Cylindrical Roller Bearings

THE M SERIES designated by the letter M satisfies most commercial applications and is available in a broad range of sizes and types up to 20" (508 mm) outside diameter.

THE MAX-PAK OR W-60000 SERIES is designed for applications with very heavy radial loads and where space for the bearing may be limited. The envelope dimensions are the same as the M series.

THE MOJ SERIES offers economical journal roller assemblies without inner or outer rings for operation in very limited space.

SPECIAL BEARINGS are available for the chain and mast guide, steel mill, rear wheel and pinion applications. Other bearings can be engineered for special requirements.

Tapered Roller Bearings

SINGLE ROW TAPERED ROLLER BEARINGS are available in many different series with straight and flanged cups up to 20" (508 mm) diameter.

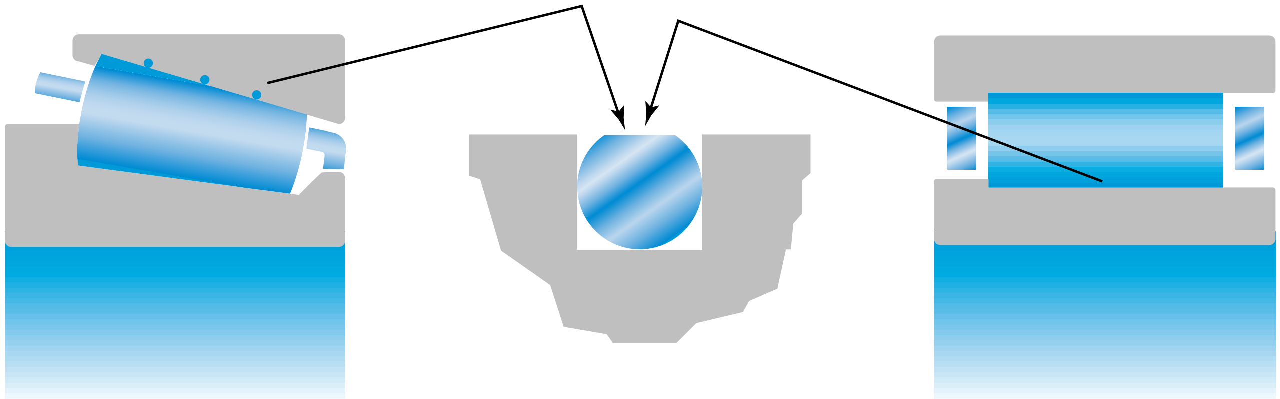
TWO ROW TAPERED ROLLER BEARINGS are available in many different series and configurations up to 20" (508 mm) outside diameter.

Two Row Bearing

Two Row Spacer Assembly

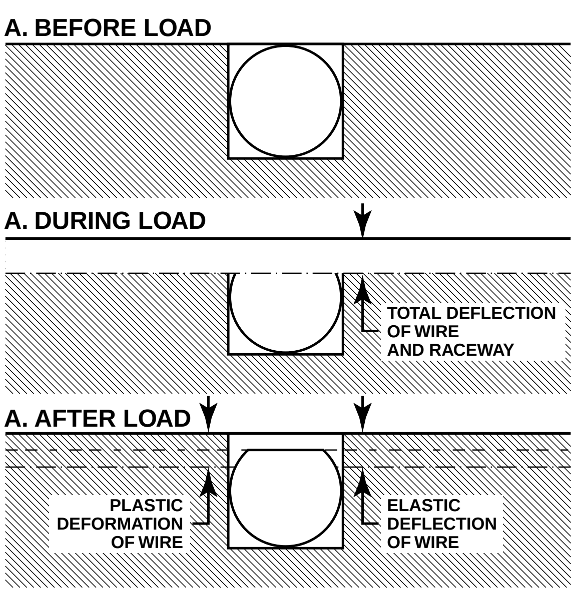
FOUR ROW TAPERED ROLLER BEARING ASSEMBLIES engineered for steel mill applications are available up to 20" (508 mm) outside diameter.

Signature Bearing



Tapered and cylindrical Signature roller bearings are used for measuring load, alignment, and load distribution in an application environment. Actual shaft and bearing housing and deflection information, often difficult to determine using analytical methods, is also included with the Signature bearing measurements. Signature bearings provide an accurate, comprehensive, and cost effective means of measurement with no modifications to the application unit required.

The Signature bearings use for measurement the difference in yield strengths between the hardened steel bearing components and the lower yield strength material inserted in grooves on the bearing rolling surface. The lower strength material takes a permanent set at bearing operating loads. The analysis information is permanently registered on the plastically deformed ductile inserts; therefore, no wire leads or recording devices are required.



The deflection of the raceway surface and the insert is depicted here graphically. As shown in view A, the race and insert surfaces are flush at the start of the test. In view B, the race and insert surfaces are deflected to some new level under load. The insert in the position does not provide the support for the load, but rather yields to the level of the supporting raceway. In view C, the load has been removed and the raceway surface has returned to its original unloaded position. The elastic limit of the race material was higher than the applied stress. The insert has not returned to its original unloaded position since its elastic limit has been exceeded. The total deflection of the insert under load is made up of both an elastic and plastic component. The elastic deflection will remain the same for all stresses exceeding the elastic limit of the insert. It is the variation in plastic deformation with load that is used for analysis.

A variety of insert configurations are available on the Signature bearing. The number and location of the inserts will be selected to meet the needs of the user. The standard configuration uses three ductile inserts spaced axially along the track as shown in the illustrations. The two end inserts are the best indicators of axial alignment. With the inserts positioned in the stationary raceway they record both maximum load as well as load zone distribution.

The Signature bearing is ideal for verifying the load conditions of your new design in the field or for use as a quality assurance check in production. Details of the Signature bearing are reported in SAE paper 850765, and can be obtained from NTN Sales.

Glossary of Symbols

A	Cylindrical bearing inner ring raceway diameter
a_1	Life adjustment factor for reliability
a_2	Life adjustment factor for material
a_3	Life adjustment factor for lubrication
a_4	Life adjustment factor for misalignment
a_5	Life adjustment factor for load zone size
B	Bearing inner ring bore
C	Cylindrical bearing outer ring raceway diameter
C(90)	Bearing radial rating @ 90 x 10^6 cycles
CA(90)	Bearing thrust rating @ 90 x 10^6 cycles
CA_{lim}	Limiting thrust rating for cylindrical roller bearings
CCW	Counterclockwise
CF	Centrifugal force
CW	Clockwise
D	Bearing outside diameter
E	Modulus of elasticity
F	Force
F_a	Thrust (Axial) component of F_n or axial force
F_n	Normal force
F_r	Radial force
F_s	Separating component of F_n
F_t	Tangentail component of F_n
f_t	Thrust factor for cylindrical roller bearing thrust rating
f_{pl}	Preload factor
G	Subscript for ring gear
H	Housing O.D.
HP	Horsepower
IF	Interference fit
J	Hollow shaft I.D.
K	Ratio of radial to thrust rating for tapered roller bearings
L_{10}	Bearing life @ 90% reliability level
L_n	Bearing life @ RL reliability level
L'_{10}	Adjusted bearing life @ 90% reliability level
L'_n	Adjusted bearing life @ RL reliability level
LH	Left hand
MPD	Mean pitch diameter
N_n	Number of teeth in gear “n”
n	Subscript index
P	Equivalent radial load for tapered roller bearings
P	Subscript for pinion
PD	Pitch diameter
p	Radial contact pressure
Q	Torque
R_n	Bearing “n” radial reaction
RL	Reliability level
RH	Right hand
r	Radius
S	Rotational speed (rpm)
T_1	Belt tension-tight side
T_2	Belt tension-loose side
TR_n	Thrust reaction of tapered bearing “n”
W	Gear face width
Wt	Weight
α (alpha)	1/2 included cup angle
β (beta)	Pitch angle for straight, zerol, and spiral bevel gears
β (beta)	Face angle of hypoid pinion and root angle of hypoid gear
δ_i (delta)	Change inner ring raceway diameter
δ_o (delta)	Change outer ring raceway diameter
υ (nu)	Poisson’s ratio
Σ (sigma)	Summation
ϕ (phi)	Normal pressure angle
ϕ_r (phi)	Pressure angle in plane of rotation
ψ (psi)	Helix or spiral angle

INTRODUCTION

The selection of the proper bearings for all mechanical systems is essential to the functional and commercial success of that system. The bearings must not only be of the right type, but also the correct size to assure reliability and cost effectiveness. The bearings must be installed properly, supplied with the correct lubricant, and provided with a compatible environment for the system to be successful. This catalog is designed to provide wuidelines for the engineer to follow in making proper bearing selection and in establishing an operating environment that will lead to reliable system performance. Because it is impossible to cover all aspects of bearing selection within any text due to the vast number of variables encountered, NTN maintains a staff of Bearing Application Engineers to assist customers in making bearing selections for applications of all kinds. We urge our customers to take advantage of this service. Application engineering assistance may be obtained by calling NTN Sales, or by contacting:

NTN Bearing Corporation of America
Application Engineering Department
1600 E. Bishop Court
P.O. Box 7604
Mt. Prospect, IL 60056-7604
708-298-7500 (Fax: 708-699-9744)

BEARING LIFE DEFINITION

All roller bearings have a finite life. Therefore, it is necessary to develop techniques to estimate their lives. Theoretical bearing life is defined as the time (measured in revolutions) to the initial occurrence of rolling contact fatigue on either raceway or any rolling element. Rolling contact fatigue is subsurface initiated damage that occurs after many revolutions of the bearing. When a bearing is rotated under load, the raceways and rolling elements are subjected to cyclic Hertzian stresses as they pass through the load zone. After millions of cycles, microscopic cracks form beneath the bearing surfaces. As the bearing continues to operate, the cracks eventually propagate to the surface causing small particles of steel to break away from the surface. This type of damage is called spalling. See Figure 1.

