

LECTURE NOTES
ENT345
ROLLING-CONTACT BEARINGS
Lecture 13
10/12/2015

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SCHOOL OF MECHATRONIC ENGINEERING
UniMAP

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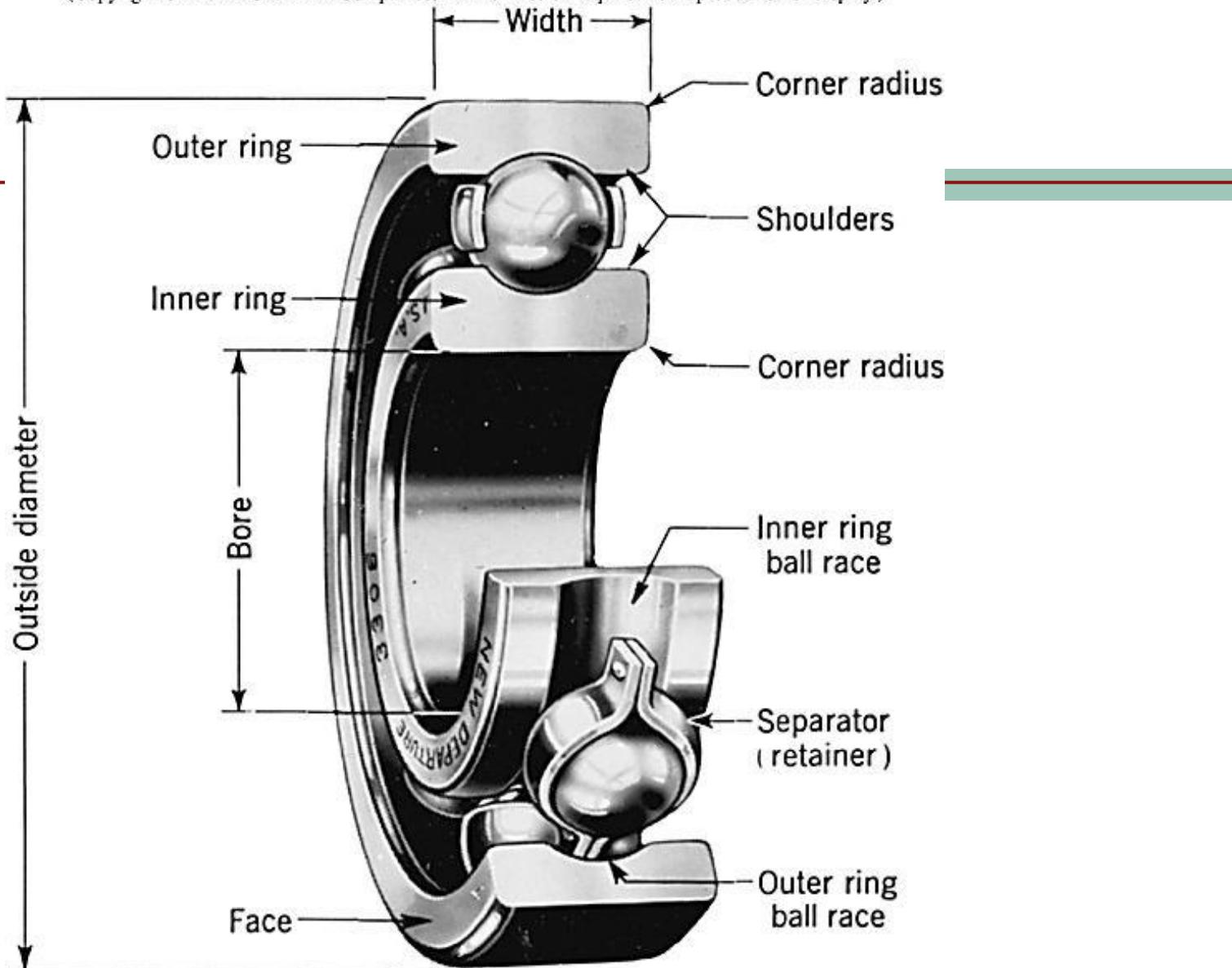
Rolling-contact bearings

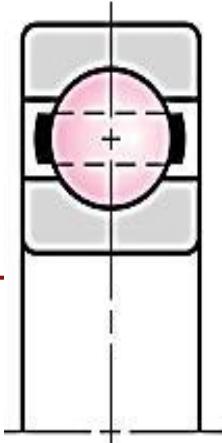
CO3:

**ABILITY TO EVALUATE
MECHANICAL COMPONENTS FOR
SELECTED MECHANICAL SYSTEMS**

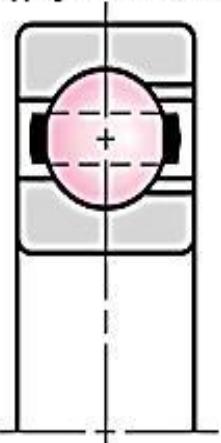
Outline

- Bearing types
- Bearing life
- Bearing load life at rated reliability
- Bearing survival: reliability versus life
- Relating load, life, and reliability
- Combined radial and thrust loading.
- Variable loading
- Selection of ball and cylindrical roller bearings
- Selection of tapered roller bearings
- Design assessment for selected rolling-contact bearings.

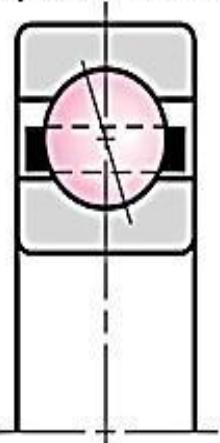




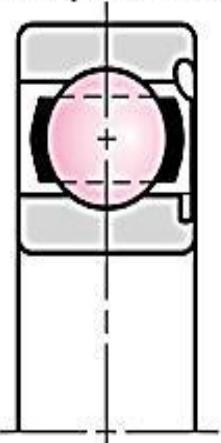
(a)
Deep groove



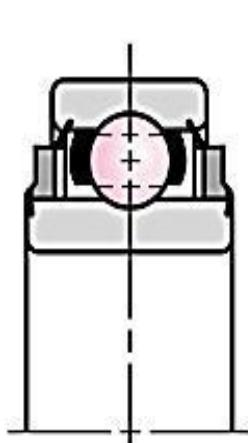
(b)
Filling notch



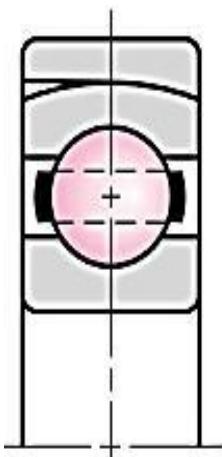
(c)
Angular contact



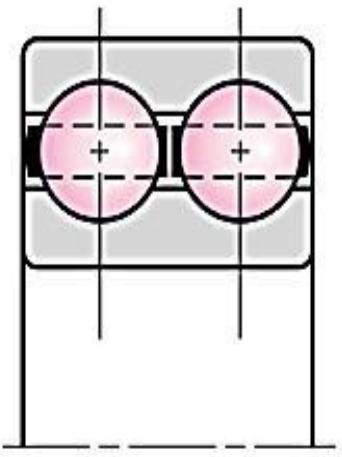
(d)
Shielded



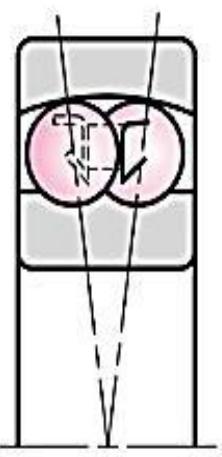
(e)
Sealed



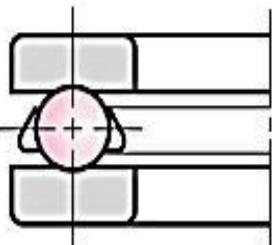
(f)
External
self-aligning



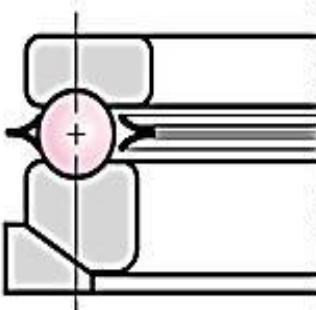
(g)
Double row



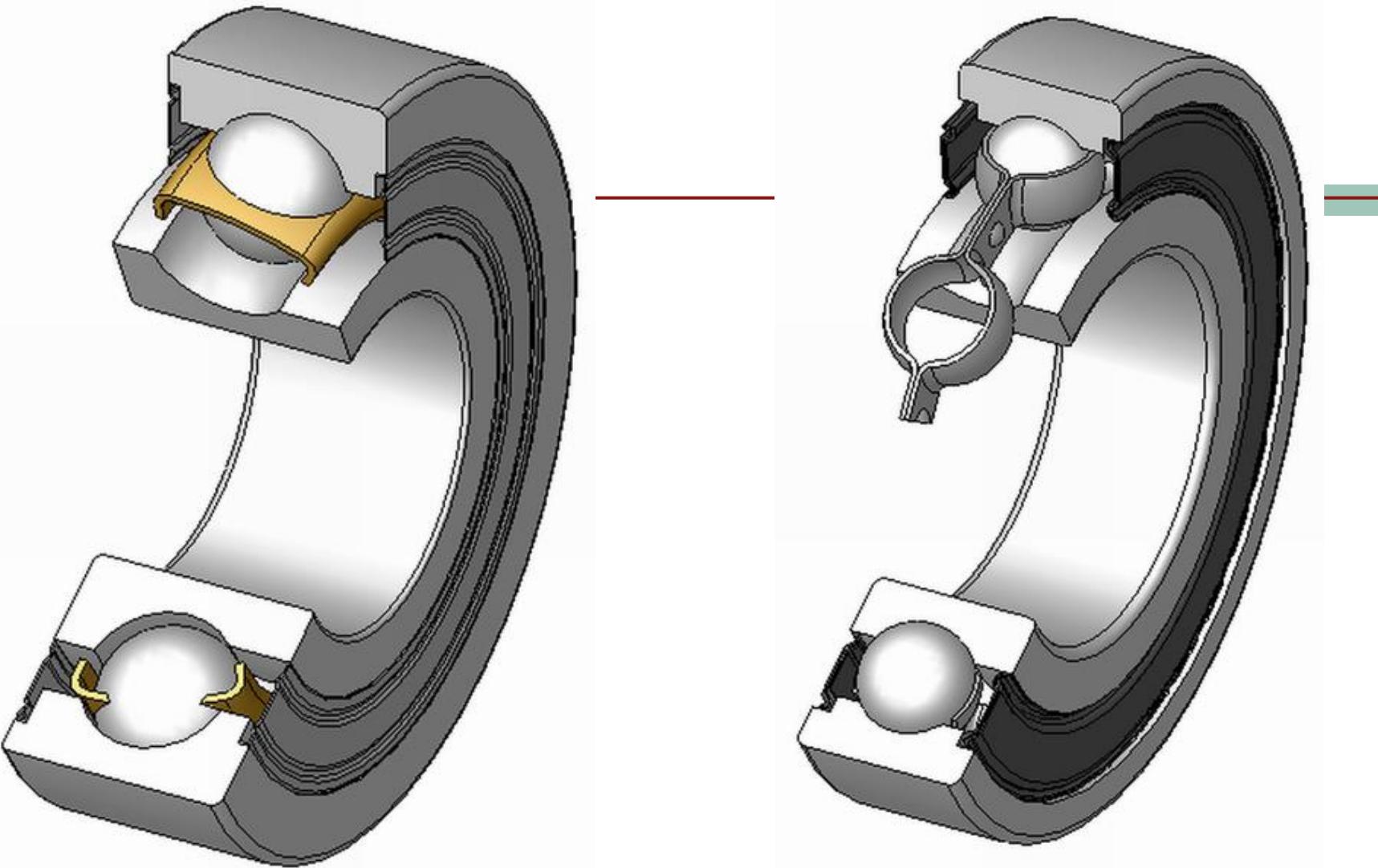
(h)
Self-aligning



(i)
Thrust



(j)
Self-aligning thrust

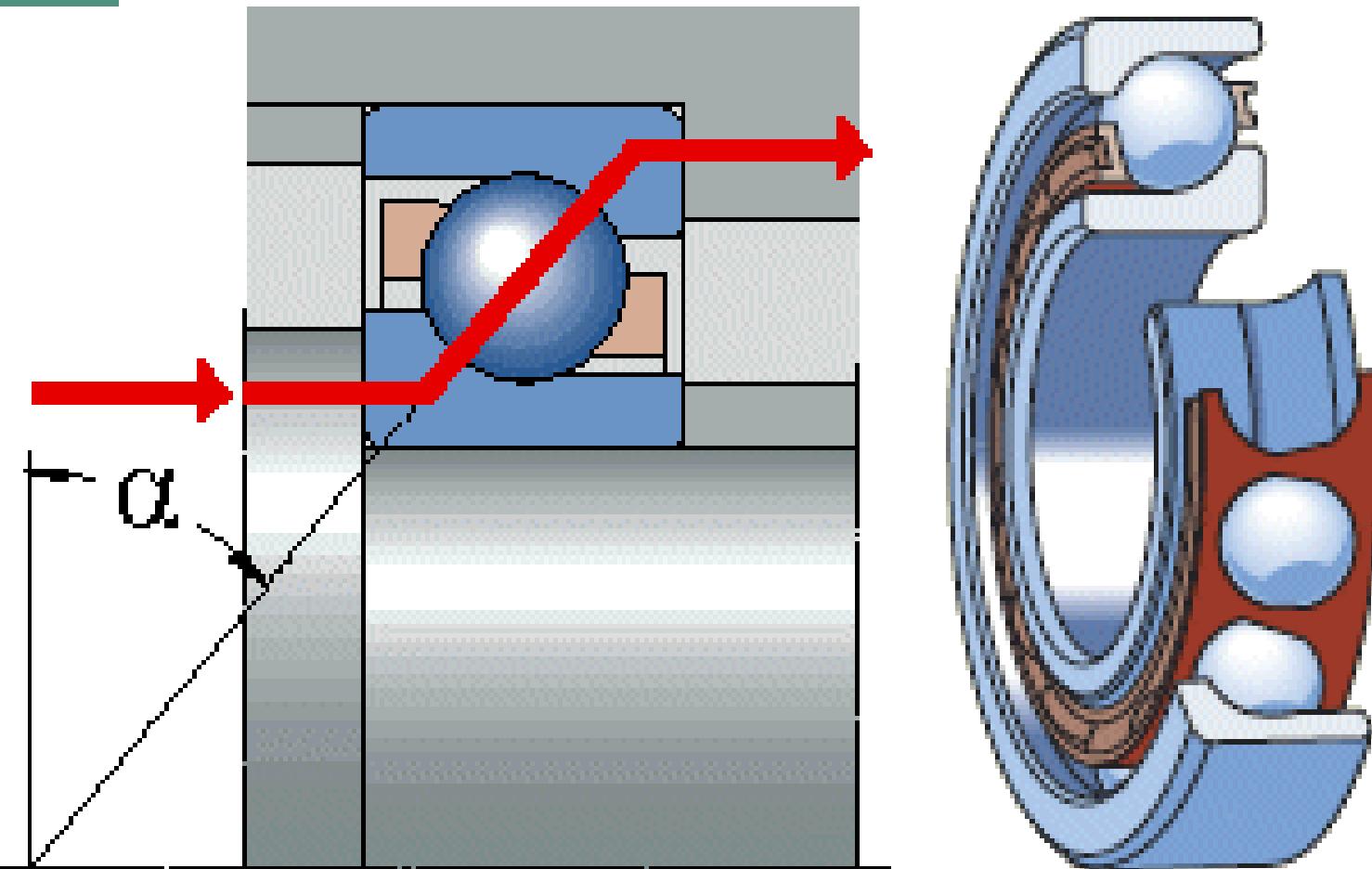


- **Angular contact bearing
groove bearing**
- **Radial and axial loads**

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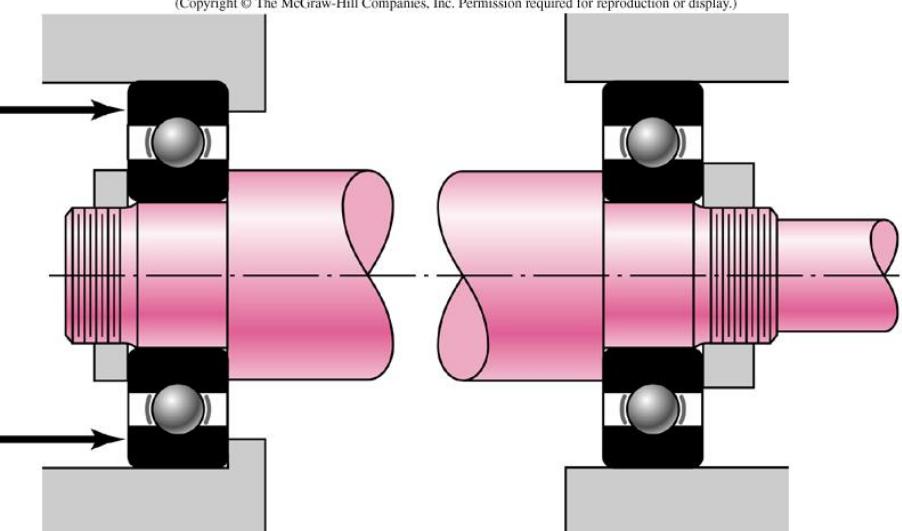
Deep

