3. CALCULATING THE PERFORMANCE OF YOUR BEARINGS

Summary of symbols used to determine applied loads

SYMBOL	DESCRIPTION	UNITS
b	Tooth length	mm, in
d_c	Distance between gear centers	mm, in
D_m	Mean diameter or effective working diameter of a sprocket, pulley, wheel, or tire	mm, in
D_m	Mean diameter or effective working diameter of gear (D_{mG}), pinion (D_{mP}), or worm (D_{mW})	mm, in
D_p	Pitch diameter of gear (D_{pG}) pinion (D_{pP}), or worm (D_{pW})	mm, in
f_B	Belt or chain pull factor	
F_a	Axial (thrust) force on gear (F_{aG}), pinion (F_{aP}), or worm (F_{aW})	N, lbf
F_b	Belt or chain pull	N, lbf
F_o	Centrifugal force	N, lbf
F_s	Separating force on gear (F_{sG}), pinion (F_{sP}), or worm (F_{sW})	N, lbf
F_t	Tangential force on gear (F_{tG}), pinion (F_{tP}), or worm (F_{tW})	N, lbf
F_{te}	Tractive effort on vehicle wheels	N, lbf
F_w	Force of unbalance	N, lbf
G	Gear, used as a subscript	
H	Power	kW, hp
L	Lead. Axial advance of a helix for one complete revolution	mm, in
M	Moment	N-m, lbf.in
m	Gearing ratio	
N	Number of teeth in gear (N_G), pinion (N_P), or sprocket (N_S)	
n	Rotational speed of gear (n_G), pinion (n_P) or worm (n_W)	rev/min
p	Pitch. Distance between similar equally spaced tooth surfaces along the pitch circle	mm, in
P	Pinion, used as a subscript	
r	Radius to center of mass	mm, in
T	Torque	N-m, lbf.in
V	Linear velocity or speed	km/h, mph
V_r	Rubbing or surface velocity	m/s, ft/min
W	Worm gear, used as a subscript	
γ (gamma)	(1) Bevel gearing - pitch angle of gear (γ_G) or pinion (γ_P)	degree
	(2) Hypoid gearing - face angle of pinion (γ_P) and root angle of gear (γ_G)	degree
η (eta)	Efficiency	decimal fraction
λ (lambda)	Worm gearing - lead angle	degree
μ (mu)	Coefficient of friction	
π (pi)	The ratio of the circumference of a circle to its diameter (π = 3.1416)	
ϕ (phi)	Normal tooth pressure angle for gear (ϕ_G) or pinion (ϕ_P)	degree
ϕ_x (phi$_x$)	Axial tooth pressure angle	degree
ψ (psi)	(1) Helical gearing - helix angle for gear (ψ_G) or pinion (ψ_P)	degree
	(2) Spiral bevel and hypoid gearing - spiral angle for gear (ψ_G) or pinion (ψ_P)	degree

TIMKEN

A. Determination of applied loads

1. Gearing

1.1. Spur gearing (Fig. 3-1)

Tangential force

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

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

Separating force

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

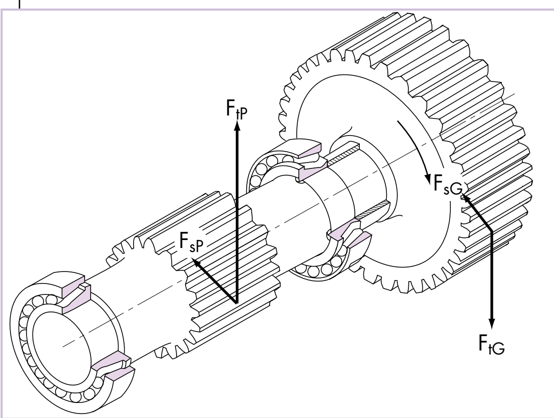


Fig. 3-1
Spur gearing.

1.2. Single helical gearing (Fig. 3-2)

Tangential force

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

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

Separating force

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

Thrust force

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

Note: for double helical (herringbone) gearing $F_{aG} = 0$

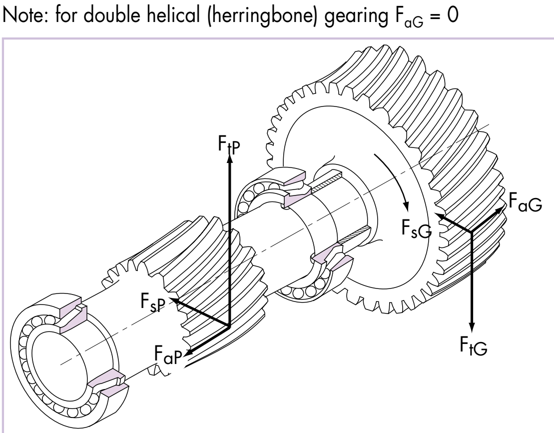


Fig. 3-2
Helical gearing.

1.3. Straight bevel and zerol gearing with zero degrees spiral (Fig. 3-4)

In straight bevel and zerol gearing, the gear forces tend to push the pinion and gear out of mesh such that the direction of the thrust and separating forces are always the same regardless of direction of rotation. (Fig. 3-3)

In calculating the tangential force, F_{tP} or F_{tG}, for bevel gearing, the pinion or gear mean diameter, D_{mP} or D_{mG}, is used instead of the pitch diameter, D_{pP} or D_{pG}. The mean diameter is calculated as follows:

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

or

$D_{mP} = D_{pP} - b \sin \gamma_P$

In straight bevel and zerol gearing

$F_{tP} = F_{tG}$

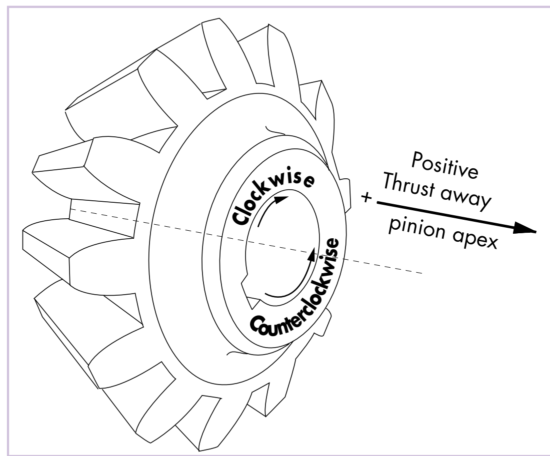


Fig. 3-3
Straight bevel and zerol gears - thrust and separating forces are always in same direction regardless of direction of rotation.

Pinion

Tangential force

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

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

Thrust force

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

Separating force

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

Gear

Tangential force

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

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

Thrust force

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

Separating force

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

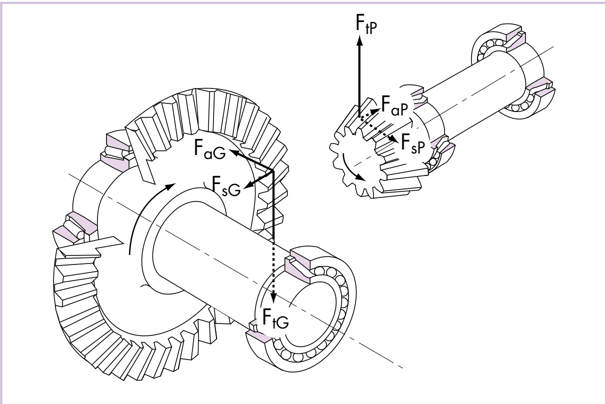


Fig. 3-4
Straight bevel gearing.

1.4. Spiral bevel and hypoid gearing (Fig. 3-6)

In spiral bevel and hypoid gearing, the direction of the thrust and separating forces depends upon spiral angle, hand of spiral, direction of rotation, and whether the gear is driving or driven (see Table 3-A). The hand of the spiral is determined by noting whether the tooth curvature on the near face of the gear (fig. 3-5) inclines to the left or right from the shaft axis. Direction of rotation is determined by viewing toward the gear or pinion apex.

In spiral bevel gearing

$F_{tP} = F_{tG}$

In hypoid gearing

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

Hypoid pinion effective working diameter

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

Tangential force

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

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

Hypoid gear effective working diameter

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

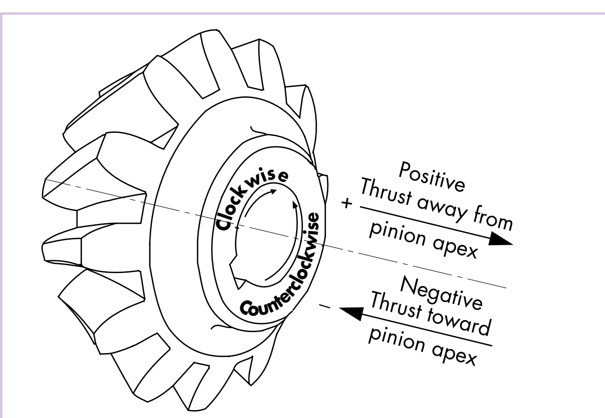


Fig. 3-5
Spiral bevel and hypoid gears - the direction of thrust and separating forces depends upon spiral angle, hand of spiral, direction of rotation, and whether the gear is driving or driven.

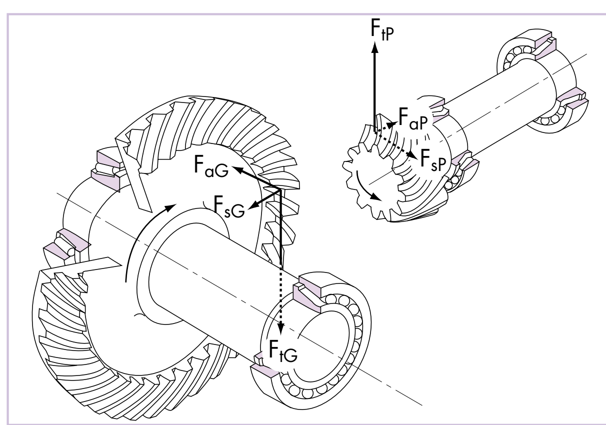


Fig. 3-6
Spiral bevel and hypoid gearing.

Driving member rotation	Thrust force	Separating force
Right hand spiral clockwise or Left hand spiral counterclockwise	Driving member $F_{aP} = \frac{F_{tP}}{\cos\psi_P}(\tan\phi_P \sin\gamma_P - \sin\psi_P \cos\gamma_P)$	Driving member $F_{sP} = \frac{F_{tP}}{\cos\psi_P}(\tan\phi_P \cos\gamma_P + \sin\psi_P \sin\gamma_P)$
	Driven member $F_{aG} = \frac{F_{tG}}{\cos\psi_G}(\tan\phi_G \sin\gamma_G + \sin\psi_G \cos\gamma_G)$	Driven member $F_{sG} = \frac{F_{tG}}{\cos\psi_G}(\tan\phi_G \cos\gamma_G - \sin\psi_G \sin\gamma_G)$
Right hand spiral counterclockwise or Left hand spiral clockwise	Driving member $F_{aP} = \frac{F_{tP}}{\cos\psi_P}(\tan\phi_P \sin\gamma_P + \sin\psi_P \cos\gamma_P)$	Driving member $F_{sP} = \frac{F_{tP}}{\cos\psi_P}(\tan\phi_P \cos\gamma_P - \sin\psi_P \sin\gamma_P)$
	Driven member $F_{aG} = \frac{F_{tG}}{\cos\psi_G}(\tan\phi_G \sin\gamma_G - \sin\psi_G \cos\gamma_G)$	Driven member $F_{sG} = \frac{F_{tG}}{\cos\psi_G}(\tan\phi_G \cos\gamma_G + \sin\psi_G \sin\gamma_G)$

Table 3A
Spiral bevel and hypoid gearing equations.