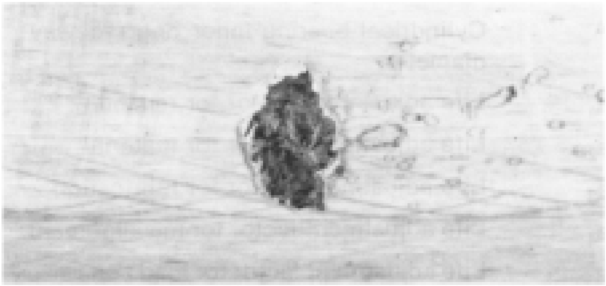


FIGURE 1

The laboratory criterion used to define the fatigue life of a bearing is the time period until either raceway or any rolling element develops a spall with an area of 0.01 in^2 (6 mm^2). This definition is necessary for a meaningful comparison of bearing lives under controlled conditions. However, in many applications, a spall of this size may have no immediate or short term adverse effect on total system performance. The size of a spall before a bearing becomes unsuitable for further use is dependent on the nature of the application and how much noise, vibration, or both can be tolerated. The time when a bearing becomes unsuitable for further service is sometimes referred to as its useful life in contrast to its fatigue life. The length of the period between the fatigue life and the useful life is a function of the stress level, the steel alloy and its heat-treatment, and the lubrication. Further information on this subject may be obtained from the NTN Application Engineering Department.

It is impossible to predict the exact fatigue life of an individual bearing. A group of apparently identical bearings subjected to the same conditions of load, speed, lubrication, and temperature will produce a considerable scatter of fatigue lives. Therefore, statistical methods are required to predict the life of the group. The Weibull distribution is generally used to evaluate these types of data. It is common practice to specify the life of the group at the L_{10} level which is the life that 90% of the group will achieve or exceed. Stating this another way, 10% of the group will have experienced fatigue of one or more components at the L_{10} level.

Many other factors besides fatigue may effect bearing performance. These include lubrication, misalignment, contamination, internal operating clearance, etc. Evaluation of these parameters is addressed in the life adjustment factor portion of the Bearing Life Calculations section, page 20.

Bearing Load Ratings

As previously defined, the fatigue life of a rolling bearing is determined by the number of revolutions under load that a bearing experiences prior to the initiation of rolling contact fatigue. Because of the natural scatter of lives in a group of bearings operating under identical conditions, the life of the group is specified at some reliability level, usually 90%. In order to evaluate the life of a bearing in a specific application, a radial load rating has been established for each bearing size. This load rating is based on a 90% survival expectation of a group of bearings operating under a constant radial load for a specific number of revolutions. It is common industry practice to specify the load rating for roller bearings at 90 million revolutions (3000 hrs @ 500 rpm). This rating is designated by the symbol "C(90)". These load ratings are tabulated in the appropriate product line sections of this catalog. The use of the load rating to estimate bearing life for a specific application is covered in the Bearing Life Calculations section, page 20.

To verify load ratings in the laboratory, it is necessary to control the other variables which affect the fatigue life of a bearing. Typical test conditions established by NTN-Bower for fatigue life comparisons are shown below. These conditions may be adjusted according to bearing size and type.

Reliability:	90%
Load:	2.0 x C(90)
Lubrication:	SAE 30 weight oil
Temperature:	150° to 180° F
Speed:	1800 rmp
Alignment:	0 to 0.001 Radian
Load Zone:	180°
Spall Size:	0.01 in^2 (6 mm^2)

BEARING SELECTION

Introduction

The prime factors in bearing selection are a total system reliability for its design life and the cost effectivenesss. To achieve such reliability, the bearings must be of the proper type and size. The selection process must consider all factors which will affect bearing performance and cost. These factors include:

- Magnitude and direction of loads
- Speed of rotation
- Required life
- Available Space
- Lubrication
- Shaft and housing designs
- Alignment
- Adjustment
- Temperature
- Environment

It is impossible to select any one of these factors as being the most critical. All must be considered in every bearing application. Each application will dictate their relative importance which will in turn guide the engineer toward proper bearing selection. We recommend that the NTN Application Engineering Deparment be consulted on all bearing applications.

Life Calculation Methods

Standard methods for estimating bearing lives have been developed for most applications. Such methods include:

- Maximum horsepower
- Skid torque
- Tractive effort
- Design load
- Work schedule

Whenever possible, the bearing selection for new applications should be based on a comparison of the calculated lives of bearings in similar successful applications using the same method. For example, in truck applications, the wheel bearing life calculations may be based on the design GVW (Gross Vehicle Weight) at 40 mph and the power train on tractive effort methods or specific route schedules. Design bogies are established for each method to assure commercial success of the vehicle. This procedure has proven to be successful in selecting bearings for many different applications. Ongoing programs update calculation methods to make them more realistically correlate with actual field conditions. An engineer must be careful when comparing new and old application calculations that the methods and the bearing ratings are identical. NTN-Bower has established life goals (measured in hours or vehicle roll miles) based on the calculated loads and speeds from the standard evaluation methods. This information is available from the NTN Application Engineering Department.

Load Analysis

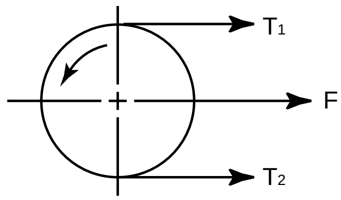
In many applications, the load and speed considerations are critical to the bearing selection. Methods of analyzing load sources and the resolution of these loads into bearing reactions are presented below. Frequently, the methods to evaluate the magnitude of the load and the speed are based on a history of performance of similar equipment. Such standard approaches are essential when the bearings are exposed to a full spectrum of loads and speeds and/or a wide variety of work schedules.

The first step in the process is to determine the magnitude and direction of the loads which the bearings are required to support. Loads may originate from a variety of sources including dead weight, belts, chains, sprockets, gears, imbalance, etc. Each load source is discussed below:

Dead weight may be either concentrated or distributed over a given area. For most bearing applications, distributed loads may be resolved into a single concentrated load acting vertically through the center of gravity. For example, the location of the center of gravity in an automobile will determine load distribution between the four wheels. The load at each wheel is distributed over the area of contact between the tire and the road. This load may be considered concentrated at the geometric center of the contact area acting normal to the road surface.

Belts are encountered in a wide variety of industrial applications. They are used for both power transmission and conveyor systems. Power transmission belts may be flat, "V" sectioned, or cogged for timing applications. Conveyor belts are normally flat for moving palletized loads or contoured to a trough shape for bulk materials. Friction between the drive pulley and the belt transmits the motive power in all applications except for cogged timing belts. To assure that sufficient frictional forces exist, the belts must be installed with the proper amount of preload tension. Belt manufacturers provide guidelines to establish the correct value for the preload.

The resultant force created on the drive and idler pulleys in any belt system must include the preload tension, the forces caused by the driving horsepower, and the weight of the material being transported in the case of conveyor systems. When the belt wrap is around 180°, formula (1) approximates the force which must be supported by the pulley bearings.



DRIVE PULLEY
FIGURE 2

$$F = T_1 + T_2 = \frac{126050 \times HP \times f_{pl}}{S \times PD} \qquad (1)$$

where
- T_1 = Tension on the tight side lb.
- T_2 = Tension on the slack side lb.
- HP = Horsepower
- S = Speed in rpm
- PD = Pulley pitch diameter in.
- f_{pl} = Preload factor
- f_{pl} = 1.1 to 1.2 cogged belts
- f_{pl} = 1.5 to 2.0 V-belts
- f_{pl} = 2.0 to 4.0 flat belts

The relatively wide ranges for the f_{pl} factor are due to the variations in field practices for setting the preload on the belt. Experience with similar installations is necessary for a closer approximation for f_{pl}. Note that in static conditions $T_1 = T_2$ = preload tension.

When the belt wrap varies significantly from 180°, the vector sum of T_1 and T_2 should be used to calculate F.

Chain and sprocket drives do not rely on friction to transmit the motive power to the chain and therefore may have zero or only a small preload. Formula (1) given above for belts is still valid for many chain and sprocket drives using f_{pl} in the range of 1.0 to 1.2. Some sprocket drives, such as used in crawler tractors, may have a heavy preload from hydraulic and/or mechanical systems to keep the track taut. The f_{pl} factor must be significantly increased to account for this preload. For further information, consult with the NTN Application Engineering Department.

Spur gears are the most common type used for positive power transmission between parallel shafts. The faces of the teeth are nearly always of involute form with a pressure angle of 14-1/2°, 20°, or 25°. The tooth surfaces are parallel to the axis of rotation.