Angular contact bearing

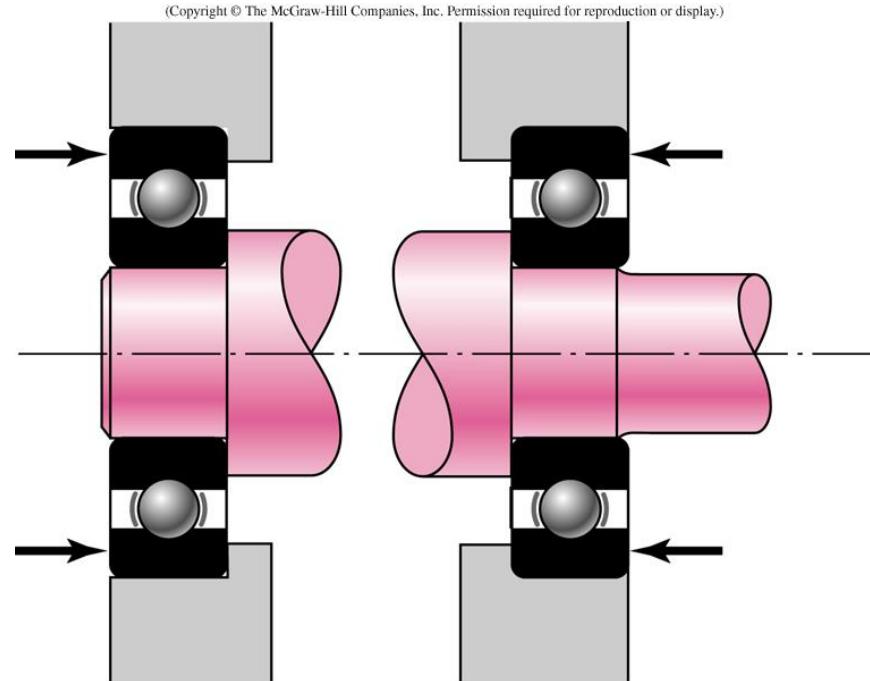


Bearing mounting

■ Common bearing mounting

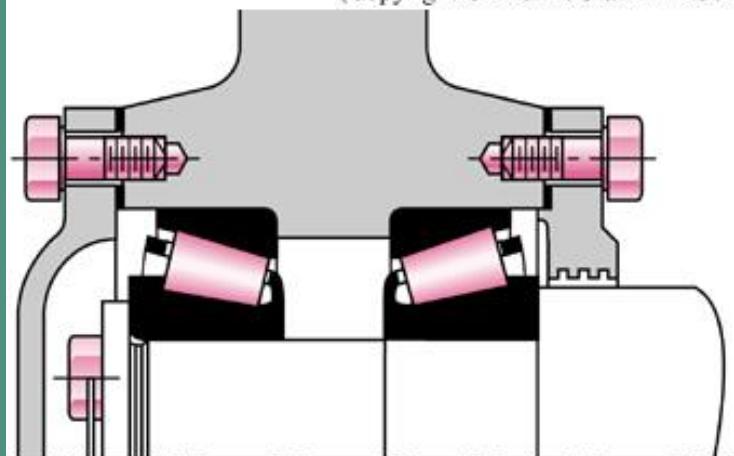


■ An alternative bearing mounting

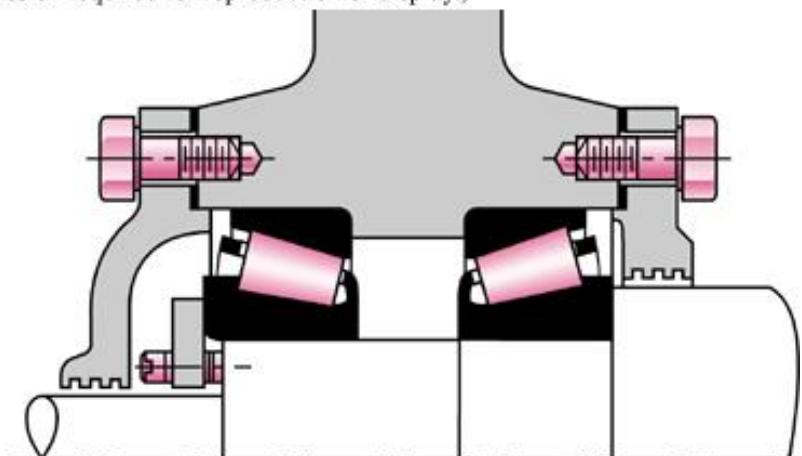


Two bearing mountings

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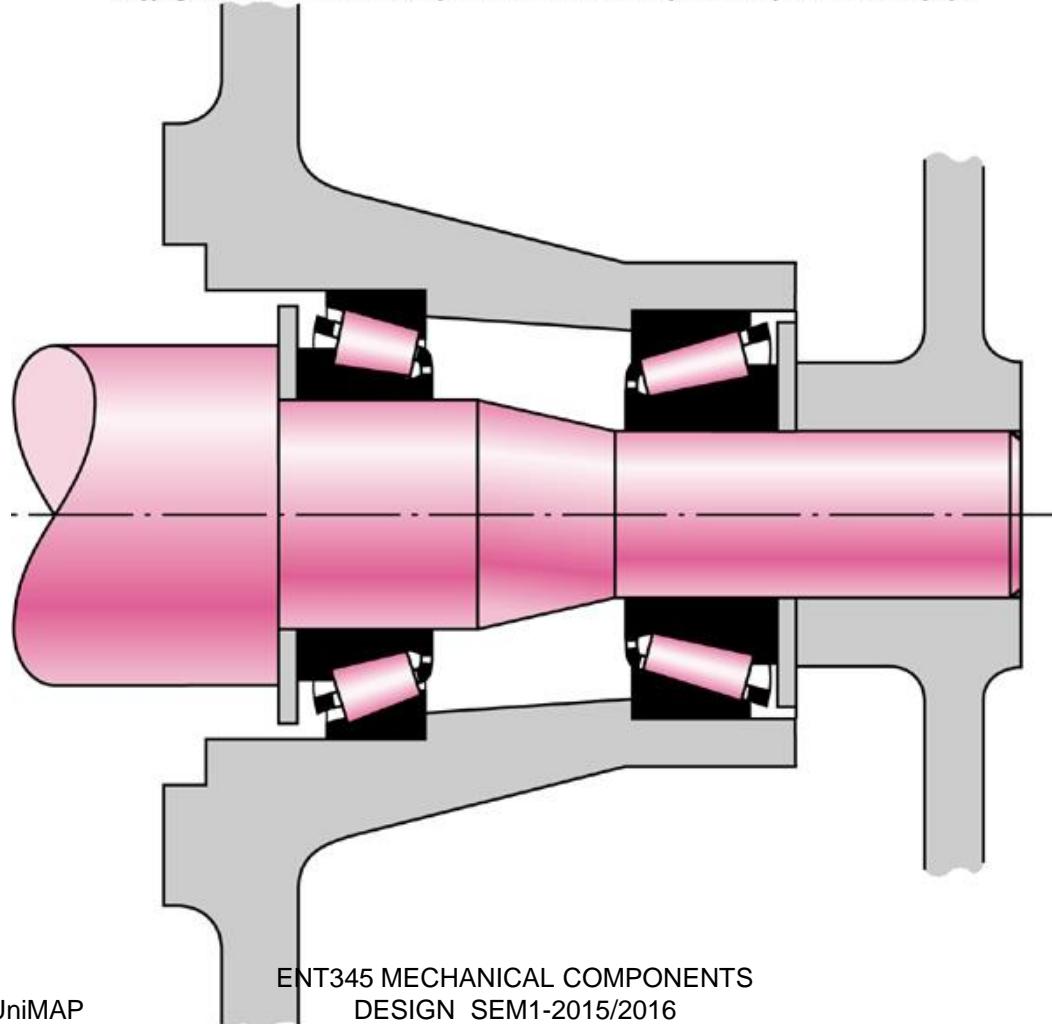
(a)



(b)

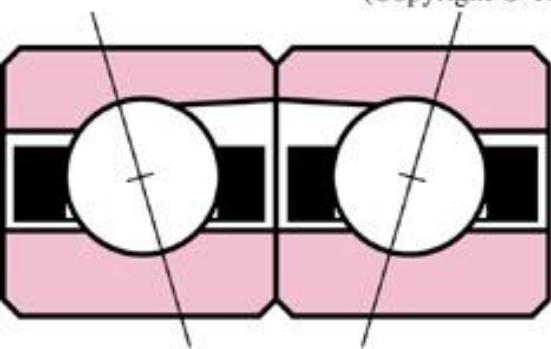
Mounting for a washing machine spindle

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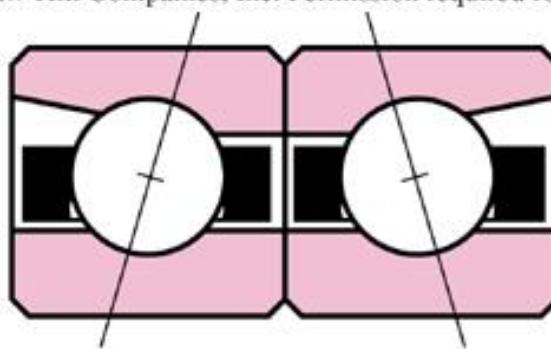


Arrangements of angular ball bearings

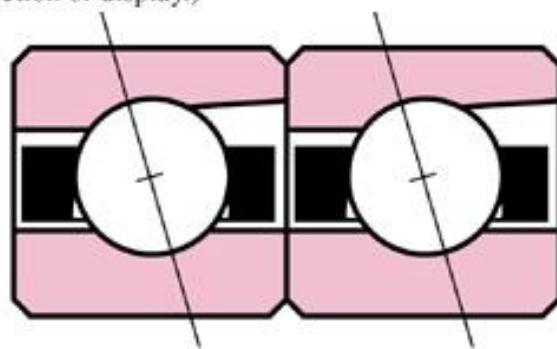
a) DF mounting.



b) DB mounting.



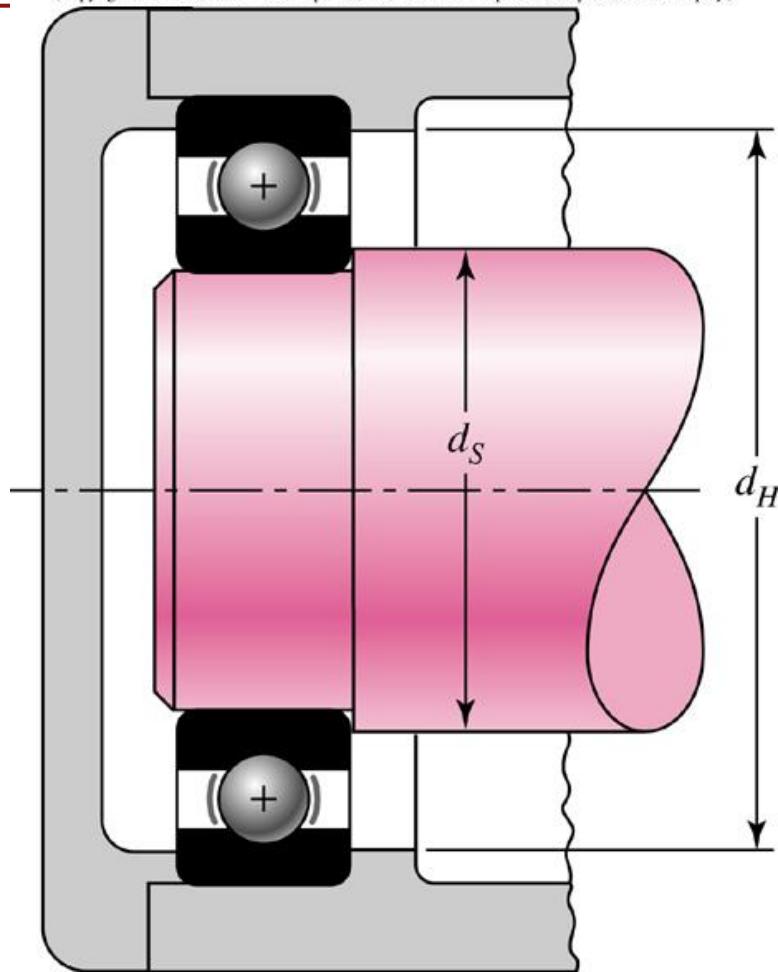
c) DT mounting.



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Shaft and housing

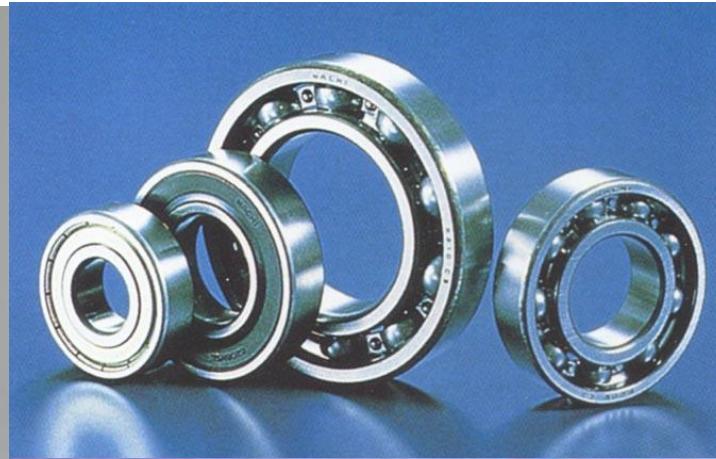
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Sleeve bearing



Ball bearing



Tapered roller bearing



Bearing Life Definitions

- **Bearing Failure:** Spalling or pitting of an area of 0.01 in²
- **Life:** Number of revolutions (or hours @ given speed) required for failure.
 - For one bearing
- **Rating Life:** *Life* required for 10% of sample to fail.
 - For a group of bearings
 - Also called *Minimum Life* or L_{10} *Life*
- **Median Life:** Average life required for 50% of sample to fail.
 - For many groups of bearings
 - Also called *Average Life* or *Average Median Life*
 - *Median Life* is typically 4 or 5 times the L_{10} *Life*

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.

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 $0.0001d$.

- 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.

Load-Life Relationship

- Nominally identical groups of bearings are tested to the life-failure criterion at different loads.
- A plot of load vs. life on log-log scale is approximately linear.
- Using a regression equation to represent the line,

$$FL^{1/a} = \text{constant} \quad (11-1)$$

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



Fig. 11-4

Load-Life Relationship

- Applying Eq. (11–1) to two load-life conditions,

$$F_1 L_1^{1/a} = F_2 L_2^{1/a} \quad (11-2)$$

- Denoting condition 1 with R for catalog rating conditions, and condition 2 with D for the desired design conditions,

$$F_R L_R^{1/a} = F_D L_D^{1/a} \quad (a)$$

- The units of L are revolutions. If life \mathcal{L} is given in hours at a given speed n in rev/min, applying a conversion of 60 min/h,

$$L = 60 \mathcal{L} n \quad (b)$$

- Solving Eq. (a) for F_R , which is just another notation for the catalog load rating,

$$C_{10} = F_R = F_D \left(\frac{L_D}{L_R} \right)^{1/a} = F_D \left(\frac{\mathcal{L}_D n_D 60}{\mathcal{L}_R n_R 60} \right)^{1/a} \quad (11-3)$$

Load-Life Relationship

$$C_{10} = F_R = F_D \left(\frac{L_D}{L_R} \right)^{1/a} = F_D \left(\frac{\mathcal{L}_D n_D 60}{\mathcal{L}_R n_R 60} \right)^{1/a} \quad (11-3)$$

- The desired design load F_D and life L_D come from the problem statement.
- The rated life L_R will be stated by the specific bearing manufacturer. Many catalogs rate at $L_R = 10^6$ revolutions.
- The catalog load rating C_{10} is used to find a suitable bearing in the catalog.
 - It is often convenient to define a dimensionless *multiple of rating life*

$$x_D = L_D / L_R$$

Example 11-1

Consider SKF, which rates in bearings for 1 million revolutions, so that L_{10} life is $60L_R n_R = 10^6$ revolutions. The $L_R n_R 60$ product produces a familiar number. Timken, for example, uses $90(10^6)$ revolutions.

