Chapter 11
Rolling Contact Bearing

- Bearing Types
- Bearing Life
- Bearing Load
- Bearing Survival
- The Reliability Goal
- Selection of Ball and Straight Roller Bearings
- Selection of Tapered Roller Bearings

Introduction

- A **Bearing** is a device to permit constrained relative motion between two parts, typically rotation or linear movement.
- Objective of bearing:

  To provide relative positioning and rotational freedom while transmitting a load between two parts. Example: a shaft and its housing.
- Objective of lubrication: To reduce:

  1. The friction
  2. The wear
  3. The heat between two surfaces moving relative to each other.

Classification of Bearings

Bearings are classified depending upon the load.

Bearings are also classified depending upon the type of contact.

1. Sliding contact bearing
   - journal bearing
   - plane bearing
2. Antifriction bearing (Rolling contact bearing)
   - Ball bearing
   - roller bearing

Function of Bearings

- A bearing permits relative motion between two machine members while minimizing frictional resistance.
- A bearing consists of an inner and outer member separated either by a thin film of lubricant or a rolling element.
- A bearing bears the load.
- It locates the moving parts in correct position.
- It provides free motion to the moving part by reducing friction.

Rolling Contact Bearing

- Rolling contact bearings are also known as antifriction bearings.
- The load, speed, and operating viscosity of the lubricant affect the friction characteristics of a rolling bearing.
- These bearings provide coefficients of friction between 0.001 and 0.002.
- The designer must deal with such matters as fatigue, friction, heat, lubrication, kinematics problems, material properties, machining tolerances, assembly, use and cost.

Rolling Contact bearing…

- Advantages of rolling contact bearing:
  1. Low starting and good operating friction torque.
  2. Ease of lubrication
  3. Requiring less axial space.
  4. Generally, taking both radial and axial loads.
  5. Rapid replacement
  6. Warning of impending failure by increasing noisiness.
  7. Good low-temperature starting.

Rolling Contact bearing…

- Disadvantages of rolling –element bearings:

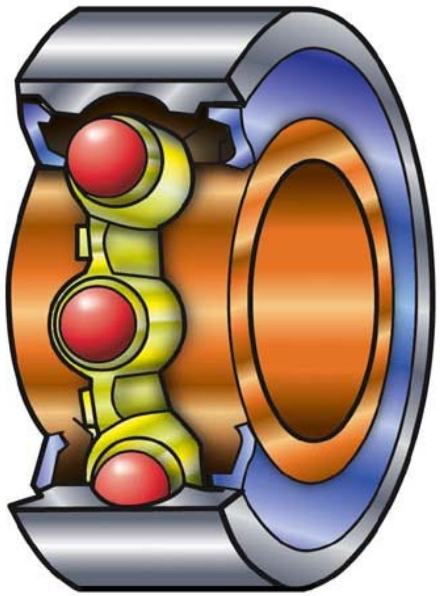
1. Greater diametral space.
2. More severe alignment requirements.
3. Higher initial cost.
4. Noisier normal operation.
5. Finite life due to eventual failure by fatigue.
6. Ease of damage by foreign matter.
7. Poor damping ability.

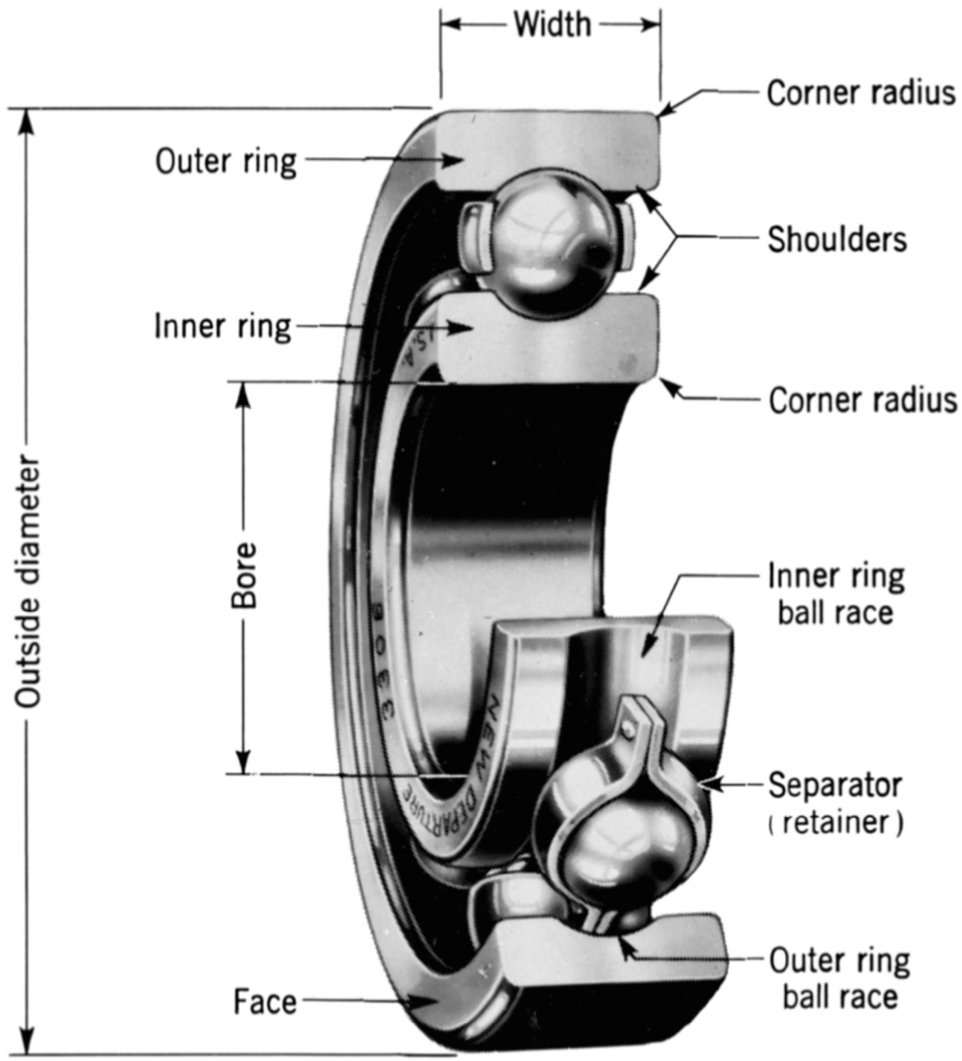
Bearing types

- Most rolling bearings are categorized in one of the three groups:
  1. Pure radial loads
  2. Pure thrust loads (axial loads)
  3. Combination of the two kinds of loads

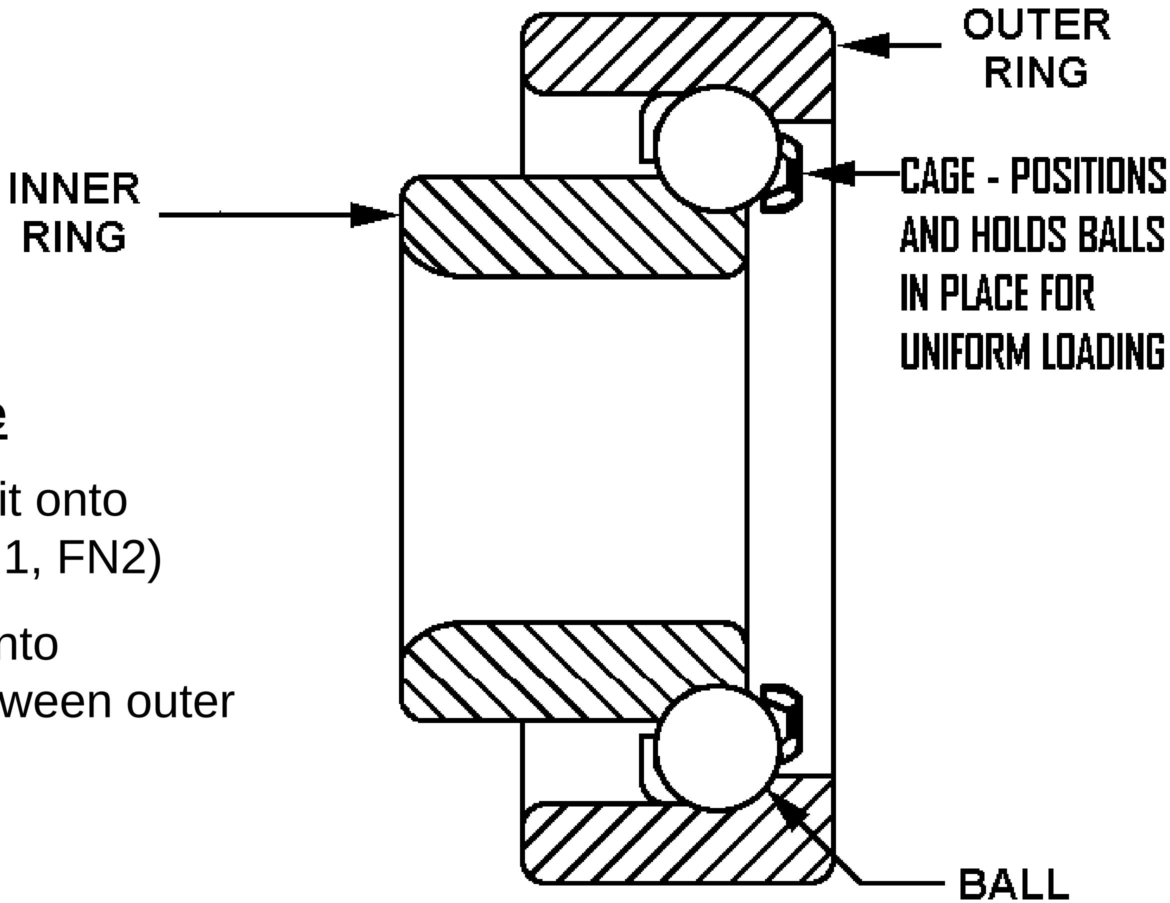
- There are two types of rolling bearings:
  1. Ball bearings
  2. Roller bearings.

Ball Bearing





Ball Bearings



How to Assemble

-Inner race press fit onto shaft shoulder (FN1, FN2)

-Assembly slides into housing (RC2) between outer race and housing

Types of Ball Bearings

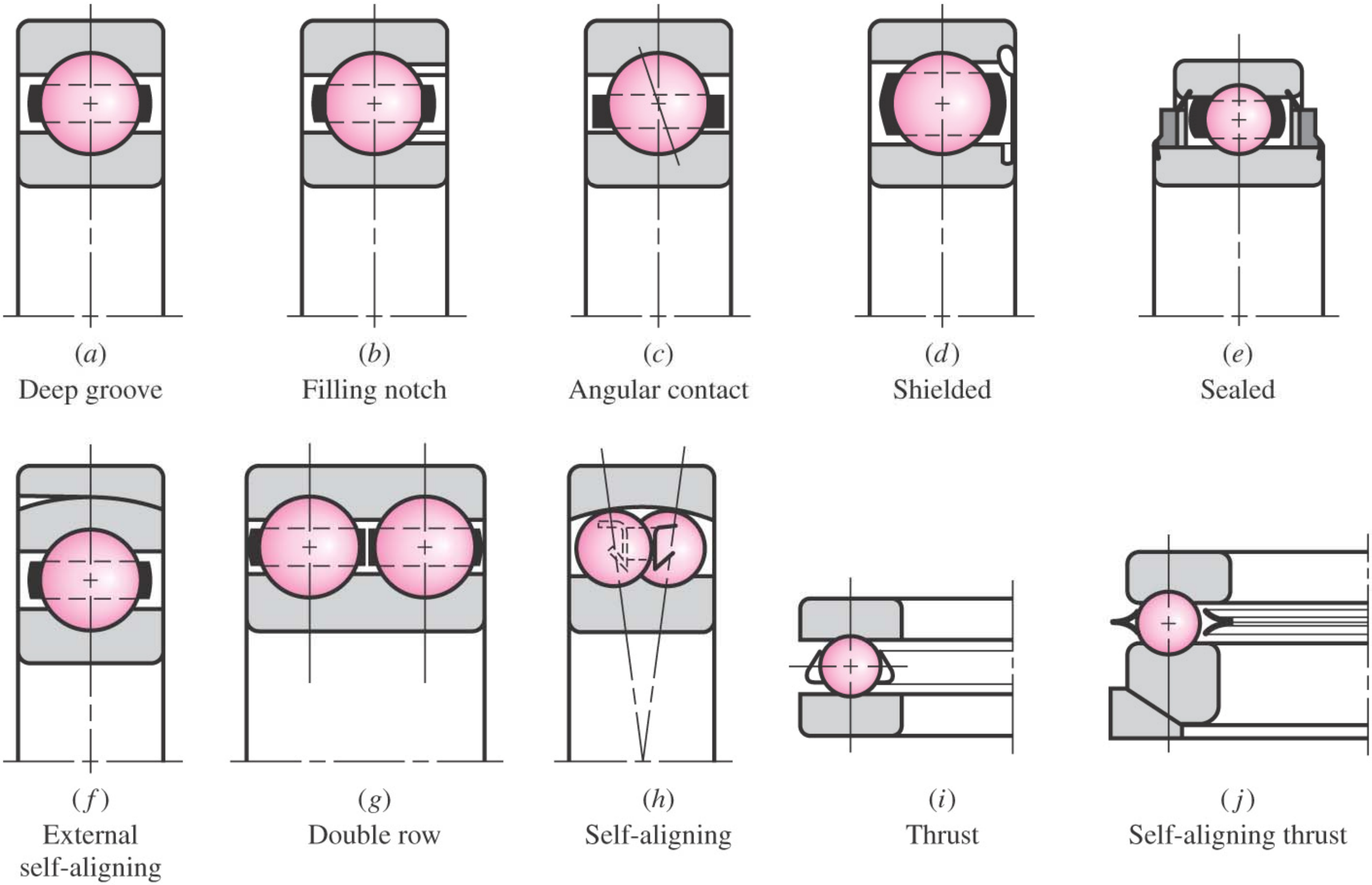
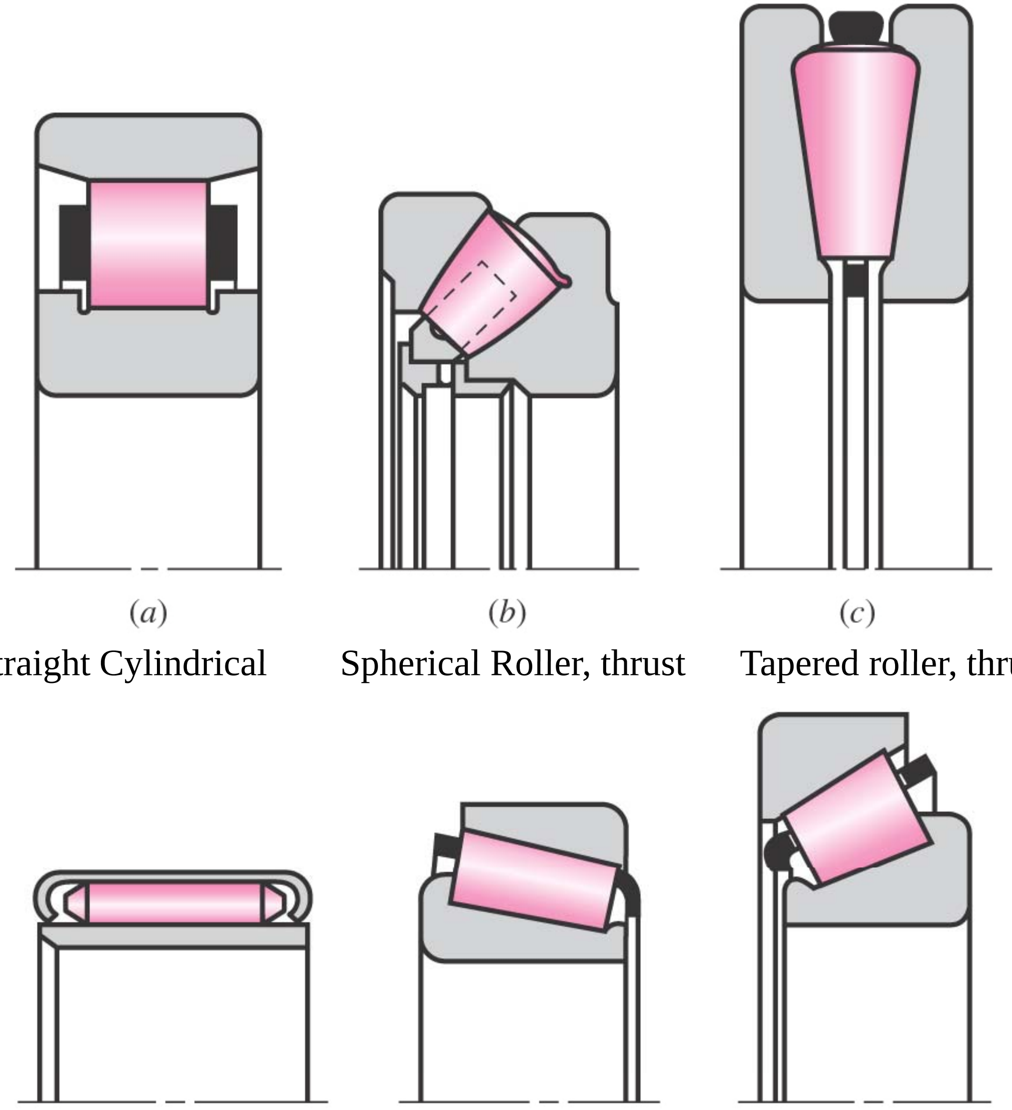


Fig. 11–2

Types of Roller Bearings



(a) Straight Cylindrical
(b) Spherical Roller, thrust
(c) Tapered roller, thrust
(d) Needle
(e) Tapered roller
(f) Steep-angle tapered roller

Fig. 11–3

All these bearings may be obtained with shields on one or both sides. The shields are not a complete closure but do offer a measure of protection against dirt.
The spherical-roller thrust is useful where heavy loads and misalignment occur. Needle bearings are very useful where radial space is limited.
Tapered roller bearings combine the advantages of ball and straight roller bearings.

