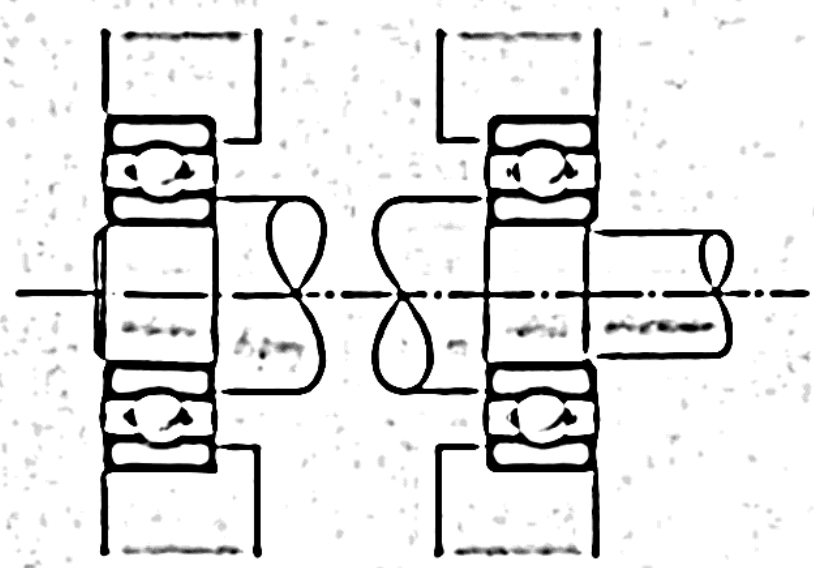


Figure 11-21: Non-locating (floating) arrangement;

Missing details?

Not to directly apply in actual designs. Refer to catalogs.

- For the so-called cantilevered shafts, refer to Figures 11-22 and 11-23, for example; or catalogs.

Misalignment
The permissible misalignment of a bearing depends on its type, and other design details of the bearing.

See Table 7-2 for typical permissible misalignments or maximum slopes.

Preloading
Although many design applications require little attention to stiffness or bearings, there are plenty of applications requiring high stiffness, high natural frequencies, low deflection, and low noise level.

Under such circumstances, preloading on bearings is to be considered.

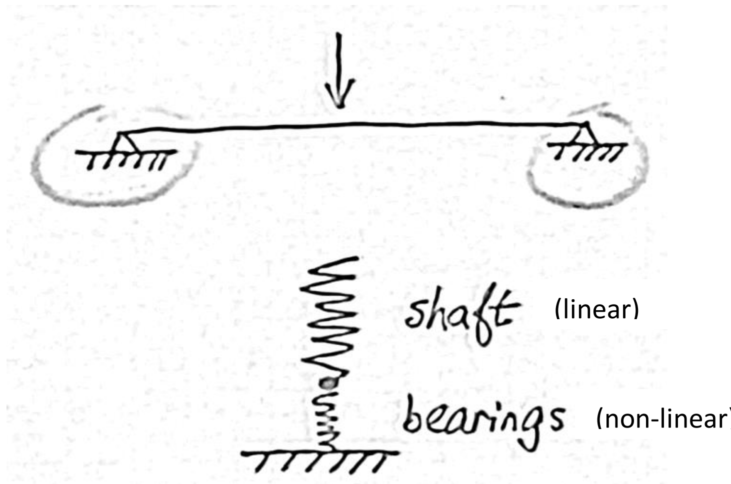
Preloading can also remove internal clearance, and increase fatigue life, amongst others.

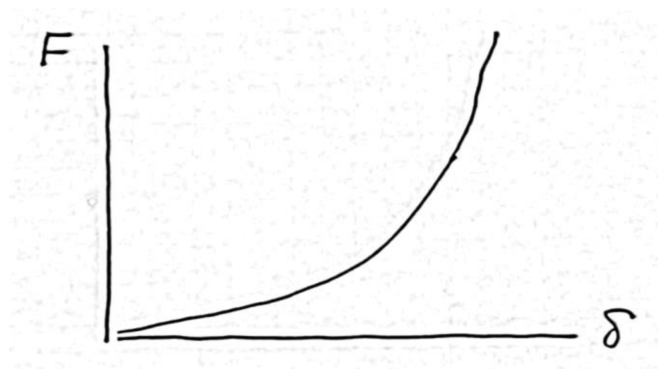
Too much preload causes early or premature failure. The key is to apply the appropriate amount of preload. Catalogs usually provide details of how much preload to apply, and how to apply.

