	Run	
1	min		Stop
50	sec	Run	
1	min		Stop
1	min	Run	
1	min		Stop
90	sec	Run	
1	min		Stop
2	min	Run	
1	min		Stop
3	min	Run*	
1	min		Stop
4	min	Run*	
1	min		Stop
6	min	Run*	
1	min		Stop
10	min	Run*	
20	min		Stop

⟹ Then run until temperature stabilizes. At this step of the cycle, as well as at the other steps marked *, closely watch the curve's shape. If it tends to be vertical, stop 15 minutes and run again at 75 percent of max. speed until the temperature stabilizes again. Then restart the cycle from the beginning at max. speed.

TABLE 20.
RUN-IN SUGGESTIONS FOR SYNTHETIC GREASE-LUBRICATED TAPERED ROLLER BEARINGS WITH MULTI-SPEED SPINDLES

25% Max. Speed			50% Max. Speed			75% Max Speed			Max. Speed		
Time	Action		Time	Action		Time	Action		Time	Action	
1 min	Run		1 min	Run		1 min	Run		1 min	Run	
1 min		Stop	1 min		Stop	1 min		Stop	1 min		Stop
1 min	Run		1 min	Run		1 min	Run		1 min	Run	
1 min		Stop	1 min		Stop	1 min		Stop	1 min		Stop
2 min	Run		2 min	Run		2 min	Run		2 min	Run	
1 min		Stop	1 min		Stop	1 min		Stop	1 min		Stop
3 min	Run		3 min	Run		3 min	Run		3 min	Run	
5 min		Stop	5 min		Stop	5 min		Stop	5 min		Stop
⟹ Then run until temperature stabilizes.			⟹ Then run until temperature stabilizes.			⟹ Then run and closely watch the curve's shape during running, until stabilization. If it tends to be vertical, stop 15 minutes and run again at same speed.			⟹ Then run until temperature stabilizes. At this step of the cycle, closely watch the curve's shape. If it tends to be vertical, stop 15 minutes and run again at 75 percent of max. speed until the temperature stabilizes again. Then restart the cycle from the beginning at max. speed.		

BALL BEARINGS WITH GREASE LUBRICATION

Fig. 62 shows bearing temperature increase due to run-in procedure. The peaking temperature followed by the leveling off is a result of the new grease being worked and then stabilized for a particular condition of load and speed.

It is important that the peak temperature not exceed 55˚ C (100˚ F) above room temperature since the chemical consistency and characteristics of the grease can be permanently altered. Thus, the proper run-in procedure is to run the machine until the spindle temperature rises to 65˚ C (150˚ F) and then turn it off to allow the grease to cool. Repeat until the spindle temperature stabilizes at a temperature below 54˚ C (130˚ F).

Fig. 63 shows the typical temperature rise of the bearing once the grease has been worked in for the specific speed and load.

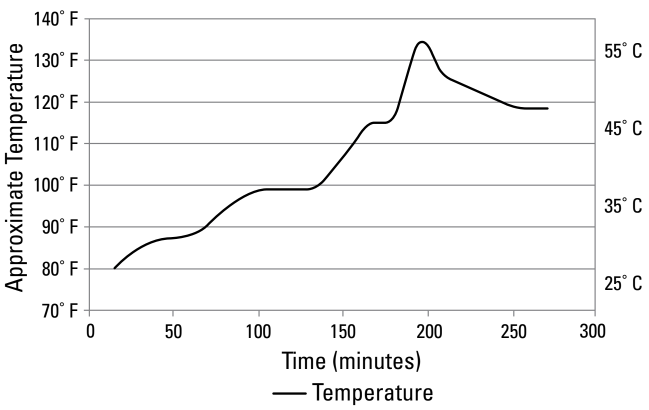


Fig. 62. Typical temperature profile during break-in (4 speed increments from 200000 dN start to 800000 dN finish).

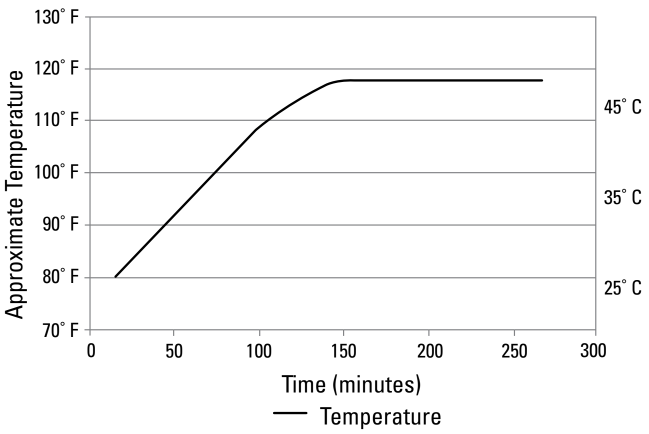


Fig. 63. Typical temperature after break-in procedure.

BALL BEARINGS WITH GREASE (FOR SPEEDS > 500000 dN)

A proper run-in procedure will provide the following results:

- Expel the excess grease from the bearings.
- Orient the lubricating film on each contact surface.
- Establish a low-equilibrium operating temperature.

RUN-IN PROCEDURE

1. Install proper quantity of grease.
2. Start at a reasonable low speed, typically 10 percent of the maximum operating speed.
3. Increase speed with small, reasonable increments when a stable temperature is reached.
4. Continue incremental increase in speed as described. If a rapid temperature increase occurs (temperature exceeds 70˚ C [158˚ F]), stop the run-in process.

   Maximum bearing temperatures should not exceed 70˚ C (158˚ F). Temperatures in excess of 70˚ C (158°F) may cause excessive bearing preloads and possible permanent grease or bearing damage.
5. Allow the system to cool to room temperature.
6. Restart procedure at the last speed prior to the temperature spike.
7. Continue repeating the above cycle until an equilibrium temperature is reached at the maximum operating speed of the application. *The ideal equilibrium operating temperature is 35˚ C to 46˚ C (95˚ F to 115˚ F).*

ALTERNATIVE RUN-IN PROCEDURE (WHEN UNABLE TO CONTROL INCREMENTAL SPEEDS)

Run-in at constant speed also is possible. In this operation, the bearing should run at full speed for about 30 seconds. After stopping, the heat in the bearing dissipates. In this way, a dangerous temperature rise is prevented. The non-running time depends on the various design factors, but it should be at least five times greater than the running time. This process is repeated until the bearing temperature becomes constant.

VERTICAL SPINDLES

For vertical axis spindles, special attention must be paid to the lubrication and sealing. Modified sealing is required to prevent the coolant from contaminating the lubricant when the spindle nose bearing is at the upper position (Fig. 64).

In the case of grease lubrication, deflectors have to be installed to prevent grease migration away from the bearing cavity. Alternatively, when oil lubrication is adopted and the nose bearing is at the lower position, a system to collect and extract the oil has to be provided to prevent leakage (Fig. 65).

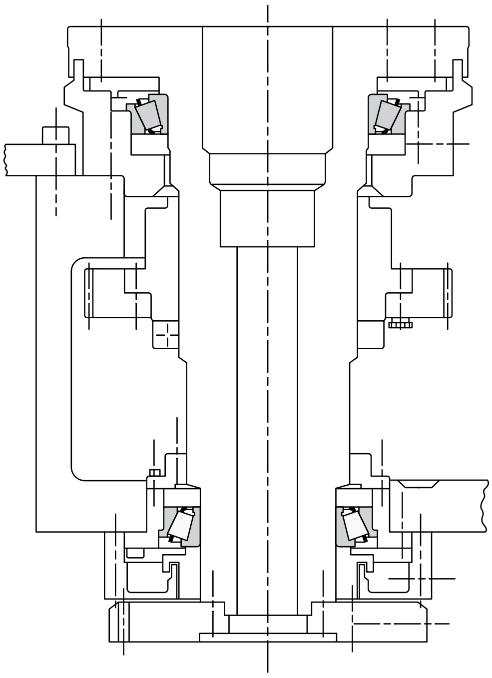


Fig. 64. Vertical axis spindle.

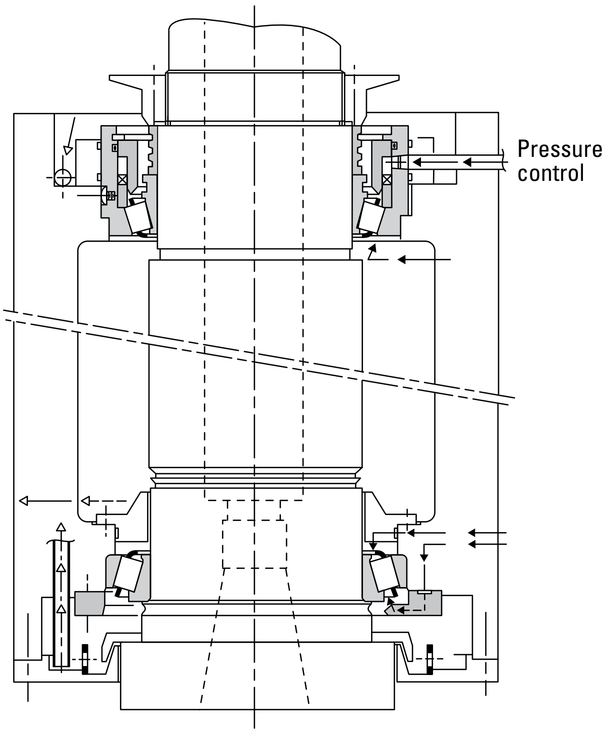


Fig. 65. Typical example of proper use of an oil-lubricated vertical axis spindle with pressure control for the oil and a means to collect and extract the oil.

HEAT GENERATION AND DISSIPATION

TAPERED ROLLER BEARINGS

HEAT GENERATION

Under normal operating conditions, most of the torque and heat generated by the bearing is due to the elastohydrodynamic losses at the contact area between rollers and races.

The following equation is used to calculate the heat generated by the bearing:

$$Q_{gen} = k_4 n\, M$$

$$M = k_1 G_1 (n\mu)^{0.62} (P_{eq})^{0.3}$$

Where:

- Q_{gen} = generated heat (W or Btu/min)
- M = running torque N.m or lbf-in.
- n = rotational speed (RPM)
- G_1 = geometry factor from bearing data tables
- μ = viscosity at operating temperature (cP)
- P_{eq} = dynamic equivalent load (N or lbf)
- k_1 = bearing torque constant
  = 2.56×10^{-6} for M in N-m
  = 3.54×10^{-5} for M in lbf-in.
- k_4 = dimensional factor to calculate heat generation rate
  = 0.105 for Qgen in W when M in N-m
  = 6.73×10^{-4} for Qgen in Btu/min when M in lbf-in.

The generated heat will be underestimated if operating speed:

$$n \le \frac{k_2}{G_2 \times \mu} \left(\frac{f_2 \times F_r}{K} \right)^{0.66} \text{RPM}$$

Where:

- G_2 = geometry factor (from bearing data tables)
- k_2 = 625 (metric) or 1700 (in.)
- K = K factor of the bearing (from bearing data tables)
- f_2 = combined load factor (Fig. 66)

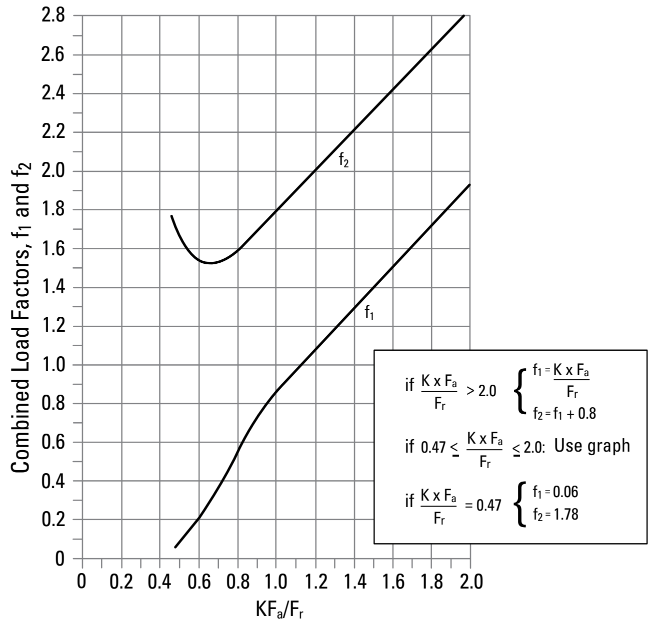
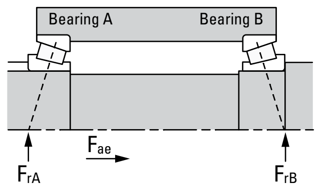
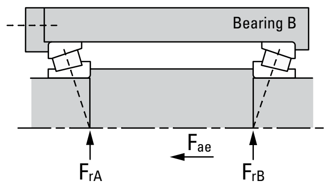


Fig. 66. Determination of combined load factors f_1 and f_2.

TABLE 21.
DETERMINATION OF DYNAMIC EQUIVALENT LOAD P_{eq}

Design (Thrust F_{ae} Onto A)	Thrust Condition	Net Axial Load	Dynamic Equivalent Load P_{eq}
	$\frac{0.47 \times F_{rA}}{K_A} \le \frac{0.47 \times F_{rB}}{K_B} + F_{ae}$	$F_{aA} = \frac{0.47 \times F_{rB}}{K_B} + F_{ae}$ $F_{aB} = \frac{0.47 \times F_{rB}}{K_B}$	$P_{eq} = \left(\frac{f_1 \times F_r}{K} \right)$ f_1 = combined load factor (see Fig. 66)
	$\frac{0.47 \times F_{rA}}{K_A} > \frac{0.47 \times F_{rB}}{K_B} + F_{ae}$	$F_{aA} = \frac{0.47 \times F_{rA}}{K_A}$ $F_{aB} = \frac{0.47 \times F_{rA}}{K_A} - F_{ae}$	

HEAT DISSIPATION

The heat dissipation rate of a bearing system is affected by many factors, and the modes of heat transfer need to be considered. Major heat transfer modes in most systems are conduction through the housing walls, convection at the inside and outside surfaces of the housing, and convection by the circulating lubricant. In many applications, overall heat dissipation can be divided into two categories:

- Heat removed by circulating oil.
- Heat removed through the housing.

Heat Dissipation by Circulating Oil

Heat dissipated by a circulating oil system is:

$$Q_{oil} = k_5 \times f \times (\theta_o - \theta_i)$$

If a circulating lubricant other than petroleum oil is used, the heat carried away by that lubricant will be:

$$Q_{oil} = k_6 \times C_p \times \rho \times f \times (\theta_o - \theta_i)$$

Where:

k_5 = dimensional factor to calculate heat carried away by a petroleum oil
k_5 = 28 for Q_{oil} in W when f in L/min and θ in °C
k_5 = 0.42 for Q_{oil} in Btu/min when f in U.S. pt/min and θ in °F

k_6 = dimensional factor to calculate heat carried away by a circulating fluid
k_6 = 1.67×10^{-5} for Q_{oil} in W
k_6 = 1.67×10^{-2} for Q_{oil} in Btu/min

Q_{oil}	= heat dissipation rate of circulating oil	W, Btu/min
θ_i	= oil inlet temperature	°C, °F
θ_o	= oil outlet temperature	°C, °F
C_p	= specific heat of lubricant	J/(kg x °C), Btu/(lb x °F)
f	= lubricant flow rate	L/min, U.S. pt/min
ρ	= lubricant density	kg/m³, lb/ft³

If lubricant flow is unrestricted on the outlet side of a bearing, the flow rate that can freely pass through the bearing depends on bearing size and internal geometry, direction of oil flow, bearing speed and lubricant properties.

A tapered roller bearing has a natural tendency to pump oil from the small end to the large end of the rollers. For maximum oil flow and heat dissipation, the oil inlet should be adjacent to the small end of the rollers.

In a splash or oil level lubrication system, heat will be carried by convection to the inner walls of the housing. The heat dissipation rate with this lubrication method can be enhanced by using cooling coils in the housing sump.

Heat Dissipation Through Housing

Heat removed through the housing is, in most cases, difficult to determine analytically. If the steady-state bearing temperature is known for one operating condition, the following method can be used to estimate the housing heat dissipation rate.

At the steady-state temperature, the total heat dissipation rate from the bearing must equal the heat generation rate of the bearing. The difference between the heat generation rate and heat dissipation rate of the oil is the heat dissipation rate of the housing at the known temperature.

Heat losses from housings are primarily by conduction and convection and are therefore nearly linearly related to temperature difference. Thus, the housing heat dissipation rate is:

$$Q_{hsg} = C (\theta_o - \theta_{ambt})$$

At the operating condition where the steady-state temperature is known, the housing heat dissipation factor can be estimated as:

$$C = \frac{Q_{gen} - Q_{oil}}{\theta_o - \theta_{ambt}}$$

BALL BEARINGS

HEAT GENERATION

Low operating temperatures, combined with adequate spindle rigidity, are important and highly desirable for precision machine tools. This is particularly true for high-speed grinding spindles where the preload of the bearings is the principal load imposed upon them. Some of the benefits derived from low operating temperatures are better dimensional stability of the processed work, less need for bearing lubrication, prevention of objectionable heat at the external surfaces of the spindle housing, and elimination of troubles due to thermal effects on mounting fits and preloads.

PRELOAD AND HEAT GENERATION

The heat developed at the bearings under load is a function of the operating speed and the bearing preload. Preloading is necessary for maximum axial and radial rigidity. Unfortunately, if speeds are increased, the bearing preload may have to be lessened to maintain proper operating temperatures at the bearing.

For high-speed operation, the bearing preload should be sufficient to maintain proper rolling friction for the balls, but not so high as to generate excessive heat. In cases where lower operating speeds are desired, bearing preloads may be increased to obtain additional bearing rigidity, provided the proper operating temperatures are maintained. Thus, a balance between heat generation and spindle rigidity dictates the amount of bearing preload that is used, commensurate with the operational speed and the bearing life required.

How bearing preload affects the operating temperature is illustrated in Fig. 67. This graph applies to 207-size, angular contact, duplexed super precision ball bearings, mounted back-to-back. Curve A is a plot of operating temperature at the bearing outside diameter for the speeds indicated, using bearings with a high built-in preload. Curve B is for bearings having a low preload. The slope of Curve A is much steeper than that of Curve B. Using bearings with a high preload, the temperature rise at the bearing outside diameter is 34˚ C (93˚ F) when operating at 3600 RPM. For the same temperature rise using bearings with low preload, an operating speed of 15300 RPM is indicated. Therefore, it is evident that for higher-speed operation, the bearing preload should be kept to the minimum necessary to ensure sufficient bearing rigidity.

For workhead spindles, the operating speeds are generally low and the loading conditions heavy. Maximum radial and axial spindle rigidity is required under these loads, making increased bearing preload mandatory.

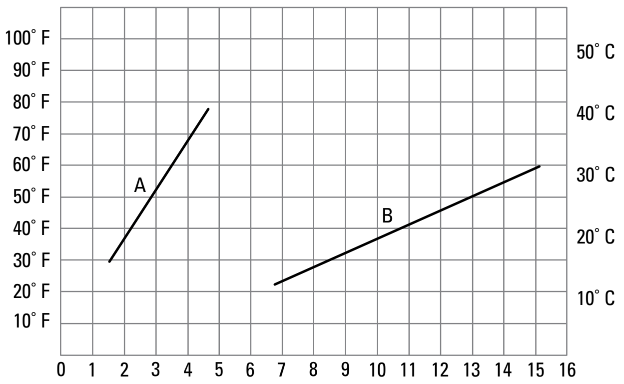


Fig. 67. Effect of preload on temperature rise.

BEARING GEOMETRY AND HEAT GENERATION

It should be noted that a bearing's internal geometry has a major impact on heat generation. High-speed designs, such as the Timken HX Series, incorporate "optimized" internal geometries that balance load-carrying capacity, stiffness and heat generation.

HEAT DISSIPATION

When ball bearing spindles are grease lubricated, the heat generated is removed only by conduction through the surrounding parts. With jet or circulating oil lubrication, generated heat is dissipated by the oil passing through the bearings as well as by conduction through the shaft and housing. Both means of removing heat from the bearings are important, but generally, dissipation through conduction is less obvious.

As an example, in an oil mist-lubricated grinding spindle, the nose or wheel-end bearings are fixed and close to the grinding coolant. The pulley-end or rear bearings are secured axially on the shaft, but permitted to float laterally in the housing to compensate for size variations due to thermal changes. Heat is conducted away from the front bearings at a faster rate because of the thermal mass of the spindle nose and the intimate contact of the outer rings with the housing shoulder, the end cover and the housing bore. This condition, coupled with oil mist lubrication and the proximity of the grinding coolant, takes away generated heat efficiently.

The rear or floating pair of bearings is not so favored. Usually, the thermal mass of the shaft at the drive-end is not so great. The drive-end possesses some heat-conduction ability, but also receives heat generated by belt friction. The absence of grinding coolant and the reduced area of conduction usually results in a slightly higher operating temperature.

TOLERANCES
TAPERED ROLLER BEARINGS

Tapered roller bearings are manufactured to a number of specifications or "classes" that define tolerances on dimensions such as bore, O.D., width and runout. The Timken Company produces bearings in both inch and metric systems. The boundary dimension tolerances applicable to these two categories of bearings differ.

The major difference between the two tolerance systems is that inch bearings have historically been manufactured to positive bore and O.D. tolerances, whereas metric bearings have been manufactured to negative tolerances.

TABLE 22.
TAPERED ROLLER BEARING PRECISION CLASSES

Tapered Roller Bearings - Precision Class				
Timken Metric	C	B	A	AA
Timken Inch	3	0	00	000
ISO/DIN	P5	P4	P2	-
ABMA Metric	C	B	A	-
ABMA Inch	3	0	00	-

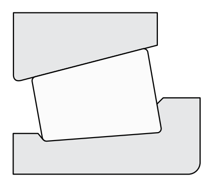
Crossed Tapered - Precision Class		
Timken Metric	S	P
Timken Inch	3	0

METRIC SYSTEM BEARINGS (ISO AND "J" PREFIX PARTS)

Timken manufactures metric system bearings to four tolerance classes. Classes C, B, A and AA are "precision" classes. These tolerances lie within those currently specified in ISO 492 with the exception of a small number of dimensions indicated in the tables. The differences normally have an insignificant effect on the mounting and performance of tapered roller bearings. The adjacent table illustrates the current ISO bearing class that corresponds approximately to each of The Timken Company metric bearing classes.

