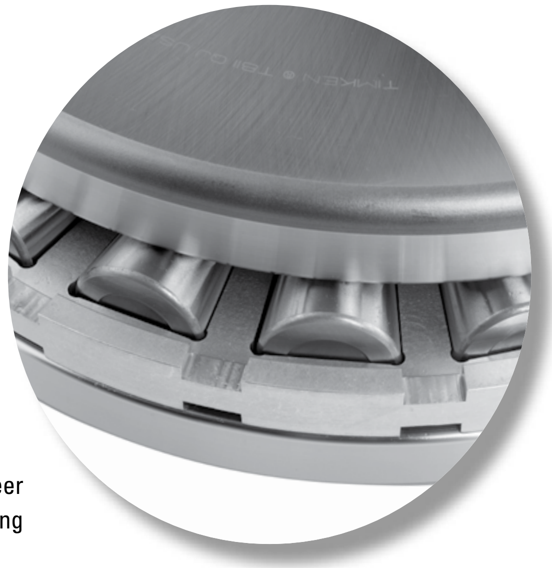

ENGINEERING

This engineering section is not intended to be comprehensive, but does serve as a useful guide in thrust bearing selection. To view the complete engineering catalog, please visit www.timken.com. To order the catalog, please contact your Timken engineer and request a copy of the Timken Engineering Manual (order no.10424).

The following topics are covered within this engineering section:

Thrust Bearing Types 14
Bearing Reactions 20
Bearing Ratings 22
System Life and Weighted Average Load and Life 28
Bearing Tolerances, Metric and Inch Systems 29
Mounting Design, Fitting Practice and Setting............. 37
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THRUST BEARING TYPES

Standard types of thrust bearings manufactured by Timken include:

TVL – Single-row angular contact thrust ball bearing.

DTVL – Double-row (two direction) angular contact thrust ball bearing.

TP – Thrust cylindrical roller bearing.

TPS – Self-aligning thrust cylindrical roller bearing.

TSR – Thrust spherical roller bearing.

TTHD – Heavy-duty thrust tapered roller bearing with two tapered raceways. Variants include:

- **TTHDSX** – where one tapered raceway has a convex outer surface for static alignment (SX).
- **TTHDSV** – where one tapered raceway has a concave outer surface for static alignment (SV).

TTHDFL – Heavy-duty thrust tapered roller bearing having one flat and one tapered raceway. Variants include:

- **TTHDFLSA** – where the flat raceway is made of two self-aligning washers (SA).
- **TTHDFLSX** – where the tapered raceway has a convex outer surface for static alignment (SX).
- **TTHDFLSV** – where the tapered raceway has a concave outer surface for static alignment (SV).

TTSP – Steering pivot thrust tapered roller bearing, off-apex design.

TTC – Steering pivot thrust tapered roller bearing, full complement (cageless).

TTD – Double-acting thrust tapered roller bearing.

TXR – Crossed roller bearing.

Each type is designed to take thrust loads. Types TVL, DTVL, TSR and TXR can accommodate radial loads as well. All types reflect advanced design concepts, with large rolling elements for maximum capacity. For some thrust roller bearings, controlled-contour rollers are used to ensure uniform, full-length contact between rollers and raceways resulting in maximum capacity.

Thrust bearings should operate under continuous load for satisfactory performance.

ANGULAR CONTACT THRUST BALL BEARINGS

Thrust ball bearings are used for lighter loads and higher speeds than thrust roller bearings.

TVL

Type TVL is a separable angular contact ball bearing primarily designed for unidirectional thrust loads. The angular contact design, however, will accommodate combined radial and thrust loads since the loads are transmitted angularly through the balls.

The bearing has two hardened and ground steel rings with ball grooves and a one-piece brass cage that spaces the ball complement. The larger ring is called the outer ring, and the smaller the inner ring. Timken standard tolerances for type TVL bearings are equivalent to ABEC 1 where applicable, but higher grades of precision are available.

Usually the inner ring is the rotating member and is shaft mounted. The outer ring is normally stationary and should be mounted with O.D. clearance to allow the bearing to assume its proper operating position. If combined loads exist, the outer ring must be radially located in the housing.

Type TVL bearings should always be operated under thrust load. Normally, this presents no problem as the bearing is usually applied on vertical shafts in oil field rotary tables and machine tool indexing tables. If constant thrust load is not present, it should be imposed by springs or other built-in devices.

Low friction, cool running and quiet operation are advantages of TVL bearings, which may be operated at relatively high speeds. TVL bearings also are less sensitive to misalignment than other types of rigid thrust bearings.

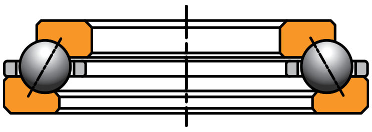


Fig. 1. Type TVL.

DTVL

Type DTVL is similar in design to TVL except that the DTVL has an additional ring and ball complement permitting it to carry moderate thrust in one direction and light thrust in the other direction.

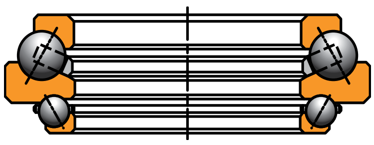


Fig. 2. Type DTVL.

THRUST CYLINDRICAL ROLLER BEARINGS

Thrust cylindrical roller bearings are designed to operate under heavy thrust loads at moderate speeds. Standard versions of these bearings can be operated at peripheral bearing O.D. speeds up to approximately 15 m/s (3000 fpm). Higher operating speeds can be attained with the incorporation of special design features. Consult your Timken engineer for these applications.

For applications where thrust loads are high, lubricants with extreme-pressure (EP) additives should be used. The preferred inlet location for the lubricant is at the bearing bore as centrifugal force will cause the lubricant to distribute radially toward the rollers.

Two types of thrust cylindrical roller bearings, TP and TPS, are available.

TP

Type TP thrust cylindrical roller bearings have two hardened and ground raceways and a window-type steel cage which retains one or more profiled rollers per pocket. When multiple rollers are used in each pocket, they are different lengths and are placed in staggered position relative to rollers in adjacent pockets to create overlapping roller paths. This minimizes wear of the raceways and therefore increases bearing life.

Because of the simplicity of their design, type TP bearings are economical. Shaft and housing seats must be square to the axis of rotation to prevent initial misalignment problems.

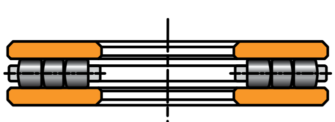


Fig. 3. Type TP.

TPS

Type TPS bearings have a lower race comprised of two rings, with the contacting faces spherically ground to provide an aligning feature. As a result, the TPS bearing is self-adjusting to static misalignment. Its use is not, however, suggested for operating conditions where alignment is continuously changing (dynamic misalignment).

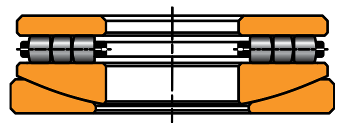


Fig. 4. Type TPS.

THRUST SPHERICAL ROLLER BEARINGS

Thrust spherical roller bearings are designed with spherically contoured rollers arranged in a steep angular configuration to achieve a high-thrust capacity with low friction and continuous roller alignment. In addition to thrust loads, they can accommodate moderate radial loads. Maximum allowable bearing O.D. speeds are typically in the 25-30 m/s (5000-6000 fpm) range, depending on size and operating temperature. They represent a combination of radial and thrust bearings, designed to operate even if shaft and housing are, or become, misaligned under load. Thrust spherical roller bearings are preferred when conditions include heavy loads, difficulties in establishing or maintaining housing alignment or when shaft deflection can be expected.

Shaft deflections and housing distortions caused by shock or heavy loads (which lead to misalignment) are compensated for by the internal self-alignment of the bearing elements during operation. Elevated edge stress on rollers, a condition that limits service life on other types of bearings, does not develop in thrust spherical roller bearings.

The thrust spherical roller bearing achieves high-thrust capacity and allows axial misalignment between the inner ring and the outer ring of up to ±2.5˚. Timken thrust spherical roller bearings are now offered exclusively with maximum capacity E-type cage construction (EM-finger type machined bronze cage, EJ- window type steel cage). Those having a bore size less than 320 mm (12.598 in.) are typically offered as TSR-EJ designs, while those with larger bores are typically designated as TSR-EM.

The inherent compensation for misalignment, provided by the spherical roller bearings, offers the designer the opportunity to use weldments for housing frames instead of complex castings. This eliminates high-cost machining operations. When castings are preferred, bore alignment is less critical if spherical roller bearings are specified. Should extreme conditions of loading and/or speed under misalignment be anticipated, contact your Timken engineer before ordering.

TSR-EJ

TSR-EJ bearings use window-type steel cages that wrap around an extension on the inner race to provide a retention means for the cage and rollers. This construction unitizes the cage and roller assembly with the inner ring, and hence simplifies bearing mounting and handling.

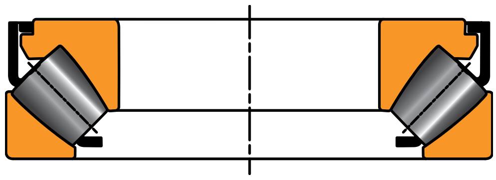


Fig. 5. Type TSR-EJ.

TSR-EM

TSR-EM bearings use finger-type brass cages. The brass cage design provides improved lubrication characteristics over a steel cage and in some cases allows for an additional roller, resulting in higher dynamic load rating. TSR-EM bearings have a roller retention ring, also known as the cage band, mounted and secured to the inner ring to retain the rollers.

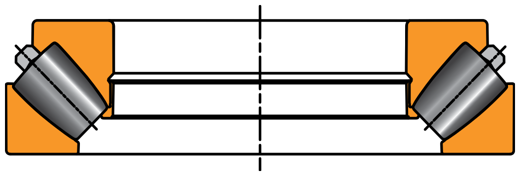


Fig. 6. Type TSR-EM.

THRUST TAPERED ROLLER BEARINGS

Thrust tapered roller bearings come in various types and within each type, there are typically several variations. The variation is denoted by a suffix in the bearing type as noted below.

D	Double acting
HD	Heavy duty
FL	Flat or freelateral
K	Keyway
SA	Spherical alignment
SV	Spherical concave ring outer profile
SX	Spherical convex ring outer profile
W	Oil slots

TTHD, TTHDSX AND TTHDSV

Type TTHD heavy-duty thrust tapered roller bearings have an identical pair of hardened and ground steel rings with tapered raceways, controlled-contour tapered rollers and typically a cage to equally space the rollers. The raceways of both rings and the tapered rollers have a common apex at the bearing center, providing true rolling motion. As a result, maximum speed ratings for TTHD bearings are higher than those of most other thrust bearing types. Type TTHD bearings also can be supplied with a full complement of rollers for low-speed, heavily loaded applications. Full-complement designs offer the highest capacity at somewhat reduced speed capability. Applications for full-complement bearings should be reviewed by your Timken engineer for help in selection of the proper bearing.

TTHD bearings are well-suited for applications where high thrust and/or heavy shock loads are applied and radial positioning is critical. Typical applications for TTHD bearings include crane hooks, oil well swivels, pulp refiners, extruders and piercing mill thrust blocks.

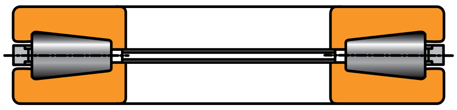


Fig. 7. Type TTHD.

Type TTHDSX and TTHDSV thrust tapered roller bearings have tapered raceways and a full complement of rollers. They are commonly known as screw down bearings in the metals industry. Outer raceways for TTHDSX and TTHDSV bearings have convex and concave outer surfaces, respectively, for the purpose of set-up alignment. They do not have a conventional bore, but are provided with center inserts for attachment purposes as well as lifting.

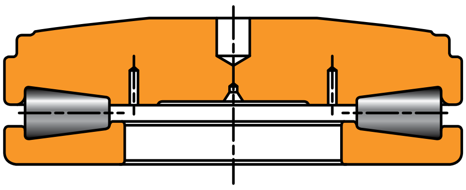


Fig. 8. Type TTHDSX.

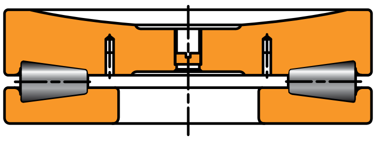


Fig. 9. Type TTHDSV.

TTHDFL, TTHDFLSA, TTHDFLSX AND TTHDFLSV

Types TTHDFL, TTHDFLSA, TTHDFLSX and TTHDFLSV heavy-duty thrust bearings have one tapered raceway, one flat raceway and controlled-contour rollers to optimize stress distribution over the contact surface. These designs combine features offering the highest possible capacity of any thrust bearing of their size and providing superior static thrust capacity. The designs were originally developed for metal scewdown rolling mill (breaker block) applications. They also are used in heavily loaded extruders, cone crushers and other applications where a wide range of operating conditions are found.

Type TTHDFL bearings typically use brass cages for smaller sizes and pin-type cages for larger sizes. The pin-type cage includes hardened pins which are inserted through the rollers, allowing closer roller spacing to maximize capacity. Smaller sizes typically use pocket-type machined brass cages. Both the brass and pin-type cages are designed to permit a full flow of lubricant to all critical surfaces, providing cooler operation.

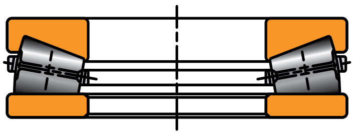


Fig. 10. Type TTHDFL.

Type TTHDFLSA bearings are similar to TTHDFL, except that the bottom race assembly is comprised of two rings, with the contacting faces spherically ground. As a result, the TTHDFLSA bearing is self-adjusting to static misalignment. It should not be used for operating conditions where alignment is continuously changing (dynamic misalignment).

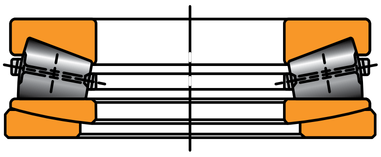


Fig. 11. Type TTHDFLSA.

Types TTHDFLSX and TTHDFLSV are full-complement designs having one raceway with either a convex or concave outer surface for the purpose of static alignment. They are commonly known as screw down bearings in the metals industry. They do not have a conventional bore, but are provided with center inserts for lifting and assembly. The full-complement design offers the highest capacity, but a reduced speed capability compared to other thrust bearings having a flat raceway.

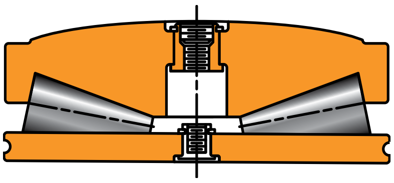


Fig. 12. Type TTHDFLSX.

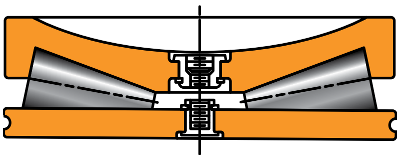


Fig. 13. Type TTHDFLSV.

TTSP, TTSPS AND TTSPL

Types TTSP, TTSPS and TTSPL thrust bearings consist of two tapered races, rollers, cage and outside retainer. The retainer holds the assembly together for shipping and installation. The raceways are off-apex, which means they do not provide true rolling motion. These thrust bearing types are used extensively in oscillating steering pivot applications.

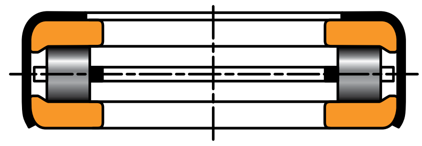


Fig. 14. Type TTSP.

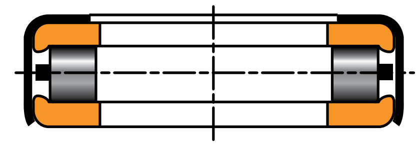


Fig. 15. Type TTSPS.

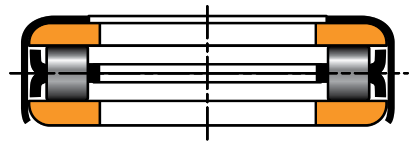


Fig. 16. Type TTSPL.

TTC, TTCS AND TTCL

Types TTC, TTCS and TTCL are cageless thrust bearings that consist of two tapered thrust raceways, a full complement of tapered rollers and an outside retainer. The outside retainer holds the assembly together for shipping and installation. These types are specifically designed for slow speed and oscillating applications and are identical with the exception of retainer construction.

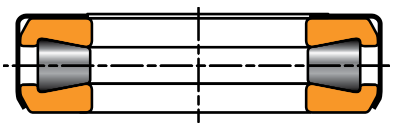


Fig. 17. Type TTC.

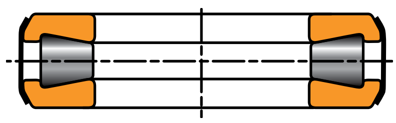


Fig. 18. Type TTCS.

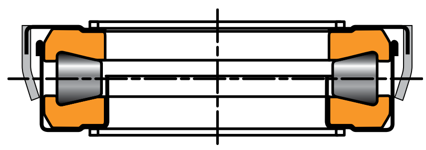


Fig. 19. Type TTCL.

TTD

Type TTD bearings are double-acting thrust tapered roller bearings that can take thrust loads in both axial directions. The inner ring is one piece having two separate raceways, one on each of the outer surfaces. These raceways can be either flat or tapered. For a flat inner raceway, the mating outer ring raceway is tapered and for a tapered inner raceway, the outer ring is flat. The outer rings and cage roller assembly are separable and are not interchangeable. Variations of the TTD bearing include the following features:

- **TTDW** with oil slots.
- **TTDWK** with oil slots and keyway.
- **TTDK** keyway (see variants in figs. 21-22).
- **TTDFL** with flat outer ring raceway.
- **TTDFLK** with flat inner ring raceway and keyway.

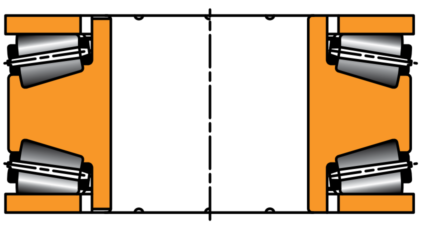


Fig. 20. Type TTDW.

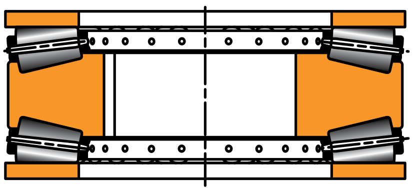


Fig. 21. Type TTDK 1.