Tangential Component $$F_t = \frac{Q \times 2}{PD} \quad (2)$$

Separating Component $$F_s = F_t \times \tan\phi \quad (3)$$

Normal Force $$F_n = \frac{F_t}{\cos\phi} \quad (4)$$

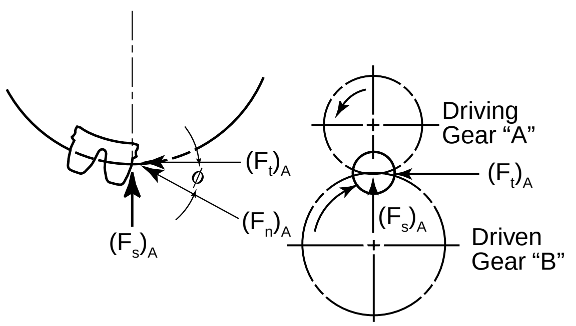
where Q = Torque (lb in)

PD = Gear pitch diameter (in)

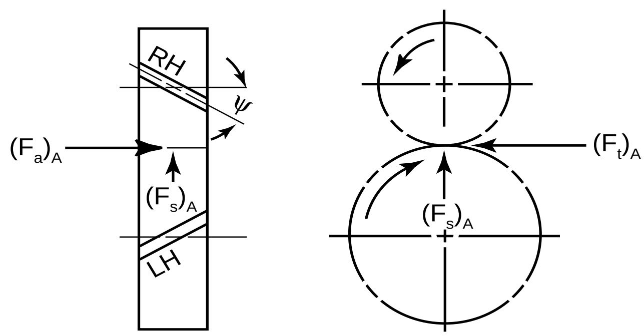
ø = Normal pressure angle (deg)

The direction of the thrust components may be determined from Figure 4. The direction of the tangential and separating components is the same as shown for spur gears in Figure 3.

	Hobbed		Shaped	
Tangential Component	$F_t = \frac{Q \times 2}{PD}$	(5)	$F_t = \frac{Q \times 2}{PD}$	(8)
Separating Component	$F_s = \frac{F_t \times \tan\phi}{\cos\psi}$	(9)	$F_s = F_t \times \tan\phi_r$	(9)
Axial (Thrust) Component	$F_a = F_t \times \tan\psi$	(10)	$F_a = F_t \times \tan\psi$	(10)



SPUR GEARS
FIGURE 3



HELICAL GEARS
FIGURE 4

The tangential component is sometimes referred to as the working component since it is directly proportional to the torque transmitted by the shaft. Spur gears may also be operated at a spread center distance in which case the operating pressure angle will increase above the theoretical value. In some bearing load calculations, an engineer may find it convenient to use the normal force.

Helical gears are similar to spur gears except that the teeth form a helix at the pitch diameter of the gear. Helical gears are formed by either hobbing or shaping. The tooth profile and the pressure angle are defined normal to the tooth surface for hobbed gears and in the plane of rotation for shaped gears. The two types will not mesh with each other.

Straight Bevel, Zerol Bevel, Spiral Bevel and Hypoid Gears are used to transmit power between non-parallel shafts; the most common angle between the shafts being 90°. The axes of rotation of the straight, zerol, and spiral bevel gears are coplanar while the axes of the hypoid gears are offset. The pitch diameter is defined at the heel (large end) of the ring gear. Because the load is distributed across the face of the tooth, the mean pitch diameter (defined in equation 11) is used in calculating the gear forces. The mean pitch diameter of the pinion is calculated by equation 12. The tangential components of the gear force are determined for the pinion and the gear by equations 13 and 14. Table I provides the formulas for the separating and thrust components of the ring gear and pinion forces.

$$MPD_G = PD - W \sin\beta_G \quad (11)$$

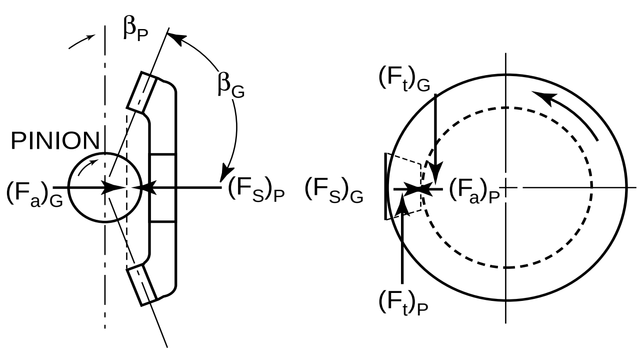
$$MPD_P = MPD_G \times \frac{N_P}{N_G} \times \frac{\cos\psi_G}{\cos\psi_P} \quad (12)$$

$$(F_t)_P = \frac{Q \times 2}{MPD_P} \quad (13)$$

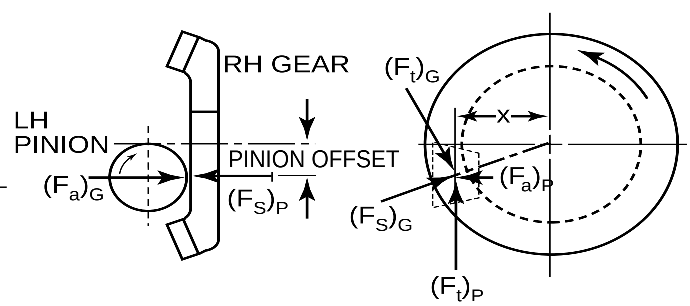
$$(F_t)_G = (Ft)_P \times \frac{\cos\psi_G}{\cos\psi_P} \quad (14)$$

TABLE I

Driving Member Hand & Rotation	Axial Component (Thrust)	Separating Component
RH/CW OR LH/CCW	**Driving Member** $F_a = \frac{F_t}{\cos\psi}$ (tan ϕ sin β– sinψ cos β)	**Driving Member** $F_s = \frac{F_t}{\cos\psi}$ (tan ϕ cos β+ sinψ sin β)
	Driven Member $F_a = \frac{F_t}{\cos\psi}$ (tan ϕ sin β+ sinψ cos β)	**Driven Member** $F_s = \frac{F_t}{\cos\psi}$ (tan ϕ cos β– sinψ sin β)
RH/CCW OR LH/CW	**Driving Member** $F_a = \frac{F_t}{\cos\psi}$ (tan ϕ sin β+ sinψ cos β)	**Driving Member** $F_s = \frac{F_t}{\cos\psi}$ (tan ϕ cos β– sinψ sin β)
	Driven Member $F_a = \frac{F_t}{\cos\psi}$ (tan ϕ sin β– sinψ cos β)	**Driven Member** $F_s = \frac{F_t}{\cos\psi}$ (tan ϕ cos β+ sinψ sin β)



STRAIGHT, ZEROL, AND SPIRAL BEVEL GEARS
FIGURE 5



HYPOID GEARS
FIGURE 6

1. The appropriate values of ϕ, ψ, and β for the driving and driven member must be used, respectively.
2. A posititive (+) value indicates the gears are separating.
3. A negative (–) value indicates the gears are being drawn together.
4. The load point on a hypoid pinion is determined from the offset and the MPD_G as shown in Figure 6.

$$x = \left[\left(\frac{MPD_G}{2}\right)^2 - offset^2\right]^{1/2} \quad (15)$$

5. For straight and zerol bevel gears, ψ = 0, therefore simplifying the equations in Table I.
6. For hypoid gears, β equals the face angle of the pinion and the root angle of the gear.

An Imbalance Force is generated when a mass rotates on an axis from its center of gravity. This imbalance, called a centrifugal force, will put an additional load on the support bearings. This load direction will remain stationary in regard to the rotating ring. The magnitude of the centrifugal force may be determined from equation 16.

$$\text{C.F.} = \frac{\text{Wt} \times \text{r} \times \text{S}^2}{3.52 \times 10^4}\ \text{lb.} \qquad (16)$$

The evaluation of a combination of rotating loads and stationary loads is a complex calculation and should be referred to the NTN Application Engineering Department.

THE CALCULATION OF BEARING LOADS

Before the actual bearing loads can be calculated, the bearing spread must be defined. For a shaft supported on two bearings, the bearing spread is defined as the distance between the two points which are considered to be the center of support for the load on the bearing. For cylindrical roller bearings, the point is defined as the intersection of the axis of rotation of the bearings and a plane normal to the axis through the midpoint of the roller length. See Figure 7.

For tapered roller bearings, the load on the bearing is considered to be normal to the shaft at a point which is the intersection of the axis of rotation and a line which is projected normal to the cup surface from the midpoint of the roller contact. This point is called the effective load center for a single row tapered roller bearing and is located at dimension “a” from the back face of the cone. This dimension “a” is tabulated for each cone in the dimensional data of the series listing of tapered roller bearings. For double row tapered roller bearings, the geometric center of the pair is used as the load center unless the external thrust load is sufficient to unseat one row in which case the effective center of the loaded row is used.

Single row tapered roller bearings may be mounted in either a direct mounting (Figure 8) or an indirect mounting (Figure 9). The direct mounting is frequently found in countershafts of transmissions in order to provide and end play adjustment through the stationary cups. The indirect mounting is common in wheel assemblies in order to provide greater stability to the assembly and, also, to allow for end play adjustment through the stationary cones. Certain thermal considerations may also influence the design and/or the end play recommendation. For further information, please contact the NTN Application Engineering Department.