If you desire a life of **5000 h** at **1725 rev/min** with a load of **2 kN** with a reliability of **90 percent**, for which catalog rating would you search in an SKF catalog.

$$\begin{aligned}C_{10} &= F_D \left(\frac{L_D n_D 60}{L_R n_R 60} \right)^{1/a} \\&= 2 \left[\frac{5000(1725)60}{10^6} \right]^{1/3} \\&= 16.02 kN\end{aligned}$$

Relating Load, Life, and Reliability

- Catalog information is at point A , at coordinates C_{10} and $x_{10}=L_{10}/L_{10}=1$, on the 0.90 reliability contour.
- The design information is at point D , at coordinates F_D and x_D , on the $R=R_D$ reliability contour.
- The designer must move from point D to point A via point B .

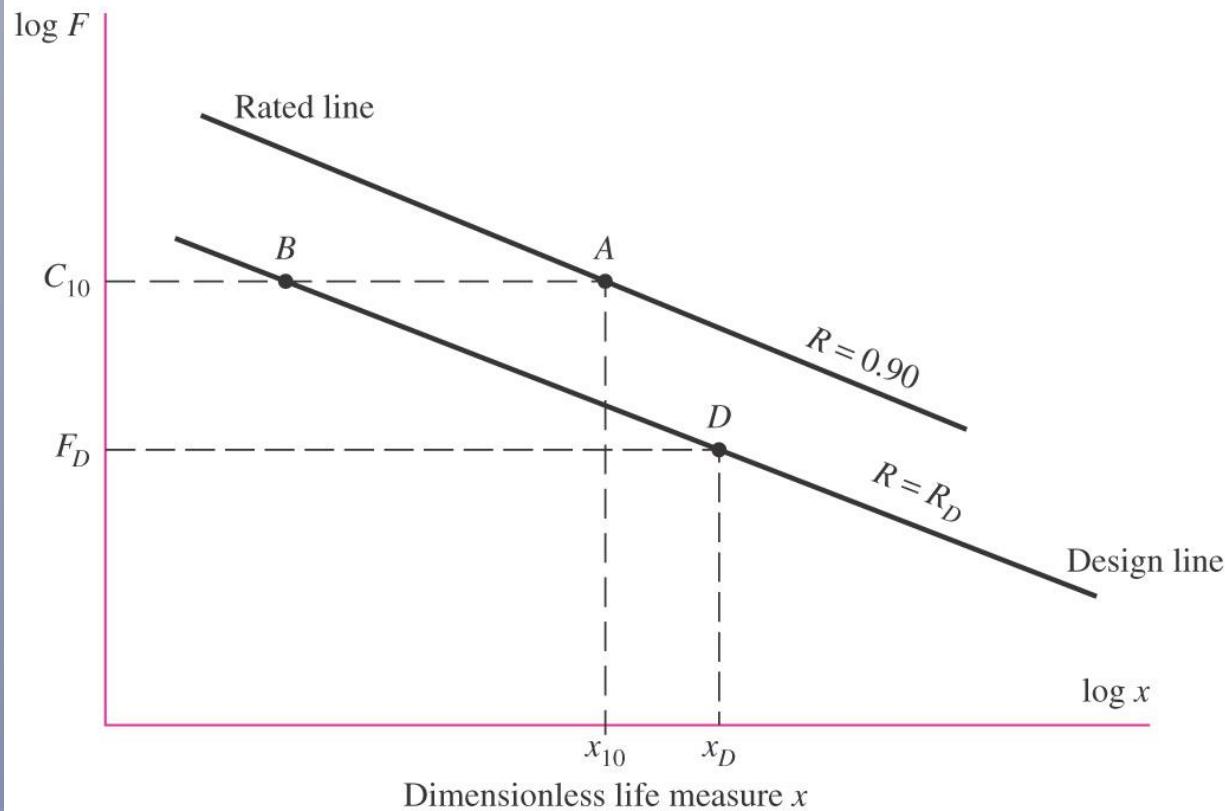


Fig. 11–5

Relating Load, Life, and Reliability

- Along a constant reliability contour (BD), Eq. (11–2) applies:

$$F_B x_B^{1/a} = F_D x_D^{1/a}$$
$$F_B = F_D \left(\frac{x_D}{x_B} \right)^{1/a} \quad (a)$$

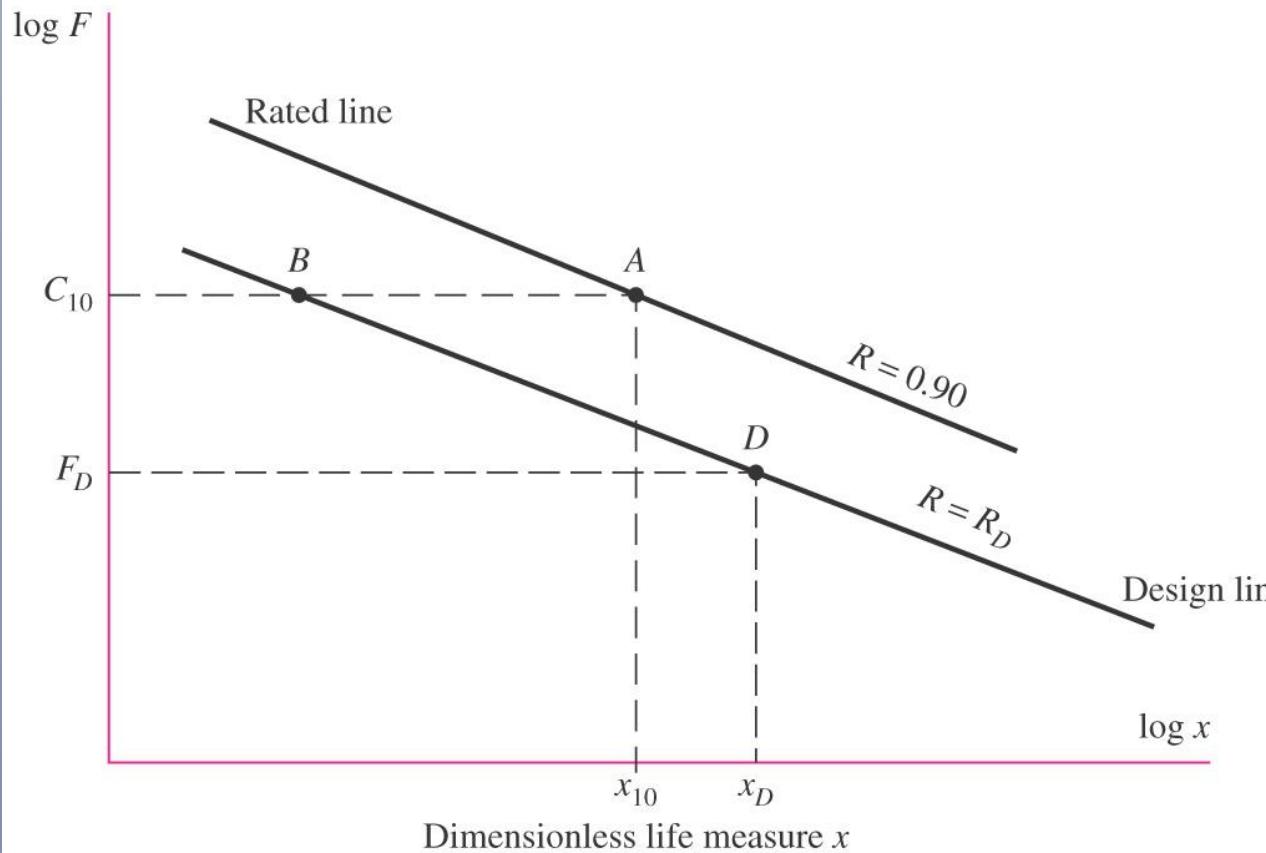


Fig. 11–5

Relating Load, Life, and Reliability

- Along a constant load line (AB), Eq. (11–4) applies:

$$R_D = \exp \left[- \left(\frac{x_B - x_0}{\theta - x_0} \right)^b \right]$$

- Solving for x_B ,

$$x_B = x_0 + (\theta - x_0) \left(\ln \frac{1}{R_D} \right)^{1/b}$$

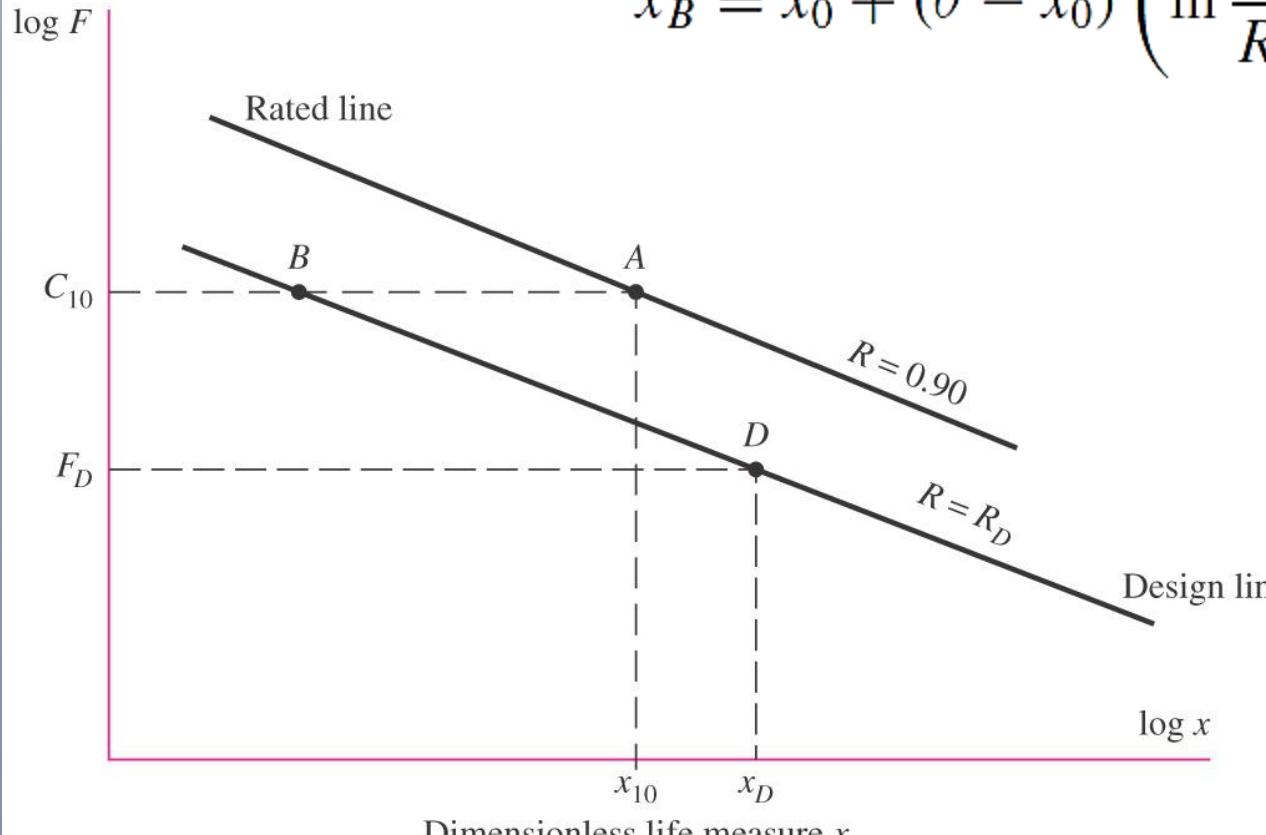


Fig. 11–5

Relating Load, Life, and Reliability

- Substituting x_B into Eq. (a),

$$F_B = F_D \left(\frac{x_D}{x_B} \right)^{1/a} = F_D \left[\frac{x_D}{x_0 + (\theta - x_0) [\ln(1/R_D)]^{1/b}} \right]^{1/a}$$

- Noting that $F_B = C_{10}$, and including an application factor a_f ,

$$C_{10} = a_f F_D \left[\frac{x_D}{x_0 + (\theta - x_0) [\ln(1/R_D)]^{1/b}} \right]^{1/a} \quad (11-9)$$

- Note that when $R_D = 0.90$, the denominator equals one and the equation reduces to Eq. (11-3).

Weibull Parameters

- The Weibull parameters x_0 , θ , and b are usually provided by the catalog.
- Typical values of Weibull parameters are given on p. 601 at the beginning of the end-of-chapter problems, and shown below.
- Manufacturer 1 parameters are common for tapered roller bearings
- Manufacturer 2 parameters are common for ball and straight roller bearings

Manufacturer	Rating Life, Revolutions	Weibull Parameters		
		x_0	θ	b
1	$90(10^6)$	0	4.48	1.5
2	$1(10^6)$	0.02	4.459	1.483

Relating Load, Life, and Reliability

$$C_{10} = a_f F_D \left[\frac{x_D}{x_0 + (\theta - x_0) [\ln(1/R_D)]^{1/b}} \right]^{1/a} \quad (11-9)$$

- Eq. (11–9) can be simplified slightly for calculator entry. Note that

$$\ln \frac{1}{R_D} = \ln \frac{1}{1 - p_f} = \ln(1 + p_f + \dots) \approx p_f = 1 - R_D$$

where p_f is the probability for failure

- Thus Eq. (11–9) can be approximated by

$$C_{10} \approx a_f F_D \left[\frac{x_D}{x_0 + (\theta - x_0)(1 - R_D)^{1/b}} \right]^{1/a} \quad R \geq 0.90 \quad (11-10)$$

Example 11-3

The design load on a ball bearing is 1840N and an application factor of 1.2 is appropriate. The speed of the shaft is to be 300rev/min, the life to be 30 kh with 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 $x_0=0.002$, $(\theta-x_0)=4.439$, and $b=1.483$

Solution

$$x_D = \frac{L}{L_{10}} = \frac{60L_D n_D}{60L_R n_R} = \frac{60(30000)300}{10^6} = 540$$

The design life is 540 times the L_{10} life. For a ball bearing, $a=3$,

$$C_{10} = F_D \left[\frac{x_D}{x_0 + (\theta - x_0)(\ln 1/R_D)^{1/b}} \right]^{1/a}$$

$$C_{10} = (1.2)(1.84) \left[\frac{540}{0.02 + 4.459[(1-0.99)]^{1/1.483}} \right]^{1/3}$$

$$C_{10} = 29.716kN$$

Example

- A certain application requires a ball bearing with the inner ring rotating, with a design life of 30000 h at a speed of 300 rev/min. The radial load is 1.898 kN and an application factor of 1.2 is appropriate. The reliability goal (R) is 0.90.
- Find the multiple of rating the required, x_D , and the catalog rating C_{10} with which to enter a bearing table.
- **Choose a 02-series deep-groove ball bearing** from Table 11-2 and estimate the reliability in use.
- Solution
The deep-groove 02-series ball bearing

$$x_D = \frac{30000(300)(60)}{10^6} = 540$$

Example

The design radial load F_D is

$$F_D = 1.2(1.898\text{kN}) = 2.278\text{kN}$$

Weibull parameter;

$$X_o = 0.02$$

$$(\theta - x_0) = 4.439$$

$$b = 1.43$$

$$C_{10} = F_D \left[\frac{x_D}{x_0 + (\theta - x_0)(\ln 1/R_D)^{1/b}} \right]^{1/a}$$

$$C_{10} = 2.278 \left[\frac{540}{0.02 + 4.459[\ln(1/0.9)]^{1/1.483}} \right]^{1/3}$$

$$C_{10} = 18.59\text{kN}$$

- $C_{10} = 18.59\text{kN}$
- From Table 11-2, choose a 02-30mm with $C_{10} = 19.5\text{kN}$

$$x_B = x_0 + (\theta - x_0) \left(\ln \frac{1}{R_D} \right)^{1/b}$$

$$F_B = F_D \left(\frac{x_D}{x_B} \right)^{1/a} \Rightarrow \frac{x_D}{x_B} = \left(\frac{F_B}{F_D} \right)^a \Rightarrow$$

$$x_B = \frac{x_D}{\left(\frac{F_B}{F_D} \right)^a} = \frac{540}{\left(\frac{19.5}{2.278} \right)^3} = 0.857$$

$$R = \exp \left[- \left(\frac{x - x_0}{\theta - x_0} \right)^b \right] =$$

$$R = \exp \left\{ - \left[\frac{0.857 - 0.02}{4.459} \right]^{1.438} \right\} = 0.913$$

Example

An angular-contact, inner ring rotating, 02-series ball bearing is required for an application in which the life requirement is 50000 h at 480 rev/min. The design radial load is 610 lbf. The application factor is 1.4. The reliability goal is 0.90.