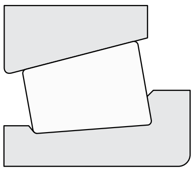
TABLE 23.
METRIC BEARING TOLERENCES - CONE BORE

Cone Bore			Precision Bearing Class							
Bearing Types[(1)]	Bore Over	Bore Incl.	C Max.	C Min.	B Max.	B Min.	A Max.	A Min.	AA Max.	AA Min.
	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.
	10.000	**18.000**	**0.000**	**-0.007**	**0.000**	**-0.005**	**0.000**	**-0.005**	**0.000**	**-0.005**
	0.3937	0.7087	0.0000	-0.0003	0.0000	-0.0002	0.0000	-0.0002	0.0000	-0.0002
	18.000	**30.000**	**0.000**	**-0.008**	**0.000**	**-0.006**	**0.000**	**-0.006**	**0.000**	**-0.006**
	0.7087	1.1811	0.0000	-0.0003	0.0000	-0.0002	0.0000	-0.0002	0.0000	-0.0002
	30.000	**50.000**	**0.000**	**-0.010**	**0.000**	**-0.008**	**0.000**	**-0.008**	**0.000**	**-0.008**
	1.1811	1.9685	0.0000	-0.0004	0.0000	-0.0003	0.0000	-0.0003	0.0000	-0.0003
	50.000	**80.000**	**0.000**	**-0.012**	**0.000**	**-0.009**	**0.000**	**-0.008**	**0.000**	**-0.008**
	1.9685	3.1496	0.0000	-0.0005	0.0000	-0.0004	0.0000	-0.0003	0.0000	-0.0003
	80.000	**120.000**	**0.000**	**-0.015**	**0.000**	**-0.010**	**0.000**	**-0.008**	**0.000**	**-0.008**
	3.1496	4.7244	0.0000	-0.0006	0.0000	-0.0004	0.0000	-0.0003	0.0000	-0.0003
	120.000	**180.000**	**0.000**	**-0.018**	**0.000**	**-0.013**	**0.000**	**-0.008**	**0.000**	**-0.008**
	4.7244	7.0866	0.0000	-0.0007	0.0000	-0.0005	0.0000	-0.0003	0.0000	-0.0003
	180.000	**250.000**	**0.000**	**-0.022**	**0.000**	**-0.015**	**0.000**	**-0.008**	**0.000**	**-0.008**
	7.0866	9.8425	0.0000	-0.0009	0.0000	-0.0006	0.0000	-0.0003	0.0000	-0.0003
TS TSF	**250.000**	**265.000**	**0.000**	**-0.022**	**0.000**	**-0.015**	**0.000**	**-0.008**	**0.000**	**-0.008**
	9.8425	10.4331	0.0000	-0.0009	0.0000	-0.0006	0.0000	-0.0003	0.0000	-0.0003
	265.000	**315.000**	**0.000**	**-0.022**	**0.000**	**-0.015**	**0.000**	**-0.008**	**0.000**	**-0.008**
	10.4331	12.4016	0.0000	-0.0009	0.0000	-0.0006	0.0000	-0.0003	0.0000	-0.0003
	315.000	**400.000**	**0.000**	**-0.025**	–	–	–	–	–	–
	12.4016	15.7480	0.0000	-0.0010	–	–	–	–	–	–
	400.000	**500.000**	**0.000**	**-0.025**	–	–	–	–	–	–
	15.7480	19.6850	0.0000	-0.0010	–	–	–	–	–	–
	500.000	**630.000**	**0.000**	**-0.030**	–	–	–	–	–	–
	19.6850	24.8031	0.0000	-0.0012	–	–	–	–	–	–
	630.000	**800.000**	**0.000**	**-0.040**	–	–	–	–	–	–
	24.8031	31.4961	0.0000	-0.0016	–	–	–	–	–	–
	800.000	**1000.000**	**0.000**	**-0.050**	–	–	–	–	–	–
	31.4961	39.3701	0.0000	-0.0020	–	–	–	–	–	–
	1000.000	**1200.000**	**0.000**	**-0.060**	–	–	–	–	–	–
	39.3701	47.2441	0.0000	-0.0024	–	–	–	–	–	–
	1200.000	**1600.000**	**0.000**	**-0.080**	–	–	–	–	–	–
	47.2441	62.9921	0.0000	-0.0031	–	–	–	–	–	–



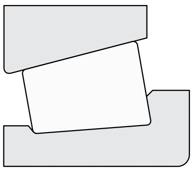
(1)Not all types and sizes are listed in this catalog. Contact your Timken representative for further information.

TABLE 24.
METRIC BEARING TOLERENCES - CUP O.D.



Cup O.D.			Precision Bearing Class							
Bearing Types[1]	Cup O.D. Over	Incl.	C Max.	C Min.	B Max.	B Min.	A Max.	A Min.	AA Max.	AA Min.
	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.
TS TSF	**10.000**	**18.000**	–	–	–	–	**0.000**	**-0.008**	**0.000**	**-0.008**
	0.3937	0.7087	–	–	–	–	0.0000	-0.0003	0.0000	-0.0003
	18.000	**30.000**	**0.000**	**-0.008**	**0.000**	**-0.006**	**0.000**	**-0.008**	**0.000**	**-0.008**
	0.7087	1.1811	0.0000	-0.0003	0.0000	-0.0002	0.0000	-0.0003	0.0000	-0.0003
	30.000	**50.000**	**0.000**	**-0.009**	**0.000**	**-0.007**	**0.000**	**-0.008**	**0.000**	**-0.008**
	1.1811	1.9685	0.0000	-0.0004	0.0000	-0.0003	0.0000	-0.0003	0.0000	-0.0003
	50.000	**80.000**	**0.000**	**-0.011**	**0.000**	**-0.009**	**0.000**	**-0.008**	**0.000**	**-0.008**
	1.9685	3.1496	0.0000	-0.0004	0.0000	-0.0004	0.0000	-0.0003	0.0000	-0.0003
	80.000	**120.000**	**0.000**	**-0.013**	**0.000**	**-0.010**	**0.000**	**-0.008**	**0.000**	**-0.008**
	3.1496	4.7244	0.0000	-0.0005	0.0000	-0.0004	0.0000	-0.0003	0.0000	-0.0003
	120.000	**150.000**	**0.000**	**-0.015**	**0.000**	**-0.011**	**0.000**	**-0.008**	**0.000**	**-0.008**
	4.7244	5.9055	0.0000	-0.0006	0.0000	-0.0004	0.0000	-0.0003	0.0000	-0.0003
	150.000	**180.000**	**0.000**	**-0.018**	**0.000**	**-0.013**	**0.000**	**-0.008**	**0.000**	**-0.008**
	5.9055	7.0866	0.0000	-0.0007	0.0000	-0.0005	0.0000	-0.0003	0.0000	-0.0003
	180.000	**250.000**	**0.000**	**-0.020**	**0.000**	**-0.015**	**0.000**	**-0.008**	**0.000**	**-0.008**
	7.0866	9.8425	0.0000	-0.0008	0.0000	-0.0006	0.0000	-0.0003	0.0000	-0.0003
	250.000	**265.000**	**0.000**	**-0.025**	**0.000**	**-0.018**	**0.000**	**-0.008**	**0.000**	**-0.008**
	9.8425	10.4331	0.0000	-0.0010	0.0000	-0.0007	0.0000	-0.0003	0.0000	-0.0003
	265.000	**315.000**	**0.000**	**-0.025**	**0.000**	**-0.018**	**0.000**	**-0.008**	**0.000**	**-0.008**
	10.4331	12.4016	0.0000	-0.0010	0.0000	-0.0007	0.0000	-0.0003	0.0000	-0.0003
	315.000	**400.000**	**0.000**	**-0.028**	**0.000**	**-0.020**	–	–	–	–
	12.4016	15.7480	0.0000	-0.0011	0.0000	-0.0008	–	–	–	–
	400.000	**500.000**	**0.000**	**-0.030**	–	–	–	–	–	–
	15.7480	19.6850	0.0000	-0.0012	–	–	–	–	–	–
	500.000	**630.000**	**0.000**	**-0.035**	–	–	–	–	–	–
	19.6850	24.8031	0.0000	-0.0014	–	–	–	–	–	–
	630.000	**800.000**	**0.000**	**-0.040**	–	–	–	–	–	–
	24.8031	31.4961	0.0000	-0.0016	–	–	–	–	–	–
	800.000	**1000.000**	**0.000**	**-0.050**	–	–	–	–	–	–
	31.4961	39.3701	0.0000	-0.0020	–	–	–	–	–	–
	1000.000	**1200.000**	**0.000**	**-0.060**	–	–	–	–	–	–
	39.3701	47.2441	0.0000	-0.0024	–	–	–	–	–	–
	1200.000	**1600.000**	**0.000**	**-0.080**	–	–	–	–	–	–
	47.2441	62.9921	0.0000	-0.0031	–	–	–	–	–	–

TABLE 25.
METRIC BEARING TOLERANCES - CONE WIDTH



Cone Width			Precision Bearing Class							
Bearing Types[1]	Bore Over	Incl.	C Max.	C Min.	B Max.	B Min.	A Max.	A Min.	AA Max.	AA Min.
	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.
TS TSF	**10.000**	**50.000**	**0.000**	**-0.200**	**0.000**	**-0.200**	**0.000**	**-0.200**	**0.000**	**-0.200**
	0.3937	1.9685	0.0000	-0.0079	0.0000	-0.0079	0.0000	-0.0079	0.0000	-0.0079
	50.000	**120.000**	**0.000**	**-0.300**	**0.000**	**-0.300**	**0.000**	**-0.300**	**0.000**	**-0.300**
	1.9685	4.7244	0.0000	-0.0118	0.0000	-0.0118	0.0000	-0.0118	0.0000	-0.0118
	120.000	**180.000**	**0.000**	**-0.300**	**0.000**	**-0.300**	**0.000**	**-0.300**	**0.000**	**-0.300**
	4.7244	7.0866	0.0000	-0.0118	0.0000	-0.0118	0.0000	-0.0118	0.0000	-0.0118
	180.000	**250.000**	**0.000**	**-0.350**	**0.000**	**-0.350**	**0.000**	**-0.350**	**0.000**	**-0.350**
	7.0866	9.8425	0.0000	-0.0138	0.0000	-0.0138	0.0000	-0.0138	0.0000	-0.0138
	250.000	**265.000**	**0.000**	**-0.350**	**0.000**	**-0.350**	**0.000**	**-0.350**	**0.000**	**-0.350**
	9.8425	10.4331	0.0000	-0.0138	0.0000	-0.0138	0.0000	-0.0138	0.0000	-0.0138
	265.000	**315.000**	**0.000**	**-0.350**	**0.000**	**-0.350**	**0.000**	**-0.350**	**0.000**	**-0.350**
	10.4331	12.4016	0.0000	-0.0138	0.0000	-0.0138	0.0000	-0.0138	0.0000	-0.0138
	315.000	**500.000**	**0.000**	**-0.350**	–	–	–	–	–	–
	12.4016	19.6850	0.0000	-0.0138	–	–	–	–	–	–
	500.000	**630.000**	**0.000**	**-0.350**	–	–	–	–	–	–
	19.6850	24.8031	0.0000	-0.0138	–	–	–	–	–	–
	630.000	**1200.000**	**0.000**	**-0.350**	–	–	–	–	–	–
	24.8031	47.2441	0.0000	-0.0138	–	–	–	–	–	–
	1200.000	**1600.000**	**0.000**	**-0.350**	–	–	–	–	–	–
	47.2441	62.9921	0.0000	-0.0138	–	–	–	–	–	–

[1]Not all types and sizes are listed in this catalog. Contact your Timken representative for further information.

TABLE 26.
METRIC BEARING TOLERANCES - CUP WIDTH

Cup Width			Precision Bearing Class							
Bearing Types(1)	Cup O.D.		C		B		A		AA	
	Over	Incl.	Max.	Min.	Max.	Min.	Max.	Min.	Max.	Min.
	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.
TS TSF	**10.000**	**80.000**	**0.000**	**-0.150**	**0.000**	**-0.150**	**0.000**	**-0.150**	**0.000**	**-0.150**
	0.3937	3.1496	0.0000	-0.0059	0.0000	-0.0059	0.0000	-0.0059	0.0000	-0.0059
	80.000	**150.000**	**0.000**	**-0.200**	**0.000**	**-0.200**	**0.000**	**-0.200**	**0.000**	**-0.200**
	3.1496	5.9055	0.0000	-0.0079	0.0000	-0.0079	0.0000	-0.0079	0.0000	-0.0079
	150.000	**180.000**	**0.000**	**-0.250**	**0.000**	**-0.250**	**0.000**	**-0.250**	**0.000**	**-0.250**
	5.9055	7.0866	0.0000	-0.0098	0.0000	-0.0098	0.0000	-0.0098	0.0000	-0.0098
	180.000	**250.000**	**0.000**	**-0.250**	**0.000**	**-0.250**	**0.000**	**-0.250**	**0.000**	**-0.250**
	7.0866	9.8425	0.0000	-0.0098	0.0000	-0.0098	0.0000	-0.0098	0.0000	-0.0098
	250.000	**265.000**	**0.000**	**-0.300**	**0.000**	**-0.300**	**0.000**	**-0.300**	**0.000**	**-0.300**
	9.8425	10.4331	0.0000	-0.0118	0.0000	-0.0118	0.0000	-0.0118	0.0000	-0.0118
	265.000	**315.000**	**0.000**	**-0.300**	**0.000**	**-0.300**	**0.000**	**-0.300**	**0.000**	**-0.300**
	10.4331	12.4016	0.0000	-0.0118	0.0000	-0.0118	0.0000	-0.0118	0.0000	-0.0118
	315.000	**400.000**	**0.000**	**-0.300**	**0.000**	**-0.300**	–	–	–	–
	12.4016	15.7480	0.0000	-0.0118	0.0000	-0.0118	–	–	–	–
	400.000	**500.000**	**0.000**	**-0.350**	–	–	–	–	–	–
	15.7480	19.6850	0.0000	-0.0138	–	–	–	–	–	–
	500.000	**800.000**	**0.000**	**-0.350**	–	–	–	–	–	–
	19.6850	31.4961	0.0000	-0.0138	–	–	–	–	–	–
	800.000	**1200.000**	**0.000**	**-0.400**	–	–	–	–	–	–
	31.4961	47.2441	0.0000	-0.0157	–	–	–	–	–	–
	1200.000	**1600.000**	**0.000**	**-0.400**	–	–	–	–	–	–
	47.2441	62.9921	0.0000	-0.0157	–	–	–	–	–	–

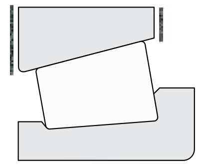
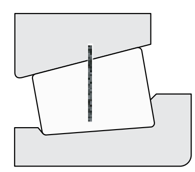


TABLE 27.
METRIC BEARING TOLERANCES - CONE STAND

Cone Stand			Precision Bearing Class							
Bearing Types(1)	Bore		C		B		A		AA	
	Over	Incl.	Max.	Min.	Max.	Min.	Max.	Min.	Max.	Min.
	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.
TS TSF	**10.000**	**80.000**	**+0.100**	**-0.100**	(2)	(2)	(2)	(2)	(2)	(2)
	0.3937	3.1496	+0.0039	-0.0039						
	80.000	**120.000**	**+0.100**	**-0.100**						
	3.1496	4.7244	+0.0039	-0.0039						
	120.000	**180.000**	**+0.100**	**-0.100**						
	4.7244	7.0866	+0.0039	-0.0039						
	180.000	**250.000**	**+0.100**	**-0.150**						
	7.0866	9.8425	+0.0039	-0.0059						
	250.000	**265.000**	**+0.100**	**-0.150**						
	9.8425	10.4331	+0.0039	-0.0059						
	265.000	**315.000**	**+0.100**	**-0.150**						
	10.4331	12.4016	+0.0039	-0.0059						
	315.000	**400.000**	**+0.150**	**-0.150**	–	–	–	–	–	–
	12.4016	15.7480	+0.0059	-0.0059	–	–	–	–	–	–
	400.000	–	(2)	(2)	–	–	–	–	–	–
	15.7480	–			–	–	–	–	–	–



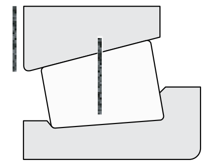
Cone Stand. Cone stand is a measure of the variation in cone raceway size, taper and roller diameter. This is checked by measuring the axial location of the reference surface of a master cup or other type gage with respect to the reference cone face.

(1)Not all types and sizes are listed in this catalog. Contact your Timken representative for further information.
(2)These sizes manufactured as matched assemblies only.

TABLE 28.
METRIC BEARING TOLERANCES - CUP STAND

Cup Stand			Precision Bearing Class							
Bearing Types[1]	Bore Over	Incl.	C		B		A		AA	
			Max.	Min.	Max.	Min.	Max.	Min.	Max.	Min.
	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.
	10.000	**18.000**	**+0.100**	**-0.100**						
	0.3937	0.7087	+0.0039	-0.0039						
	18.000	**80.000**	**+0.100**	**-0.100**						
	0.7087	3.1496	+0.0039	-0.0039						
	80.000	**120.000**	**+0.100**	**-0.100**	(2)	(2)	(2)	(2)	(2)	(2)
	3.1496	4.7244	+0.0039	-0.0039						
TS TSF[3]	**120.000**	**265.000**	**+0.100**	**-0.150**						
	4.7244	10.4331	+0.0039	-0.0059						
	265.000	**315.000**	**+0.100**	**-0.150**						
	10.4331	12.4016	+0.0039	-0.0059						
	315.000	**400.000**	**+0.100**	**-0.150**	–	–	–	–	–	–
	12.4016	15.7480	+0.0039	-0.0059	–	–	–	–	–	–
	400.000	–	(2)	(2)	–	–	–	–	–	–
	15.7480	–			–	–	–	–	–	–

[1]Not all types and sizes are listed in this catalog. Contact your Timken representative for further information.
[2]These sizes manufactured as matched assemblies only.
[3]Stand for flanged cup is measured from flange backface (seating face).



Cup Stand. Cup stand is a measure of the variation in cup I.D. size and taper. This is checked by measuring the axial location of the reference surface of a master plug or other type gage with respect to the reference face of the cup.

TABLE 29.
METRIC BEARING TOLERANCES - OVERALL BEARING WIDTH

Overall Bearing Width			Precision Bearing Class							
Bearing Types[1]	Bore Over	Incl.	C		B		A		AA	
			Max.	Min.	Max.	Min.	Max.	Min.	Max.	Min.
	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.
	10.000	**80.000**	**+0.200**	**-0.200**	**+0.200**	**-0.200**	**+0.200**	**-0.200**	**+0.200**	**-0.200**
	0.3937	3.1496	+0.0079	-0.0079	+0.0079	-0.0079	+0.0079	-0.0079	+0.0079	-0.0079
	80.000	**120.000**	**+0.200**	**-0.200**	**+0.200**	**-0.200**	**+0.200**	**-0.200**	**+0.200**	**-0.200**
	3.1496	4.7244	+0.0079	-0.0079	+0.0079	-0.0079	+0.0079	-0.0079	+0.0079	-0.0079
	120.000	**180.000**	**+0.350**	**-0.250**	**+0.200**	**-0.250**	**+0.200**	**-0.250**	**+0.200**	**-0.250**
	4.7244	7.0866	+0.0138	-0.0098	+0.0079	-0.0098	+0.0079	-0.0098	+0.0079	-0.0098
	180.000	**250.000**	**+0.350**	**-0.250**	**+0.200**	**-0.300**	**+0.200**	**-0.300**	**+0.200**	**-0.300**
	7.0866	9.8425	+0.0138	-0.0098	+0.0079	-0.0118	+0.0079	-0.0118	+0.0079	-0.0118
	250.000	**265.000**	**+0.350**	**-0.300**	**+0.200**	**-0.300**	**+0.200**	**-0.300**	**+0.200**	**-0.300**
	9.8425	10.4331	+0.0138	-0.0118	+0.0079	-0.0118	+0.0079	-0.0118	+0.0079	-0.0118
TS TSF[2]	**265.000**	**315.000**	**+0.350**	**-0.300**	**+0.200**	**-0.300**	**+0.200**	**-0.300**	**+0.200**	**-0.300**
	10.4331	12.4016	+0.0138	-0.0118	+0.0079	-0.0118	+0.0079	-0.0118	+0.0079	-0.0118
	315.000	**500.000**	**+0.350**	**-0.300**	–	–	–	–	–	–
	12.4016	19.6850	+0.0138	-0.0118	–	–	–	–	–	–
	500.000	**800.000**	**+0.350**	**-0.400**	–	–	–	–	–	–
	19.6850	31.4961	+0.0138	-0.0157	–	–	–	–	–	–
	800.000	**1000.000**	**+0.350**	**-0.400**	–	–	–	–	–	–
	31.4961	39.3701	+0.0138	-0.0157	–	–	–	–	–	–
	1000.000	**1200.000**	**+0.350**	**-0.450**	–	–	–	–	–	–
	39.3701	47.2441	+0.0138	-0.0177	–	–	–	–	–	–
	1200.000	**1600.000**	**+0.350**	**-0.500**	–	–	–	–	–	–
	47.2441	62.9921	+0.0138	-0.0197	–	–	–	–	–	–

[1]Not all types and sizes are listed in this catalog. Contact your Timken representative for further information.
[2]For bearing type TSF, the tolerance applies to the dimension T_1.

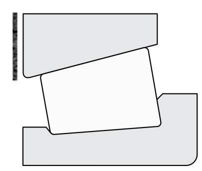
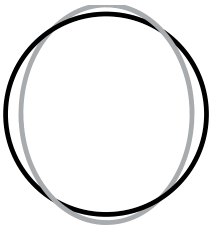


TABLE 30.
METRIC BEARING TOLERANCES - ASSEMBLED BEARING MAXIMUM RADIAL RUNOUT

Assembled Bearing Maximum Radial Runout			Precision Bearing Class			
Bearing Types[1]	Bore Over	Bore Incl.	C	B	A	AA
	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.
TS TSF	**10.000** 0.3937	**18.000** 0.7087	**0.005** –	**0.003** 0.0001	**0.002** 0.00008	**0.001** 0.00004
	18.000 0.7087	**30.000** 1.1811	**0.005** 0.0002	**0.003** 0.0001	**0.002** 0.00008	**0.001** 0.00004
	30.000 1.1811	**50.000** 1.9685	**0.006** 0.0002	**0.003** 0.0001	**0.002** 0.00008	**0.001** 0.00004
	50.000 1.9685	**80.000** 3.1496	**0.006** 0.0002	**0.004** 0.0002	**0.002** 0.00008	**0.001** 0.00004
	80.000 3.1496	**120.000** 4.7244	**0.006** 0.0002	**0.004** 0.0002	**0.002** 0.00008	**0.001** 0.00004
	120.000 4.7244	**150.000** 5.9055	**0.007** 0.0003	**0.004** 0.0002	**0.002** 0.00008	**0.001** 0.00004
	150.000 5.9055	**180.000** 7.0866	**0.008** 0.0003	**0.004** 0.0002	**0.002** 0.00008	**0.001** 0.00004
	180.000 7.0866	**250.000** 9.8425	**0.010** 0.0004	**0.005** 0.0002	**0.002** 0.00008	**0.001** 0.00004
	250.000 9.8425	**265.000** 10.4331	**0.011** 0.0004	**0.005** 0.0002	**0.002** 0.00008	**0.001** 0.00004
	265.000 10.4331	**315.000** 12.4016	**0.011** 0.0004	**0.005** 0.0002	**0.002** 0.00008	**0.001** 0.00004
	315.000 12.4016	**400.000** 15.7480	**0.013** 0.0005	**0.005** 0.0002	**–** –	**–** –
	400.000 15.7480	**500.000** 19.6850	**0.018** 0.0007	**–** –	**–** –	**–** –
	500.000 19.6850	**630.000** 24.8031	**0.025** 0.0010	**–** –	**–** –	**–** –
	630.000 24.8031	**800.000** 31.4961	**0.035** 0.0014	**–** –	**–** –	**–** –
	800.000 31.4961	**1000.000** 39.3701	**0.050** 0.0020	**–** –	**–** –	**–** –
	1000.000 39.3701	**1200.000** 47.2441	**0.060** 0.0024	**–** –	**–** –	**–** –
	1200.000 47.2441	**1600.000** 62.9921	**0.080** 0.0031	**–** –	**–** –	**–** –

[1]Not all types and sizes are listed in this catalog. Contact your Timken representative for further information.