1.5. Straight worm gearing (Fig. 3-7)

Worm

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

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

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

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

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

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

Worm gear

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

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

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

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

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

Separating force $$F_{sG} = \frac{F_{tW}\sin\phi}{\cos\phi\sin\lambda + \mu\cos\lambda}$$

where:

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

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

Metric system

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

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

Inch system

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

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

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

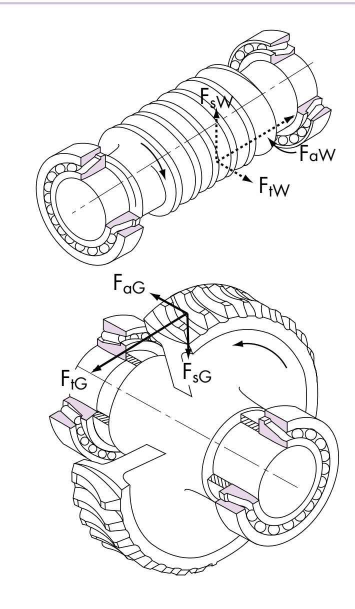


Fig. 3-7
Straight worm gearing.

1.6. Double enveloping worm gearing

Worm

Tangential force $F_{tW} = \dfrac{(1.91 \times 10^7)\,H}{D_{mW}\,n_W}$ (newtons)

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

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

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

Separating force $F_{sW} = \dfrac{0.98\,F_{tG}\tan\phi}{\cos\lambda}$

Worm gear

Tangential force $F_{tG} = \dfrac{(1.91 \times 10^7)\,H\,m\,\eta}{D_{pG}\,n_W}$ (newtons)

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

or

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

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

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

Thrust force $F_{aG} = \dfrac{(1.91 \times 10^7)\,H}{D_{mW}\,n_W}$ (newtons)

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

Separating force $F_{sG} = \dfrac{0.98\,F_{tG}\tan\phi}{\cos\lambda}$

where:

η = efficiency (refer to manufacturer's catalog)

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

Lead angle at center of worm

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

2. Belt and chain drive factors (Fig. 3-8)

Due to the variations of belt tightness as set by various operators, an exact equation relating total belt pull to tension F_1 on the tight side and tension F_2 on the slack side (fig. 3-8), is difficult to establish. The following equation and table 3-B may be used to estimate the total pull from various types of belt and pulley, and chain and sprocket designs:

$F_b = \dfrac{(1.91 \times 10^7)\,H\,f_B}{D_m\,n}$ (newtons)

$= \dfrac{(1.26 \times 10^5)\,H\,f_B}{D_m\,n}$ (pounds-force)

Standard roller chain sprocket mean diameter

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

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

Table 3-B
Belt or chain pull factor based on 180 degrees angle of wrap.

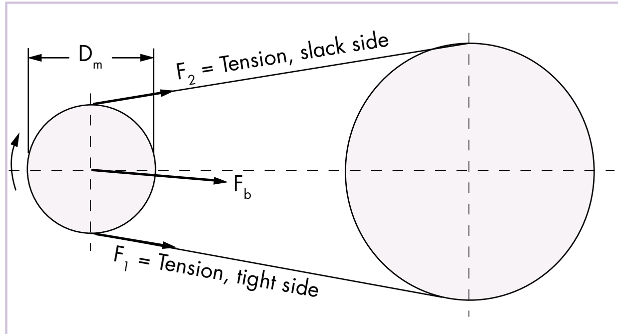


Fig. 3-8
Belt or chain drive.

3. Centrifugal force

Centrifugal force resulting from imbalance in a rotating member:

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

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

4. Shock loads

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

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

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

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

5. General formulas

5.1. Tractive effort and wheel speed

The relationships of tractive effort, power, wheel speed and vehicle speed are:

Metric system

$$H = \frac{F_{te}\ V}{3600} \quad \text{(kW)}$$

$$n = \frac{5300\ V}{D_m} \quad \text{(rev/min)}$$

Inch system

$$H = \frac{F_{te}\ V}{375} \quad \text{(hp)}$$

$$n = \frac{336\ V}{D_m} \quad \text{(rev/min)}$$

5.2. Torque to power relationship

Metric system

$$T = \frac{60\,000\ H}{2\pi\ n} \quad \text{(N-m)}$$

$$H = \frac{2\pi\ n\ \ T}{60\,000} \quad \text{(kW)}$$

Inch system

$$T = \frac{395\,877\ H}{2\pi\ n} \quad \text{(lbf.in)}$$

$$H = \frac{2\pi\ n\ T}{395\,877} \quad \text{(hp)}$$

6. Bearing reactions

6.1. Effective spread

When a load is applied to a tapered roller bearing, the internal forces at each roller body to cup contact act normal to the raceway (see Fig. 1-5, page 4). These forces have radial and axial components. With the exception of the special case of pure thrust loads, the cone and the shaft will experience moments imposed by the asymmetrical axial components of the forces on the rollers.
It can be demonstrated mathematically that if the shaft is modeled as being supported at its effective bearing center, rather than at its geometric bearing center, the bearing moment may be ignored when calculating radial loads on the bearing. Then only externally applied loads need to be considered, and moments are taken about the effective centers of the bearings to determine bearing loads or reactions.
Fig. 3-9 shows single-row bearings in a "direct" and "indirect" mounting configuration. The choice of whether to use direct or indirect mounting depends upon the application and duty.

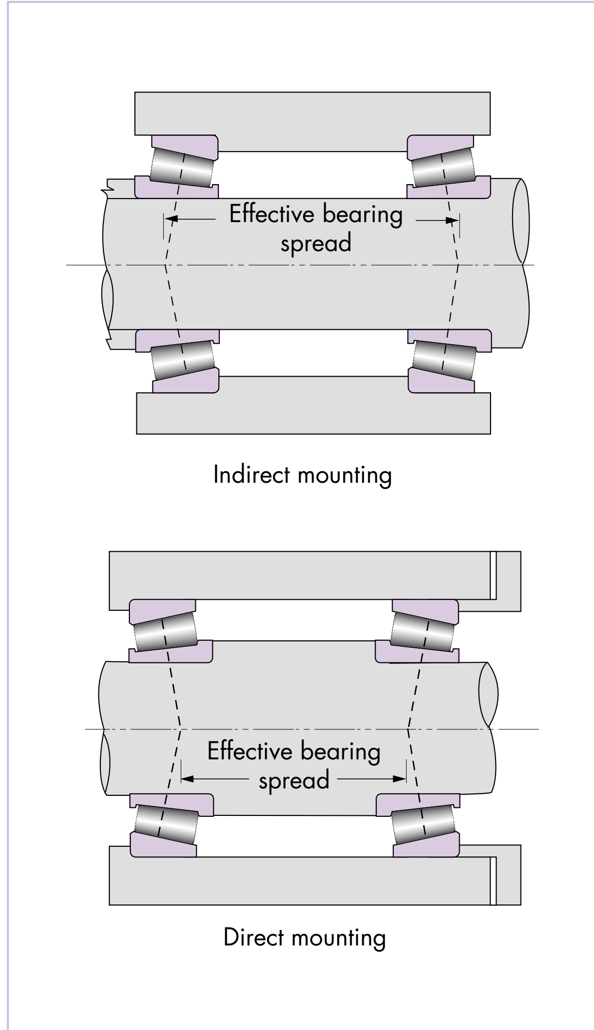


Fig. 3-9
Choice of mounting configuration for single-row bearings, showing position of effective load carrying centers.

6.2. Shaft on two supports

Simple beam equations are used to translate the externally applied forces on a shaft into bearing reactions acting at the bearing effective centers.
With two-row bearings, the geometric center of the bearing is considered to be the support point except where the thrust force is large enough to unload one row. Then the effective center of the loaded row is used as the point about which bearing load reactions are calculated. These approaches approximate the load distribution within a two-row bearing, assuming rigid shaft and housing. However, these are statically indeterminate problems in which shaft and support rigidity can significantly influence bearing loading and require the use of computer programs for solution.

6.3. Shaft on three or more supports

The equations of static equilibrium are insufficient to solve bearing reactions on a shaft having more than two supports. Such cases can be solved using computer programs if adequate information is available.
In such problems, the deflections of the shaft, bearings and housings affect the distribution of loads. Any variance in these parameters can significantly affect bearing reactions.

6.4. Calculation example

Symbols used in calculation examples

a_e	Effective bearing spread	mm, in
A, B, ...	Bearing position, used as subscripts	
c_1, c_2, ...	Linear distance (positive or negative)	mm, in
F	Applied force	N, lbf
F_r	Radial bearing load	N, lbf
h	Horizontal (used as subscript)	
H	Power	kW,hp
K	K-factor from bearing tables	
M	Moment	N-mm, lbf.in
v	Vertical (used as subscript)	
θ_1, θ_2, θ_3	Gear mesh angle relative to plane of reference defined in figure 3-10	degree

Bearing radial reactions - Shaft on two supports

Bearing radial loads are determined by:

1. Resolving forces applied to the shaft into horizontal and vertical components relative to a convenient reference plane.

2. Taking moments about the opposite support.

3. Combining the horizontal and vertical reactions at each support into one resultant load.

Shown are equations for the case of a shaft on two supports with gear forces F_t (tangential), F_s (separating), and F_a (thrust), an external radial load F, and an external moment M. The loads are applied at arbitrary angles (θ_1, θ_2, and θ_3) relative to the reference plane indicated in figure 3-10. Using the principle of superposition, the equations for vertical and horizontal reactions (F_{rv} and F_{rh}) can be expanded to include any number of gears, external forces or moments. Use signs as determined from gear force equation.