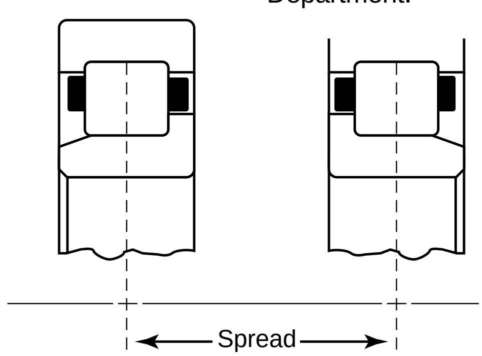


Figure 7

DIRECT MOUNTING

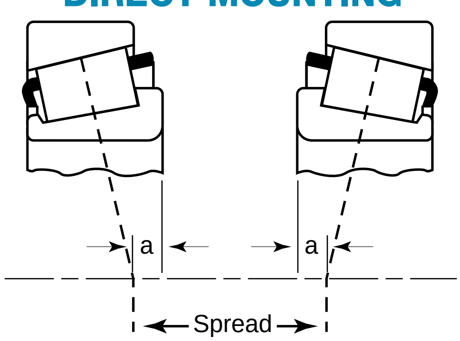


Figure 8

INDIRECT MOUNTING

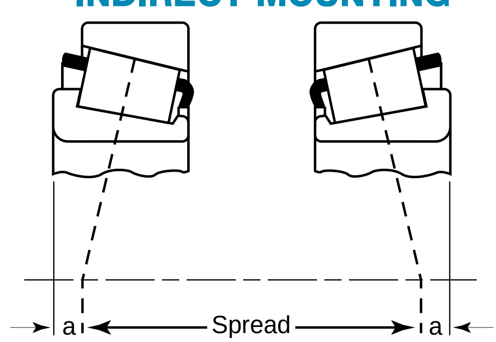


Figure 9

A SIMPLIFIED METHOD FOR FIGURING BEARING LOADS

The simplified method for solving bearing loads described below is merely a condensed or consolidated version of standard methods of basic mechanics. It makes full use of the basic laws or equilibrium, namely, for any system of forces;

Where: $\Sigma F = 0$
$\Sigma M = 0$
ΣF = Summation of forces
ΣM = Summatiion of moments about an arbitrary point

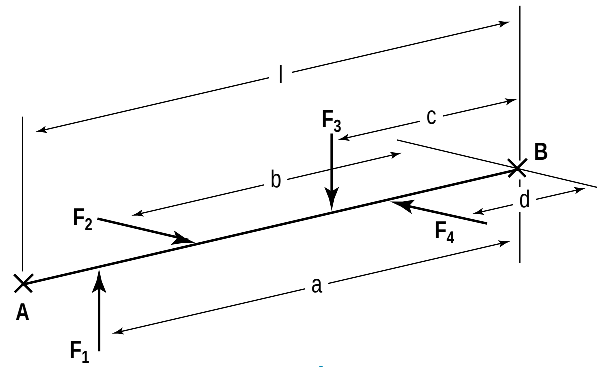
Combining these laws with the Pythagorean theorem, the required bearing loads are easily determined. It must be remembered that the applied loads and moments in conjunction with the bearing reactions create equilibrium for the system. The following rules provide an orderly procedure which will minimize the chance of error.

1. Break all forces into components that may be projected onto one of two convenient planes passing through the shaft centerline and at right angles to each other. These convenient planes will normally be horizontal and vertical and will, hereafter, be referred to as such.

2. The sign of the moment of a force about a point in its plane will be regarded as positive if the sense of rotation is counterclockwise and negative if the sense of rotation is clockwise.

3. Always use the right hand bearing as the moment-center.

4. Solve for the left bearing load components by taking moments of all the forces about the right hand bearing and DIVIDING THEIR ALGEBRAIC SUM BY THE BEARING SPREAD. Combine the equations for the horizontal and vertical components by the Pythagorean theorem and solve for the bearing load.



Example 1:

Vertical Component — Horizontal Component

$$R_A = \left[\left(\frac{\overbrace{-F_1 \times a + F_3 \times c}^{\pm V_A}}{l} \right)^2 + \left(\frac{\overbrace{F_2 \times b - F_4 \times d}^{\pm H_A}}{l} \right)^2 \right]^{1/2} \quad (17)$$

In any pair of bearings, the second bearing load (R_B) may be found by the summation of forces. This summation will include the components of R_A, remembering that the reaction of R_A must be used as the load on the shaft, hence, the load components of R_A must be multiplied by minus one.

$$R_B = \left[\left(-F_1 + F_3 \mp V_A\right)^2 + \left(F_2 - F_4 \mp H_A\right)^2 \right]^{1/2} \quad (18)$$

By locating equation 18 near equation 17, the equation for R_B may be set up by taking the load figures directly from the equation for R_A without further reference to the diagram.

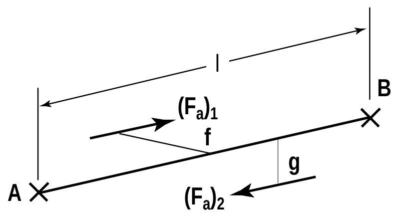
$$R_A = \left[\left(\frac{\overbrace{-F_1 \times a + F_3 \times c}^{\pm V_A}}{l} \right)^2 + \left(\frac{\overbrace{F_2 \times b - F_4 \times d}^{\pm H_A}}{l} \right)^2 \right]^{1/2} \quad (17)$$

$$R_B = \left[\left(-F_1 + F_3 \mp V_A\right)^2 + \left(F_2 - F_4 \mp H_A\right)^2 \right]^{1/2} \quad (18)$$

Note that the sign of the individual forces is the same for R_B as it was in R_A while the sings for the components V_A and H_A have been reversed as previously explained.

SPECIAL CASES

1. **Thrust Forces.** Thrust forces are reduced to components in the two specified planes and moments are taken about the right hand bearing to solve R_A. When solving for the second bearing load, it must be remembered that the thrust components are parallel to the axis of the shaft and, therefore, do not enter into the summation of the horizontal or vertical forces.

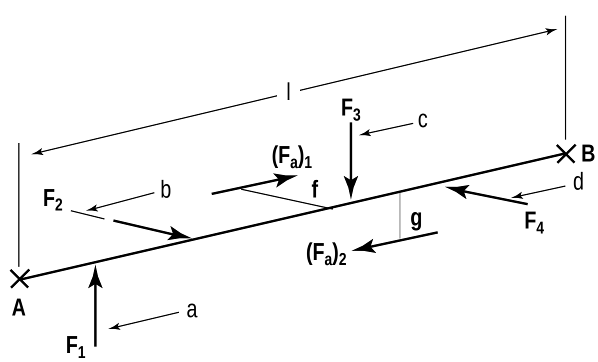


Example 2:

$$R_A = \left[\left(\overbrace{\frac{-(F_a)_2 \times g}{l}}^{V_A} \right)^2 + \left(\overbrace{\frac{-(F_a)_1 \times f}{l}}^{H_A} \right)^2 \right]^{1/2} \quad (19)$$

$$R_B = \left[(+V_A)^2 + (+H_A)^2 \right]^{1/2} \quad (20)$$

Combine examples 1 and 2.

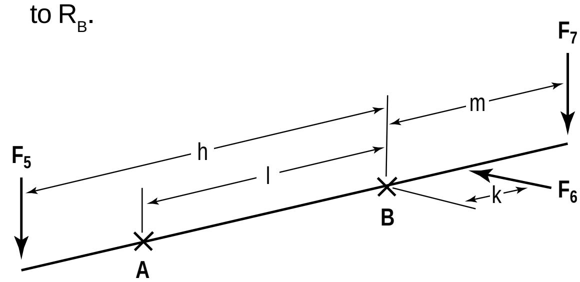


$$R_A = \left[\left(\overbrace{\frac{-F_1 \times a + F_3 \times c - (F_a)_2 \times g}{l}}^{\pm V_A} \right)^2 + \left(\overbrace{\frac{F_2 \times b - F_4 \times d - (F_a)_1 \times f}{l}}^{\pm H_A} \right)^2 \right]^{1/2} \quad (21)$$

$$R_B = \left[(-F_1 + F_3 \mp V_A)^2 + (F_2 - F_4 \mp H_A)^2 \right]^{1/2} \quad (22)$$

2. **Overhanging Forces.** Definition: An overhanging force is any force so located (1) as to not be between the two support points, and (2) as to not have one of the supports between it and the moment-center. Thus, when the right hand support is used as the moment-center, all forces to the right of the right hand support (moment-center) are overhanging forces.

Rule: When carrying the value of the overhanging force down to solve for R_B, the sign must be reversed. This is obvious from the fact that a shaft loading consisting of only an overhanging force, the two support reactions are of the opposite sense. It may be necessary to refer to a diagram here to avoid missing an overhanging force with reference to R_B.

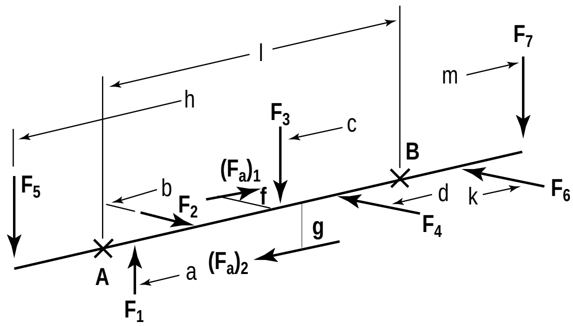


Example 3:

$$R_A = \left[\left(\overbrace{\frac{F_5 \times h - F_7 \times m}{l}}^{\pm V_A} \right)^2 + \left(\overbrace{\frac{F_6 \times k}{l}}^{\pm H_A} \right)^2 \right]^{1/2} \quad (23)$$

$$R_B = \left[(F_5 + F_7 \mp V_A)^2 + (-F_6 \mp H_A)^2 \right]^{1/2} \quad (24)$$

Note: By definition, F_6 and F_7 are overhanging forces and therefore require a change in sign in solving for R_B by summation of forces. Also, by definition, F_5 is **not** considered an overhanging force.