Runout. Runout is a measure of rotational accuracy expressed by Total Indicator Reading (T.I.R.). Total displacement is measured by an instrument sensing against a moving surface, or moved with respect to a fixed surface. A radial runout measurement includes both roundness errors and the centering error of the surface that the instrument head senses against.

METRIC SYSTEM BEARINGS (TXR PREFIX PARTS)

TABLE 31.
METRIC BEARING TOLERANCES - OUTER RACE O.D.

Outer Race O.D.			Precision Bearing Class			
Bearing Types	O.D. Over	O.D. Incl.	S Max.	S Min.	P Max.	P Min.
	mm / in.	mm / in.	mm / in.	mm / in.	mm / in.	mm / in.
TXR	18.000 / 0.7087	30.000 / 1.1811	0.000 / 0.0000	-0.010 / -0.0004	0.000 / 0.0000	-0.008 / -0.0003
	30.000 / 1.1811	50.000 / 1.9685	0.000 / 0.0000	-0.013 / -0.0005	0.000 / 0.0000	-0.010 / -0.0004
	50.000 / 1.9685	80.000 / 3.1496	0.000 / 0.0000	-0.015 / -0.0006	0.000 / 0.0000	-0.010 / -0.0004
	80.000 / 3.1496	120.000 / 4.7244	0.000 / 0.0000	-0.020 / -0.0008	0.000 / 0.0000	-0.013 / -0.0005
	120.000 / 4.7244	180.000 / 7.0866	0.000 / 0.0000	-0.025 / -0.0010	0.000 / 0.0000	-0.013 / -0.0005
	180.000 / 7.0866	250.000 / 9.8425	0.000 / 0.0000	-0.030 / -0.0012	0.000 / 0.0000	-0.015 / -0.0006
	250.000 / 9.8425	400.000 / 15.7480	0.000 / 0.0000	-0.040 / -0.0016	0.000 / 0.0000	-0.020 / -0.0008
	400.000 / 15.7480	630.000 / 24.8031	0.000 / 0.0000	-0.050 / -0.0020	0.000 / 0.0000	-0.025 / -0.0010
	630.000 / 24.8031	800.000 / 31.4961	0.000 / 0.0000	-0.060 / -0.0024	– / –	– / –
	800.000 / 31.4961	1000.000 / 39.3701	0.000 / 0.0000	-0.080 / -0.0031	– / –	– / –
	1000.000 / 39.3701	1200.000 / 47.2441	0.000 / 0.0000	-0.100 / -0.0039	– / –	– / –
	1200.000 / 47.2441	1600.000 / 62.9921	0.000 / 0.0000	-0.120 / -0.0047	– / –	– / –
	1600.000 / 62.9921	2000.000 / 78.7402	0.000 / 0.0000	-0.140 / -0.0055	– / –	– / –
	2000.000 / 78.7402	– / –	0.000 / 0.0000	-0.140 / -0.0055	– / –	– / –

TABLE 32.
METRIC BEARING TOLERANCES - INNER RACE BORE

Inner Race Bore			Precision Bearing Class			
Bearing Types	Bore Over	Bore Incl.	S Max.	S Min.	P Max.	P Min.
	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.
TXR	**10.000** 0.3937	**50.000** 1.9685	**0.000** 0.0000	**-0.013** -0.0005	**0.000** 0.0000	**-0.010** -0.0004
	50.000 1.9685	**80.000** 3.1496	**0.000** 0.0000	**-0.015** -0.0006	**0.000** 0.0000	**-0.013** -0.0005
	80.000 3.1496	**120.000** 4.7244	**0.000** 0.0000	**-0.020** -0.0008	**0.000** 0.0000	**-0.013** -0.0005
	120.000 4.7244	**180.000** 7.0866	**0.000** 0.0000	**-0.025** -0.0010	**0.000** 0.0000	**-0.013** -0.0005
	180.000 7.0866	**250.000** 9.8425	**0.000** 0.0000	**-0.030** -0.0012	**0.000** 0.0000	**-0.015** -0.0006
	250.000 9.8425	**315.000** 12.4016	**0.000** 0.0000	**-0.035** -0.0014	**0.000** 0.0000	**-0.018** -0.0007
	315.000 12.4016	**400.000** 15.7480	**0.000** 0.0000	**-0.040** -0.0016	**0.000** 0.0000	**-0.020** -0.0008
	400.000 15.7480	**500.000** 19.6850	**0.000** 0.0000	**-0.045** -0.0018	**0.000** 0.0000	**-0.025** -0.0010
	500.000 19.6850	**630.000** 24.8031	**0.000** 0.0000	**-0.050** -0.0020	**0.000** 0.0000	**-0.030** -0.0012
	630.000 24.8031	**800.000** 31.4961	**0.000** 0.0000	**-0.060** -0.0024	**–** –	**–** –
	800.000 31.4961	**1000.000** 39.3701	**0.000** 0.0000	**-0.080** -0.0031	**–** –	**–** –
	1000.000 39.3701	**1200.000** 47.2441	**0.000** 0.0000	**-0.100** -0.0039	**–** –	**–** –
	1200.000 47.2441	**1600.000** 62.9921	**0.000** 0.0000	**-0.120** -0.0047	**–** –	**–** –
	1600.000 62.9921	**2000.000** 78.7402	**0.000** 0.0000	**-0.140** -0.0055	**–** –	**–** –
	2000.000 78.7402	**–** –	**0.000** 0.0000	**-0.140** -0.0055	**–** –	**–** –

INCH SYSTEM BEARINGS

Inch system bearings are manufactured to a number of tolerance classes. Classes 3, 0, 00 and 000 are "precision" classes.

TABLE 33.
INCH BEARING TOLERECES - CONE BORE

Cone Bore			Precision Bearing Class							
Bearing Types(1)	Bore		3		0		00		000	
	Over	Incl.	Max.	Min.	Max.	Min.	Max.	Min.	Max.	Min.
	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.
TS TSF TSL(2) TDI TDIT TDO TNA	**0.000**	**76.200**	**+0.013**	**0.000**	**+0.013**	**0.000**	**+0.008**	**0.000**	**+0.008**	**0.000**
	0.0000	3.0000	+0.0005	0.0000	+0.0005	0.0000	+0.0003	0.0000	+0.0003	0.0000
	76.200	**304.800**	**+0.013**	**0.000**	**+0.013**	**0.000**	**+0.008**	**0.000**	**+0.008**	**0.000**
	3.0000	12.0000	+0.0005	0.0000	+0.0005	0.0000	+0.0003	0.0000	+0.0003	0.0000
	304.800	**609.600**	**+0.025**	**0.000**	–	–	–	–	–	–
	12.0000	24.0000	+0.0010	0.0000	–	–	–	–	–	–
	609.600	**914.400**	**+0.038**	**0.000**	–	–	–	–	–	–
	24.0000	36.0000	+0.0015	0.0000	–	–	–	–	–	–
	914.400	**1219.200**	**+0.051**	**0.000**	–	–	–	–	–	–
	36.0000	48.0000	+0.0020	0.0000	–	–	–	–	–	–
	1219.200		**+0.076**	**0.000**	–	–	–	–	–	–
	48.0000		+0.0030	0.0000	–	–	–	–	–	–

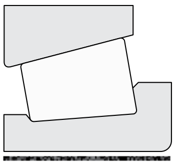
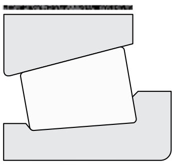


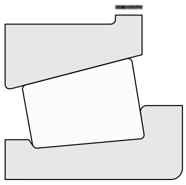
TABLE 34.
INCH SYSTEM BEARINGS - CUP O.D.

Cup O.D.			Precision Bearing Class							
Bearing Types(1)	Cup O.D.		3		0		00		000	
	Over	Incl.	Max.	Min.	Max.	Min.	Max.	Min.	Max.	Min.
	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.
TS TSF TSL TDI TDIT TDO TNA TNASW TNASWE	**0.000**	**304.800**	**+0.013**	**0.000**	**+0.013**	**0.000**	**+0.008**	**0.000**	**+0.008**	**0.000**
	0.0000	12.0000	+0.0005	0.0000	+0.0005	0.0000	+0.0003	0.0000	+0.0003	0.0000
	304.800	**609.600**	**+0.025**	**0.000**	**+0.013**	**0.000**	**+0.008**	**0.000**	**+0.008**	**0.000**
	12.0000	24.0000	+0.0010	0.0000	+0.0005	0.0000	+0.0003	0.0000	+0.0003	0.0000
	609.600	**914.400**	**+0.038**	**0.000**	–	–	–	–	–	–
	24.0000	36.0000	+0.0015	0.0000	–	–	–	–	–	–
	914.400	**1219.200**	**+0.051**	**0.000**	–	–	–	–	–	–
	36.0000	48.0000	+0.0020	0.0000	–	–	–	–	–	–
	1219.200		**+0.076**	**0.000**	–	–	–	–	–	–
	48.0000		+0.0030	0.0000	–	–	–	–	–	–



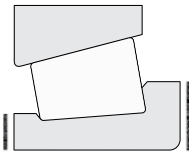
(1)Not all types and sizes are listed in this catalog. Contact your Timken representative for further information.
(2)For TSL bearings these are the normal tolerances of cone bore. However, bore size can be slightly reduced at large end due to tight fit assembly of the seal on the rib. This should not have any effect on the performance of the bearing.

TABLE 35.
INCH SYSTEM BEARINGS - CUP FLANGE O.D.



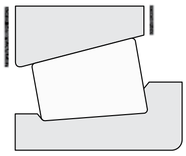
Cup Flange O.D.			Precision Bearing Class							
Bearing Types(1)	Cup O.D. Over	Incl.	3		0		00		000	
			Max.	Min.	Max.	Min.	Max.	Min.	Max.	Min.
	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.
TSF	**0.000**	**304.800**	**+0.051**	**0.000**	**+0.051**	**0.000**	**+0.051**	**0.000**	**+0.051**	**0.000**
	0.0000	12.0000	+0.0020	0.0000	+0.0020	0.0000	+0.0020	0.0000	+0.0020	0.0000
	304.800	**609.600**	**+0.076**	**0.000**	**+0.051**	**0.000**	**+0.051**	**0.000**	**+0.051**	**0.000**
	12.0000	24.0000	+0.0030	0.0000	+0.0020	0.0000	+0.0020	0.0000	+0.0020	0.0000
	609.600	**914.400**	**+0.102**	**0.000**	–	–	–	–	–	–
	24.0000	36.0000	+0.0040	0.0000	–	–	–	–	–	–
	914.400		**+0.127**	**0.000**	–	–	–	–	–	–
	36.0000		+0.0050	0.0000	–	–	–	–	–	–

TABLE 36.
INCH SYSTEM BEARINGS - CONE WIDTH



Cone Width			Precision Bearing Class							
Bearing Types(1)	Bore Over	Incl.	3		0		00		000	
			Max.	Min.	Max.	Min.	Max.	Min.	Max.	Min.
	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.
TS TSF TSL TDI TDIT TDO	All sizes		**+0.076**	**-0.254**	**+0.076**	**-0.254**	**+0.076**	**-0.254**	**+0.076**	**-0.254**
			+0.0030	-0.0100	+0.0030	-0.0100	+0.0030	-0.0100	+0.0030	-0.0100

TABLE 37.
INCH SYSTEM BEARINGS - CUP WIDTH

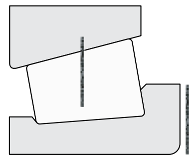


Cup Width			Precision Bearing Class							
Bearing Types(1)	Bore Over	Incl.	3		0		00		000	
			Max.	Min.	Max.	Min.	Max.	Min.	Max.	Min.
	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.
All types	All sizes		**+0.051**	**-0.254**	**+0.051**	**-0.254**	**+0.051**	**-0.254**	**+0.051**	**-0.254**
			+0.0020	-0.0100	+0.0020	-0.0100	+0.0020	-0.0100	+0.0020	-0.0100

(1)Not all types and sizes are listed in this catalog. Contact your Timken representative for further information.

TABLE 38.
INCH SYSTEM BEARINGS - CONE STAND

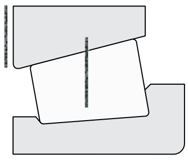
Cone Stand			Precision Bearing Class							
Bearing Types(1)	Bore		3		0		00		000	
	Over	Incl.	Max.	Min.	Max.	Min.	Max.	Min.	Max.	Min.
	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.
TS TSF TSL TDI TDIT TDO	**0.000**	**101.600**	**+0.102**	**-0.102**						
	0.0000	4.0000	+0.0040	-0.0040						
	101.600	**266.700**	**+0.102**	**-0.102**						
	4.0000	10.5000	+0.0040	-0.0040	(2)	(2)	(2)	(2)	(2)	(2)
	266.700	**304.800**	**+0.102**	**-0.102**						
	10.5000	12.0000	+0.0040	-0.0040						
	304.800	**406.400**	**+0.102**	**-0.102**	–	–	–	–	–	–
	12.0000	16.0000	+0.0040	-0.0040	–	–	–	–	–	–
	406.400		(2)	(2)	–	–	–	–	–	–
	16.0000				–	–	–	–	–	–



Cone Stand. Cone stand is a measure of the variation in cone raceway size, taper and roller diameter. This is checked by measuring the axial location of the reference surface of a master cup or other type gage with respect to the reference cone face.

TABLE 39.
INCH SYSTEM BEARINGS - CUP STAND

Cup Stand			Precision Bearing Class							
Bearing Types(1)	Bore		3		0		00		000	
	Over	Incl.	Max.	Min.	Max.	Min.	Max.	Min.	Max.	Min.
	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.
TS TSF(3) TSL TDI TDIT	**0.000**	**101.600**	**+0.102**	**-0.102**						
	0.0000	4.0000	+0.0040	-0.0040						
	101.600	**266.700**	**+0.102**	**-0.102**						
	4.0000	10.5000	+0.0040	-0.0040	(2)	(2)	(2)	(2)	(2)	(2)
	266.700	**304.800**	**+0.102**	**-0.102**						
	10.5000	12.0000	+0.0040	-0.0040						
	304.800	**406.400**	**+0.102**	**-0.102**	–	–	–	–	–	–
	12.0000	16.0000	+0.0040	-0.0040	–	–	–	–	–	–
	406.400		(2)	(2)	–	–	–	–	–	–
	16.0000				–	–	–	–	–	–



Cup Stand. Cup stand is a measure of the variation in cup I.D. size and taper. This is checked by measuring the axial location of the reference surface of a master plug or other type gage with respect to the reference face of the cup.

(1)Not all types and sizes are listed in this catalog. Please contact your Timken representative for further information.
(2)These sizes manufactured as matched assemblies only.
(3)Stand for flanged cup is measured from flange backface (seating face).

TABLE 40.
INCH SYSTEM BEARINGS - OVERALL BEARING WIDTH

Overall Bearing Width					Bearing Class							
Bearing Types[1]	Bore		O.D.		3		0		00		000	
	Over	Incl.	Over	Incl.	Max.	Min.	Max.	Min.	Max.	Min.	Max.	Min.
	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.
TS TSF[2] TSL	**0.000**	**101.600**	**–**	**–**	**+0.203**	**-0.203**	**+0.203**	**-0.203**	**+0.203**	**-0.203**	**+0.203**	**-0.203**
	0.0000	4.0000	–	–	+0.0080	-0.0080	+0.0080	-0.0080	+0.0080	-0.0080	+0.0080	-0.0080
	101.600	**304.800**	**–**	**–**	**+0.203**	**-0.203**	**+0.203**	**-0.203**	**+0.203**	**-0.203**	**+0.203**	**-0.203**
	4.0000	12.0000	–	–	+0.0080	-0.0080	+0.0080	-0.0080	+0.0080	-0.0080	+0.0080	-0.0080
	304.800	**609.600**	**0.000**	**508.000**	**+0.203**	**-0.203**	**–**	**–**	**–**	**–**	**–**	**–**
	12.0000	24.0000	0.0000	20.0000	+0.0080	-0.0080	–	–	–	–	–	–
	304.800	**609.600**	**508.000**		**+0.381**	**-0.381**	**–**	**–**	**–**	**–**	**–**	**–**
	12.0000	24.0000	20.0000		+0.0150	-0.0150	–	–	–	–	–	–
	609.600		**–**	**–**	**+0.381**	**-0.381**	**–**	**–**	**–**	**–**	**–**	**–**
	24.0000		–	–	+0.0150	-0.0150	–	–	–	–	–	–
TNA TNASW TNASWE	**0.000**	**127.000**	**–**	**–**	**+0.254**	**0.000**	**–**	**–**	**–**	**–**	**–**	**–**
	0.0000	5.0000	–	–	+0.0100	0.0000	–	–	–	–	–	–
	127.000		**–**	**–**	**+0.762**	**0.000**	**–**	**–**	**–**	**–**	**–**	**–**
	5.0000		–	–	+0.0300	0.0000	–	–	–	–	–	–
TDI TDIT TDO	**0.000**	**101.600**	**–**	**–**	**+0.406**	**-0.406**	**+0.406**	**-0.406**	**+0.406**	**-0.406**	**+0.406**	**-0.406**
	0.0000	4.0000	–	–	+0.0160	-0.0160	+0.0160	-0.0160	+0.0160	-0.0160	+0.0160	-0.0160
	101.600	**304.800**	**–**	**–**	**+0.406**	**-0.406**	**+0.406**	**-0.406**	**+0.406**	**-0.406**	**+0.406**	**-0.406**
	4.0000	12.0000	–	–	+0.0160	-0.0160	+0.0160	-0.0160	+0.0160	-0.0160	+0.0160	-0.0160
	304.800	**609.600**	**0.000**	**508.000**	**+0.406**	**-0.406**	**–**	**–**	**–**	**–**	**–**	**–**
	12.0000	24.0000	0.0000	20.0000	+0.0160	-0.0160	–	–	–	–	–	–
	304.800	**609.600**	**508.000**		**+0.762**	**-0.762**	**–**	**–**	**–**	**–**	**–**	**–**
	12.0000	24.0000	20.0000		+0.0300	-0.0300	–	–	–	–	–	–
	609.600	**–**	**–**	**–**	**+0.762**	**-0.762**	**–**	**–**	**–**	**–**	**–**	**–**
	24.0000	–	–	–	+0.0300	-0.0300	–	–	–	–	–	–

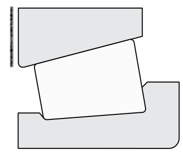
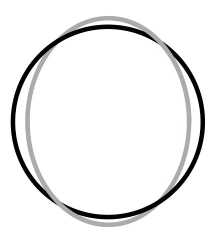


TABLE 41.
INCH SYSTEM BEARINGS - ASSEMBLED BEARING MAXIMUM RADIAL RUNOUT

Assembled Bearing Maximum Radial Runout			Bearing Class			
Bearing Types[1]	Cup O.D. Over	Incl.	3	0	00	000
	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.
TS TSF TSL TDI TDIT TDO TNA TNASW TNASWE	**0.000**	**266.700**	**0.008**	**0.004**	**0.002**	**0.001**
	0.0000	10.5000	0.0003	0.00015	0.000075	0.000040
	266.700	**304.800**	**0.008**	**0.004**	**0.002**	**0.001**
	10.5000	12.0000	0.0003	0.00015	0.000075	0.000040
	304.800	**609.600**	**0.018**	**–**	**–**	**–**
	12.0000	24.0000	0.0007	–	–	–
	609.600	**914.400**	**0.051**	**–**	**–**	**–**
	24.0000	36.0000	0.0020	–	–	–
	914.400		**0.076**	**–**	**–**	**–**
	36.0000		0.0030	–	–	–



Runout. Runout is a measure of rotational accuracy expressed by Total Indicator Reading (T.I.R.). Total displacement is measured by an instrument sensing against a moving surface, or moved with respect to a fixed surface. A radial runout measurement includes both roundness errors and the centering error of the surface that the instrument head senses against.

[1]Not all types and sizes are listed in this catalog. Please contact your Timken representative for further information.
[2]For bearing type TSF, the tolerance applies to the dimension T_1.

ENGINEERING

TOLERANCES

TABLE 42.
INCH SYSTEM BEARINGS - OUTER RACE O.D.

Outer Race O.D.			Precision Bearing Class			
Bearing Types	O.D.		3		0	
	Over	Incl.	Max.	Min.	Max.	Min.
	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.
TXR	**–**	**304.800**	**0.025**	**0.000**	**0.127**	**0.000**
	–	12.0000	0.0010	0.0000	0.0005	0.0000
	304.800	**609.600**	**0.051**	**0.000**	**0.025**	**0.000**
	12.0000	24.0000	0.0020	0.0000	0.0010	0.0000
	609.600	**914.400**	**0.076**	**0.000**	**–**	**–**
	24.0000	36.000	0.0030	0.0000	–	–
	914.400	**1219.200**	**0.102**	**0.000**	**–**	**–**
	36.0000	48.0000	0.0040	0.0000	–	–
	1219.200	**1524.000**	**0.127**	**0.000**	**–**	**–**
	48.0000	60.0000	0.0050	0.0000	–	–
	1524.000	**–**	**0.127**	**0.000**	**–**	**–**
	60.0000	–	0.0050	0.0000	–	–

TABLE 43.
INCH SYSTEM BEARINGS - INNER RACE BORE

Inner Race Bore			Precision Bearing Class			
Bearing Types	Bore		3		0	
	Over	Incl.	Max.	Min.	Max.	Min.
	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.
TXR	**–**	**76.200**	**0.013**	**0.000**	**0.013**	**0.000**
	–	3.0000	0.0005	0.0000	0.0005	0.0000
	76.200	**304.800**	**0.025**	**0.000**	**0.013**	**0.000**
	3.0000	12.0000	0.0010	0.0000	0.0005	0.0000
	304.800	**609.600**	**0.051**	**0.000**	**0.025**	**0.000**
	12.0000	24.0000	0.0020	0.0000	0.0010	0.0000
	609.600	**914.400**	**0.076**	**0.000**	**–**	**–**
	24.0000	36.0000	0.0030	0.0000	–	–
	914.400	**1219.200**	**0.102**	**0.000**	**–**	**–**
	36.0000	48.0000	0.0040	0.0000	–	–
	1219.200	**–**	**0.127**	**0.000**	**–**	**–**
	48.0000	–	0.0050	0.0000	–	–

BALL BEARINGS

The Annular Bearing Engineers' Committee has established five classes of tolerances for ball bearings, known as ABEC 1, ABEC 3, ABEC 5, ABEC 7 and ABEC 9. The highest number indicates the class with the most exacting tolerances. Every ball bearing manufactured by Timken is made to close tolerances, adhering to the established ABEC standards.

In general, these standards are equivalent to the comparable classes of tolerance established by the International Organization for Standardization, known as ISO P0 (ABEC 1), ISO P5 (ABEC 5), ISO P4 (ABEC 7) and ISO P2 (ABEC 9).

Timken manufactures a complete line of super precision ball bearings made to ABEC 7 (ISO P4) and ABEC 7/9 (ISO P4/P2) tolerances for applications involving high speeds, extreme accuracy and rigidity. The range of such equipment includes high-grade machine tools, jet engines, computer hardware, robotics and space exploration vehicles. Machine tool bearings, basically single-row construction, are available in four series, named ultra-light (9300/71900/ISO 19), extra-light (9100/7000/ISO 10), light (200/7200/ISO 02) and medium (300/7300/ISO 03), providing a considerable range in external dimension relationships.