Find the multiple of rating life x_D required and the catalog rating C_{10} with which to enter Table 11-2.

Choose a bearing and estimate the existing reliability in service.

Solution

The angular-contact 02-series ball bearing, the rating life

$$x_D = \frac{L}{L_{10}} = \frac{60L_D n_D}{60L_R n_R} = \frac{60(50000)480}{10^6} = 1440$$

$$F_D = 1.4(610) = 854 \text{ lbf} = 3.80 \text{ kN}$$

$$C_{10} = 854 \left\{ \frac{1440}{0.02 + 4.459[\ln(1/0.9)]^{1/1.483}} \right\}^{1/3}$$

$$C_{10} = 9665 \text{ lbf} = 43 \text{ kN}$$

Table 11-2 Select a 02-55mm with
 $C_{10}=46.2 \text{ kN}$

Example

$$R = \exp\left[-\left(\frac{x - x_0}{\theta - x_0}\right)^b\right] =$$
$$R = \exp\left\{-\left[\frac{14409(3.8/46.2)^3 - 0.02}{4.459}\right]^{1.438}\right\} = 0.927$$

Combined Reliability of Multiple Bearings

- If the combined reliability of multiple bearings on a shaft, or in a gearbox, is desired, then the total reliability is equal to the product of the individual reliabilities.
- For two bearings on a shaft, $R = R_A R_B$
- If the bearings are to be identical, each bearing should have a reliability equal to the square root of the total desired reliability.
- If the bearings are not identical, their reliabilities need not be identical, so long as the total reliability is realized.

Dimension-Series Code

- ABMA standardized *dimension-series code* represents the relative size of the boundary dimensions of the bearing cross section for metric bearings.
- Two digit series number
- First digit designates the width series
- Second digit designates the diameter series
- Specific dimensions are tabulated in catalogs under a specific series

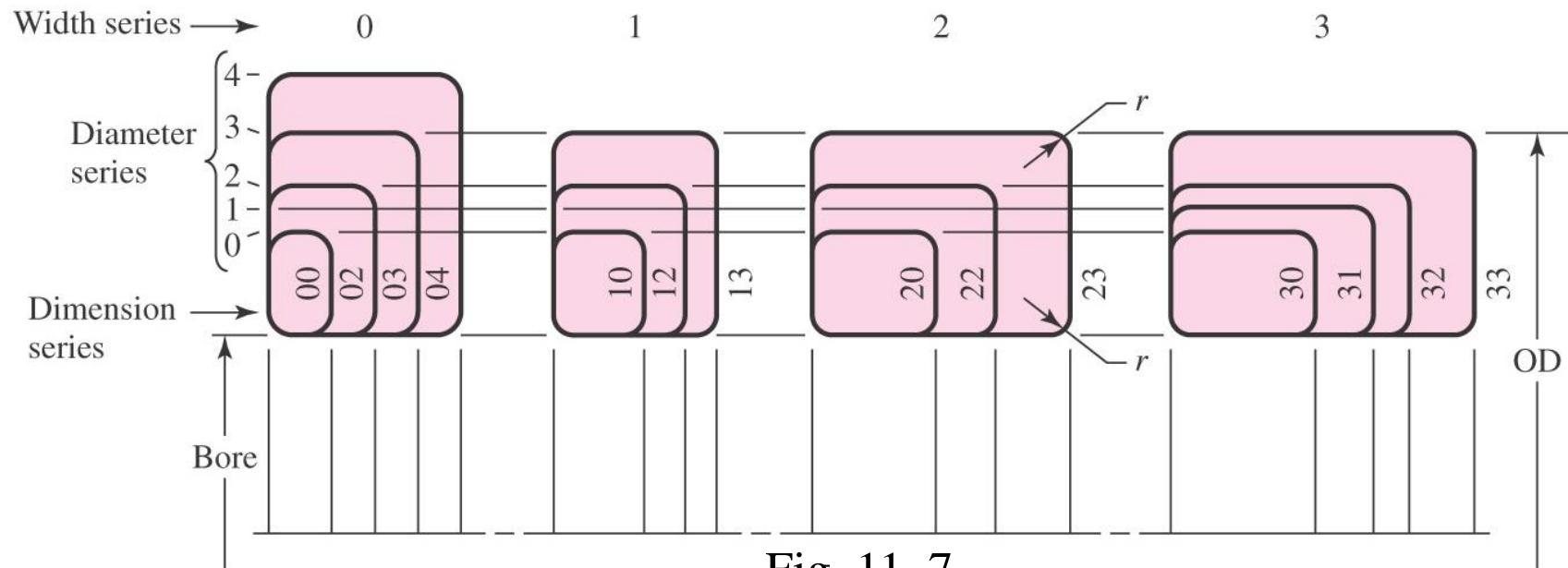


Fig. 11-7

Representative Catalog Data for Ball Bearings (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		Load Ratings, kN		Angular Contact	
				d_S	d_H	Deep Groove	Angular Contact	C_{10}	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

Representative Catalog Data for Cylindrical Roller Bearings

(Table 11–3)

02-Series					03-Series				
Bore, mm	OD, mm	Width, mm	Load Rating, kN		OD, mm	Width, mm	Load Rating, kN		
			C ₁₀	C ₀			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	

Combined radial and thrust loading

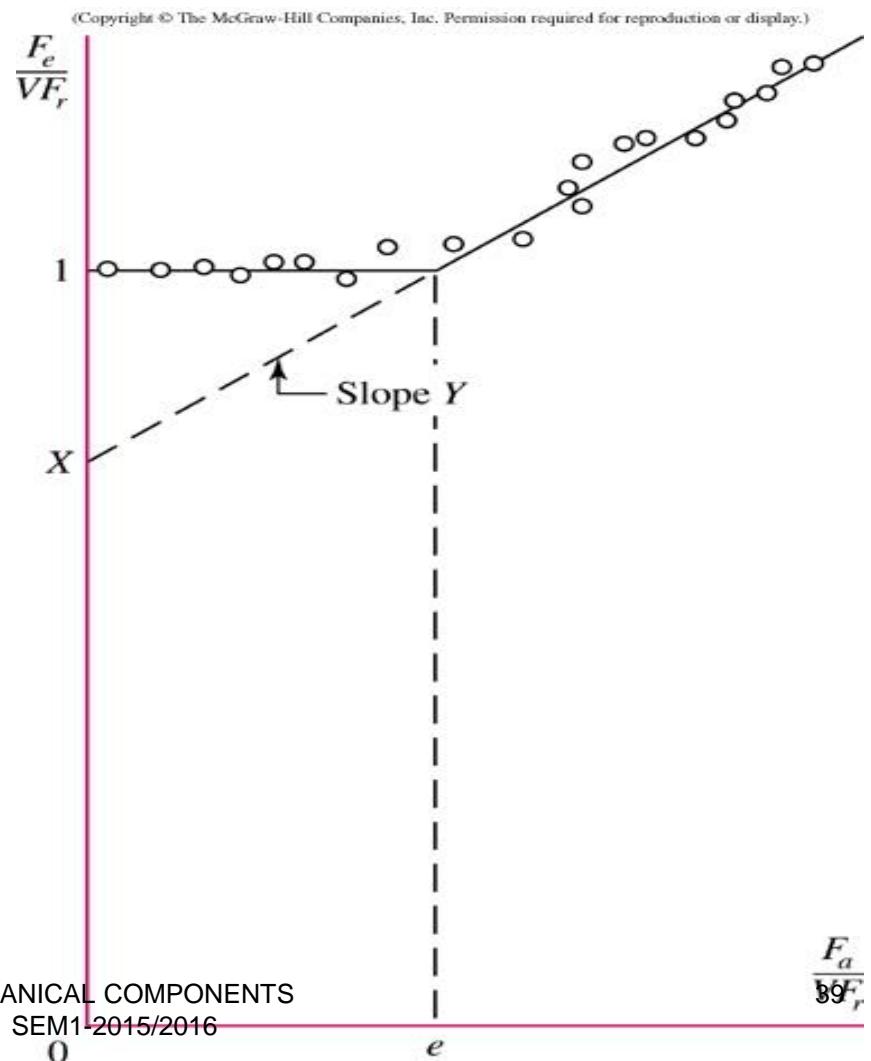
A ball bearing is capable of resisting radial loading and a thrust loading. Furthermore, these can be combined.

F_a = the axial thrust load.

F_r = the radial load.

F_e = equivalent radial load that does the same damage as the combined radial and thrust loads together.

The abscissa e is defined by the intersection of the two lines.



Combined Radial and Thrust Loading

- The data can be approximated by two straight lines

$$\frac{F_e}{VF_r} = 1 \quad \text{when } \frac{F_a}{VF_r} \leq e$$

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

- X is the ordinate intercept and Y is the slope
- Basically indicates that F_e equals F_r for smaller ratios of F_a/F_r , then begins to rise when F_a/F_r exceeds some amount e

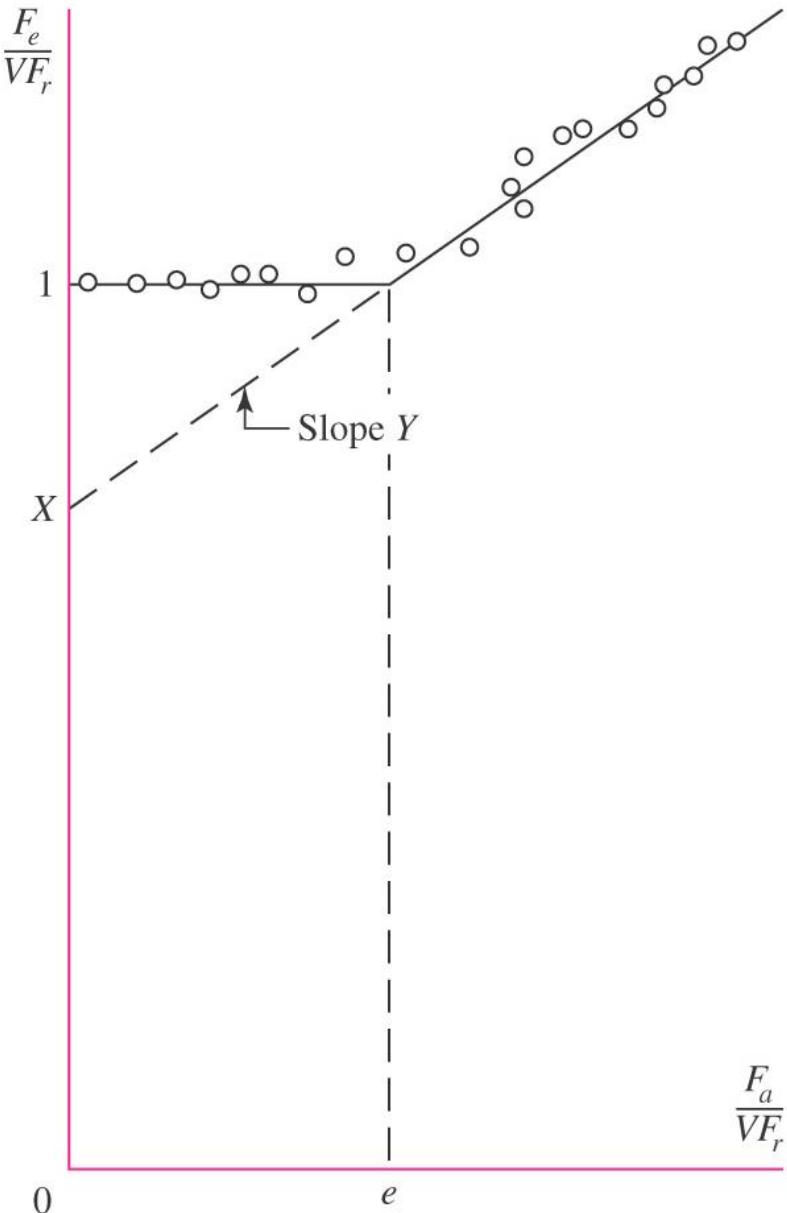


Fig. 11-6

Combined Radial and Thrust Loading

- It is common to express the two equations as a single equation

$$F_e = X_i V F_r + Y_i F_a \quad (11-12)$$

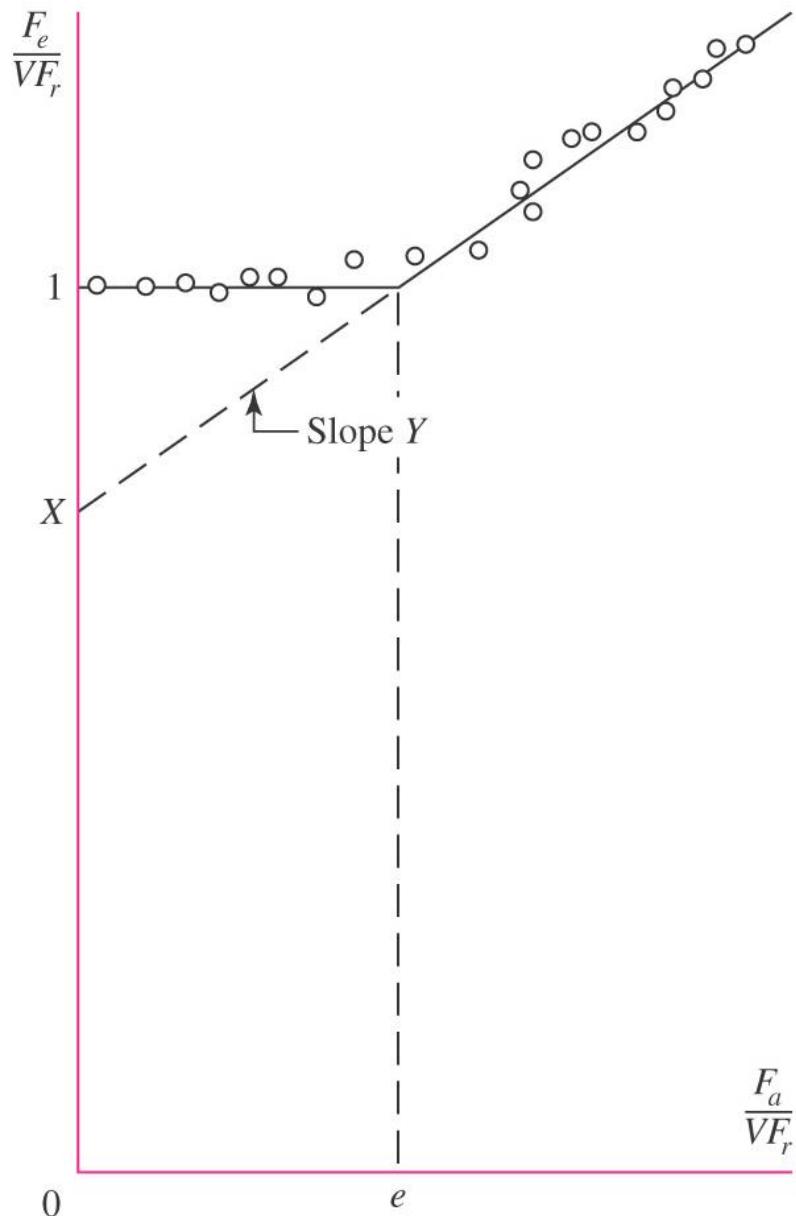
where

$$i = 1 \text{ when } F_a / (V F_r) \leq e$$

$$i = 2 \text{ when } F_a / (V F_r) > e$$

- X and Y factors depend on geometry and construction of the specific bearing.

Fig. 11-6



Equivalent Radial Load Factors for Ball Bearings

$$F_e = X_i V F_r + Y_i F_a \quad (11-12)$$

- X and Y for specific bearing obtained from bearing catalog.
- Table 11–1 gives representative values in a manner common to many catalogs.

Table 11–1

F_a/C_0	e	$F_a/(V F_r) \leq e$		$F_a/(V F_r) > e$	
		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 $F_a/C_0 < 0.014$.

Equivalent Radial Load Factors for Ball Bearings

$$F_e = X_i V F_r + Y_i F_a \quad (11-12)$$

Table 11–1

F_a/C_0	e	$F_a/(V F_r) \leq e$		$F_a/(V F_r) > e$	
		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

- X and Y are functions of e , which is a function of F_a/C_0 .
- C_0 is the *basic static load rating*, which is tabulated in the catalog.