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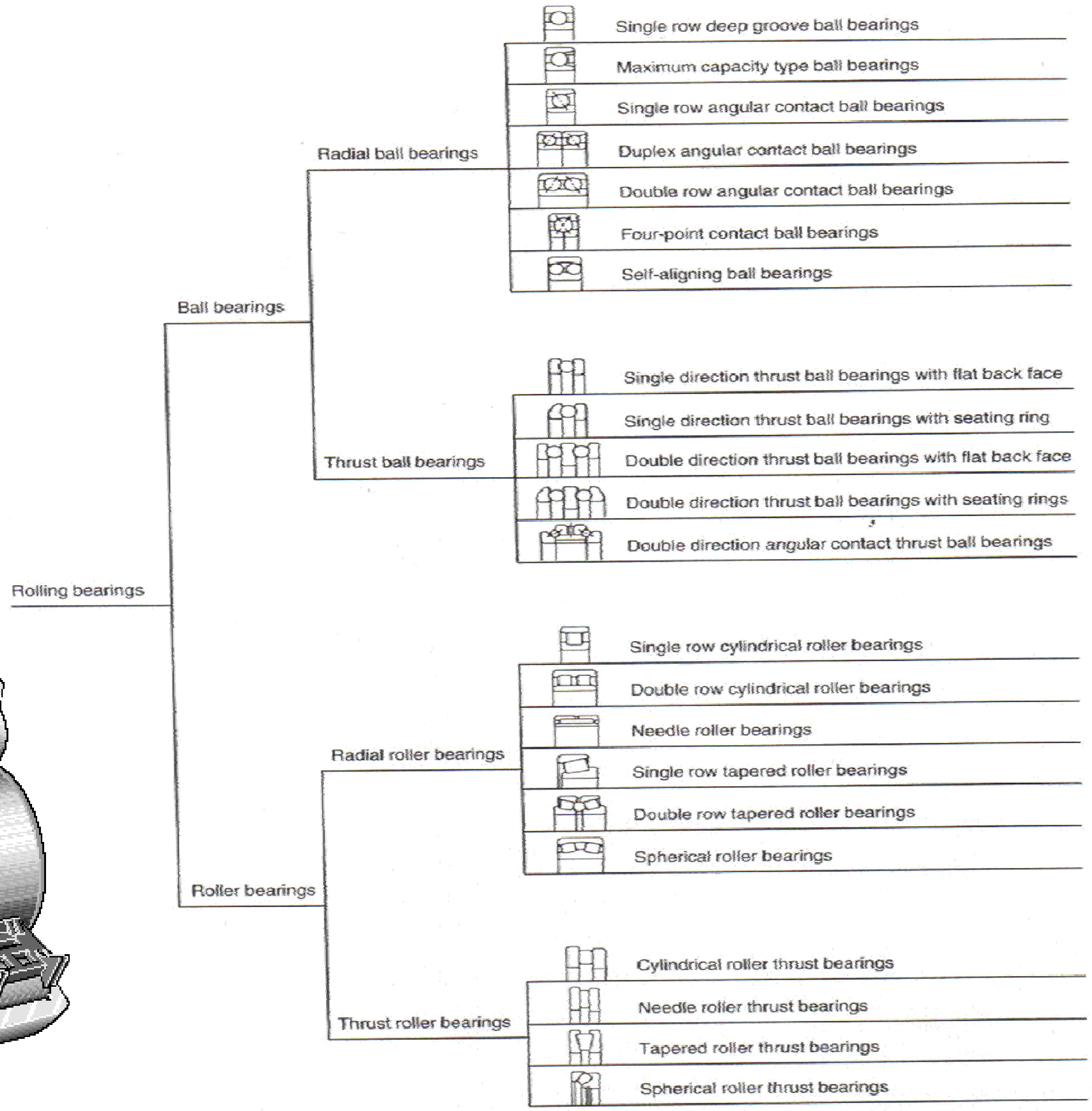

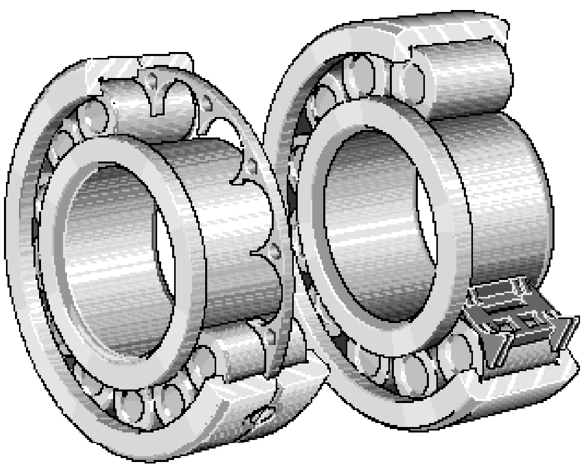

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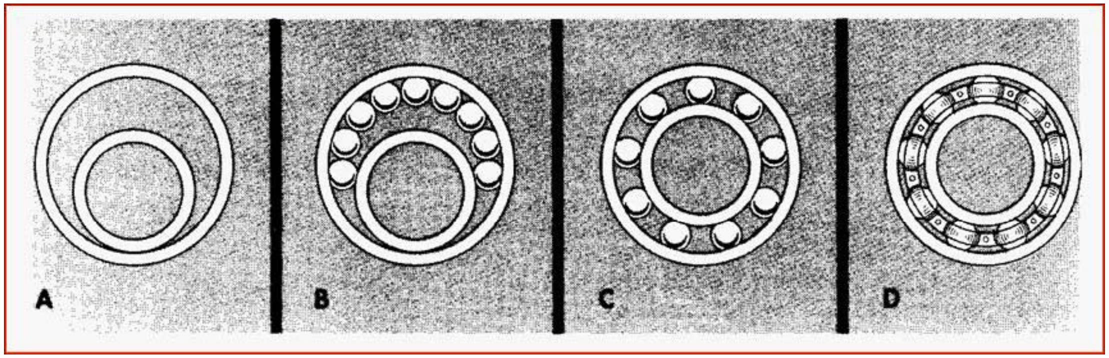
Ball Bearings

1. Deep groove (Conrad) bearing

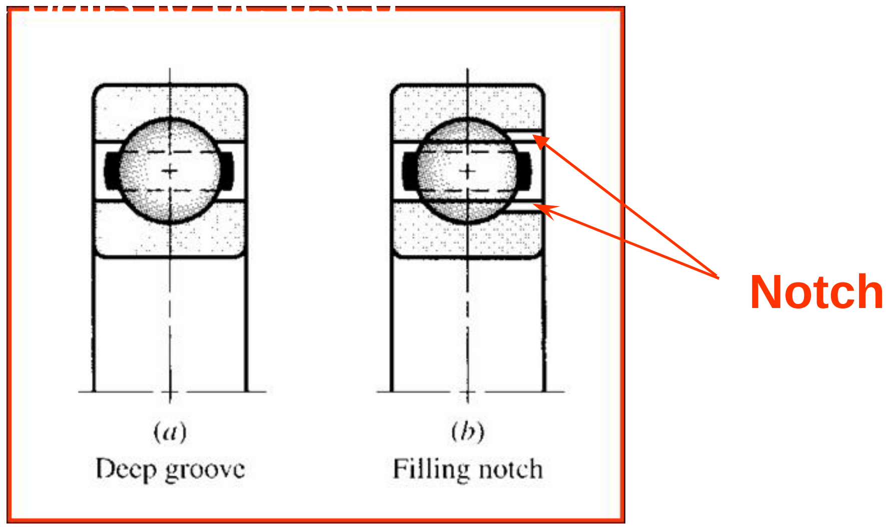


Load capacity is limited by the number of balls

Primarily designed to support radial loads, the thrust capacity is about 25% of radial load capacity

Ball Bearings

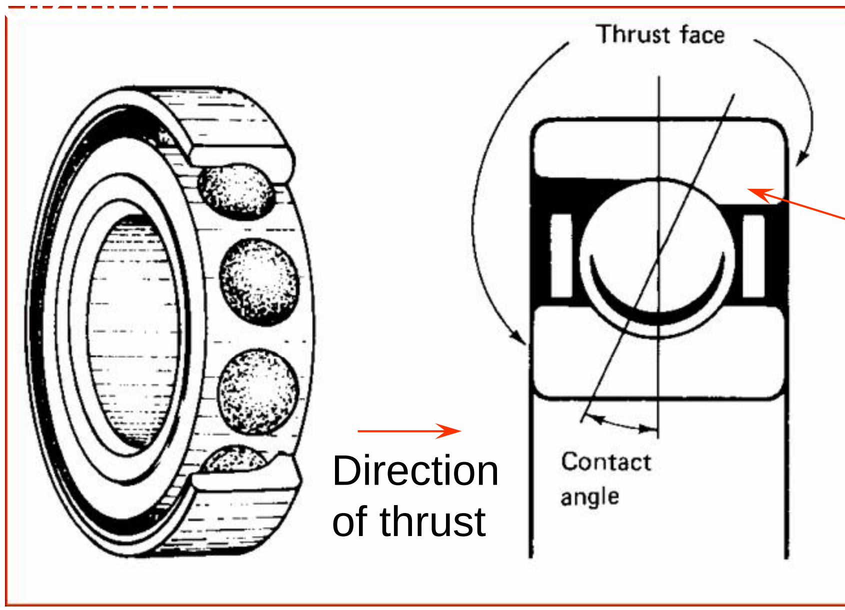
2. Filling notch or maximum capacity ball bearings



- Thrust load capacity drops to 70% (2 directions) of radial load capacity.

Ball Bearings

3. Angular contact bearings (AC)



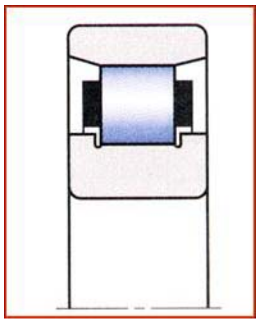
Extra support
in the back

Used for high radial and thrust load applications

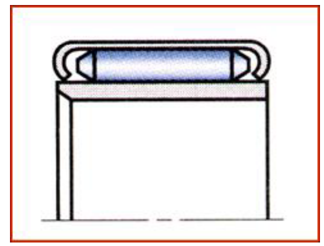
Roller Bearings

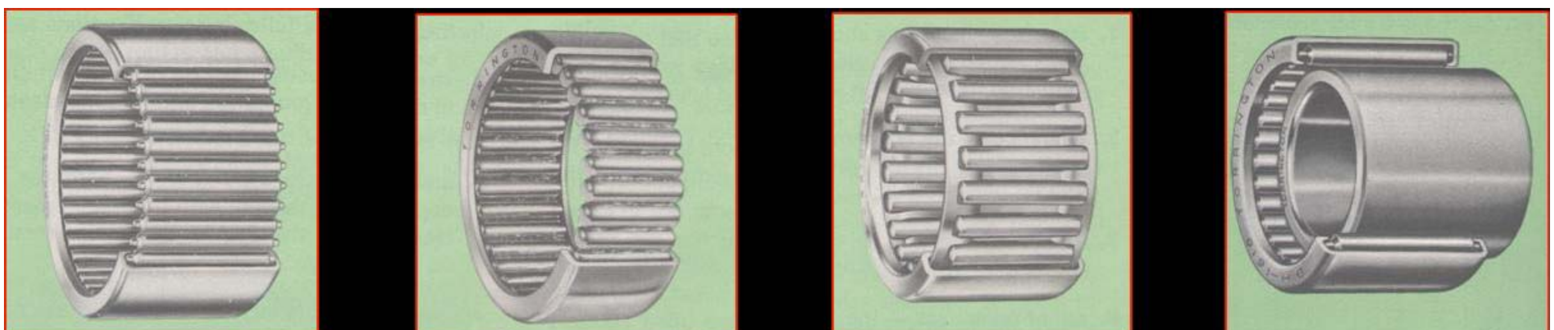
Roller bearings have higher load capacity than ball bearings, load is transmitted through line contact instead of point

Straight cylindrical roller



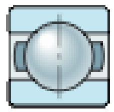
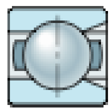
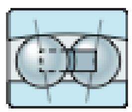
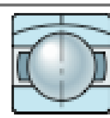
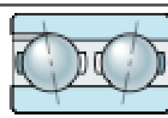
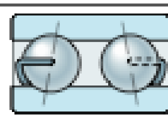
Needle type





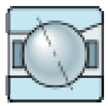
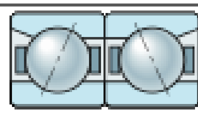
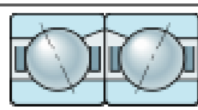
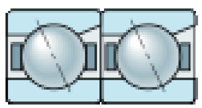
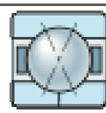
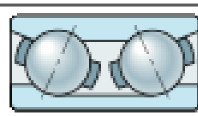
Mechanically retained rollers | Greased retained rollers | Caged | With inner race

Radial Ball Bearings

Type		Approximate range of bore sizes, mm		Relative capacity		Limiting speed factor	Tolerance to mis alignment
		Mini mum	Maxi mum	Radial	Thrust		
Conrad or deep groove		3	1060	1.00	[a]0.7	1.0	±0° 15′
Maximum capacity or filling notch		10	130	1.2–1.4	[a]0.2	1.0	+0° 3′
Self aligning, internal		5	120	0.7	[b]0.2	1.0	±2° 30′
Self aligning, external				1.0	[a]0.7	1.0	High
Double row, maximum		6	110	1.5	[a]0.2	1.0	+0° 3′
Double row, deep groove		6	110	1.5	[a]1.4	1.0	0°

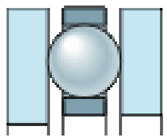
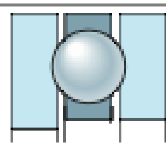
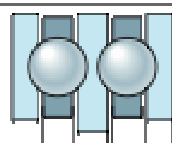
Characteristics of representative radial ball bearings.

Angular-Contact Ball Bearings

Type		Approximate maximum bore size, mm	Relative capacity		Limiting speed factor	Tolerance to mis-alignment
			Radial	Thrust		
One-directional thrust		320	[b]1.00-1.15	[a,b]1.5-2.3	[b]1.1-3.0	±0°2′
Duplex, back to back		320	1.85	[c]1.5	3.0	0°
Duplex, face to face		320	1.85	[c]1.5	3.0	0°
Duplex, tandem		320	1.85	[d]2.1	3.0	0°
Two directional or split ring		110	1.15	[c]1.5	3.0	±0°2′
Double row		140	1.5	[c]1.85	0.8	0°

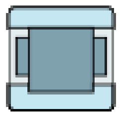
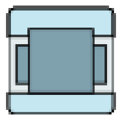
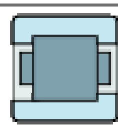
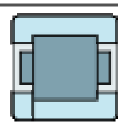
Characteristics of representative angular-contact ball bearings.

Thrust Ball Bearings

Type		Approximate range of bore sizes, mm		Relative thrust	Limiting speed	Tolerance to mis-
		Minimum	Maximum			
One directional, flat race		6.45	88.9	[a]0.7	0.10	[b]0°
One directional, grooved race		6.45	1180	[a]1.5	0.30	0°
Two directional, grooved race		15	220	[c]1.5	0.30	0°

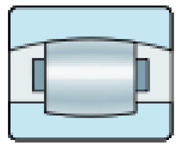
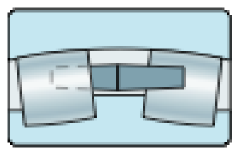
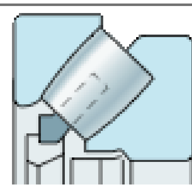
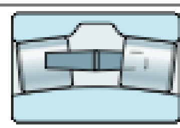
Characteristics of representative thrust ball bearings.

Cylindrical Roller Bearings

Type		Approximate range of bore sizes, mm: Minimum	Approximate range of bore sizes, mm: Maximum	Relative capacity: Radial	Relative capacity: thrust	Limiting speed factor	Tolerance to misalignment
Separable outer ring, nonlocating (N)		10	320	1.55	0	1.20	±0°5′
Separable inner ring, nonlocating (NU)		12	500	1.55	0	1.20	±0°5′
Separable inner ring, one-direction locating (NJ)		12	320	1.55	[a]Locating	1.15	±0°5′
Separable inner ring, two-direction locating		20	320	1.55	[b]Locating	1.15	±0°5′

Characteristics of representative cylindrical roller bearings.

Spherical Roller Bearings

Type	Approximate range of bore sizes, mm: Mini mm	Approximate range of bore sizes, mm: Maxi mm	Relative capacity: Radial	Relative capacity: Thrust	Limiting speed factor	Tolerance to mis alignment
Single row, barrel or convex 	20	320	2.10	0.20	0.50	±2°
Double row, barrel or convex 	25	250	2.40	0.70	0.50	±1°30′
Thrust 	85	360	[a]0.10 [b]0.10	[a]1.80 [b]2.40	0.35 0.50	±3°
Double row, concave 	50	130	2.40	0.70	0.50	±1°30′

[a]Symmetric rollers.
[b]Asymmetric rollers.

Characteristics of representative spherical roller bearings.

- When the ball or roller of rolling-contact bearings rolls, contact stresses occur on the inner ring, the rolling element, and on the outer ring.
- Common life measures are
  - **Number of revolutions until the first tangible evidence of fatigue**
  - **Number of hours of until the first tangible evidence of fatigue**
- Under ideal conditions, the fatigue failure consists of spalling of the load-carrying surfaces. The American Bearing Manufacturers Association (ABMA) standard states that the failure criterion is the first evidence of fatigue. The fatigue criterion used by the Timken Company laboratories is the spalling or pitting of an area of 0.01 in^2.
- The rating life of a group of nominally identical ball or roller bearings is defined as the number of revolutions (or hours at a constant speed) that 90 percent of a group of bearings will achieve or exceed before the failure criterion develops also termed as minimum life, L_{10} life, and B_{10} life.
- The most commonly used rating life is 10^6 revolutions.

Bearing life

- Contact stresses occur on the inner ring, the rolling element, and on the outer ring, when the ball or roller of rolling-contact bearings rolls.
- If:
  - Bearing is clean and properly lubricated
  - Mounted and sealed against the entrance of dust and dirt,
  - Maintained in this condition
  - Operated at reasonable temperatures.

  Then:

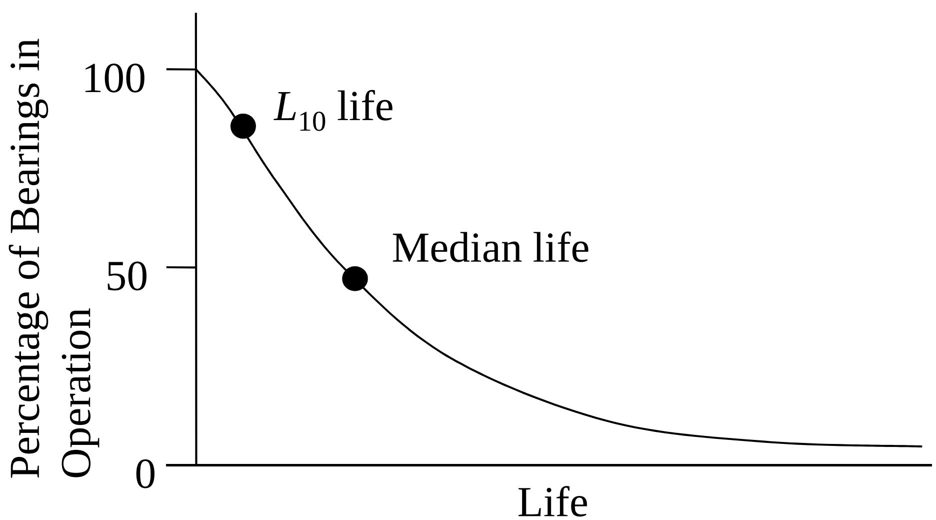
  The only cause of failure is metal fatigue.