Combine examples 1, 2, and 3.

$$R_A = \left[\left(\overbrace{\frac{-F_1 \times a + F_3 \times c + F_5 \times h - (F_a)_2 \times g - F_7 \times m}{l}}^{\pm V_A} \right)^2 + \left(\overbrace{\frac{F_2 \times b - F_4 \times d - (F_a)_1 \times f + F_6 \times k}{l}}^{\pm H_A} \right)^2 \right]^{1/2} \quad (25)$$

$$R_B = \left[\left(-F_1 + F_3 + F_5 + F_7 \mp V_A \right)^2 + \left(F_2 - F_4 - F_6 \mp H_A \right)^2 \right]^{1/2} \quad (26)$$

SUGGESTIONS:

1. If the overhanging forces are always located at the end of each component in the equation, the possibility of overlooking them and the accompanying sign change will be reduced.

2. It will be much easier to learn one set of rules and always use the right hand support as the moment-center; however, the left hand support may be used if it is necessary. When using the left hand support as the moment-center, the signs for clockwise and counterclockwise rotation must be reversed. All other rules remain the same. Be sure to follow the strict definition of an overhanging force.

COMBINED LOADING EQUATIONS

Bearings are frequently required to support a combination of radial and thrust loads. In order to calculate the bearing life under such conditions, it is necessary to calculate an Equivalent Radial Load. The theoretical bearing life under combined radial and thrust loading conditions will be the same as that which would be expected under a pure radial load equal to the Equivalent Radial Load.

Cylindrical roller bearings with opposed solid ribs on the inner and outer rings will support light to moderate thrust loads. The maximum thrust load that a cylindrical roller bearing will support is defined later in this section. Field experience and laboratory tests have proven that as long as the applied thrust load is less than the applied radial load and less than the limiting thrust rating, the fatigue life of the bearing will not be adversely affected. Therefore, the fatigue life of a cylindrical roller bearing under such combined loading conditions will be equivalent to the life under the applied radial load. The Equivalent Radial Load concept is not applicable to cylindrical roller bearings.

Tapered roller bearings, due to their basic design, generate a thrust reaction when subjected to a radial load. The magnitude of this thrust reaction is a function of the load, the included cup angle, and the size of the load zone within the bearing. For convenience in load and life calculations, a "K" factor has been assigned to each tapered bearing series. This factor is defined as:

$$K = 0.389 \cot \alpha \quad (27)$$

$$\text{or} \quad K = \frac{\text{Radial Rating}}{\text{Thrust Rating}} = \frac{C(90)}{CA(90)} \quad (28)$$

$$\text{Where } \alpha = 1/2 \text{ included cup angle}$$

When the load on bearing (A) is pure radial (R_A) and the load zone within the bearing is 180° or less, the approximate thrust reaction (TR_A) is:

$$TR_A = \frac{0.47\ R_A}{K_A} \quad (29)$$

When the load zone on bearing (B) approaches 360° due to a combined radial load (R_B) and an external thrust load, its approximate thrust reaction is:

$$TR_B = \frac{0.60\ R_B}{K_B} \quad (30)$$

These thrust reactions are a critical part of the Equivalent Radial Load equations for tapered roller bearings.

The general AFBMA equation for the equivalent radial load is:

$$P = X\,F_r + Y\,F_a \qquad (31)$$

where P = Equivalent radial load

F_r = Applied radial load

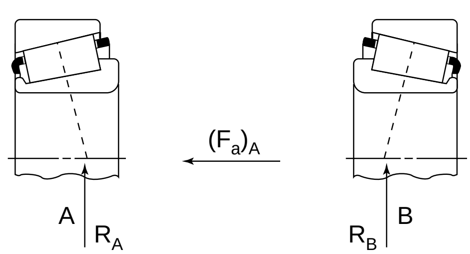
F_a = Applied thrust load

X = Radial load factor

Y = Thrust load factor

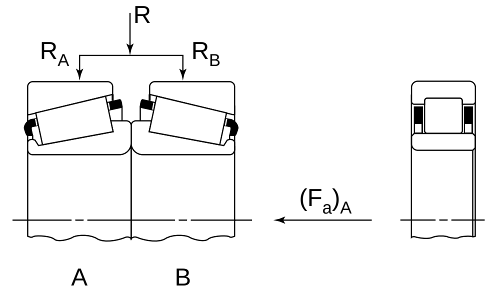
In the calculation of the equivalent radial load for a tapered bearing, the algebraic sum of all external thrust loads and the thrust reactions of the bearings must be considered. All factors are automatically included in the Equivalent Radial Load formulas shown in Table II. Note, when the calculated Equivalent Radial Load is less than the applied radial load, the radial load alone is used to estimate the bearing life.

TABLE II
EQUIVALENT RADIAL LOAD FORMULAS
SINGLE ROW MOUNTING

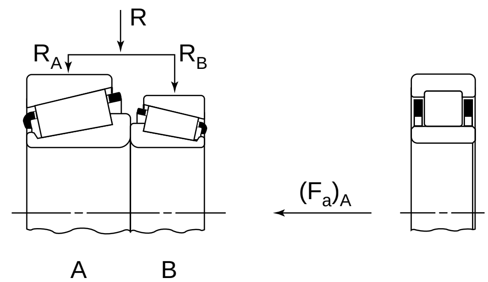


Thrust Condition	Equivalent Radial Load
$\frac{0.47R_A}{K_A} < \frac{0.47R_B}{K_B} + (F_a)_A$	$P_A = 0.40R_A + K_A\left(\frac{0.47R_B}{K_B} + (F_a)_A\right)$ $P_B = R_B$
$\frac{0.47R_A}{K_A} > \frac{0.47R_B}{K_B} + (F_a)_A$	$P_A = R_A$ $P_B = 0.40R_B + K_B\left(\frac{0.47R_A}{K_A} - (F_a)_A\right)$

MOUNTING OF TWO ROW ASSEMBLY



Thrust Condition	Two Row Identical Series
$(F_a)_A < \frac{0.6R}{K_A}$	$P_A = \frac{R}{2} + 0.83\,K_A(F_a)_A$ $P_B = \frac{R}{2} - 0.83\,K_A(F_a)_A$
$(F_a)_A > \frac{0.6R}{K_A}$	$P_A = 0.4R + K_A(F_a)_A$ $P_B = 0$



Thrust Condition	Two Row Dissimilar Series
$(F_a)_A < \frac{0.6R}{K_A}$	$P_A = \frac{K_A}{K_A + K_B}\left(R + 1.67\,K_B(F_a)_A\right)$ $P_B = \frac{K_B}{K_A + K_B}\left(R - 1.67\,K_A(F_a)_A\right)$
$(F_a)_A > \frac{0.6R}{K_A}$	$P_A = 0.4R + K_A(F_a)_A$ $P_B = 0$

where

R = Total radial load—lbs.

R_A = Radial load, brg. A—lbs.

R_B = Radial load, brg. B—lbs.

$(F_a)_A$ = External thrust on brg. A*—lbs.

K_A = Factor K brg. A

K_B = Factor K brg. B

P_A = Equivalent radial load, brg. A—lbs.

P_B = Equivalent radial load, brg. B—lbs.

* When there are no external thrust loads $F_a = 0$ in equations above.

BEARING LIFE CALCULATIONS

The previous sections have established the methods of determining the bearing loads and speeds for various applications. The next step in the bearing selection process is to evaluate the expected bearing life so that it may be compared to the design bogie. Traditionally, the 90% reliability level has been used to evaluate the fatigue life of a bearing in a specific application. The basic life equations are:

$$L_{10} = \left(\frac{C\,(90)}{P}\right)^{10/3} \times 90 \times 10^{6} \text{ Revolutions of either ring} \qquad (32)$$

$$\text{or } L_{10} = \left(\frac{C\,(90)}{P}\right)^{10/3} \times \frac{500}{S} \times 3000 \text{ Hours} \qquad (33)$$

where: C(90) = Dynamic load rating
P = Equivalent radial load
S = rpm

These equations are valid for either inner or outer ring rotation. If both rings are rotating S is equal to the algebraic difference of the rpm of the inner and outer rings.

Classical subsurface fatigue is not the only factor limiting bearing life. Modern technology provides a basis for evaluating the effects on fatigue life of alternate bearing materials, lubrication, misalignment, and the size of the load zone within the installed bearing. Also, some applications may require a more critical reliability factor rather than the 90% level. To take these factors into account NTN-Bower has developed the following adjusted life equation:

$$L_n = a_1 \times a_2 \times a_3 \times a_4 \times a_5 \times L_{10} \qquad (34)$$

where L_{10} = Value from (32) or (33)
RL = % Reliability level
n = 100 - RL
a_1 = Reliability factor
a_2 = Material factor
a_3 = Lubrication factor
a_4 = Misalignment factor
a_5 = Load zone factor

Each of these factors is defined below:

a_1—Reliability Factor

As previously defined, normal industry practice and the radial load ratings in this catalog are based on the 90% reliability level. In some applications, a more stringent reliability level may be required. As defined by AFBMA, the reliability factor a_1 is:

$$a_1 = 4.48 \times \left[\text{In}\,\frac{100}{R}\right]^{2/3} \qquad (35)$$

For convenience, specific values are shown in Table III.