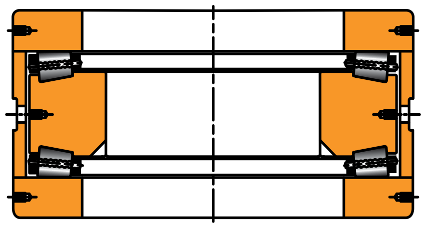


Fig. 22. Type TTDK 2.

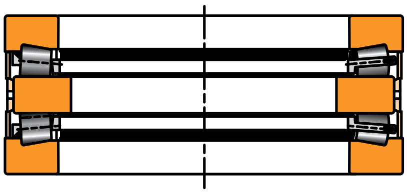


Fig. 23. Type TTDFLK.

TXR – CROSSED ROLLER BEARINGS

A crossed roller bearing is two sets of bearing rings and rollers brought together at right angles with alternate rollers facing opposite directions. TXR bearings have a section height not much greater than that of a TS bearing. The steep angle, tapered geometry of the bearing results in a total effective bearing spread many times greater than the width of the bearing itself. This type of bearing offers a high resistance to overturning moments.

The normal design of the bearing is type TXRDO, which has a double outer ring and two inner rings, with rollers spaced by polymer cages. Another design, Type TXRDI, has a double inner-ring and two outer rings. Crossed roller bearings are manufactured in precision classes. The crossed roller bearing is ideal for machine tool applications such as vertical boring mills, grinding machines, and other similar applications.

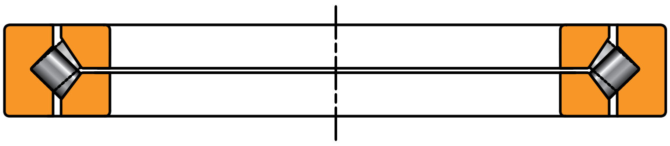


Fig. 24. Type TXR crossed roller bearings.

BEARING REACTIONS

DYNAMIC EQUIVALENT THRUST LOAD (P_a)

To calculate the fatigue life of a thrust bearing, it is necessary to calculate a dynamic equivalent thrust load, designated as P_a. The dynamic equivalent thrust load is defined as the single thrust load that, if applied to the bearing, will result in the same life as the combined radial and thrust loading under which the bearing operates. For thrust ball, thrust spherical and thrust tapered roller bearings, the existence of radial loads introduces complex load calculations that must be carefully considered. If the radial load (F_r) is zero, the dynamic equivalent thrust load will be equal to the applied thrust load (F_a).

THRUST BALL, CYLINDRICAL AND TAPERED ROLLER BEARINGS

Thrust cylindrical roller bearings, as well as most thrust ball and thrust tapered roller bearings, are designed to carry thrust load only. The dynamic equivalent thrust load is equal to the applied thrust load (F_a) for these pure thrust applications. For thrust ball and thrust tapered roller bearing applications where radial load is applied, load calculations become much more complex. Please consult your Timken engineer for a review of bearing selection and application.

ANGULAR CONTACT THRUST BALL BEARINGS

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

$$P_a = X F_r + Y F_a$$

For standard TVL and DTVL bearings having a 50° contact angle, X = 0.76 and Y = 1.00. Minimim F_a/F_r ratio to maintain proper operation for these applications is 1.56.

THRUST SPHERICAL ROLLER BEARINGS

Thrust spherical roller bearing dynamic loads are determined by:

$$P_a = 1.2F_r + F_a$$

Radial load (F_r) of a thrust spherical roller bearing is proportional to the applied axial load (F_a) such that $F_r \leq 0.55\ F_a$. The steep roller angle induces a thrust load ($F_{ai} = 1.2F_r$) when a radial load is applied. This thrust load must be resisted by another thrust bearing on the shaft or by an axial load greater than F_{ai}.

STATIC AXIAL EQUIVALENT LOADS

To compare the load on a non-rotating bearing with the basic static capacity, it is necessary to determine the static equivalent load. In the case of thrust bearings, the static equivalent thrust load is used. The static axial equivalent load is defined as the pure thrust load that produces the same contact pressure in the center of the most heavily stressed rolling element as the actual combined load. The static axial equivalent load is dependent on the bearing type selected. For bearings such as thrust cylindrical roller bearings and most thrust tapered roller bearings that are designed to accommodate thrust loading only, the static axial equivalent load is equal to the applied load. For thrust tapered roller bearings where a radial load or moment is applied, please consult your Timken engineer.

THRUST BALL, CYLINDRICAL AND TAPERED ROLLER BEARINGS

Thrust cylindrical roller bearings, as well as most thrust ball and thrust tapered roller bearings, are designed to carry thrust load only. The static axial equivalent load is equal to the applied thrust load for these pure thrust applications. For thrust ball and thrust tapered roller bearing applications where radial load is applied, load calculations become much more complex. Please consult your Timken engineer for these applications.

ANGULAR CONTACT THRUST BALL BEARINGS

Angular contact thrust ball bearings use the same equation for equivalent static and dynamic loading.

$$P_{oa} = X_o F_r + Y_o F_a$$

For standard TVL and DVL bearings having a 50° contact angle, X = 0.76 and Y = 1.00.

THRUST SPHERICAL ROLLER BEARINGS

The following equation is used for thrust spherical roller bearings:

$$P_{oa} = F_a + 2.7\ F_r$$

Thrust spherical roller bearings require a minimum thrust load for proper operation. P_{oa} should not be greater than 0.5 C_{oa}. If conditions exceed this, consult your Timken engineer.

MINIMUM BEARING LOAD

THRUST SPHERICAL ROLLER BEARINGS

Centrifugal force in thrust spherical roller bearings tends to propel the rollers outward. The bearing geometry converts this force to induced thrust component, which must be overcome by an axial load. This induced thrust (F_{ac}) is given by:

$$F_{ac} = Kc\ n^2 \times 10^{-5} \text{ (lbf per RPM)}$$

Kc = centrifugal force constant found in product tables pages 87-91

The minimum required working thrust load on a thrust spherical roller bearing ($F_{a\,min}$) is then computed by:

$$F_{a\,min} = 1.2\ F_r + F_{ac} >= C_{0a}/1000 \text{ (lbf)}$$

In addition to meeting the above calculated value, the minimum required working thrust load ($F_{a\,min}$) should be equal to or greater than 0.1 percent of the static thrust load rating (C_{0a}).

BEARING RATINGS

There are two fundamental load ratings for bearings, a dynamic load rating and a static load rating. The dynamic load rating is used to estimate the 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.

DYNAMIC LOAD RATING

Published dynamic load ratings for Timken bearings are typically based on a rated life of one million revolutions. This rating, designated as C, is defined as the radial load under which a population of bearings will achieve an L_{10} life of one million revolutions. For Timken tapered roller bearings, the dynamic load rating is more commonly based on a rated life of 90 million revolutions, with the designation of C_{90}. This rating is the radial load under which a population of bearings will achieve an L_{10} life of 90 million revolutions. For tapered roller bearings, the dynamic thrust rating also is published and is designated as C_{a90}. The C_{a90} rating is the thrust load under which a population of bearings will achieve an L_{10} life of 90 million revolutions. The dynamic load rating of a bearing is a function of material cleanliness as well as the internal bearing geometry, which includes raceway angles, contact length between rolling elements and raceways, and the number and size of rolling elements.

STATIC LOAD RATING

The basic static radial load rating and thrust load rating 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 is critical, or if a pronounced shock load is present, a lower load limit should be applied. For more information on selecting a bearing for static load conditions, consult your Timken engineer.

BEARING 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 section deals primarily with bearing life as related to material-associated fatigue. Bearing life is defined as the length of time, or number of revolutions, until a fatigue spall of 6 mm^2 (0.01 in.^{2}) develops. Since 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 the accepted standard for predicting 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 and the dynamic load rating is based on one million cycles:

$$L_{10} = \left(\frac{C}{P_r}\right)^e (1x10^6) \quad \text{revolutions}$$

or

$$L_{10} = \left(\frac{C}{P_r}\right)^e \left(\frac{1x10^6}{60n}\right) \quad \text{hours}$$

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

$$L_{10} = \left(\frac{C_a}{P_a}\right)^e (1x10^6) \quad \text{revolutions}$$

or

$$L_{10} = \left(\frac{C_a}{P_a}\right)^e \left(\frac{1x10^6}{60n}\right) \quad \text{hours}$$

e = 3 for ball bearings
= $^{10}/_3$ for tapered, cylindrical and spherical roller bearings

Tapered roller bearings typically use a dynamic load rating based on 90 million cycles, denoted as C_{90}, changing the equations as follows:

$$L_{10} = \left(\frac{C_{90}}{P_r}\right)^{10/3} (90x10^6) \quad \text{revolutions}$$

or $$L_{10} = \left(\frac{C_{90}}{P_r}\right)^{10/3} \left(\frac{90x10^6}{60n}\right) \quad \text{hours}$$

and $$L_{10} = \left(\frac{C_{a90}}{P_a}\right)^{10/3} (90x10^6) \quad \text{revolutions}$$

or $$L_{10} = \left(\frac{C_{a90}}{P_a}\right)^{10/3} \frac{90x10^6}{60n} \quad \text{hours}$$

The traditional form of the equations based on dynamic load ratings of one million cycles is most common and will, therefore, be used throughout the rest of this section. The dynamic equivalent load equations and the life adjustment factors defined in subsequent sections 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 or factors that affect bearing performance. The approach whereby these factors are considered in the bearing analysis and selection has been termed Bearing Systems Analysis (BSA). For thrust bearings, these factors are currently only applied to thrust tapered and thrust spherical roller bearings. The ABMA expanded bearing life equation is:

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

The Timken expanded bearing life equation is:

$$L_{na} = a_1 a_2 a_{3d} a_{3l} a_{3m} a_{3p} \left(\frac{C}{P_r}\right)^e (1x10^6) \quad \text{revolutions}$$

Where: e = 3 for ball bearings
= $^{10}/_3$ for tapered, cylindrical and spherical roller bearings

RELIABILITY LIFE FACTOR (a_1)

Reliability, in the context of bearing life for a group of apparently identical bearings operating under the same conditions, is the percentage of the group that is expected to attain or exceed a specified life. The reliability of an individual bearing is the probability that the bearing will attain or exceed a specified life.

The reliability life adjustment factor is:

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

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 table below lists the reliability factors for commonly used reliability values.

TABLE 1. 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: 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 predicted using the above adjustment factor. For a more accurate prediction of bearing lives at high levels of reliability, consult your Timken engineer.

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® bearing). 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 engineer for applicability of the material factor.

DEBRIS LIFE FACTOR (a_{3d})

Debris within a lubrication system reduces the life of a roller 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 40 μm oil filtration, and measured ISO cleanliness levels of approximately 15/12, which is typical of cleanliness 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 cleanliness level to more accurately reflect the expected bearing performance.

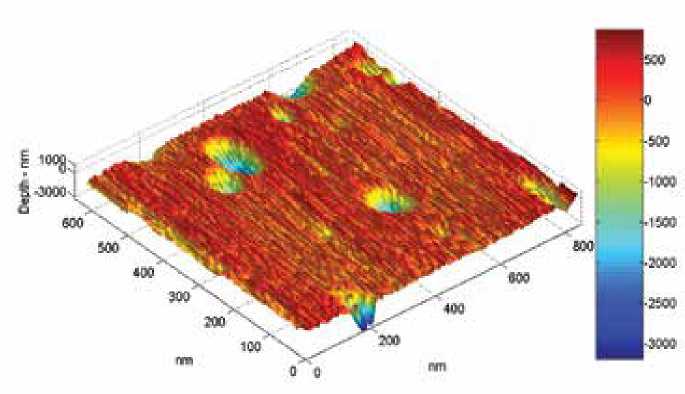


Fig. 25. Surface map of a bearing raceway with debris denting.

A more accurate option for predicting bearing life in a debris environment is to perform a Debris Signature Analysis™. 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 degradation. Soft, ductile particles can cause differing levels of 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 of 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 engineer.

LUBRICATION LIFE FACTOR (a_{3l})

The influence of lubrication film on bearing performance is related to the reduction or prevention of asperity (metal-metal) contact between the bearing surfaces. Extensive testing has been done at at our technology centers to quantify the effects of the lubrication-related parameters on bearing life. It has been found that the roller 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 bearing life (L_{10a}):

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

The a_{3l} maximum is 2.88 for all bearings. The a_{3l} minimum is 0.200 for case-carburized 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 40 µm filter to provide a realistic level of lubricant cleanness for most applications.

Geometry factor (Cg)

C_g is given for most part numbers that are available in the bearing catalogs on www.timken.com. The geometry factor also includes the material effects and load zone considerations for non-tapered roller bearings, as these also are inherent to the bearing design. However, it should be noted that the primary effect of the load zone is on roller 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.

The geometry factor (C_g) is not applicable to our DuraSpexx™ product. For more information on our DuraSpexx product, consult your Timken engineer.

Load factor (C_l)

The C_l factor can be obtained from fig. 26. 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 Dynamic Equivalent Bearing Loads (P_r) section.

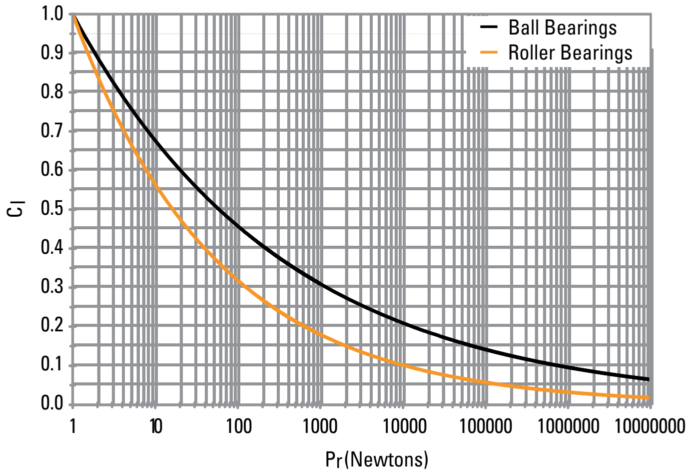


Fig. 26. Load factor (C_l) vs. dynamic equivalent bearing load (P_r).

Speed factor (C_s)

C_s can be determined from fig. 27, where rev/min (RPM) is the rotational speed of the inner ring relative to the outer ring.

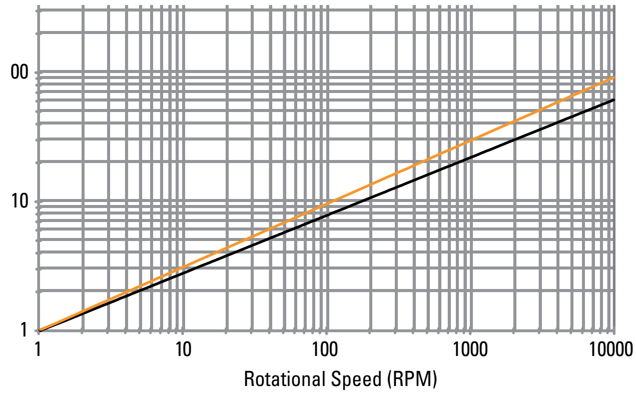


Fig. 27. Speed factor (C_s) vs. rotational speed.

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. 28. The viscosity factor (C_v) can then be determined from figs. 28 and 29 shown here.

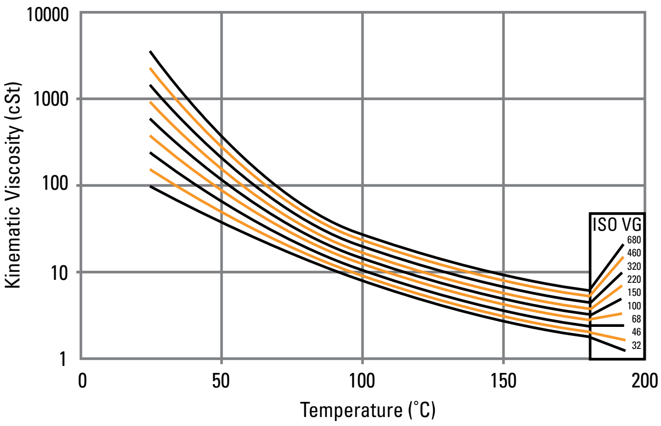


Fig. 28. Temperature vs. kinematic viscosity.

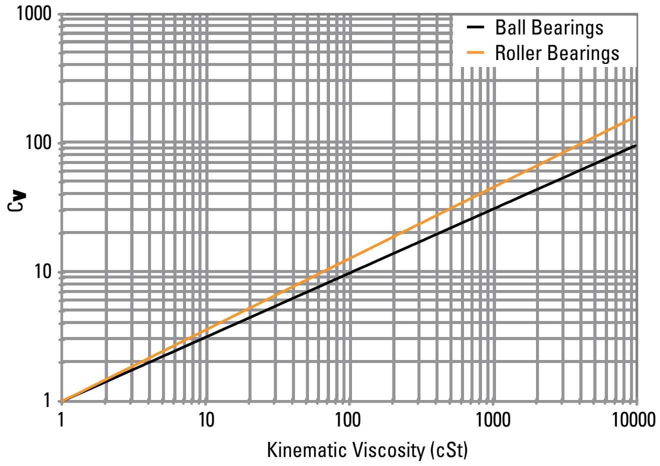


Fig. 29. Viscosity factor (C_v) vs. kinematic viscosity.

LOW-LOAD LIFE FACTOR (a_{3p})

Bearing life tests show 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 to predict the life increase expected when operating under low-bearing loads. Fig. 30 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.