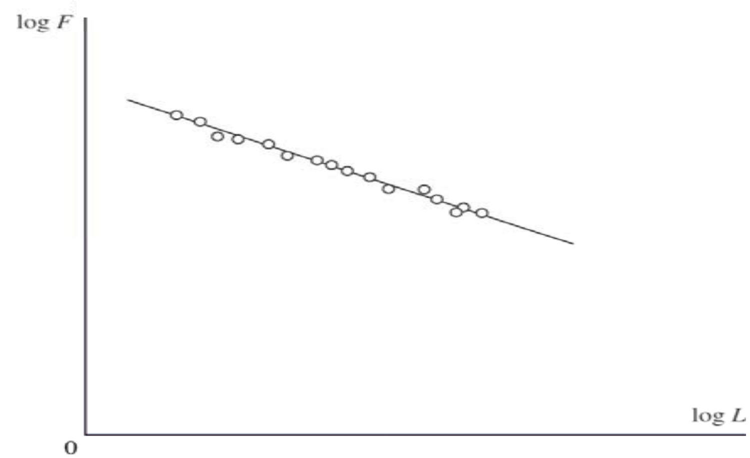
Bearing Life…

- American Bearing Manufacturers Association (ABMA) established the following definitions associated with the life of bearing.
  - **Bearing life**: the number of revolutions or hours at some uniform speed at which the bearing operates until fatigue failure.
  - **Rating life L_{10}**: The number of revolutions (or hours at a uniform speed) that 90% of a group of identical roller bearings will complete or exceed before the first evidence of fatigue develops.
  - **Median life**: refers to the life that 50% of the group of bearings would complete or exceed. It is about 5 times of L_{10} life.

❑ <u>The basic load rating C</u> is defined as: the constant radial load which a group of apparently identical bearings can endure for a rating life of one million revolutions of the inner ring (stationary load and stationary outer ring)

- The definition of the rating life L_{10} is based on a 90% reliability(or 10% failure)
- A typical curve of bearing life expectancy is shown in the figures





Load Rating Definitions

- *Catalog Load Rating*, C_{10}: Constant radial load that causes 10% of a group of bearings to fail at the bearing manufacturer's rating life.
  - Depends on type, geometry, accuracy of fabrication, and material of bearing
  - Also called Basic Dynamic Load Rating, and Basic Dynamic Capacity
- *Basic Load Rating, C*: A catalog load rating based on a rating life of 10^6 revolutions of the inner ring.
  - The radial load that would be necessary to cause failure at such a low life is unrealistically high.
  - The Basic Load Rating is a reference value, not an actual load.

Bearing Load Life at Rated Reliability

- The load-life function at 0.90 reliability using a regression equation is

$$FL^{1/a} = \text{constant}$$

Where
a = 3 for ball bearings
a = 10/3 for roller bearings (cylindrical and tapered roller)

- A catalog load rating is defined as the radial load that causes 10 percent of a group of bearing to fail at the bearing manufacturer's rating life.
- In selecting a bearing for a given application, it is necessary to relate the desired load and life requirements to the published catalog load rating corresponding to the catalog rating life.

$$F_R(\mathcal{L}_R n_R 60)^{1/a} = F_D(\mathcal{L}_D n_D 60)^{1/a}$$

catalog rating, lbf or kN
rating life in hours
rating speed, rev/min
desired speed, rev/min
desired life, hours
desired radial load, lbf or kN

- The expression for a catalog load rating as a function of the desire load, desired life, and catalog rating life is then

$$C_{10} = F_R = F_D\left(\frac{L_D}{L_R}\right)^{1/a} = F_D\left(\frac{\mathcal{L}_D n_D 60}{\mathcal{L}_R n_R 60}\right)^{1/a}$$

Load Rating Definitions

- *Static Load Rating, C_o*:
  Static radial load which corresponds to a permanent deformation of rolling element and race at the most heavily stressed contact of 0.0001*d*.
  - *d* = diameter of roller
  - Used to check for permanent deformation
  - Used in combining radial and thrust loads into an equivalent radial load
- *Equivalent Radial Load, F_e*:
  Constant stationary load applied to bearing with rotating inner ring which gives the same life as actual load and rotation conditions.

Bearing Load Life at Rated Reliability

- Experiments show that two groups of identical bearings tested under different loads F_1 and F_2 will have respectively lives L_1 and L_2 according to the relation:

$$\frac{L_1}{L_2} = \left(\frac{F_2}{F_1}\right)^a \qquad (11\text{-}1)$$

Where

L = the life of millions of revolutions or the life of hours at a given constant speed n, in rev/min.

a = 3 for ball bearings

10/3 for roller bearings (cylindrical and tapered roller)

- A bearing manufacturer may choose a rated cycle value of 10^6 revolutions
- Timken Company choose a rate of $90(10^6)$ revolutions which is based on 3000 hours at a speed of 500 rev/min.

$$L_{10} = (3000h)(60\,\mathrm{min}/h)(500 rev/\mathrm{min}) = 90(10^6) rev.$$

- Equation 11-1 can be written in more useful way as:

$$FL^{1/a} = const \qquad (11\text{-}2)$$

- Basic load rating - Catalog load rating C_{10}, denotes the tenth percentile rating life for a particular bearing in the catalog.
- From equation (11-2)

$$F_1 L_1^{1/a} = F_2 L_2^{1/a}$$

- Put L_1 as L_{10} and F_1 as C_{10}, we can write the above equation as:

$$C_{10} L_{10}^{1/a} = F L^{1/a}$$

- Finally, we can write the equation in the general form as:

$$C_{10} = F_D \left(\frac{L_D n_D 60}{L_R n_R 60} \right)^{1/a} \qquad (10\text{-}3)$$

Where

C_{10}: catalog rating (lbf or kN)

L_R: rating life (hours)

n_R: rating speed (rev/min)

F_D: desired radial load (lbf or kN)

L_D: desired life (hours)

n_D: desired speed (rev/min)

Bearing Survival: Reliability Vs Life

- If the machine is assembled with a total of N bearings, each having the same reliability R, then the reliability of the group must be:

$$R_N = R^N$$

- Suppose we have a gear-reduction unit consisting of six bearings, all loaded so that the L_{10} lives are equal.
- For example, if the reliability of each bearing is 90%, the reliability of all the bearings in the assembly is:

$$R_6 = (0.90)^6 = 0.531$$

This points up the need to select bearings having reliabilities greater than 90%

Reliability based on Weibull distribution

Ball Bearing

$$R = \exp\left[-\left(\frac{L/L_{10} - 0.02}{4.439} \right)^{1.483} \right]$$

$$L_{10} = \frac{L}{0.02 + 4.439\left[\ln(1/R)\right]^{1/1.483}}$$

$$C_{10} = F_D \left\{ \frac{(L_D n_D / L_R n_R)}{0.02 + 4.439\left[\ln(1/R)\right]^{1/1.483}} \right\}^{1/3}$$

Roller Bearing

$$R = \exp\left[-\left(\frac{L/L_{10}}{4.48} \right)^{1.5} \right]$$

$$L_{10} = \frac{L}{4.48\left[\ln(1/R)\right]^{1/1.5}}$$

$$C_{10} = F_D \left\{ \frac{(L_D n_D / L_R n_R)}{4.48\left[\ln(1/R)\right]^{1/1.5}} \right\}^{3/10}$$

$$C_{10} = F_D \left(\frac{L_D n_D 60}{L_R n_R 60} \right)^{1/a}$$

- Example:

  A certain application requires a bearing to last for 1800 h with a reliability of 99%. What should be the rated life of the bearing selected for this application?

- Solution:

  Given: L=1800 h , R=0.99

  Required: L_{10}

  For Ball Bearing

$$L_{10} = \frac{L}{0.02 + 4.439\left[\ln(1/R)\right]^{1/1.483}} = 8197h$$

  For Tapered Bearing

$$L_{10} = \frac{L}{4.48\left[\ln(1/R)\right]^{1/1.5}} = 8627h$$

- Example

  A roller bearing is to be selected to withstand a radial load of 4 kN and have an L_{10} life of 1200 h at a speed of 600 rev/min. What load rating would you look for in searching the Timken Engineering Journal?

- Solution

  Given:

  F_D=4kN, L_D=1200 h, n_D=600 rev/min,

  L_R=3000 h, n_R=500 rev/min,

  Roller bearing $\Rightarrow a = 10/3$

  Required: load rating C_{10}

$$C_{10} = F_D\left(\frac{L_D n_D 60}{L_R n_R 60}\right)^{1/a} = 4\left(\frac{(1200)(600)(60)}{(3000)(500)(60)}\right)^{3/10} = 3.21kN$$

- Example

  What load rating would be used if the application in the pervious example is to have a reliability of 99 percent?

- Solution:

  From equation 11-10:

$$C_{10} = F_D \left\{ \frac{\left(L_D n_D / L_R n_R\right)}{4.48\left[\ln\left(1/R\right)\right]^{1/1.5}} \right\}^{3/10} = 4\left\{ \frac{\left((1200)(600)/(3000)(500)\right)}{4.48\left[\ln(1/0.99)\right]^{1/1.5}} \right\}^{3/10} = 5.136kN$$

  This value let us enter the catalog with C_{10}=5.136 kN

EXAMPLE 11–3 The design load on a ball bearing is 1840 N and an application factor of 1.2 is appropriate. The speed of the shaft is to be 300 rev/min, the life to be 30 kh with a reliability of 0.99. What is the C_{10} catalog entry to be sought (or exceeded) when searching for a deep-groove bearing in a manufacturer's catalog on the basis of 10^6 revolutions for rating life? The Weibull parameters are $x_0 = 0.02$, $(\theta - x_0) = 4.439$, and $b = 1.483$.

Solution

$$x_D = \frac{L}{L_{10}} = \frac{60L_D n_D}{60L_R n_R} = \frac{60(30\,000)300}{10^6} = 540$$

Thus, the design life is 540 times the L_{10} life. For a ball bearing, $a = 3$. Then, from Eq. (11–7),

Answer

$$C_{10} = (1.2)(1.84)\left[\frac{540}{0.02 + 4.439(1 - 0.99)^{1/1.483}}\right]^{1/3} = 29.7 \text{ kN}$$

11.6 Selection of Ball and Straight Roller Bearings

- Ball bearings are usually operated with some combination of radial and thrust load.
- Since, catalog ratings are based only on radial load, it is convenient to define an equivalent radial load F_e that will have the same effect on bearing life as do the applied loads

- An equivalent ABMA radial load equation for ball bearings is the maximum of the two values:
- $F_e = VF_r$
- $F_e = XVF_r + YF_a$
- Where

F_a : applied axial thrust load

F_r : applied radial load

F_e : equivalent radial load

V : a rotation factor

X : a radial factor

Y : a thrust factor

- A rotation factor V is defined such that:

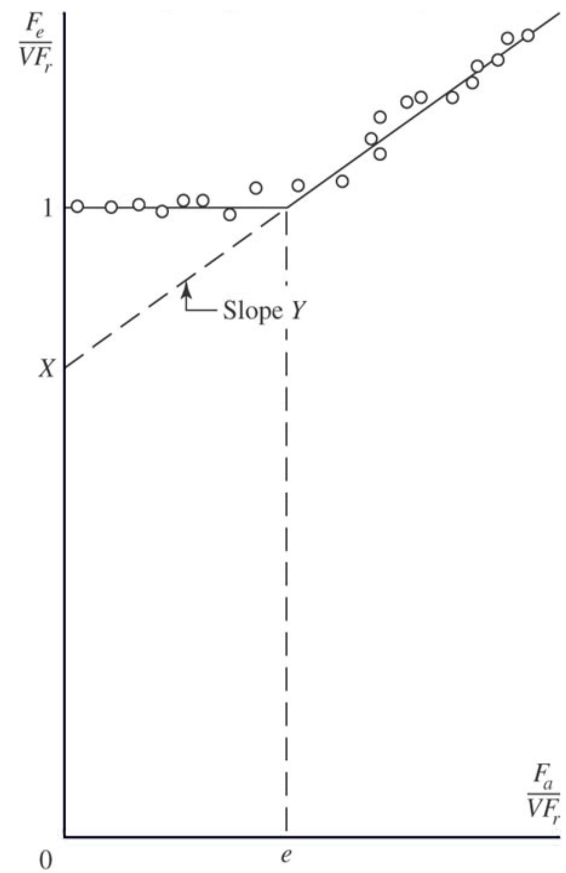
When The inner ring rotates $\Rightarrow V = 1.0$

The outer ring rotates $\Rightarrow V = 1.2$

-
- Let

$$\frac{F_e}{VF_r} = 1 \qquad when \quad \frac{F_a}{VF_r} \le e$$

$$\frac{F_e}{VF_r} = X + Y\frac{F_a}{VF_r} \quad when \quad \frac{F_a}{VF_r} > e$$



The pervious equations can be written in one single equation as:

$$F_e = X_i VF_r + Y_i F_a$$

$$where \quad \begin{matrix} i = 1 & when & \frac{F_a}{VF_r} \le e \\ i = 2 & when & \frac{F_a}{VF_r} > e \end{matrix}$$

Table 11–1

Equivalent Radial Load Factors for Ball Bearings

		$F_a/(VF_r) \le e$		$F_a/(VF_r) > e$	
F_a/C_0	e	X_1	Y_1	X_2	Y_2
0.014*	0.19	1.00	0	0.56	2.30
0.021	0.21	1.00	0	0.56	2.15
0.028	0.22	1.00	0	0.56	1.99
0.042	0.24	1.00	0	0.56	1.85
0.056	0.26	1.00	0	0.56	1.71
0.070	0.27	1.00	0	0.56	1.63
0.084	0.28	1.00	0	0.56	1.55
0.110	0.30	1.00	0	0.56	1.45
0.17	0.34	1.00	0	0.56	1.31
0.28	0.38	1.00	0	0.56	1.15
0.42	0.42	1.00	0	0.56	1.04
0.56	0.44	1.00	0	0.56	1.00

*Use 0.014 if $F_a/C_0 < 0.014$.

Table 11–2

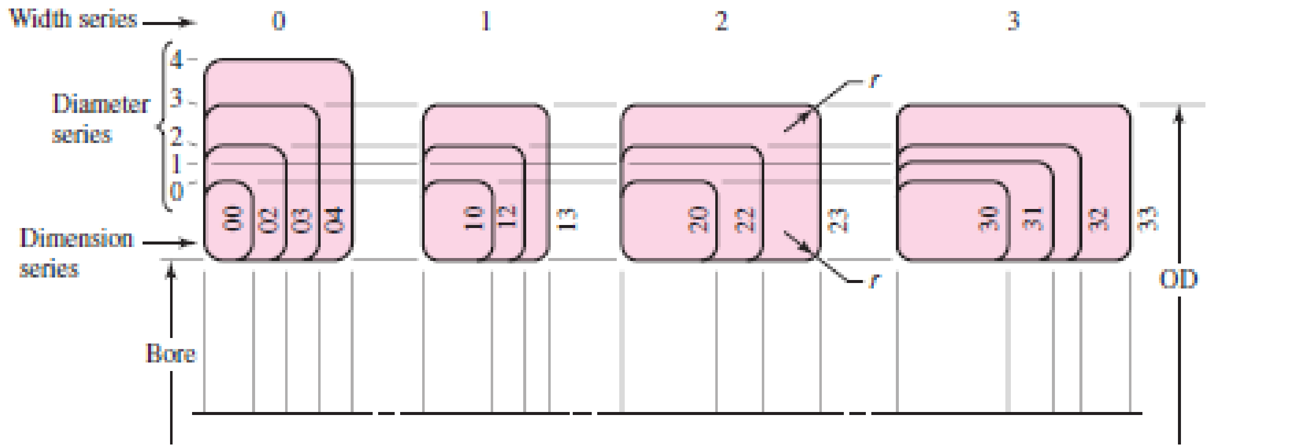
Dimensions and Load Ratings for Single-Row 02-Series Deep-Groove and Angular-Contact Ball Bearings

Bore, mm	OD, mm	Width, mm	Fillet Radius, mm	Shoulder Diameter, mm d_S	Shoulder Diameter, mm d_H	Load Ratings, kN Deep Groove C_{10}	Load Ratings, kN Deep Groove C_0	Load Ratings, kN Angular Contact C_{10}	Load Ratings, kN Angular Contact C_0
10	30	9	0.6	12.5	27	5.07	2.24	4.94	2.12
12	32	10	0.6	14.5	28	6.89	3.10	7.02	3.05
15	35	11	0.6	17.5	31	7.80	3.55	8.06	3.65
17	40	12	0.6	19.5	34	9.56	4.50	9.95	4.75
20	47	14	1.0	25	41	12.7	6.20	13.3	6.55
25	52	15	1.0	30	47	14.0	6.95	14.8	7.65
30	62	16	1.0	35	55	19.5	10.0	20.3	11.0
35	72	17	1.0	41	65	25.5	13.7	27.0	15.0
40	80	18	1.0	46	72	30.7	16.6	31.9	18.6
45	85	19	1.0	52	77	33.2	18.6	35.8	21.2
50	90	20	1.0	56	82	35.1	19.6	37.7	22.8
55	100	21	1.5	63	90	43.6	25.0	46.2	28.5
60	110	22	1.5	70	99	47.5	28.0	55.9	35.5
65	120	23	1.5	74	109	55.9	34.0	63.7	41.5
70	125	24	1.5	79	114	61.8	37.5	68.9	45.5
75	130	25	1.5	86	119	66.3	40.5	71.5	49.0
80	140	26	2.0	93	127	70.2	45.0	80.6	55.0
85	150	28	2.0	99	136	83.2	53.0	90.4	63.0
90	160	30	2.0	104	146	95.6	62.0	106	73.5
95	170	32	2.0	110	156	108	69.5	121	85.0

The ABMA has established standard boundary dimensions for bearings which define the bearing bore, the outside diameter OD, the width, and the fillet sizes on the shaft and housing shoulders.