The chart below shows the various classes of tolerances for 35.000 mm (1.3779 in.) bore size, light-series bearings (207). To meet the exacting requirements of the machine tool industry, even ABEC 9/ISO P2 tolerances do not represent the ultimate, since some special applications require even higher precision.

SYSTEM TOLERANCES

Before determining which type and classification of Timken super precision ball bearing is best suited for a particular application, it is important to explore all relevant details of the bearing characteristics, tolerances and mounting, as listed in the catalog data tables. Although cost is another factor, it is not economical to attempt the use of low-precision bearings on an application where extra-high speeds and ultra-precision bearings are required.

Timken precision bearings are manufactured to close tolerances to help assure consistent performance and interchangeability. To take full advantage of this precision product, it is expected that equally close tolerances be used in the production of mounting components (housings, shafts, spacers, etc.). Therefore, special consideration must be given to the particular details relating to proper shaft and housing fits and the housing design.

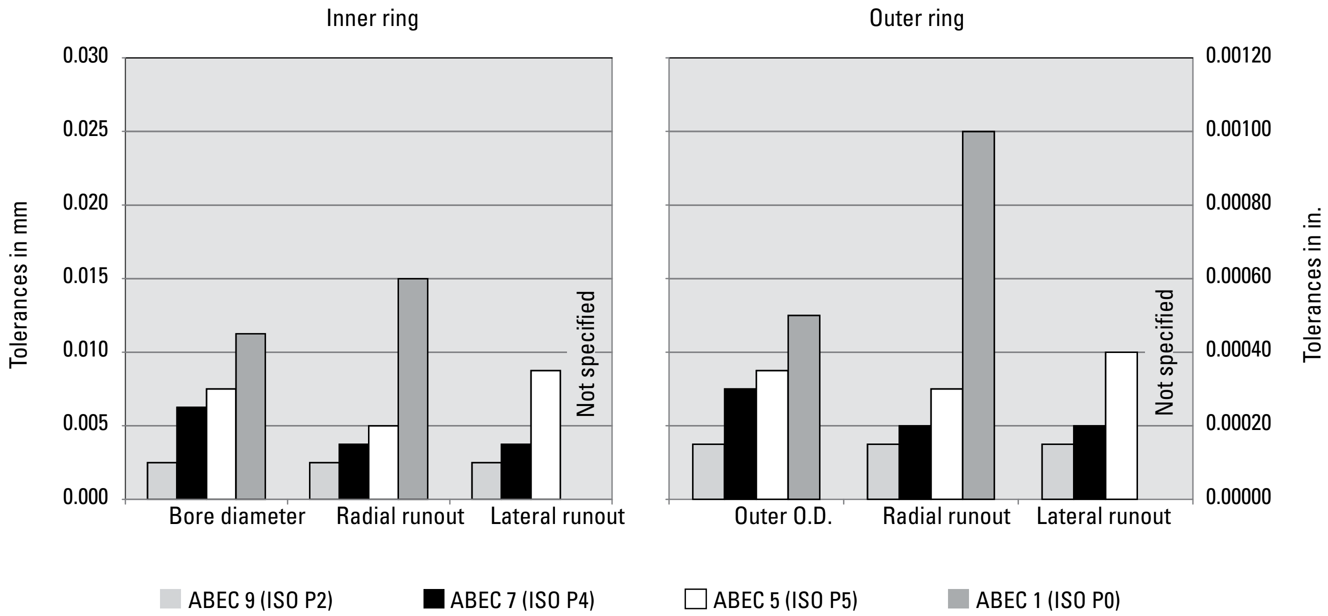


Fig. 68. Comparison of tolerances among ABEC/ISO classes (light series, 35.000 mm bore [1.3779 in.] type).

OUTER, INNER RINGS ABEC 5, 7, 9 – ISO P5, P4, P2

Values of tolerances for super precision ball bearings are shown below. This catalog lists the Timken sizes manufactured to MV (ISO P4) and MM/MMV (ISO P4S) levels.

TABLE 44 .
STANDARD ABEC/ISO TOLERANCES – INNER RING

Bearing Bore		Δ_{dmp} Bore Diameter[1] **+0.000 mm,** +0.0000 in.			V_{Bs} Width Variation			K_{ia} Raceway Radial Runout			S_{ia} Raceway Axial Runout			S_d Face Runout With Bore			Δ_{Bs} Width Inner Rings **+0.000 mm,** +0.0000 in.
		ABEC/ISO			ABEC/ISO			ABEC/ISO			ABEC/ISO			ABEC/ISO			ABEC/ISO
Over	Incl.	5/P5	7/P4[2]	9/P2	5/P5	7/P4	9/P2[2]	5/P5	7/P4	9/P2[2]	5/P5	7/P4	9/P2[2]	5/P5	7/P4	9/P2[2]	5/P5, 7/P4, 9/P2[2]
mm in.	**mm** in.	**mm** in.	**mm** in.	**mm** in.	**mm** in.	**mm** in.	**mm** in.	**mm** in.	**mm** in.	**mm** in.	**mm** in.	**mm** in.	**mm** in.	**mm** in.	**mm** in.	**mm** in.	**mm** in.
2.500	**10.000**	**-0.005**	**-0.004**	**-0.0025**	**0.005**	**0.0025**	**0.0015**	**0.004**	**0.0025**	**0.0015**	**0.007**	**0.003**	**0.0015**	**0.007**	**0.003**	**0.0015**	**-0.040**
0.0984	0.3937	-0.0002	-0.0002	-0.0001	0.0002	0.0001	0.0001	0.0002	0.0001	0.0001	0.0003	0.0001	0.00005	0.0003	0.0001	0.00005	-0.0015
10.000	**18.000**	**-0.005**	**-0.004**	**-0.0025**	**0.005**	**0.0025**	**0.0015**	**0.004**	**0.0025**	**0.0015**	**0.007**	**0.003**	**0.0015**	**0.007**	**0.003**	**0.0015**	**-0.080**
0.3937	0.7087	-0.0002	-0.0002	-0.0001	0.0002	0.0001	0.0001	0.0002	0.0001	0.0001	0.0003	0.0001	0.00005	0.0003	0.0001	0.00005	-0.0030
18.000	**30.000**	**-0.006**	**-0.005**	**-0.0025**	**0.005**	**0.0025**	**0.0015**	**0.004**	**0.003**	**0.0025**	**0.008**	**0.004**	**0.0025**	**0.008**	**0.004**	**0.0015**	**-0.120**
0.7087	1.1811	-0.0002	-0.0002	-0.0001	0.0002	0.0001	0.0001	0.0002	0.0001	0.0001	0.0003	0.0002	0.0001	0.0003	0.0002	0.00005	-0.0050
30.000	**50.000**	**-0.008**	**-0.006**	**-0.0025**	**0.005**	**0.003**	**0.0015**	**0.005**	**0.004**	**0.0025**	**0.008**	**0.004**	**0.0025**	**0.008**	**0.004**	**0.0015**	**-0.120**
1.1811	1.9685	-0.0003	-0.0002	-0.0001	0.0002	0.0001	0.0001	0.0002	0.00015	0.0001	0.0003	0.0002	0.0001	0.0003	0.0002	0.00005	-0.0050
50.000	**80.000**	**-0.009**	**-0.007**	**-0.004**	**0.006**	**0.004**	**0.0015**	**0.005**	**0.004**	**0.0025**	**0.008**	**0.005**	**0.0025**	**0.008**	**0.005**	**0.0015**	**-0.150**
1.9685	3.1496	-0.0004	-0.0003	-0.00015	0.0002	0.00015	0.0001	0.0002	0.00015	0.0001	0.0003	0.0002	0.0001	0.0003	0.0002	0.00005	-0.0060
80.000	**120.000**	**-0.010**	**-0.008**	**-0.005**	**0.007**	**0.004**	**0.0025**	**0.006**	**0.005**	**0.0025**	**0.009**	**0.005**	**0.0025**	**0.009**	**0.005**	**0.0025**	**-0.200**
3.1496	4.7244	-0.0004	-0.0003	-0.0002	0.0003	0.00015	0.0001	0.0002	0.0002	0.0001	0.00035	0.0002	0.0001	0.00035	0.0002	0.0001	-0.0080
120.000	**150.000**	**-0.013**	**-0.010**	**-0.007**	**0.008**	**0.005**	**0.0025**	**0.008**	**0.006**	**0.0025**	**0.010**	**0.007**	**0.0025**	**0.010**	**0.006**	**0.0025**	**-0.250**
4.7244	5.9055	-0.0005	-0.0004	-0.0003	0.0003	0.0002	0.0001	0.0003	0.00025	0.0001	0.0004	0.0003	0.0001	0.0004	0.00025	0.0001	-0.0100
150.000	**180.000**	**-0.013**	**-0.010**	**-0.007**	**0.008**	**0.005**	**0.004**	**0.008**	**0.006**	**0.005**	**0.010**	**0.007**	**0.005**	**0.010**	**0.006**	**0.004**	**-0.250**
5.9055	7.0866	-0.0005	-0.0004	-0.0003	0.0003	0.0002	0.00015	0.0003	0.00025	0.0002	0.0004	0.0003	0.0002	0.0004	0.00025	0.00015	-0.0100
180.000	**250.000**	**-0.015**	**-0.012**	**-0.008**	**0.010**	**0.006**	**0.005**	**0.010**	**0.008**	**0.005**	**0.013**	**0.008**	**0.005**	**0.011**	**0.007**	**0.005**	**-0.300**
7.0866	9.8425	-0.0006	-0.0004	-0.0003	0.0004	0.00025	0.0002	0.0004	0.0003	0.0002	0.0005	0.0003	0.0002	0.00045	0.0003	0.0002	-0.0120

The tolerances in this table are in conformance with ANSI ABMA Standard 20 - 1996.

[1] d_{min} (the smallest single diameter of a bore) and d_{max} (the largest single diameter of a bore) may fall outside limits shown. $\frac{d_{min} + d_{max}}{2}$ must be within bore diameter tabulated.
[2] MM/MMV (P4S) tolerance.

ABMA / ISO Symbols - Inner Ring

- Δ_{dmp} Single plane mean bore diameter deviation from basic bore diameter, e.g., bore tolerance for a basically tapered bore, Δ_{dmp} refers only to the theoretical small bore end of the bore
- K_{ia} Radial runout of assembled bearing inner ring, e.g., radial runout of raceway
- V_{Bs} Inner ring width variation, e.g., parallelism
- S_d Inner ring reference face runout with bore, e.g., squareness - bore to face
- S_{ia} Axial runout of assembled bearing inner ring, e.g., lateral (axial) runout of raceway
- Δ_{Bs} Inner ring width deviation from basic, e.g., width tolerance

WIDTH TOLERANCES

The width tolerances for individual inner and outer rings are shown in the tables below. To allow for the preload grinding on bearings for various preloads, the total width tolerances of duplex sets are as shown in the table to the right. The total width tolerance is proportional to the number of bearings. Note the Timken values are significantly tighter than ABMA/ISO requirements.

TABLE 45.
PRELOADED DUPLEX SET WIDTH TOLERANCE

Nominal Bore		Width Tolerance (Duplex Set)			
		ABMA/ISO		Timken	
Over	Incl.	Max.	Min.	Max.	Min.
mm in.	mm in.	mm in.	mm in.	mm in.	mm in.
0.000 0.0000	**80.000** 3.1496	**0.000** 0.0000	**-0.500** -0.0196	**0.000** 0.0000	**-0.250** -0.0100
80.000 3.1496	**180.000** 7.0866	**0.000** 0.0000	**-0.760** -0.0300	**0.000** 0.0000	**-0.250** -0.0100
180.000 7.0866	**250.000** 9.8425	**0.000** 0.0000	**-1.000** -0.0394	**0.000** 0.0000	**-0.250** -0.0100

TABLE 46.
STANDARD ABEC/ISO TOLERANCES – OUTER RING

Bearing O.D.		Δ_{Dmp} Outside Diameter[1] **+0.000 mm**, +0.0000 in.			V_{Cs} Width Variation			K_{ea} Raceway Radial Runout			S_{ea} Raceway Axial Runout			S_D Face Runout With O.D.			Δ_{Cs} Width Outer Rings **+0.000 mm**, +0.0000 in.
		ABEC/ISO			ABEC/ISO			ABEC/ISO			ABEC/ISO			ABEC/ISO			ABEC/ISO
Over	Incl.	5/P5	7/P4[3]	9/P2	5/P5	7/P4	9/P2[3]	5/P5	7/P4	9/P2[3]	5/P5	7/P4	9/P2[3]	5/P5	7/P4	9/P2[3]	5/P5, 7/P4, 9/P2[3]
mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.
6.000 0.2362	**18.000** 0.7087	**-0.005** -0.0002	**-0.004** -0.00015	**-0.0025** -0.0001	**0.005** 0.0002	**0.0025** 0.0001	**0.0015** 0.00005	**0.005** 0.0002	**0.003** 0.0001	**0.0015** 0.00005	**0.008** 0.0003	**0.005** 0.0002	**0.0015** 0.00005	**0.008** 0.0003	**0.004** 0.00015	**0.0015** 0.00005	
18.000 0.7087	**30.000** 1.1811	**-0.006** -0.00025	**-0.005** -0.0002	**-0.004** -0.00015	**0.005** 0.0002	**0.0025** 0.0001	**0.0015** 0.00005	**0.006** 0.00025	**0.004** 0.00015	**0.0025** 0.0001	**0.008** 0.0003	**0.005** 0.0002	**0.0025** 0.0001	**0.008** 0.0003	**0.004** 0.00015	**0.0015** 0.00005	
30.000 1.1811	**50.000** 1.9685	**-0.007** -0.0003	**-0.006** -0.0002	**-0.004** -0.00015	**0.005** 0.0002	**0.0025** 0.0001	**0.0015** 0.00005	**0.007** 0.0003	**0.005** 0.0002	**0.0025** 0.0001	**0.008** 0.0003	**0.005** 0.0002	**0.0025** 0.0001	**0.008** 0.0003	**0.004** 0.00015	**0.0015** 0.00005	
50.000 1.9685	**80.000** 3.1496	**-0.009** -0.00035	**-0.007** -0.0003	**-0.004** -0.00015	**0.006** 0.00025	**0.003** 0.0001	**0.0015** 0.00005	**0.008** 0.0003	**0.005** 0.0002	**0.004** 0.00015	**0.010** 0.0004	**0.005** 0.0002	**0.004** 0.00015	**0.008** 0.0003	**0.004** 0.00015	**0.0015** 0.00005	
80.000 3.1496	**120.000** 4.7244	**-0.010** -0.0004	**-0.008** -0.0003	**-0.005** -0.0002	**0.008** 0.0003	**0.004** 0.00015	**0.0025** 0.0001	**0.010** 0.0004	**0.006** 0.00025	**0.005** 0.0002	**0.011** 0.00045	**0.006** 0.00025	**0.005** 0.0002	**0.009** 0.00035	**0.005** 0.0002	**0.0025** 0.0001	(2)
120.000 4.7244	**150.000** 5.9055	**-0.011** -0.00045	**-0.009** -0.00035	**-0.005** -0.0002	**0.008** 0.0003	**0.005** 0.0002	**0.0025** 0.0001	**0.011** 0.00045	**0.007** 0.0003	**0.005** 0.0002	**0.013** 0.0005	**0.007** 0.0003	**0.005** 0.0002	**0.010** 0.0004	**0.005** 0.0002	**0.0025** 0.0001	
150.000 5.9055	**180.000** 7.0866	**-0.013** -0.0005	**-0.010** -0.0004	**-0.007** -0.0003	**0.008** 0.0003	**0.005** 0.0002	**0.0025** 0.0001	**0.013** 0.0005	**0.008** 0.0003	**0.005** 0.0002	**0.014** 0.00055	**0.008** 0.0003	**0.005** 0.0002	**0.010** 0.0004	**0.005** 0.0002	**0.0025** 0.0001	
180.000 7.0866	**250.000** 9.8425	**-0.015** -0.0006	**-0.011** -0.00045	**-0.008** -0.0003	**0.010** 0.0004	**0.007** 0.0003	**0.004** 0.00015	**0.015** 0.0006	**0.010** 0.0004	**0.007** 0.0003	**0.015** 0.0006	**0.010** 0.0004	**0.007** 0.0003	**0.011** 0.00045	**0.007** 0.0003	**0.004** 0.00015	
250.000 9.8425	**315.000** 12.4016	**-0.018** -0.0007	**-0.013** -0.0005	**-0.008** -0.0003	**0.011** 0.00045	**0.007** 0.0003	**0.005** 0.0002	**0.018** 0.0007	**0.011** 0.00045	**0.007** 0.0003	**0.018** 0.0007	**0.010** 0.0004	**0.007** 0.0003	**0.013** 0.0005	**0.008** 0.0003	**0.005** 0.0002	
315.000 12.4016	**400.000** 15.7480	**-0.020** -0.0008	**-0.015** -0.0006	**-0.010** -0.0004	**0.013** 0.0005	**0.008** 0.0003	**0.007** 0.0003	**0.020** 0.0008	**0.013** 0.0005	**0.008** 0.0003	**0.020** 0.0008	**0.013** 0.0005	**0.008** 0.0003	**0.013** 0.0005	**0.010** 0.0004	**0.007** 0.0003	

The tolerances in this table are in conformance with ANSI ABMA Standard 20 - 1996.

(1)D_{min} (the smallest single diameter of an O.D.) and D_{max} (the largest single diameter of an O.D.) may fall outside limits shown. $\frac{D_{min} + D_{max}}{2}$ must be within outside diameter tabulated.

(2)Identical to Δ_{Bs} of inner ring of same bearing.

(3)MM/MMV (P4S) tolerance

ABMA / ISO Symbols - Outer Ring

- Δ_{Dmp} Single plane mean outside diameter deviation from basic outside diameter, e.g., O.D. tolerance
- K_{ea} Radial runout of assembled bearing outer ring, e.g., radial runout of raceway
- V_{Cs} Outer ring width variation, e.g., parallelism
- S_D Outer ring reference face runout with O.D., e.g., squareness O.D. to face
- S_{ea} Axial runout of assembled bearing outer ring, e.g., lateral (axial) runout of raceway
- Δ_{Cs} Outer ring width deviation from basic, e.g., width tolerance

FITTING PRACTICES

GENERAL GUIDELINES FOR TAPERED ROLLER BEARINGS

The design of a tapered roller bearing permits the setting to be achieved during installation (or during running when using a Hydra-Rib), irrespective of the inner and outer ring fits on shaft and housing. This allows the use of the widest possible machining tolerances for shaft and housing and the use of the best possible fits for the inner and outer rings to match the duty of the bearing.

The fitting practice will depend upon the following parameters:

- Precision class of the bearing.
- Type of layout.
- Type and direction of loads.
- Running conditions (vibrations, high speeds).
- Shaft and housing sections and materials.
- Mounting and setting conditions.

Certain table fits may not be adequate for light shaft and housing sections, shafts other than steel, nonferrous housings, critical operation conditions such as high speed, unusual thermal or loading conditions or a combination thereof. Also assembly procedures and the means and ease of obtaining the bearing setting may require special fits. In these cases, experience should be used as a guideline or your Timken representative should be consulted for review and suggestions.

Precision class bearings should be mounted on shafts and in housings which are similarly finished to at least the same precision limits as the bearing bore and O.D. High-quality surface finishes should also be provided.

In the machine tool industry, where almost 100 percent of cases are rotating shaft applications, the general rule is to tight-fit both the inner and outer rings for simple layouts to eliminate any undesirable radial clearance.

Tapered roller bearing envelope tolerances can be adjusted to the needs of a specific application.

NON-FERROUS HOUSINGS

Care should be taken when pressing cups into aluminum or magnesium housings to avoid metal pick up. This may result in unsatisfactory fits, backing, and alignment from debris trapped between the cup and backing shoulder. Preferably, the cup should be frozen or the housing heated, or both, during assembly. Also, a special lubricant may be used to ease assembly. In some cases, cups are mounted in steel inserts which are attached to the aluminum or magnesium housings. Table fits may then be used. Where the cup is fitted directly into an aluminum housing, it is suggested that a minimum tight fit of 1.0 μm per mm (0.0010 in. per in.) of cup outside diameter be used. For a magnesium housing, a minimum tight fit of 1.5 μm per mm (0.0015 in. per in.) of cup outside diameter is suggested.

PRECISION CLASS TAPERED ROLLER BEARINGS

HYDRA-RIB

The Hydra-Rib cup is designed to be mounted either as a flanged cup or shouldered against the cup backface. The 50.000 mm (1.968 in.) bore assembly does not have a flanged cup and the cup backface must be mounted against the housing shoulder.

SUGGESTED FITTING GUIDELINES FOR FERROUS SHAFT AND HOUSING

For heavy loads, high speed or shock, contact your Timken representative for further information.