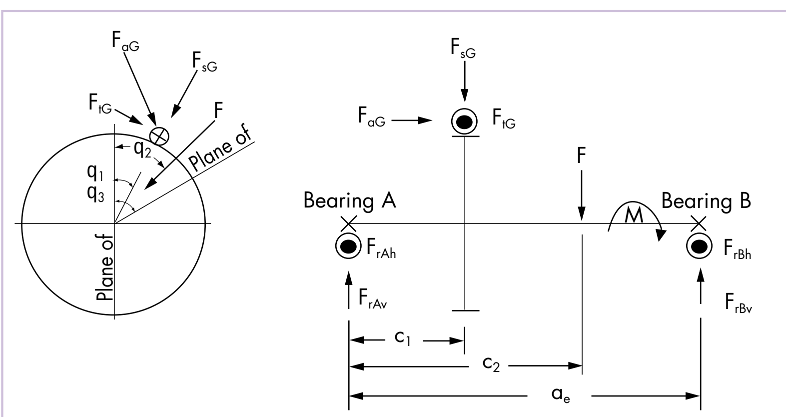


Fig. 3-10
Bearing radial reactions.

Vertical reaction component at bearing position B

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

Horizontal reaction component at bearing position B

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

Vertical reaction component at bearing position A

$$F_{rAv} = F_{sG}\cos\theta_1 + F_{tG}\sin\theta_1 + F\cos\theta_2 - F_{rBv}$$

Horizontal reaction component at bearing position A

$$F_{rAh} = F_{sG}\sin\theta_1 - F_{tG}\cos\theta_1 + F\sin\theta_2 - F_{rBh}$$

Resultant radial reaction

$$F_{rA} = \left(F_{rAv}^2 + F_{rAh}^2\right)^{1/2}$$

$$F_{rB} = \left(F_{rBv}^2 + F_{rBh}^2\right)^{1/2}$$

See page 62 for examples of bearing life calculation.

B. Bearing life

1. Dynamic conditions

1.1. Nominal or catalog life

1.1.1. Bearing life

Many different performance criteria dictate bearing selection. These include bearing fatigue life, rotational precision, power requirements, temperature limits, speed capabilities, sound, etc. This guide deals with bearing life related to material associated fatigue spalling.

Bearing failure mode may not be fatigue

There are other factors that limit bearing life if not specially considered in the initial design analysis, such as inadequate lubrication, improper mounting, poor sealing, extreme temperatures, high speeds, and unusual vibrations (translational and torsional). Also, proper handling and maintenance must be provided. These factors will not be addressed in this guide, but if present in any application, a Timken Company sales engineer or representative should be consulted.

Bearing life is defined here as the length of time, or the number of revolutions, until a fatigue spall of a specific size develops.

Since metal fatigue is a statistical phenomenon, the life of an individual bearing is impossible to predetermine precisely. Bearings that may appear to be identical can exhibit considerable life scatter when tested under identical conditions. Thus it is necessary to base life predictions on a statistical evaluation of a large number of bearings operating under similar conditions. The Weibull distribution function is commonly used to predict the life of a bearing at any given reliability level.

1.1.2. Rating life (L_{10})

Rating life, L_{10}, is the life that 90 percent of a group of identical bearings will complete or exceed before the area of fatigue spalling reaches a defined criterion. The L_{10} life is also associated with 90 percent reliability for a single bearing under a certain load.

The life of a properly applied and lubricated tapered roller bearing is normally reached after repeated stressing produces a fatigue spall of a specific size on one of the contacting surfaces. The limiting criterion for fatigue used in Timken laboratories is a spalled area of 6 mm^2 (0.01 in^2). This is an arbitrary designation and, depending upon the application, bearing useful life may extend considerably beyond this point. If a sample of apparently identical bearings is run under

1.1.3. Bearing life equations

The following factors also help to visualize the effects of load and speed on bearing life:

- Doubling the load reduces life to approximately one-tenth. Reducing the load by one-half increases life approximately ten times.
- Doubling the speed reduces hours of life by one-half. Reducing the speed by one-half doubles hours of life.

With increased emphasis on the relationship between the reference conditions and the actual environment in which the bearing operates in the machine, the traditional life equations have been expanded to include certain additional variables that affect bearing performance. Technology permits the quantitative evaluation of environmental differences, such as lubrication, load zone and alignment, in the form of various life adjustment factors. These factors, plus a factor for useful life, are considered in the bearing analysis and selection approach by The Timken Company.

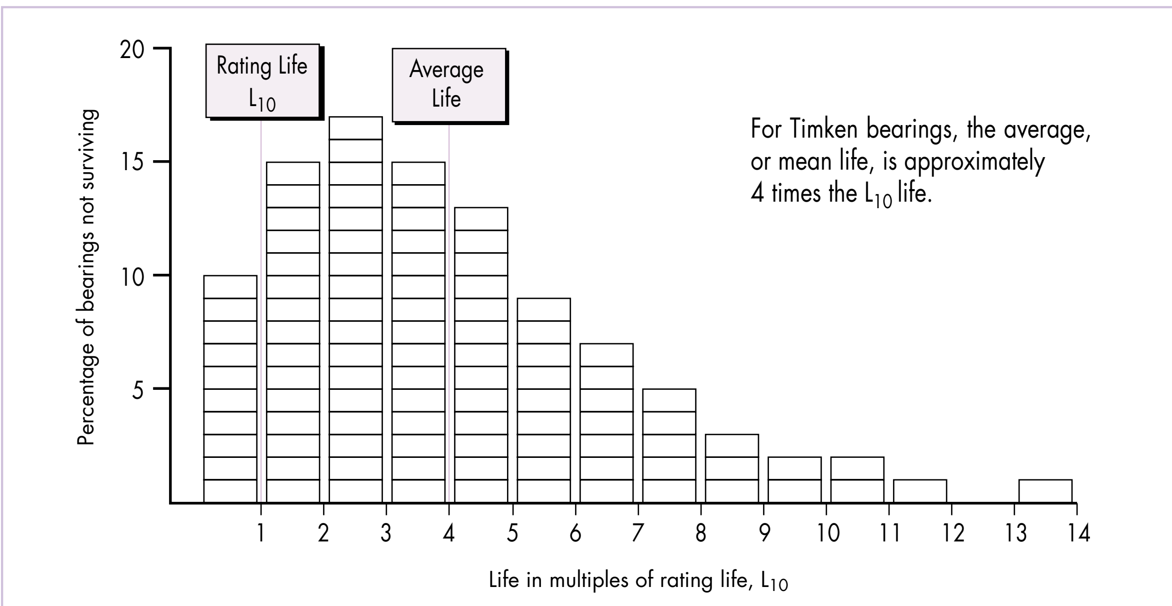


Fig. 3-11
Theoretical life frequency distribution of one hundred apparently identical bearings operating under similar conditions.

specified laboratory conditions until a material associated fatigue spall of 6 mm^2 (0.01 in^2) develops on each bearing, 90 percent of these bearings are expected to exhibit lives greater than the rating life. Then, only 10 percent would have lives less than the rating life. The example (fig. 3-11), shows bearing life scatter following a Weibull distribution function with a dispersion parameter (slope) equal to 1.5. From hundreds of such tested groups, L_{10} life estimates are determined. Likewise, rating life and load rating are established and verified.

To assure consistent quality, worldwide, The Timken Company conducts extensive bearing fatigue life tests in laboratories in the United States and in England. This testing results in confidence in Timken ratings.

Bearing life adjustment equations are:

$$L_{na} = a_1\,a_2\,a_3\,a_4 \left(\frac{C_{90}}{P}\right)^{10/3} (90 \times 10^6) \quad \text{(revolutions)}$$

or

$$L_{na} = a_1\,a_2\,a_3\,a_4 \left(\frac{C_{90}}{P}\right)^{10/3} \left(\frac{1.5 \times 10^6}{n}\right) \quad \text{(hours)}$$

where:

a_1 = life adjustment factor for reliability
a_2 = life adjustment factor for bearing material
a_3 = life adjustment factor for environmental conditions
a_4 = life adjustment factor for useful life (spall size)

For the case of a pure external thrust load, F_a, the previous equation becomes:

$$L_{na} = a_1\, a_2\, a_3\, a_4 \left(\frac{C_{a90}}{F_a}\right)^{10/3} \left(\frac{1.5 \times 10^6}{n}\right) \text{ (hours)}$$

Traditional L_{10} life calculations are based on bearing capacity, dynamic equivalent radial load (see page 60) and speed. The Timken Company method of calculating L_{10} life is based on a C_{90} load rating, which is the load under which population of bearings will achieve an L_{10} life of 90 million revolutions. The ISO method is based on a C_1 load rating, which produces a population L_{10} life of 1 million revolutions. While these two methods correctly account for the differences in basis, other differences can affect the calculation of bearing life. For instance, the two methods of calculating dynamic equivalent radial load (pages 57) can yield slight differences that are accentuated in the life equations by the exponent 10/3. In addition, it is important to distinguish between the ISO L_{10} life calculation method and the ISO bearing rating. Comparisons between bearing lives should only be made for values calculated on the same basis (C_1 or C_{90}) and the same rating formula (Timken or ISO). The two methods are listed below.

1) The Timken Company method

$$L_{10} = \left(\frac{C_{90}}{P}\right)^{10/3} 90 \times 10^6 \quad \text{(revolutions)} \qquad (1)$$

$$L_{10} = \left(\frac{C_{90}}{P}\right)^{10/3} \left(\frac{1.5 \times 10^6}{n}\right) \quad \text{(hours)} \qquad (2)$$

where:

L_{10} = rating life or catalog life (life expectancy associated with 90% reliability)

C_{90} = basic dynamic radial load rating of a single row bearing for an L_{10} life of 90 million revolutions (3,000 hours at 500 rev/min)

P = dynamic equivalent radial load (see page 60)

n = speed of rotation, rev/min

Note: for pure thrust loading and for thrust bearings, equations 1 and 2 become:

$$L_{10} = \left(\frac{C_{a90}}{F_{ae}}\right)^{10/3} 90 \times 10^6 \quad \text{(revolutions)} \qquad (1a)$$

$$L_{10} = \left(\frac{C_{a90}}{F_{ae}}\right)^{10/3} \left(\frac{1.5 \times 10^6}{n}\right) \quad \text{(hours)} \qquad (2a)$$

where:

C_{a90} = basic dynamic thrust rating for an L_{10} life of 90 million revolutions

F_{ae} = external thrust load

2) The ISO method (ISO 281)

$$L_{10} = \left(\frac{C_1}{P}\right)^{10/3} 1 \times 10^6 \quad \text{(revolutions)} \qquad (3)$$

$$L_{10} = \left(\frac{C_1}{P}\right)^{10/3} \left(\frac{1 \times 10^6}{60\, n}\right) \quad \text{(hours)} \qquad (4)$$

where:

C_1 = basic dynamic radial load rating for an L_{10} life of 1 million revolutions

Note: The C_1 ratings used in equations 3 and 4 and listed in the Bearing Data Tables are Timken C_{90} ratings modified for an L_{10} of 1 million revolutions and not ISO 281 ratings.