Figure 11–7

The basic ABMA plan for boundary dimensions. These apply to ball bearings, straight roller bearings, and spherical roller bearings, but not to inch-series ball bearings or tapered roller bearings. The contour of the corner is not specified. It may be rounded or chamfered, but it must be small enough to clear the fillet radius specified in the standards.



recommendations are based on the ratio of the thrust component F_a to the *basic static*

The bearings are identified by a 2-digit number called the dimension-series code.

First digit : the width series

Second digit: the diameter series

Bearing series

- Three main series;
- extra light (100)
- light (200)--most common series for average applications;
- medium (300)--can handle 33% more load than the 200 series

Bearing series

- There is a heavy series, which can handle 20-30% more than the 300 series but it is available in only a few types and sizes.
- There are other series, but they are not commonly used.

Bearing series

Note: all have the same ID

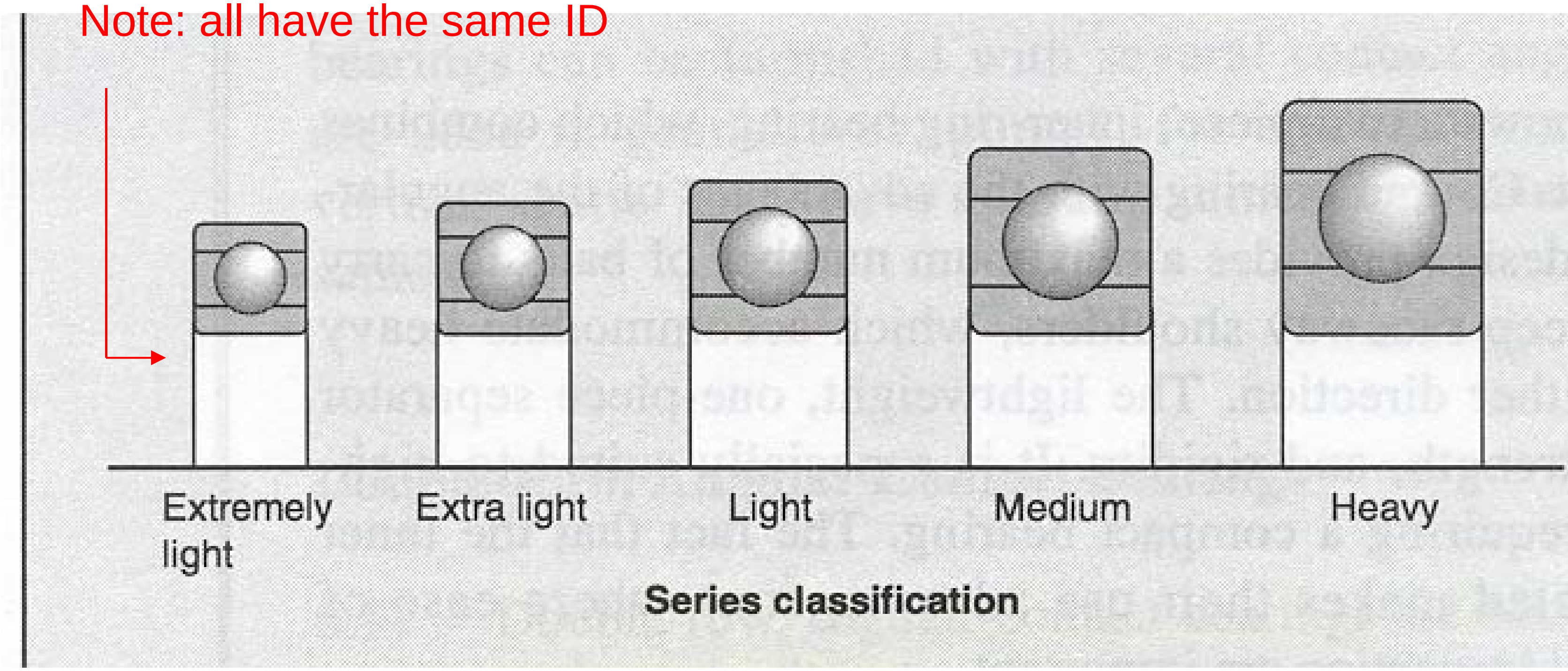
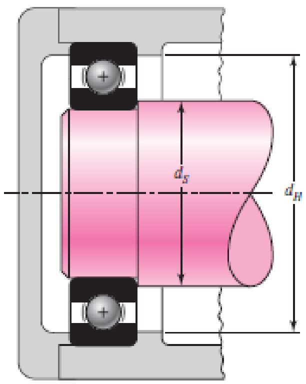


Table 11–2

Dimensions and Load Ratings for Single-Row 02-Series Deep-Groove and Angular-Contact Ball Bearings



Bore, mm	OD, mm	Width, mm	Fillet Radius, mm	Shoulder Diameter, mm d_S	Shoulder Diameter, mm d_H	Load Ratings, kN Deep Groove C_{10}	Load Ratings, kN Deep Groove C_0	Load Ratings, kN Angular Contact C_{10}	Load Ratings, kN Angular Contact C_0
10	30	9	0.6	12.5	27	5.07	2.24	4.94	2.12
12	32	10	0.6	14.5	28	6.89	3.10	7.02	3.05
15	35	11	0.6	17.5	31	7.80	3.55	8.06	3.65
17	40	12	0.6	19.5	34	9.56	4.50	9.95	4.75
20	47	14	1.0	25	41	12.7	6.20	13.3	6.55
25	52	15	1.0	30	47	14.0	6.95	14.8	7.65
30	62	16	1.0	35	55	19.5	10.0	20.3	11.0
35	72	17	1.0	41	65	25.5	13.7	27.0	15.0
40	80	18	1.0	46	72	30.7	16.6	31.9	18.6
45	85	19	1.0	52	77	33.2	18.6	35.8	21.2
50	90	20	1.0	56	82	35.1	19.6	37.7	22.8
55	100	21	1.5	63	90	43.6	25.0	46.2	28.5
60	110	22	1.5	70	99	47.5	28.0	55.9	35.5
65	120	23	1.5	74	109	55.9	34.0	63.7	41.5
70	125	24	1.5	79	114	61.8	37.5	68.9	45.5
75	130	25	1.5	86	119	66.3	40.5	71.5	49.0
80	140	26	2.0	93	127	70.2	45.0	80.6	55.0
85	150	28	2.0	99	136	83.2	53.0	90.4	63.0
90	160	30	2.0	104	146	95.6	62.0	106	73.5
95	170	32	2.0	110	156	108	69.5	121	85.0

Table 11–3

Dimensions and Basic Load Ratings for Cylindrical Roller Bearings

	02-Series				03-Series			
Bore, mm	OD, mm	Width, mm	Load Rating, kN C_{10}	C_0	OD, mm	Width, mm	Load Rating, kN C_{10}	C_0
25	52	15	16.8	8.8	62	17	28.6	15.0
30	62	16	22.4	12.0	72	19	36.9	20.0
35	72	17	31.9	17.6	80	21	44.6	27.1
40	80	18	41.8	24.0	90	23	56.1	32.5
45	85	19	44.0	25.5	100	25	72.1	45.4
50	90	20	45.7	27.5	110	27	88.0	52.0
55	100	21	56.1	34.0	120	29	102	67.2
60	110	22	64.4	43.1	130	31	123	76.5
65	120	23	76.5	51.2	140	33	138	85.0
70	125	24	79.2	51.2	150	35	151	102
75	130	25	93.1	63.2	160	37	183	125
80	140	26	106	69.4	170	39	190	125
85	150	28	119	78.3	180	41	212	149
90	160	30	142	100	190	43	242	160
95	170	32	165	112	200	45	264	189
100	180	34	183	125	215	47	303	220
110	200	38	229	167	240	50	391	304
120	215	40	260	183	260	55	457	340
130	230	40	270	193	280	58	539	408
140	250	42	319	240	300	62	682	454
150	270	45	446	260	320	65	781	502

Table 11–4

Bearing-Life Recommendations for Various Classes of Machinery

Type of Application	Life, kh
Instruments and apparatus for infrequent use	Up to 0.5
Aircraft engines	0.5–2
Machines for short or intermittent operation where service interruption is of minor importance	4–8
Machines for intermittent service where reliable operation is of great importance	8–14
Machines for 8-h service that are not always fully utilized	14–20
Machines for 8-h service that are fully utilized	20–30
Machines for continuous 24-h service	50–60
Machines for continuous 24-h service where reliability is of extreme importance	100–200

Table 11–5

Load-Application Factors

Type of Application	Load Factor
Precision gearing	1.0–1.1
Commercial gearing	1.1–1.3
Applications with poor bearing seals	1.2
Machinery with no impact	1.0–1.2
Machinery with light impact	1.2–1.5
Machinery with moderate impact	1.5–3.0

Use as a factors of safety

Example

- An SKF 6210 angular-contact ball bearing has an axial load F_a of 400 lbf and a radial load F_r of 500 lbf applied with the outer ring stationary. The basic load rating C_0 is 4450 lbf and the basic load rating C_{10} is 7900 lbf. Estimate the L_{10} life at a speed of 720 rev/min.

Solution

- Outer ring stationary $\Rightarrow V=1$, $F_a/C_0=400/4450=0.09$
- From table 11-1, using interpolation $\Rightarrow$ e = 0.285
- $F_a / VF_r = 400/ [(1)500]=0.8 > e \Rightarrow$ From equation (11-9) $i = 2$
- $F_e = X_2VF_r + Y_2F_a$
- $X_2=0.56$, and $Y_2=1.527$ by interpolation
- $F_e = (0.56)(1)(500)+1.527(400)=890.8$ lbf
- With $L_D=L_{10}$ and $F_D=F_e$
- Using equation 11-3 to find L_{10}

$$C_{10} = F_D\left(\frac{L_D n_D 60}{L_R n_R 60}\right)^{1/a}$$

$$L_D = L_{10} = \frac{60 L_R n_R}{60 n_D}\left(\frac{C_{10}}{F_e}\right)^a = \frac{10^6}{60(720)}\left(\frac{7900}{890.8}\right)^3 = 16150h$$

EXAMPLE 11–4 An SKF 6210 angular-contact ball bearing has an axial load F_a of 1780 N and a radial load F_r of 2225 N applied with the outer ring stationary. The basic static load rating C_0 is 19 800 N and the basic load rating C_{10} is 35 150 N. Estimate the L_{10} life at a speed of 720 rev/min.

Solution $V = 1$ and $F_a/C_0 = 1780/19\ 800 = 0.090$. Interpolate for e in Table 11–1:

F_a/C_0	e	
0.084	0.28	
0.090	e	from which $e = 0.285$
0.110	0.30	

$F_a/(VF_r) = 1780/[(1)2225] = 0.8 > 0.285$. Thus, interpolate for Y_2:

F_a/C_0	Y_2	
0.084	1.55	
0.090	Y_2	from which $Y_2 = 1.527$
0.110	1.45	

From Eq. (11–9),

$$F_e = X_2VF_r + Y_2F_a = 0.56(1)2225 + 1.527(1780) = 3964 \text{ N}$$

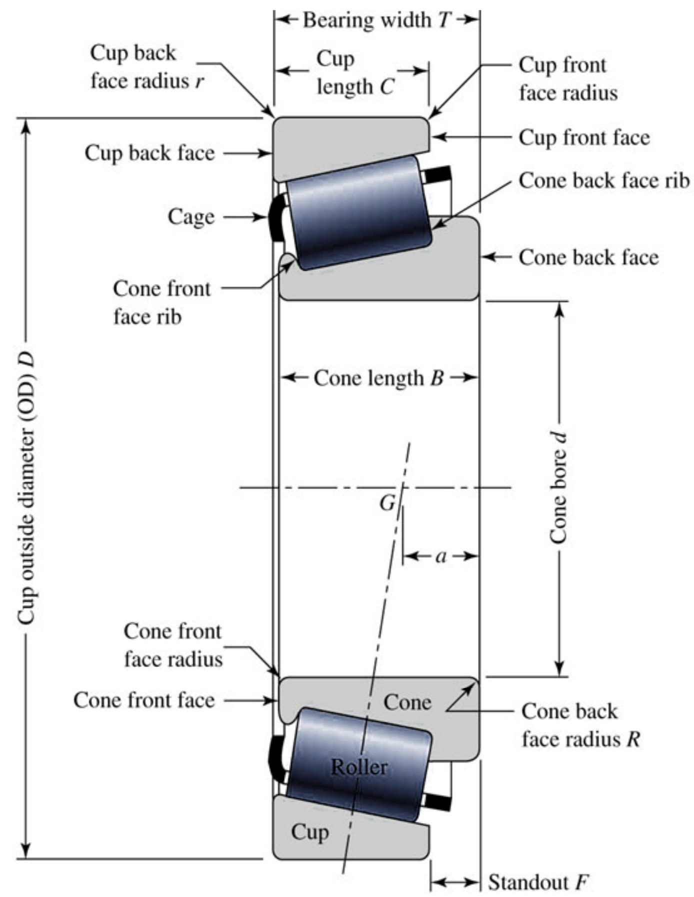
With $L_D = L_{10}$ and $F_D = F_e$, solving Eq. (11–3) for L_{10} gives

Answer

$$L_{10} = \frac{60L_Rn_R}{60n_D}\left(\frac{C_{10}}{F_e}\right)^a = \frac{10^6}{60(720)}\left(\frac{35\ 150}{3964}\right)^3 = 161\ 395 \text{ h}$$

11.9 Selection of Tapered Roller Bearings

- Tapered roller bearings have a number of features that make them complicated
- The tapered roller bearing assembly consist of four components as shown in figure 11.3:
  - Cone (inner ring)
  - Cup (outer ring)
  - Tapered rollers
  - Cage (spacer retainer)



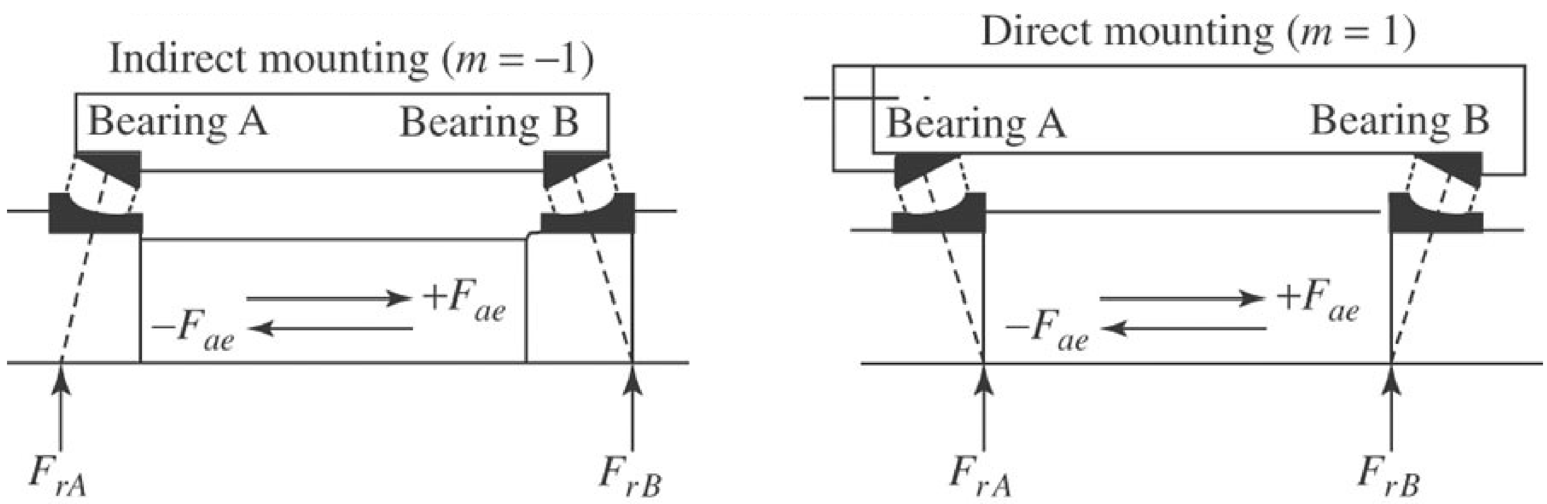
- A tapered roller bearing carries both radial and thrust (axial) loads, or any combination of the two.

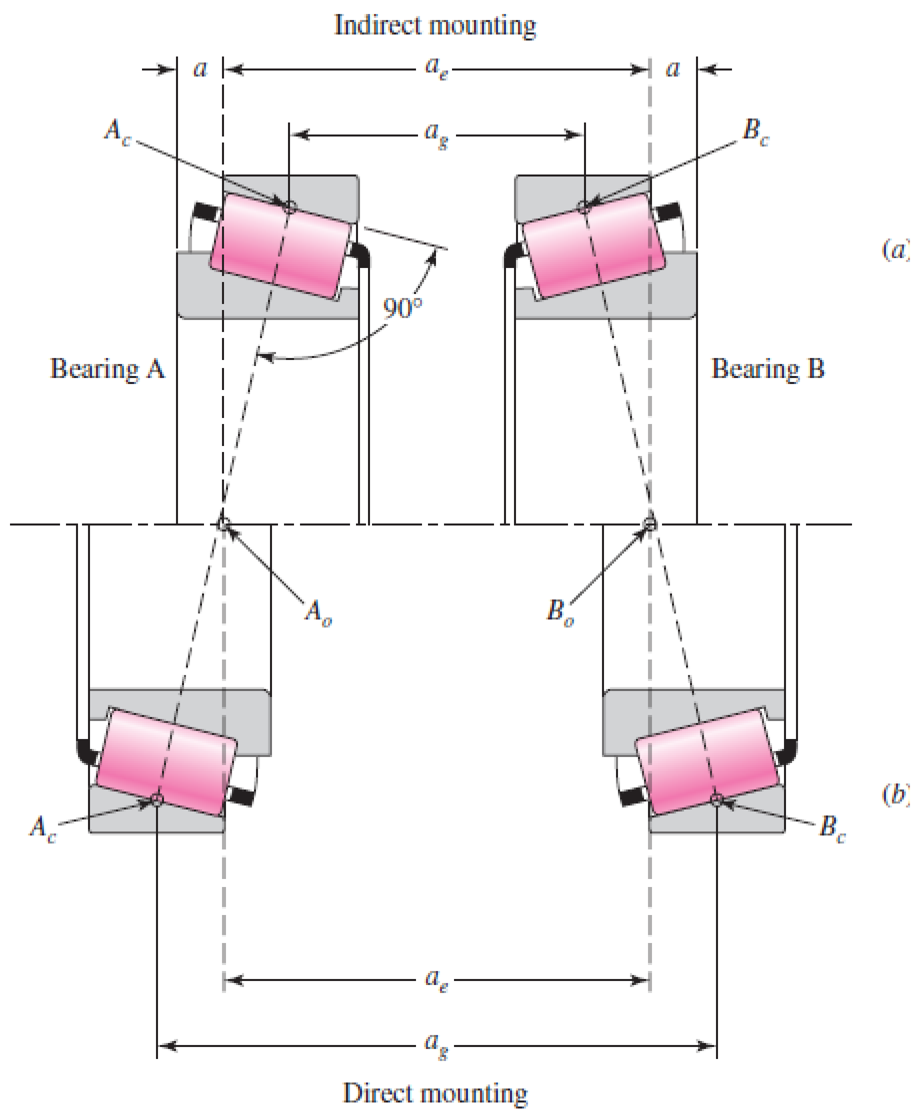
- If there is no external thrust load, there is still a thrust reaction within the bearing due to the taper shape of the bearing.