TABLE 47.
SHAFT O.D. - CLASS C AND CLASS B

Bearing Bore Range		Class C				Class B			
Over	Incl.	Bearing Bore Tolerance	Symbol	Shaft O.D. Deviation	Resultant Fit	Bearing Bore Tolerance	Symbol	Shaft O.D. Deviation	Resultant Fit
mm in.	mm in.	mm in.		mm in.	mm in.	mm in.		mm in.	mm in.
30.000	**50.000**	**-0.010**	k5	**+0.013**	**0.023T**	**-0.08**	k5	**+0.013**	**0.021T**
		0.000		**+0.002**	**0.002T**	**0.000**		**+0.002**	**0.002T**
1.1811	1.9685	-0.0004		+0.0005	0.0009T	-0.0003		+0.0005	0.0008T
		0.0000		+0.0001	0.0001T	0.0000		+0.0001	0.0001T
50.000	**80.000**	**-0.012**	k5	**+0.015**	**0.027T**	**-0.009**	k5	**+0.015**	**0.024T**
		0.000		**+0.002**	**0.002T**	**0.000**		**+0.002**	**0.002T**
1.9685	3.1496	-0.0005		+0.0006	0.0011T	-0.0004		+0.0006	0.0010T
		0.0000		+0.0001	0.0001T	0.0000		+0.0001	0.0001T
80.000	**120.000**	**-0.015**	k5	**+0.018**	**0.033T**	**-0.010**	k5	**+0.018**	**0.028T**
		0.000		**+0.003**	**0.003T**	**0.000**		**+0.003**	**0.003T**
3.1496	4.7244	-0.0006		+0.0007	0.0013T	-0.0004		+0.0007	0.0011T
		0.0000		+0.0001	0.0001T	0.0000		+0.0001	0.0001T
120.000	**180.000**	**-0.018**	k5	**+0.021**	**0.039T**	**-0.013**	k5	**+0.021**	**0.034T**
		0.000		**+0.003**	**0.003T**	**0.000**		**+0.003**	**0.003T**
4.7244	7.0866	-0.0007		+0.0008	0.0015T	-0.0005		+0.0008	0.0013T
		0.0000		+0.0001	0.0001T	0.0000		+0.0001	0.0001T
180.000	**250.000**	**-0.022**	k5	**+0.024**	**0.046T**	**-0.015**	k5	**+0.024**	**0.039T**
		0.000		**+0.004**	**0.004T**	**0.000**		**+0.004**	**0.004T**
7.0866	9.8425	-0.0009		+0.0010	0.0018T	-0.0006		+0.0010	0.0016T
		0.0000		+0.0002	0.0002T	0.0000		+0.0002	0.0002T

Deviation from nominal (maximum) bearing bore and resultant fit.
T= Tight
L = Loose

TABLE 48.
HOUSING BORE - CLASS C AND CLASS B

Bearing O.D. Range		Class C				Class B			
Over	Incl.	Bearing O.D. Tolerance	Symbol	Housing Bore Deviation	Resultant Fit	Bearing O.D. Tolerance	Symbol	Housing Bore Deviation	Resultant Fit
mm in.	mm in.	mm in.		mm in.	mm in.	mm in.		mm in.	mm in.
80.000	**120.000**	**0.000**	M5	**-0.023**	**0.023T**	**0.000**	M5	**-0.023**	**0.023T**
		-0.013		**-0.008**	**0.005L**	**-0.010**		**-0.008**	**0.002L**
3.1496	4.7244	0.0000		-0.0009	0.0009T	0.000 0		-0.0009	0.0009T
		-0.0005		-0.0003	0.0002L	-0.0004		-0.0003	0.0001L
120.000	**150.000**	**0.000**	M5	**-0.027**	**0.027T**	**0.000**	M5	**-0.027**	**0.027T**
		-0.015		**-0.009**	**0.006L**	**-0.011**		**-0.009**	**0.002L**
4.7244	5.9055	0.0000		-0.0011	0.0011T	0.0000		-0.0011	0.0011T
		-0.0006		-0.0004	0.0002	-0.0004		-0.0004	0.0001L
150.000	**180.000**	**0.000**	M5	**-0.027**	**0.027T**	**0.000**	M5	**-0.027**	**0.027T**
		-0.018		**-0.010**	**0.009L**	**-0.013**		**-0.009**	**0.004L**
5.9055	7.0866	0.0000		-0.0011	0.0011T	0.0000		-0.0012	0.0011T
		-0.0007		-0.0004	0.0004L	-0.0005		-0.0004	0.0002L
180.000	**250.000**	**0.000**	M5	**-0.031**	**0.031T**	**0.000**	M5	**-0.031**	**0.031T**
		-0.020		**-0.011**	**0.009L**	**-0.015**		**-0.011**	**0.004L**
7.0866	9.8425	0.0000		-0.0012	0.0012T	0.0000		-0.0012	0.0012T
		-0.0008		-0.0004	0.0004L	-0.0006		-0.0004	0.0002L
250.000	**350.000**	**0.000**	M5	**-0.036**	**0.036T**	–	–	–	–
		-0.025		**-0.013**	**0.012L**	–	–	–	–
9.8425	10.4331	0.0000		-0.0014	0.0014T	–	–	–	–
		-0.0010		-0.0005	0.0005L	–	–	–	–

Deviation from nominal (maximum) bearing bore and resultant fit.
T= Tight
L = Loose

PRECISION CLASS TAPERED ROLLER BEARINGS (EXCEPT HYDRA-RIB AND TXR BEARINGS)

SUGGESTED FITTING GUIDELINES FOR FERROUS SHAFT AND HOUSING

For heavy loads, high speed or shock, contact your Timken representative for further information.

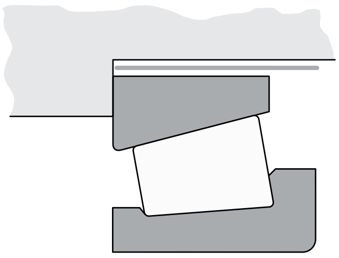


TABLE 49.
HOUSING BORE METRIC BEARINGS (ISO AND J PREFIX) - CLASS C

Bearing O.D.			Class C								
Range		Tolerance	Non-Adjustable Or In Carrier			Floating			Adjustable		
Over	Incl.		Symbol	Housing Bore Deviation	Resultant Fit	Symbol	Housing Bore Deviation	Resultant Fit	Symbol	Housing Bore Deviation	Resultant Fit
mm in.	mm in.	mm in.		mm in.	mm in.		mm in.	mm in.		mm in.	mm in.
18.000	**30.000**	**0.000**	N5	**-0.021**	**0.021T**	G5	**+0.007**	**0.007L**	K5	**-0.008**	**0.008T**
		-0.008		**-0.012**	**0.004T**		**+0.016**	**0.024L**		**+0.001**	**0.009L**
0.7087	1.1811	0.0000		-0.0008	0.0008T		+0.0003	0.0003L		-0.0003	0.0003T
		-0.0003		-0.0004	0.0002T		+0.0006	0.0009L		+0.00004	0.0004L
30.000	**50.000**	**0.000**	N5	**-0.024**	**0.024T**	G5	**+0.009**	**0.009L**	K5	**-0.009**	**0.009T**
		-0.009		**-0.013**	**0.004T**		**+0.020**	**0.029L**		**+0.002**	**0.011L**
1.1811	1.9685	0.0000		-0.0009	0.0009T		+0.00035	0.0004L		-0.0004	0.0003T
		-0.0004		-0.0005	0.0002T		+0.0008	0.0011L		+0.0001	0.0004L
50.000	**80.000**	**0.000**	N5	**-0.028**	**0.028T**	G5	**+0.010**	**0.010L**	K5	**-0.010**	**0.010T**
		-0.011		**-0.015**	**0.004T**		**+0.023**	**0.034L**		**+0.003**	**0.014L**
1.9685	3.1496	0.0000		-0.0011	0.0011T		+0.0004	0.0004L		-0.0004	0.0004T
		-0.00045		-0.0006	0.0002T		+0.0009	0.0013L		+0.0001	0.0006L
80.000	**120.000**	**0.000**	N5	**-0.033**	**0.033T**	G5	**+0.012**	**0.012L**	K5	**-0.013**	**0.013T**
		-0.013		**-0.018**	**0.005T**		**+0.027**	**0.040L**		**+0.002**	**0.015L**
3.1496	4.7244	0.0000		-0.0013	0.0013T		+0.0005	0.0008L		-0.0005	0.0005T
		-0.0005		-0.0007	0.0002T		+0.0011	0.0016L		+0.0001	0.0006L
120.000	**150.000**	**0.000**	N5	**-0.039**	**0.039T**	G5	**+0.014**	**0.014L**	K5	**-0.015**	**0.015T**
		-0.015		**-0.021**	**0.006T**		**+0.032**	**0.047L**		**+0.003**	**0.018L**
4.7244	5.9055	0.0000		-0.0015	0.0015T		+0.0006	0.0006L		-0.0006	0.0006T
		-0.0006		-0.0008	0.0002T		+0.0013	0.0019L		+0.0001	0.0007L
150.000	**180.000**	**0.000**	N5	**-0.039**	**0.039T**	G5	**+0.014**	**0.014L**	K5	**-0.015**	**0.015T**
		-0.018		**-0.021**	**0.003T**		**+0.032**	**0.050L**		**+0.003**	**0.021L**
5.9055	7.0866	0.0000		-0.0015	0.0015T		+0.0006	0.0006L		-0.0006	0.0006T
		-0.0007		-0.0008	0.0001T		+0.0013	0.0020T		+0.0001	0.0008L
180.000	**250.000**	**0.000**	N5	**-0.045**	**0.045T**	G5	**+0.015**	**0.015L**	K5	**-0.018**	**0.018T**
		-0.020		**-0.025**	**0.005T**		**+0.035**	**0.055L**		**+0.002**	**0.027L**
7.0866	9.8425	0.0000		-0.0018	0.0018T		+0.0006	0.0006L		-0.0007	0.0007T
		-0.0008		-0.0010	0.0002T		+0.0014	0.0022L		+0.0001	0.0011L
250.000	**315.000**	**0.000**	N5	**-0.050**	**0.050T**	G5	**+0.017**	**0.017L**	K5	**-0.020**	**0.020T**
		-0.025		**-0.027**	**0.002T**		**+0.040**	**0.065L**		**+0.003**	**0.028L**
9.8425	12.4016	0.0000		-0.0020	0.0020T		+0.0007	0.0007L		-0.0008	0.0008T
		-0.0010		-0.0011	0.0001T		+0.0016	0.0023L		+0.0001	0.0011L

Deviation from nominal (maximum) bearing bore and resultant fit.
T= Tight
L = Loose

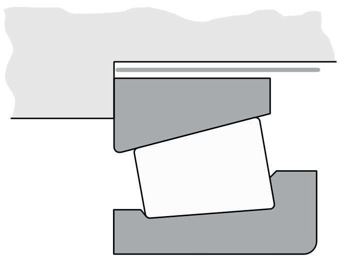


TABLE 50.
HOUSING BORE METRIC BEARINGS (ISO AND J PREFIX) - CLASS B

Bearing O.D.			Class B								
Range		Tolerance	Non-Adjustable Or In Carrier			Floating			Adjustable		
Over	Incl.		Symbol	Housing Bore Deviation	Resultant Fit	Symbol	Housing Bore Deviation	Resultant Fit	Symbol	Housing Bore Deviation	Resultant Fit
mm in.	mm in.	mm in.		mm in.	mm in.		mm in.	mm in.		mm in.	mm in.
18.000	**30.000**	**0.000**	M5	**-0.014**	**0.014T**	G5	**+0.007**	**0.007L**	K5	**-0.008**	**0.008T**
		-0.006		**-0.005**	**0.001L**		**+0.016**	**0.022L**		**+0.001**	**0.007L**
0.7087	1.1811	0.0000		-0.0006	0.0006T		+0.0003	0.0003L		-0.0003	0.0003T
		0.0002		-0.0002	0.00004L		+0.0006	0.0009L		+0.00004	0.0003L
30.000	**50.000**	**0.000**	M5	**-0.016**	**0.016T**	G5	**+0.009**	**0.009L**	K5	**-0.009**	**0.009T**
		-0.007		**-0.005**	**0.002L**		**+0.020**	**0.027L**		**+0.002**	**0.009L**
1.1811	1.9685	0.0000		-0.0006	0.0006T		+0.0004	0.0004L		-0.0004	0.0004T
		-0.0003		-0.0002	0.0001L		+0.0008	0.0011L		+0.0001	0.0004L
50.000	**80.000**	**0.000**	M5	**-0.019**	**0.019T**	G5	**+0.010**	**0.010L**	K5	**-0.010**	**0.010T**
		-0.009		**-0.006**	**0.003L**		**+0.023**	**0.032L**		**+0.003**	**0.012L**
1.9685	3.1496	0.000		-0.0008	0.0008T		+0.0004	0.0004L		-0.0004	0.0004T
		-0.0004		-0.0002	0.0001L		+0.0009	0.0013L		+0.0001	0.0005L
80.000	**120.000**	**0.000**	M5	**-0.023**	**0.023T**	G5	**+0.012**	**0.012L**	K5	**-0.013**	**0.013T**
		-0.010		**-0.008**	**0.002L**		**+0.027**	**0.037L**		**+0.002**	**0.012L**
3.1496	4.7244	0.000		0.0009	0.0009T		+0.0005	0.0004L		-0.0005	0.0005T
		-0.0004		0.0003	0.0001L		+0.0011	0.0015L		+0.0001	0.0005L
120.000	**150.000**	**0.000**	M5	**-0.027**	**0.027T**	G5	**+0.014**	**0.014L**	K5	**-0.015**	**0.015T**
		-0.011		**-0.009**	**0.002L**		**+0.032**	**0.043L**		**+0.003**	**0.012L**
4.7244	5.9055	0.000		-0.0011	0.0011T		+0.0006	0.0006L		-0.0006	0.0006T
		-0.0004		-0.0004	0.0001L		+0.0013	0.0017L		+0.0001	0.0005L
150.000	**180.000**	**0.000**	M5	**-0.027**	**0.027T**	G5	**+0.014**	**0.014L**	K5	**-0.015**	**0.015T**
		-0.013		**-0.009**	**0.004L**		**+0.032**	**0.045L**		**+0.003**	**0.016L**
5.9055	7.0866	0.000		-0.0011	0.0011T		+0.0006	0.0006L		-0.0006	0.0006T
		-0.0005		-0.0004	0.0002L		+0.0013	0.0018L		+0.0001	0.0006L
180.000	**250.000**	**0.000**	M5	**-0.031**	**0.031T**	G5	**+0.015**	**0.015L**	K5	**-0.018**	**0.018T**
		-0.015		**-0.011**	**0.004L**		**+0.035**	**0.050L**		**+0.002**	**0.017L**
7.0866	9.8425	0.000		-0.0012	0.0012T		+0.0006	0.0006L		-0.0007	0.0007T
		0.0006		-0.0004	0.0002L		+0.0014	0.0020L		+0.0001	0.0007L
250.000	**315.000**	**0.000**	M5	**-0.036**	**0.036T**	G5	**+0.017**	**0.017L**	K5	**-0.020**	**0.020T**
		-0.018		**-0.013**	**0.005L**		**+0.040**	**0.058L**		**+0.003**	**0.021L**
9.8425	12.4016	0.000		-0.0014	0.0014T		+0.0007	0.0007L		-0.0008	0.0008T
		-0.0007		-0.0005	0.0002L		+0.0016	0.0023L		+0.0001	0.0008L

Deviation from nominal (maximum) bearing bore and resultant fit.
T= Tight
L = Loose

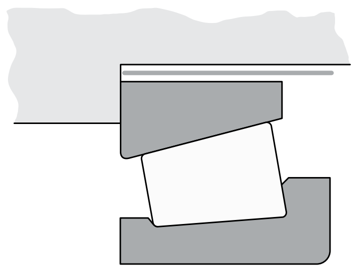


TABLE 51.
HOUSING BORE INCH BEARINGS (ISO AND J PREFIX) - CLASS 3 AND 0

Bearing O.D.			Class 3 and 0(1)					
Range		Tolerance	Non-Adjustable Or In Carrier		Floating		Adjustable	
Over	Incl.		Housing Bore Deviation	Resultant Fit	Housing Bore Deviation	Resultant Fit	Housing Bore Deviation	Resultant Fit
mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.
–	**152.400**	**+0.013**	**-0.013**	**0.026T**	**+0.025**	**0.012L**	**0.000**	**0.013T**
		0.000	**0.000**	**0.000**	**+0.038**	**0.038L**	**+0.013**	**0.013L**
–	6.0000	+0.0005	-0.0005	0.0010T	+0.0010	0.0005L	0.000	0.0005T
		0.0000	0.0000	0.0000	+0.0015	0.0015L	+0.0005	0.0005L
152.400	**304.800**	**+0.013**	**-0.025**	**0.038T**	**+0.025**	**0.012L**	**0.000**	**0.013T**
		0.000	**0.000**	**0.000**	**+0.038**	**0.038L**	**+0.025**	**0.025L**
6.0000	12.0000	+0.0005	-0.0010	0.0015T	+0.0010	0.0005L	0.0000	0.0005T
		0.0000	0.0000	0.0000	+0.0015	0.0015L	+0.0010	0.0010L
304.800	**609.600**	**+0.025**	**-0.025**	**0.050T**	**+0.038**	**0.013L**	**0.000**	**0.025T**
		0.000	**0.000**	**0.000**	**+0.064**	**0.064L**	**+0.025**	**0.025L**
12.0000	24.0000	+0.0010	-0.0010	0.0020T	+0.0015	0.0005L	0.0000	0.0010T
		0.0000	0.0000	0.0000	+0.0025	0.0025L	+0.0010	0.0010L
609.600	**914.400**	**+0.038**	**-0.038**	**0.076T**	**+0.051**	**0.013L**	**,0000**	**0.038T**
		0.000	**0.000**	**0.000**	**+0.089**	**0.089L**	**+0.038**	**0.038L**
24.0000	36.0000	+0.0015	-0.0015	0.0030T	+0.0020	0.0005L	0.0000	0.0015T
		0.0000	0.0000	0.0000	+0.0035	0.0035L	+0.0015	0.0015L

(1)Class 0 made only to 304.800 mm (12.0000 in.) O.D.
Deviation from nominal (maximum) bearing bore and resultant fit.
T= Tight
L = Loose

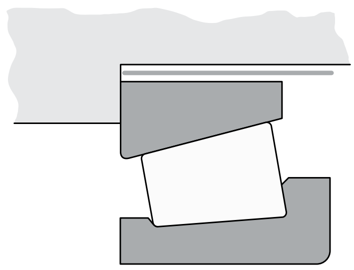


TABLE 52.
HOUSING BORE INCH BEARINGS - CLASS A AND AA

Bearing O.D.			Class A and AA					
Range		Tolerance	Non-Adjustable Or In Carrier		Floating		Adjustable	
Over	Incl.		Housing Bore Deviation	Resultant Fit	Housing Bore Deviation	Resultant Fit	Housing Bore Deviation	Resultant Fit
mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.
0.000	**315.000**	**-0.000** **-0.008**	**-0.016** **-0.008**	**0.016T** **0.000**	**+0.008** **+0.016**	**0.008L** **0.024L**	**-0.008** **-0.000**	**0.008T** **0.008L**
0.0000	12.4016	-0.0000 -0.0003	-0.0006 -0.0003	0.0006T 0.0000	+0.0003 +0.0006	0.0003L 0.0009L	-0.0003 -0.0000	0.0003T 0.0003L

Deviation from nominal (maximum) bearing bore and resultant fit.
T= Tight
L = Loose

TABLE 53.
HOUSING BORE INCH BEARINGS - CLASS 00 AND 000

Bearing O.D.			Class 00 and 000					
Range		Tolerance	Non-Adjustable Or In Carrier		Floating		Adjustable	
Over	Incl.		Housing Bore Deviation	Resultant Fit	Housing Bore Deviation	Resultant Fit	Housing Bore Deviation	Resultant Fit
mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.
0.000	**304.800**	**+0.008** **0.000**	**-0.008** **0.000**	**0.016T** **0.000**	**+0.015** **+0.023**	**0.007L** **0.023L**	**0.000** **+0.008**	**0.008T** **0.008L**
0.0000	12.0000	+0.0003 0.0000	-0.0003 0.0000	0.0006T 0.0000	+0.0006 +0.0009	0.0003L 0.0009L	0.0000 +0.0003	0.0003T 0.0003L

Deviation from nominal (maximum) bearing bore and resultant fit.
T= Tight
L = Loose

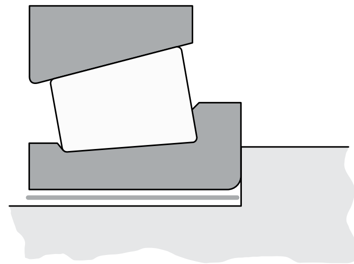


TABLE 54.
SHAFT O.D. METRIC BEARINGS (ISO AND J PREFIX) - CLASS C AND CLASS B

Bearing Bore Range		Class C				Class B			
Over	Incl.	Bearing Bore Tolerance	Symbol	Shaft O.D. Deviation	Resultant Fit	Bearing Bore Tolerance	Symbol	Shaft O.D. Deviation	Resultant Fit
mm in.	mm in.	mm in.		mm in.	mm in.	mm in.		mm in.	mm in.
10.000	**18.000**	**-0.007**	k5	**+0.009**	**0.016T**	**-0.005**	k5	**+0.009**	**0.014T**
		0.000		**+0.001**	**0.001T**	**0.000**		**+0.001**	**0.001T**
0.3937	0.7087	-0.0003		+0.0004	0.0006T	-0.0002		+0.0004	0.0006T
		0.0000		+0.00004	0.00004T	0.0000		+0.00004	0.00004T
18.000	**30.000**	**-0.008**	k5	**+0.011**	**0.019T**	**-0.006**	k5	**+0.011**	**0.017T**
		0.000		**+0.002**	**0.002T**	**0.000**		**+0.002**	**0.002T**
0.7087	1.1811	-0.0003		+0.0005	0.0007T	-0.0002		+0.0004	0.0007T
		0.0000		+0.0001	0.0001T	0.0000		+0.0001	0.0001T
30.000	**50.000**	**-0.010**	k5	**+0.013**	**0.023T**	**-0.008**	k5	**+0.013**	**0.021T**
		0.000		**+0.002**	**0.002T**	**0.000**		**+0.002**	**0.002T**
1.1811	1.9685	-0.0004		+0.0005	0.0009T	-0.0003		+0.0005	0.0008T
		0.0000		+0.0001	0.0001T	0.0000		+0.0001	0.0001T
50.000	**80.000**	**-0.012**	k5	**+0.015**	**0.027T**	**-0.009**	k5	**+0.015**	**0.024T**
		0.000		**+0.003**	**0.002T**	**0.000**		**+0.002**	**0.002T**
1.9685	3.1496	-0.0005		+0.0006	0.0011T	-0.0004		+0.0006	0.0009T
		0.0000		+0.0001	0.0001T	0.0000		+0.0001	0.0001T
80.000	**120.000**	**-0.015**	k5	**+0.018**	**0.033T**	**-0.010**	k5	**+0.018**	**0.028T**
		0.000		**+0.003**	**0.003T**	**0.000**		**+0.003**	**0.003T**
3.1496	4.7244	-0.0006		+0.0007	0.0013T	-0.0004		+0.0007	0.0011T
		0.0000		+0.0001	0.0001T	0.0000		+0.0001	0.0001T
120.000	**180.000**	**-0.018**	k5	**+0.021**	**0.039T**	**-0.013**	k5	**+0.021**	**0.034T**
		0.000		**+0.003**	**0.003T**	**0.000**		**+0.003**	**0.003T**
4.7244	7.0866	-0.0007		+0.0008	0.0015T	-0.0005		+0.0008	0.0013T
		0.0000		+0.0001	0.0001T	0.0000		+0.0001	0.0001T
180.000	**250.000**	**-0.022**	k5	**+0.024**	**0.046T**	**-0.015**	k5	**+0.024**	**0.039T**
		0.000		**+0.004**	**0.004T**	**0.000**		**+0.004**	**0.004T**
7.0866	9.8425	-0.0009		+0.0009	0.0018T	-0.0006		+0.0009	0.0015T
		0.0000		+0.0002	0.0002T	0.0000		+0.0002	0.0002T
250.000	**315.000**	**-0.022**	k5	**+0.027**	**0.049T**	**-0.015**	k5	**+0.027**	**0.042T**
		0.000		**+0.004**	**0.004T**	**0.000**		**+0.004**	**0.004T**
9.8425	12.4016	-0.0009		+0.0011	0.0019T	-0.0006		+0.0011	0.0017T
		0.0000		+0.0002	0.0002T	0.0000		+0.0002	0.0002T

Deviation from nominal (maximum) bearing bore and resultant fit.
T= Tight
L = Loose

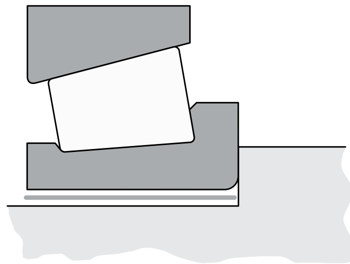


TABLE 55.
SHAFT O.D. METRIC BEARINGS (ISO AND J PREFIX) - CLASS A AND AA

Bearing Bore Range Over	Bearing Bore Range Incl.	Class A And AA Bearing Bore Tolerance	Symbol	Shaft O.D. Deviation	Resultant Fit
mm in.	mm in.	mm in.		mm in.	mm in.
10.000	**18.000**	**-0.005**	k4	**+0.006**	**0.011T**
		0.000		**+0.001**	**0.001T**
0.3937	0.7087	-0.0002		+0.0002	0.0004T
		0.0000		+0.00004	0.00004T
18.000	**30.000**	**-0.006**	k4	**+0.008**	**0.014T**
		0.000		**+0.002**	**0.002T**
0.7087	1.1811	-0.0002		+0.0003	0.0006T
		0.0000		+0.0001	0.0001T
30.000	**315.000**	**-0.008**		**+0.013**	**0.021T**
		0.000		**+0.005**	**0.005T**
1.1811	12.4016	-0.0003		+0.0005	0.0008T
		0.0000		+0.0002	0.0002T

Deviation from nominal (maximum) bearing bore and resultant fit.
T= Tight
L = Loose

TABLE 56.
SHAFT O.D. INCH BEARINGS - CLASS 3 AND 0 • CLASS 00 AND 000

Bearing Bore Range Over	Bearing Bore Range Incl.	Class 3 And 0[1] Bearing Bore Tolerance	Class 3 And 0[1] Shaft O.D. Deviation	Class 3 And 0[1] Resultant Fit	Class 00 And 000 Bearing Bore Tolerance μm (0.0001 In.)	Class 00 And 000 Shaft O.D. Deviation	Class 00 And 000 Resultant Fit
mm in.	mm in.	mm in.	mm in.		mm in.	mm in.	
–	**304.800**	**0.000**	**+0.030**	**0.030T**	**0.000**	**+0.020**	**0.020T**
		+0.013	**+0.018**	**0.005T**	**+0.008**	**+0.013**	**0.005T**
–	12.0000	0.0000	+0.0012	0.0012T	0.0000	+0.0008	0.0008T
		+0.0005	+0.0007	0.0002T	+0.0003	+0.0005	0.0002T
304.800	**609.600**	**0.000**	**+0.064**	**0.064T**	–	–	–
		+0.025	**+0.038**	**0.013T**			
12.0000	24.0000	0.0000	+0.0025	0.0025T	–	–	–
		+0.0010	+0.0015	0.0005T			
609.600	**914.400**	**0.000**	**+0.102**	**0.102T**	–	–	–
		+0.038	**+0.064**	**0.026T**			
24.0000	36.0000	0.0000	+0.0040	0.0040T	–	–	–
		+0.0015	+0.0025	0.0010T			

[1]Class 0 made only to 304.800 mm (12.0000 in.) O.D.
Deviation from nominal (maximum) bearing bore and resultant fit.
T= Tight
L = Loose

PRECISION CLASS TAPERED ROLLER BEARINGS

TXR

SUGGESTED FITTING GUIDELINES FOR FERROUS SHAFT AND HOUSING

For heavy loads, high speed or shock, contact your Timken representative for further information.