1.1.4. Bearing equivalent loads and required ratings

Tapered roller bearings are ideally suited to carry all types of loadings - radial, thrust and any combination of both. Due to the tapered design of the bearing, a radial load will induce a thrust reaction within the bearing that must be opposed by an equal or greater thrust reaction to keep the cones and cups from separating. The number of rollers in contact as a result of this ratio determines the load zone in the bearing. If all the rollers are in contact, the load zone is referred to as being 360 degrees.
When only a radial load is applied to a tapered roller bearing, it is assumed that half the rollers support the load and the load zone is 180 degrees. In this case, induced bearing thrust is:

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

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

1.1.5. Dynamic equivalent radial load

The basic dynamic radial load rating, C_{90}, is assumed to be the radial load carrying capacity with a 180-degree load zone in the bearing. When the thrust load on a bearing exceeds the induced thrust, $F_{a(180)}$, a dynamic equivalent radial load must be used to calculate bearing life.
The dynamic equivalent radial load is that radial load which, if applied to a bearing, will give the same life as the bearing will attain under the actual loading (combined axial and thrust).
The equations presented give close approximations of the dynamic equivalent radial load assuming a 180-degree load zone in one bearing and 180 degrees or more in the opposite bearing. More exact calculations using computer programs can be used to account for parameters such as bearing spring rate, setting and supporting housing stiffness.

The approximate equation is:

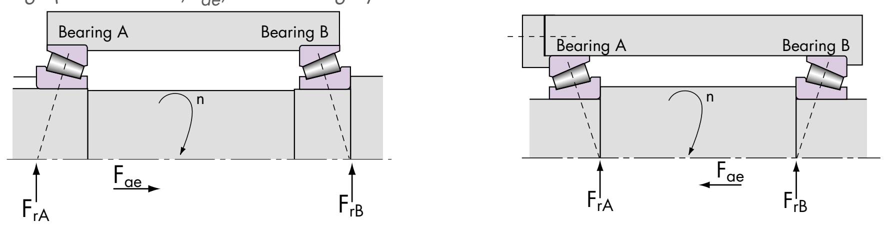
$$P = XF_r + YF_a$$

The following tables give the equations to determine bearing thrust load and the dynamic equivalent radial loads for various designs. The Timken method along with ISO method are shown. The factors necessary to perform the calculations are shown in the bearing tables.

1.1.6. Single row equations

Combined radial and thrust load

Design (external thrust, F_{ae}, onto bearing A)



ISO method

Thrust condition 1

$$\frac{0.5\,F_{rA}}{Y_A} \leq \frac{0.5\,F_{rB}}{Y_B} + F_{ae}$$

Net bearing thrust load

$$F_{aA} = \frac{0.5\,F_{rB}}{Y_B} + F_{ae}$$

$$F_{aB} = \frac{0.5\,F_{rB}}{Y_B}$$

Dynamic equivalent radial load

if $\dfrac{F_{aA}}{F_{rA}} \leq e_A$

$P_A = F_{rA}$

if $\dfrac{F_{aA}}{F_{rA}} > e_A$

$P_A = 0.4\,F_{rA} + Y_A\,F_{aA}$

$P_B = F_{rB}$

Thrust condition 2

$$\frac{0.5\,F_{rA}}{Y_A} > \frac{0.5\,F_{rB}}{Y_B} + F_{ae}$$

Net bearing thrust load

$$F_{aA} = \frac{0.5\,F_{rA}}{Y_A}$$

$$F_{aB} = \frac{0.5\,F_{rA}}{Y_A} - F_{ae}$$

Dynamic equivalent radial load

$P_A = F_{rA}$

if $\dfrac{F_{aB}}{F_{rB}} \leq e_B$, $P_B = F_{rB}$

if $\dfrac{F_{aB}}{F_{rB}} > e_B$

$P_B = 0.4\,F_{rB} + Y_B\,F_{aB}$

L_{10} life

$$L_{10A} = \frac{10^6}{60n}\left(\frac{C_{1A}}{P_A}\right)^{10/3} \quad \text{(hours)}$$

$$L_{10B} = \frac{10^6}{60n}\left(\frac{C_{1B}}{P_B}\right)^{10/3} \quad \text{(hours)}$$

Timken method

Thrust condition 1

$$\frac{0.47\,F_{rA}}{K_A} \leq \frac{0.47\,F_{rB}}{K_B} + F_{ae}$$

Net bearing thrust load

$$F_{aA} = \frac{0.47\,F_{rB}}{K_B} + F_{ae}$$

$$F_{aB} = \frac{0.47\,F_{rB}}{K_B}$$

Dynamic equivalent radial load

$P_A = 0.4\,F_{rA} + K_A\,F_{aA}$

if $P_A < F_{rA}$, $P_A = F_{rA}$

$P_B = F_{rB}$

Thrust condition 2

$$\frac{0.47\,F_{rA}}{K_A} > \frac{0.47\,F_{rB}}{K_B} + F_{ae}$$

Net bearing thrust load

$$F_{aA} = \frac{0.47\,F_{rA}}{K_A}$$

$$F_{aB} = \frac{0.47\,F_{rA}}{K_A} - F_{ae}$$

Dynamic equivalent radial load

$P_A = F_{rA}$

$P_B = 0.4\,F_{rB} + K_B\,F_{aB}$

if $P_B < F_{rB}$, $P_B = F_{rB}$

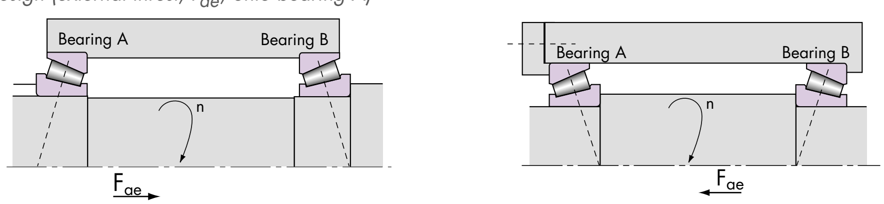
L_{10} life

$$L_{10A} = \left(\frac{C_{90A}}{P_A}\right)^{10/3} \times 3000 \times \frac{500}{n} \quad \text{(hours)}$$

$$L_{10B} = \left(\frac{C_{90B}}{P_B}\right)^{10/3} \times 3000 \times \frac{500}{n} \quad \text{(hours)}$$

Thrust load only

Design (external thrust, F_{ae}, onto bearing A)



ISO method

Thrust condition

$F_{aA} = F_{ae}$
$F_{aB} = 0$

Thrust load

$F_{aA} = F_{ae}$
$F_{aB} = 0$

Dynamic equivalent load

$P_A = Y_A\,F_{aA}$
$P_B = 0$

L_{10} life

$$L_{10A} = \frac{10^6}{60n}\left(\frac{C_{1A}}{P_A}\right)^{10/3} \quad \text{(hours)}$$

$$L_{10B} = \frac{10^6}{60n}\left(\frac{C_{1B}}{P_B}\right)^{10/3} \quad \text{(hours)}$$

Timken method

Thrust condition

$F_{aA} = F_{ae}$
$F_{aB} = 0$

Thrust load

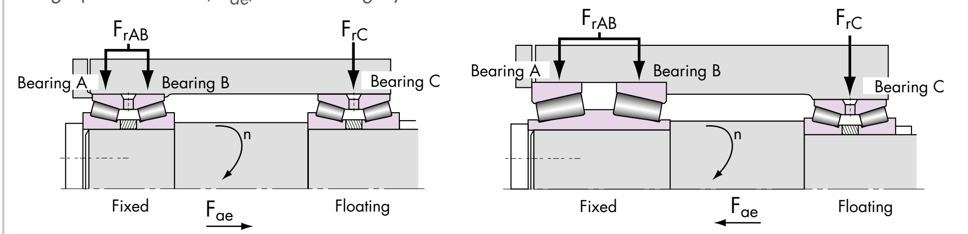
$F_{aA} = F_{ae}$
$F_{aB} = 0$

L_{10} life

$$L_{10A} = \left(\frac{C_{a90A}}{F_{aA}}\right)^{10/3} \times 3000 \times \frac{500}{n} \quad \text{(hours)}$$

$$L_{10B} = \left(\frac{C_{a90B}}{F_{aB}}\right)^{10/3} \times 3000 \times \frac{500}{n} \quad \text{(hours)}$$

Similar bearing series, $K_A = K_B$



Design (external thrust, F_{ae}, onto bearing A)

ISO method

Thrust condition

$$\frac{F_{ae}}{F_r} \leq e \qquad\qquad \frac{F_{ae}}{F_r} > e$$

Dynamic equivalent radial load

$P_{AB} = F_{rAB} + Y_{1AB}\,F_{ae}$ | $P_{AB} = 0.67\,F_{rAB} + Y_{2AB}\,F_{ae}$

$P_C = F_{rC}$ | $P_C = F_{rC}$

L_{10} life

$$L_{10AB} = \frac{10^6}{60n}\left(\frac{C_{1\,(2)}}{P_{AB}}\right)^{10/3} \text{ (hours)}$$

$$L_{10C} = \frac{10^6}{60n}\left(\frac{C_{1\,(2)}}{P_C}\right)^{10/3} \text{ (hours)}$$

Timken method

Thrust condition

$$F_{ae} > \frac{0.6\,F_{rAB}}{K_A} \qquad\qquad F_{ae} \leq \frac{0.6\,F_{rAB}}{K_A}$$

Dynamic equivalent radial load

$P_A = 0.4\,F_{rAB} + K_A\,F_{ae}$
$P_B = 0$
$P_C = F_{rC}$

$P_A = 0.5\,F_{rAB} + 0.83\,K_A\,F_{ae}$
$P_B = 0.5\,F_{rAB} - 0.83\,K_A\,F_{ae}$
$P_C = F_{rC}$

L_{10} life

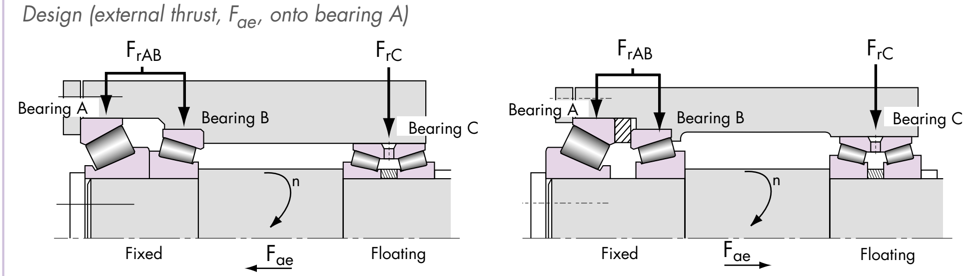
$$L_{10A} = \left(\frac{C_{90A}}{P_A}\right)^{10/3} \times 3000 \times \frac{500}{n} \text{ (hours)}$$

$$L_{10B} = \left(\frac{C_{90B}}{P_B}\right)^{10/3} \times 3000 \times \frac{500}{n} \text{ (hours)}$$

$$L_{10C} = \left(\frac{C_{90(2)C}}{P_C}\right)^{10/3} \times 3000 \times \frac{500}{n} \text{ (hours)}$$