- To avoid the separation of the races and the rollers, the thrust load must be resisted by an equal and opposite force. This is can be done by use at least two tapered roller bearings on a shaft

•

- In this case, the two bearings can be mounted either:
  - The cone fronts facing each other (It is called indirect mounting)
  - The cone backs facing each other (It is called direct mounting)





Indirect mounting
a
a_e
a
A_c
a_g
B_c
(a)
90°
Bearing A
Bearing B
A_o
B_o
(b)
A_c
B_c
a_e
a_g
Direct mounting

- The thrust component F_a produced by a pure radial load F_r is specified by the Timken Company as:

$$F_a = \frac{0.47F_r}{K}$$

Where K is the ratio of the radial load rating of the bearing to the thrust load rating

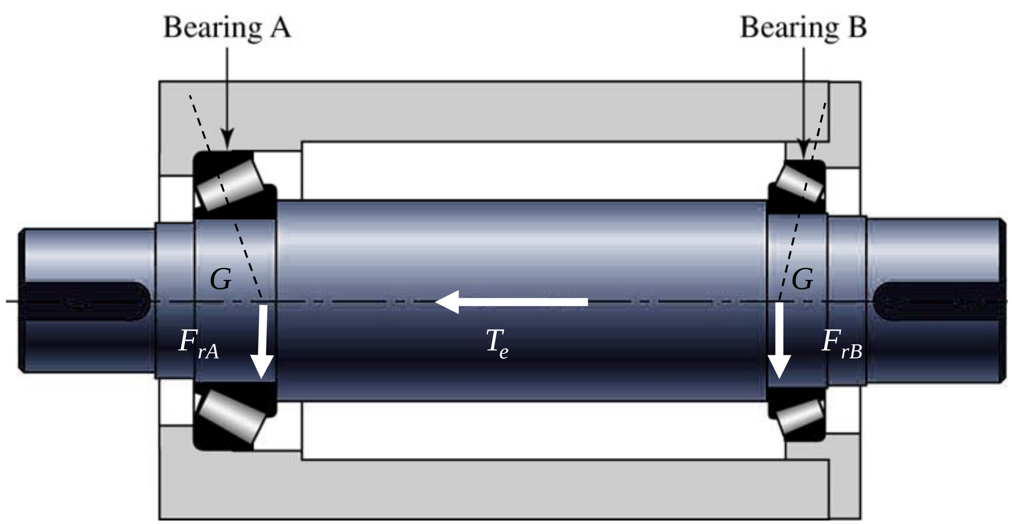
In design of bearing, the initial value of K is chosen to be

$K = 1.50$ for radial bearing

$K = 0.75$ for steep-angle bearings

After the analysis is done, the correction value for K is reselected from the catalogue and the analysis is repeated.

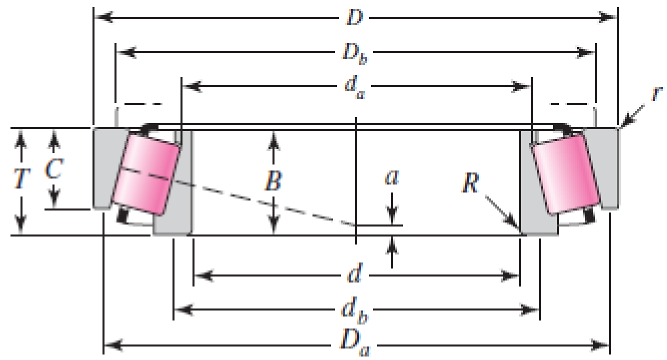
- The figure shows a typical bearing mounting subjected to an **external thrust load** T_e.
- The radial reactions F_{rA} and F_{rB} are computed by taking moments about the effective load centers G.
- The distance for the value of a which is shown in figure 11.3 is obtained from the catalog rating sheets.



SINGLE-ROW STRAIGHT BORE

bore	outside diameter	width	rating at 500 rpm for 3000 hours L_{10}		factor	eff. load center	part numbers		cone				cup			
			one-row radial	thrust			cone	cup	max shaft fillet radius	width	backing shoulder diameters		max housing fillet radius	width	backing shoulder diameters	
d	D	T	N lbf	N lbf	K	a②			R①	B	d_b	d_a	r①	C	D_b	D_a
25.000 0.9843	**52.000** 2.0472	**16.250** 0.6398	**8190** 1840	**5260** 1180	**1.56**	**−3.6** −0.14	**◆30205**	**◆30205**	**1.0** 0.04	**15.000** 0.5906	**30.5** 1.20	**29.0** 1.14	**1.0** 0.04	**13.000** 0.5118	**46.0** 1.81	**48.5** 1.91
25.000 0.9843	**52.000** 2.0472	**19.250** 0.7579	**9520** 2140	**9510** 2140	**1.00**	**−3.0** −0.12	**◆32205-B**	**◆32205-B**	**1.0** 0.04	**18.000** 0.7087	**34.0** 1.34	**31.0** 1.22	**1.0** 0.04	**15.000** 0.5906	**43.5** 1.71	**49.5** 1.95
25.000 0.9843	**52.000** 2.0472	**22.000** 0.8661	**13200** 2980	**7960** 1790	**1.66**	**−7.6** −0.30	**◆33205**	**◆33205**	**1.0** 0.04	**22.000** 0.8661	**34.0** 1.34	**30.5** 1.20	**1.0** 0.04	**18.000** 0.7087	**44.5** 1.75	**49.0** 1.93
25.000 0.9843	**62.000** 2.4409	**18.250** 0.7185	**13000** 2930	**6680** 1500	**1.95**	**−5.1** −0.20	**◆30305**	**◆30305**	**1.5** 0.06	**17.000** 0.6693	**32.5** 1.28	**30.0** 1.18	**1.5** 0.06	**15.000** 0.5906	**55.0** 2.17	**57.0** 2.24
25.000 0.9843	**62.000** 2.4409	**25.250** 0.9941	**17400** 3910	**8930** 2010	**1.95**	**−9.7** −0.38	**◆32305**	**◆32305**	**1.5** 0.06	**24.000** 0.9449	**35.0** 1.38	**31.5** 1.24	**1.5** 0.06	**20.000** 0.7874	**54.0** 2.13	**57.0** 2.24

SINGLE-ROW STRAIGHT BORE



bore	outside diameter	width	rating at 500 rpm for 3000 hours L_{10}		factor	eff. load center	part numbers		cone				cup			
			one-row radial	thrust			cone	cup	max shaft fillet radius	width	backing shoulder diameters		max housing fillet radius	width	backing shoulder diameters	
d	D	T	N lbf	N lbf	K	a②			R①	B	d_b	d_a	r①	C	D_b	D_a
25.400 1.0000	**65.088** 2.5625	**22.225** 0.8750	**13100** 2950	**16400** 3690	**0.80**	**−2.3** −0.09	**23100**	**23256**	**1.5** 0.06	**21.463** 0.8450	**39.0** 1.54	**34.5** 1.36	**1.5** 0.06	**15.875** 0.6250	**53.0** 2.09	**63.0** 2.48
25.400 1.0000	**66.421** 2.6150	**23.812** 0.9375	**18400** 4140	**8000** 1800	**2.30**	**−9.4** −0.37	**2687**	**2631**	**1.3** 0.05	**25.433** 1.0013	**33.5** 1.32	**31.5** 1.24	**1.3** 0.05	**19.050** 0.7500	**58.0** 2.28	**60.0** 2.36
25.400 1.0000	**68.262** 2.6875	**22.225** 0.8750	**15300** 3440	**10900** 2450	**1.40**	**−5.1** −0.20	**02473**	**02420**	**0.8** 0.03	**22.225** 0.8750	**34.5** 1.36	**33.5** 1.32	**1.5** 0.06	**17.462** 0.6875	**59.0** 2.32	**63.0** 2.48
25.400 1.0000	**72.233** 2.8438	**25.400** 1.0000	**18400** 4140	**17200** 3870	**1.07**	**−4.6** −0.18	**HM88630**	**HM88610**	**0.8** 0.03	**25.400** 1.0000	**39.5** 1.56	**39.5** 1.56	**2.3** 0.09	**19.842** 0.7812	**60.0** 2.36	**69.0** 2.72

- The equivalent radial load on the bearings
  - Bearing A: $$F_{eA} = 0.4F_{rA} + K_A\left(\frac{0.47F_{rB}}{K_B} + T_e\right)$$
  - Bearing B: $$F_{eB} = 0.4F_{rB} + K_B\left(\frac{0.47F_{rA}}{K_A} - T_e\right)$$
- After we calculate the value of F_{eA} and F_{eB}, we check the following:
  - IF $F_{rA} > F_{eA} \Rightarrow$ use F_{rA} for design of baring A.
  - IF $F_{rA} < F_{eA} \Rightarrow$ use F_{eA} for design of baring A.
  - IF $F_{rB} > F_{eB} \Rightarrow$ use F_{rB} for design of baring B.
  - IF $F_{rB} < F_{eB} \Rightarrow$ use F_{eB} for design of baring B.

Steps in Tapered Roller Bearing Selection

- Given required bearing life L_{10} and load arranegment
- Determine applied radial F_{ex} and thrust T_{ex} loads
- Distribute radial loads to bearings: F_{rA} and F_{rB} using estimated "a" value to locate load center
- Estimate Bearing K factors for induced thrust
- Calculate equivalent radial loads F_{eA} and F_{eB} using estimated K factor
- Calculate rated load using actual load and desired life
- Select bearing from those available (in table)
- Check bearing loads using actual K factor and "a" values

Example

- The gear-reduction unit is arranged to rotate the cup while the cone is stationary. Bearing A takes the thrust load of 250 lb and, in addition, has a radial load 875 lb. Bearing B is subjected to a pure radial load of 625 lb. The speed is 150 rev/min. The desired L_{10} life is 90 kh. The desired shaft diameters are 1.125 in at A and 1 in at B. Select suitable tapered roller bearings, using an application factor of unity.

Solution

- Given:

  At Bearing A: T_e=250 lb, F_{rA} =875 lb, d_A=1.125 in.

  At Bearing B: F_{rB} =625 lb, d_B=1 in.

  L_{10} = 90 kh, n_D=150 rev/min. R=.9

  Assume an initial value for K_A and K_B as 1.5

$$F_{eA} = 0.4F_{rA} + K_A\left(\frac{0.47F_{rB}}{K_B} + T_e\right)$$

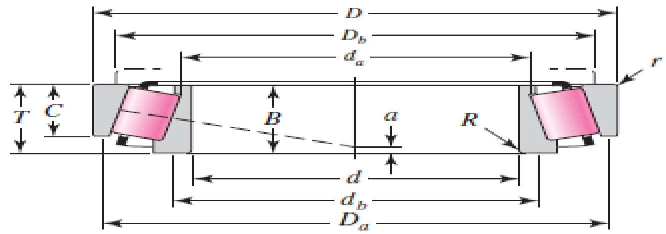
$$= 0.4(875) + 1.5\left(\frac{0.47(625)}{1.5} + 250\right) = 1020lb$$

- Since $F_{eA} > F_{rA} \Rightarrow$ use F_{eA} for design of bearing A.
- Form equation 11-3

$$C_{10} = F_D \left(\frac{L_D n_D 60}{L_R n_R 60} \right)^{1/a} = 1020 \left(\frac{(9000)(150)(60)}{(3000)(500)(60)} \right)^{3/10} = 1970 lb$$

- Now, go figure 11-15 and with (d_A=1.125 in) select form columns 4 and 5 the value of C_{10} that is equal or near to the value that we calculate in equation above
- Thus, we select 15590 cone, a 15520 cup, and K_A=1.69
- Calculate F_{eA} and then C_{10} again with the corrected value of K_A, we get: C_{10}=2130 lb.

SINGLE-ROW STRAIGHT BORE



bore d	outside diameter D	width T	rating at 500 rpm for 3000 hours L_{10}: one-row radial N lbf	rating at 500 rpm for 3000 hours L_{10}: thrust N lbf	factor K	eff. load center a②	part numbers: cone	part numbers: cup	cone: max shaft fillet radius R①	cone: width B	cone: backing shoulder diameters d_b	cone: backing shoulder diameters d_a	cup: max housing fillet radius r①	cup: width C	cup: backing shoulder diameters D_b	cup: backing shoulder diameters D_a
25.400 1.0000	**65.088** 2.5625	**22.225** 0.8750	**13100** 2950	**16400** 3690	**0.80**	**−2.3** −0.09	**23100**	**23256**	**1.5** 0.06	**21.463** 0.8450	**39.0** 1.54	**34.5** 1.36	**1.5** 0.06	**15.875** 0.6250	**53.0** 2.09	**63.0** 2.48
25.400 1.0000	**66.421** 2.6150	**23.812** 0.9375	**18400** 4140	**8000** 1800	**2.30**	**−9.4** −0.37	**2687**	**2631**	**1.3** 0.05	**25.433** 1.0013	**33.5** 1.32	**31.5** 1.24	**1.3** 0.05	**19.050** 0.7500	**58.0** 2.28	**60.0** 2.36
25.400 1.0000	**68.262** 2.6875	**22.225** 0.8750	**15300** 3440	**10900** 2450	**1.40**	**−5.1** −0.20	**02473**	**02420**	**0.8** 0.03	**22.225** 0.8750	**34.5** 1.36	**33.5** 1.32	**1.5** 0.06	**17.462** 0.6875	**59.0** 2.32	**63.0** 2.48
25.400 1.0000	**72.233** 2.8438	**25.400** 1.0000	**18400** 4140	**17200** 3870	**1.07**	**−4.6** −0.18	**HM88630**	**HM88610**	**0.8** 0.03	**25.400** 1.0000	**39.5** 1.56	**39.5** 1.56	**2.3** 0.09	**19.842** 0.7812	**60.0** 2.36	**69.0** 2.72
25.400 1.0000	**72.626** 2.8593	**30.162** 1.1875	**22700** 5110	**13000** 2910	**1.76**	**−10.2** −0.40	**3189**	**3120**	**0.8** 0.03	**29.997** 1.1810	**35.5** 1.40	**35.0** 1.38	**3.3** 0.13	**23.812** 0.9375	**61.0** 2.40	**67.0** 2.64
26.157 1.0298	**62.000** 2.4409	**19.050** 0.7500	**12100** 2730	**7280** 1640	**1.67**	**−5.8** −0.23	**15103**	**15245**	**0.8** 0.03	**20.638** 0.8125	**33.0** 1.30	**32.5** 1.28	**1.3** 0.05	**14.288** 0.5625	**55.0** 2.17	**58.0** 2.28
26.162 1.0300	**63.100** 2.4843	**23.812** 0.9375	**18400** 4140	**8000** 1800	**2.30**	**−9.4** −0.37	**2682**	**2630**	**1.5** 0.06	**25.433** 1.0013	**34.5** 1.36	**32.0** 1.26	**0.8** 0.03	**19.050** 0.7500	**57.0** 2.24	**59.0** 2.32
26.162 1.0300	**66.421** 2.6150	**23.812** 0.9375	**18400** 4140	**8000** 1800	**2.30**	**−9.4** −0.37	**2682**	**2631**	**1.5** 0.06	**25.433** 1.0013	**34.5** 1.36	**32.0** 1.26	**1.3** 0.05	**19.050** 0.7500	**58.0** 2.28	**60.0** 2.36
26.975 1.0620	**58.738** 2.3125	**19.050** 0.7500	**11600** 2610	**6560** 1470	**1.77**	**−5.8** −0.23	**1987**	**1932**	**0.8** 0.03	**19.355** 0.7620	**32.5** 1.28	**31.5** 1.24	**1.3** 0.05	**15.080** 0.5937	**52.0** 2.05	**54.0** 2.13
† **26.988** † 1.0625	**50.292** 1.9800	**14.224** 0.5600	**7210** 1620	**4620** 1040	**1.56**	**−3.3** −0.13	**L44649**	**L44610**	**3.5** 0.14	**14.732** 0.5800	**37.5** 1.48	**31.0** 1.22	**1.3** 0.05	**10.668** 0.4200	**44.5** 1.75	**47.0** 1.85
† **26.988** † 1.0625	**60.325** 2.3750	**19.842** 0.7812	**11000** 2480	**6550** 1470	**1.69**	**−5.1** −0.20	**15580**	**15523**	**3.5** 0.14	**17.462** 0.6875	**38.5** 1.52	**32.0** 1.26	**1.5** 0.06	**15.875** 0.6250	**51.0** 2.01	**54.0** 2.13
† **26.988** † 1.0625	**62.000** 2.4409	**19.050** 0.7500	**12100** 2730	**7280** 1640	**1.67**	**−5.8** −0.23	**15106**	**15245**	**0.8** 0.03	**20.638** 0.8125	**33.5** 1.32	**33.0** 1.30	**1.3** 0.05	**14.288** 0.5625	**55.0** 2.17	**58.0** 2.28
† **26.988** † 1.0625	**66.421** 2.6150	**23.812** 0.9375	**18400** 4140	**8000** 1800	**2.30**	**−9.4** −0.37	**2688**	**2631**	**1.5** 0.06	**25.433** 1.0013	**35.0** 1.38	**33.0** 1.30	**1.3** 0.05	**19.050** 0.7500	**58.0** 2.28	**60.0** 2.36
28.575 1.1250	**56.896** 2.2400	**19.845** 0.7813	**11600** 2610	**6560** 1470	**1.77**	**−5.8** −0.23	**1985**	**1930**	**0.8** 0.03	**19.355** 0.7620	**34.0** 1.34	**33.5** 1.32	**0.8** 0.03	**15.875** 0.6250	**51.0** 2.01	**54.0** 2.11
28.575 1.1250	**57.150** 2.2500	**17.462** 0.6875	**11000** 2480	**6550** 1470	**1.69**	**−5.1** −0.20	**15590**	**15520**	**3.5** 0.14	**17.462** 0.6875	**39.5** 1.56	**33.5** 1.32	**1.5** 0.06	**13.495** 0.5313	**51.0** 2.01	**53.0** 2.09
28.575 1.1250	**58.738** 2.3125	**19.050** 0.7500	**11600** 2610	**6560** 1470	**1.77**	**−5.8** −0.23	**1985**	**1932**	**0.8** 0.03	**19.355** 0.7620	**34.0** 1.34	**33.5** 1.32	**1.3** 0.05	**15.080** 0.5937	**52.0** 2.05	**54.0** 2.13
28.575 1.1250	**58.738** 2.3125	**19.050** 0.7500	**11600** 2610	**6560** 1470	**1.77**	**−5.8** −0.23	**1988**	**1932**	**3.5** 0.14	**19.355** 0.7620	**39.5** 1.56	**33.5** 1.32	**1.3** 0.05	**15.080** 0.5937	**52.0** 2.05	**54.0** 2.13
28.575 1.1250	**60.325** 2.3750	**19.842** 0.7812	**11000** 2480	**6550** 1470	**1.69**	**−5.1** −0.20	**15590**	**15523**	**3.5** 0.14	**17.462** 0.6875	**39.5** 1.56	**33.5** 1.32	**1.5** 0.06	**15.875** 0.6250	**51.0** 2.01	**54.0** 2.13
28.575 1.1250	**60.325** 2.3750	**19.845** 0.7813	**11600** 2610	**6560** 1470	**1.77**	**−5.8** −0.23	**1985**	**1931**	**0.5** 0.03	**19.355** 0.7620	**34.0** 1.34	**33.5** 1.32	**1.3** 0.05	**15.875** 0.6250	**52.0** 2.05	**55.0** 2.17

- Again go to figure 11-15 and with (d_A=1.125 in) select form column 4 the value of C_{10} that is equal or near to the value that we calculate in equation above we have the same values: 15590 cone, a 15520 cup, and K_A=1.69.