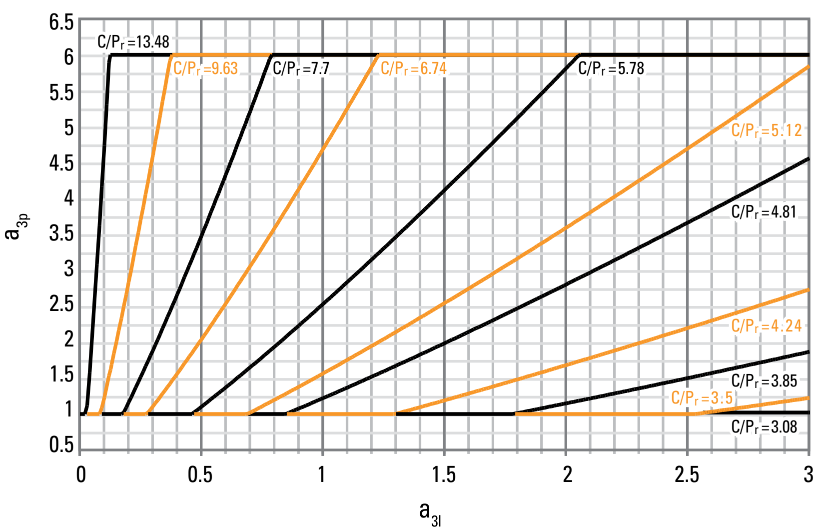


Fig. 30. Low-load life adjustment factor.

GREASE LUBRICATION FACTOR (C_{gr})

Over time, grease degradation causes a reduction in lubricant film thickness. Consequently, a reduction factor (C_{gr}) should be used to adjust for this effect.

$$C_{gr} = 0.79$$

MISALIGNMENT LIFE FACTOR (a_{3m})

Accurate alignment of the shaft relative to the housing is critical for bearing performance. As misalignment increases under moderate to heavy loads, high contact stresses can be generated at the edges of contact between the raceway and rolling element. Special profiling of the raceway or rolling element can, in most cases, offset the effects of misalignment as shown in fig. 31. This figure shows the roller-to-inner ring contact stress of a tapered roller bearing under a misaligned condition with and without special profiling. The profiling significantly reduces the edge stress, resulting in improved bearing performance. The misalignment factor takes into account the effects of profiling on bearing life.

The misalignment factor for thrust spherical roller bearings is 1.0 due to their self-aligning capabilities. The allowable misalignment of a thrust spherical roller bearing is ± 2.5 degrees. Life will be reduced if these limits are exceeded. For misalignment factors for other thrust bearing types, contact your Timken engineer.

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 engineer for more information.

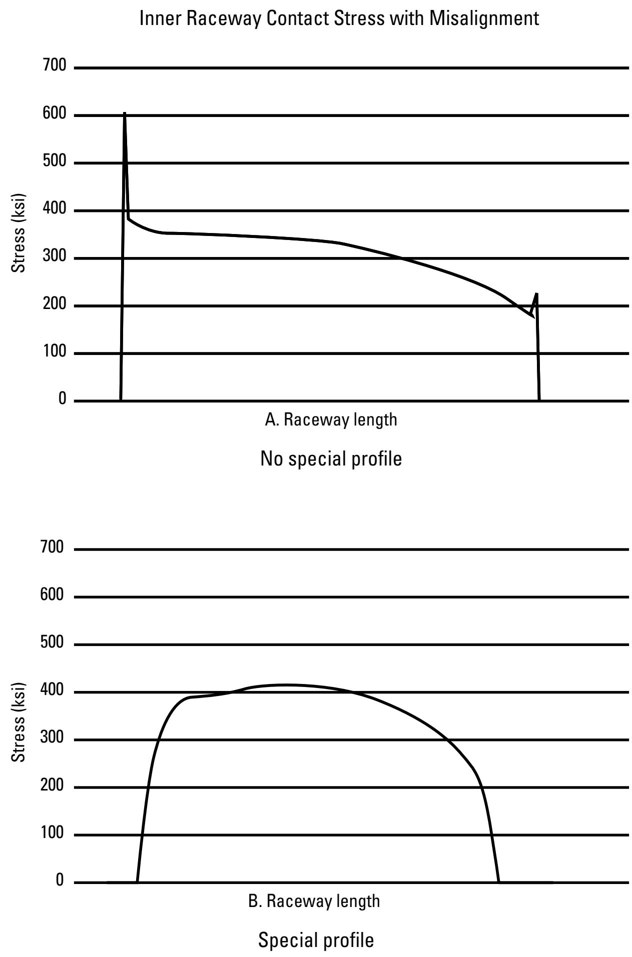


Fig. 31. Tapered roller bearing contact stress under misaligned condition.

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 LOAD AND LIFE 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 by 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 speeds 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:

$$F_{wt} = [(n_1 t_1 F_1^{10/3} + \ldots n_n t_n F_n^{10/3}) / n_a]^{0.3}$$

Uniformly increasing load, constant speed:

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

Use of the weighted average load in the bearing life equation does not take into account the effects of different speeds on the lubrication factor a_{3l}. For load cycles with varying speeds, it is recommended that life calculations be made for each condition and that the life for each condition be plugged into the weighted average life equation.

WEIGHTED AVERAGE LIFE

$$L_{nwt} = 1/\left\{[t_1 / (L_n)_1] + [t_2 / (L_n)_2] + \ldots [t_n / (L_n)_n]\right\}$$

BEARING TOLERANCES, METRIC AND INCH SYSTEMS

Ball and roller bearings are manufactured to a number of specifications, with each having classes that define dimensional tolerances such as inside diameter, outside diameter, width and runout. In addition, bearings are produced in both inch and metric systems with the boundary dimension tolerances being different for these two systems. The major difference between the two systems is that inch bearings have historically been manufactured to positive bore and O.D. tolerances, whereas metric bearings have been manufactured to corresponding standard negative tolerances.

The following table summarizes the different specifications and classes for ball, tapered roller, cylindrical roller and spherical roller bearings. For the purpose of this catalog, ISO specifications are shown for ball, cylindrical roller and spherical roller bearings. Timken specifications are shown for tapered roller bearings. Timken® thrust tapered roller bearings comply with current ABMA inch system standard 23.2. Standard Timken® ball, spherical roller and cylindrical roller thrust bearings maintain normal metric system tolerances according to the current ISO standard 199.

TABLE 2. BEARING SPECIFICATIONS AND CLASSES

System	Specification	Bearing Type	Standard Bearing Class		Precision Bearing Class	
Metric	Timken	Tapered Roller Bearings	K	N	C	B
	ISO/DIN	All Bearing Types	P0	P6	P5	P4
	ABMA	Cylindrical, Spherical Roller Bearings	RBEC 1	RBEC 3	RBEC 5	RBEC 7
		Ball Bearings	ABEC 1	ABEC 3	ABEC 5	ABEC 7
		Tapered Roller Bearings (Not XR)	K	N	C	B
		Crossed Roller Bearings	–	–	S	P
Inch	Timken	Tapered Roller Bearings	4	2	3	0
	ABMA	Tapered Roller Bearings	4	2	3	0

The term deviation is defined as the difference between a single ring dimension and the nominal dimension. For metric tolerances, the nominal dimension is at a +0 mm (0 in.) tolerance. The deviation is the tolerance range for the listed parameter. Variation is defined as the difference between the largest and smallest measurements of a given parameter for an individual ring.

Boundary dimension tolerances for Timken thrust bearings are listed in the following tables (pages 30-35). These tolerances are provided for use in selecting bearings for general applications in conjunction with the bearing mounting and fitting practices offered in later sections.

THRUST BALL BEARING TOLERANCES

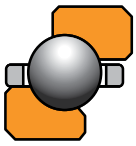
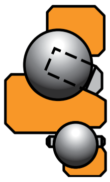



TABLE 3. THRUST BALL BEARING TOLERANCES – TYPES TVL AND DTVL

Bore			O.D.			Width		
Bearing Bore		Tolerance(1)	Bearing O.D.		Tolerance(1)	Bearing Width		Tolerance
Over	Incl.		Over	Incl.		Over	Incl.	Max.
mm in.	**mm** in.	**mm** in.	**mm** in.	**mm** in.	**mm** in.	**mm** in.	**mm** in.	**mm** in.
0.000 0.0000	**504.825** 19.8750	**-0.076** -0.0030	**0.000** 0.0000	**584.000** 23.0000	**-0.076** -0.0030	All Sizes		**±0.381** ±0.0150
504.825 19.8750	**1524.000** 60.0000	**-0.127** -0.0050	**584.000** 23.0000	**1778.000** 70.0000	**-0.127** -0.0050			– –

(1)The tolerances in this table conform to ABMA Standard 21.2.

THRUST SPHERICAL ROLLER BEARING TOLERANCES


TABLE 4. THRUST SPHERICAL ROLLER BEARING TOLERANCES

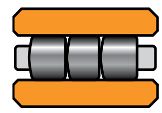
Bore			O.D.			Width			
Bearing Bore		Tolerance(1)	Bearing O.D.		Tolerance(1)	Bearing Width		Tolerance	
Over	Incl.		Over	Incl.		Over	Incl.	Max.	Min.
mm in.	**mm** in.	**mm** in.	**mm** in.	**mm** in.	**mm** in.	**mm** in.	**mm** in.	**mm** in.	**mm** in.
80.000 3.1496	**120.000** 4.7244	**-0.020** -0.0008	**120.000** 4.7244	**150.000** 5.9055	**-0.020** -0.0080	**80.000** 3.1496	**120.000** 4.7244	**+0.094** +0.0037	**-0.254** -0.0100
120.000 4.7244	**180.000** 7.0866	**-0.025** -0.0010	**150.000** 5.9055	**180.000** 7.0866	**-0.025** -0.0010	**120.000** 4.7244	**180.000** 7.0866	**+0.109** +0.0043	**-0.300** -0.0118
180.000 7.0866	**250.000** 9.8425	**-0.030** -0.0012	**180.000** 7.0866	**250.000** 9.8425	**-0.030** -0.0012	**180.000** 7.0866	**250.000** 9.8425	**+0.130** +0.0051	**-0.366** -0.0144
250.000 9.8425	**315.000** 12.4016	**-0.036** -0.0014	**250.000** 9.8425	**315.000** 12.4016	**-0.036** -0.0014	**250.000** 9.8425	**315.000** 12.4016	**+0.155** +0.0061	**-0.434** -0.0171
315.000 12.4016	**400.000** 15.7480	**-0.041** -0.0016	**315.000** 12.4016	**400.000** 15.7480	**-0.041** -0.0016	**315.000** 12.4016	**400.000** 15.7480	**+0.170** +0.0067	**-0.480** -0.0189
400.000 15.7480	**500.000** 19.6850	**-0.046** -0.0018	**400.000** 15.7480	**500.000** 19.6850	**-0.046** -0.0018	**400.000** 15.7480	**500.000** 19.6850	**+0.185** +0.0073	**-0.526** -0.0207
500.000 19.6850	**630.000** 24.8031	**-0.051** -0.0020	**500.000** 19.6850	**630.000** 24.8031	**-0.051** -0.0020	**500.000** 19.6850	**and up**	**+0.203** +0.0080	**-0.584** -0.0230
630.000 24.8031	**800.000** 31.4961	**-0.076** -0.0030	**630.000** 24.8031	**800.000** 31.4961	**-0.076** -0.0030	– –	– –	– –	– –
800.000 31.4961	**1000.000** 39.3701	**-0.102** -0.0040	**800.000** 31.4961	**1000.000** 39.3701	**-0.102** -0.0040	– –	– –	– –	– –
1000.000 39.3701	**1250.000** 49.2126	**-0.127** -0.0050	**1000.000** 39.3701	**1250.000** 49.2126	**-0.127** -0.0050	– –	– –	– –	– –
– –	– –	– –	**1600.000** 62.9921	**-0.165** -0.0065	**0.193** 0.0076	– –	– –	– –	– –
– –	– –	– –	**2000.000** 78.7402	**-0.203** -0.0080	**0.229** 0.009	– –	– –	– –	– –

(1)Tolerance range is from +0 to value listed.

THRUST CYLINDRICAL ROLLER BEARING TOLERANCES

TABLE 5. THRUST CYLINDRICAL ROLLER BEARING TOLERANCES – TYPE TP

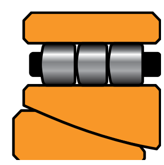
Bore			O.D.			Width		
Bearing Bore Over	Bearing Bore Incl.	Tolerance[1]	Bearing O.D. Over	Bearing O.D. Incl.	Tolerance[1]	Bearing Width Over	Bearing Width Incl.	Tolerance Max.
mm in.	**mm** in.	**mm** in.	**mm** in.	**mm** in.	**mm** in.	**mm** in.	**mm** in.	**mm** in.
50.800 2.0000	**76.200** 3.0000	**-0.025** -0.0010	**127.000** 5.0000	**254.000** 10.0000	**+0.038** +0.0015	**0.000** 0.0000	**50.800** 2.0000	**-0.152** -0.0060
76.200 3.0000	**88.900** 3.5000	**-0.030** -0.0012	**254.000** 10.0000	**457.200** 18.0000	**+0.051** +0.0020	**50.800** 2.0000	**76.200** 3.0000	**-0.203** -0.0080
88.900 3.5000	**228.600** 9.0000	**-0.038** -0.0015	**457.200** 18.0000	**660.400** 26.0000	**+0.640** +0.0025	**76.200** 3.0000	**152.400** 6.0000	**-0.254** -0.0100
228.600 9.0000	**304.800** 12.0000	**-0.046** -0.0018	**660.400** 26.0000	**863.600** 34.0000	**+0.076** +0.0030	**152.400** 6.0000	**254.000** 10.0000	**-0.381** -0.0150
304.800 12.0000	**457.200** 18.0000	**-0.051** -0.0020	**863.600** 34.0000	**1117.600** 44.0000	**+0.102** +0.0040	**254.000** 10.0000	**457.200** 18.0000	**-0.508** -0.0200
457.200 18.0000	**558.800** 22.0000	**-0.064** -0.0025				**457.200** 18.0000	**762.000** 30.0000	**-0.635** -0.0250
558.800 22.0000	**762.000** 30.0000	**-0.076** -0.0030						



[1]The tolerances in this table conform to ABMA Standard 21.2.

TABLE 6. THRUST CYLINDRICAL ROLLER BEARING TOLERANCES – TYPE TPS

Bore			O.D.			Width		
Bearing Bore Over	Bearing Bore Incl.	Tolerance[1]	Bearing O.D. Over	Bearing O.D. Incl.	Tolerance[1]	Bearing Width Over	Bearing Width Incl.	Tolerance Max.
mm in.	**mm** in.	**mm** in.	**mm** in.	**mm** in.	**mm** in.	**mm** in.	**mm** in.	**mm** in.
50.800 2.0000	**76.200** 3.0000	**-0.025** -0.0010	**127.000** 5.0000	**266.700** 10.5000	**+0.048** +0.0019	**0.000** 0.0000	**50.800** 2.0000	**-0.203** -0.0080
76.200 3.0000	**88.900** 3.5000	**-0.030** -0.0012	**266.700** 10.5000	**323.850** 12.7500	**+0.053** +0.0021	**50.800** 2.0000	**76.200** 3.0000	**-0.254** -0.0100
88.900 3.5000	**228.600** 9.0000	**-0.038** -0.0015	**323.850** 12.7500	**431.800** 17.0000	**+0.058** +0.0023	**76.200** 3.0000	**152.400** 6.0000	**-0.381** -0.0150
228.600 9.0000	**304.800** 12.0000	**-0.046** -0.0018	**431.800** 17.0000	**685.800** 27.0000	**+0.064** +0.0025	**152.400** 6.0000	**254.000** 10.0000	**-0.508** -0.0200
304.800 12.0000	**457.200** 18.0000	**-0.051** -0.0020	**685.800** 27.0000	**889.000** 35.0000	**+0.076** +0.0030	**254.000** 10.0000	**457.200** 18.0000	**-0.635** -0.0250
457.200 18.0000	**558.800** 22.0000	**-0.064** -0.0025				**457.200** 18.0000	**762.000** 30.0000	**-0.762** -0.0300
558.800 22.0000	**762.000** 30.0000	**-0.076** -0.0030						

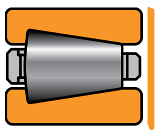


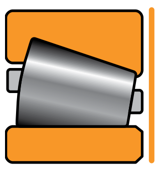
[1]The tolerances in this table conform to ABMA Standard 21.2.