$C_{90\,(2)}$ = dynamic radial load rating for 2 rows

Dissimilar bearing series $K_A \neq K_B$



Design (external thrust, F_{ae}, onto bearing A)

Timken method

Thrust condition

$$F_{ae} > \frac{0.6\,F_{rAB}}{K_A} \qquad\qquad F_{ae} \leq \frac{0.6\,F_{rAB}}{K_A}$$

Dynamic equivalent radial load

$P_A = 0.4\,F_{rAB} + K_A\,F_{ae}$

$P_B = 0$

$P_C = F_{rC}$

$$P_A = \frac{K_A}{K_A + K_B}\left(F_{rAB} + 1.67\,K_B\,F_{ae}\right)$$

$$P_B = \frac{K_B}{K_A + K_B}\left(F_{rAB} - 1.67\,K_A\,F_{ae}\right)$$

$P_C = F_{rC}$

L_{10} life

$$L_{10A} = \left(\frac{C_{90A}}{P_A}\right)^{10/3} \times 3000 \times \frac{500}{n} \text{ (hours)}$$

$$L_{10B} = \left(\frac{C_{90B}}{P_B}\right)^{10/3} \times 3000 \times \frac{500}{n} \text{ (hours)}$$

$$L_{10C} = \left(\frac{C_{90(2)c}}{P_C}\right)^{10/3} \times 3000 \times \frac{500}{n} \text{ (hours)}$$

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1.2. Adjusted life

1.2.1. General equation

With the increased emphasis on the relationship between rating reference conditions and the actual environment in which the bearing operates, the traditional life equations have been expanded to include certain additional variables that affect bearing performance.
The expanded bearing life equation becomes:

$L_{na} = a_1\ a_2\ a_3\ a_4\ L_{10}$

L_{na} = adjusted rating life for a reliability of (100 – n) percent

a_1 = life adjustment factor for reliability

a_2 = life adjustment factor for material

a_3 = life adjustment factor for environmental conditions

a_4 = life adjustment factor for useful life

L_{10} = rating life from equations 1 to 4 page 56

1.2.2. Factor for reliability - 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.

Rating life, L_{10}, for an individual bearing, or a group of identical bearings operating under the same conditions, is the life associated with 90 percent reliability. Some bearing applications require a reliability other than 90 percent. A life adjustment factor for determining a reliability other than 90 percent is:

$$a_1 = 4.48 \left(\ln \frac{100}{R} \right)^{2/3}$$

ln = natural logrithium (Base e)

Multiply the calculated L_{10} rating life by a_1 to obtain the L_n life, which is the life for reliability of R percent. By definition, $a_1 = 1$ for a reliability of 90 percent so, for reliabilities greater than 90 percent, $a_1 < 1$ and for reliabilities less than 90 percent, $a_1 > 1$.

1.2.3. Factor for material - a_2

For Timken bearings manufactured from electric-arc furnace, ladle refined, bearing quality alloy steel, a_2 is generally = 1.
Bearings can also be manufactured from premium steels that contain fewer and smaller inclusion impurities than standard bearing steels and provide the benefit of extending bearing fatigue life where it is limited by non-metallic inclusions. A higher value can then be applied for the factor a_2.

1.2.4. Factor for environmental conditions - a_3

Calculated life can be modified to take account of different environmental conditions, on a comparative basis, by using the factor a_3 which is comprised of three separate factors:

$a_3 = a_{3k}\ a_{3\ell}\ a_{3m}$

a_{3k} = life adjustment factor for load zone
$a_{3\ell}$ = life adjustment factor for lubrication
a_{3m} = life adjustment factor for alignment

a_{3k} - load zone factor

Load zone is the loaded portion of the raceway measured in degrees (fig. 3-12). It is a direct indication of how many rollers share the applied load.
Load zone is a function of the amount of endplay (internal clearance) or preload within the bearing system. This, in turn, is a function of the initial setting, internal geometry of the bearing, the load applied and deformation of components (shaft, bearing, housing).

$a_{3k} = 1$ – The nominal or "catalog" L_{10} life assumes a minimum of 180° load zone in the bearing.

$a_{3k} \neq 1$ – Depending on endplay or preload, to quantify a_{3k} requires computer analysis by The Timken Company.

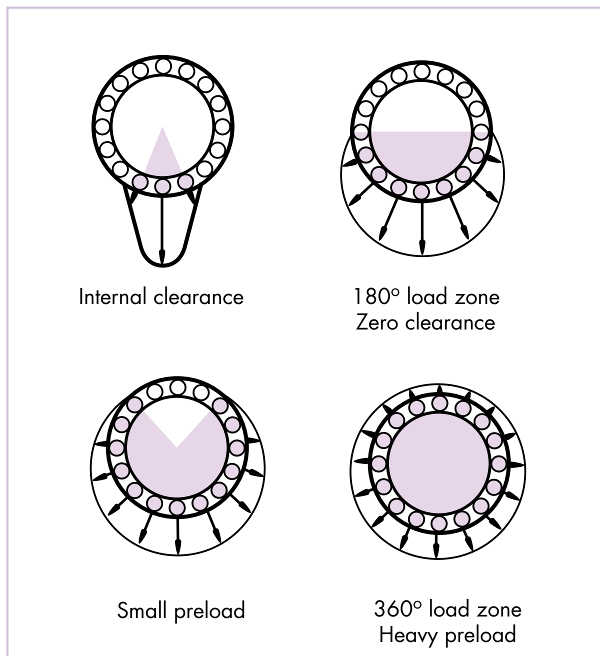

Fig. 3-12
Load zone effect - radial load applied.

a_{3m} - alignment factor

For optimum performance and life, the races of a tapered roller bearing should be perfectly aligned. However, this is generally impractical due to misalignment between shaft and housing seats and also deflection under load (fig. 3-13).

$a_{3m} = 1$ – For catalog life calculations, it is assumed that alignment is equivalent to the rating reference condition of 0.0005 radians.

$a_{3m} < 1$

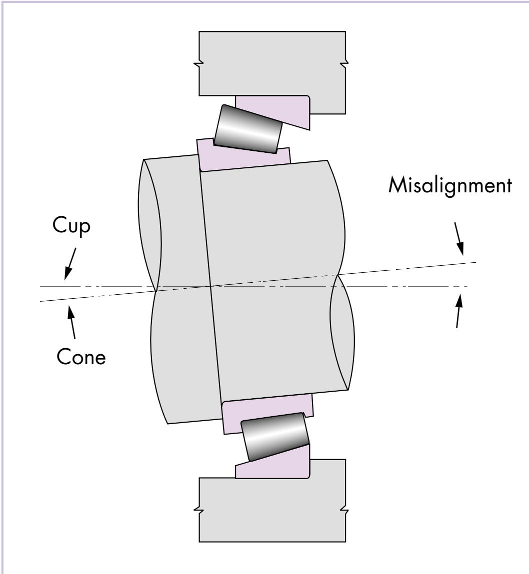

Fig. 3-13
Misalignment.

If misalignment is greater than 0.0005 radians, then bearing performance will be affected. However, the predicted life is dependent on such factors as bearing internal geometry, load zone and applied load. P900 bearings can be tailored to suit particular application conditions, like misalignment with component profiling. Quantifying a_{3m}, for actual operating conditions or to determine the benefits of P900, requires a computer analysis by Timken.

$a_{3\ell}$ - lubrication factor

Ongoing research conducted by The Timken Company has demonstrated that bearing life calculated from only speed and load, may be very different from actual life when the operating environment differs perceptibly from laboratory conditions. Historically, The Timken Company has calculated the catalogue life adjustment factor for lubrication ($a3_\ell$) as a function of three parameters:

- Bearing speed
- Bearing operating temperature
- Oil viscosity

These parameters are needed to determine the elastohydrodynamic (EHL) lubricant in the rolling contact region of rolling element bearings. During the last decade, extensive testing has been done to quantify the effects of other lubrication related parameters on bearing life. Roller and raceway surface finish relative to lubricant film thickness have the most notable effect. Other factors include bearing geometry, material, loads and load zone.
The following equation provides a simple method to calculate the lubrication factor for an accurate prediction of the influence of lubrication on bearing life (L10a).

$a_{3\ell} = C_g \cdot x\ C_\ell \cdot x\ C_j \cdot x\ C_s \cdot x\ C_v \cdot x\ C_{gr}$

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

Note: The $a3_\ell$ maximum is 2.88 for all bearings. The a3l minimum is 0.20 for case carburized bearings and 0.06 for through hardened bearings.

A lubricant contamination factor is not included in the lubrication factor because our endurance tests are run with a 40 µm filter to provide a realistic level of lubricant cleanness.

Geometry factor - C_g

C_g Is given for each cone part number in the TS bearing tables (pages 164 to 256). Note that this factor is not applicable to our P900 bearing concept (see page 64).

Load factor - C_ℓ

The C_ℓ factor is obtained from figure 3-14. Note that the factor is different for case carburized and through hardened bearings. F_a is the thrust load on each bearing which is determined from the calculation method on page 64. Separate curves are given for loads given in Newtons or pounds.
It is necessary to resolve all loads on the shaft into bearing radial loads (F_{rA}, F_{rB}) and one external thrust load (F_{ae}) before calculating the thrust load for each bearing.

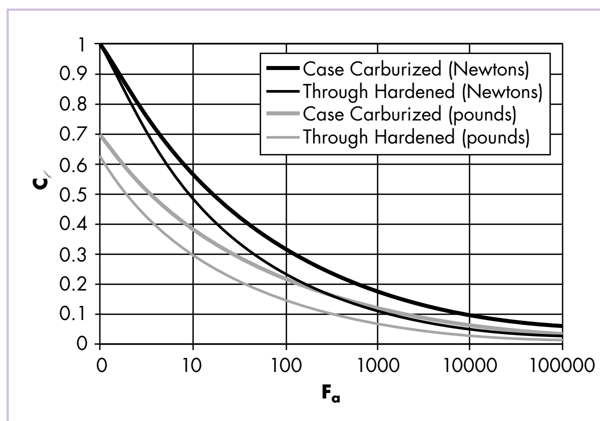

Fig. 3-14
Load factor (C_ℓ).

Load zone factor - C_j

a) Calculate X, where $X = \dfrac{F_r}{F_a\ K}$

b) If X > 2.13, the bearing load zone is less than 180°, then:
For case carburized bearings, C_j = 0.747
For through hardened bearings, C_j = 0.691

If X < 2.13, the bearing load zone is larger than 180° and C_j. can be determined from figure 3-15.