- For bearing B:

$$F_{eB} = 0.4F_{rB} + K_B\left(\frac{0.47F_{rA}}{K_A} - T_e\right)$$

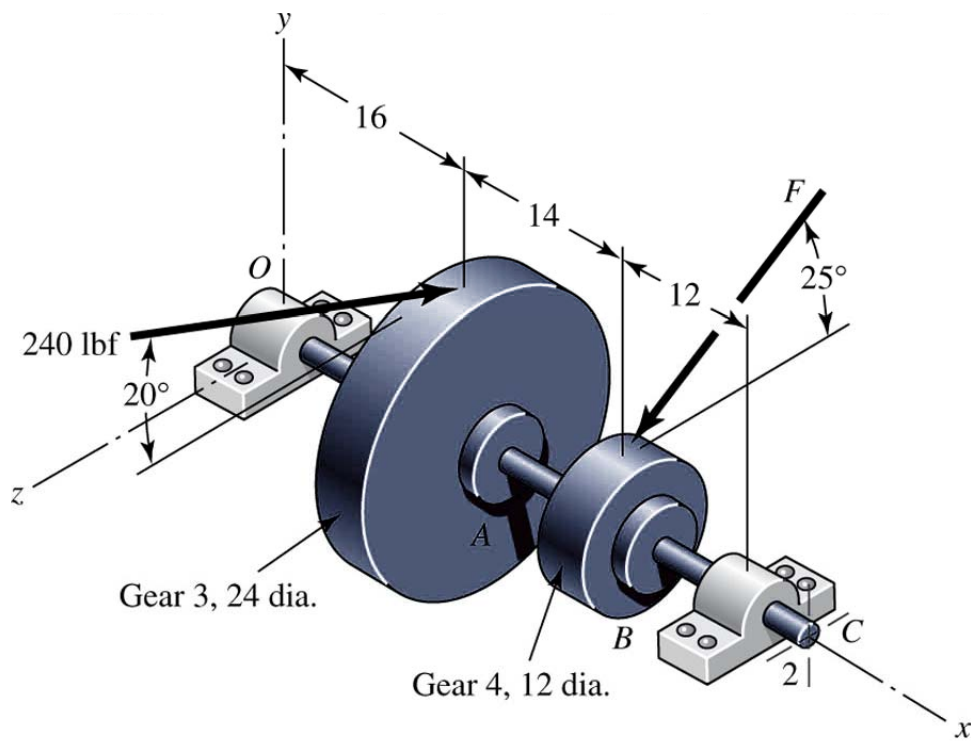
$$= 0.4(625) + 1.5\left(\frac{0.47(875)}{1.69} - 250\right) = 240lb$$

- Since $F_{\mathrm{rB}} > F_{\mathrm{eB}} \Rightarrow$ use F_{rB} for design of bearing B
- Form equation 11-3

$$C_{10} = F_D\left(\frac{L_D n_D 60}{L_R n_R 60}\right)^{1/a} = 625\left(\frac{(9000)(150)(60)}{(3000)(500)(60)}\right)^{3/10} = 1210lb$$

- Now, go to figure 11-15 and with (d_B=1 in) there are five bearings from which to choose. The one at the top of the list has the smallest rating, the smallest OD, and the narrowest width.
- Thus, we select 07100 Cone, a 07196 Cup, C_{10}=1570 lb and K_B=1.45
- Calculate F_{eB} and then C_{10} again with the corrected value of K_B, we get the same C_{10}=1210 lb. So the selection for bearing B is: 07100 Cone, a 07196 Cup, C_{10}=1570 lb and K_B=1.45.

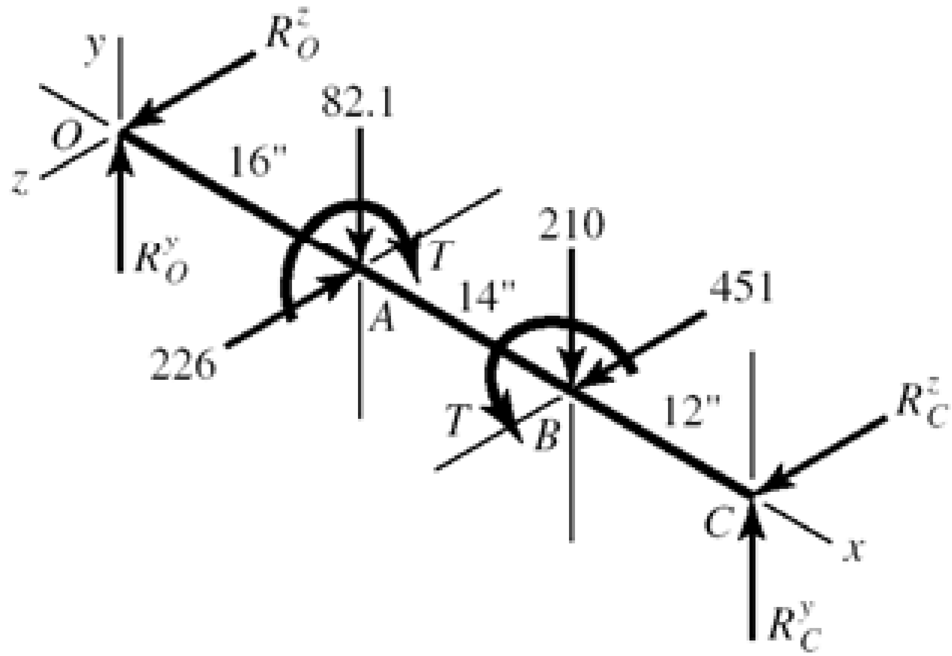
Problem 11-13



- A gear-reduction unit uses the countershaft depicted in the figure. Find the two bearing reactions. The bearings are to be angular-contact ball bearings, having a desired life of 40 kh when used at 200 rev/min. Use 1.2 for the application factor and a reliability goal for the bearing pair of 0.95. Select the bearings from Table 11-2.

Solution

- The bearings are to be angular-contact ball bearings, having a desired life of :
  - $L_D = 40$ kh
  - $n_D = 200$ rev/min.
  - Application factor $f = 1.2$
  - Reliability goal $\boldsymbol{R_2} = 0.95$
  - Select the bearings from Table 11-2.



- To find the reliability for each bearing:

$$\boldsymbol{R_2} = (R)^2 \Rightarrow R=(\boldsymbol{R_2})^{1/2}==(0.95)^{1/2} = 0.975$$

- The torque at gear A:

$$T = 240(12)(\cos 20°) = 2706 \text{ lbf} \cdot \text{in}$$

- The force at gear B can be found by:

$$F = \frac{2706}{6\cos 25^{\text{o}}} = 498 \text{ lbf}$$

- In *xy*-plane:

$$\sum M_O = -82.1(16) - 210(30) + 42R_C^y = 0$$

$$R_C^y = 181 \text{ lbf}$$

$$R_O^y = 82 + 210 - 181 = 111 \text{ lbf}$$

In *xz*-plane:

$$\sum M_O = 226(16) - 452(30) - 42R_c^z = 0$$

$$R_C^z = -237 \text{ lbf}$$

$$R_O^z = 226 - 451 + 237 = 12 \text{ lbf}$$

$$R_O = (111^2 + 12^2)^{1/2} = 112 \text{ lbf} \quad \textit{Ans.}$$

$$R_C = (181^2 + 237^2)^{1/2} = 298 \text{ lbf} \quad \textit{Ans.}$$

$$F_{eO} = 1.2(112) = 134.4 \text{ lbf}$$

$$F_{eC} = 1.2(298) = 357.6 \text{ lbf}$$

$$x_D = \frac{40\,000(200)(60)}{10^6} = 480$$

$$(C_{10})_O = 134.4\left\{\frac{480}{0.02 + 4.439[\ln(1/0.975)]^{1/1.483}}\right\}^{1/3}$$

$$= 1438 \text{ lbf} \quad \text{or} \quad 6.398 \text{ kN}$$

$$(C_{10})_C = 357.6\left\{\frac{480}{0.02 + 4.439[\ln(1/0.975)]^{1/1.483}}\right\}^{1/3}$$

$$= 3825 \text{ lbf} \quad \text{or} \quad 17.02 \text{ kN}$$

- Bearing at *O*:

  Choose a deep-groove 02-12 mm with C_{10} = 6.89 kN and C_0 = 3.10 kN.

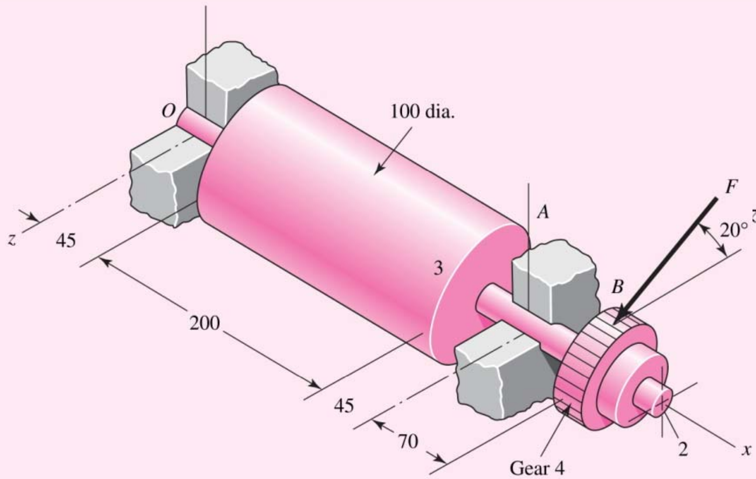
- Bearing at *C*:

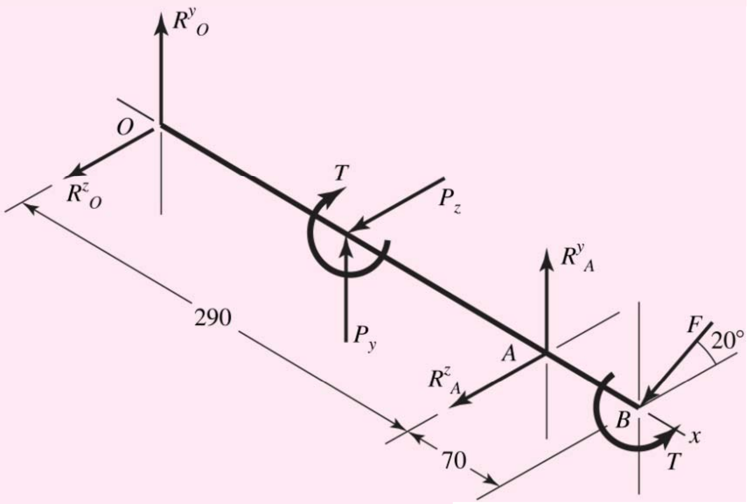
  Choose a deep-groove 02-30 mm with C_{10} = 19.5 kN and C_0 = 10.0 kN.

- Note:

  It may be an advantage to use identical 02-30 mm bearings in a gear-reduction unit.

Shown in Figure 11–12 is a gear-driven squeeze roll that mates with an idler roll. The roll is designed to exert a normal force of 5.25 N/mm of roll length and a pull of 4.2 N/mm on the material being processed. The roll speed is 300 rev/min, and a design life of 30 kh is desired. Use an application factor of 1.2, and select a pair of angular-contact 02-series ball bearings from Table 11–2 to be mounted at 0 and *A*. Use the same size bearings at both locations and a combined reliability of at least 0.92.





Assume concentrated forces as shown.

$$P_z = 200(4.2) = 840 \text{ N}$$

$$P_y = 200(5.25) = 1050 \text{ N}$$

$$T = 840(50) = 42\,000 \text{ N} \cdot \text{mm}$$

$$\sum T^x = -42\,000 + 38F\cos 20^\circ = 0$$

$$F = \frac{42\,000}{38(0.940)} = 1176.2 \text{ N}$$

$$\sum M_O^z = 145P_y + 290R_A^y - 360F\sin 20^\circ = 0;$$

thus

$$145(1050) + 290R_A^y - 360(1176.2)(0.342) = 0$$

$$R_A^y = -25.6 \text{ N}$$

$$\sum M_O^y = -145P_z - 290R_A^z - 360F\cos 20^\circ = 0;$$

thus

$$-145(840) - 290R_A^z - 360(1176.2)(0.940) = 0$$

$$R_A^z = -1792 \text{ N};$$

$$R_A = [(-1792)^2 + (-25.6)^2]^{1/2} = 1792 \text{ N}$$

$$\sum F^z = R_O^z + P_z + R_A^z + F\cos 20^\circ = 0$$

$$R_O^z + 840 - 1792 + 1176.2(0.940) = 0$$

$$R_O^z = -153.3 \text{ N}$$

$$\sum F^y = R_O^y + P_y + R_A^y - F\sin 20^\circ = 0$$

$$R_O^y + 1050 - 25.6 - 1176.2(0.342) = 0$$

$$R_O^y = -622 \text{ N}$$

$$R_O = [(-153.3)^2 + (-622)^2]^{1/2} = 640.6 \text{ N}$$

So the reaction at A governs.

Reliability Goal: $\sqrt{0.92} = 0.96$

$$F_D = 1.2(1792) = 2150.4 \text{ N}$$

$$x_D = 30\,000(300)(60/10^6) = 540$$

$$C_{10} = 2150.4\left\{\frac{540}{0.02 + 4.439[\ln(1/0.96)]^{1/1.483}}\right\}^{1/3}$$

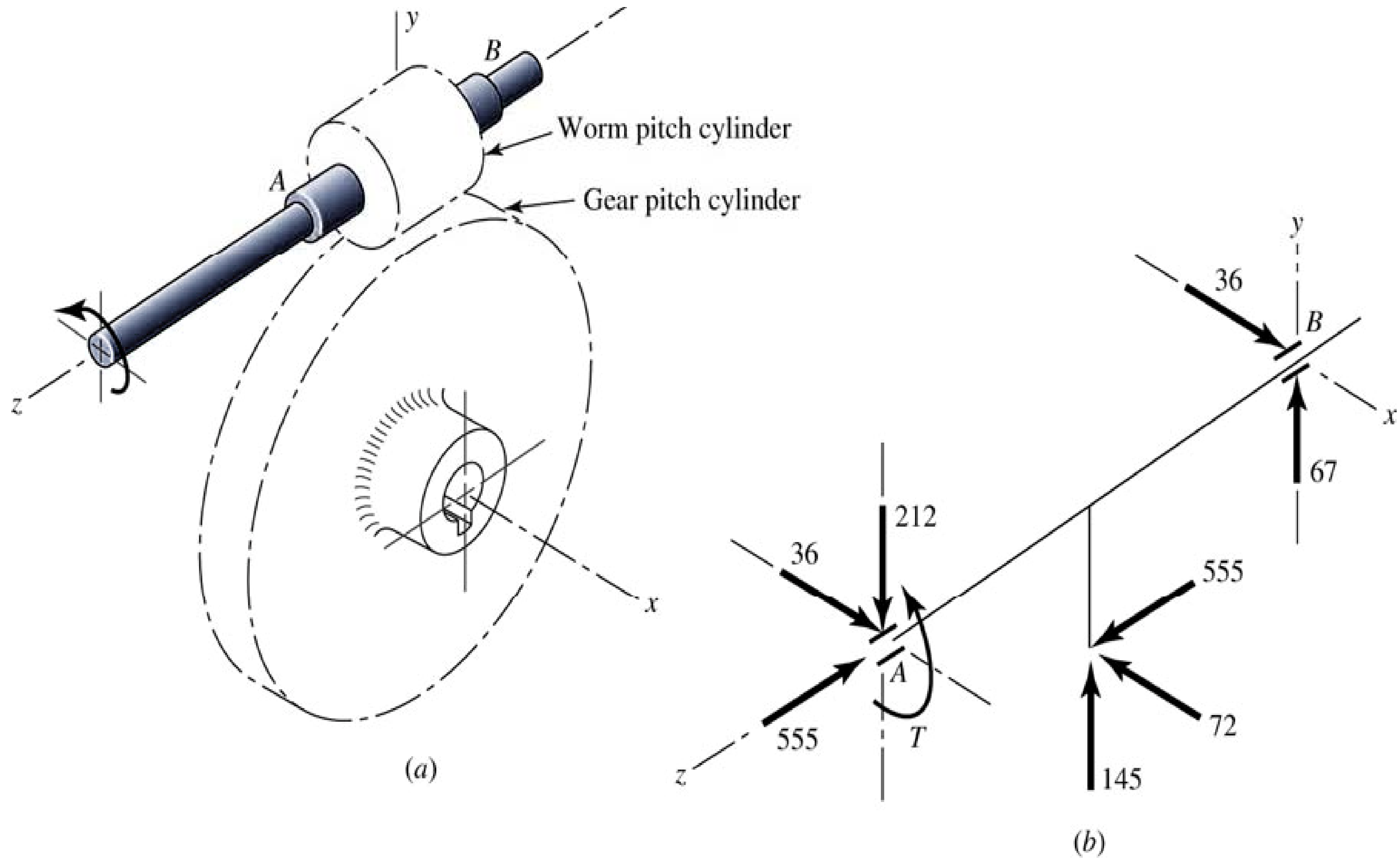
$$= 21.59 \text{ kN}$$

A 02–35 bearing will do.