TABLE III

Reliability Level %	L_n	Life Adjustment Factor a_1
90	L_{10}	1.00
95	L_5	0.62
96	L_4	0.53
97	L_3	0.44
98	L_2	0.33
99	L_1	0.21

a_2—Material Factor

Most NTN-Bower bearings are manufactured from carburizing grades of alloy steels processed to meet exacting bearing quality standards. A few special products utilize alternate materials specifically selected for their intended applications. All load ratings published in this catalog reflect the use of bearing quality alloy steel. Therefore, the material factor, a_2, is equal to 1.0.

In some applications, it may not be possible to find a standard bearing with adequate fatigue life within the boundary restraints. To avoid the necessity of a redesign of the entire system, bearings manufactured from premium materials have longer fatigue life due to fewer and more widely separated non-metallic inclusions in the steel matrix which reduces the number and severity of possible fatigue initiation sites. Materials which have these properties include Consumable Electrode Vacuum Melt (CEVM) and Electro-Slag Remelt (ESR) steels. NTN has established material life adjustment factors, a_2, for these premium steels as shown in Table IV.

TABLE IV

Material	Life Adjustment Factor a_2
CEVM	2.0
ESR	2.0

a_3—Lubrication Factor

The lubricant selected for the application, the operating temperature, and the bearing load and speed combine to affect bearing life. When any of these deviate substantially from the base conditions, the expected bearing life can be adjusted by the lubrication life factor a_3. In general, higher viscosity lubricants, higher speeds, and lower temperatures yield an adjustment factor greater than 1.0 (a_3>1.0). Figures 10 through 13 are used to approximate the lubrication factor—a_3. This procedure is intended only to provide a ballpark figure for a_3. For a more exact determination of a_3, contact NTN Application Engineering Department.

a_4—Misalignment Factor

Although bearings should be perfectly aligned, some degree of misalignment is virtually always present in an application. A small degree of misalignment is allowed for in the bearing ratios. However, the factor, a_4, should be considered when misalignment exceeds a value of 0.001 radian. Misalignment is a measurement of the angle between the axis of rotation of the outer ring. Figure 14 is used to estimate the misalignment factor—a_4 for cylindrical and tapered roller bearings. For a more exact evaluation, contact NTN Application Engineering Department.

a_5—Load Distribution Factor

The distribution of load within a bearing is a function of mounted clearance, support stiffness, and the magnitude of the load. For a given application there exists an optimum mounted internal clearance to maximize a bearing's fatigue life. The proper selection of the fitting practice for cylindrical roller bearings with preset radial clearance is critical to bearing performance. For adjustable tapered roller bearings, the opportunity exists to optimize bearing performance through adjustment methods.

The technique used to estimate the influence of internal clearance on fatigue life involves the computer analysis of many variables. The bearing user should consult the NTN Application Engineering Department for evaluation of the load distribution factor.

a_3—Lubrication Factor

Figure 10

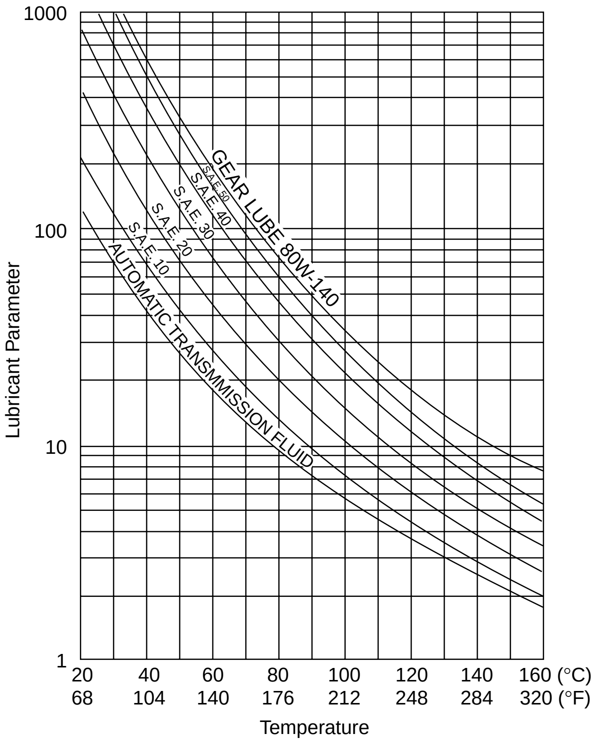


Figure 11

Cylindrical Bearing Film Thickness Chart

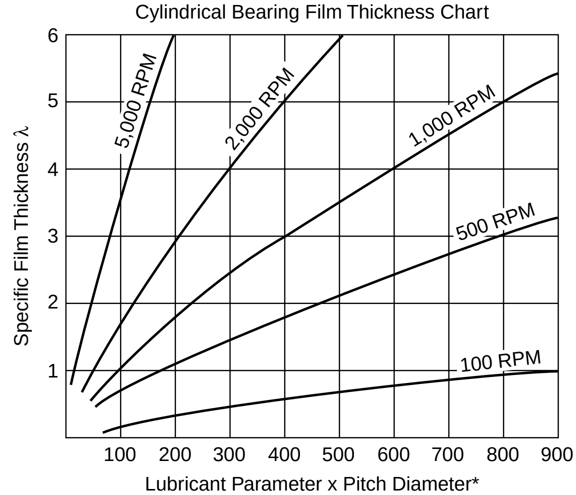


Figure 12

Tapered Bearing Film Thickness Chart

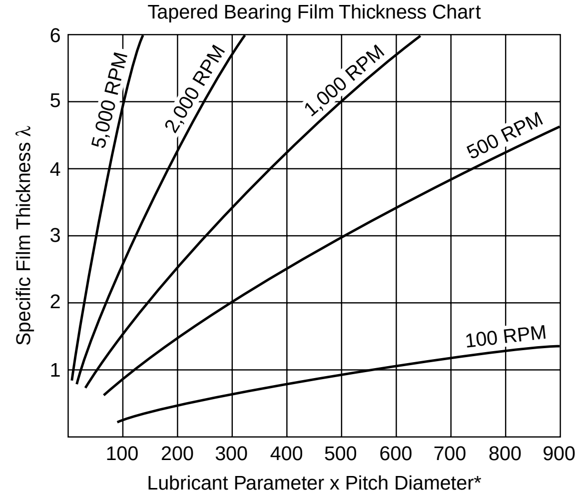
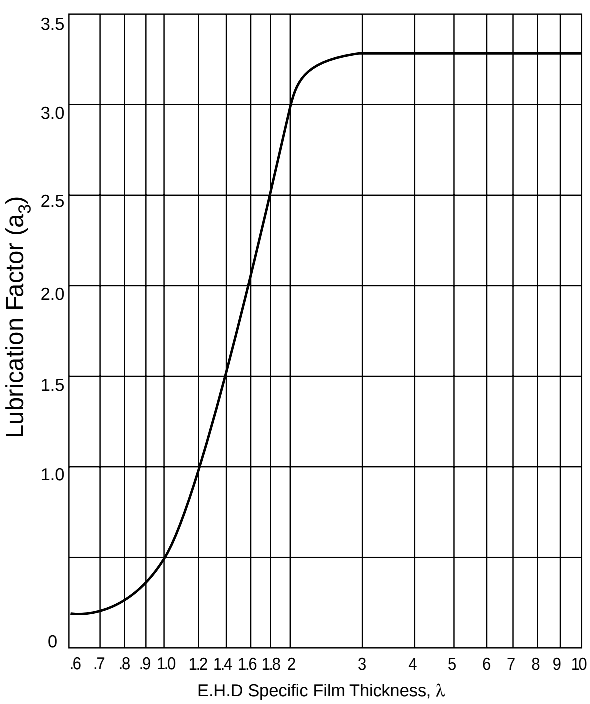


Figure 13

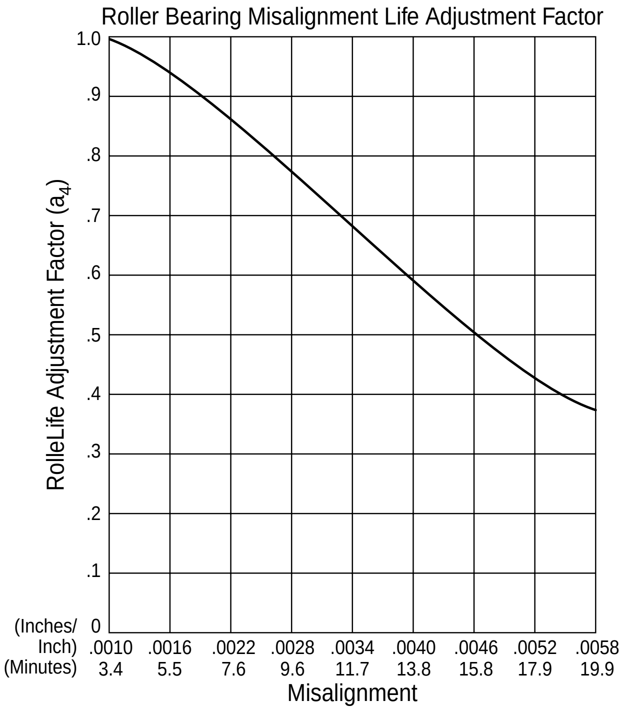


INSTRUCTIONS

1. Determine Lubricant Parameter according to temperature and type of Lubricant from Figure 10.
2. Multiply Lubricant Parameter by Bearing Pitch Diameter*.
3. Determine Specific Film Thickness "λ" from Figure 11 or 12.
4. Determine Lubrication Factor "a_3" from Figure 13.

$$*\text{Pitch Diameter (in.)} = \frac{\text{Bore Diameter} + \text{Outside Diameter}}{2}$$

Figure 14



WEIGHTED LIFE EQUATION

Bearing selection is sometimes based on life expectancy at maximum load and speed requirements. However, in some applications, the load and/or speed may vary at different time intervals. Therefore, a more economical bearing selection can be considered if these variations are evaluated to determine a weighted life for the bearing.