THRUST TAPERED ROLLER BEARING TOLERANCES

INCH BEARINGS

Bore tolerances





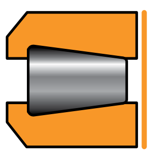


TABLE 7. THRUST TAPERED ROLLER BEARINGS – BORE TOLERANCES

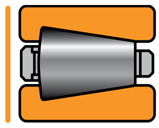
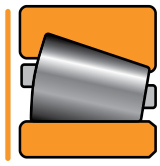
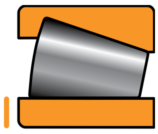
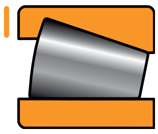
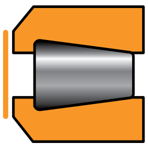
Bearing Types	Bore		Tolerance	
	Over	Incl.	Max.	Min.
	mm in.	**mm** in.	**mm** in.	**mm** in.
TTHD TTHDFL TTHDFLSA TTDW TTDK TTHDSX-1 TTHDSX-2 TTHDSV-1 TTHDSV-2	**0.000** 0.0000	**304.800** 12.0000	**+0.025** +0.0010	**+0.000** +0.0000
	304.800 12.0000	**609.600** 24.0000	**+0.051** +0.0020	**+0.000** +0.0000
	609.600 24.0000	**914.400** 36.0000	**+0.076** +0.0030	**+0.000** +0.0000
	914.400 36.0000	**1219.200** 48.0000	**+0.102** +0.0040	**+0.000** +0.0000
	1219.200 48.0000	–	**+0.127** +0.0050	**+0.000** +0.0000
TTSP TTSPS TTC TTCS TTCL	**0.000** 0.0000	**25.400** 1.0000	**+0.076** +0.0030	**-0.076** -0.0030
	25.400 1.0000	**76.200** 3.0000	**+0.102** +0.0040	**-0.102** -0.0040
	76.200 3.0000	– –	**+0.127** +0.0050	**-0.127** -0.0050

INCH BEARINGS

Outside diameter tolerances

TABLE 8. THRUST TAPERED ROLLER BEARINGS – OUTSIDE DIAMETER TOLERANCES

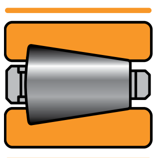
Bearing Types	O.D. Over	O.D. Incl.	Tolerance Max.	Tolerance Min.
	mm in.	mm in.	mm in.	mm in.
TTHD TTHDFL TTHDFLSA TTDW TTDK TTHDSX-1 TTHDSX-2 TTHDSV-1 TTHDSV-2	**0.000** 0.0000	**304.800** 12.0000	**+0.025** +0.0010	**+0.000** +0.0000
	304.800 12.0000	**609.600** 24.0000	**+0.051** +0.0020	**+0.000** +0.0000
	609.600 24.0000	**914.400** 36.0000	**+0.076** +0.0030	**+0.000** +0.0000
	914.400 36.0000	**1219.200** 48.0000	**+0.102** +0.0040	**+0.000** +0.0000
	1219.200 48.0000	–	**+0.127** +0.0050	**+0.000** +0.0000
TTHDFLSX-1 TTHDFLSX-2 TTHDFLSX-3 TTHDFLSV-1 TTHDFLSV-2 TTHDDV	**0.000** 0.0000	**317.500** 12.5000	**+0.000** +0.0000	**-0.025** -0.0010
	317.500 12.5000	647.700 25.5000	**+0.000** +0.0000	**-0.051** -0.0020
TTHDFLSX-1 TTHDFLSX-2 TTHDFLSX-3 TTHDFLSV-1 TTHDFLSV-2 TTHDDV	**0.000** 0.0000	**520.700** 20.5000	**+0.000** +0.0000	**-0.127** -0.0050
	520.700 20.5000	647.700 25.5000	**+0.000** +0.0000	**-0.254** -0.0100
TTSP TTSPS TTC TTCS TTCL	**0.000** 0.0000	**127.000** 5.0000	**+0.254** +0.0100	**0.000** 0.0000
	127.000 5.0000	**203.200** 8.0000	**+0.381** +0.0150	**0.000** 0.0000
	203.200 8.0000	–	**+0.508** +0.2000	**0.000** 0.0000

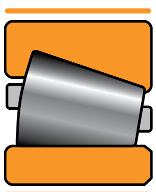






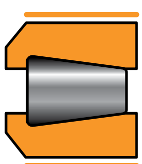
INCH BEARINGS

Width tolerances

TABLE 9. THRUST TAPERED ROLLER BEARING TOLERANCES - WIDTH (INCH)







Bearing Types	Bore		Tolerance	
	Over	Incl.	Max.	Min.
	mm in.	**mm** in.	**mm** in.	**mm** in.
TTHD TTHDFL TTHDFLSA TTHDSX-1 TTHDSX-2 TTHDSV-1 TTHDSV-2 TTHDFLSX-1 TTHDFLSX-2 TTHDFLSX-3 TTHDFLSV-1 TTHDFLSV-2 TTHDDV	All Sizes		**+0.381** +0.015	**-0.381** -0.015
TTDW TTDK	All Sizes		**+0.762** +0.030	**-0.762** -0.030
TTSP TTSPS TTC TTCS TTCL	**0.000** 0.0000	**76.200** 3.0000	**+0.254** +0.0100	**-0.254** -0.0100
	76.200 3.0000	**127.000** 5.0000	**+0.381** +0.0150	**-0.381** -0.0150
	127.000 5.0000	–	**+0.508** +0.2000	**-0.508** -0.2000

METRIC BEARINGS

TABLE 10. THRUST TAPERED ROLLER BEARINGS – BORE TOLERANCES

Bearing Types	Bore		Tolerance	
	Over	Incl.	Max.	Min.
	mm in.	mm in.	mm in.	mm in.
TTDFLK	**80.000** 3.1496	**120.000** 4.7244	**+0.000** +0.0000	**-0.020** -0.0008
	120.000 4.7244	**180.000** 7.0866	**+0.000** +0.0000	**-0.025** -0.0010
	180.000 7.0866	**250.000** 9.8425	**+0.000** +0.0000	**-0.030** -0.0012
	250.000 9.8425	**315.000** 12.4016	**+0.000** +0.0000	**-0.035** -0.0014
	315.000 12.4016	400.000 15.7480	**+0.000** +0.0000	**-0.040** -0.0016
	400.000 15.7480	**500.000** 19.6850	**+0.000** +0.0000	**-0.045** -0.0018
	500.000 19.6850	630.000 24.8031	**+0.000** +0.0000	**-0.050** -0.0020

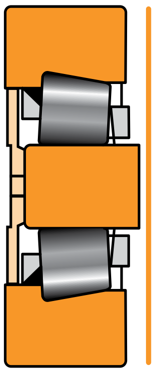
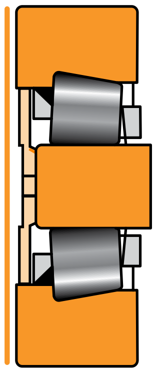


TABLE 11. THRUST TAPERED ROLLER BEARINGS – OUTSIDE DIAMETER TOLERANCES

Bearing Types	O.D.		Tolerance	
	Over	Incl.	Max.	Min.
	mm in.	mm in.	mm in.	mm in.
TTDFLK	**80.000** 3.1496	**120.000** 4.7244	**+0.000** +0.0000	**-0.022** -0.0009
	120.000 4.7244	**180.000** 7.0866	**+0.000** +0.0000	**-0.025** -0.0010
	180.000 7.0866	**250.000** 9.8425	**+0.000** +0.0000	**-0.030** -0.0012
	250.000 9.8425	**315.000** 12.4016	**+0.000** +0.0000	**-0.035** -0.0014
	315.000 12.4016	400.000 15.7480	**+0.000** +0.0000	**-0.040** -0.0016
	400.000 15.7480	**500.000** 19.6850	**+0.000** +0.0000	**-0.045** -0.0018
	500.000 19.6850	**630.000** 24.8031	**+0.000** +0.0000	**-0.050** -0.0020
	630.000 24.8031	**800.000** 31.4961	**+0.000** +0.0000	**-0.075** -0.0030
	800.000 31.4961	**1000.000** 39.3701	**+0.000** +0.0000	**-0.100** -0.0039
	1000.000 39.3701	1250.000 49.2126	**+0.000** +0.0000	**-0.125** -0.0049
	1250.000 49.2126	1600.000 62.9921	**+0.000** +0.0000	**-0.160** -0.0063



Width tolerances

Please contract your Timken engineer for information on the metric thrust bearing width tolerances.

MOUNTING DESIGN, FITTING PRACTICE AND SETTING

To achieve expected bearing performance, it is critical to follow proper mounting design, fitting practices, settings and installation procedures. While there are different practices between thrust tapered roller, cylindrical roller, spherical roller and ball bearings, there are many similarities that apply to all. These similarities are summarized in the sections below, followed by a summary of practices specific to each bearing type.

MOUNTING DESIGN

All bearing types are typically mounted onto a shaft and into a housing where the shaft and housing have surfaces supporting the rings. These surfaces establish the axial location and alignment under all operating conditions. It is essential that a shoulder be square with the bearing ring and of sufficient diameter and axial section to provide adequate backing of the bearing raceway. It also must be of sufficient section to resist axial movement and excessive deflection under loading. Wear resistance at the interface with the bearing rings must be considered.

It is highly recommended that roller bearing shaft seats be ground to a surface finish of 1.6 μm (65 μin) Ra maximum. Ball bearing seats should be 0.8 μm (32 μin) for shafts under 2 inches and 1.6 μm (65 μin) for all other sizes.

When shaft seats are turned, a tighter heavy-duty fit should be selected to ensure interference fit pressure and to prevent rotation. The shaft diameter should be turned to a finish of 3.2 μm (125 μin) Ra maximum.

Housing inside diameters should be finished to 3.2 μm (125 μin) Ra maximum.

ANGULAR CONTACT THRUST BALL BEARINGS – TYPES TVL AND DTVL

The TVL is a separable single-row angular contact ball bearing designed for unidirectional axial loads. The angular contact design, however, will accommodate combined radial and axial loads since the loads are transmitted angularly through the balls. The DTVL is similar in design to TVL except that the DTVL has an additional ring and ball complement permitting it to carry moderate forces in both directions.

Both TVL and DTVL are used extensively in rotary table applications in the oil and gas drilling industry. Rotary table operation generates upward and downward axial loads while being supported and positioned by two main thrust bearings, often of the angular contact thrust ball type. The upper or main position takes the predominant downward axial loads. The lower position, which also is known as the hold down bearing, handles the upward axial load and the majority of the radial loading due to gear forces or dynamic imbalance of the rotating components, fixtures and drill pipe.

An example of arrangements of the angular contact thrust ball bearings includes using one size TVL in the main position, and another size in the lower position, as illustrated in fig. 32. Another popular mounting arrangement is to use a single DTVL as a triple-ring combination bearing to handle thrust loads in both directions at the same time (see fig. 33).

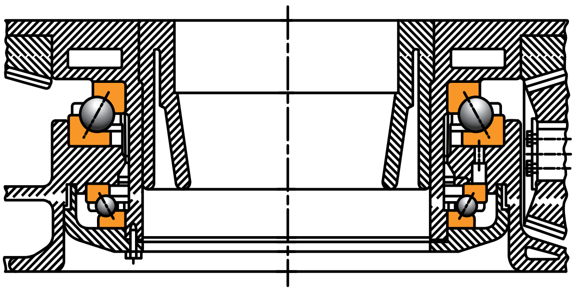


Fig. 32. Large TVL in main position, small TVL in lower position.

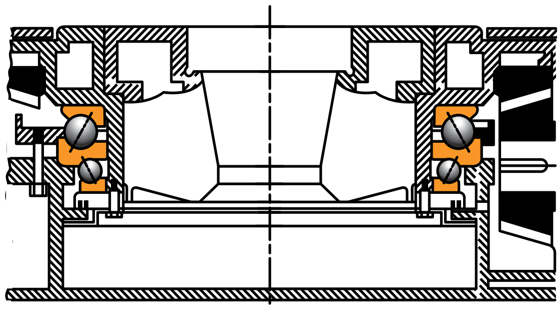


Fig. 33. DTVL mounting to accommodate bi-directional loads.

THRUST CYLINDRICAL ROLLER BEARINGS – TYPES TP AND TPS

Thrust cylindrical roller bearings are generally used in applications where high axial loads are present. Timken TP and TPS thrust cylindrical roller bearings are used in a variety of heavy industrial equipment and challenging thrust applications. Mineral and aggregate crushers and pulverizers are typical examples where thrust cylindrical roller bearings are used in primary thrust support positions to handle the loads applied during the compressive breakdown of aggregate (see fig. 34). Dependent on mounting and axial force applications, these bearings can accommodate moderate overturning moments.

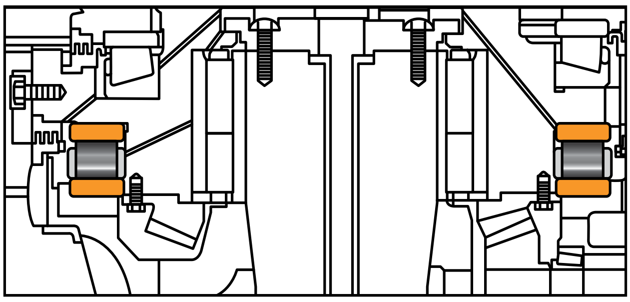


Fig. 34. Typical mounting arrangement of thrust cylindrical roller bearing in crusher application.

Mounting is typically handled by hoisting mechanisms in a shop environment, but may be assembled manually in the field during replacement situations. Mounting of TP and TPS bearings with loose fitting practice on both the shaft and housing is common to allow ease of installation. However, depending on bearing reaction torque, anti-rotation features may be required.

THRUST SPHERICAL ROLLER BEARINGS – TYPE TSR

Thrust spherical roller bearings are used to support axial force in a wide variety of industrial machinery. They can be mounted at axial positions on vertical shafts (e.g. crushers), or mounted horizontally as in long product mill, flat product mill, and cold mill works or intermediate rolls with axial shifting. These assemblies are best suited for applications where accommodation of heavy roll bending and high misalignment is required. Timken thrust spherical roller bearings are capable of handling misalignment between the inner and outer ring of up to 2.5 degrees in either direction.

Bearing outer rings must be mounted with a loose fit to isolate radial loads when used as pure thrust bearings. When used in a shaft position and reacting to radial and axial forces, special housing fitting practice is required. To support axial loads in both directions, thrust spherical roller bearings are often mounted in pairs. In such situations, a spring system maintains the outer races in contact with the rollers on the unloaded row. An axial clearance must be established during mounting using a shim pack between the chock and the cover. Housing components must be designed to accommodate preload springs or precision axial clearance setting.

A cartridge or adapter ring is sometimes used with the inner rings tight fitted on a sleeve and the sleeve loose fitted and keyed on the shaft (see figs. 35 and 36 for typical mountings of EJ and EM styles respectively).

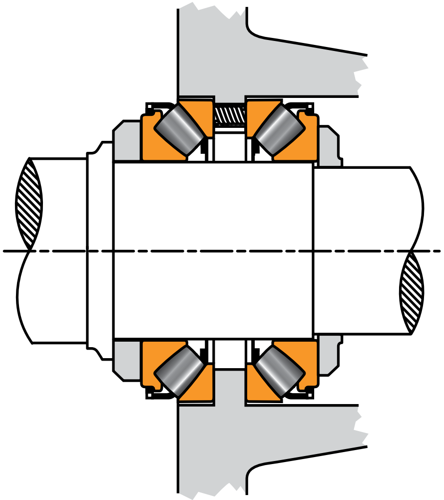


Fig. 35. Back-to-back mounting arrangement of a TSR-EJ bearing set.

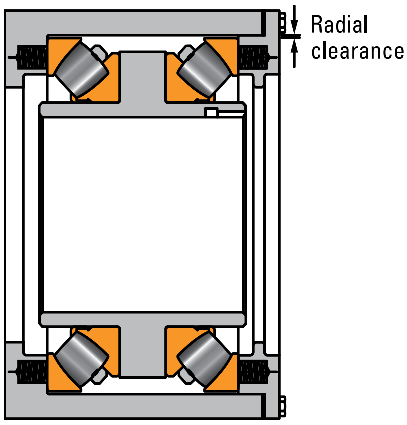


Fig. 36. Face-to-face mounting arrangement of a TSR-EM bearing set.

To maximize axial load support in both directions, thrust spherical roller bearings are often mounted in a tandem face-to-face arrangement (see fig. 36-37). This configuration is common in rollneck applications in the metals industry. In such cases, the inner rings can be clamped in position against each other using inner ring spacers. In applications where surrounding components are mounted in close proximity to the bearing, special care must be taken so that such components do not encroach on the cage or rollers, and so that adequate clearance from the cage and rolling elements is maintained. If there is concern in this regard when mounting thrust spherical roller bearings, contact your Timken representative for support.

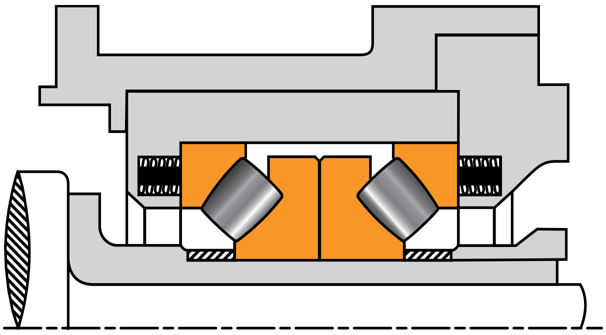


Fig. 37. Typical tandem mount with inner ring spacers.

THRUST TAPERED ROLLER BEARINGS – TYPE TTHD

Thrust tapered roller bearings of type TTHD or TTHDFL are used in a variety of applications such as plastic extruder thrust blocks, oil rig swivels, marine drives and machine tool tables. When mounted, the bearing should be square to the shaft and housing. The backing diameter must be sufficient in the radial direction to support the full length of the rollers, both at the large and small roller ends, and of sufficient axial section to prevent misalignment due to distortion.

In general, the rotating race is mounted with a tight shaft fit, and the stationary race is mounted with a loose housing fit. For TTHDFL, the flat race may be loose fit or tight fit on its outer diameter depending on customer preference.

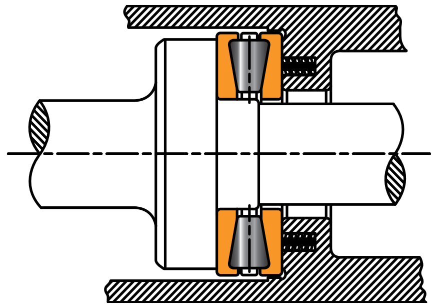


Fig. 38. Typical mounting of a spring-loaded TTHD.

The use of springs is sometimes required on horizontal axis applications where reversing axial loads or shock loads are encountered. Fig. 38 shows a spring-loaded TTHD bearing to keep the housing-supported race in contact with the rolling elements at all times.

THRUST TAPERED ROLLER BEARINGS – SCREWDOWN SYSTEMS – TYPES TTHDSX/SV AND TTHDFLSX/SV

Screwdown bearings of these types are used predominantly in metal mills. The bearings used in screwdown systems include single-row tapered thrust designs that are available in a variety of configurations (see pages 117-126 for further details). The heavy-duty thrust bearing makes the connection between the screwdown and the top roll chock, as shown in fig. 39.

The operating speed of screwdown systems is very low during gap adjustment. Modern mills will either use the electromechanical screwdown system in conjunction with a hydraulic roll force cylinder, or will solely use the hydraulic roll force cylinder. The primary benefit of hydraulic roll force cylinders is their fast response time compared to the electro-mechanical screwdown systems, but the mechanical system gives more precise location with small displacements.

When the mechanical system is used, the screwdown thrust bearing is applied between the main mill screw and top chock. The loads transmitted through these screwdown bearings are extremely high, typically equivalent to half of the mill's separating force, which can be several thousand tons. The operating speed is basically zero as the screw's rotational speed is very slow during adjustment. For this reason, the bearing selection is based on its static capacity (C_0).

Below are a few important considerations to keep in mind when mounting screwdown bearings:

1. **Bearing cartridge:** The bearing is mounted in a cartridge primarily to contain the lubricant needed for the assembly, but also to unitize the entire bearing assembly.
2. **Tapered-bottom race:** If the bottom race is tapered (TTHDSX/SV), then a 3 mm (0.120 in.) radial clearance is suggested relative to the O.D. of the race to ensure that the bottom race will self-align with respect to the upper tapered race. Otherwise, the roller ends will not be properly seated against both the upper and lower large ribs simultaneously. A piloting bushing is pressed into the cartridge and is used for centering the upper race and rollers. The bottom race will be centered by the upper race and roller set.