Answer *Decision:* Specify an angular-contact 02–35 mm ball bearing for the locations at A and O. Check combined reliability.

Problem 11-14

- The worm shaft shown in part *a* of the figure transmits 1.35 hp at 600 rev/min. A static force analysis gave the results shown in part *b* of the figure. Bearing *A* is to be an angular-contact ball bearing mounted to take the 555-lbf thrust load. The bearing at *B* is to take only the radial load, so a straight roller bearing will be employed. Use an application factor of 1.3, a desired life of 25 kh, and a reliability goal, combined, of 0.99. Specify each bearing.



- Figure P11-14: (*a*) Worm and worm gear; (*b*) force analysis of worm shaft, forces in pounds.

Solution

- The worm shaft transmits 1.35 hp at 600 rev/min.
- Bearing *A* is to be an angular-contact ball bearing mounted to take the 555-lbf thrust load. Bearing *B* takes only radial load, so a straight roller bearing will be employed.
- Application factor $f = 1.3$
- n_D =25 kh, R= 0.99.

Bearing at A (Ball)

- $F_r = (362 + 2122)^{1/2} = 215$ lbf = 0.957 kN
- $F_a = 555$ lbf = 2.47 kN

- Trial #1:

  Since we do not have a value for the specification of the bearing, we start by selecting one of the bearing from the table.

  Therefore, select a 02-85 mm angular-contact with $C_{10} = 90.4$ kN and $C_0 = 63.0$ kN.

- $F_a / C_0 = 2.47 / 63.0 = 0.0392$
  Form table 11-1: by interpolation e = 0.236

- $F_a / VF_r = 2.47/ [(1)0.957]=2.581 > e$
  $\Rightarrow$ From equation (11-9) $i = 2$

- $F_e=X_2VF_r+Y_2F_a$
  X_2=0.56, and Y_2=1.88 by interpolation
  $F_e = (0.56)(1)(0.957)+1.88(2.47)=5.18$ kN

- With the application factor F_D=1.3(5.18)=6.73kN

$$C_{10} = F_D \left\{ \frac{(L_D n_D / L_R n_R)}{0.02 + 4.439[\ln(1/R)]^{1/1.483}} \right\}^{1/3}$$

$$L_D n_D / L_R n_R = \frac{L}{L_{10}} = \frac{(25000)(600)(60)rev}{(10^6)rev} = 900$$

$$\therefore C_{10} = 6.73 \left\{ \frac{900}{0.02 + 4.439[\ln(1/0.99)]^{1/1.483}} \right\}^{1/3} = 107.7kN$$

- We found that: $(C_{10})_{\text{calculated}} > (C_{10})_{\text{selected from the table}}$

107.7 kN > 90.4kN,

Therefore, we try another bearing with the specification:

Select a 02-95 mm angular-contact with C_{10} = 121 kN and C_0 = 85.0 kN.

- Trial #2:

  Tentatively select a 02-95 mm angular-contact ball with C_{10} = 121 kN and C_0 = 85 kN.
- F_a / C_0 = 2.47 / 85.0 = 0.02906

  Form table 11-1: by interpolation e = 0.222
- F_a / VF_r = 2.47/ [(1)0.957]=2.581 > e

  $\Rightarrow$ From equation (11-9) $i = 2$
- $F_e = X_2 V F_r + Y_2 F_a$

  X_2=0.56, and Y_2=1.98 by interpolation

  F_e = (0.56)(1)(0.957) + 1.98(2.47) = 5.427 kN
- With the application factor F_D=1.3(5.427) = 7.05kN

$$C_{10} = F_D \left\{ \frac{\left(L_D n_D / L_R n_R\right)}{0.02 + 4.439\left[\ln\left(1/R\right)\right]^{1/1.483}} \right\}^{1/3}$$

$$L_D n_D / L_R n_R = \frac{L}{L_{10}} = \frac{(25000)(600)(60)rev}{\left(10^6\right)rev} = 900$$

$$\therefore C_{10} = 7.05 \left\{ \frac{900}{0.02 + 4.439\left[\ln\left(1/0.99\right)\right]^{1/1.483}} \right\}^{1/3} = 112.82kN$$

- We found that: $(C_{10})_{\text{calculated}} < (C_{10})_{\text{selected from the table}}$

  112.82 kN < 121kN

Therefore the specification of the bearing A is:

A 02 series angular-contact with Bore 95 mm, C_{10} = 121 kN, and C_0 = 85.0 kN.

Bearing at B (Roller):

- F_r = (362 + 672)$^{1/2}$ = 76.06 lbf = 0.3385 kN
- F_D =1.3(F_r) = 0.44 kN

$$C_{10} = F_D \left\{ \frac{\left(L_D n_D / L_R n_R\right)}{0.02 + 4.439\left[\ln\left(1/R\right)\right]^{1/1.483}} \right\}^{1/3}$$

$$L_D n_D / L_R n_R = \frac{L}{L_{10}} = \frac{(25000)(600)(60)rev}{\left(10^6\right)rev} = 900$$

$$\therefore C_{10} = 0.44 \left\{ \frac{900}{0.02 + 4.439\left[\ln\left(1/0.99\right)\right]^{1/1.483}} \right\}^{3/10} = 5.34 kN$$

- From table 11-3, using a 02 series cylindrical roller bearing with Bore 25 mm, C_{10} = 16.8 kN, and C_0 = 8.8 kN.

EXAMPLE 11–8 The shaft depicted in Fig. 11–18*a* carries a helical gear with a tangential force of 3980 N, a radial force of 1770 N, and a thrust force of 1690 N at the pitch cylinder with directions shown. The pitch diameter of the gear is 200 mm. The shaft runs at a speed of 800 rev/min, and the span (effective spread) between the direct-mount bearings is 150 mm. The design life is to be 5000 h and an application factor of 1 is appropriate. If the reliability of the bearing set is to be 0.99, select suitable single-row tapered-roller Timken bearings.

Solution The reactions in the xz plane from Fig. 11–18*b* are

$$R_{zA} = \frac{3980(50)}{150} = 1327\,\text{N}$$

$$R_{zB} = \frac{3980(100)}{150} = 2653\ \text{N}$$

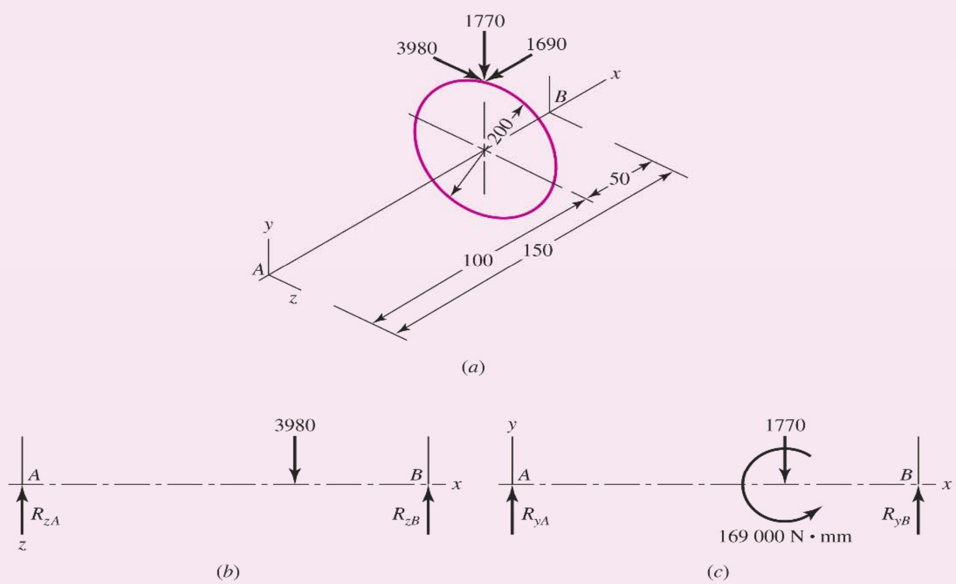
The reactions in the xy plane from Fig. 11–18*c* are

$$R_{yA} = \frac{1770(50)}{150} + \frac{169\ 000}{150} = 1716.7 = 1717\ \text{N}$$

$$R_{yB} = \frac{1770(100)}{150} - \frac{169\ 000}{150} = 53.3\ \text{N}$$

Figure 11–18

Essential geometry of helical gear and shaft. Length dimensions in mm, loads in N, couple in N · mm. (*a*) Sketch (not to scale) showing thrust, radial, and tangential forces. (*b*) Forces in *xz* plane. (*c*) Forces in *xy* plane.



The radial loads F_{rA} and F_{rB} are the vector additions of R_{yA} and R_{zA}, and R_{yB} and R_{zB}, respectively:

$$F_{rA} = \left(R_{zA}^2 + R_{yA}^2\right)^{1/2} = (1327^2 + 1717^2)^{1/2} = 2170\,\text{N}$$
$$F_{rB} = \left(R_{zB}^2 + R_{yB}^2\right)^{1/2} = (2653^2 + 53.3^2)^{1/2} = 2654\,\text{N}$$

Trial 1: With direct mounting of the bearings and application of the external thrust to the shaft, the squeezed bearing is bearing A as labeled in Fig. 11–18a. Using K of 1.5 as the initial guess for each bearing, the induced loads from the bearings are

$$F_{iA} = \frac{0.47F_{rA}}{K_A} = \frac{0.47(2170)}{1.5} = 680\text{ N}$$

$$F_{iB} = \frac{0.47F_{rB}}{K_B} = \frac{0.47(2654)}{1.5} = 832\text{ N}$$

Since F_{iA} is clearly less than $F_{iB} + F_{ae}$, bearing A carries the net thrust load, and Eq. (11–16) is applicable. Therefore, the dynamic equivalent loads are

$$F_{eA} = 0.4F_{rA} + K_A(F_{iB} + F_{ae}) = 0.4(2170) + 1.5(832 + 1690) = 4651\text{ N}$$
$$F_{eB} = F_{rB} = 2654\text{ N}$$

The multiple of rating life is

$$x_D = \frac{L_D}{L_R} = \frac{\mathcal{L}_D n_D 60}{L_R} = \frac{(5000)(800)(60)}{90(10^6)} = 2.67$$

Estimate R_D as $\sqrt{0.99} = 0.995$ for each bearing. For bearing A, from Eq. (11–7) the catalog entry C_{10} should equal or exceed

$$C_{10} = (1)(4651)\left[\frac{2.67}{(4.48)(1-0.995)^{2/3}}\right]^{3/10} = 11\,486\text{ N}$$

From Fig. 11–15, tentatively select type TS 15100 cone and 15245 cup, which will work: $K_A = 1.67$, $C_{10} = 12\,100$ N.

For bearing B, from Eq. (11–7), the catalog entry C_{10} should equal or exceed

$$C_{10} = (1)2654\left[\frac{2.67}{(4.48)(1-0.995)^{2/3}}\right]^{3/10} = 6554 \text{ N}$$

Tentatively select the bearing identical to bearing A, which will work: $K_B = 1.67$, $C_{10} = 12\,100$ N.

Trial 2: Repeat the process with $K_A = K_B = 1.67$ from tentative bearing selection.

$$F_{iA} = \frac{0.47F_{rA}}{K_A} = \frac{0.47(2170)}{1.67} = 611 \text{ N}$$

$$F_{iB} = \frac{0.47F_{rB}}{K_B} = \frac{0.47(2654)}{1.67} = 747 \text{ N}$$

Since F_{iA} is still less than $F_{iB} + F_{ae}$, Eq. (11–16) is still applicable.

$$F_{eA} = 0.4F_{rA} + K_A(F_{iB} + F_{ae}) = 0.4(2170) + 1.67(747 + 1690) = 4938 \text{ N}$$

$$F_{eB} = F_{rB} = 2654 \text{ N}$$

For bearing A, from Eq. (11–7) the corrected catalog entry C_{10} should equal or exceed

$$C_{10} = (1)(4938)\left[\frac{2.67}{(4.48)(1-0.995)^{2/3}}\right]^{3/10} = 12\,195 \text{ N}$$

Although this catalog entry exceeds slightly the tentative selection for bearing A, we will keep it since the reliability of bearing B exceeds 0.995. In the next section we will quantitatively show that the combined reliability of bearing A and B will exceed the reliability goal of 0.99.

For bearing B, $F_{eB} = F_{rB} = 2654$ N. From Eq. (11–7),

$$C_{10} = (1)2654\left[\frac{2.67}{(4.48)(1-0.995)^{2/3}}\right]^{3/10} = 6554 \text{ N}$$

Select cone and cup 15100 and 15245, respectively, for both bearing A and B. Note from Fig. 11–14 the effective load center is located at $a = -5.8$ mm, that is, 5.8 mm into the cup from the back. Thus the shoulder-to-shoulder dimension should be $150 - 2(5.8) = 138.4$ mm. Note that in each iteration of Eq. (11–7) to find the catalog load rating, the bracketed portion of the equation is identical and need not be re-entered on a calculator each time.

EXAMPLE 11–11 Consider a constrained housing as depicted in Fig. 11–19 with two direct-mount tapered roller bearings resisting an external thrust F_{ae} of 8000 N. The shaft speed is 950 rev/min, the desired life is 10 000 h, the expected shaft diameter is approximately 1 in. The reliability goal is 0.95. The application factor is appropriately $a_f = 1$.

(*a*) Choose a suitable tapered roller bearing for A.

(*b*) Choose a suitable tapered roller bearing for B.

(*c*) Find the reliabilities R_A, R_B, and R.

Solution (*a*) By inspection, note that the left bearing carries the axial load and is properly labeled as bearing A. The bearing reactions at A are

$$F_{rA} = F_{rB} = 0$$

$$F_{aA} = F_{ae} = 8000 \text{ N}$$

Since bearing B is unloaded, we will start with $R = R_A = 0.95$.

With no radial loads, there are no induced thrust loads. Eq. (11–16) is applicable.

$$F_{eA} = 0.4F_{rA} + K_A(F_{iB} + F_{ae}) = K_A F_{ae}$$

If we set $K_A = 1$, we can find C_{10} in the thrust column and avoid iteration:

$$F_{eA} = (1)8000 = 8000 \text{ N}$$

$$F_{eB} = F_{rB} = 0$$

The multiple of rating life is

$$x_D = \frac{L_D}{L_R} = \frac{\mathcal{L}_D n_D 60}{L_R} = \frac{(10\,000)(950)(60)}{90(10^6)} = 6.333$$

Then, from Eq. (11–7), for bearing A

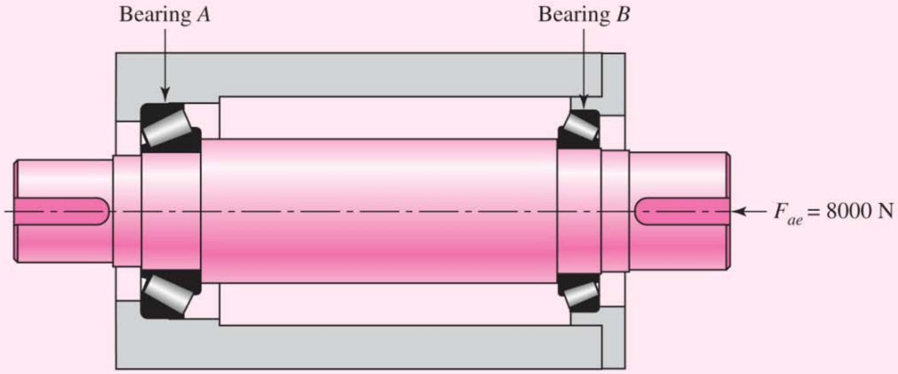
$$C_{10} = a_f F_{eA}\left[\frac{x_D}{4.48(1 - R_D)^{2/3}}\right]^{3/10}$$

$$= (1)8000\left[\frac{6.33}{4.48(1 - 0.95)^{2/3}}\right]^{3/10} = 16\,159 \text{ N}$$

Answer Figure 11–15 presents one possibility in the 1-in bore (25.4-mm) size: cone, HM88630, cup HM88610 with a thrust rating $(C_{10})_a = 17\,200$ N.

Figure 11–19

The constrained housing of Ex. 11–11.



Answer (*b*) Bearing *B* experiences no load, and the cheapest bearing of this bore size will do, including a ball or roller bearing.