• To determine a weighted bearing L10 life in hours where the life at various conditions has been determined and a work schedule is known, use equation (36).

$$L_{WT} = \frac{1}{\frac{T_1}{L_{10_1}} + \frac{T_2}{L_{10_2}} + \text{L} + \frac{T_n}{L_{10_n}}}$$

$$L_{10} = \text{Life in Hours}$$

$$T_1, T_2, \text{L } T_n = \text{Time in \% of Total Time occurring during a loading cycle}$$

$$L_{WT} = \text{Weighted } L_{10} \text{ Life}$$

Example:

Given: Selected bearing has C(90) = 7225 lbs for rear countershaft position on five speed truck transmission. Operating schedule tabulated above.

Truck Operating Schedule

Gear	Load(P) lbs	Speed(S) rpm	Time(T) %	Life(L_{10}) hrs
1st	16190	100	3	1019
2nd	8550	400	5	2139
3rd	5850	900	30	3369
4th	3840	1200	42	10279
5th	2880	1500	20	21453

Problem: Determine weighted L_{10} life of selected bearing

$$L_{WT} = \frac{1}{\frac{.03}{1019} + \frac{.05}{2139} + \frac{.30}{3369} + \frac{.42}{10279} + \frac{.20}{21453}}$$

$$= 5207 \text{ hrs.}$$

THRUST RATING OF CYLINDRICAL ROLLER BEARINGS

Cylindrical roller bearings with opposed integral ribs on the inner and outer rings can support light to moderate thrust loads. The mechanism for supporting the thrust load in a cylindrical roller bearing is different from that in any other type of rolling bearing. In a ball bearing, the thrust load, as well as the radial load, is carried through the rolling contact between the balls and the raceways. In a tapered roller bearing, the major portion of the thrust load is carried on the rolling contact between the O.D. of the rollers and the raceways and the balance of sliding contact of the spherical head against the large cone flange. The cylindrical roller bearing can only support thrust loads on the ends of the rollers in a sliding contact with the raceway ribs, thus limiting thrust load carrying capabilities.

Several important factors must be considered when using cylindrical roller bearings in thrust applications. The thrust reactions at the diametrically opposed raceway ribs create a radial overturning moment on the roller and the sliding action creates a cirumferential skewing moment. To overcome the radial moment and stabilize the roller, the applied radial load must be greater than the thrust load. The longer rollers in wide series cylindrical roller bearings are more adversely affected by the skewing moment and, therefore, are more restricted in thrust capabilities. The shaft alignment must be within 0.0001 radian of the true position to obtain equal load sharing between the rollers. Because of the sliding action, the lubricant must provide an adequate film between the roller ends and the raceway ribs; high viscosity oil is preferred.

TABLE VI

Bearing Type	(PD x S)*	
Cylindrical Roller Bearings	**Narrow Series**	**Wide Series**
X-Bar Cage	450,000	350,000
Fibron Cage	450,000	350,000
One piece steel Cage	400,000	300,000
Composite steel Cage	250,000	200,000
Full complement	200,000	150,000
Tapered Roller Bearings		
K > 1.0	400,000	
K < 1.09	300,000	

$$*PD = \text{Bearing pitch diameter} = \frac{\text{Bore} + \text{O.D.}}{2}$$

$$S = \text{Speed in rpm}$$

For tapered roller bearings use the maximum bore and minimum O.D. available in the series.

LIMITING SPEEDS

Because of the many factors involved in determining the speed capabilities of a rolling bearing, it is impossible to develop a simple formula to establish an exact value for the limiting speed. Besides the precision of the bearing itself, the magnitude and direction of the load, the type of cage, the type of lubricant and lubrication system, the rate of heat dissipation, the alignment, the mounting practice, and the balance of the rotating components all play a significant role. Therefore, only very general guidelines may be given on this subject.

The bearing industry has traditionally used guidelines based on the DN value (bore mm x rpm). This approach negates the radial section of the bearing and the cup angle in tapered roller bearings. A better approximation may be obtained by using the bearing pitch diameter instead of the bore. Table VI may be used as a general guideline to establish a limiting speed. Since each application must be evaluated on its own merits, it is recommended the NTN Application Engineering Department be consulted when the speed approaches the limiting value.

HEAVILY LOADED APPLICATIONS

Laboratory tests and field experience have proven that the life-load exponent is not constant in heavily loaded applications. As stress levels increase above a specific value, the exponent increases above the 10/3 for roller bearings. This phenomenon is due to greater sensitivity of the steel to the higher stress level. The evaluation is quite complex and must be processed with a computer program since it is dependent on load, bearing geometry, and load zone. When standard calculations indicate a life less than 10,000,000 cycles, the application should be reviewed with the NTN Application Engineering Department.

EFFECTS OF FITTING PRACTICE

Cylindrical roller bearings are manufactured with a preset amount of radial clearance. They are available in two styles, the standard series and the “A” series. The standard series is designed to be installed with a press fit on one ring and a tap fit on the other as defined in the cylindrical roller bearing fitting practice section of this catalog, pages 80-97. The “A” series is designed for a press fit on the inner ring and a heavy press fit on the outer ring which are required for heavy duty applications.

The press fit of either the inner ring or the outer ring reduces the radial clearance within the bearing. This reduction in clearance has been compensated for at the time of bearing manufacture. Therefore, it is essential that the recommended fitting practices be adhered to to assure that the bearing will operate with the proper installed clearance.

The inner ring will expand according to equation (38) for the general case.

$$\delta_i = \frac{p_i A}{E_1}\left[\frac{2 \times B^2}{A^2 - B^2}\right] \tag{38}$$

where
- δ_i = Expansion of inner ring raceway diameter (in)
- p_i = Radial contact pressure between inner ring and shaft (psi)
- A = Inner ring raceway diameter (in)
- B = Inner ring bore (in)
- E_1 = Inner ring modulus of elasticity = 29 x 10^6 psi

For a solid steel shaft equation (38) reduces to:

$$\delta_i = \frac{B}{A}(IF)_i \tag{39}$$

The outer ring will contract according to equation (40) for the general case.

$$\delta_o = \frac{-p_o C}{E_1}\left[\frac{2 \times D^2}{D^2 - C^2}\right] \tag{40}$$

where
- δ_o = Contraction of outer ring raceway (in)
- p_o = Radial contact pressure between outer ring and housing (psi)
- C = Outer ring raceway diameter (in)
- D = Outer ring O.D. (in)
- E_1 = Outer ring modulus of elasticity = 29 x 10^6 psi

For massive steel housing equation (40) reduces to

$$\delta_o = \frac{-C}{D}(IF)_o \tag{41}$$

For the general case, p_i and p_o may be solved for from the following equations, respectively:

$$(IF)_i = \frac{p_i B}{E_1}\left[\frac{A^2 + B^2}{A^2 - B^2} + \nu_1\right] + \frac{p_i B}{E_2}\left[\frac{B^2 + J^2}{B^2 - J^2} - \nu_2\right] \tag{42}$$

$$(IF)_o = \frac{p_o D}{E_1}\left[\frac{D^2 + C^2}{D^2 - C^2} - \nu_1\right] + \frac{p_o D}{E_3}\left[\frac{H^2 + D^2}{H^2 - D^2} + \nu_3\right] \tag{43}$$

where
- $(IF)_i$ = Interference fit of inner ring on shaft (in)
- $(IF)_o$ = Interference fit of outer ring in housing (in)
- ν_1 = Poisson's ratio for bearing rings = 0.27
- E_2 = Modulus of elasticity for shaft (psi)
- ν_2 = Poisson's ratio for shaft
- E_3 = Modulus of elasticity for housing (psi)
- ν_3 = Poisson's ratio for housing
- A = Inner ring raceway
- B = Inner ring bore
- C = Outer ring raceway diameter
- D = Outer ring O.D.
- J = Hollow shaft bore
- H = Housing O.D.

Tapered roller bearings have a more complex reaction to interference fits. Not only do the bearing raceways change in a radial direction, but, due to the tapered relationship of the raceways, there is also an expansion of bearing width which may effect the bearing setting. Please consult NTN Application Engineering Department for further information.

Lubrication

The following information on lubrication is intended only as a general guide. Due to the complexity of the subject, a qualified lubrication engineer should be consulted for recommendations on specific applications.

To obtain the full, calculated life of a bearing in an application, it is essential to select an adequate lubricant viscosity and method of lubrication.

The necessary data and formula to adjust bearing life for oil film thickness, based on the Elastohydrodynamic Theory (EHD), is provided in the “Bearing Selection” section under “Life Adjustment Factors” on page 21. Bearing life adjustment evaluation for grease lubrication is not given since other factors must be considered, including bearing load, humidity conditions, service life required and frequency of re-lubrication.