TABLE 57.
SHAFT O.D. METRIC BEARINGS - CLASS S • CLASS P

Bearing Bore Range		Class S		Class P	
Over	Incl.	Max.	Min.	Max.	Min.
mm in.	mm in.	mm in.	mm in.	mm in.	mm in.
– –	**50.000** 1.9685	**0.020T** 0.0008T	**0.007T** 0.0003T	**0.014T** 0.0006T	**0.004T** 0.0002T
50.000 1.9685	**80.000** 3.1496	**0.025T** 0.0010T	**0.010T** 0.0004T	**0.017T** 0.0007T	**0.004T** 0.0002T
80.000 3.1496	**120.000** 4.7244	**0.033T** 0.0013T	**0.013T** 0.0005T	**0.017T** 0.0007T	**0.004T** 0.0002T
120.000 4.7244	**180.000** 7.0866	**0.052T** 0.0021T	**0.027T** 0.0011T	**0.017T** 0.0007T	**0.004T** 0.0002T
180.000 7.0866	**250.000** 9.8425	**0.060T** 0.0024T	**0.030T** 0.0012T	**0.020T** 0.0008T	**0.004T** 0.0002T
250.000 9.8425	**315.000** 12.4016	**0.070T** 0.0028T	**0.035T** 0.0014T	**0.022T** 0.0009T	**0.004T** 0.0002T
315.000 12.4016	**400.000** 15.7480	**0.077T** 0.0030T	**0.037T** 0.0015T	**0.024T** 0.0009T	**0.004T** 0.0002T
400.000 15.7480	**500.000** 19.6850	**0.085T** 0.0034T	**0.040T** 0.0016T	**0.030T** 0.0012T	**0.004T** 0.0002T

Deviation from nominal (maximum) bearing bore and resultant fit.
T= Tight
L = Loose

TABLE 58.
HOUSING BORE METRIC BEARINGS - CLASS S • CLASS P

Bearing O.D. Range		Class S		Class P	
Over	Incl.	Max.	Min.	Max.	Min.
mm in.	mm in.	mm in.	mm in.	mm in.	mm in.
– –	**50.000** 1.9685	**0.020T** 0.0008T	**0.007T** 0.0003T	**0.014T** 0.0006T	**0.004T** 0.0002T
50.000 1.9685	**80.000** 3.1496	**0.025T** 0.0010T	**0.010T** 0.0004T	**0.017T** 0.0007T	**0.004T** 0.0002T
80.000 3.1496	**120.000** 4.7244	**0.033T** 0.0013T	**0.013T** 0.0005T	**0.017T** 0.0007T	**0.004T** 0.0002T
120.000 4.7244	**180.000** 7.0866	**0.052T** 0.0021T	**0.027T** 0.0011T	**0.017T** 0.0007T	**0.004T** 0.0002T
180.000 7.0866	**250.000** 9.8425	**0.060T** 0.0024T	**0.030T** 0.0012T	**0.020T** 0.0008T	**0.004T** 0.0002T
250.000 9.8425	**315.000** 12.4016	**0.070T** 0.0028T	**0.035T** 0.0014T	**0.022T** 0.0009T	**0.004T** 0.0002T
315.000 12.4016	**400.000** 15.7480	**0.077T** 0.0030T	**0.037T** 0.0015T	**0.024T** 0.0009T	**0.004T** 0.0002T
400.000 15.7480	**500.000** 19.6850	**0.085T** 0.0034T	**0.040T** 0.0016T	**0.030T** 0.0012T	**0.004T** 0.0002T

Deviation from nominal (maximum) bearing bore and resultant fit.
T= Tight
L = Loose

TABLE 59.
SHAFT O.D. INCH BEARINGS - CLASS 3 • CLASS 0

Bearing Bore Range		Class 3		Class 0	
Over	Incl.	Max.	Min.	Max.	Min.
mm in.	mm in.	mm in.	mm in.	mm in.	mm in.
– –	304.800 12.0000	0.037T 0.0015T	0.013T 0.0005T	0.020T 0.0008T	0.007T 0.0003T
304.800 12.0000	609.600 24.0000	0.077T 0.0030T	0.025T 0.0010T	0.037T 0.0015T	0.013T 0.0005T
609.600 24.0000	914.400 36.0000	0.114T 0.0045T	0.037T 0.0015T	– –	– –
914.400 36.0000	1219.200 48.0000	0.152T 0.0060T	0.051T 0.0020T	– –	– –
1219.200 48.0000	– –	0.191T 0.0075T	0.064T 0.0025T	– –	– –

Deviation from nominal (maximum) bearing bore and resultant fit.
T= Tight
L = Loose

TABLE 60.
HOUSING BORE INCH BEARINGS - CLASS 3 • CLASS 0

Bearing O.D. Range		Class 3		Class 0	
Over	Incl.	Max.	Min.	Max.	Min.
mm in.	mm in.	mm in.	mm in.	mm in.	mm in.
– –	304.800 12.0000	0.037T 0.0015T	0.013T 0.0005T	0.020T 0.0008T	0.007T 0.0003T
304.800 12.0000	609.600 24.0000	0.077T 0.0030T	0.025T 0.0010T	0.037T 0.0015T	0.013T 0.0005T
609.600 24.0000	914.400 36.0000	0.114T 0.0045T	0.037T 0.0015T	– –	– –
914.400 36.0000	1219.200 48.0000	0.152T 0.0060T	0.051T 0.0020T	– –	– –
1219.200 48.0000	– –	0.191T 0.0075T	0.064T 0.0025T	– –	– –

Deviation from nominal (maximum) bearing bore and resultant fit.
T= Tight
L = Loose

SHAFT AND HOUSING CONSIDERATIONS
TAPERED ROLLER BEARINGS

In general, machining bearing seats and shoulders in spindles and housings requires careful consideration of the following form and orientation characteristics. The first four characteristics apply to the seats of bearing rings.

- Circularity (roundness) of each seat at every cross section.
- Cylindricity of each seat. Cylindricity includes the taper, roundness and other form characteristics of the seat.
- Coaxiality of the inner ring seats on the spindle and coaxiality of the outer ring seats in the housing. Coaxiality includes offset misalignment and angular misalignment between seats.
- Angularity of each bearing ring seat. This is a consideration when an inner ring seat is tapered.

The following two characteristics apply to the shoulders corresponding to each bearing seat.

- **Perpendicularity** (squareness) of each shoulder to its corresponding bearing seat, or as a more practical measure, perpendicularity of each shoulder to the spindle or housing centerline established from the two bearing seats.
- **Flatness** of each shoulder. A practical way of assessing the combined perpendicularity and flatness of each shoulder is to measure the total runout of the shoulder relative to the spindle or housing centerline. The runout of the face of the adjusting nuts, if used, should also be measured.

The tolerances to which these characteristics should be held are dependent upon the class, size and application of the bearing. *In general, these tolerances should be no greater than the total indicator reading (T.I.R.) of the assembled bearing.*

Some of the characteristics can be difficult to measure precisely. The individual user may elect to measure a subset of these characteristics (roundness and taper as an alternative to cylindricity). The individual user must determine the degree of effort and expense to be invested in the measurements. That determination should be based on the intended application of the bearing and the level of confidence in the machining process employed to manufacture the spindle and housing.

SURFACE FINISHES – PRECISION BEARINGS

Precision class bearings should be mounted on shafts and in housings that are finished to at least the same precision limits as the bearing bore or outside diameter.

Furthermore, high-quality surface finishes together with close machining tolerances of bearing seats also must be provided. The following tabulations give some guidelines for all these criteria.

TABLE 61.
SUGGESTED TAPERED ROLLER BEARING SHAFT AND HOUSING FINISHES

All Sizes	Bearing Class			
	C 3	B 0	A 00	AA 000
	µm µin	**µm** µin	**µm** µin	**µm** µin
Shaft - Ra	**0.8** 32	**0.6** 24	**0.4** 16	**0.2** 8
Housing - Ra	**1.6** 63	**0.8** 32	**0.6** 24	**0.4** 16

BALL BEARINGS

SHAFT FITS

The main purpose of the shaft fit is to assure a proper attachment of the inner ring to the shaft. Under normal conditions of shaft rotation, a loosely fitted inner ring will creep on the shaft, leading to wear and fretting. This condition will be further aggravated by increase of load or speed. To prevent creeping or slipping, the inner ring should be mounted firmly in place and held securely against the shaft shoulder. However, it is important that the shaft fit should not result in any undue tightening of the bearing. An excessive interference fit of the bearing bore with the shaft could result in a proportionate expansion of the bearing inner ring which could disturb the internal fit of the bearing and lead to heating and increased power consumption.

As a general rule, it is suggested that the shaft size and tolerance for seating super precision ball bearings (ABEC 7/ISO P4 and ABEC 9/ ISO P2) be the same as the bearing bore. In the case of preloaded bearings, the suggested shaft fit is line-to-line, since an excessively tight fit expands the bearing inner ring and increases the bearing preload, which can lead to overheating. For example, a duplex pair of 2MM9111WI DUL bearings with 16 kg (35 lbs.) built-in preload, when mounted on a shaft that provides an interference fit of 0.010 mm (0.0004 in.), will increase the preload to approximately 86 kg (180 lbs.), which could result in elevated operating temperatures.

TABLE 62.
SHAFT FIT EXAMPLE: MMV (ABEC 7/ISO P4)

Bore Size		Shaft Diameter		Resulting Mounting Fit	
Max	Min.	Max.	Min.	Loose	Tight
mm in.	mm in.	mm in.	mm in.	mm in.	mm in.
55.000	**54.994**	**55.000**	**54.994**	**0.004**	**0.006**
2.1654	2.1651	2.1654	2.1651	0.0002	0.0003

HOUSING FITS

Under normal conditions of rotating shaft, the outer ring is stationary and should be mounted with a hand push to a light tapping fit. Should the housing be the rotating member, the same fundamental considerations apply in mounting the outer ring as in the case of an inner ring mounted on a rotating shaft. Contact your Timken representative for outer ring rotation requirements.

As a general rule, the minimum housing bore dimension for super precision ball bearings may be established as the same as the maximum bearing outside diameter. If the bearing O.D. tolerance is 0.008 mm (0.0003 in.), the maximum housing bore should be established as 0.008 mm (0.0003 in.) larger than the minimum housing bore dimensions.

TABLE 63.
HOUSING BORE FIT EXAMPLE: MMV (ABEC 7/ISO P4)

Outside Diameter		Housing Bore		Resulting Mounting Fit		Average Fit	
Max	Min.	Max.	Min.	Loose	Tight	Loose	Loose
mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.
90.000	**89.992**	**90.007**	**90.000**	**0.015**	**0.000**	**0.006**	**0.006**
3.5433	3.5430	3.5436	3.5433	0.0006	0.0000	0.0003	0.0003

Tables covering suggested shaft and housing seat dimensions for super precision (ABEC 7/ISO P4) ball bearings are shown with part numbers in the product pages.

To accomplish the optimum mounting condition, it is important to follow the tabulated tolerances, except when deviations are suggested by your Timken representative. It is equally important that all shaft and housing shoulders be square and properly relieved to assure accurate seating and positioning of the bearings in the mounting.

On high-speed applications where nearby heat input is along the shaft, it is extremely important that the floating bearings can move axially to compensate for thermal changes. Ball bearings cannot float axially if they are restricted by tight housing bores or by the radial expansion of the bearing itself due to temperature differentials. Therefore, in such cases, the suggested housing mounting fit for the floating bearings is slightly looser than the tabulated average fit.

Likewise, in spring-loaded ball bearing applications, the housing mounting fit must be free enough to permit axial movement of the bearings under the spring pressure during all conditions of operation. The suggested housing dimensions to ensure proper "float" of the bearings under average conditions are listed in the product pages.

SHAFT AND HOUSING TOLERANCES

SHAFT GEOMETRY REQUIREMENTS

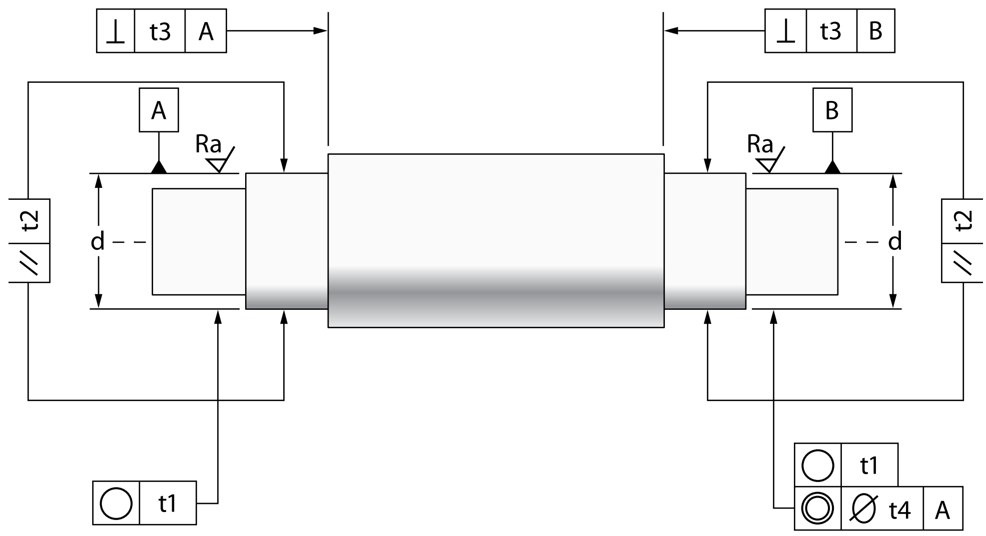


Fig. 69. Shaft Tolerances.

TABLE 64.
DEFINITION OF TOLERANCE SYMBOLS

Description	Symbol	Tolerance Value	MV ABEC 7 (ISO P4)	MMV/MM ABEC 7/9 (ISO P4S)	MMX ABEC 9 (ISO P2)
Roundness	○	t1	IT2	IT1	IT0
Parallelism	//	t2	IT2	IT1	IT0
Squareness	⊥	t3	IT2	IT1	IT0
Concentricity	◎	t4	IT3	IT2	IT2
Surface Finish	Ra		16 (µin.)	or	0.4 µm

TABLE 65.
SHAFT SURFACE FINISH SPECIFICATIONS

Shaft Journal Diameter (d) mm		Units – Micrometer (µm)			
>	≤	IT0	IT1	IT2	IT3
—	**10**	0.6	1.0	1.5	2.5
10	**18**	0.8	1.2	2.0	3.0
18	**30**	1.0	1.5	2.5	4.0
30	**50**	1.0	1.5	2.5	4.0
50	**80**	1.2	2.0	3.0	5.0
80	**120**	1.5	2.5	4.0	6.0
120	**180**	2.0	3.5	5.0	8.0
180	**250**	3.0	4.5	7.0	10.0
250	**315**	—	6.0	8.0	12.0

Shaft Journal Diameter (d) mm		Units – Microinches (µin.)			
>	≤	IT0	IT1	IT2	IT3
—	**10**	20	40	60	100
10	**18**	30	50	80	120
18	**30**	40	60	100	160
30	**50**	40	60	100	160
50	**80**	50	80	120	200
80	**120**	60	100	160	240
120	**180**	80	140	200	310
180	**250**	120	180	280	390
250	**315**	—	240	310	470

Reference ISO 286.

HOUSING GEOMETRY REQUIREMENTS

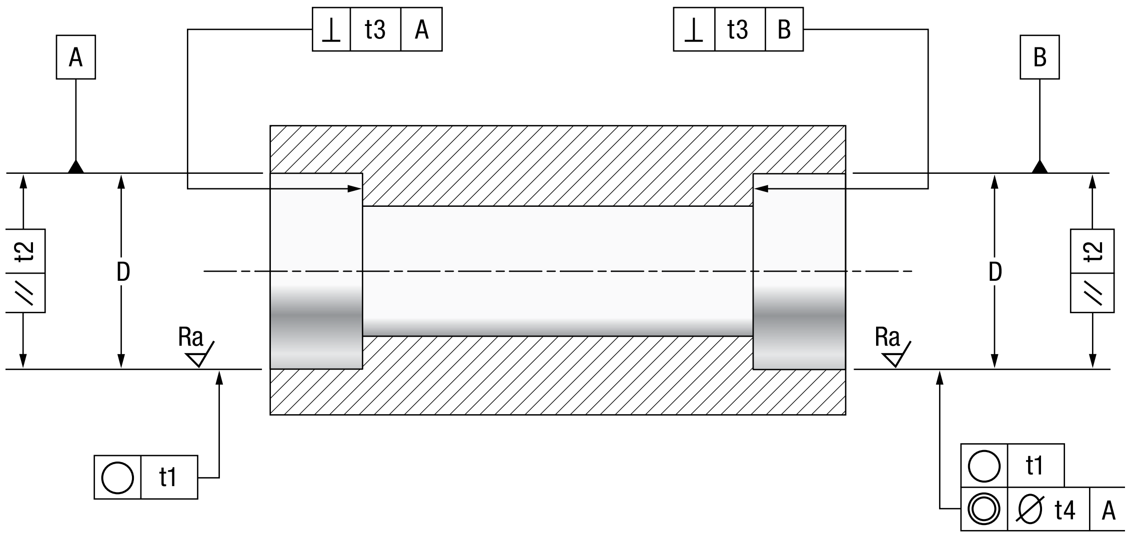


Fig. 70. Housing Tolerances.

TABLE 66.
DEFINITION OF TOLERANCE SYMBOLS

Description	Symbol	Tolerance Value	MV ABEC 7 (ISO P4)	MMV/MM ABEC 7/9 (ISO P4S)	MMX ABEC 9 (ISOP2)
Roundness	○	t1	IT2	IT1	IT0
Parallelism	//	t2	IT2	IT1	IT0
Squareness	⊥	t3	IT2	IT1	IT0
Concentricity	◎	t4	IT3	IT2	IT2
Surface Finish	Ra		16 (µin.)	or	0.4 µm

TABLE 67.
HOUSING SURFACE FINISH SPECIFICATIONS

Housing Journal Diameter (D) mm		Units – Micrometer (µm)			
>	≤	IT0	IT1	IT2	IT3
10	**18**	0.8	1.2	2.0	3.0
18	**30**	1.0	1.5	2.5	4.0
30	**50**	1.0	1.5	2.5	4.0
50	**80**	1.2	2.0	3.0	5.0
80	**120**	1.5	2.5	4.0	6.0
120	**180**	2.0	3.5	5.0	8.0
180	**250**	3.0	4.5	7.0	10.0
250	**315**	3.5	6.0	8.0	12.0
315	**400**	4.5	6.0	8.0	12.0

Housing Journal Diameter (D) mm		Units – Microinches (µin.)			
>	≤	IT0	IT1	IT2	IT3
10	**18**	30	50	80	120
18	**30**	40	60	100	160
30	**50**	40	60	100	160
50	**80**	50	80	120	200
80	**120**	60	100	160	240
120	**180**	80	140	200	310
180	**250**	120	180	280	390
250	**315**	140	240	310	470
315	**400**	180	240	310	470

Reference ISO 286.

MOUNTING DESIGNS

Obtaining good spindle accuracy depends not only on selecting the proper precision bearings but also on the following factors:

- Good design and machining of the components that support the bearing (roundness and alignment of the seats, squareness of backing shoulders of both the spindle and the housing, and surface finish).
- Correct use of information given on bearings.
- Correct fitting practices.
- Appropriate bearing setting.

Selection of the most appropriate mounting design is largely dictated by optimizing the stiffness, speedability and ease of assembly.

DESIGN AND ACCURACY OF MOUNTING SURFACES

The total runout of a spindle-bearing-housing system is a combination of the runout of each component. A precision bearing will assume the shape of the spindle and perpetuate whatever runout is present. If the runout is caused by a defective housing, the spindle and bearing will simply transmit the error to the workpiece. Therefore, particular attention needs to be paid to the design and accuracy of the mounting surfaces.

The primary function of the inner or outer ring seat and abutment is to positively establish the location and alignment of the bearing under all loading and operating conditions. To achieve optimum bearing performance, it is essential to design housing seats and abutments that are round and square in alignment with the spindle axis. Shoulders must be of sufficient section and design to resist axial deflection under load. The shoulder diameters should be respected to help obtain optimum bearing performance.

HOUSING DESIGN

Housings are usually made of cast iron or steel and are generally heat-treated to lessen possible distortion. For smaller high-speed applications, steel housings are preferred.

The bore of the housing should be ground or bored and checked at a number of points throughout its length and diameter to ensure that it is round and does not taper.

It is preferable to mount the bearings in one casting; this permits machining the two housing bores in one setting, as well as accurate alignment of the bearings.

In many cases of machine design, it is advantageous to employ a subhousing or a steel sleeve between the outer ring of the bearing and the machine frame, thus allowing assembly of the bearings on the shaft and insertion of the entire unit into the machine frame. This method also provides a surface of proper hardness where machine frames are made of a material that has a low Brinell value, such as aluminum and other soft metals.

Shaft shoulders and housing shoulders should be square and true, and should be of such diameters as to meet the suggestions shown with the part numbers given. The choice between fillets and undercut reliefs rests with the individual shaft design and conditions surrounding its normal use. Suggested housing geometry requirements are discussed on pages 83 and 85.