(*c*) The actual reliability of bearing *A*, from Eq. (11–21), is

Answer

$$R_A \doteq 1 - \left\{ \frac{x_D}{4.48[C_{10}/(a_f F_D)]^{10/3}} \right\}^{3/2}$$

$$\doteq 1 - \left\{ \frac{6.333}{4.48\,[17\,200/(1 \times 8000)]^{10/3}} \right\}^{3/2} = 0.963$$

which is greater than 0.95, as one would expect. For bearing *B*,

Answer

$$F_D = F_{eB} = 0$$

$$R_B \doteq 1 - \left[\frac{6.333}{0.85(17\,200/0)^{10/3}} \right]^{3/2} = 1 - 0 = 1$$

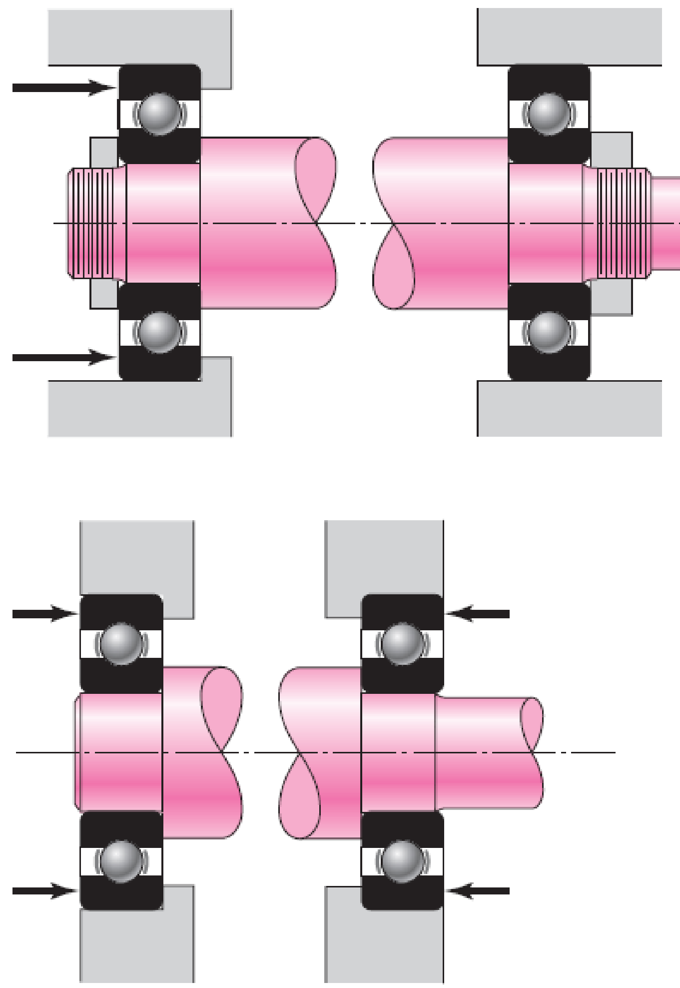
as one would expect. The combined reliability of bearings *A* and *B* as a pair is

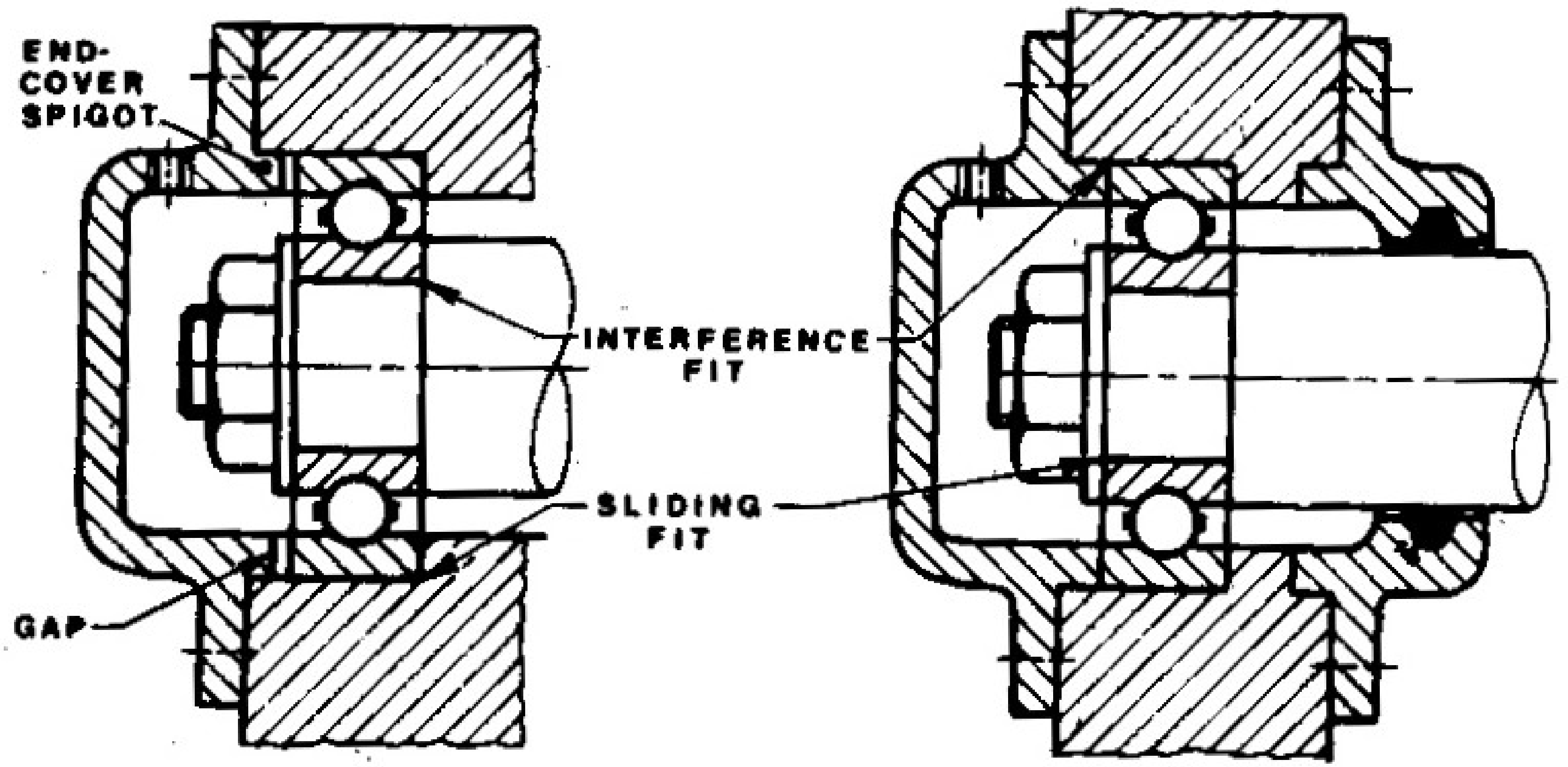
Answer

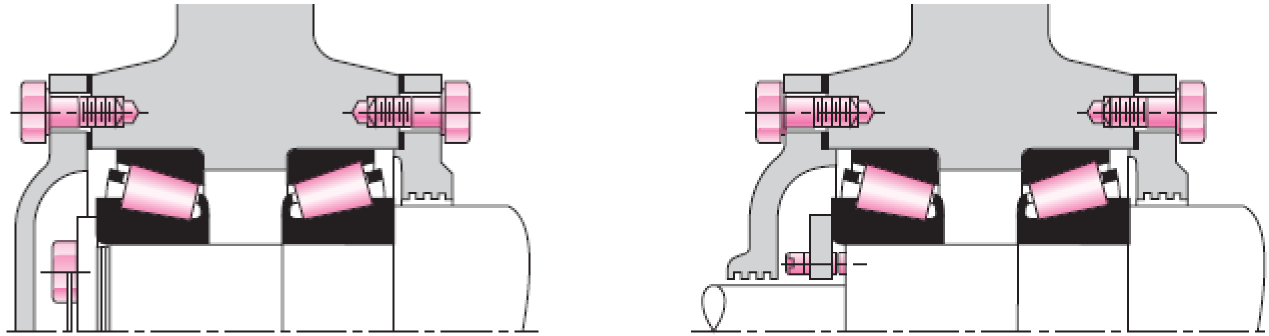
$$R = R_A R_B = 0.963(1) = 0.963$$

which is greater than the reliability goal of 0.95, as one would expect.

Mounting and Enclosure




END-
COVER
SPIGOT
INTERFERENCE
FIT
SLIDING
FIT
GAP



(*a*)

(*b*)

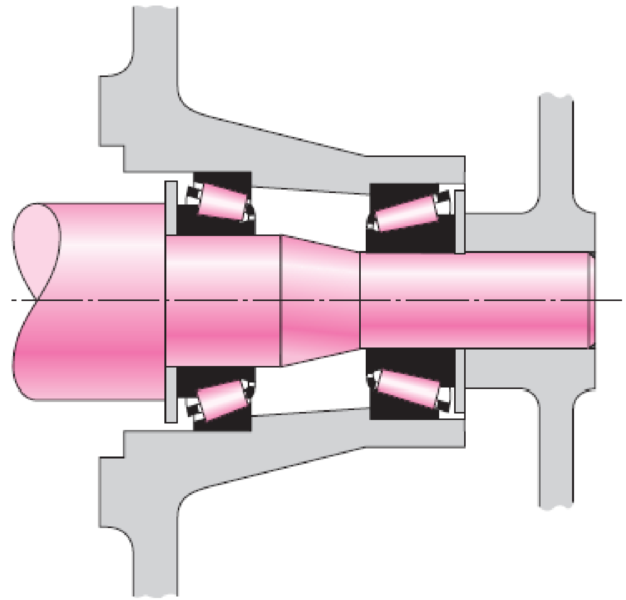


Figure 11–24

Arrangements of angular ball bearings. (*a*) DF mounting; (*b*) DB mounting; (*c*) DT mounting. *(Courtesy of The Timken Company.)*

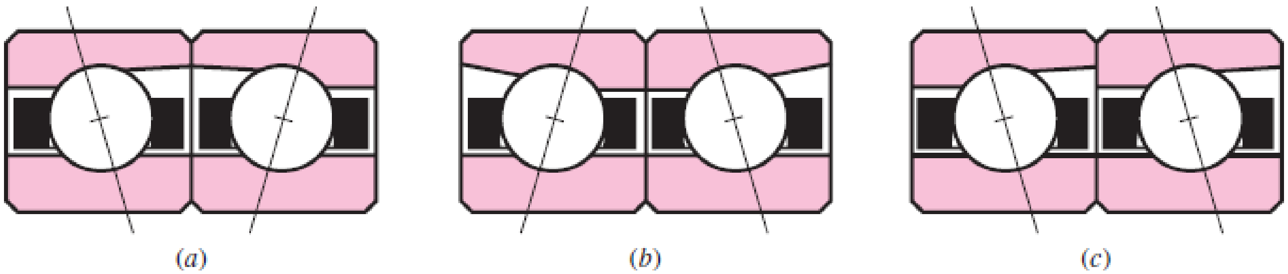
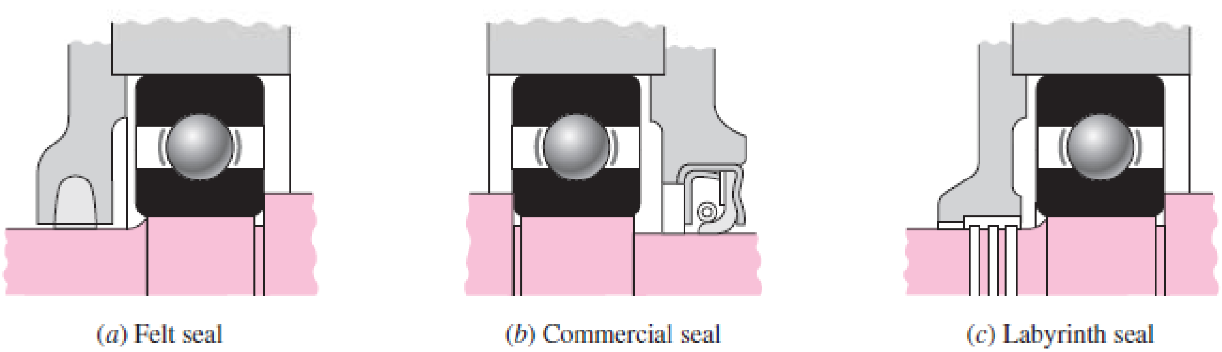
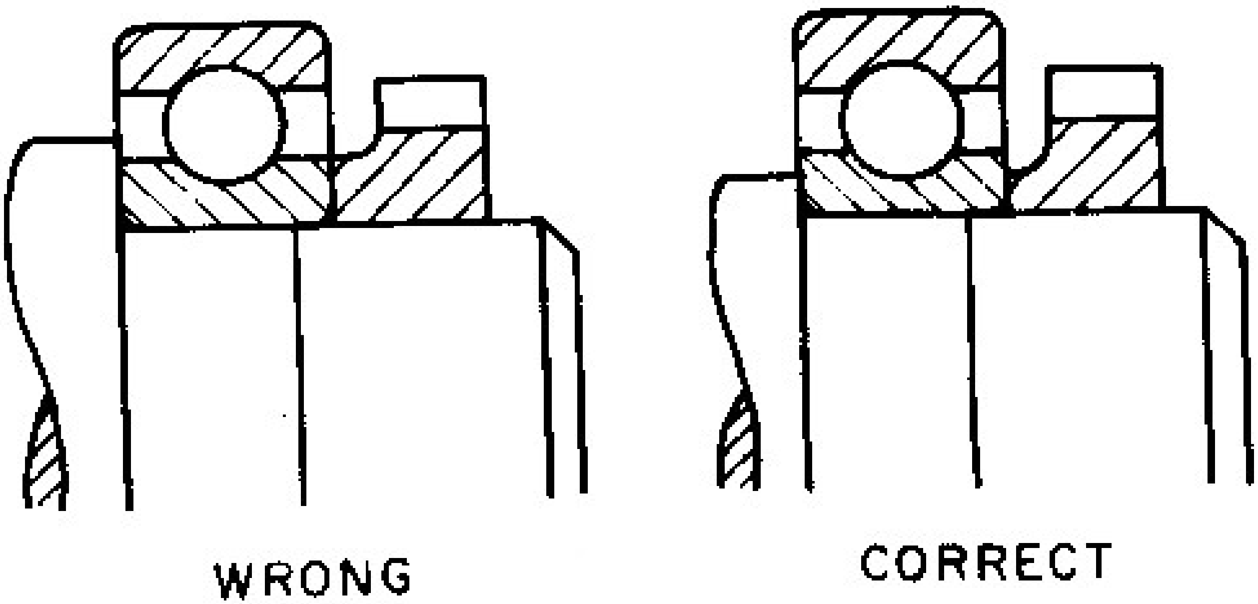


Figure 11–26

Typical sealing methods. *(General Motors Corp. Used with permission, GM Media Archives.)*



Correct assembly



PREPARATION FOR MOUNTING AND DISMOUNTING.

Before mounting, all the necessary parts, tool, equipment and data need to be at hand. It is also recommended that any drawings or instruction be studied to determine the correct order in which to assemble the various components.

Housing, shafts, seals and other components of the bearing arrangement need to be checked to see that they are clean, particularly any threaded holes, leads or grooves where remnants of previous machining operation might have collected.

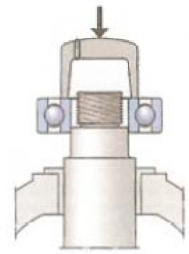
MOUNTING METHOD

Depending on the bearing type, and size, mechanical, thermal and hydraulic methods are used for mounting.

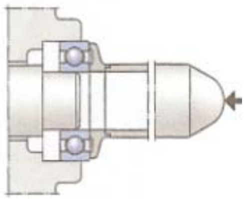
1. Cold Mounting Method.
2. Hot Mounting Method

COLD MOUNTING

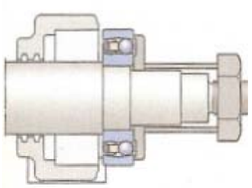
If the fit is not too tight, small bearings may be driven into position by applying light hammer blows to a sleeve placed against the bearing ring face.



The blows should be evenly distributed around the ring to prevent the bearing from tilting or skewing.



The use of the mounting dolly instead of a sleeve allows the mounting force to be applied centrally.



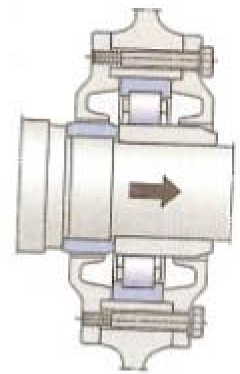
HOT MOUNTING

It is generally not possible to mount larger bearing in the cold state, as the force required to mount a bearing increases very considerably with increasing bearing size .

The inner rings or the housings are there fore heated prior to mounting.

Bearing should not be heated to more than 125c as otherwise dimensional changes caused by alterations in the structure of the bearing material may occur.

Bearing fitted with shields or seals should not be heated above 80c because of their grease fill or seal material.



DISMOUNTING METHOD

If the bearings are to be used again after removal, the force used to dismount them must never be applied through the rolling elements.

With separable bearings, the ring with the rolling element and cage assembly can be removed independently of the other ring.

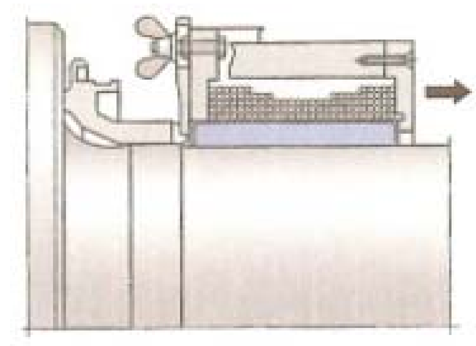
To dismount a bearing having an interference fit, the tools described in the following section may be used, the choice of tools will depend on bearing type, size and fit.

1. Cold dismounting.
2. Hot dismounting.

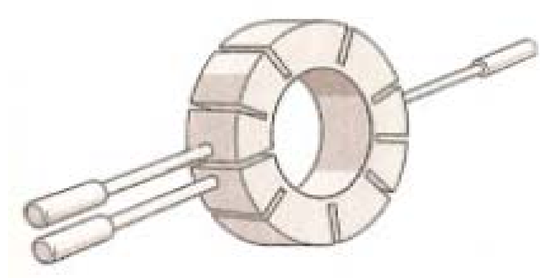
HOT DISMOUNTING

Special induction heaters have been developed to dismount the inner ring of cylindrical roller bearing having no flanges or only one flange.

They heat the inner ring rapidly without heating the shaft to any degree, so that the expanded ring can easily be removed.



Fig



BEARING LUBRICATION

Lubrication reduces friction. It also prevents wear and corrosion, and guards against

solid and liquid contamination. Theoretically, a properly lubricated bearing operating

under ideal conditions will last forever. This is not possible in reality, of course. But a

properly lubricated bearing has the best chance of achieving its maximum service life.

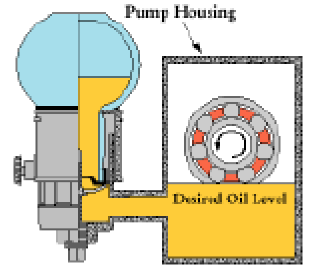
LUBRICANT SUPLLY SYSTEM

Oil and grease require different types of supply systems.

Several oil and grease supply systems exist that meet the needs of various bearing applications.

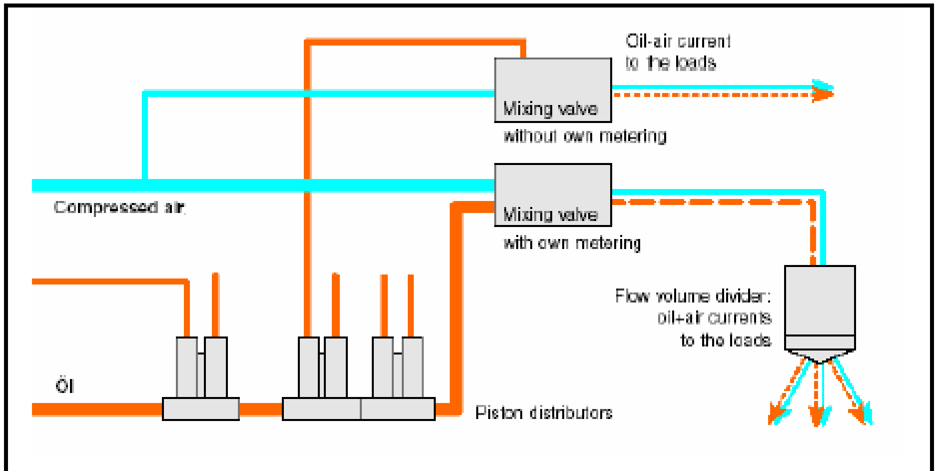
Oil supply systems include: oil baths, circulating oil systems, spray or mist systems, and the wick feed method.

Grease supply systems include: housings (with or without grease fittings), grease chamber lubrication, and the grease quantity regulator.



Oil bath with constant level oiler

LUBRICANT SUPLLY SYSTEM



Oil mist lubrication