Bearing Life Recommendations (Table 11–4)

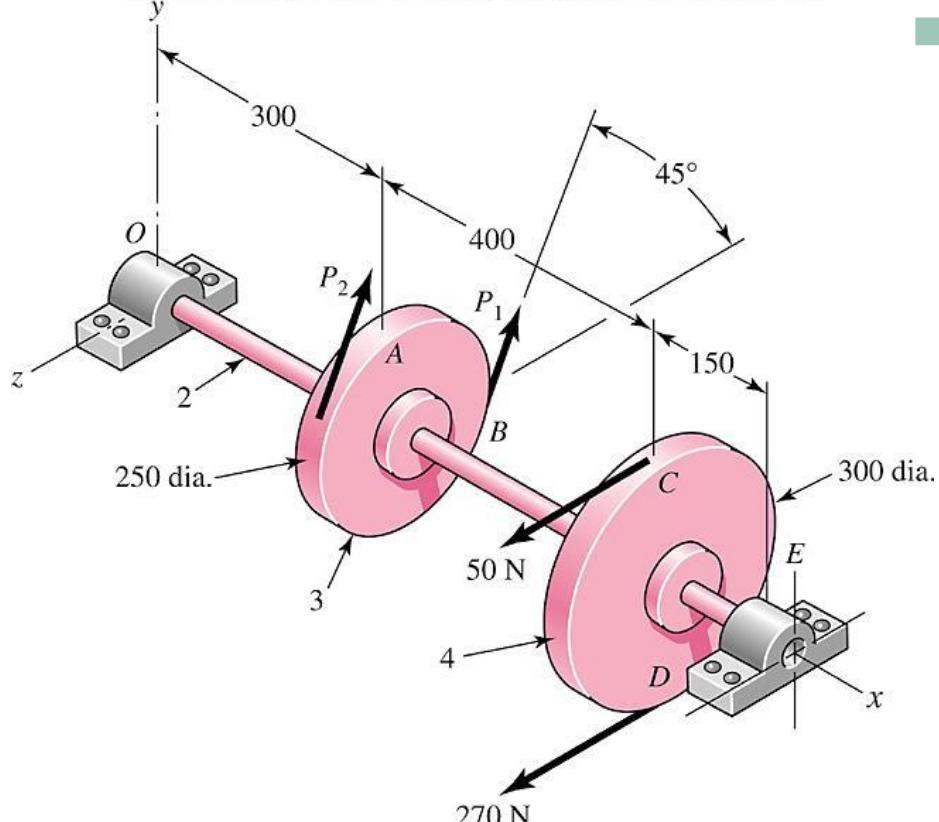
Type of Application	Life, kh
Instruments and apparatus for infrequent use	Up to 0.5
Aircraft engines	0.5–2
Machines for short or intermittent operation where service interruption is of minor importance	4–8
Machines for intermittent service where reliable operation is of great importance	8–14
Machines 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

Recommended Load Application Factors (Table 11–5)

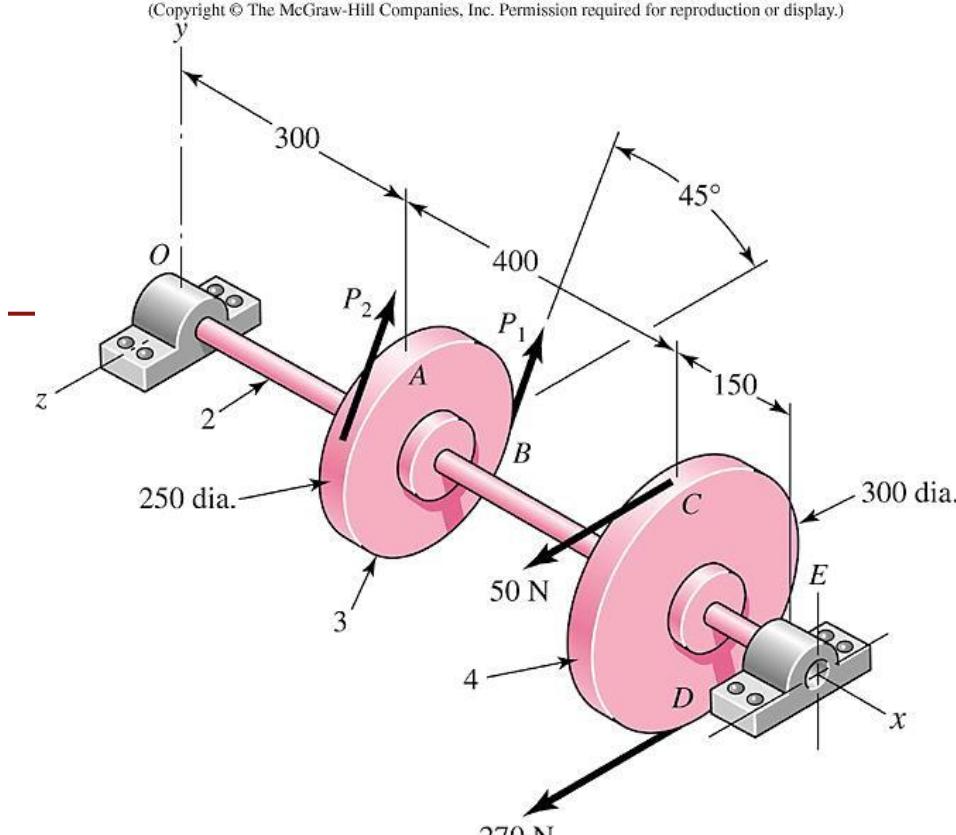
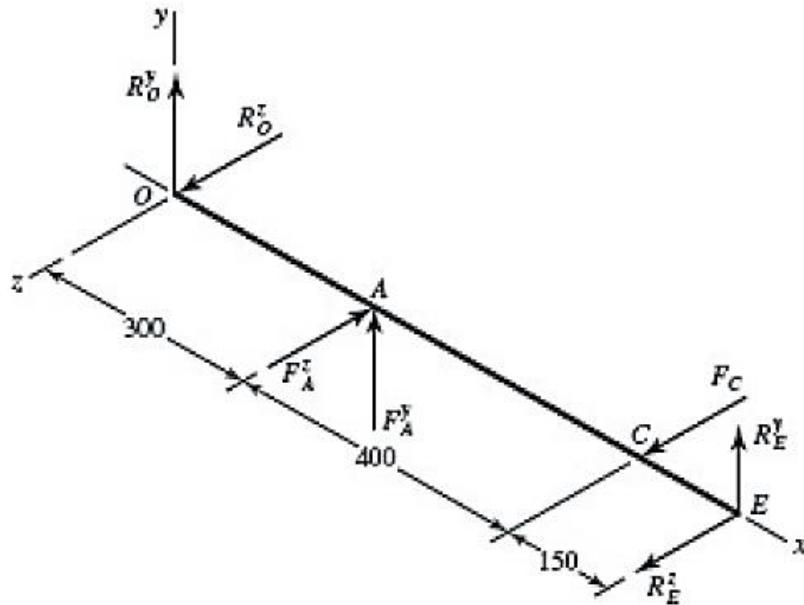
Type of Application	Load Factor
Precision gearing	1.0–1.1
Commercial gearing	1.1–1.3
Applications with poor bearing seals	1.2
Machinery with no impact	1.0–1.2
Machinery with light impact	1.2–1.5
Machinery with moderate impact	1.5–3.0

Problem 11-33

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- The figure is schematic drawing of a countershaft that supports two V-belt pulleys. The countershaft runs at 1500 rev/min and the bearings are to have a life of 60 kh at a combined reliability of 0.98. The belt tension on the loose side of pulley A is 15 percent of tension on the tight sight. Select deep-groove bearings from Table 11-2 for use at O and E, using an application factor of unity.



$$\text{Torque (T)} = (270N - 50N)150\text{mm} = 33000 \text{ Nmm}$$

$$T = (P_1 - P_2)125 = (P_1 - 0.15P_1)125 \text{ Nmm}$$

$$P_1 = 310.6 \text{ N}$$

$$P_2 = 0.15(310.6)\text{N} = 46.6 \text{ N}$$

$$P_1 + P_2 = 310.6 + 46.6 = 357.2 \text{ N}$$

$$F_A^Y = F_A^Z = 357.2 \sin 45^\circ = 252.6 \text{ N}$$

$$\sum M_{0Z} = 850R_{Ey} + (300(252.6)) = 0$$

$$R_{Ey} = -89.2 \text{ N}$$

$$\sum F_y = 252.6 - 89.2 + R_{oy} = 0$$

$$R_{oy} = -163.4 \text{ N}$$

$$\sum M_{oy} = -850R_{EZ} - 700(320) + 300(252.6) = 0$$

$$R_{oE} = -174.4 \text{ N}$$

$$\sum z = -174.4 + 320 - R_{oz} = 0$$

$$R_{oz} = 107 \text{ N}$$

$$R_o = \sqrt{(-163.4)^2 + (107)^2} = 195N$$

$$R_E = \sqrt{(-89.2)^2 + (-174.4)^2} = 196N$$

The radial loads the same at O and E

The reliability of the individual bearings is $R = \sqrt{0.98} = 0.9899$

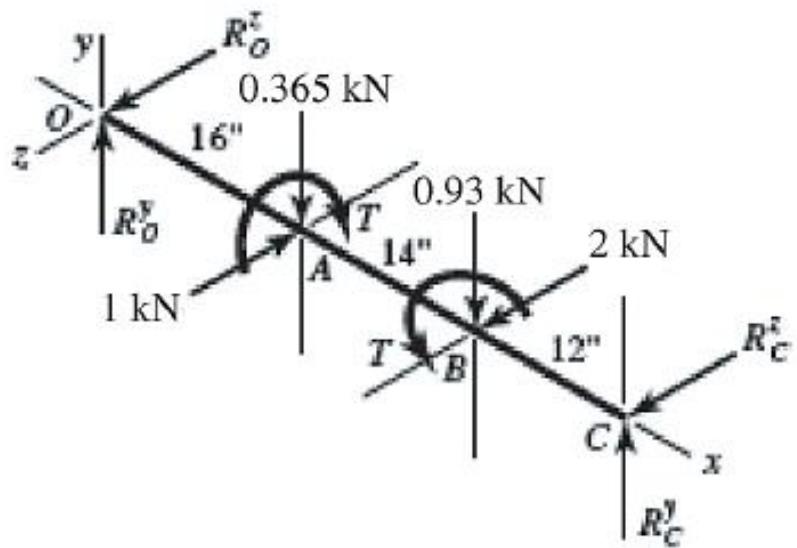
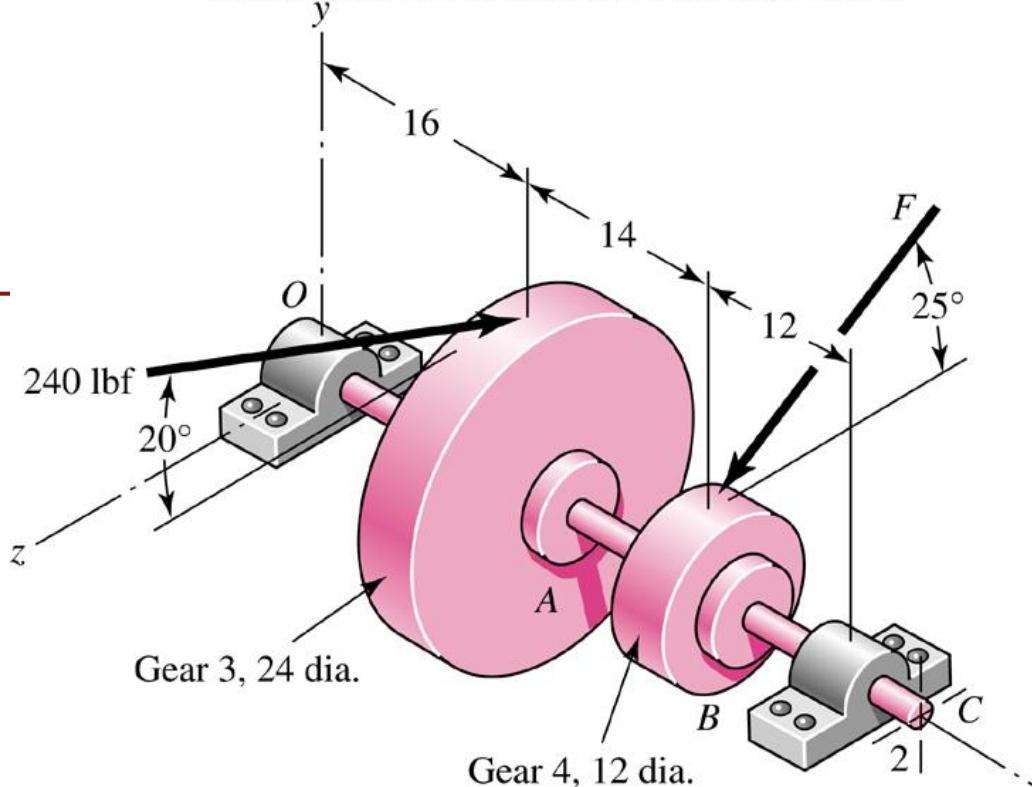
$$X_D = \frac{60000(1500)(60)}{10^6} = 5400N$$

$$C_{10} = (1)(0.186) \left\{ \frac{5400}{0.02 + 4.439[\ln(1/0.9899)]^{1/1.483}} \right\}^{1/3} = 5.7kN$$

Table 11 – 2 select an 02 – 20 mm deep groove with a load rating of 6.89 kN

Problem 11-34

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 application factor and reliability goal for the bearing pair of 0.95. Select the bearings from Table 11-2.



$$T = 300(1068 \cos 20^\circ) = 301.1 \text{ kN}$$

$$F = \frac{T}{\frac{300}{2} \cos 25^\circ} = 2.2 \text{ kN}$$

xy plane

$$\sum M_{z_o} = -400(0.36) - 750(0.93) + 1050R_c y = 0$$

$$R_c y = 0.803 \text{ kN}$$

$$R_o y = 0.365 + 0.93 - 0.803 = 0.492 \text{ kN}$$

xz-plane

$$\sum M_{y_o} = 1(400) - 2(750) - 1050R_c z = 0$$

$$R_c z = -1.05 \text{ kN}$$

$$R_o z = 1 - 2 + 1.05 = 0.05 \text{ kN}$$

$$R_o = (0.492^2 + 0.05^2)^{1/2} = 0.495 \text{ kN}$$

$$R_c = (0.803^2 + 1.05^2)^{1/2} = 1.322 \text{ kN}$$

$$x_D = \frac{50000(200)(60)}{10^6} = 480$$

$$R = \sqrt{0.95} = 0.975$$

$$(C_{10})_O = 1.2(0.495) \left\{ \frac{480}{0.02 + 4.439 \left[\ln \left(\frac{1}{0.975} \right) \right]^{1.483}} \right\}^{\frac{1}{3}} = 6.355 \text{ kN}$$

$$(C_{10})_C = 1.2(1.322) \left\{ \frac{480}{0.02 + 4.439 \left[\ln \left(\frac{1}{0.975} \right) \right]^{1.483}} \right\}^{\frac{1}{3}} = 16.964 \text{ kN}$$

At O choose bearing with type a deep-groove 02-12 mm

At C choose bearing with type a deep-groove 02-30 mm

Example 11–4

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 static load rating C_0 is 4450 lbf and the basic load rating C_{10} is 7900 lbf. Estimate the \mathcal{L}_{10} life at a speed of 720 rev/min.