Table 7-2

Typical Maximum Ranges for Slopes and Transverse Deflections

Slopes	
Tapered roller	0.0005–0.0012 rad
Cylindrical roller	0.0008–0.0012 rad
Deep-groove ball	0.001–0.003 rad
Spherical roller	0.026–0.052 rad
Self-align ball	0.026–0.052 rad
Uncrowned spur gear	<0.0005 rad
Transverse Deflections	
Spur gears with $P < 10$ teeth/in	0.010 in
Spur gears with $11 < P < 19$	0.005 in
Spur gears with $20 < P < 50$	0.003 in


shaft
(linear)
bearings
(non-linear)


F
δ

11-9 Selection of Tapered Roller Bearings

Figure 11-13: terminology

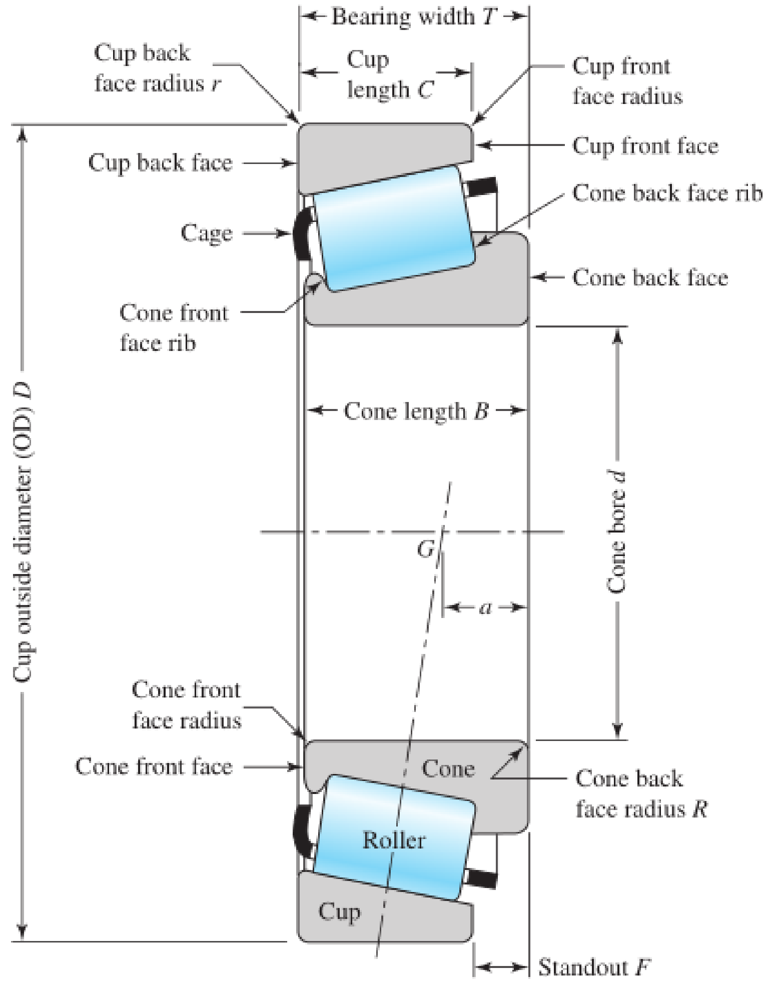


Figure 11–13

Nomenclature of a tapered roller bearing. Point G is the location of the effective load center; use this point to estimate the radial bearing load. *(Courtesy of The Timken Company.)*

<u>A few notes about the terminology:</u>

We don't call them inner and outer rings (Inner = cone, Outer = cup)

The larger end of the cone is the back face.

The smaller end of the cone is the front face.

G is the point of load application (radial and axial).

When G is inside the back face of a cone as shown, a will be given a negative value.

- Figure 11-14: *direct mounting* (back-to-back in terms of cone faces and *indirect mounting* (front-to-front in terms of cone faces).
- A_0 (point G of bearing A) and B_0 (point G or bearing B) are the points of application of loads, or locations of bearings or supports for analyses; a_e is the effective length between bearings or supports.