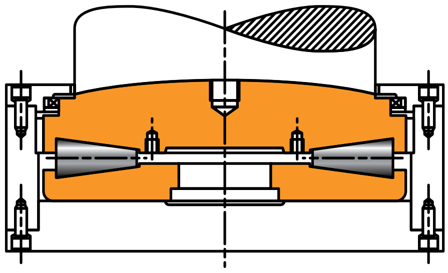


Fig. 39. Typical screwdown support configuration using a TTHDSX thrust bearing.

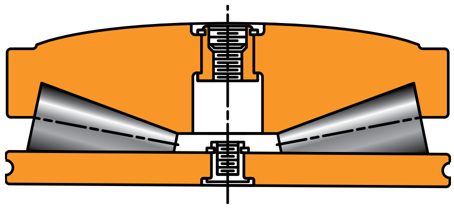


Fig. 40. TTHDFLSX convex upper race design.

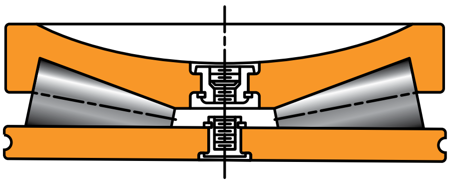


Fig. 41. TTHDFLSV concave upper race design.

3. **Flat-bottom race**: If the bottom race is flat (TTHDFLSX/SV), as in figs. 40-41, then apply close fit as per fitting practice guidelines. The flat race permits radial self-aligning of the rollers and conical raceways.
4. **Sealing:** An oil seal is mounted in the upper plate that is bolted to the cartridge to keep contaminants from entering the bearing assembly.
5. **Lubrication:** Adequate lubrication is maintained by filling the bearing with high-quality EP grease having a viscosity of approximately 450 cSt at 40º C (104º F).

DOUBLE-ACTING HEAVY DUTY THRUST TAPERED ROLLER BEARINGS – TYPES TTDWK AND TTDFLK

The TTDWK or TTDFLK double-acting thrust tapered roller bearing is an excellent choice where extremely high axial loads are anticipated.

Double-acting thrust tapered roller bearings are commonly used in strip mills that generate particularly large thrust forces, as is the case in cross rolling systems.

The TTDWK (fig. 42) bearing includes two flat raceways – one on each side and one tapered double-race thrust ring at the center of the bearing, as well as two sets of rollers that are retained as a unit in a pinned cage.

The TTDFLK (fig. 43), a variant to this TTDWK configuration, uses two tapered raceways (one on each side) and a flat, double-race thrust ring at its center.

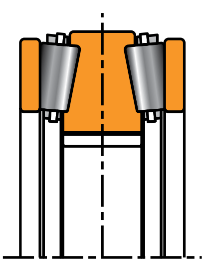


Fig. 42. Typical TTDWK assembly (with flat outer raceways).

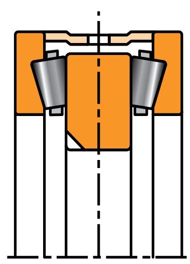


Fig. 43. Typical TTDFLK assembly (with flat double inner raceways).

The TTDWK double-acting thrust bearing is usually mounted in combination with a radial bearing at the fixed position (fig. 44). Such an assembly is fitted in a separate housing that will be mounted on the chock. The outer races are not axially clamped, but adjusted to obtain the required axial clearance, allowing the springs to develop the correct axial force to seat the unloaded row. A keyway is generally provided in the center double-race ring to stop it from rotating on the roll neck.

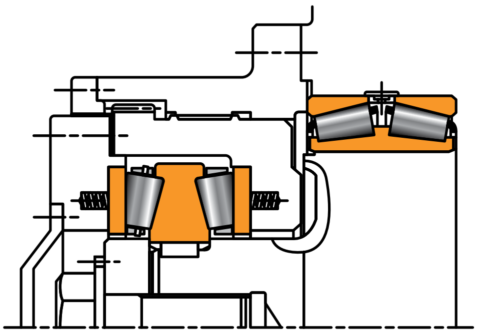


Fig. 44. Typical TTDWK thrust bearing arrangement.

The TTDFLK bearing, on the other hand, is preset and does not require adjustment during mounting. If the bearing is supplied without a spacer, then the same spring arrangement and adjustment as the TTDWK must be used.

The assembly must be axially clamped using metal shims or a compressible gasket, as shown in fig. 45. This bearing can also be ordered without the spacer and then mount it like the TTDWK (fig. 46).

These double-acting bearings (TTDWK and TTDFLK) can only be installed as a unit. Take care to ensure that the flat races are correctly centered when lifting or lowering this bearing into the housing.

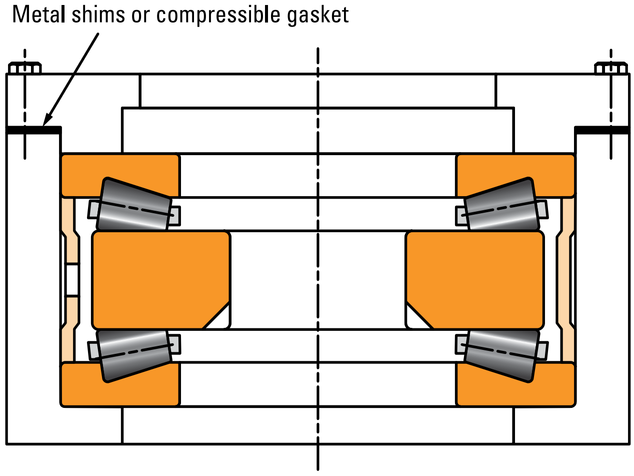


Fig. 45. TTDFLK thrust bearing mounted in housing.

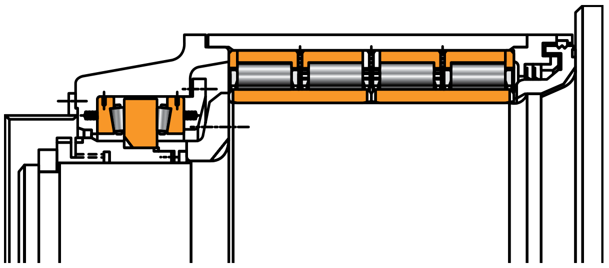


Fig. 46. TTDFLK thrust assembly typical mounting.

CROSSED ROLLER BEARINGS

TXR (DO)

A typical mounting arrangement for the type TXRDO crossed roller bearing is shown in fig. 47.

The arrangement shown is for lubrication by oil circulation in conjunction with an oil level maintained within the bearing. 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. 48) should be machined to give a mean of the suggested interference fits (pages 48-49).

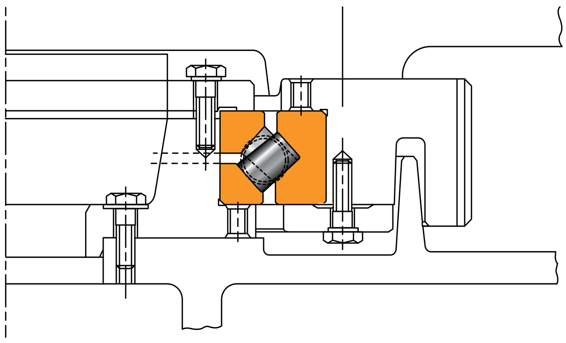


Fig. 47. Typical mounting arrangement of a TXRDO bearing.

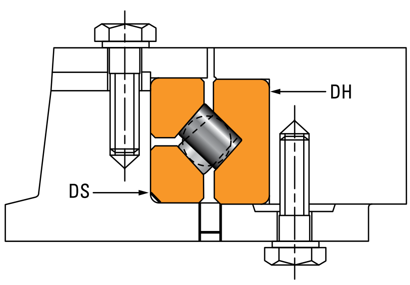


Fig. 48. Fitting and setting of TXR bearing.

The bearing is adjusted externally by segments beneath the top inner-ring clamping plate to get the required preload.

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

AUXILIARY EQUIPMENT AND OTHER BEARING TYPE MOUNTING PROCEDURES

For industry-standard bearing types, please refer to the following Timken catalogs for mounting procedures – Timken® Tapered Roller Bearing Catalog (order no. 10481), Timken® Cylindrical Roller Bearing Catalog (order no. 10447), Timken® Spherical Roller Bearing Catalog (order no. 10446) and the Timken® Engineering Manual (order no. 10424).

FITTING PRACTICE

As a general guideline, bearing rings mounted on a rotating member should have an interference fit. For some thrust bearing applications, the ring is pinned to the rotating shaft. Loose fits may permit the ring to creep or turn and wear the mating surface and backing shoulder. This wear can result in excessive bearing looseness which can lead to damage of the bearing, shaft or housing. Many thrust bearing applications have outer rings mounted with a clearance to insulate them from radial loads and to allow axial float.

The choice of fitting practices will mainly depend upon the following parameters:

- Precision class of the bearing.
- Rotating or stationary ring.
- Type of layout (single- or double-row bearings).
- Type and direction of load (continuous/alternate rotating, overturning moments).
- Particular running conditions like shocks, vibrations, overloading or high speed.
- Capability for machining the seats (grinding, turning or boring).
- Shaft and housing section and material.
- Mounting and setting conditions.

General fitting practice guidelines for thrust bearings having a bore less than 304.8 mm (12 in.) are:

Rotating race

- Use a tight fit with horizontal shafts; vertical shafts may consider split or loose fit.
- Use a clearance with housing.

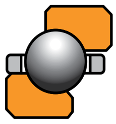
Stationary race

- Use a loose fit on shaft and clearance with housing.

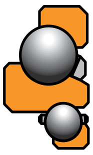
For bore sizes greater than 304.8 mm (12 in.), contact your Timken engineer. Detailed fitting practices for various thrust bearing types are listed in the following tables 12-26.

ANGULAR CONTACT THRUST BALL BEARING FITS

Shaft and housing diameters are shown as variance from nominal dimensions.



TVL



DTVL

TABLE 12. SHAFT FITS – ANGULAR CONTACT THRUST BALL BEARINGS – TYPE TVL AND DTVL

Bearing Bore Nominal		Shaft Diameter			
		Interference Fit[1]		Loose Fit[2]	
Over	Incl.	Max.	Min.	Max.	Min.
mm in.	mm in.	mm in.	mm in.	mm in.	mm in.
0.000 0.0000	**504.825** 19.8750	**+0.076** +0.0030	**+0.000** +0.0000	**-0.152** -0.0060	**-0.076** -0.0030
504.825 19.8750	**1524.000** 60.0000	**+0.127** +0.0050	**+0.000** +0.0000	**-0.254** -0.0100	**-0.127** -0.0050

[1]Dowel pin suggested.
[2]Dowel pin required.

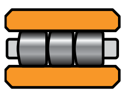
TABLE 13. HOUSING FITS – ANGULAR CONTACT THRUST BALL BEARINGS – TYPE TVL AND DTVL

Bearing O.D. Nominal		Housing Diameter			
		Interference Fit[1]		Loose Fit[2]	
Over	Incl.	Max.	Min.	Max.	Min.
mm in.	mm in.	mm in.	mm in.	mm in.	mm in.
0.000 0.0000	**584.000** 23.0000	**-0.152** -0.0060	**-0.076** -0.0030	**+0.152** +0.0060	**0.076** 0.0030
584.000 23.0000	**1778.000** 70.0000	**-0.254** -0.0100	**-0.127** -0.0050	**+0.254** +0.0100	**0.127** 0.0050

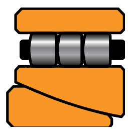
[1]Dowel pin suggested.
[2]Dowel pin required.

THRUST CYLINDRICAL ROLLER BEARING FITS

Tolerances for housing bore and for shaft diameters shown as variance from nominal bearing dimension.



TP



TPS

TABLE 14. SHAFT FITS – THRUST CYLINDRICAL ROLLER BEARING – TYPE TP AND TPS

Bearing Bore Nominal		Shaft Diameter	
Over	Incl.	Max.	Min.
mm in.	mm in.	mm in.	mm in.
47.625 1.8750	**53.975** 2.1250	**-0.025** -0.0010	**-0.051** -0.0020
53.975 2.1250	**63.500** 2.5000	**-0.028** -0.0011	**-0.053** -0.0021
63.500 2.5000	**76.200** 3.0000	**-0.030** -0.0012	**-0.056** -0.0022
76.200 3.0000	**88.900** 3.5000	**-0.033** -0.0012	**-0.058** -0.0023
88.900 3.5000	**177.800** 7.0000	**-0.038** -0.0015	**-0.064** -0.0025
177.800 7.0000	**228.600** 9.0000	**-0.038** -0.0015	**-0.076** -0.0030
228.600 9.0000	**304.800** 12.0000	**-0.046** -0.0018	**-0.084** -0.0330
304.800 12.0000	**381.000** 15.0000	**-0.051** -0.0020	**-0.089** -0.0035
381.000 15.0000	**482.600** 19.0000	**-0.051** -0.0020	**-0.102** -0.0040
482.600 19.0000	**584.200** 23.0000	**-0.064** -0.0025	**-0.114** -0.0045
584.200 23.0000	**762.000** 30.0000	**-0.076** -0.0030	**-0.140** -0.0055

TABLE 15. HOUSING FITS – THRUST CYLINDRICAL ROLLER BEARING – TYPE TP

Bearing O.D. Nominal		Housing Diameter Deviation from D	
Over	Incl.	Max.	Min.
mm in.	mm in.	mm in.	mm in.
115.092 4.5312	**254.000** 10.0000	**+0.076** +0.0030	**+0.038** +0.0015
254.000 10.0000	**457.200** 18.0000	**+0.102** +0.0040	**+0.051** +0.002
457.200 18.0000	**558.800** 22.0000	**+0.127** +0.0050	**+0.064** +0.0025
558.800 22.0000	**660.400** 26.0000	**+0.140** +0.0055	**+0.064** +0.0025
660.400 26.0000	**711.200** 28.0000	**+0.152** +0.0060	**+0.076** +0.0030
711.200 28.0000	**863.600** 34.0000	**+0.178** +0.0070	**+0.076** +0.0030
863.600 34.0000	**965.200** 38.0000	**+0.203** +0.0080	**+0.089** +0.0035
965.200 38.0000	**1117.600** 44.0000	**+0.229** +0.0090	**+0.102** +0.0040

TABLE 16. HOUSING FITS – THRUST CYLINDRICAL ROLLER BEARING – TYPE TPS

Bearing O.D. Nominal		Housing Diameter Deviation from D	
Over	Incl.	Max.	Min.
mm in.	mm in.	mm in.	mm in.
50.800 2.0000	**60.325** 2.3750	**+0.038** +0.0015	**+0.013** +0.0005
60.325 2.3750	**82.550** 3.2500	**+0.043** +0.0017	**+0.018** +0.0007
82.550 3.2500	**93.663** 3.6875	**+0.048** +0.0019	**+0.023** +0.0009
93.663 3.6875	**101.600** 4.0000	**+0.053** +0.0021	**+0.028** +0.0011
101.600 4.0000	**115.092** 4.5312	**+0.071** +0.0028	**+0.033** +0.0013
115.092 4.5312	**254.000** 10.0000	**+0.076** +0.0030	**+0.038** +0.0015
254.000 10.0000	**457.200** 18.0000	**+0.102** +0.0040	**+0.051** +0.0020
457.200 18.0000	**558.800** 22.0000	**+0.127** +0.0050	**+0.064** +0.0025
558.800 22.0000	**660.400** 26.0000	**+0.140** +0.0055	**+0.064** +0.0025
660.400 26.0000	**711.200** 28.0000	**+0.152** +0.0060	**+0.076** +0.0030
711.200 28.0000	**863.600** 34.0000	**+0.178** +0.0070	**+0.076** +0.0030
863.600 34.0000	**965.200** 38.0000	**+0.203** +0.0080	**+0.089** +0.0035
965.200 38.0000	**1117.600** 44.0000	**+0.229** +0.0090	**+0.102** +0.0040

THRUST SPHERICAL ROLLER BEARING FITS

Tolerances for housing bore and for shaft diameters are shown as variance from nominal bearing dimension.

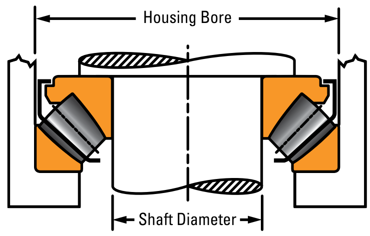


TABLE 17. SHAFT FITS – THRUST SPHERICAL ROLLER BEARINGSS

Bearing Bore Nominal		Shaft Diameter			
		Stationary Load		Rotation Load	
Over	Incl.	Max.	Min.	Max.	Min.
mm in.	mm in.	mm in.	mm in.	mm in.	mm in.
80.000 3.1496	**120.000** 4.7244	**+0.013** +0.0005	**-0.010** -0.0004	**+0.025** +0.0010	**+0.003** +0.0001
120.000 4.7244	**180.000** 7.0866	**+0.015** +0.0006	**-0.010** -0.0004	**+0.028** +0.0011	**+0.003** +0.0001
180.000 7.0866	**200.000** 7.8740	**+0.018** +0.0007	**-0.013** -0.0005	**+0.036** +0.0014	**+0.005** +0.0002
200.000 7.8740	**240.000** 9.4488	**+0.018** +0.0007	**-0.013** -0.0005	**+0.046** +0.0018	**+0.015** +0.0006
240.000 9.4488	**315.000** 12.4016	**+0.018** +0.0007	**-0.015** -0.0006	**+0.051** +0.0020	**+0.020** +0.0008
315.000 12.4016	**400.000** 15.7480	**+0.018** +0.0007	**-0.018** -0.0007	**+0.056** +0.0022	**+0.020** +0.0008
400.000 15.7480	**500.000** 19.6850	**+0.023** +0.0009	**-0.018** -0.0007	**+0.086** +0.0034	**+0.046** +0.0018
500.000 19.6850	**630.000** 24.8031	**+0.023** +0.0009	**-0.020** -0.0008	**+0.086** +0.0034	**+0.043** +0.0017

TABLE 18. HOUSING FITS – THRUST SPHERICAL ROLLER BEARINGS

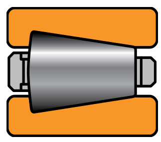
Bearing O.D. Nominal		Housing Bore					
		Springs in Housing Light Radial Load		Combined Axial & Radial Load			
				Stationary Outer Ring		Rotating Outer Ring	
Over	Incl.	Max.	Min.	Max.	Min.	Max.	Min.
mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.
180.000 7.0866	**250.000** 9.8425	**+0.061** +0.0024	**+0.015** +0.0006	**+0.028** +0.0011	**-0.018** -0.0007	**+0.013** +0.0005	**-0.033** -0.0013
250.000 9.8425	**315.000** 12.4016	**+0.069** +0.0027	**+0.018** +0.0007	**+0.033** +0.0013	**-0.018** -0.0007	**+0.015** +0.0006	**-0.036** -0.0014
315.000 12.4016	**400.000** 15.7480	**+0.074** +0.0029	**+0.018** +0.0007	**+0.038** +0.0015	**-0.018** -0.0007	**+0.015** +0.0006	**-0.041** -0.0016
400.000 15.7480	**500.000** 19.6850	**+0.084** +0.0033	**+0.020** +0.0008	**+0.041** +0.0016	**-0.023** -0.0009	**+0.018** +0.0007	**-0.046** -0.0018
500.000 19.6850	**630.000** 24.8031	**+0.091** +0.0036	**+0.023** +0.0009	**+0.046** +0.018	**-0.023** -0.0009	**+0.020** +0.0008	**-0.048** -0.0019
630.000 24.8031	**800.000** 31.4960	**+0.102** +0.0040	**+0.023** +0.0009	**+0.051** +0.0020	**-0.023** -0.0009	**+0.023** +0.0009	**-0.051** -0.0020
800.000 31.4960	**1000.000** 39.3700	**+0.109** +0.0043	**+0.025** +0.0010	**+0.058** +0.0023	**-0.025** -0.0010	**+0.025** +0.0010	**-0.058** -0.0023
1000.000 39.3700	**1250.000** 49.2126	**+0.122** +0.0048	**+0.028** +0.0011	**+0.066** +0.0026	**-0.028** -0.0011	**+0.030** +0.0012	**-0.064** -0.0025

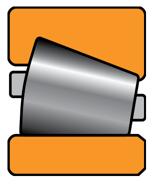
NOTE

When application calls for thrust loads only, the housing must be relieved by 1.588 mm (0.0625 in.) on diameter so that no radial load is carried on the bearing.