Solution

$V = 1$ and $F_a/C_0 = 400/4450 = 0.090$. Interpolate for e in Table 11–1:

F_a/C_0	e
0.084	0.28
0.090	e
0.110	0.30

from which $e = 0.285$

Example 11–4 (continued)

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

F_a/C_0	Y_2
0.084	1.55
0.090	Y_2
0.110	1.45

From Eq. (11–9),

$$F_e = X_2 V F_r + Y_2 F_a = 0.56(1)2225 + 1.527(1780) = 3964 \text{ kN}$$

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

Answer

$$\mathcal{L}_{10} = \frac{60\mathcal{L}_R n_R}{60n_D} \left(\frac{C_{10}}{F_e} \right)^a = \frac{10^6}{60(720)} \left(\frac{35\ 150}{3964} \right)^3 = 161\ 395 \text{ h}$$

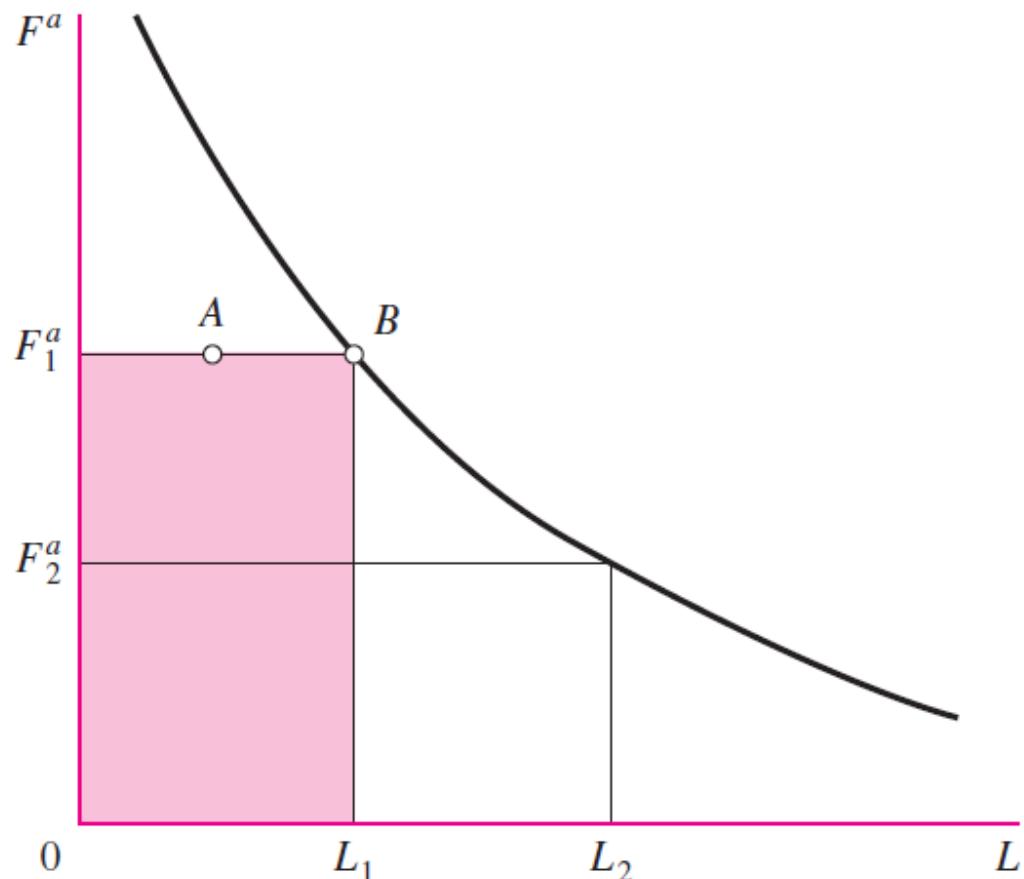
Variable Loading

$$F^a L = \text{constant} = K$$

(a)

Figure 11-9

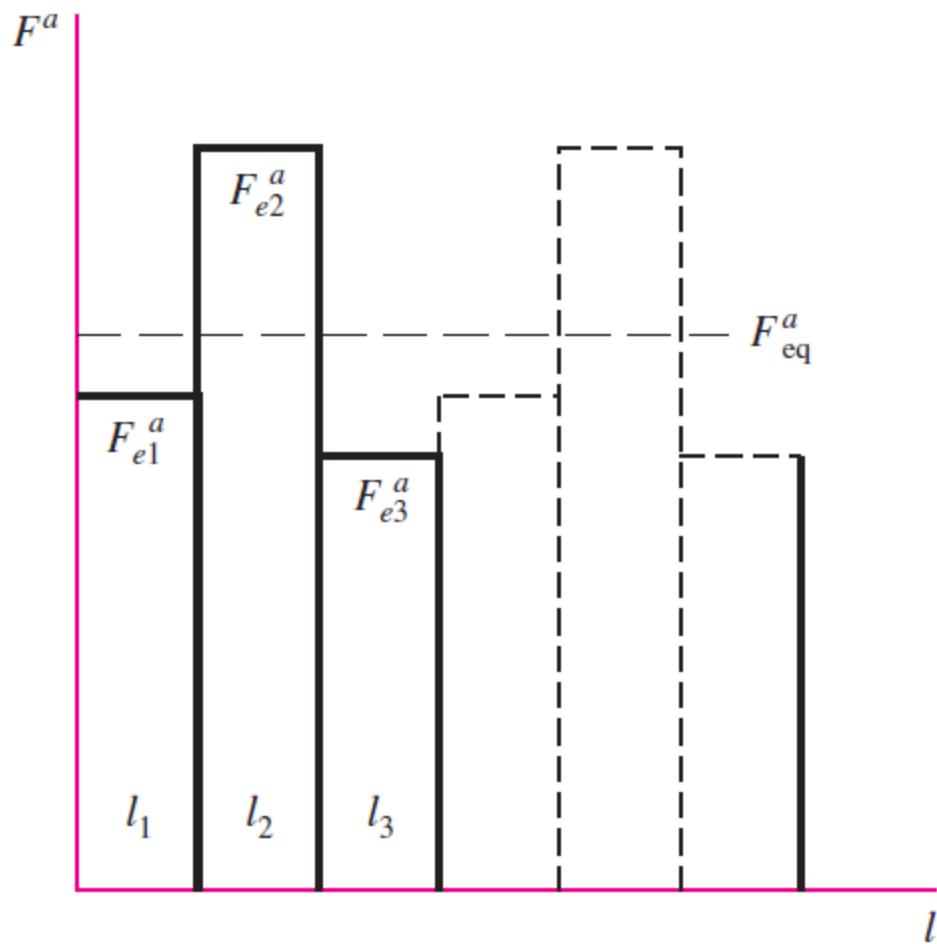
Plot of F^a as ordinate and L as abscissa for $F^a L = \text{constant}$. The linear damage hypothesis says that in the case of load F_1 , the area under the curve from $L = 0$ to $L = L_A$ is a measure of the damage $D = F_1^a L_A$. The complete damage to failure is measured by $C_{10}^a L_B$.



Variable Loading with Piecewise Constant Loading

Figure 11-10

A three-part piecewise-continuous periodic loading cycle involving loads F_{e1} , F_{e2} , and F_{e3} . F_{eq} is the equivalent steady load inflicting the same damage when run for $l_1 + l_2 + l_3$ revolutions, doing the same damage D per period.



$$D = F_{e1}^a l_1 + F_{e2}^a l_2 + F_{e3}^a l_3 \quad (b)$$

$$D = F_{eq}^a (l_1 + l_2 + l_3) \quad (c)$$

Variable Loading with Piecewise Constant Loading

$$D = F_{e1}^a l_1 + F_{e2}^a l_2 + F_{e3}^a l_3 \quad (b)$$

$$D = F_{\text{eq}}^a (l_1 + l_2 + l_3) \quad (c)$$

$$F_{\text{eq}} = \left[\frac{F_{e1}^a l_1 + F_{e2}^a l_2 + F_{e3}^a l_3}{l_1 + l_2 + l_3} \right]^{1/a} = \left[\sum f_i F_{ei}^a \right]^{1/a} \quad (11-13)$$

$$F_{\text{eq}} = \left[\frac{\sum n_i t_i F_{ei}^a}{\sum n_i t_i} \right]^{1/a} \quad (11-14)$$

$$F_{\text{eq}} = \left[\sum f_i (a_{fi} F_{ei})^a \right]^{1/a} \quad L_{\text{eq}} = \frac{K}{F_{\text{eq}}^a} \quad (11-15)$$

Example 11–5

A ball bearing is run at four piecewise continuous steady loads as shown in the following table. Columns (1), (2), and (5) to (8) to (9) are given.

(1) Time Fraction	(2) Speed, rev/min	(3) Product, Column (1) × (2)	(4) Turns Fraction, (3)/ $\sum(3)$	(5) F_{ri} , N	(6) F_{ai} , N	(7) F_{ei} , N	(8) a_{fi}	(9) F_{ei} , N
0.1	2000	200	0.077	2700	1350	3573	1.10	3930
0.1	3000	300	0.115	1350	1350	2817	1.25	3521
0.3	3000	900	0.346	3375	1350	3951	1.10	4346
0.5	2400	<u>1200</u>	<u>0.462</u>	1688	1350	3006	1.25	3758
		<u>2600</u>	<u>1.000</u>					

Columns 1 and 2 are multiplied to obtain column 3. The column 3 entry is divided by the sum of column 3, 2600, to give column 4. Columns 5, 6, and 7 are the radial, axial, and equivalent loads respectively. Column 8 is the appropriate application factor. Column 9 is the product of columns 7 and 8.

Example 11–5 (continued)

Solution

From Eq. (11–13), with $a = 3$, the equivalent radial load F_e is

$$F_e = [0.077(3930)^3 + 0.115(3521)^3 + 0.346(4346)^3 + 0.462(3758)^3]^{1/3} = 3971 \text{ N}$$

Answer

Variable Loading with Piecewise Constant Loading

$$F_{\text{eq}}^a L_{\text{eq}} = F_{e1}^a l_1 + F_{e2}^a l_2 + F_{e3}^a l_3$$

$$K = F_{e1}^a L_1 = F_{e2}^a L_2 = F_{e3}^a L_3$$

$$K = F_{e1}^a l_1 + F_{e2}^a l_2 + F_{e3}^a l_3 = \frac{K}{L_1} l_1 + \frac{K}{L_2} l_2 + \frac{K}{L_3} l_3 = K \sum \frac{l_i}{L_i}$$

$$\sum \frac{l_i}{L_i} = 1 \quad (11-16)$$

Variable Loading with Periodic Variation

$$dD = F^a d\theta$$

$$D = \int dD = \int_0^\phi F^a d\theta = F_{\text{eq}}^a \phi$$

$$F_{\text{eq}} = \left[\frac{1}{\phi} \int_0^\phi F^a d\theta \right]^{1/a} \quad L_{\text{eq}} = \frac{K}{F_{\text{eq}}^a} \quad (11-17)$$

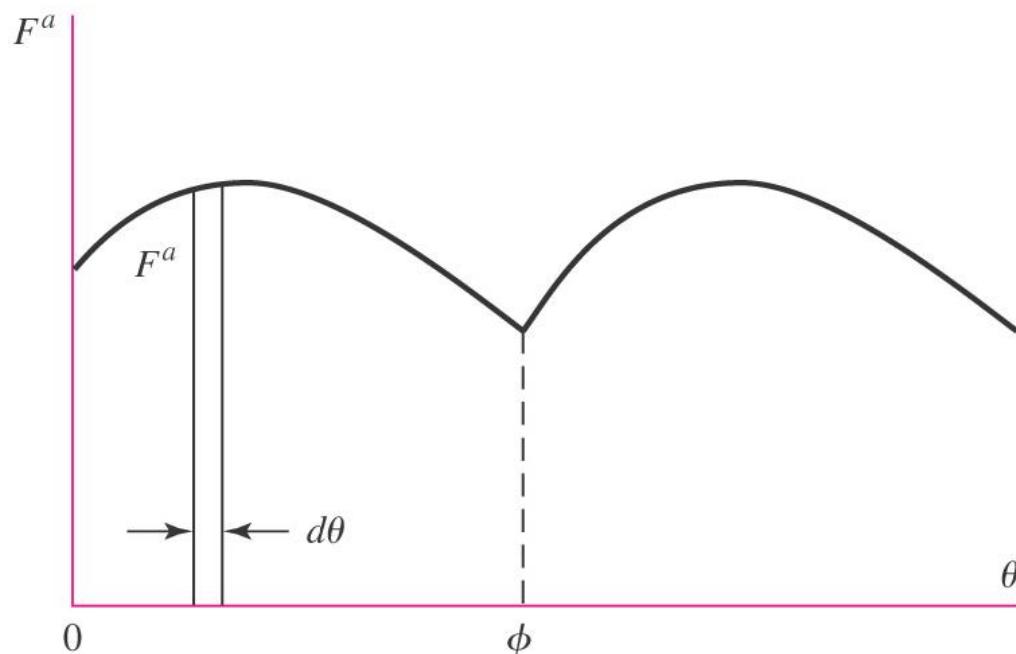


Fig. 11-11

Example 11–6

The operation of a particular rotary pump involves a power demand of $P = \bar{P} + A' \sin \theta$ where \bar{P} is the average power. The bearings feel the same variation as $F = \bar{F} + A \sin \theta$. Develop an application factor a_f for this application of ball bearings.

Solution

From Eq. (11–17), with $a = 3$,

$$\begin{aligned} F_{\text{eq}} &= \left(\frac{1}{2\pi} \int_0^{2\pi} F^a d\theta \right)^{1/a} = \left(\frac{1}{2\pi} \int_0^{2\pi} (\bar{F} + A \sin \theta)^3 d\theta \right)^{1/3} \\ &= \left[\frac{1}{2\pi} \left(\int_0^{2\pi} \bar{F}^3 d\theta + 3\bar{F}^2 A \int_0^{2\pi} \sin \theta d\theta + 3\bar{F}A^2 \int_0^{2\pi} \sin^2 \theta d\theta \right. \right. \\ &\quad \left. \left. + A^3 \int_0^{2\pi} \sin^3 \theta d\theta \right) \right]^{1/3} \\ F_{\text{eq}} &= \left[\frac{1}{2\pi} (2\pi\bar{F}^3 + 0 + 3\pi\bar{F}A^2 + 0) \right]^{1/3} = \bar{F} \left[1 + \frac{3}{2} \left(\frac{A}{\bar{F}} \right)^2 \right]^{1/3} \end{aligned}$$

Example 11–6 (continued)

In terms of \bar{F} , the application factor is

Answer $a_f = \left[1 + \frac{3}{2} \left(\frac{A}{\bar{F}} \right)^2 \right]^{1/3}$

We can present the result in tabular form:

A/\bar{F}	a_f
0	1
0.2	1.02
0.4	1.07
0.6	1.15
0.8	1.25
1.0	1.36

Example 11–7

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.

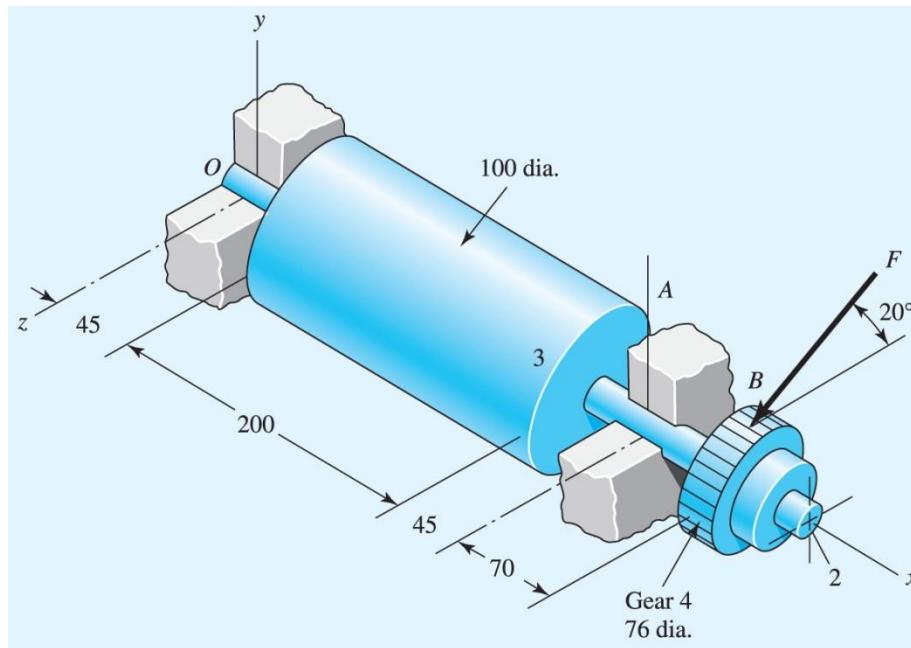
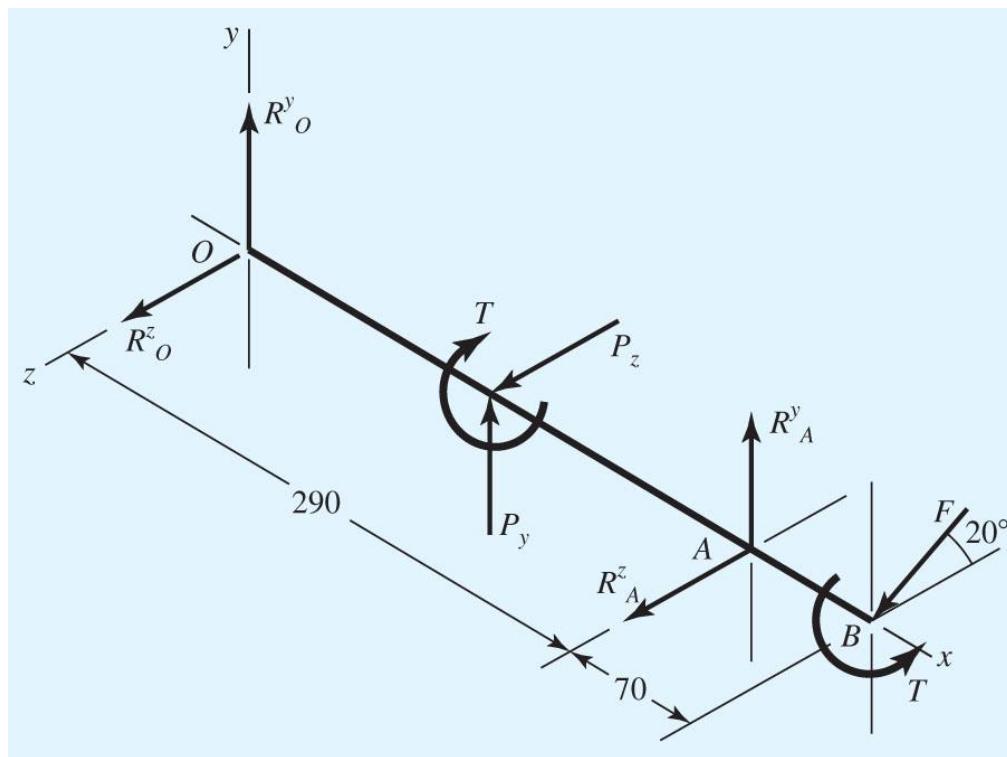


Fig. 11–12

Example 11–7



Example 11–7

Solution

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

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

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

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

Example 11–7

thus

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

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

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

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

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

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

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

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

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

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

Example 11–7 (continued)

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.