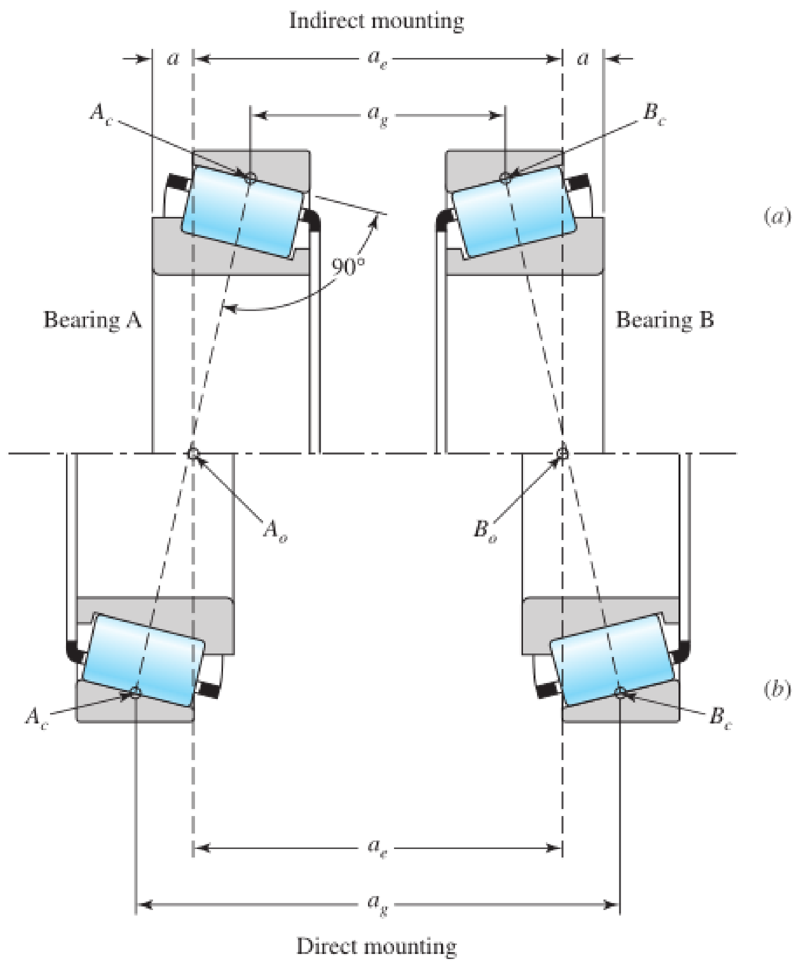


Figure 11–14

Comparison of mounting stability between indirect and direct mountings. *(Courtesy of The Timken Company.)*

- How to select tapered roller bearings?

<u>The induced thrusts</u>, see Figure 11-16.

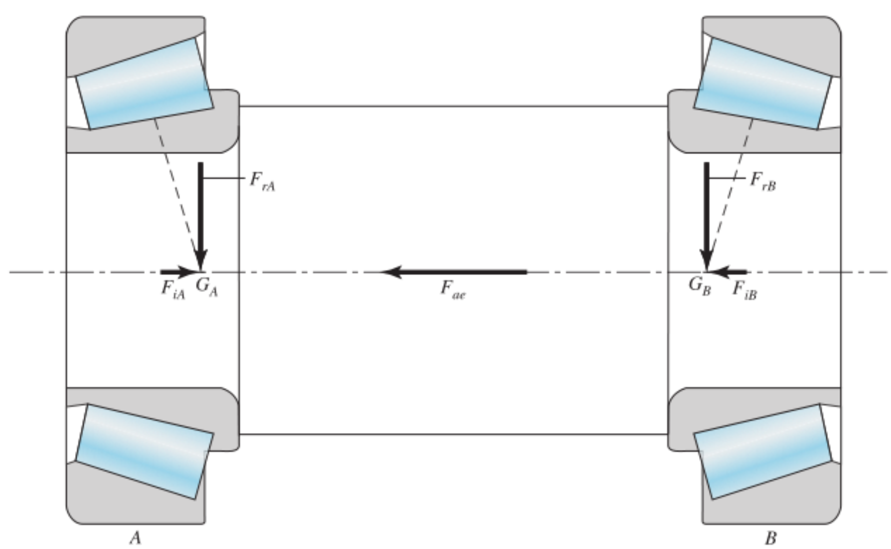


Figure 11–16

Direct-mounted tapered roller bearings, showing radial, induced thrust, and external thrust loads.

From FBD of the shaft: F_{rA} and F_{rB} are the radial loads, and F_{ae} is the externally applied axial load.

F_{iA} and F_{iB} are the *induced* axial loads.

The value of an induced axial load depends on the geometry of the bearing, and the radial load on it.

$$F_i = \frac{0.47F_r}{K} \qquad (11\text{–}18)$$

K is found from catalogs.