THRUST TAPERED ROLLER BEARING FITS

Tolerances for housing bore and shaft diameters are shown as variance from nominal bearing dimension.


TTHD


TTHDFL

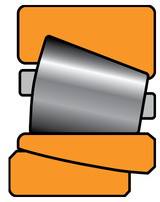

TTHDFLSA

TABLE 19. FITTING GUIDELINES – THRUST TAPERED ROLLER BEARINGS – TYPE TTHD

Bore		Rotating Ring						Stationary Ring	
		Class 2			Class 3			Class 2 and 3	
Over	Incl.	Tolerance	Shaft O.D. Deviation	Resultant Fit	Tolerance	Shaft O.D. Deviation	Resultant Fit		
mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.	mm in.		
0.000 0.0000	**304.800** 12.0000	**0.000** **+0.025** 0.0000 +0.0010	**+0.076** **+0.050** +0.0030 +0.0020	**0.076T** **0.025T** 0.0030T 0.0010T	**0.000** **+0.013** 0.0000 +0.0005	**+0.051** **+0.038** +0.0020 +0.0015	**0.051T** **0.025T** 0.0020T 0.0010T	All sizes	Provide a minimum radial clearance of 2.5 mm (0.1 in.) between ring bore and shaft O.D.
304.800 12.0000	**609.600** 24.0000	**0.000** **+0.051** 0.0000 +0.0020	**+0.152** **+0.102** +0.0060 +0.0040	**0.152T** **0.051T** 0.0060T 0.0020T	**0.000** **+0.025** 0.0000 +0.0010	**+0.102** **+0.076** +0.0040 +0.0030	**0.102T** **0.051T** 0.0040T 0.0020T		
609.600 24.0000	**914.400** 36.0000	**0.000** **+0.076** 0.0000 +0.0030	**+0.204** **+0.127** +0.0080 +0.0050	**0.204T** **0.051T** 0.0080T 0.0020T	**0.000** **+0.038** 0.0000 +0.0015	**+0.127** **+0.089** +0.0050 +0.0035	**0.127T** **0.051T** 0.0050T 0.0020T		
914.400 36.0000	**1219.200** 48.0000	**0.000** **+0.102** 0.0000 +0.0040	**+0.254** **+0.153** +0.0100 +0.0060	**0.254T** **0.051T** 0.0100T 0.0020T	**0.000** **+0.051** 0.0000 +0.0020	**+0.153** **+0.102** +0.0060 +0.0040	**0.153T** **0.051T** 0.0060T 0.0020T		
1219.200 48.0000		**0.000** **+0.127** 0.0000 +0.0050	**+0.305** **+0.178** +0.0120 +0.0070	**0.305T** **0.051T** 0.0120T 0.0020T	**0.000** **+0.076** 0.0000 +0.0030	**+0.204** **+0.127** +0.0080 +0.0050	**0.204T** **0.051T** 0.0080T 0.0020T		

- Rotating ring O.D. must have a minimum radial clearance of 2.5 mm (0.1 in.).
- TTHD stationary ring O.D. must have a minimum loose fit of 0.25 to 0.37 mm (0.01 to 0.015 in.).
- TTHDFL ring when stationary may be loose fit on its O.D. (same as the TTHD) or may be 0.025 to 0.076 mm (0.001 to 0.003 in.) tight.

TABLE 20. SHAFT FITS – THRUST TAPERED ROLLER BEARINGS TYPE TTHDFL AND TTHDFLSA

Bearing Bore Nominal		Shaft Diameter
Over	Incl.	Min.(1)
mm in.	mm in.	mm in.
0.000 0.0000	**304.800** 12.0000	**-0.051** -0.0020
304.800 12.0000	**508.000** 20.0000	**-0.051** -0.0020
508.000 20.0000	**711.200** 28.0000	**-0.076** -0.0030
711.200 28.0000	**1219.200** 48.0000	**-0.102** -0.0040
1219.200 48.0000	**1727.200** 68.0000	**-0.127** -0.0050

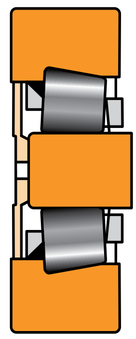
(1)Tolerance range is from +0 to value listed.

TABLE 21. HOUSING FITS – THRUST TAPERED ROLLER BEARINGS TYPE TTHDFL AND TTHDFLSA

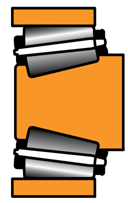
Bearing O.D. Nominal		Housing Bore	
Over	Incl.	Max.	Min.
mm in.	mm in.	mm in.	mm in.
161.925 6.3750	**265.113** 10.4375	**+0.060** +0.0025	**+0.025** +0.0010
265.113 10.3475	**317.500** 12.5000	**+0.076** +0.0030	**+0.025** +0.0010
317.500 12.5000	**482.600** 19.0000	**+0.102** +0.0040	**+0.051** +0.0020
482.600 19.0000	**603.250** 23.7500	**+0.113** +0.0045	**+0.051** +0.0020
603.250 23.7500	**711.200** 28.0000	**+0.152** +0.0060	**+0.076** +0.0030
711.200 28.0000	**838.200** 33.0000	**+0.178** +0.0070	**+0.076** +0.0030

TABLE 22. SHAFT FITS – THRUST TAPERED ROLLER BEARINGS – TYPE TTD, TTDW, TTDWK, TTDF, TTDFLK

Bore Range Over	 Incl.	Bore Tolerance	Inner Race Seat Deviation	Resultant Fit
mm in.	**mm** in.	**mm** in.	**mm** in.	**mm** in.
0.000 0.0000	**76.200** 3.0000	**0.000** **+13** +0.0000 +0.0005	**-51** **-76** -0.0020 -0.0030	**51L** **89L** 0.0020L 0.0035L
76.200 3.0000	**101.600** 4.0000	**0.000** **+25** 0.0000 +0.0010	**-76** **-102** -0.0030 -0.0040	**76L** **127L** 0.0030L 0.0050L
101.600 4.0000	**127.000** 5.0000	**0.000** **+25** 0.0000 +0.0010	**-102** **-127** -0.0040 -0.0050	**102L** **152L** 0.0040L 0.0060L
127.000 5.0000	**152.400** 6.0000	**0.000** **+25** 0.0000 +0.0010	**-127** **-152** -0.0050 -0.0060	**127L** **177L** 0.0050L 0.0070L
152.400 6.0000	**203.200** 8.0000	**0.000** **+25** 0.0000 +0.0010	**-152** **-178** -0.0060 -0.0070	**152L** **203L** 0.0060L 0.0080L
203.200 8.0000	**304.800** 12.0000	**0.000** **+25** 0.0000 +0.0010	**-178** **-203** -0.0070 -0.0080	**178L** **228L** 0.0070L 0.0090L
304.800 12.0000	**609.600** 24.0000	**0.000** **+51** 0.0000 +0.0020	**-203** **-254** -0.0080 -0.0100	**203L** **305L** 0.0080L 0.0120L
609.600 24.0000	**914.400** 36.0000	**0.000** **+76** 0.0000 +0.0030	**-254** **-330** -0.0100 -0.0130	**254L** **406L** 0.0100L 0.0160L
914.400 36.0000	**1219.200** 48.0000	**0.000** **+102** 0.0000 +0.0040	**-305** **-406** -0.0120 -0.0160	**305L** **508L** 0.0120L 0.0200L
1219.200 48.0000		**0.000** **+127** 0.0000 +0.0050	**-305** **-432** -0.0120 -0.0170	**305L** **559L** 0.0120L 0.0220L



TTDFLK



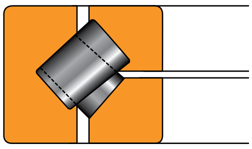
TTDWK

NOTE

When one ring is piloted by the housing, sufficient clearances must be allowed at the outside diameter of the other ring as well as at the bore of both rings to prevent cross-loading of the rollers. For most applications, this clearance is approximately 1.588 mm (0.0625 in.).

PRECISION CLASS TXR TAPERED ROLLER BEARING FITS

Tolerances for housing bore and shaft diameters are shown as variance from nominal bearing dimension.



TXR

TABLE 23. PRECISION CLASS TXR TAPERED ROLLER BEARINGS – SHAFT DIAMETER TXR CLASSES S AND P (METRIC)

Bearing Bore Range Over	Bearing Bore Range Incl.	Class S Max.	Class S Min.	Class P Max.	Class P 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

TABLE 24. PRECISION CLASS TXR TAPERED ROLLER BEARINGS – HOUSING BORE – TXR CLASSES S AND P (METRIC)

Bearing Bore Range Over	Bearing Bore Range Incl.	Class S Max.	Class S Min.	Class P Max.	Class P 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

TABLE 25. PRECISION CLASS TXR TAPERED ROLLER BEARINGS - SHAFT DIAMETER TXR CLASSES 3 AND 0 (INCH)

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	– –	– –

TABLE 26. PRECISION CLASS TXR TAPERED ROLLER BEARINGS - HOUSING BORE TXR CLASSES 3 AND 0 (INCH)

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	– –	– –

SETTING

Thrust bearings are typically set up against another bearing with the setting determined by the application requirements. Most thrust bearings operate under a preload condition.

Correct bearing mounting and fitting practices are key components of proper bearing setting.

INSTALLATION

Proper bearing installation, including cleanliness of the components, as well as use of proper tools, is critical to bearing performance.

Cleanliness of the bearing and mating components is essential for a bearing to achieve maximum service life. Burrs, foreign material and any raised portions of the components mating with the bearing can cause misalignment. Care should be taken to avoid these conditions. Shafts and housings, including lubrication holes, should be thoroughly cleaned before bearing installation. If blind holes are present, insert a magnetic rod to remove metal chips that might have accumulated during manufacture. An air hose may be used on shafts and housings, but should not be used on bearings. Bearings in their shipping containers are typically coated with a rust-inhibitive oil. This oil is compatible with most lubricants and does not need to be removed prior to installation.

WARNING

Failure to observe the following warnings could create a risk of death or serious injury.

Never spin a bearing with compressed air. The components may be forcefully expelled.

Proper maintenance and handling practices are critical. Always follow installation instructions and maintain proper lubrication.

Adequate tools must be used to properly fit the inner rings onto the shaft and outer rings into the housing to avoid damage. Direct impact on the rings must be avoided. Inspection of fillets and undercuts should be completed prior to assembly to ensure proper clearance with the bearing.

If applications require a tight interference fit of one or both rings, it is acceptable to heat or cool rings to ease assembly. Standard bearings should not be heated above 120° C (250° F) or cooled below -55° C (-65° F). Precision bearings should not be heated above 65° C (150° F) or cooled below -30° C (-20° F). An alternate method of mounting, generally used on smaller sizes, is to press the bearing onto the shaft or into the housing using an arbor press.

For more information on these installation procedures, please contact your Timken engineer.

WARNING

Failure to observe the following warnings could create a risk of death or serious injury.

Proper maintenance and handling practices are critical. Always follow installation instructions and maintain proper lubrication.

Overheated bearings can ignite explosive atmospheres. Special care must be taken to properly select, install, maintain, and lubricate housed unit bearings that are used in or near atmospheres that may contain explosive levels of combustible gases or accumulations of dust such as from grain, coal, or other combustible materials.

Consult your equipment designer or supplier for installation and maintenance instructions.

NOTE:

The products cataloged are application specific. Any use in applications other than those intended could lead to equipment failure or to reduced equipment life.

Use of improper bearing fits may cause damage to equipment.

Do not use damaged bearings. The use of a damaged bearing can result in equipment damage.

BEARING OPERATION
OPERATING TEMPERATURES

Bearings operate in a wide range of applications and environments. In most cases, bearing operating temperature is not an issue. Some applications, however, operate at extreme speeds or in extreme temperature environments. In these cases, care must be taken not to exceed the temperature limits of the bearing. Minimum temperature limits are primarily based on lubricant capability. Maximum temperature limits are most often based on material and/or lubricant constraints, but also may be based on accuracy requirements of the equipment that the bearings are built into. These constraints/limitations are discussed below.

BEARING MATERIAL LIMITATIONS

Standard bearing steels with a standard heat treatment cannot maintain a minimum hardness of 58 HRC much above 120° C (250° F).

Dimensional stability of Timken bearings is managed through the proper selection of an appropriate heat-treat process. Standard Timken ball bearings are dimensionally stabilized from -54° C (-65° F) up to 120° C (250° F). Upon request, these bearings can be ordered to higher levels of stability as listed below. These designations are in agreement with DIN Standard 623.

TABLE 27.

Stability Designation	Maximum Operating Temperature	
	°C	°F
S0	**150**	302
S1	**200**	392
S2	**250**	482
S3	**300**	572
S4	**350**	662

With dimensionally stabilized product, there still may be some changes in dimensions during service as a result of microstructural transformations. These transformations include the continued tempering of martensite and decomposition of retained austenite. The magnitude of change depends on the operating temperature, the time at temperature and the composition and heat-treatment of the steel.

Temperatures exceeding the limits shown in table 27 require special high-temperature steel. Consult your Timken engineer for availability of specific part numbers for non-standard heat stability or high-temperature steel grades.

Suggested materials for use in balls, rings and rollers at various operating temperatures are listed in table 28. Also listed are chemical composition suggestions, hardness suggestions and dimensional stability information.

Operating temperature affects lubricant film thickness and setting, both of which directly influence bearing life. Extremely high temperatures can result in a reduced film thickness that can lead to asperity contact between contacting surfaces.

Operating temperature also can affect performance of cages, seals and shields, which in turn can affect bearing performance. Materials for these components and their operating temperature ranges are shown in table 28.

LUBRICATION LIMITATIONS

Starting torque in grease-lubricated applications typically increases significantly at cold temperatures. Starting torque is not primarily a function of the consistency or channel properties of the grease. Most often, it is a function of the rheological properties of the grease.

The high-temperature limit for greases is generally a function of the thermal and oxidation stability of the base oil in the grease and the effectiveness of the oxidation inhibitors.

See the LUBRICATION section on page 55 for more information on lubrication limitations.

EQUIPMENT REQUIREMENTS

The equipment designer must evaluate the effects of temperature on the performance of the equipment being designed. Precision machine tool spindles, for example, can be very sensitive to thermal expansions. For some spindles, it is important that the temperature rise over ambient be held to 20° C to 35° C (36° F to 45° F).

Most industrial equipment can operate at considerably higher temperatures. Thermal ratings on gear drives, for example, are based on 93° C (200° F). Equipment such as gas turbines operates continuously at temperatures above 100° C (212° F). Running at high temperatures for extended periods of time, however, may affect shaft and housing fits if the shaft and housing are not machined and heat-treated properly.

ENGINEERING

BEARING OPERATION

Although bearings can operate satisfactorily up to 120° C (250° F), an upper temperature limit of 80° C to 95° C (176° F to 203° F) is more practical. Higher operating temperatures increase the risk of damage from transient temperature spikes. Prototype testing of the application can help define the operating temperature range and should be conducted if possible. It is the responsibility of the equipment designer to weigh all relevant factors and make the final determination of satisfactory operating temperature.

Table 28 provides standard operating temperatures for common bearing component materials. They should be used for reference purposes only. Other bearing component materials are available on request. Contact your Timken engineer for more information.