Tapered Roller Bearings

- Straight roller bearings can carry large radial loads, but no axial load.
- Ball bearings can carry moderate radial loads, and small axial loads.
- Tapered roller bearings rely on roller tipped at an angle to allow them to carry large radial and large axial loads.
- Tapered roller bearings were popularized by the Timken Company.

Tapered Roller Bearings

- Two separable parts
 - Cone assembly
 - Cone (inner ring)
 - Rollers
 - Cage
 - Cup (outer ring)
- Rollers are tapered so virtual apex is on shaft centerline
- Taper allows for pure rolling of angled rollers
- Distance a locates the effective axial location for force analysis

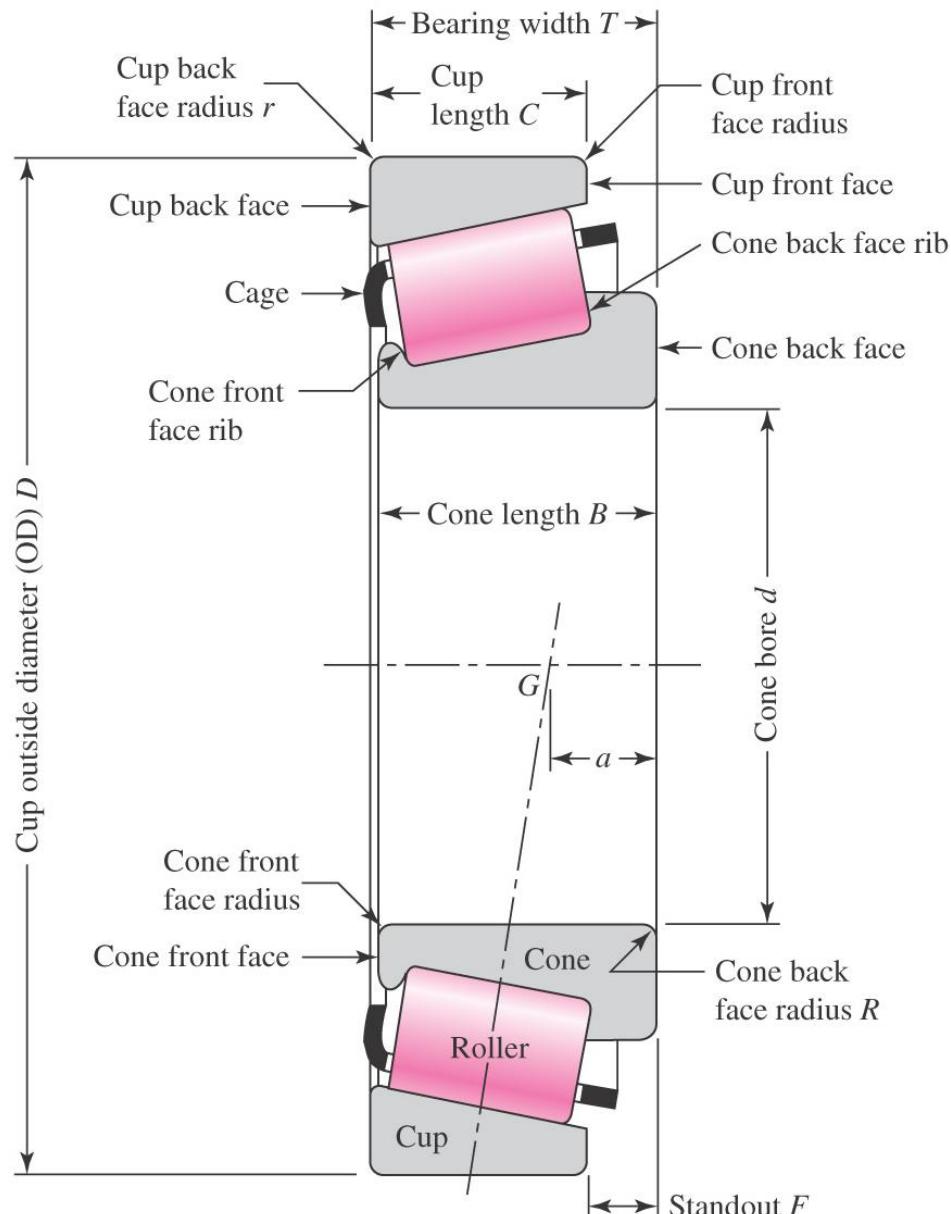


Fig. 11–13

Mounting Directions of Tapered Roller Bearings

- Mount pairs in opposite directions to counter the axial loads
- Can be mounted in *direct mounting* or *indirect mounting* configurations
- For the same effective spread a_e , direct mounting requires greater geometric spread a_g
- For the same geometric spread a_g , direct mounting provides smaller effect spread a_e

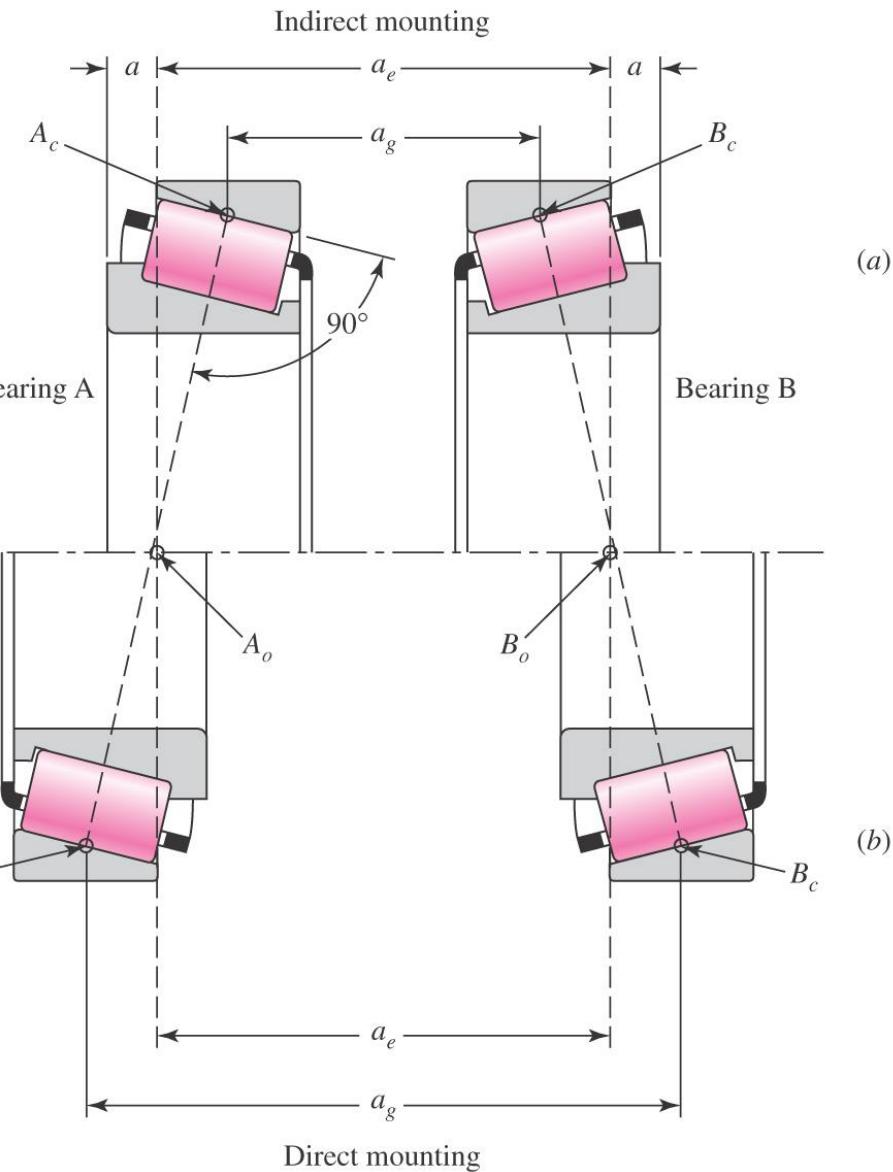
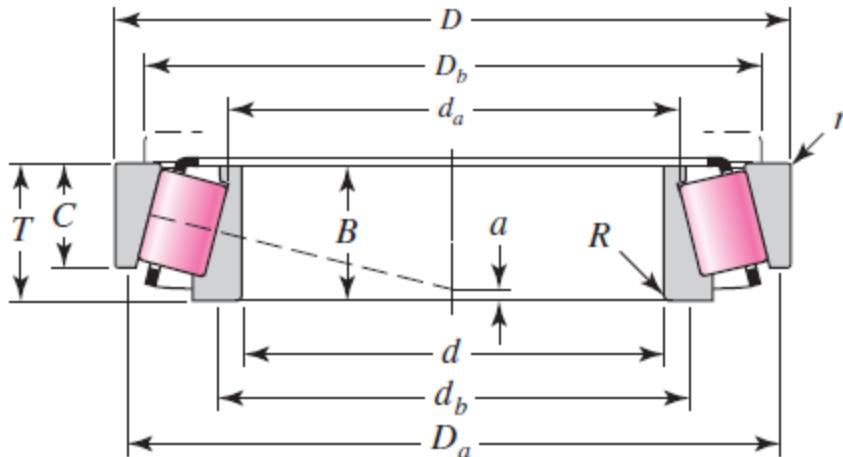


Fig. 11-14

Typical Catalog Data (Fig. 11–15)

SINGLE-ROW STRAIGHT BORE



Typical Catalog Data (Fig. 11–15)

bore d	outside diameter D	width T	rating at 500 rpm for 3000 hours L ₁₀		fac- tor K	eff. load center a②	part numbers		max shaft fillet radius R^①	width B	backing shoulder diameters		max hos- ting fillet radius r^①	width C	cup	
			one- row radial N lbf	thrust N lbf			cone	cup			width	width	backing shoulder diameters			
											d_b	d_a			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
25.159 0.9905	50.005 1.9687	13.495 0.5313	6990 1570	4810 1080	1.45	-2.8 -0.11	07096	07196	1.5 0.06	14.260 0.5614	31.5 1.24	29.5 1.16	1.0 0.04	9.525 0.3750	44.5 1.75	47.0 1.85
25.400 1.0000	50.005 1.9687	13.495 0.5313	6990 1570	4810 1080	1.45	-2.8 -0.11	07100	07196	1.0 0.04	14.260 0.5614	30.5 1.20	29.5 1.16	1.0 0.04	9.525 0.3750	44.5 1.75	47.0 1.85
25.400 1.0000	50.005 1.9687	13.495 0.5313	6990 1570	4810 1080	1.45	-2.8 -0.11	07100-S	07196	1.5 0.06	14.260 0.5614	31.5 1.24	29.5 1.16	1.0 0.04	9.525 0.3750	44.5 1.75	47.0 1.85
25.400 1.0000	50.292 1.9800	14.224 0.5600	7210 1620	4620 1040	1.56	-3.3 -0.13	L44642	L44610	3.5 0.14	14.732 0.5800	36.0 1.42	29.5 1.16	1.3 0.05	10.668 0.4200	44.5 1.75	47.0 1.85
25.400 1.0000	50.292 1.9800	14.224 0.5600	7210 1620	4620 1040	1.56	-3.3 -0.13	L44643	L44610	1.3 0.05	14.732 0.5800	31.5 1.24	29.5 1.16	1.3 0.05	10.668 0.4200	44.5 1.75	47.0 1.85
25.400 1.0000	51.994 2.0470	15.011 0.5910	6990 1570	4810 1080	1.45	-2.8 -0.11	07100	07204	1.0 0.04	14.260 0.5614	30.5 1.20	29.5 1.16	1.3 0.05	10.668 0.4200	44.5 1.75	47.0 1.85
25.400 1.0000	56.896 2.2400	19.368 0.7625	10900 2450	5740 1290	1.90	-6.9 -0.27	1780	1729	0.8 0.03	19.837 0.7810	30.5 1.20	30.0 1.18	1.3 0.05	15.875 0.6250	49.0 1.93	51.0 2.01

Typical Catalog Data (Fig. 11–15 continued)

bore d	outside diameter D	width T	rating at 500 rpm for 3000 hours L ₁₀		fac- tor K	eff. load center a^②	part numbers		max shaft fillet radius R^①	width B	backing shoulder diameters		max hos- ting fillet radius r^①	cup		backing shoulder diameters D_b		
			one- row radial N lbf	thrust N lbf			cone	cup			d_b	d_a		C	D_b	D_a		
d	D	T	N lbf	N lbf														
25.400 1.0000	62.000 2.4409	19.050 0.7500	12100 2730	7280 1640	1.67	-5.8 -0.23	15102	15245	1.5 0.06	20.638 0.8125	34.0 1.34	31.5 1.24	1.3 0.05	14.288 0.5625	55.0 2.17	58.0 2.28		
25.400 1.0000	62.000 2.4409	20.638 0.8125	12100 2730	7280 1640	1.67	-5.8 -0.23	15101	15244	0.8 0.03	20.638 0.8125	32.5 1.28	31.5 1.24	1.3 0.05	15.875 0.6250	55.0 2.17	58.0 2.28		
25.400 1.0000	63.500 2.5000	20.638 0.8125	12100 2730	7280 1640	1.67	-5.8 -0.23	15101	15250	0.8 0.03	20.638 0.8125	32.5 1.28	31.5 1.24	1.3 0.05	15.875 0.6250	56.0 2.20	59.0 2.32		
25.400 1.0000	63.500 2.5000	20.638 0.8125	12100 2730	7280 1640	1.67	-5.8 -0.23	15101	15250	0.8 0.03	20.638 0.8125	32.5 1.28	31.5 1.24	1.5 0.06	15.875 0.6250	55.0 2.17	59.0 2.32		
25.400 1.0000	64.292 2.5312	21.433 0.8438	14500 3250	13500 3040	1.07	-3.3 -0.13	M86643	M86610	1.5 0.06	21.433 0.8438	38.0 1.50	36.5 1.44	1.5 0.06	16.670 0.6563	54.0 2.13	61.0 2.40		
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		

Typical Catalog Data (Fig. 11–15 continued)

- ① These maximum fillet radii will be cleared by the bearing corners.
- ② Minus value indicates center is inside cone backface.
- † For standard class **ONLY**, the maximum metric size is a whole mm value.
- ★ For "J" part tolerances—see metric tolerances, page 73, and fitting practice, page 65.
- ◆ ISO cone and cup combinations are designated with a common part number and should be purchased as an assembly.
For ISO bearing tolerances—see metric tolerances, page 73, and fitting practice, page 65.