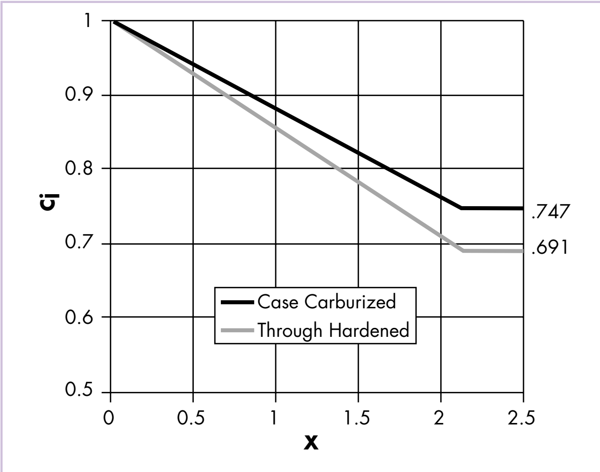

Fig. 3-15
Load zone factor (C_i).

Speed factor - C_s

C_s is determined from figure 3-16 where rev/min (RPM) is the rotational speed of the inner race relative to the outer race.

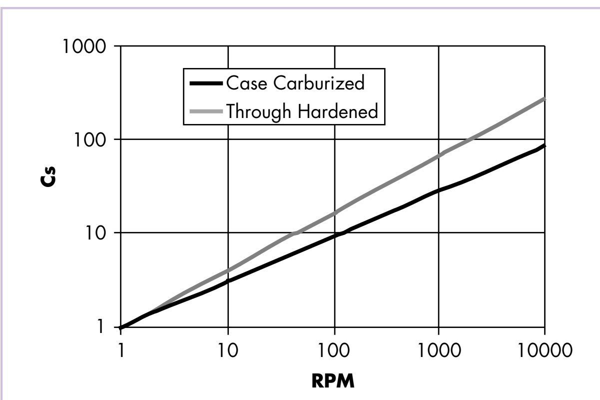

Fig. 3-16
Speed factor (C_s).

Viscosity factor - C_v

The kinematic viscosity lubricant [Centistokes (cSt)] is taken at the operating temperature of the bearings. The operating viscosity can be estimated by using figure 5-7, page 120 in Section 5 "Lubricating your bearings." Viscosity factor (C_v) can then be determined from figure 3-17.

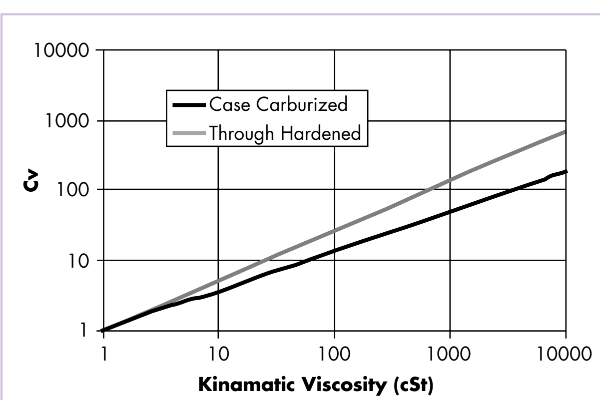

Fig. 3-17
Viscosity factor (C_v).

Grease lubrication factor - C_{gr}

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

For case carburized bearings, $C_{gr} = 0.79$
For through hardened bearings, $C_{gr} = 0.74$

1.2.5 Factor for useful life - a_4

The limiting criterion for fatigue is a spalled area of 6 mm^2 (0.01 in^2). This is the reference condition in The Timken Company rating, $a_4 = 1$.
If a larger limit for area of fatigue spall can be reasonably established for a particular application, then a higher value of a_4 can be applied.

1.2.6. Select-A-Nalysis™

Bearing Systems Analysis analyzes the effect many real life variables have on bearing performance, in addition to the load and speed considerations used in the traditional catalog life calculation approach.
The Timken Company's unique computer program, Select-A-Nalysis, adds sophisticated bearing selection logic to that analytical tool.
Bearing Systems Analysis allows the designer to quantify differences in bearing performance due to changes in the operating environment.
The selection procedure can be either performance or price oriented.

1.3. System life and weighted average load and life

1.3.1. System life

System reliability is the probability that all of several 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 an application, the L_{10} system life for a number of bearings each having a different L_{10} life is:

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

1.3.2. Weighted average load and life equations

In many applications bearings are subjected to variable conditions of loading, and bearing selection is often made on the basis of maximum load and speed.
However, under these conditions a more meaningful analysis may be made examining the loading cycle to determine the weighted average load.
Bearing selection based on weighted average loading will take into account variations in speed, load and proportion of

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time during which the variable loads and speed occur. However, it is still necessary to consider extreme loading conditions to evaluate bearing contact stresses and alignment.

Weighted average load

Variable speed, load and proportion time:

$$F_{wt} = \left(\frac{n_1\, T_1\, F_1^{\;10/3} + \ldots + n_n\, T_n\, F_n^{\;10/3}}{n_a} \right)^{0.3}$$

where, during each condition in a load cycle:
T = proportion of total time
F = load applied
n = speed of rotation, rev/min
n_a = assumed (arbitrary) speed of rotation for use in bearing life equations. For convenience, 500 rev/min is normally used.

Uniformly increasing load, constant speed:

$$F_{wt} = \left[\frac{3}{13} \left(\frac{F_{max}^{\;13/3} - F_{min}^{\;13/3}}{F_{max} - F_{min}} \right) \right]^{0.3}$$

where, during a load cycle:

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

Note: The above formulas do not allow the use of the life modifying factor for lubrication $a_{3\ell}$, except in the case of constant speed. Therefore, when these equations are used in the bearing selection process, the design L_{10} bearing life should be based on a similar successful machine that operates in the same environment. Life calculations for both machines must be performed on the same basis. To allow for varying lubrication conditions in a load cycle, it is necessary to perform the weighted average life calculation:

Weighted average life

$$L_{10wt} = \frac{1}{\dfrac{T_1}{(L_{10})_1} + \dfrac{T_2}{(L_{10})_2} + \ldots + \dfrac{T_n}{(L_{10})_n}}$$

where, during a load cycle:

T = proportion of total time
L_{10} = calculated L_{10} bearing life (page 55) for each condition

1.3.3. Ratios of bearing life to loads, power and speeds

In applications subjected to variable conditions of loading, bearing life is calculated for one condition. Life for any other condition can easily be calculated by taking the ratio of certain variables. To use these ratios, the bearing load must vary proportionally with power, speed or both. Nevertheless, this applies only to catalog lives or adjusted lives by any life adjustment factors.

The following relationships in table 3-C hold under stated specific conditions:

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

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

Table 3-C
Life ratio equations.

P = Load, torque or tangential gear force

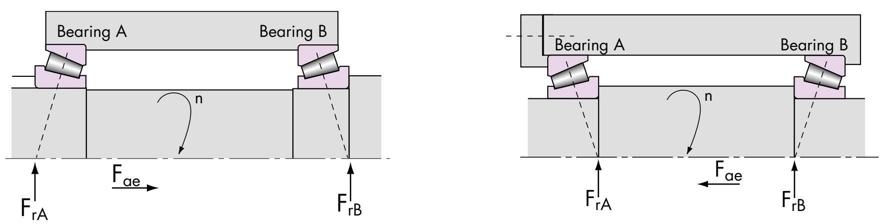
1.3.4. Life calculation examples

Combined radial and thrust load

A

32012X
C_{1A} = 89600 N
Y_A = 1.39
e_A = 0.43
C_{90A} = 23200 N
K_A = 1.36

Design (external thrust, F_{ae}, onto bearing A)



B

32011X
C_{1B} = 88000 N
Y_B = 1.48
e_B = 0.41
C_{90B} = 22800 N
K_B = 1.44

Speed n = 600 rev/min
Operating temperature = 60°C
Oil viscosity = VG46

F_{ae} = 4000 N
F_{rA} = 9000 N
F_{rB} = 7000 N

ISO method

Thrust condition

$$\frac{0.5 \times 9000}{1.39} < \frac{0.5 \times 7000}{1.48} + 4000$$

Net bearing thrust load

$$F_{aA} = \frac{0.5 \times 7000}{1.48} + 4000$$

$$F_{aA} = 6365 \text{ N}$$

$$F_{aB} = \frac{0.5 \times 7000}{1.48}$$

$$F_{aB} = 2365 \text{ N}$$

Timken method

Thrust conditon

$$\frac{0.47 \times 9000}{1.36} < \frac{0.47 \times 7000}{1.44} + 4000$$

Net bearing thrust load

$$F_{aA} = \frac{0.47 \times 7000}{1.44} + 4000$$

$$F_{aA} = 6285 \text{ N}$$

$$F_{aB} = \frac{0.47 \times 7000}{1.44}$$

$$F_{aB} = 2285 \text{ N}$$

ISO method

Dynamic equivalent radial load

$\frac{6365}{9000} = 0.707$ $e_A = 0.43$

$0.707 > 0.43$

$P_A = 0.4 \times 9000 + 1.39 \times 6365$

$P_A = 12447$ N

$P_B = F_{rB} = 7000$ N

L_{10} life

$$L_{10A} = \frac{10^6}{60 \times 600} \left(\frac{89600}{12447}\right)^{10/3} = 20006 \text{ hours}$$

$$L_{10B} = \frac{10^6}{60 \times 600} \left(\frac{88000}{7000}\right)^{10/3} = 128325 \text{ hours}$$

Life adjustment for lubrication

$a_{3\ell A} = 0.04138 \times (6365)^{-0.3131} \times 0.830 \times (600)^{0.6136} \times (20)^{0.7136} = 0.951$

$a_{3\ell B} = 0.03874 \times (2365)^{-0.3131} \times 0.690 \times (600)^{0.6136} \times (20)^{0.7136} = 1.009$

$L_{10aA} = 20006 \times 0.951 = 19026$ hours
$L_{10aB} = 128325 \times 1.009 = 129480$ hours

Timken method

Dynamic equivalent radial load

$P_A = 0.4 \times 9000 + 1.36 \times 6285$

$P_A = 12147$ N

$P_B = F_{rB} = 7000$ N

L_{10} life

$$L_{10A} = \left(\frac{23200}{12147}\right)^{10/3} \times 3000 \times \frac{500}{600} = 21610 \text{ hours}$$

$$L_{10B} = \left(\frac{22800}{7000}\right)^{10/3} \times 3000 \times \frac{500}{600} = 128054 \text{ hours}$$

Life adjustment for lubrication

$a_{3\ell A} = 0.04138 \times (6285)^{-0.3131} \times 0.830 \times (600)^{0.6136} \times (20)^{0.7136} = 0.954$

$a_{3\ell B} = 0.03874 \times (2285)^{-0.3131} \times 0.690 \times (600)^{0.6136} \times (20)^{0.7136} = 1.020$

$L_{10aA} = 21610 \times 0.954 = 20616$ hours
$L_{10aB} = 128054 \times 1.020 = 130615$ hours

2. Static conditions

2.1. Static rating

The static radial load rating C_0 is based on a maximum contact stress within a non-rotating bearing of 4,000 MPa (580,000 psi) at the center of contact and a 180° load zone (loaded portion of the raceway).