Where screws are used to fasten end caps into the main housing, adequate section should be left between the screw hole and the housing bore. This is required to prevent distortion of the housing bore when the screws are tightened and the covers or others parts are pulled tightly into place.

Prior to assembly, shafts and housings, as well as all lubricant holes and channels, should be cleaned thoroughly to remove all chips and particles that may be carried by the lubricant into the bearings and cause bearing damage.

HOUSING SEALS

A labyrinth combination of slinger and end cover provides a highly effective seal against the intrusion of foreign matter. This seal is suggested for use over a wide range of speeds. For slower-speed applications, a combination of slinger and a commercial contact-type seal is usually employed.

Slingers should be machined all over to assure true-running. Their diameters should be concentric with the bore. The outside diameter of the slinger is often tapered to throw off cutting compounds, coolants, etc., from the point at which such liquids may enter the spindle. A drip or run-off groove adjacent to the open lip of the end cover is highly desirable and practical.

The axial clearances of the internal faces between slinger and end cover should be about 1.600 mm (0.0629 in.). The first radial clearance opening on any design through which liquid may pass should be made very close, about 0.089 mm (0.0035 in.) on a side. The inner radial clearances should be between 0.380 mm (0.0149 in.) and 0.190 mm (0.0075 in.).

SHAFTS

Shafts are preferably made from hardened and ground steel; and, where suitable, a hardness of 45-50 HRC has been successful. When designing a spindle or shaft, it is highly desirable to plan so that it can be ground all over in one setting as a final operation. This promotes true balance and running accuracy, which are critical in high-speed applications. Suggested shaft geometry can be found on pages 84 and 85.

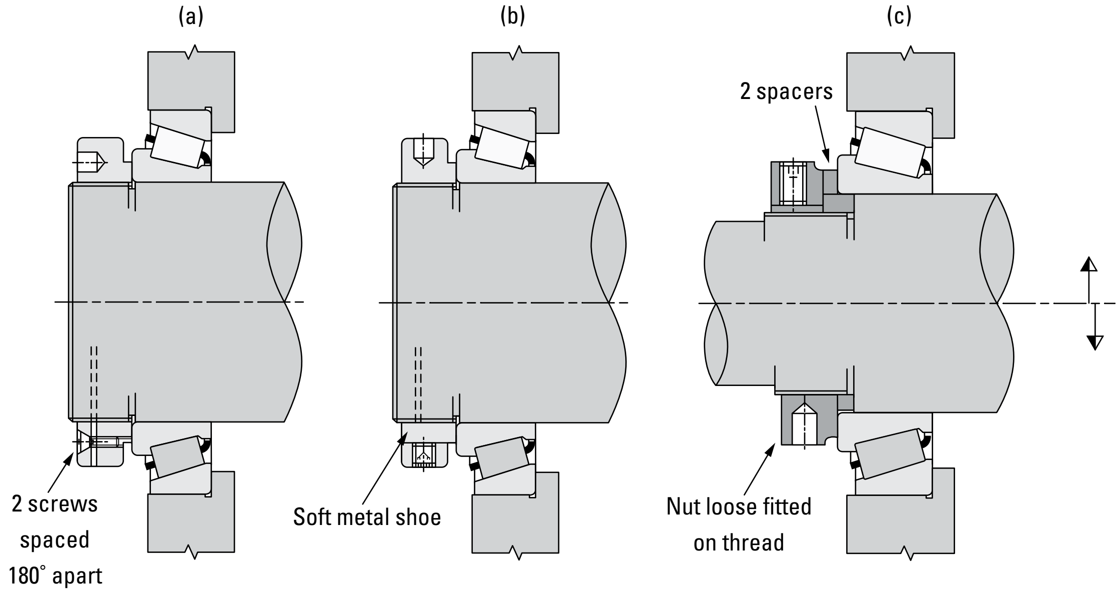


Fig. 71.
Locking devices.

LOCKING DEVICES

In most cases, simple 2TS(F) spindle layouts are adjusted by correct positioning of the tail bearing cone. A commonly used device is a precision adjusting nut. A locking device must be provided to properly maintain the nut after setting: either axially by means of two screws 180 degrees opposite pinching the threads (Fig. 71a), or radially by pressure of a screw on a soft metal shoe (Fig. 71b).

For improved accuracy, a ground spacer in conjunction with a square-ground spindle shoulder and a locking precision nut also can be used (Fig. 70). Good parallelism of the ground spacer faces as well as the squareness of the spindle shoulder will ensure a perfect positioning of the cone backface. This mounting configuration also offers assurance that the initially defined setting cannot be interfered with by the final user. Fig. 71c shows two different solutions with ground spacers. Note the practicality of the above centerline solution, which allows the spacer to both increase or decrease the initial setting.

A well-known method of providing good spindle alignment, roundness and backing squareness is to grind the cone seats and the backing shoulders during the same operation (Fig. 73). In this method, the grinding of the square backing of the adjusting nut (if any) also can be achieved by locking the nut on its thread. This eliminates any possible default of the nut due to internal thread clearance.

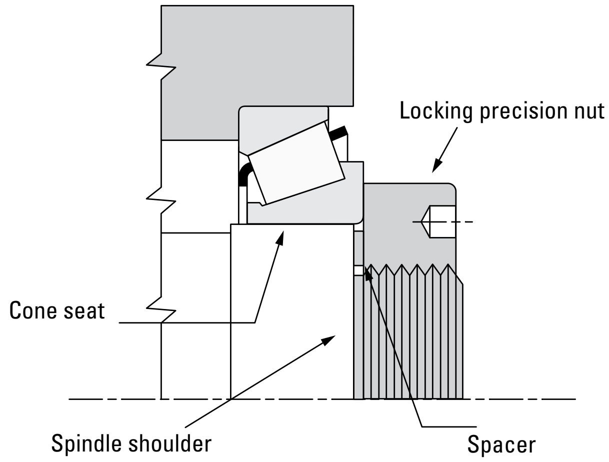


Fig. 72. Using ground spacer and spindle shoulder together with a precision nut for improved accuracy.

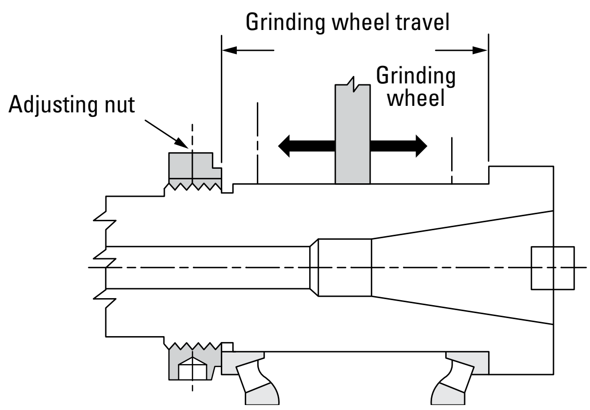


Fig. 73. Grinding of cone shaft and backing shoulders.

TAPERED ROLLER BEARINGS

Tapered roller bearings are generally used in two fundamental spindle design configurations:

- Three-support mountings for heavily loaded or long spindles.
- Simple mounting of two single-row bearings.

THREE-SUPPORT MOUNTING

Fig. 74 shows the "box type" mounting using three bearings. The two nose bearings are located axially (fixed position) and accept axial forces in both directions, while the tail bearing is fitted as a floating position to accommodate the thermal expansion of the spindle. The floating position can be supported either by a tapered roller bearing or a cylindrical roller bearing.

This kind of arrangement is mainly used for special heavy machines running at low or medium speeds, or for long spindle designs.

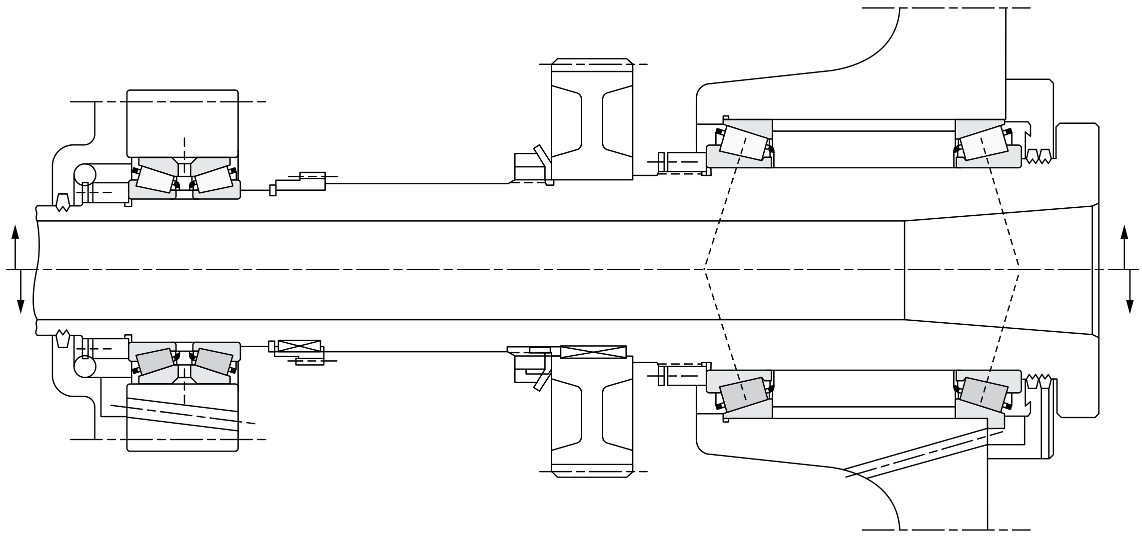


Fig. 74. "Box-type" mounting with a TDO at the floating position.

SIMPLE MOUNTING

The evolution of two single-row bearing arrangements for spindles, discussed below, is directly related to the speed requirements and, consequently, the lubrication modes (see page 41).

TS and TSF Arrangement

The spindle is supported by one bearing at the nose position and a second one at the tail position. This layout offers the advantage of being a simple isostatic design that allows easy machining of adjacent parts. The mounting and setting procedures can be achieved without any specific tooling.

Static stiffness calculations of the spindle-bearing system allow the optimum bearing spread to be determined precisely for each mounting, as a function of the overhung value of the spindle nose. A good approximation, however, is to consider that the distance between bearing centers should be of two and a half to three times the spindle nose diameter. This represents an optimum value not only for stiffness, but also for thermal equilibrium.

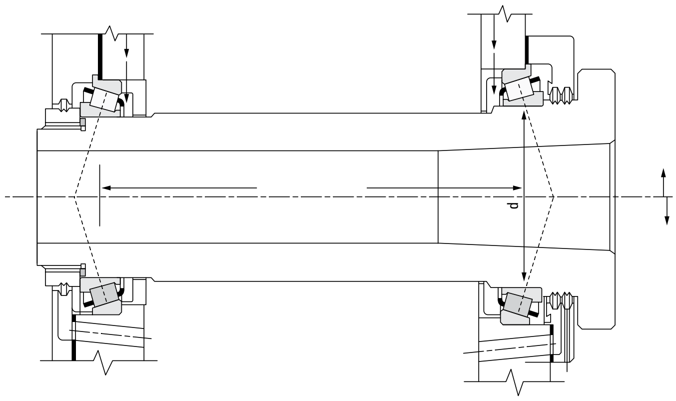


Fig. 75. Simple mounting with a pair of TS or TSF bearings.

Fig. 75 represents the simplest layout of a two single-row bearing concept. The view above the centerline shows flanged cups (Type TSF) allowing a through-bore machining concept for the housing, which offers increased accuracy with no need for cup backing shoulders. The arrangement shown below the centerline uses two single-row bearings (Type TS).

The bearings are adjusted by means of a ground spacer locked by a precision nut. Lubrication is often achieved by oil circulation, which enters through radial oil inlets or special high-speed grease.

As shown below, the next evolution of this arrangement consists of improving the lubrication system by using appropriate jets for oil inlets and cooling (Fig. 76 and Fig. 77).

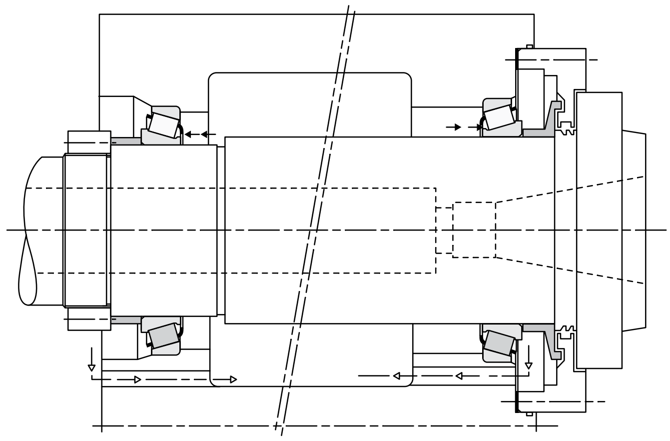


Fig. 76. Simple paired TS mounting with oil inlet at the small end of the rollers.

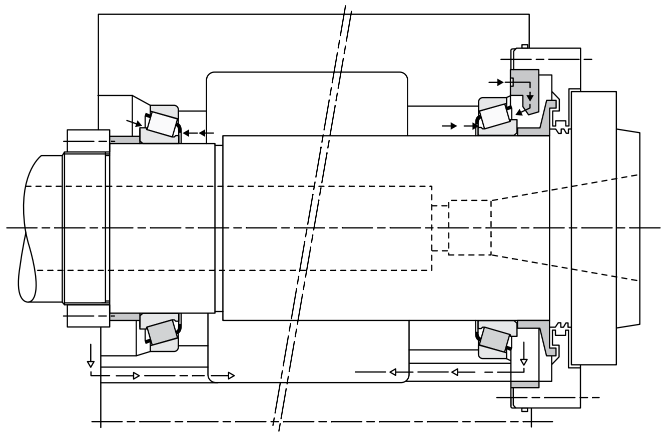


Fig. 77. Simple paired TS mounting with oil jets at both ends of the rollers for inlet and cooling.

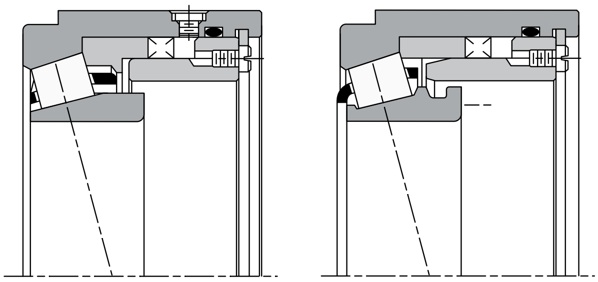


Fig. 78. Two designs of the Hydra-Rib bearing.

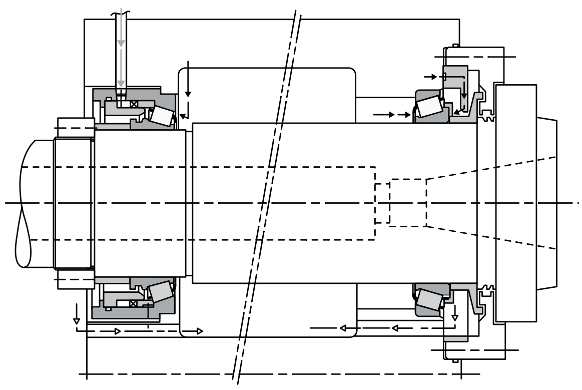


Fig. 80. Simple mounting with a Hydra-Rib bearing cooled by an axial oil inlet and a TSMA bearing with oil jets at both ends of the rollers for inlet and cooling.

TS(F) and Hydra-Rib

A typical spindle arrangement is the combination of a Hydra-Rib bearing with a single-row TS bearing (Fig. 79). The Hydra-Rib bearing is fitted at the tail position and the TS bearing at the nose position of the spindle. The outer ring rib simplifies the lubrication at high speed since the natural flow of the oil under centrifugal effect feeds the oil to the rib. A simple axial oil inlet above the cage on the small roller end is therefore sufficient for lubricating the Hydra-Rib bearing.

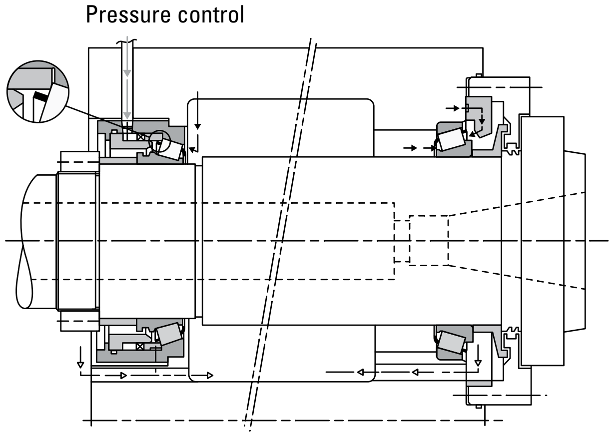
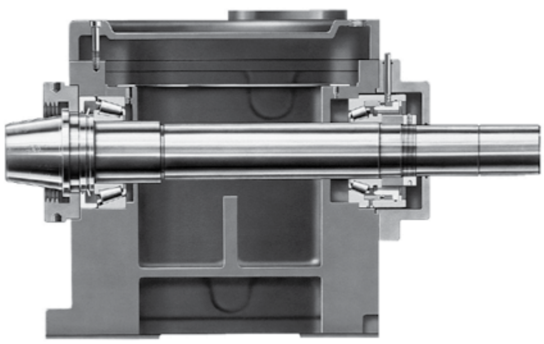


Fig. 79. Simple mounting with a Hydra-Rib cooled by an axial oil inlet and a TS bearing with oil jets at both end of the rollers for inlet and cooling.

TSMA and Hydra-Rib

Fig. 80 shows the same arrangement with a TSMA bearing. This arrangement allows the widest range of operating speeds, under optimum preload.



TXR(DO)

A typical mounting arrangement for the type TXRDO crossed roller bearing is shown in Fig. 81.

The arrangement shown is for lubrication by oil circulation in conjunction with an oil level. It can, however, be designed for grease lubrication with appropriate sealing arrangements.

The bore of the housing (DH) and the diameter of the spigot (DS) (Fig. 82) should be machined to give a mean of the suggested interference fits (pages 80-81).

The bearing is adjusted externally by segments beneath the top inner ring clamping plate (Fig. 82) to get the required preload.

Your Timken representative should be consulted for more details about the use of crossed roller bearings.

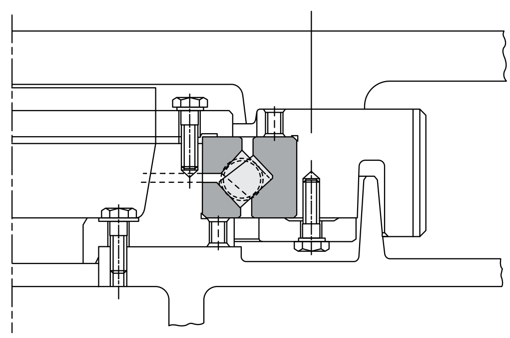


Fig. 81. Typical mounting arrangement of a TXRDO bearing.

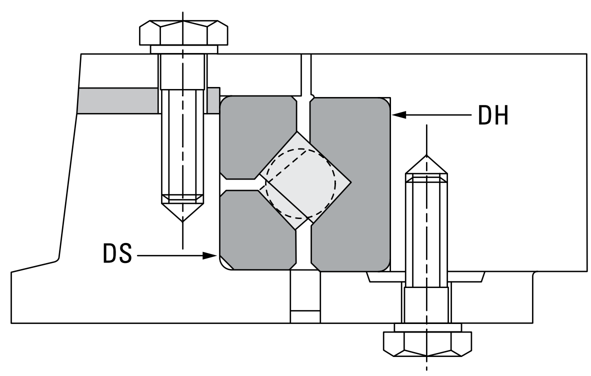


Fig. 82. Fitting and setting of TXR bearings.

DUPLEX BALL BEARINGS

BACK-TO-BACK MOUNTING, DB OR ("O") (CONTACT ANGLES DIVERGING TOWARD SHAFT CENTERLINE)

Before mounting, there is clearance between the two adjacent inner ring faces. After mounting, these faces are clamped together to provide an internal preload on each bearing. This arrangement is well-suited for pulleys, sheaves and other applications where there are overturning loads and also all floating positions where thermal expansion of the shaft occurs. It also provides axial and radial rigidity and equal axial capacity in either direction when used in a fixed location. Back-to-back is the most commonly used of all duplex arrangements. Timken pairs for back-to-back mounting should be ordered as DU. Example: 2MM207WI-DU. Also available as two single flush-ground bearings, e.g., -SU (two bearings).

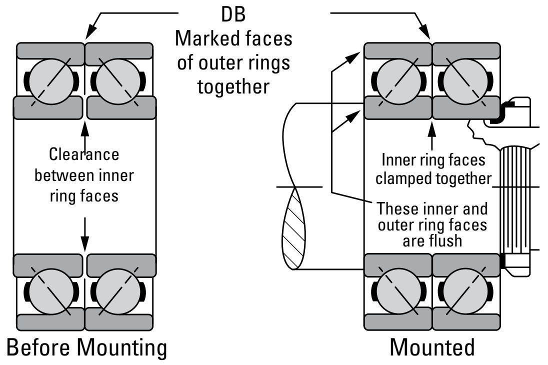


Fig. 83. Back-to-back bearing assemblies before and after mounting.

FACE-TO-FACE MOUNTING, DF OR ("X") (CONTACT ANGLES CONVERGING TOWARD SHAFT CENTERLINE)

Before mounting, there is clearance between the two adjacent outer ring faces. After mounting, these faces are clamped together between the housing shoulder and cover plate shoulder, providing an internal preload on each bearing. This arrangement provides equal axial capacity in either direction as well as radial and axial rigidity. Since the face-to-face mounting has inherent disadvantages of low resistance to moment loading and thermal instability, it should not be considered unless a significantly more convenient method of assembly or disassembly occurs from its use. Timken pairs for face-to-face mounting should be ordered as DU. Example: 2MM212WI-DU. Also available as two single flush-ground bearings, e.g., -SU (two bearings).

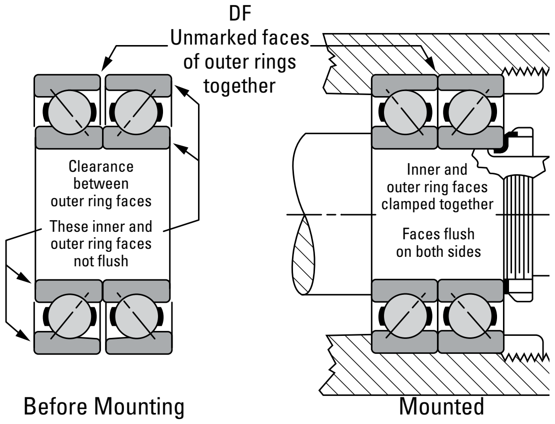


Fig. 84. Face-to-face bearing assemblies before and after mounting.

TANDEM MOUNTING, DT

Before mounting, the inner ring faces of each bearing are offset from the outer ring faces. After mounting, when an axial load is applied equal to that of twice the normal preload, the inner and outer ring faces are brought into alignment on both sides. This arrangement provides double axial capacity in one direction only. More than two bearings can be used in tandem if additional axial capacity is required. Timken pairs for tandem mounting should be specified as DU. Example: 2MM205WI-DU. Also available as two single flush-ground bearings, e.g., -SU (two bearings).