Bearing lubricants basically are used to:

- Provide a minimum lubricant film thickness that will separate the contacting surfaces at bearing operating temperature and speed
- Reduce friction and thus prevent wear
- Dissipate heat generated within the bearing
- Protect the contacting surfaces from corrosion within the bearing
- Remove or seal out foreign material from the bearing

To select an adequate bearing lubricant, it is necessary to be familiar with the environment in which the bearing will operate. Lubricant selection is influenced by:

- Bearing operating temperatures
- Bearing operating speeds
- Lubrication requirements of related components
- Compatibility with sealing devices
- Method and amount of lubrication required for the bearing

OIL VS. GREASE

Lubricants for roller bearings in commercial applications are of two basic types, oil or grease. While oil is the preferred lubricant because it has the desirable characteristics of a fluid, both have their advantages and limitations:

Oil

- Suitable for all speeds—but must be used for extremely high speeds
- For elevated temperatures—where the oil is circulated to cool the bearing
- For extremely low temperatures
- To provide a clean, filtered environment
- For a closed lubrication system—where related components require lubrication in addition to the bearings
- For critical applications—where the quantity of the lubricant must be controlled
- For more positive feeding of lubricant to heavily loaded contact surfaces
- For low running torque condition use an oil mist lubrication system

Grease

- For extremely low to moderate speeds
- For low to moderate loads
- For moderate temperatures
- As an aid in excluding severe contamination because of its consistency
- For less complicated lubrication systems
- For simple, positive lubrication as in a self-contained, sealed, pre-lubricated unit
- For a simplified housing design
- For ease of sealing

OIL

Oil, the preferred lubricant for roller bearings, consists of either petroleum fluids refined from crude oil or synthetic fluids produced by chemical synthesis. Most commercial lubricating oils are available with an additive or combination of additives to meet various environmental or operating conditions. Common types of additives and their primary functions are:

- **Oxidation inhibitor:**

  Retards oil deterioration and formation of sludge, carbon and varnish

- **Rust inhibitor:**

  Protects lubricated surfaces from rust and corrosion

- **Detergent—dispersant:**

  Reduces and controls degradation products and helps maintain cleanliness of lubricated surfaces

- **Defoaming agent:**

  Prevents formation of air bubbles

- **Extreme Pressure (EP) additive:**

  Prevents high friction, wear or scoring under various conditions of sliding or marginal lubrication

- **Viscosity Index (VI) improver:**

  Reduces the affect of temperature changes on oil viscosity

- **Pour—Point Depressant:**

  Lowers the solidification point of oil

The above list is not meant to imply that all or any of these specific additives mentioned are always required. Proper use of additives is fundamental to obtaining long and satisfactory roller bearing service. It is recommended that a reputable oil company be consulted for the specific operating conditions under consideration. Special attention should be given to stability over the operating temperature range of the oil and to possible chemical changes in the oil from storage or service conditions.

The oil lubrication systems most commonly used in commercial applications are:

- **Splash Feed System.** In many transmission and gear box systems, sufficient splash is generated by the gears to lubricate the bearings. However, if excessive contaminants are generated by the gears or if the system cannot be cleaned frequently, contaminants may cause serious damage to the bearings. It is recommended that magnetic drain plugs be used in these systems.

- **Oil Circulating System.** This system is used for the same speed ranges as the Oil Drop Feed System. However, it is designed for use when excessive heat or contamination must be removed from the bearing. To meet the contamination problem, a suitable filter should be incorporated into the system.

- **Oil Mist System.** This system is recommended for use when the speeds are extremely high, provided the air which atomizes the oil is clean and dry.

- **Constant Oil Level.** In low and medium speed applications, a constant oil level system is used. The oil level should immerse approximately fifty percent of the lowest roller when the bearing is stationary.

- **Drop Feed System.** When the speed is too high for the oil level system, the drop feed system is often used. In this case, the oil is fed into the bearing in droplet form. It moves through the bearing and out the drain, which is located on the side opposite the oil supply. It is not recommended where contamination is a problem or where good cooling is required.

GREASE

Greases in general use for roller bearings are composed of oil thickened with a metallic soap base, in various proportions, to form a desired consistency. The oil is of a specified viscosity no lower than 70 SUS (Saybolt Universal Seconds) at 100° F. The soap base type may be sodium (soda), calcium (lime), lithium, calcium complex, aluminum complex or various synthetic and non-soap base types. Properties of some of the soap base types are:

- **Sodium—**good stability at the higher permissible speed and temperature ranges; not water resistant
- **Calcium—**inexpensive; good water resistance; limited to temperatures under 150° F.
- **Lithium—**generally stable at higher temperatures, good water resistance, good internal cohesion, “multi-purpose”.

Sodium and mixed sodium-calcium soap greases are considered good “general purpose” lubricants. Calcium, lithium and non-soap greases are used where water resistance is required.

Synthetic oil greases are more expensive than petroleum oil greases and are used where it is desirable to broaden the temperature range beyond that of petroleum base greases.

- Silicone oil greases are used for both high and low temperature operation (-100° F to +450° F), but have a limited load carrying capacity
- Ester oil greases cover a wide temperature range (-100° to +350° F)
- Di-ester oil greases cover the low temperature range to -65° F

The grease consistency at bearing operating temperature is an important factor in selecting a suitable grease. Its melting point should be considerably higher than the operating temperature. Roller bearing greases in general use are a NLGI #1 or #2 grade, multipurpose, with an ASTM worked penetration number between 265-340.

The following guide applies to general applications under normal loading at operating speeds of 100—1000 rpm. For heavy loads and low speeds, the advice of a lubrication engineer should be obtained.

GREASE TEMPERATURE GUIDE

Grease Grade	Operating Temperature
#0	Below 32° F
#1	32° F—150° F
#2	150° F—250° F

GREASE CONSISTENCY CLASS

Grease Grade	ASTM Worked Penetration @ 77° F	Description
#0	355—385	Very soft
#1	310—340	Soft
#2	265—295	Moderately firm

Grease churns when used in excessive quantities, resulting in excessive temperatures, separation of the grease components and breakdown in the lubricant. Generally, the cavity in which the bearing is mounted should be kept 1/2—1/3 full for normal speeds.

A suitable grease should remain mechanically and chemically stable at operating temperature. It should not thicken, harden, separate, or become acid or alkaline to any marked degree.

Re-lubrication intervals should be established based on the experience of similar applications. The recommended grease type should be used.

HANDLING AND INSTALLATION

Improper handling practices prior to and during installation can easily damage the quality and precision built into NTN-Bower roller bearings. Although a general set of rules cannot adequately cover all the ways that a roller bearing should be handled to prevent it from becoming unserviceable, certain essential precautions and care will minimize such damage.

Prior to shipment, NTN-Bower roller bearings are thoroughly cleaned, coated with a rust preventative, and carefully packaged for protection against contamination and oxidation. A positive effort should be made to keep the bearings in this condition prior to final assembly. The bearing package should be kept closed until ready for immediate installation. If it is necessary to unwrap the bearings before that time, they should be placed on a clean surface and covered with a lint free cloth. Prior to bearing installation, housings, shafts, and other adjacent parts should be wiped clean or washed. In addition, foundry sand should be completely removed from castings.

Roller bearings should be installed in an area where a clean atmosphere exists. In addition, it is imperative that assembly benches and tools be kept clean to prevent contaminants such as dust, grit and steel chips from entering the bearing. Contamination not only causes rough and noisy operation, but usually results in premature bearing fatigue. It is much easier to keep a bearing clean than it is to wash it clean enough for service.

New bearings must be cleaned prior to installation only if they become contaminated after being removed from their original package. Light spindle oils (less than SAE 10 Viscosity) or Stoddard solvents are recommended for washing purposes. It is recommended that chlorinated solvents not be used because of rust hazards associated with certain types. Compressed air may be used to blow out foreign matter. However, care must be taken not to free spin the bearing because permanent damage may result from dirt particles scoring the rolling surfaces. The compressed air must be filtered so that it is free from moisture, otherwise it could corrode the bearing surfaces.

The bearings must be carefully inspected after cleaning to make certain they are clean enough for use. They should then be coated with a rust preventative and installed immediately or wrapped in a grease proof paper and properly labeled for future identification.

The bearing mounting must be properly designed from a functional standpoint and must have correct shaft and housing fits and shoulder heights. In addition, the design should be such that the bearings and other components can be installed as easily as possible.

Proper assembly tools such as arbor presses, pullers, and sleeves will not only facilitate assembly, but will also avoid damage to the bearings. When a roller bearing is pressed on a shaft, the inner ring must be started squarely. A “cocked” ring may score the shaft and damage the bearing. The pressure must be applied directly on the ring being pressed, avoiding all pressure through the rollers. The bearing must not be tapped in place with direct blows on the bearing ring. The preferred practice is to place a sleeve between the bearing ring and the hammer and to tap the sleeve lightly all around. Hammers that shed chips should not be used as the chips may get into the bearing recesses.

Sometimes a bearing must be heated so that it can be more easily assembled on a shaft. A convenient method of doing this is to insert a heat source such as an electric light bulb in the bore of the bearing, keeping it there until the inner ring has expanded sufficiently. Another method is to heat the bearing in a bath of hot oil. The oil must be clean and the temperature should not exceed 250° F. Higher temperatures may cause the oil to decompose and the bearing to lose its proper hardness.

Further information regarding the care and installation of roller bearings may be obtained from the NTN Application Engineering Department.