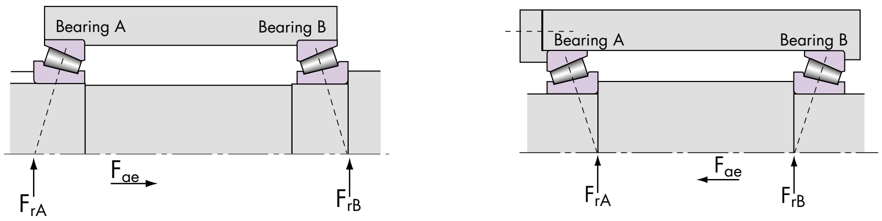
The 4,000 MPa (580,000 psi) stress level may cause visible light brinell marks on the bearing raceways. This degree of marking will not have a measurable effect on fatigue life when the bearing is subsequently rotating under a lower application load. If noise, vibration or torque are critical, a lower load limit may be required.

The following formulas may be used to calculate the static equivalent radial load on a bearing under a particular loading condition. This is then compared with the static radial rating as a criterion for selection of bearing size. However it is advisable to consult The Timken Company for qualification of bearing selection in applications where static loads prevail.

2.2. Static equivalent radial load (single-row bearings)

The static equivalent radial load is the static radial load (no rotation or oscillation) that produces the same maximum stress, at the center of contact of a roller, as the actual combined radial and thrust load applied. The equations presented give an approximation to the static equivalent radial load assuming a 180° load zone (loaded portion of the raceway) in one bearing and 180° or more in the opposing bearing.

Design (external thrust, F_{ae}, onto bearing A)



Thrust condition	Net bearing thrust load	Static equivalent radial load (P_0)
$\frac{0.47\ F_{rA}}{K_A} \leq \frac{0.47\ F_{rB}}{K_B} + F_{ae}$	$F_{aA} = \frac{0.47\ F_{rB}}{K_B} + F_{ae}$ $F_{aB} = \frac{0.47\ F_{rB}}{K_B}$	$P_{0B} = F_{rB}$ for $F_{aA} < 0.6\ F_{rA} / K_A$ $P_{0A} = 1.6\ F_{rA} - 1.269\ K_A\ F_{aA}$ for $F_{aA} > 0.6\ F_{rA} / K_A$ $P_{0A} = 0.5\ F_{rA} + 0.564\ K_A\ F_{aA}$
$\frac{0.47\ F_{rA}}{K_A} > \frac{0.47\ F_{rB}}{K_B} + F_{ae}$	$F_{aA} = \frac{0.47\ F_{rA}}{K_A}$ $F_{aB} = \frac{0.47\ F_{rA}}{K_A} - F_{ae}$	for $F_{aB} > 0.6\ F_{rB} / K_B$ $P_{0B} = 0.5\ F_{rB} + 0.564\ K_B\ F_{aB}$ for $F_{aB} < 0.6\ F_{rB} / K_B$ $P_{0B} = 1.6\ F_{rB} - 1.269\ K_B\ F_{aB}$ $P_{0A} = F_{rA}$

where:

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

Note: use the values of P_0 calculated for comparison with the static rating, C_0, even if P_0 is less than the radial applied, F_r.

2.3. Static equivalent radial load (two-row bearings)

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

$C_{0(2)} = 2C_0$

where:

$C_{0(2)}$ = two-row static radial rating
C_0 = static radial load rating of a single row bearing, type TS, from the same series (refer to part number index on page 121)

Where radial and thrust loads are applied consult a Timken Company sales engineer or representative.

3. Performance 900 (P900) bearings

P900 bearings permit critical applications to be downsized with smaller, lighter bearings, which allow upgraded power capacity, prolonged life and increased reliability.

P900 bearings can improve performance of standard bearings by a factor of 3 or more, within the same space.

P900 products offer:

- Extended life from super-clean airmelt steel
- Increased load-carrying capacity from enhanced bearing geometry
- Improved performance in thin lubricant film environments due to advanced surface finishes
- Technologically advanced analytical capabilities to apply these enhancements.

For more information on these new bearing capabilities, contact a Timken Company sales engineer or representative.

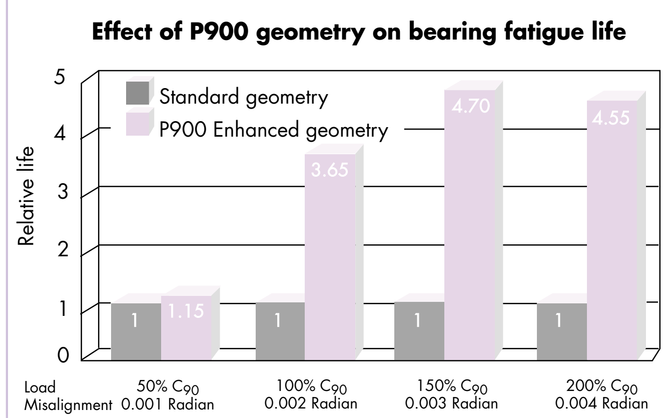


Fig. 3-15
The enhanced geometry of P900 bearings virtually eliminates edge stress concentrations caused by high loads or misalignment.

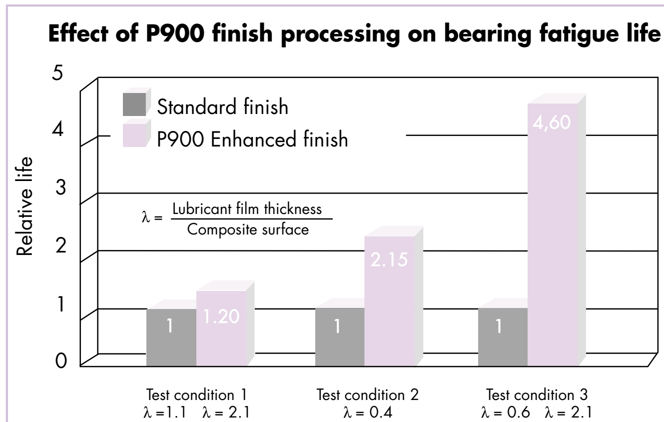


Fig. 3-16
The finishing process dramatically improves rolling contact surface finish and fatigue life when limited by surface distress. It also produces superior all-around surface topography and rounder rolling surfaces.

TM = Trademark of The Timken Company

C. Torque

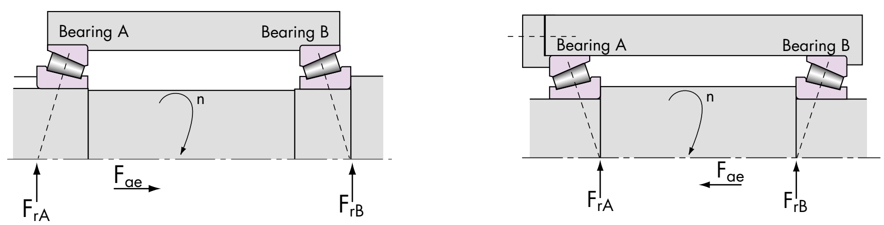
Running torque - M

The rotational resistance of a tapered roller bearing is dependent on load, speed, lubrication conditions and bearing internal characteristics.

The following formulas yield approximations to values of bearing running torque. The formulas apply to bearings lubricated by oil. For bearings lubricated by grease or oil mist, torque is usually lower although for grease lubrication this depends on amount and consistency of the grease. The formulas also assume the bearing running torque has stabilized after an initial period referred to as "running-in".

1. Single row

Design (external thrust, F_{ae}, onto bearing A)

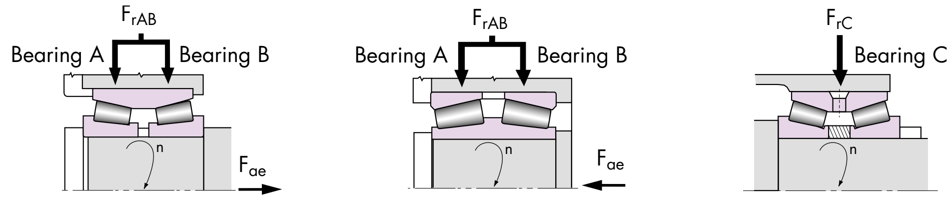


Thrust condition	Net bearing thrust load			
$\frac{0.47\,F_{rA}}{K_A} \leq \frac{0.47\,F_{rB}}{K_B} + F_{ae}$	$F_{aA} = \frac{0.47\,F_{rB}}{K_B} + F_{ae}$ $F_{aB} = \frac{0.47\,F_{rB}}{K_B}$	**1**	a) $\frac{K_A\,F_{aA}}{F_{rA}} > 2$ $f_1 = \frac{K_A\,F_{aA}}{F_{rA}}$ $f_2 = f_1 + 0.8$ b) $0.47 < \frac{K_A\,F_{aA}}{F_{rA}} \leq 2$ f_1, f_2: use graph page 67	$M_A = k_1\,G_{1A}\,(n\mu)^{0.62}\left(\frac{f_1\,F_{rA}}{K_A}\right)^{0.3}$
			c) $\frac{K_A\,F_{aA}}{F_{rA}} = 0.47$ $f_1 = 0.06$ $f_2 = 1.78$	$M_A = k_1\,G_{1A}\,(n\mu)^{0.62}\left(\frac{0.06F_{aA}}{K_A}\right)^{0.3}$
		2	$f_1 = 0.06$ $f_2 = 1.78$	$M_B = k_1\,G_{1B}\,(n\mu)^{0.62}\left(\frac{0.06F_{aB}}{K_B}\right)^{0.3}$
$\frac{0.47\,F_{rA}}{K_A} > \frac{0.47\,F_{rB}}{K_B} + F_{ae}$	$F_{aA} = \frac{0.47\,F_{rA}}{K_A}$ $F_{aB} = \frac{0.47\,F_{rA}}{K_A} - F_{ae}$	**1**	a) $\frac{K_B\,F_{aB}}{F_{rB}} > 2$ $f_1 = \frac{K_B\,F_{aB}}{F_{rB}}$ $f_2 = f_1 + 0.8$ b) $0.47 < \frac{K_B\,F_{aB}}{F_{rB}} \leq 2$ f_1, f_2: use graph page 67	$M_B = k_1\,G_{1B}\,(n\mu)^{0.62}\left(\frac{f_1\,F_{rB}}{K_B}\right)^{0.3}$
			c) $\frac{K_B\,F_{aB}}{F_{rB}} = 0.47$ $f_1 = 0.06$ $f_2 = 1.78$	$M_B = k_1\,G_{1B}\,(n\mu)^{0.62}\left(\frac{0.06F_{aB}}{K_B}\right)^{0.3}$
		2	$f_1 = 0.06$ $f_2 = 1.78$	$M_A = k_1\,G_{1A}\,(n\mu)^{0.62}\left(\frac{0.06F_{aA}}{K_A}\right)^{0.3}$

M_A or M_B will underestimate running torque if operating speed $n < \frac{k_2}{G_2\mu}\left(\frac{f_2\,F_r}{K}\right)^{2/3}$