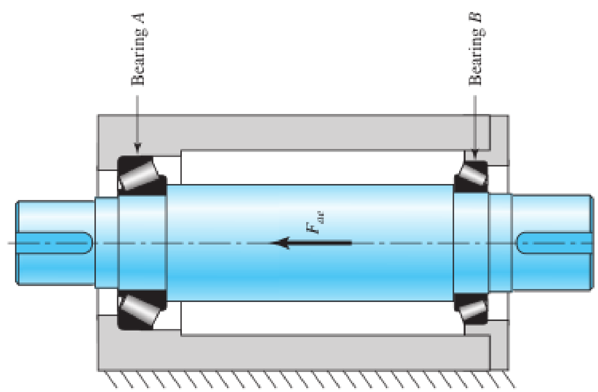
The process to determine the equivalent radial loads F_{eA} and F_{eB}:

- The pair will be labelled A and B;
- Bearing A is the one being "squeezed" by F_{ae}. Label the other at B;
- The equivalent radial loads F_{eA} and F_{eB} will then be determined by (Eq. 11-19) or (Eq. 11-20)

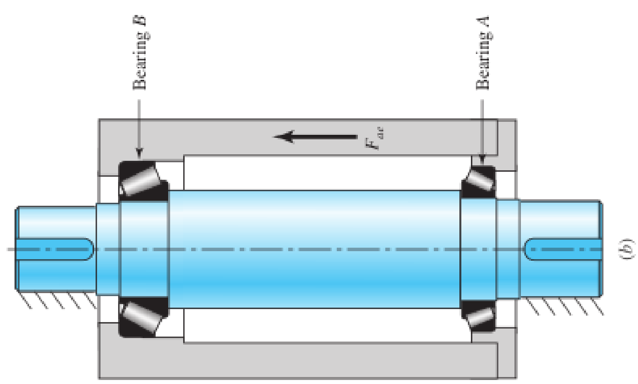
$$\text{If} \quad F_{iA} \le (F_{iB} + F_{ae}) \quad \begin{cases} F_{eA} = 0.4F_{rA} + K_A(F_{iB} + F_{ae}) & (11\text{–}19a) \\ F_{eB} = F_{rB} & (11\text{–}19b) \end{cases}$$

$$\text{If} \quad F_{iA} > (F_{iB} + F_{ae}) \quad \begin{cases} F_{eB} = 0.4F_{rB} + K_B(F_{iA} - F_{ae}) & (11\text{–}20a) \\ F_{eA} = F_{rA} & (11\text{–}20b) \end{cases}$$

Which bearing is bearing A? Figures 11-17 and 11-19:



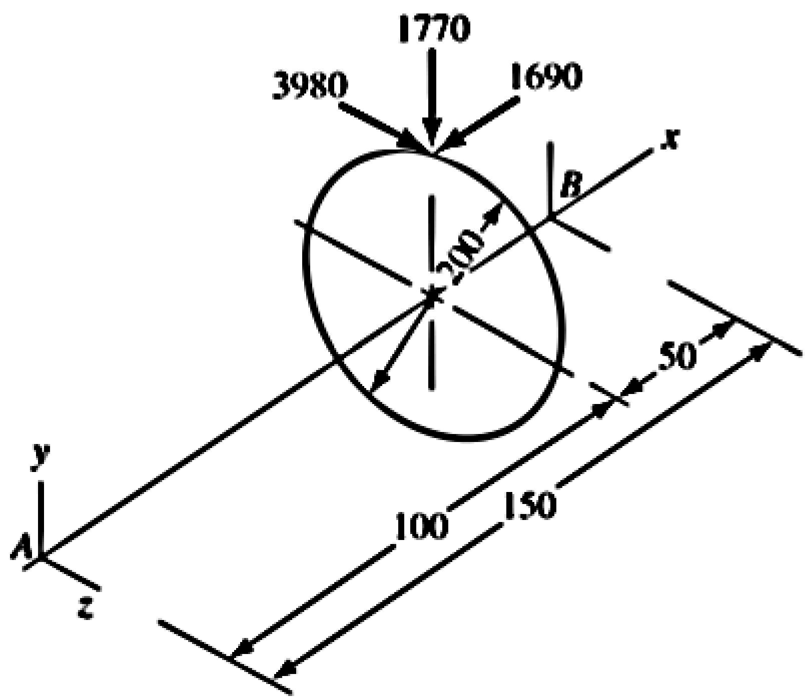
Shaft is moving, housing stationary.



Shaft is stationary, housing is moving.

- The above discussion re: induced axial load is applicable to angular-contact ball bearings. Methodology of finding F_i and F_e can be found form catalogs or manufacturers.

Example 11-8
The shaft runs at 800 rpm and is supported by two direct mounted taper-roller bearings. The design life of the bearings is to be 5000 hours. The helical gear mounted on the shaft is subject to tangential, radial, and axial loads. The reliability of the pair of bearings is set to 99% and k_a is 1.