TABLE 28. OPERATING TEMPERATURES FOR BEARING COMPONENT MATERIALS

Material	Approximate Chemical Analysis %	Temp. °F	Hard-ness HRC	-73° C -100° F	-54° C -65° F	-17° C 0° F	38° C 100° F	93° C 200° F	121° C 250° F	149° C 300° F	204° C 400° F	260° C 500° F	316° C 600° F	371° C 700° F	427° C 800° F
Low-alloy carbon-chromium bearing steels. 52100 and others per ASTM A295	1C 0.5–1.5Cr 0.35Mn	70	60		STANDARD DIMENSIONAL STABILIZATION <0.0001 in./in dimensional change in 2500 hours at 100° C (212° F). Good oxidation resistance.										
Low-alloy carbon-chromium bearing steels. 52100 and others per ASTM A295	1C 0.5–1.5Cr 0.35Mn	70 350 450	58 56 54	Heat stabilized <0.0001in./in dimensional change in 2500 hours at 149° C (300° F). When given a stabilizing heat treatment, A295 steel is suitable for many applications in the 177°-232° C (350-450° F) range; however, it is not as dimensionally stable as it is at temperatures below 177° C (350° F). If utmost stability is required, use materials in the 316° C (600° F) group below.											
Deep-hardening steels for heavy sec-tions per ASTM A485	1C 1–1.8Cr 1–1.5Mn .06Si	70 450 600	58 55 52	As heat-treated and tempered, it is stabilized, <0.0001 in./in dimensional change in 2500 hours at 149° C (300° F).											
Carburizing steels per ASTM A534 a) low alloy 4118, 8X19, 5019, 8620 (Ni-Moly grades) b) high nickel 3310	Ni-Moly: 0.2C, 0.4-2.0Mn, 0.3-0.8Cr, 0-2.0Ni, 0-0.3Mo .0.1C, 1.5Cr, 0.4Mn, 3.5Ni	70	58		Nickel-Moly grades of steel frequently used to achieve extra ductility in inner rings for locking device bearings. 3311 and others used for extra-thick-section rings.										
Corrosion-resistant 440C stainless steel per ASTM A756	1C 18Cr	70	58		Excellent corrosion resistance.										
Corrosion-resistant 440C stainless steel per ASTM A756	1C 18Cr	70 450 600	58 55 52	As heat stabilized for maximum hardness at high temperatures. Good oxidation resistance at higher temperatures. Note load capacity drops off more rapidly at higher temperatures than M50 shown below, which should be considered if loads are high, <0.0001 in./in dimensional change in 1200 hours.											
M-50 medium high speed	4Cr 4Mo 1V 0.8C	70 450 600	60 59 57	Suggested where stable high hardness at elevated temperature is required, <0.0001 in./in dimensional change in 1200 hours at 316° C (600° F).											

NOTE: Dimensional stability data shown above is the permanent metallurgical growth and/or shrinkage only. Thermal expansion effects are not included. For operating temperatures above 427° C (800° F), consult your Timken engineer.

HEAT GENERATION AND DISSIPATION

Bearing operating temperature is dependent upon a number of factors, including heat generation of all contributing heat sources, heat flow rate between sources and the ability of the system to dissipate the heat. Heat sources include such things as bearings, seals, gears, clutches and oil supply. Heat dissipation is affected by many factors, including shaft and housing materials and designs, lubricant circulation and external environmental conditions. These and other factors are discussed in the following sections.

HEAT GENERATION

Under normal operating conditions, most of the torque and heat generated by the bearing is caused by the elastohydrodynamic losses at the roller/ring contacts.

Heat generation is the product of bearing torque (M) and speed (n). The following equation is used to calculate the heat generated.

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

Where:

k_4 = 0.105 for Q_{gen} in W when M in N-m
= 6.73×10^{-4} for Q_{gen} in Btu/min when M in lbf-in.

If the bearing is tapered, the torque can be calculated using the following equation.

$$M = k_1 G_1 (n\mu)^{0.5} (Fa)^{0.3}$$

Where:

k_1 = bearing torque constant
= 7.97×10^{-6} for M in N-m
= 1.1×10^{-4} for M in lbf-in.
Fa = thrust load
μ = lubricant viscosity
G_1 = bearing geometry factor
(Part-specific; please contact your Timken representative.)

For thrust cylindrical and spherical roller bearings, the torque equations are given as follows, where the coefficients are based on series and found table 29:

$$M = \begin{cases} f_1 F_a dm + 10^{-7} f_0 (v \times n)^{2/3} dm^3 & \text{if } (v \times n) \geq 2000 \\ f_1 F_a dm + 160 \times 10^{-7} f_0 dm^3 & \text{if } (v \times n) < 2000 \end{cases}$$

Note that the viscosity is in units of centistokes and dm is the mean bearing diameter.

TABLE 29. COEFFICIENTS FOR THE TORQUE EQUATION

Bearing Type	Dimension Series	f_0	f_1
Thrust cylindrical roller bearings	11	3	0.00150
	12	4	0.00150
Thrust spherical roller bearings	92	2.5	0.00023
	93	2.5	0.00023
	94	3	0.00030

HEAT DISSIPATION

The problem of determining the heat flow from a bearing in a specific application is rather complex. In general, it can be said that factors affecting the rate of heat dissipation include the following:

1. Temperature gradient from the bearing to the housing. This is affected by size configuration of the house and any external cooling such as fans, water cooling or fan action of the rotating components.
2. Temperature gradient from the bearing to the shaft. Any other heat sources, such as gears and additional bearings and their proximity to the bearing considered, will influence the temperature of the shaft.
3. The heat carried away by a circulating oil system.

To what extent nos. 1 and 2 can be controlled will depend on the application. The heat-dissipation modes include conduction through the system, convection along the inside and outside surfaces of the system, as well as radiation exchange to and from neighboring structures. In many applications, overall heat dissipation can be divided into two categories – heat removed by circulating oil and heat removed through the structure.

Heat dissipation by circulating oil

The amount of heat removed by the lubricant can be controlled more easily. In a splash lubrication system, cooling coils may be used to control the bulk oil temperature.

The amount of heat carried away in a circulating oil system by the lubricant can be approximated from the following equations.

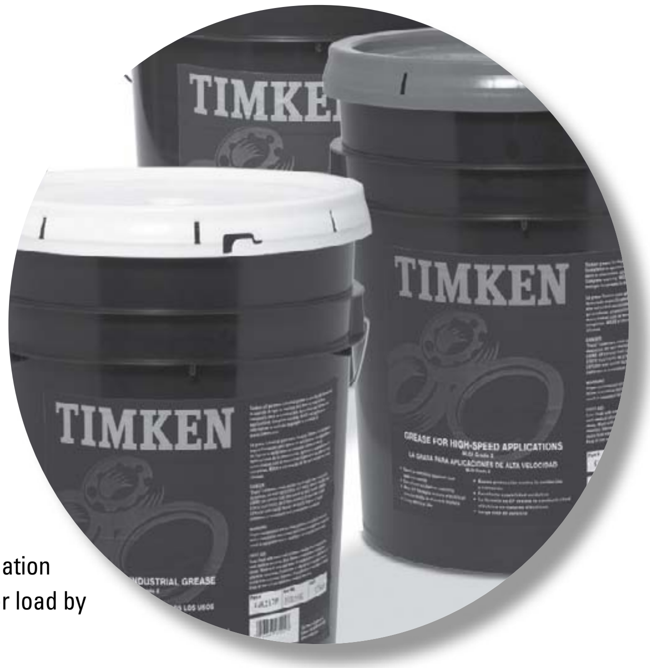
$$Q_{oil} = 1.67 \times 10^{-5}\ v\ C_p \rho\ (\theta_o - \theta_i)$$

Where:

V = oil flow rate (L/min)
Cp = Specific Heat of Lubricant (J/(kg- °C)
ρ = lubricant density (kg/m³)
θ_i = oil inlet temperature
θ_o = oil outlet temperature

DISCLAIMER

If a more thorough knowledge of bearing torque, power losses and system temperatures is needed, contact your Timken representative.



LUBRICATION

To help maintain a bearing's antifriction characteristics, lubrication is needed to:

- Minimize rolling resistance due to deformation of the rolling elements and raceway under load by separating the mating surfaces.
- Minimize sliding friction occurring between rolling elements, raceways and cage.
- Transfer heat (with oil lubrication).
- Protect from corrosion and, with grease lubrication, from contaminant ingress.

LUBRICATION

The wide range of bearing types and operating conditions precludes any simple, all-inclusive statement or guideline for selecting the proper lubricant. At the design level, the first consideration is whether oil or grease is best for the particular operation. The advantages of oil and grease are outlined in table 30. When heat must be carried away from the bearing, oil must be used. Oil is almost always preferred for very high-speed applications.

TABLE 30. ADVANTAGES OF OIL AND GREASE

Oil	Grease
Carries heat away from the bearings	Simplifies seal design and acts as a sealant
Carries away moisture and particulate matter	Permits prelubrication of sealed or shielded bearings
Easily controlled lubrication	Generally requires less frequent lubrication

European REACH compliance

Timken-branded lubricants, greases and similar products sold in stand-alone containers or delivery systems are subject to the European REACH (**R**egistration, **E**valuation, **A**uthorization and Restriction of **CH**emicals) directive. For import into the European Union, Timken can sell and provide only those lubricants and greases that are registered with ECHA (European CHemical Agency). For further information, please contact your Timken engineer.

OIL LUBRICATION

Oils used for bearing lubrication should be high-quality mineral oils or synthetic oils with similar properties. Selection of the proper type of oil depends on bearing speed, load, operating temperature and lubrication method. In addition to the above, some features and advantages of oil lubrication are:

- Oil is a better lubricant for high speeds or high temperatures. It can be cooled to help reduce bearing temperature.
- It is easier to handle and control the amount of lubricant reaching the bearing. It is harder to retain in the bearing. Lubricant losses may be higher than with grease.
- Oil can be introduced to the bearing in many ways, such as drip-feed, wick-feed, pressurized circulating systems, oil bath or air-oil mist. Each is suited for certain types of applications.
- Oil is easier to keep clean for recirculating systems.

Oil may be introduced to the bearing housing in many ways. The most common systems are:

- **Oil bath.** The housing is designed to provide a sump through which the rolling elements of the bearing will pass. Generally, the oil level should be no higher than the center point of the lowest rolling element. If speed is high, lower oil levels should be used to reduce churning. Gages or controlled elevation drains are used to achieve and maintain the proper oil level.
- **Circulating system.** This system has the advantages of:
  - An adequate supply of oil for both cooling and lubrication.
  - Metered control of the quantity of oil delivered to each bearing.
  - Removal of contaminants and moisture from the bearing by flushing action.
  - Suitability for multiple bearing installations.
  - Large reservoir, which reduces deterioration.
  - Increased lubricant life provides economical efficiency.
  - Incorporation of oil-filtering devices.
  - Positive control to deliver the lubricant where needed.
  - A typical circulating oil system consists of an oil reservoir, pump, piping and filter. A heat exchange may be required.
- **Oil-mist lubrication**. Oil-mist lubrication systems are used in high-speed, continuous-operation applications. This system permits close control of the amount of lubricant reaching the bearings. The oil may be metered, atomized by compressed air and mixed with air, or picked up from a reservoir using a venturi effect. In either case, the air is filtered and supplied under sufficient pressure to ensure adequate lubrication of the bearings. Control of this type of lubrication system is accomplished by monitoring the operating temperatures of the bearings being lubricated. The continuous passage of the pressurized air and oil through the labyrinth seals used in the system prevents the entrance of contaminants from the atmosphere to the system.

  The successful operation of this type of system is based upon the following factors:
  - Proper location of the lubricant entry ports in relation to the bearings being lubricated.
  - Avoidance of excessive pressure drops across void spaces within the system.

- Proper air pressure and oil quantity ratio to suit the particular application.
- Adequate exhaust of the air-oil mist after lubrication has been accomplished.

To ensure "wetting" of the bearings, and to prevent possible damage to the rolling elements and rings, it is imperative that the oil-mist system be turned on for several minutes before the equipment is started. The importance of "wetting" the bearing before starting cannot be overstated, and it also has particular significance for equipment that has been idled for extended periods of time.

Lubricating oils are commercially available in many forms for automotive, industrial, aircraft and other uses. Oils are classified as either petroleum types (refined from crude oil) or synthetic types (produced by chemical synthesis).

PETROLEUM OILS

Petroleum oils are made from a petroleum hydrocarbon derived from crude oil, with additives to improve certain properties. Petroleum oils are used for nearly all oil-lubricated applications of bearings.

SYNTHETIC OILS

Synthetic oils cover a broad range of categories and include polyalphaolefins, silicones, polyglycols and various esters. In general, synthetic oils are less prone to oxidation and can operate at extreme hot or cold temperatures. Physical properties, such as pressure-viscosity coefficients, tend to vary between oil types; use caution when making oil selections.

The polyalphaolefins (PAO) have a hydrocarbon chemistry that parallels petroleum oil both in chemical structures and pressure-viscosity coefficients. Therefore, PAO oil is mostly used in the oil-lubricated applications of bearings when severe temperature environments (hot and cold) are encountered or when extended lubricant life is required.

The silicone, ester and polyglycol oils have an oxygen-based chemistry that is structurally quite different from petroleum oils and PAO oils. This difference has a profound effect on its physical properties where pressure-viscosity coefficients can be lower compared to mineral and PAO oils. This means that these types of synthetic oils may actually generate a smaller elastohydrodynamic (EHD) film thickness than a mineral or PAO oil of equal viscosity at operating temperature. Reductions in bearing fatigue life and increases in bearing wear could result from this reduction of lubricant film thickness.

VISCOSITY

The selection of oil viscosity for any bearing application requires consideration of several factors: load, speed, bearing setting, type of oil and environmental factors. Since oil viscosity varies inversely with temperature, a viscosity value must always be stated with the temperature at which it was determined. High-viscosity oil is used for low-speed or high-ambient-temperature applications. Low-viscosity oil is used for high-speed or low-ambient-temperature applications.

There are several classifications of oils based on viscosity grades. The most familiar are the Society of Automotive Engineers (SAE) classifications for automotive engine and gear oils. The American Society for Testing and Materials (ASTM) and the International Organization for Standardization (ISO) have adopted standard viscosity grades for industrial fluids. Fig. 49 shows the viscosity comparisons of ISO/ASTM with SAE classification systems at 40° C (104° F).

VISCOSITY CLASSIFICATION COMPARISON

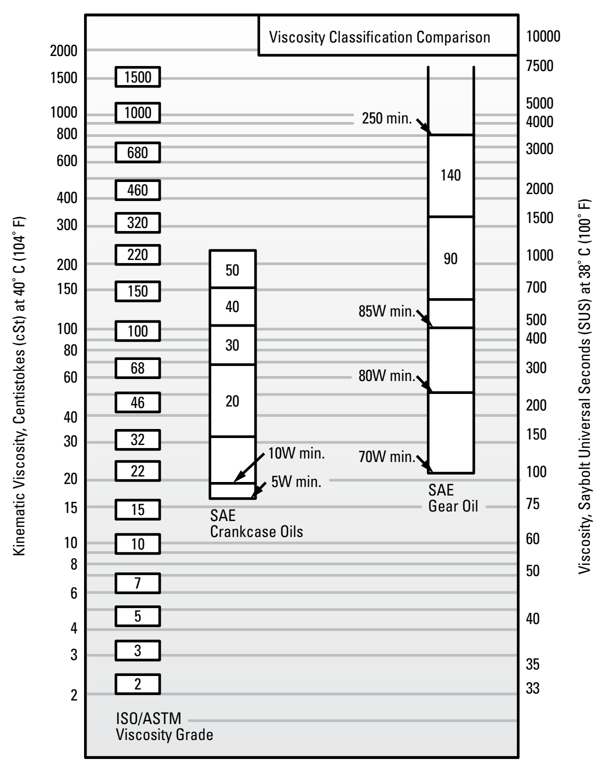


Fig. 49. Comparison between ISO/ASTM grades (ISO 3448/ASTM D2442) and SAE grades (SAE J 300-80 for crankcase oils, SAE J 306-81 for axle and manual transmission oils).

The ASTM/ISO viscosity grade system for industrial oils is depicted below.

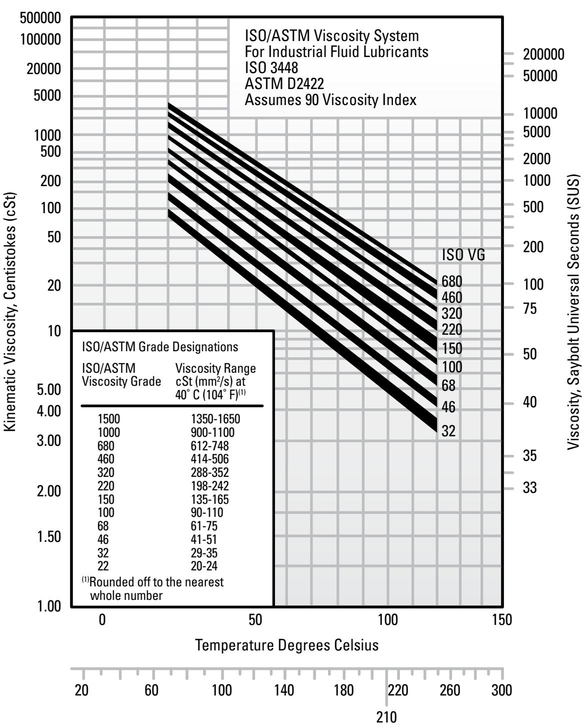


Fig. 50. Viscosity grade system for industrial oils.

TYPICAL BEARING LUBRICATION OILS

In this section, the properties and characteristics of lubricants for typical roller bearing applications are listed. These general characteristics are derived from successful performance in applications across all industries.

General-purpose rust and oxidation lubricating oil

General-purpose rust and oxidation (R&O) inhibited oils are the most common type of industrial lubricant. They are used to lubricate Timken® bearings in all types of industrial applications where conditions requiring special considerations do not exist.

TABLE 31. SUGGESTED GENERAL PURPOSE R&O LUBRICATING OIL PROPERTIES

Properties	
Base stock	Solvent-refined, high-viscosity-index petroleum oil
Additives	Corrosion and oxidation inhibitors
Viscosity index	80 min.
Pour point	-10° C max. (14° F)
Viscosity grades	ISO/ASTM 32 through 220

Some low-speed and/or high-ambient-temperature applications require the higher viscosity grades. High-speed and/or low-temperature applications require the lower viscosity grades.

Industrial extreme-pressure (EP) gear oil

Extreme-pressure gear oils are used to lubricate Timken bearings in most types of heavily loaded industrial equipment. They should be capable of withstanding abnormal shock loads that are common in heavy-duty equipment.