2. Double row

Design (external thrust, F_{ae}, onto bearing A)



a) Fixed position

Load condition	Radial load on each row F_r		
$F_{ae} > \frac{0.47\ F_{rAB}}{K_A}$	Bearing B is unloaded $F_{rA} = F_{rAB}$ $F_{aA} = F_{ae}$	$\frac{K\ F_{ae}}{F_{rAB}} > 2$ $f_1 = \frac{K\ F_{ae}}{F_{rAB}}$ $f_2 = f_1 + 0.8$ $0.47 \leq \frac{K\ F_{ae}}{F_{rAB}} \leq 2$ f_1 , f_2: use graph page 67	$M_A = k_1\ G_{1A}\ (n\mu)^{0.62} \times (F_{ae})^{0.3}$ $M_A = k_1\ G_{1A}\ (n\mu)^{0.62} \left(\frac{f_1\ F_{rAB}}{K}\right)^{0.3}$
$F_{ae} \leq \frac{0.47\ F_{rAB}}{K_A}$	$F_{rA} = \frac{F_{rAB}}{2} + 1.06\ K\ F_{ae}$ $F_{rB} = \frac{F_{rAB}}{2} - 1.06\ K\ F_{ae}$	$M = k_1\ G_1\ (n\mu)^{0.62} \left(\frac{0.060}{K}\right)^{0.3} (F_{rA}{}^{0.3} + F_{rB}{}^{0.3})$	

b) Floating position

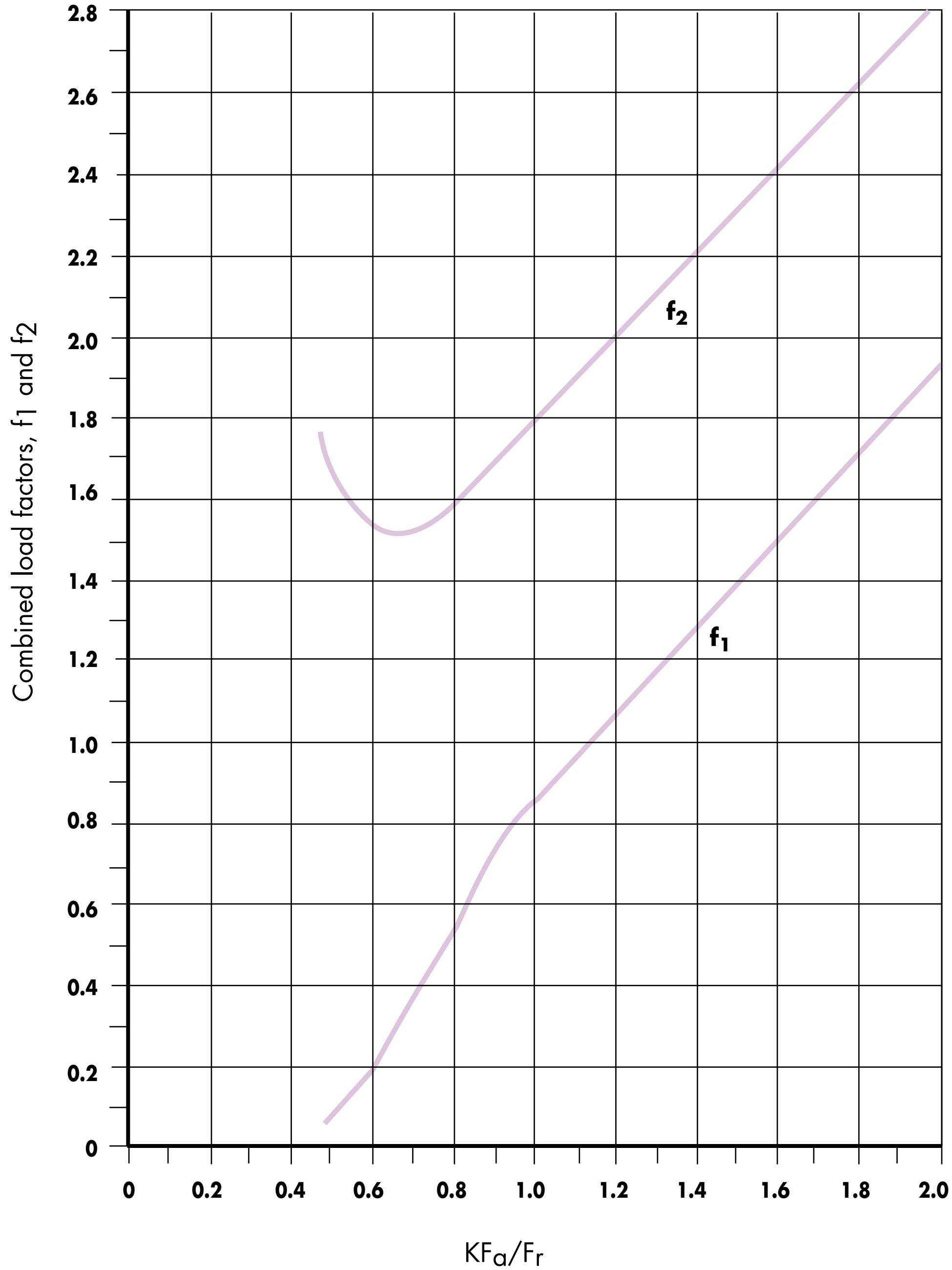
$$M_C = 2\ k_1\ G_{1C}\ (n\mu)^{0.62} \left(\frac{0.030\ F_{rC}}{K_C}\right)^{0.3}$$

M_A will underestimate running torque if operating speed $n < \frac{k_2}{G_2\mu}\left(\frac{f_2\ F_{rA}}{K}\right)^{2/3}$

M_{AB} will underestimate running torque if operating speed $n < \frac{k_2}{G_2\mu}\left(\frac{1.78\ F_{rA}}{K}\right)^{2/3}$

M_C will underestimate running torque if operating speed $n < \frac{k_2}{G_2\mu}\left(\frac{0.890\ F_{rC}}{K_C}\right)^{2/3}$

M = running torque, N.m (lbf.in)
F_r = radial load, N (lbf)
G_1 = geometry factor from bearing data tables
G_2 = geometry factor from bearing data tables
K = K-factor
n = speed of rotation, rev/min
k_1 = 2.56×10^{-6} (metric) or 3.54×10^{-5} (inch)
k_2 = 625 (metric) or 1700 (inch)
μ = lubricant dynamic viscosity at operating temperature centipoise. For grease use the base oil viscosity (fig 3-18).
f_1 = combined load factor (fig. 3-17)
f_2 = combined load factor (fig. 3-17)



Load condition	f_1 and f_2
$KF_a/F_r > 2.0$	$f_1 = KF_a/F_r$ $f_2 = f_1 + 0.8$
$0.47 \leq KF_a/F_r \leq 2.0$	Use graph above
$KF_a/F_r = 0.47$	$f_1 = 0.06$ $f_2 = 1.78$

Fig. 3-17
Determination of combined load factors f_1 and f_2.

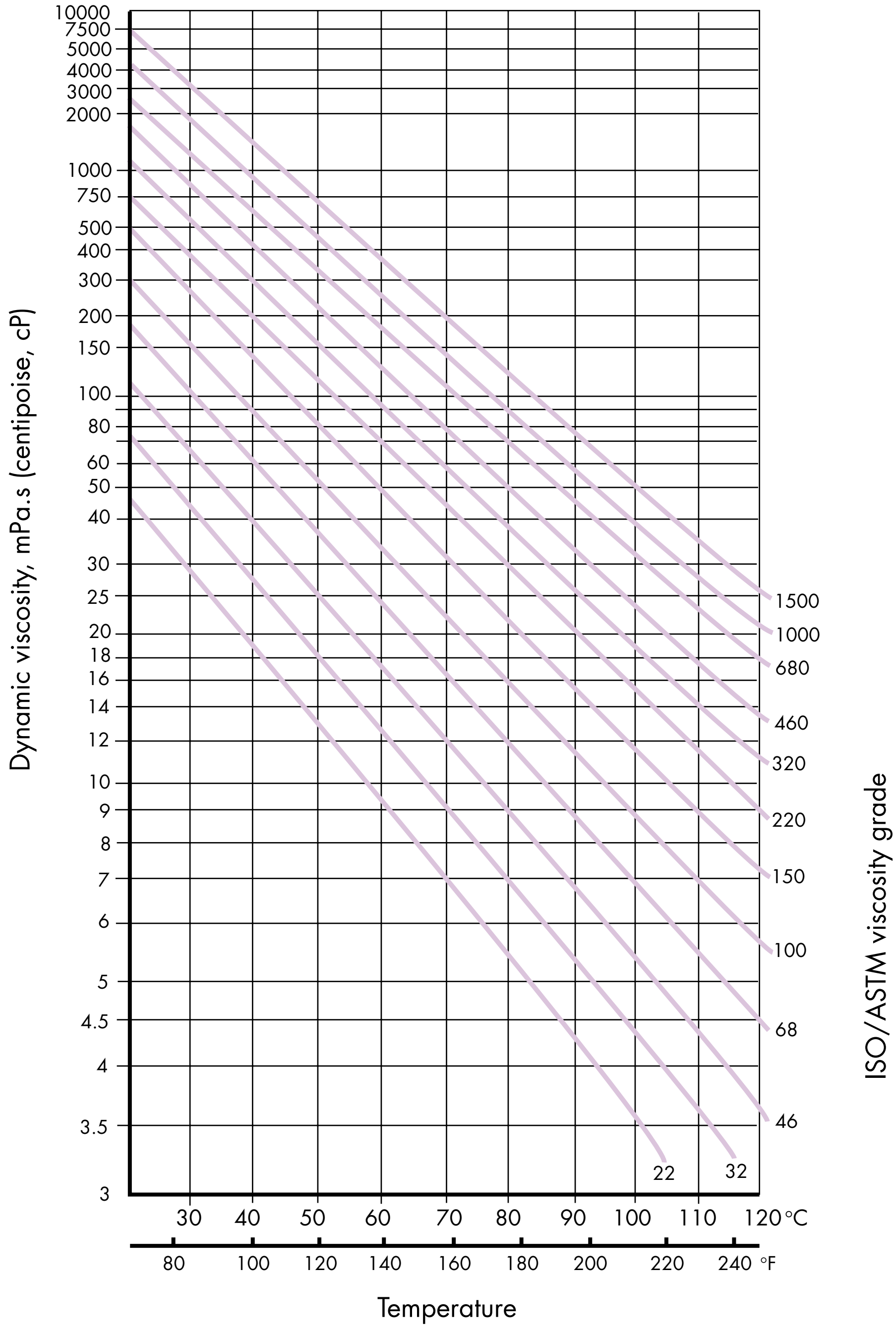


Fig. 3-18
Viscosities in mPa.s (centipoise, cP) for ISO/ASTM industrial fluid lubricant grade designations. Assumes: Viscosity Index 90; Specific Gravity 0.875 at 40°C.