Select suitable Timken tapered-roller bearings from Figure 11-15. Would the reversal of direction of the shaft's rotating require smaller/larger bearings?

<u>Solution:</u>
(1) Current direction of rotation

Assume 150-mm is the effective span; proceed with FBD's and finding support reactions. The vector sums of the support reactions are:

$R_A = 2170\ N$
$R_B = 2654\ N$
$F_{ae} = 1690\ N$

Also, $L_D = (800)(60)(5000) = 240(10^6)$ revs, and $L_{10} = (500)(60)(3000) = 90(10^6)$ revs

Reliabilities of the individual bearings are $\sqrt{0.99} = 0.995$.

Factor a_1 is then 0.175.

All bearings in Figure 11-15 have a bore of 25 mm. Select 07096/07196. It has $C_{10} = 6{,}990\ N$ and $K = 1.45$.

From (Eq. 11-18):

$$F_i = \frac{0.47F_r}{K} \qquad (11\text{–}18)$$

$$F_{iA} = \frac{(0.47)(2170)}{(1.45)} = 703.4\ N$$

$$F_{iB} = \frac{(0.47)(2654)}{(1.45)} = 860.3\ N$$

Since $F_{iA} \leq F_{iB} + F_{ae}$, (Eq. 11-19) is used to determine F_e.

$$\begin{Bmatrix} F_{eA} = 0.4F_{rA} + K_A(F_{iB} + F_{ae}) \\ F_{eB} = F_{rB} \end{Bmatrix}$$

Therefore, $F_{eA} = (0.4)(2170) + (1.45)(860.3 + 1690) = 4566\ N$
$F_{eB} = 2654\ N$

Apply basic bearing equation for C_{10}

$$\frac{C_{10}}{k_a F_D} = \left(\frac{L_D}{a_1 L_{10}}\right)^{\frac{1}{a}}$$
$$\frac{C_{10}}{(1)(4566)} = \left(\frac{(240)(10^6)}{(0.175)(90)(10^6)}\right)^{\frac{3}{10}}$$

The result is $C_{10} = 10{,}337\ N$. So, 07096/07196 is not sufficient.

Reselect 15101/15243. $C_{10} = 12{,}100\ N$, and $K = 1.67$. Repeat above calculations:

$$F_{iA} = (0.47)\left(\frac{2170}{1.67}\right) = 519.8\ N$$
$$F_{iB} = (0.47)\left(\frac{2654}{1.67}\right) = 614.1\ N$$
$$F_{eA} = (0.4)(2170) + (1.67)(614.1 + 1690) = 4716\ N$$
$$F_{eB} = 2654\ N$$

And $C_{10} = 10{,}677\ N$, which is less than 12,100 N.

For reliability assessment, recall

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

So, bearing A has $a_1 = 0.115$, $R = 99.8$; bearing B has $a_1 = 0.0169$. Since $a_1 < 0.05$, $R \geq 99.9$; combined reliability is $(0.998)(0.999) = 0.997$, which is more than 0.99.

Therefore, 15101/15243 is sufficient for both locations.

(2) Reversal in direction of rotation

Re-label the bearings as L(eft) (used to be bearing A) and R(ight) (used to be bearing B)

FBD and calculations give rise to $R_L = 1431\ N$ and $R_R = 3516\ N$.

Bearings are 15101/15243 with $C_{10} = 12{,}100\ N$ and $K = 1.67$.

$$F_{iL} = (0.47)\left(\frac{1431}{1{,}67}\right) = 402.7\ N$$
$$F_{iR} = (0.47)\left(\frac{3516}{1.67}\right) = 989.5\ N$$

Now because the right bearing s being “squeezed”.

Because $F_{iR} \leq F_{iL} + F_{ae}$, therefore (Eq. 11-19) gives

$F_{eR} = (0.4)(3516) + (1.67)(402.7 + 1690) = 4901\ N$, and $F_{eL} = 1431\ N$

Calculating the required C_{10} for the right bearing

$$\frac{C_{10}}{(1)(4901)} = \left(\frac{(240)(10^6)}{(0.175)(90)(10^6)}\right)^{\frac{3}{10}}$$

Then $C_{10} = 11{,}096\ N$, which is less than the catalog’s $12{,}100\ N$.

So, the selection is suitable for opposite direction of the shaft’s rotation.

Or, assessing the bearings’ reliabilities; the left bearing has $a_1 = 0.00217$, so $R \geq 99.9$; the right bearing has $a_1 = 0.131$, $R = 99.7$; As a result, the combined reliability is $(0.999)(0.997) = 0.997$, which is more than 0.99.

So, the 15101/15243 bearings can be used in either direction of rotation.