TABLE 32. SUGGESTED INDUSTRIAL EP GEAR OIL PROPERTIES

Properties	
Base stock	Solvent-refined, high-viscosity-index petroleum oil
Additives	Corrosion and oxidation inhibitors Extreme-pressure (EP) additive(1) - 15.8 kg (35 lb.) min.
Viscosity index	80 min.
Pour point	-10° C max. (14° F)
Viscosity grades	ISO/ASTM 100, 150, 220, 320, 460

(1) ASTM D 2782

Industrial EP gear oils should be composed of a highly refined petroleum oil-based stock plus appropriate inhibitors and additives. They should not contain materials that are corrosive or abrasive to bearings. The inhibitors should provide long-term protection from oxidation and protect the bearing from corrosion in the presence of moisture. The oils should resist foaming in service and have good water-separation properties. An EP additive protects against scoring under boundary-lubrication conditions. The viscosity grades suggested represent a wide range. High-temperature and/or slow-speed applications generally require the higher viscosity grades. Low temperatures and/or high speeds require the use of lower viscosity grades.

GREASE LUBRICATION

Grease lubrication is generally applicable to low-to-moderate speed applications that have operating temperatures within the limits of the grease. There is no universal antifriction bearing grease. Each grease has limiting properties and characteristics.

Greases consist of a base oil, a thickening agent and additives. Conventionally, bearing greases have consisted of petroleum base oils thickened to the desired consistency by some form of metallic soap. More recently synthetic base oils have been used with organic and inorganic thickeners. Table 33 summarizes the composition of typical lubricating greases.

TABLE 33. COMPOSITION OF GREASES

Base Oil	+	Thickening Agents	+	Additives	=	Lubricating Grease
Mineral oil Synthetic hydrocarbon Esters Perfluorinated oil Silicone		Soaps and complex soaps lithium, aluminum, barium, calcium Non-Soap (inorganic) microgel (clay), carbon black, silica-gel, PTFE Non-Soap (organic) Polyurea compounds		Rust inhibitors Dyes Tactifiers Metal deactivates Oxidation inhibitors Anti-wear EP		

Calcium- and aluminum-based greases have excellent water resistance and are used in industrial applications where water ingress is an issue. Lithium-based greases are multi-purpose and are used in industrial applications and wheel bearings.

Synthetic-based oils such as esters, organic esters and silicones used with conventional thickeners and additives typically have higher maximum operating temperatures than petroleum-based greases. Synthetic greases can be designed to operate in temperatures from -73° C (-100° F) to 288° C (550° F).

Below are the general characteristics of common thickeners used with petroleum base oils.

TABLE 34. GENERAL CHARACTERISTICS OF THICKENERS USED WITH PETROLEUM BASE OILS

Thickener	Typical Dropping Point		Maximum Temperature		Typical Water Resistance
	°C	°F	°C	°F	
Lithium soap	**193**	380	**121**	250	Good
Lithium complex	**260+**	500+	**149**	300	Good
Aluminum complex	**249**	480	**149**	300	Excellent
Calcium sulfonate	**299**	570	**177**	350	Excellent
Polyurea	**260**	500	**149**	300	Good

Use of the thickeners in table 34 with synthetic hydrocarbon or ester base oils increases the maximum operating temperature by approximately 10° C (50° F).

Using polyurea as a thickener for lubricating fluids is one of the most significant lubrication developments in more than 30 years. Polyurea grease performance is outstanding in a wide range of bearing applications and, in a relatively short time, it has gained acceptance as a factory-packed lubricant for ball bearings.

LOW TEMPERATURES

Starting torque in a grease-lubricated bearing at low temperatures can be critical. Some greases may function adequately as long as the bearing is operating, but resistance to initial movement may be excessive. In certain smaller machines, starting may be impossible when very cold. Under such operating circumstances, greases containing low-temperature characteristic oils are generally required.

If the operating temperature range is wide, synthetic greases offer advantages. Synthetic greases are available to provide very low starting and running torque at temperatures as low as -73° C (-100° F). In certain instances, these greases perform better in this respect than oil.

An important point concerning lubricating greases is that the starting torque is not necessarily a function of the consistency or the channel properties of the grease. Starting torque is more a function of the individual rheological properties of a particular grease and is best evaluated by application experience.

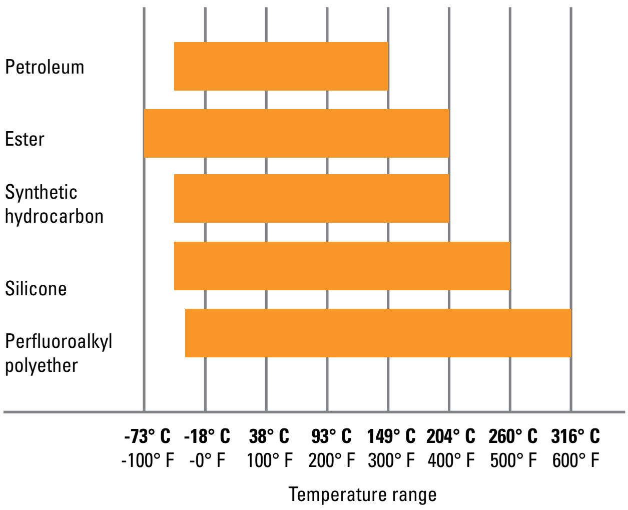
HIGH TEMPERATURES

The high temperature limit for lubricating greases is generally a function of the thermal and oxidation stability of the fluid and the effectiveness of the oxidation inhibitors. Grease temperature ranges are defined by both the dropping point of the grease thickener and composition of the base oil. Table 35 shows the temperature ranges of various base oils used in grease formulations.

A rule of thumb, developed from years of testing grease-lubricated bearings, indicates that grease life is halved for every 10° C (50° F) increase in temperature. For example, if a particular grease provides 2000 hours of life at 90° C (194° F), by raising the temperature to 100° C (212° F), reduction in life to approximately 1000 hours would result. On the other hand, 4000 hours could be expected by lowering the temperature to 80° C (176° F).

Thermal stability, oxidation resistance and temperature limitations must be considered when selecting greases for high-temperature applications. In non-relubricatable applications, highly refined mineral oils or chemically stable synthetic fluids are required as the oil component of greases for operation at temperatures above 121° C (250° F).

TABLE 35. TEMPERATURE RANGES FOR BASE OILS USED IN LUBRICATING GREASES



CONTAMINATION

Abrasive Particles

When roller bearings operate in a clean environment, the primary cause of damage is the eventual fatigue of the surfaces where rolling contact occurs. However, when particle contamination enters the bearing system, it is likely to cause damage such as bruising, which can shorten bearing life.

When dirt from the environment or metallic wear debris from some component in the application is allowed to contaminate the lubricant, wear can become the predominant cause of bearing damage. If bearing wear becomes significant, changes will occur to critical bearing dimensions that could adversely affect machine operation.

Bearings operating in a contaminated lubricant exhibit a higher initial rate of wear than those running in an uncontaminated lubricant. With no further contaminant ingress, this wear rate quickly diminishes. The contamination particles are reduced in size as they pass through the bearing contact area during normal operation.

Water

Water and moisture can be particularly conducive to bearing damage. Lubricating greases may provide a measure of protection from this contamination. Certain greases, such as calcium and aluminum-complex, are highly water-resistant.

Sodium-soap greases are water-soluble and should not be used in applications involving water.

Dissolved or suspended water in lubricating oils can exert a detrimental influence on bearing fatigue life. Water can cause bearing etching that also can reduce bearing fatigue life. The exact mechanism by which water lowers fatigue life is not fully understood. It has been suggested that water enters micro-cracks in the bearing rings that are caused by repeated stress cycles. This leads to corrosion and hydrogen embrittlement in the micro-cracks, reducing the time required for these cracks to propagate to an unacceptable-sized spall.

Water-based fluids, such as water glycol and invert emulsions, also have shown a reduction in bearing fatigue life. Although water from these sources is not the same as contamination, the results support the previous discussion concerning water-contaminated lubricants.

GREASE SELECTION

The successful use of bearing grease depends on the physical and chemical properties of the lubricant as well as application and environmental conditions. Because the choice of grease for a particular bearing under certain service conditions is often difficult to make, you should consult with your lubricant supplier or equipment maker for specific questions about lubrication requirements for your application. You also can contact your Timken engineer for general lubrication guidelines for any application.

Grease must be carefully selected with regard to its consistency at operating temperature. It should not exhibit thickening, separation of oil, acid formation or hardening to any marked degree. It should be smooth, non-fibrous and entirely free from chemically active ingredients. Its dropping point should be considerably higher than the operating temperature.

Timken® application-specific lubricants were developed by leveraging our knowledge of tribology and anti-friction bearings, and how these two elements affect overall system performance. Timken® lubricants help bearings and related components operate effectively in demanding industrial operations. High-temperature, anti-wear and water-resistant additives offer superior protection in challenging environments. Table 36 provides an overview of the Timken greases available for general applications. Contact your Timken engineer for a more detailed publication on Timken® lubrication solutions.

TABLE 36. GREASE LUBRICATION SELECTION GUIDE

ENVIRONMENT		APPLICATION
High Wear • Moderate Loads Moderate Speeds Moderate Temperatures	Timken Premium All-Purpose Industrial Grease	Agriculture • Bushings/Ball Joints Truck and Auto Wheel Bearings Heavy-Duty Industrial
Extreme Heat • Heavy Loads High Sliding Wear Dirty Environments Slow Speeds • Shock Loading	Timken Construction and Off-Highway Grease	Agriculture/Mining • Cement Plants Construction/Off Road • Rock Quarry Earth-Moving Equipment Fleet Equipment • Heavy Industry Pivot Pins/Splined Shafts
Wet and Corrosive Conditions Quiet Environments • Light Loads Moderate to High Speeds Moderate Temperatures Light Load Moderate Water	Timken Ball Bearing Pillow Block Grease	Lightly Loaded Pillow Blocks Idler Pulleys • Oven Conveyors Electric Motors • Fans • Pumps Alternators • Generators
Corrosive Media • Extreme Heat Heavy Loads • Wet Conditions Slow to Moderate Speeds	Timken Mill Grease	Aluminum Mills • Paper Mills Steel Mills • Offshore Rigs Power Generation
Incidental Food Contact Hot and Cold Temperatures Moderate to High Speeds Medium Loads	Timken Food Safe Grease	Food and Beverage Industries Pharmaceuticals
Extreme Low and High Temperatures Severe Loads Corrosive Media Slow to Moderate Speeds	Timken Synthetic Industrial Grease	Wind Energy Main Bearing Pulp and Paper Machines General Heavy Industry Marine Applications Centralized Grease Systems
Moderate Speeds Light to Moderate Loads Moderate Temperatures Moderate Water	Timken Multi-Use Lithium Grease	General Industrial Applications Pins and Bushings • Track Rollers Water Pumps Plain and Anti-Friction Bearings

This selection guide is not intended to replace the specifications by the equipment builder, who is responsible for its performance.

Many bearing applications require lubricants with special properties or lubricants formulated specifically for certain environments, such as:

- Friction oxidation (fretting corrosion).
- Chemical and solvent resistance.
- Food handling.
- Quiet running.
- Space and/or vacuum.
- Electrical conductivity.

For assistance with these or other areas requiring special lubricants, consult your Timken engineer.

GREASE USE GUIDELINES

It is important to use the proper amount of grease in the application. In typical industrial applications, the bearing cavity should be kept approximately one-third to one-half full. Less grease may result in the bearing being starved for lubrication. More grease may result in churning. Both conditions may result in excessive temperature. As the grease temperature rises, viscosity decreases and the grease becomes thinner. This can reduce the lubricating effect and increase leakage of the grease from the bearing. It also may cause the grease components to separate, leading to a general breakdown of the lubricant properties. As the grease breaks down, bearing torque increases. In the case of excess grease resulting in churning, torque may also increase due to the resistance caused by the grease.

For best results, there should be ample space in the housing to allow room for excess grease to be thrown from the bearing. However, it is equally important that the grease be retained all around the bearing. If a large void exists between the bearings, grease closures should be used to prevent the grease from leaving the bearing area.

Only in low-speed applications may the housing be entirely filled with grease. This method of lubrication is a safeguard against the entry of foreign matter, where sealing provisions are inadequate for exclusion of contaminants or moisture.

During periods of non-operation, it is often wise to completely fill the housings with grease to protect the bearing surfaces. Prior to restarting operation, remove the excess grease and restore the proper level.

Applications utilizing grease lubrication should have a grease fitting and a vent at opposite ends of the housing near the top. A drain plug should be located near the bottom of the housing to allow the old grease to purge from the bearing.

Bearings should be relubricated at regular intervals to prevent damage. Relubrication intervals are difficult to determine. If plant practice or experience with other applications is not available, consult your lubricant supplier.

Timken offers a range of lubricants to help bearings and related components operate effectively in demanding industrial operations. High-temperature, anti-wear and water-resistant additives offer greater protection in challenging environments. Timken also offers a line of single- and multi-point lubricators to simplify grease delivery.

Fig. 51. Grease can easily be packed by hand.

Fig. 52. Mechanical grease packer.

Grease application methods

Grease, in general, is easier to use than oil in industrial bearing lubrication applications. Most bearings that are initially packed with grease require periodic relubrication to operate efficiently.

Grease should be packed into the bearing so that it gets between the rolling elements – the rollers or balls. For tapered roller bearings, forcing grease through the bearing from the large end to the small end will ensure proper distribution.

Grease can be easily packed into small- and medium-size bearings by hand (fig. 51). In shops where bearings are frequently regreased, a mechanical grease packer that forces grease through the bearing under pressure may be appropriate (fig. 52). Regardless of the method, after packing the internal areas of the bearing, a small amount of grease also should be smeared on the outside of the rollers or balls.

The two primary considerations that determine the relubrication cycle are operating temperature and sealing efficiency. High-operating-temperature applications generally require more frequent regreasing. The less efficient the seals, the greater the grease loss and the more frequently grease must be added.

Grease should be added any time the amount in the bearing falls below the desired amount. The grease should be replaced when its lubrication properties have been reduced through contamination, high temperature, water, oxidation or any other factors. For additional information on appropriate regreasing cycles, consult with the equipment manufacturer or your Timken engineer.

CONSISTENCY

Greases may vary in consistency from semi-fluids that are hardly thicker than a viscous oil to solid grades almost as hard as a soft wood.

Consistency is measured by a penetrometer in which a standard weighted cone is dropped into the grease. The distance the cone penetrates (measured in tenths of a millimeter in a specific time) is the penetration number.

The National Lubricating Grease Institute (NLGI) classification of grease consistency is shown below:

TABLE 37. NLGI CLASSIFICATIONS

NLGI Grease Grades	Penetration Number
0	355-385
1	310-340
2	265-295
3	220-250
4	175-205
5	130-160
6	85-115

Grease consistency is not fixed; it normally becomes softer when sheared or “worked.” In the laboratory, this “working” is accomplished by forcing a perforated plate up and down through a closed container of grease. This “working” does not compare with the violent shearing action that takes place in a bearing and does not necessarily correlate with actual performance.

TABLE 38. GREASE COMPATIBILITY CHART

■ = Best Choice
■ = Compatible
■ = Borderline
■ = Incompatible

	Al Complex	Ba Complex	Ca Stearate	Ca 12 Hydroxy	Ca Complex	Ca Sulfonate	Clay Non-Soap	Li Stearate	Li 12 Hydroxy	Li Complex	Polyurea	Polyurea S S
Aluminum Complex	Best Choice	Incompatible	Incompatible	Compatible	Incompatible	Borderline	Incompatible	Incompatible	Incompatible	Compatible	Incompatible	Compatible
Timken Food Safe	Best Choice	Incompatible	Incompatible	Compatible	Incompatible	Borderline	Incompatible	Incompatible	Incompatible	Compatible	Incompatible	Compatible
Barium Complex	Incompatible	Best Choice	Incompatible	Compatible	Incompatible	Compatible	Incompatible	Incompatible	Incompatible	Incompatible	Incompatible	Borderline
Calcium Stearate	Incompatible	Incompatible	Best Choice	Compatible	Incompatible	Compatible	Compatible	Compatible	Borderline	Compatible	Incompatible	Compatible
Calcium 12 Hydroxy	Compatible	Compatible	Compatible	Best Choice	Borderline	Borderline	Compatible	Compatible	Compatible	Compatible	Incompatible	Compatible
Calcium Complex	Incompatible	Incompatible	Incompatible	Borderline	Best Choice	Incompatible	Incompatible	Incompatible	Incompatible	Compatible	Compatible	Compatible
Calcium Sulfonate	Borderline	Compatible	Compatible	Borderline	Incompatible	Best Choice	Incompatible	Borderline	Borderline	Compatible	Incompatible	Compatible
Timken Premium Mill Timken Heavy-Duty Moly	Borderline	Compatible	Compatible	Borderline	Incompatible	Best Choice	Incompatible	Borderline	Borderline	Compatible	Incompatible	Compatible
Clay Non-Soap	Incompatible	Incompatible	Compatible	Compatible	Incompatible	Incompatible	Best Choice	Incompatible	Incompatible	Incompatible	Incompatible	Borderline
Lithium Stearate	Incompatible	Incompatible	Compatible	Compatible	Incompatible	Borderline	Incompatible	Best Choice	Compatible	Compatible	Incompatible	Compatible
Lithium 12 Hydroxy	Incompatible	Incompatible	Borderline	Compatible	Incompatible	Borderline	Incompatible	Compatible	Best Choice	Compatible	Incompatible	Compatible
Lithium Complex	Compatible	Incompatible	Compatible	Compatible	Compatible	Compatible	Incompatible	Compatible	Compatible	Best Choice	Incompatible	Compatible
Polyurea Conventional	Incompatible	Incompatible	Incompatible	Incompatible	Compatible	Incompatible	Incompatible	Incompatible	Incompatible	Incompatible	Best Choice	Compatible
Polyurea Shear Stable	Compatible	Borderline	Compatible	Compatible	Compatible	Compatible	Borderline	Compatible	Compatible	Compatible	Compatible	Best Choice
Timken Multi-Use	Incompatible	Incompatible	Borderline	Compatible	Incompatible	Borderline	Incompatible	Compatible	Best Choice	Compatible	Incompatible	Compatible
Timken All-Purpose Timken Synthetic	Compatible	Incompatible	Compatible	Compatible	Compatible	Compatible	Incompatible	Compatible	Compatible	Best Choice	Incompatible	Compatible
Timken Pillow Block	Compatible	Borderline	Compatible	Compatible	Compatible	Compatible	Borderline	Compatible	Compatible	Compatible	Compatible	Best Choice

NOTE

Mixing greases can result in improper bearing lubrication.
Always follow the specific lubrication instructions of your equipment supplier.