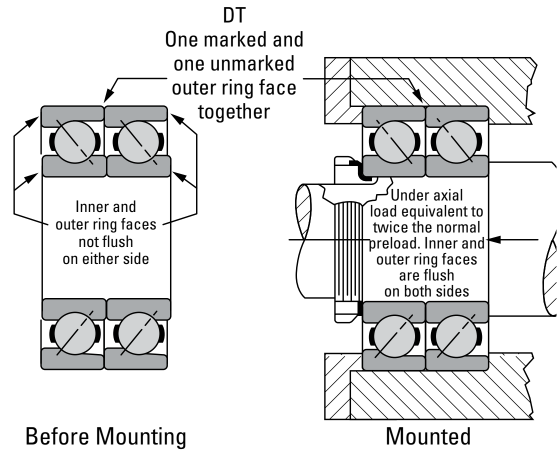


Fig. 85. Tandem bearing assemblies before and after mounting.

OTHER MOUNTINGS

Flush-ground (DU) pairs may be mounted in combination with a single flush-ground bearing as a "triplex" (TU) set shown below. Also shown below is a "quadruplex" (QU) set where three bearings in tandem are mounted back-to-back with a single bearing. These arrangements provide high capacity in one direction and also a positively rigid mounting capable of carrying a moderate amount of reverse thrust.

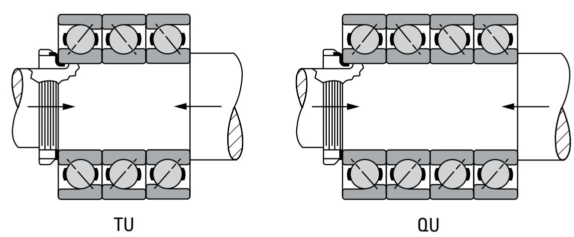


Fig. 86. Typical triplex and quadruplex bearing mountings.

BACK-TO-BACK VERSUS FACE-TO-FACE MOUNTINGS

Mountings having bearings applied in any of the face-to-face (DF) arrangements are objectionable because they provide the least rigidity. Furthermore, when the operating speeds are comparatively high, such mountings may buildup bearing preload excessively because of the temperature gradient between the housings, bearings and shafts. As this gradient increases, the bearing preload builds up, starting a detrimental cycle that may lead to premature spindle damage.

In spindle mountings, the shaft temperature usually changes at a faster rate than the housing, creating temperature differentials between the two members. These are due to their difference in mass and their respective abilities to act as heat sinks. Thus, the shaft and the inner-ring spacer expand at a faster rate than the housing and the outer-ring spacer. As the shaft expands longitudinally and the inner-ring spacer lengthens, an axial load builds up on each bearing and continues to increase until the equilibrium temperature is reached. This occurs when the temperature at the housing levels off and the heat transferred from the bearings balances the heat generated within the system. Therefore, if the housing attains an excessively high temperature, the initial bearing temperature is built up considerably.

In a face-to-face mounting, Fig. 87, the shaft expands radially and longitudinally and the inner-ring spacer lengthens, but at a faster rate than the outer-ring spacer. This thermal expansion causes an additional axial load to be imposed on both inner rings, increasing the preload of the bearings. Conversely, in back-to-back mounting, Fig. 88, the longitudinal expansion of the inner-ring spacer tends to relieve, rather than build up, the bearing preload.

The two back-to-back pairs, shown in Fig. 89, are mounted so that the two middle bearings are face-to-face. As previously observed, temperature differentials cause the preload of these inner bearings to increase during operation. This mounting operation is not suggested. In bearing mountings of the system seen in Fig. 90, undue axial loads are put on the two outer bearings as the temperature along the shaft becomes higher than at the housing. The two inner bearings unload, starting a vicious cycle of increasing temperature, preload buildup and lubricant breakdown. This also is an unacceptable mounting arrangement and is not suggested. The same bearings are shown correctly mounted in tandem and arranged back-to-back in Fig. 91. Lateral expansion of the shaft and inner-ring spacer of such mountings increases neither axial loading nor bearing preload.

Therefore, to prevent increases in preload due to the thermal expansion, back-to-back mountings are preferred for bearings on machine tool spindles. When two pairs are used, each pair should be mounted in tandem, but the combination should be arranged back-to-back as in Fig. 91.

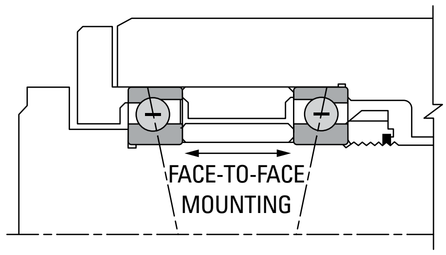


Fig. 87. DF Mounting, fixed (not suggested).

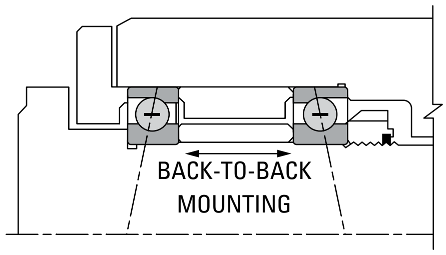


Fig. 88. DB Mounting, fixed (suggested).

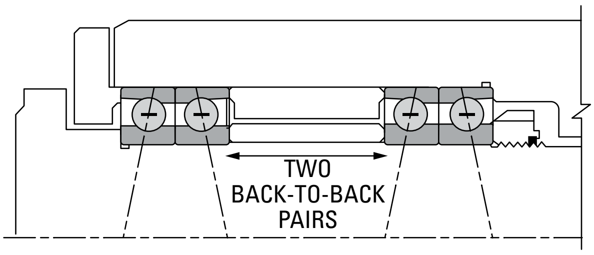


Fig. 89. DB-DB Mounting, fixed (not suggested).

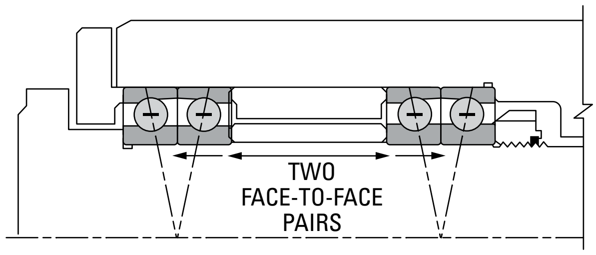


Fig. 90. DF-DF Mounting, fixed (not suggested).

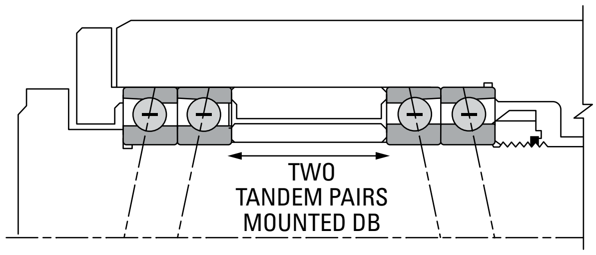


Fig. 91. DT-DB Mounting, fixed (suggested).

SPRING-LOADED MOUNTINGS

For high-speed applications, radial and axial rigidity and smooth spindle performance may be obtained by spring-loading the ball bearings with a predetermined axial load. Spring-loading allows the spindle to float laterally during temperature changes without appreciably increasing or decreasing the original spring axial load.

As the inner ring heats up during operation, it expands radially. This radial expansion applies an increasing load through the ball and outer ring and finally to the preload springs. The preload springs deflect slightly to compensate for the loads due to thermal expansion and maintain a consistent load on the spindle system.

In some applications, single, spring-loaded bearings are employed at the front and rear locations, mounted in back-to-back arrangement. Other mountings, similarly spring-loaded, have a pair of bearings installed in tandem at each end of the spindle in back-to-back arrangement (DT-DB). In either case, the spring pressure is applied to the pulley-end or rear bearing position, placing the shaft in tension between the two bearing locations.

HIGH POINTS OF RUNOUT

The correct use of the high point of runout etched on the bearing components allows the accuracy of the spindle to be optimized. The components should be mounted in the housing and on the spindle so that the high points are aligned with each other. In other words, the inner ring is fitted on the spindle so the high point of the rear ring is aligned with the high point of the nose bearing. Similarly, the high points of the outer ring are aligned in the housing.

To obtain maximum precision, and when the high points of runout of both the spindle and the housing are known, the respective high points of the bearing components should be 180 degrees opposite to those of the spindle and the housing. This will tend to neutralize the eccentricity and minimize the effect of the high spots of all components. The figures to the right show typical examples of the correct and incorrect use of the high point of runout of bearings.

The greatest accuracy can be provided by grinding the spindle nose after the bearings are installed. This procedure will produce spindle runout considerably smaller than the bearing runout.

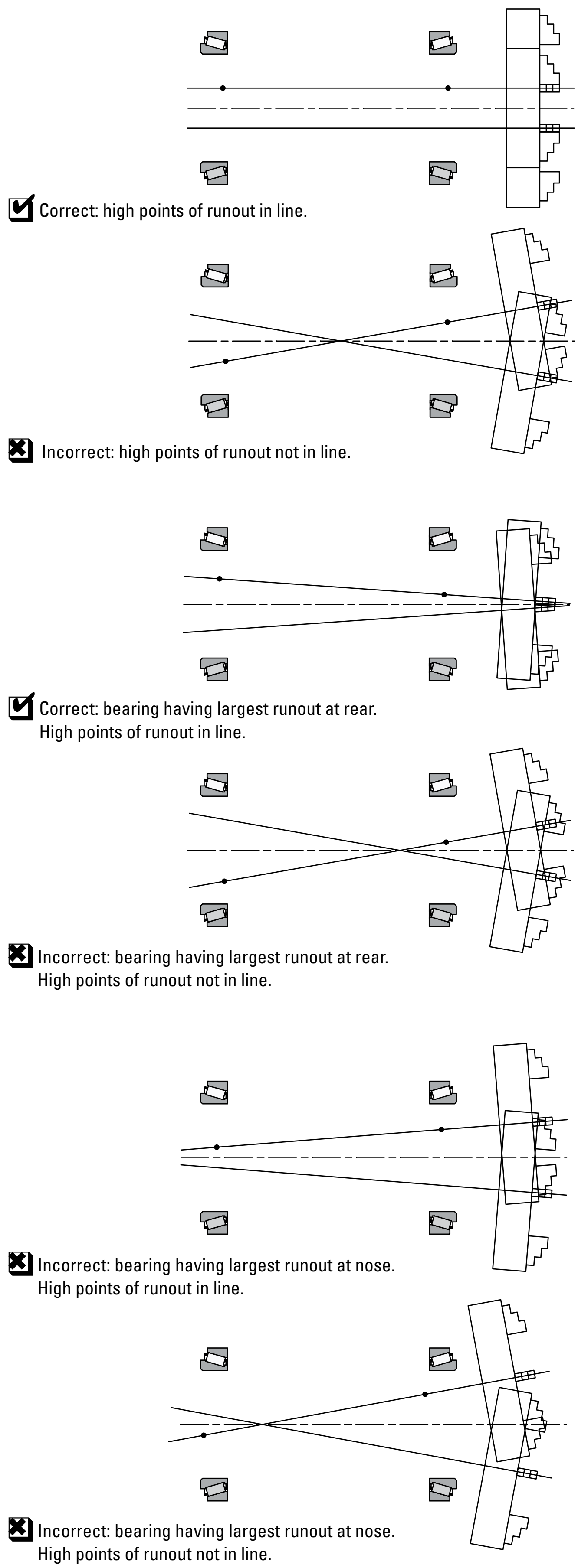


Fig. 92. The effect of bearing runout high point locations on spindle accuracy.

SETTING AND PRELOADING GUIDELINES

TAPERED ROLLER BEARINGS

The optimum operating setting of a bearing system has a direct influence on the spindle performance as far as accuracy, dynamic stiffness, operating temperature and cutting capabilities are concerned.

An operating setting range between zero and light preload is generally the optimum value for simple dual TS or TSF layouts.

To reach this range, it is important to evaluate the different parameters that will directly influence the operating setting in order to determine the cold-mounted setting:

- Rotating speed
- Applied loads
- Spindle layout
- Lubrication system
- External sources of heat

This evaluation occurs generally during the testing phase of the spindle because of the complexity of each individual parameter and the interaction of all of them during running conditions. At the same time, it also is important to consider the bearing layout and particularly the bearing spread to evaluate their effect on bearing setting.

It has been demonstrated that an optimum bearing spread for stiffness exists. In the same way, an optimum spread for thermal stability can be determined should this be the overriding factor.

Under steady-state temperature conditions, the spindle and housing temperature is not uniformly distributed. Generally, a temperature gradient of 2° C to 5° C (4° F to 9° F) exists between the spindle and housing. This phenomenon is valid for any type of bearing and has a direct influence on the bearing setting. In the case of pure radial bearings, such as cylindrical roller bearings, the radial setting will vary proportionally to the radial temperature gradient without any possibility for correction. The use of tapered roller bearings allows the radial loss of end play due to the gradient between the spindle and the housing to be compensated by the axial expansion of the spindle with respect to the housing through optimization of the bearing spread.

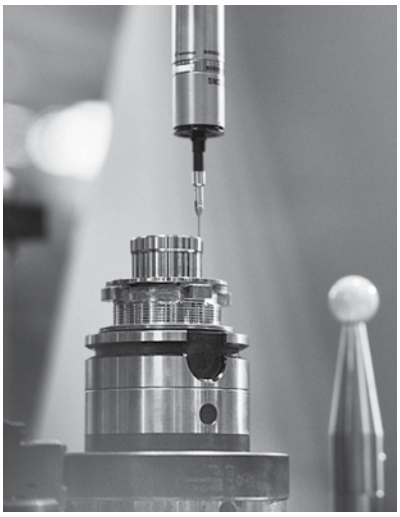


Fig. 93 shows a graphical way to determine this optimum spread. To define the optimum spread for thermal compensation or to calculate the effect on setting for a given spread in a simple 2TS(F) bearing system, the designer can use the formula below for ferrous housings and spindles.

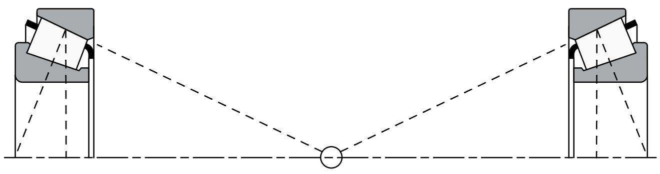


Fig. 93. Graphical determination of optimum thermal spread.

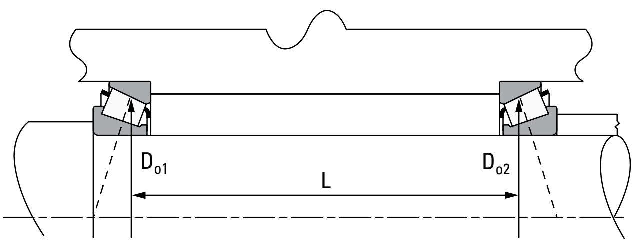


Fig. 94. Resulting distance between bearing geometric centerlines.

$$\text{Loss of end play} = 12 \times 10^{-6} \times t \times \left[\left(\frac{K_1}{0.39} \times \frac{D_{o1}}{2} \right) + \left(\frac{K_2}{0.39} \times \frac{D_{o2}}{2} \right) - L \right]$$

Where:

t	= temperature difference between shaft / inner ring rollers and housing / outer ring ($\theta_s - \theta_h$)	(°C)
K_1 and K_2	= respective K factor of bearings 1 and 2 from bearing tables	
D_{o1} and D_{o2}	= respective outer race mean diameter	(mm)
L	= distance between bearings geometric centerlines	(mm)

During the starting period, care must be taken because the radial thermal expansion is not immediately compensated by the axial expansion of the spindle. That occurs later. During this "transient period," a decrease of the axial end play or an increase of preload is generally recorded (Fig. 95). The loss of end play can be calculated by using the same formula, but ignoring the parameter "L." For this reason, it is generally recommended to initially set the bearings with a light, cold end play to avoid any bearing burn-up, due to excessive preload during the transient temperature rise. During the testing phase, it will be possible to modify this initial end play to obtain the optimum setting for the application.

Fig. 95 shows also that a three-support layout is more sensitive to thermal effects, leading to a higher temperature rise and loss of end play, than a simple arrangement because of the short bearing geometric spread at the fixed position.

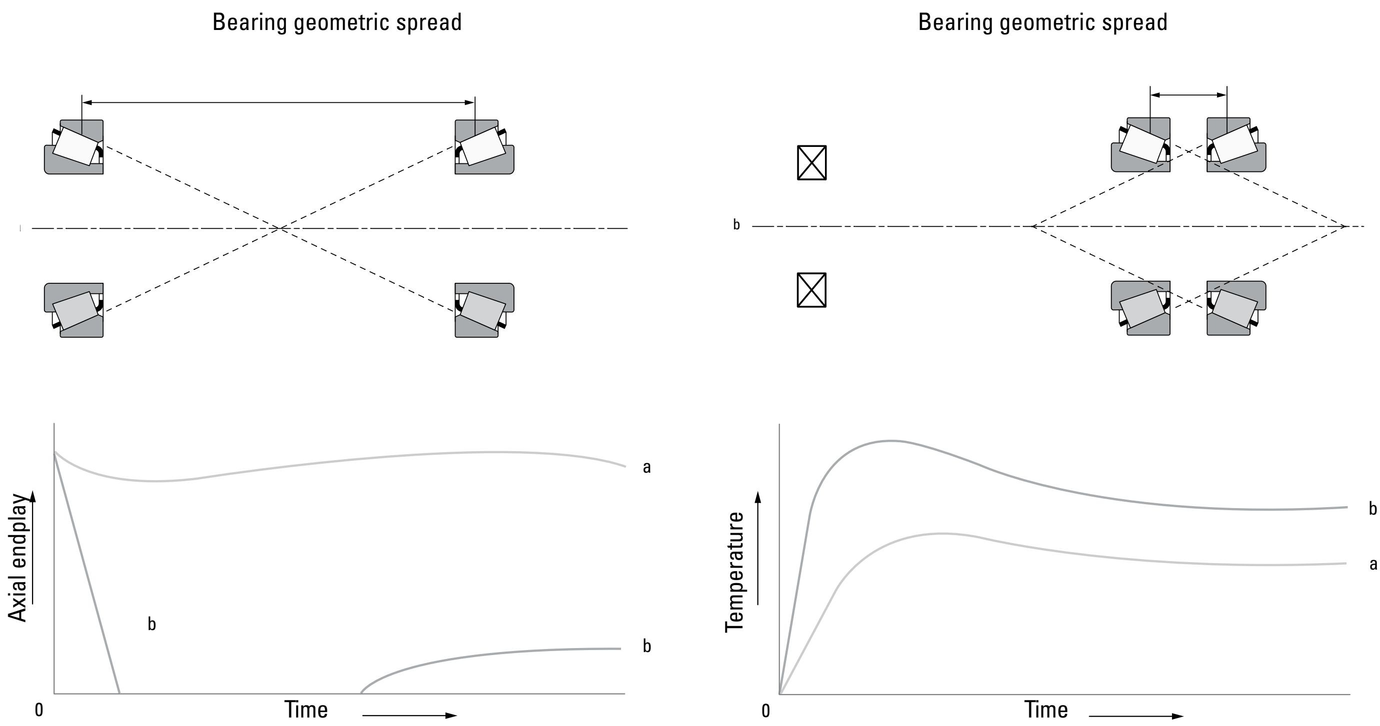


Fig. 95. Evolution of the spindle system setting and temperature during the transient period:
a) Simple mounting
b) Three-support mounting.

BALL BEARINGS

Preloading of precision ball bearings to a predetermined axial load for "universal" mounting is accomplished by grinding a certain amount of stock off faces of the inner and outer rings so that before mounting the bearing, faces on the abutting side are offset an amount equal to the deflection under "preload." When mounted, these faces are clamped together, the opposite bearing faces become flush and the bearing parts are subjected to compressive forces, bringing the balls into contact with their respective raceways to take up the initial clearances of the bearings. Thus, the preload built into the bearings is automatically obtained. The condition of a preloaded ball bearing is similar to that of one in operation under axial load. This initial axial load serves to decrease the axial and radial deflections when subsequent operational loads are imposed on the bearing assembly.

Bearings are preloaded no more than necessary. Excessive preload adds little to the rigidity of the spindle, but appreciably reduces the range of operating speeds by causing bearings to run hot at higher speeds. To meet conditions of speed, mounting arrangement and maximum rigidity consistent with low operating temperatures, Timken precision ball bearings are designed and produced with preloads varying from light to heavy and, in some instances, with negative preload.

In many cases, the amount of bearing preload is a trade-off between having the desired degree of rigidity and reducing any adverse effect preloading has on the equipment. If the operating speed is high, a heavy preload can lead to excessively high operating temperatures, resulting in shortened bearing life. For these reasons, three classes of ball bearing preloads are most commonly used – light, medium and heavy. In certain applications, such as high-speed motorized router spindles, specially preloaded, super precision ball bearings are required. Such bearings are "zero" preloaded – that is, the faces of the inner and outer rings are ground flush under negligible load.

The light, medium and heavy standard preload values for Timken super precision angular contact ball bearings and for both high and low contact angles are located with the dimension tables in Section C.

Axial deflection curves of various preload conditions for duplex pairs of 15 degree contact angle super precision ball bearings are shown in Fig. 96 and the radial deflection curves for the same bearings are shown in Fig. 97.

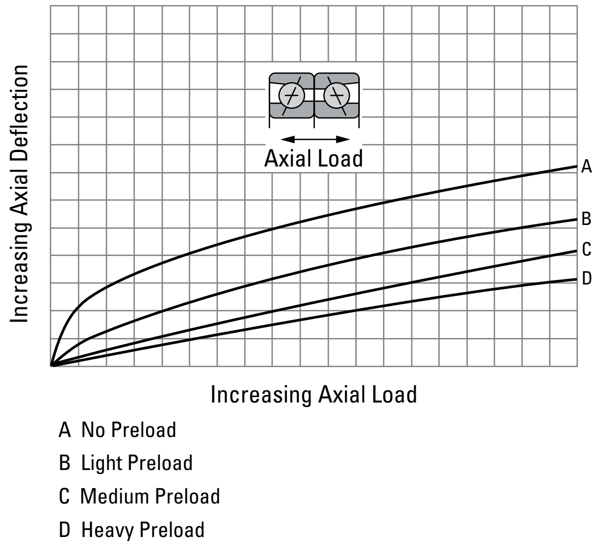


Fig. 96. Effect of most commonly used preloads on axial deflection.

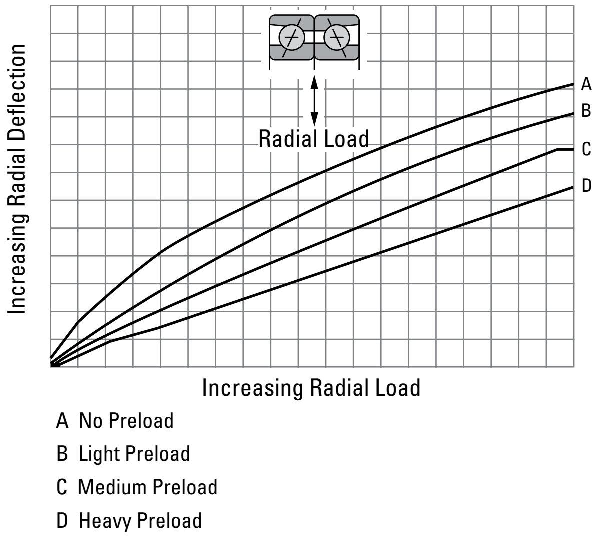


Fig. 97. Effect of most commonly used preloads on radial deflection.