Induced Thrust Load

- A radial load induces a thrust reaction due to the roller angle.

$$F_i = \frac{0.47F_r}{K} \quad (11-18)$$

- K is ratio of radial load rating to thrust load rating
- K is dependent on specific bearing, and is tabulated in catalog

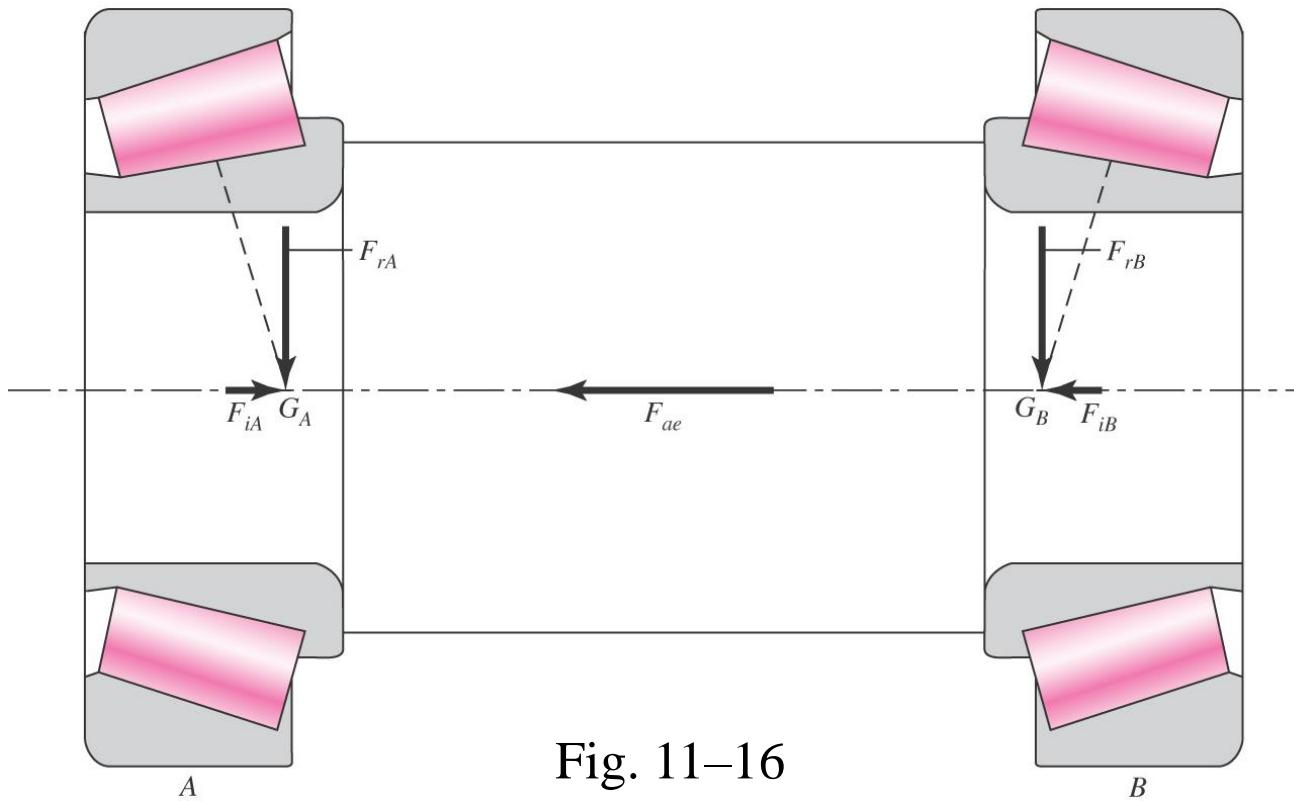


Fig. 11-16

Equivalent Radial Load

- The equivalent radial load for tapered roller bearings is found in similar form as before,

$$F_e = XVF_r + YF_a$$

- Timken recommends $X = 0.4$ and $Y = K$

$$F_e = 0.4F_r + KF_a$$

- F_a is the net axial load carried by the bearing, including induced thrust load from the other bearing and the external axial load carried by the bearing.
- Only one of the bearings will carry the external axial load

Determining Which Bearing Carries External Axial Load

- Regardless of mounting direction or shaft orientation, visually inspect to determine which bearing is being “squeezed”
- Label this bearing as *Bearing A*

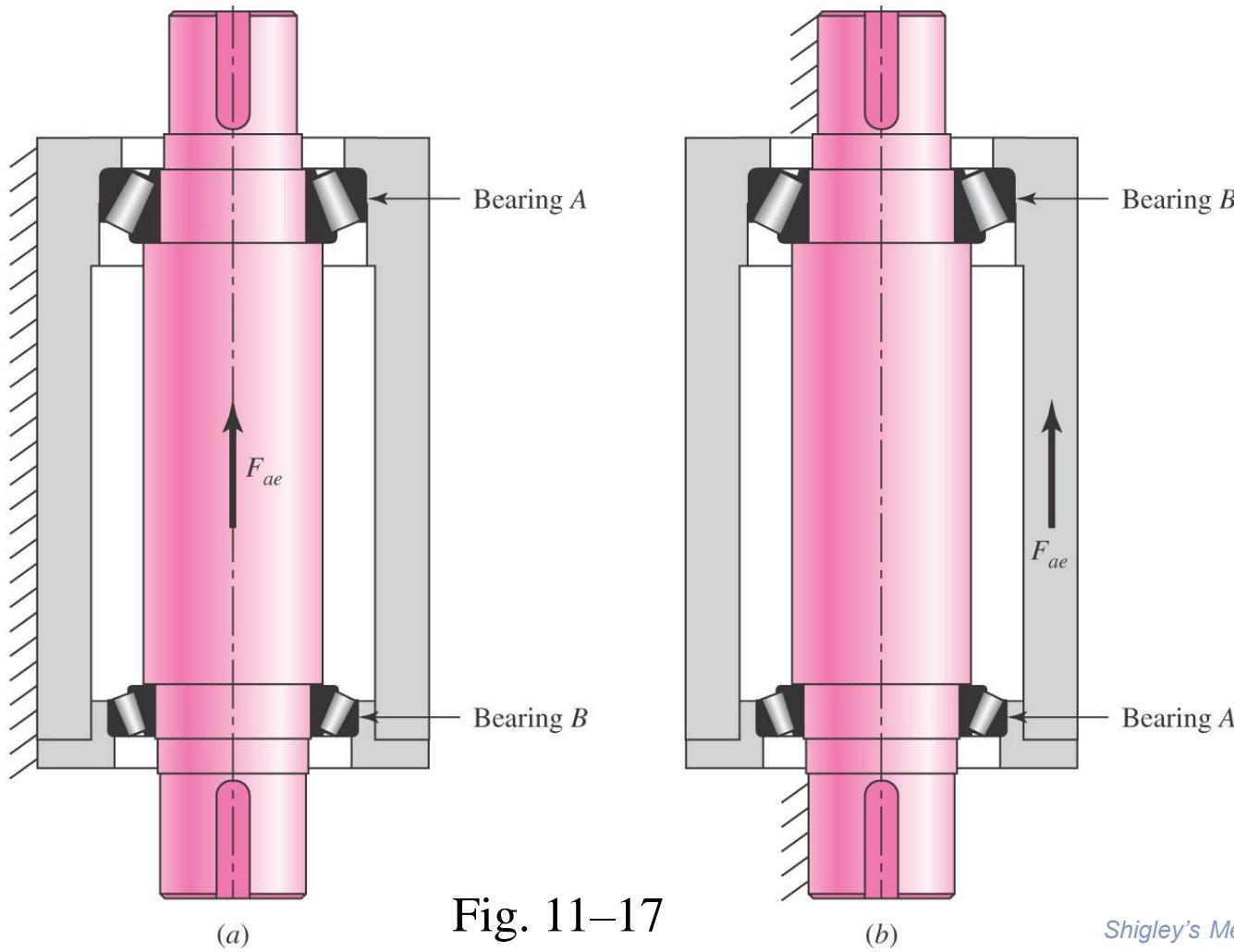


Fig. 11–17

Net Axial Load

- Generally, *Bearing A* (the squeezed bearing) carries the net axial load
- Occasionally the induced thrust from *Bearing A*, F_{iA} , is greater than the combination of the induced thrust from *Bearing B*, F_{iB} , and the external axial load F_{ae} , that is

$$F_{iA} > (F_{iB} + F_{ae})$$

- If this happens, then *Bearing B* actually carries the net axial load

Equivalent Radial Load

- Timken recommends using the full radial load for the bearing that is not carrying the net axial load.
- Equivalent radial load equation:

If $F_{iA} \leq (F_{iB} + F_{ae})$ $\begin{cases} F_{eA} = 0.4F_{rA} + K_A(F_{iB} + F_{ae}) \\ F_{eB} = F_{rB} \end{cases}$ (11-19a)

(11-19b)

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

(11-20b)

- If the equivalent radial load is less than the original radial load, then use the original radial load.

Example 11–8

The shaft depicted in Fig. 11–18a 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.

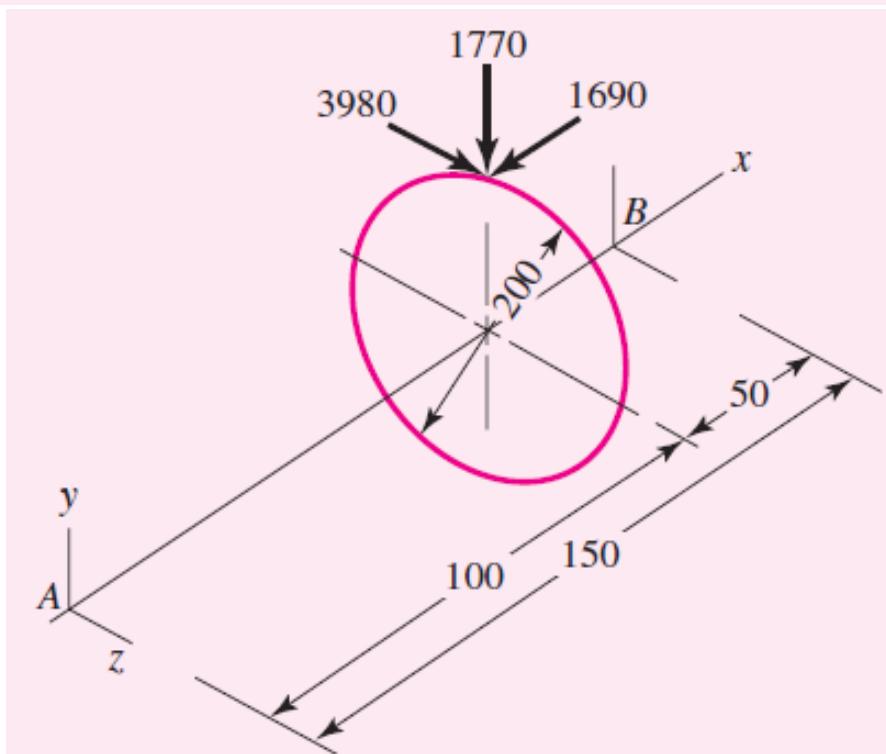


Fig. 11–18 (a)

Example 11–8 (continued)

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

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

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

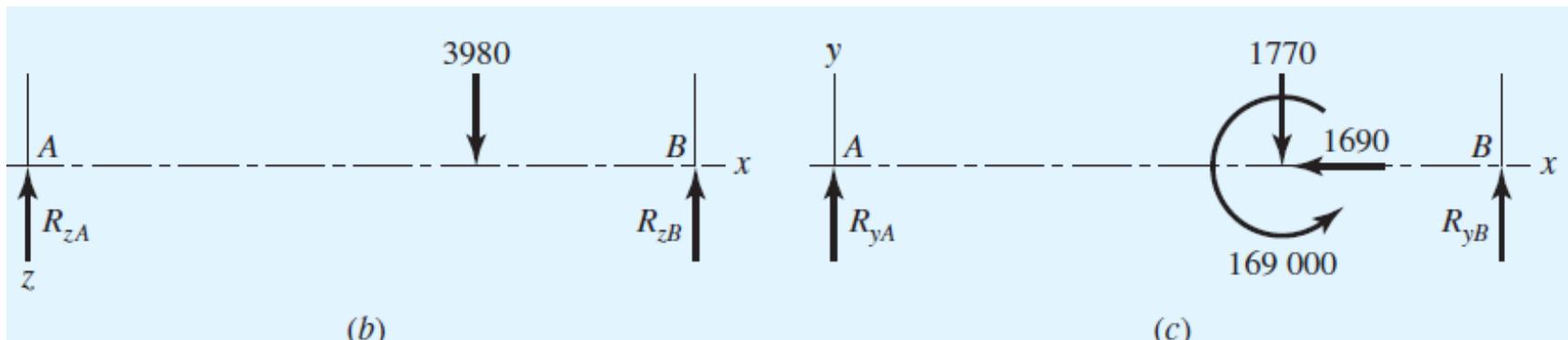


Fig. 11-18

Example 11–8 (continued)

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

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

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

Example 11–8 (continued)

Since F_{iA} is clearly less than $F_{iB} + F_{ae}$, bearing A carries the net thrust load, and Eq. (11–19) 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. (11–10) 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: $K_A = 1.67$, $C_{10} = 12\,100 \text{ N}$.

Example 11–8 (continued)

For bearing B , from Eq. (11–10), 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: $K_B = 1.67$, $C_{10} = 12\,100 \text{ N}$.

Example 11–8 (continued)

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. (11–19) 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. (11–10) the corrected catalog entry C_{10} should equal or exceed

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

Example 11–8 (continued)

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. (11–10),

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

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. (11–10) 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.

Realized Bearing Reliability

- Eq. (11–9) was previously derived to determine a suitable catalog rated load for a given design situation and reliability goal.

$$C_{10} = a_f F_D \left[\frac{x_D}{x_0 + (\theta - x_0) [\ln(1/R_D)]^{1/b}} \right]^{1/a} \quad (11-9)$$

- An actual bearing is selected from a catalog with a rating greater than C_{10} .
- Sometimes it is desirable to determine the realized reliability from the actual bearing (that was slightly higher capacity than needed).
- Solving Eq. (11–9) for the reliability,

$$R = \exp \left(- \left\{ \frac{x_D \left(\frac{a_f F_D}{C_{10}} \right)^a - x_0}{\theta - x_0} \right\}^b \right) \quad (11-21)$$

Realized Bearing Reliability

- Similarly for the alternate approximate equation, Eq. (11-10),

$$C_{10} \approx a_f F_D \left[\frac{x_D}{x_0 + (\theta - x_0)(1 - R_D)^{1/b}} \right]^{1/a} \quad R \geq 0.90 \quad (11-10)$$

$$R \approx 1 - \left\{ \frac{x_D \left(\frac{a_f F_D}{C_{10}} \right)^a - x_0}{\theta - x_0} \right\}^b \quad R \geq 0.90 \quad (11-22)$$

Example 11–9

In Ex. 11–3, the minimum required load rating for 99 percent reliability, at $x_D = L_D/L_{10} = 540$, is $C_{10} = 6696 \text{ lbf} = 29.8 \text{ kN}$. From Table 11–2 a 02-40 mm deep-groove ball bearing would satisfy the requirement. If the bore in the application had to be 70 mm or larger (selecting a 02-70 mm deep-groove ball bearing), what is the resulting reliability?

Solution

From Table 11–2, for a 02-70 mm deep-groove ball bearing, $C_{10} = 61.8 \text{ kN} = 13\,888 \text{ lbf}$. Using Eq. (11–22), recalling from Ex. 11–3 that $a_f = 1.2$, $F_D = 413 \text{ lbf}$, $x_0 = 0.02$, $(\theta - x_0) = 4.439$, and $b = 1.483$, we can write

Answer

$$R \approx 1 - \left\{ \frac{\left[540 \left[\frac{1.2(413)}{13\,888} \right]^3 - 0.02 \right]}{4.439} \right\}^{1.483} = 0.999\,963$$

which, as expected, is much higher than 0.99 from Ex. 11–3.

Realized Reliability for Tapered Roller Bearings

- Substituting typical Weibull parameters for tapered roller bearings into Eqs. (11–21) and (11–22) give realized reliability equations customized for tapered roller bearings.

$$\begin{aligned} R &= \exp \left\{ - \left[\frac{x_D}{\theta [C_{10}/(a_f F_D)]^a} \right]^b \right\} \\ &= \exp \left\{ - \left[\frac{x_D}{4.48 [C_{10}/(a_f F_D)]^{10/3}} \right]^{3/2} \right\} \end{aligned} \quad (11-23)$$

$$R \approx 1 - \left\{ \frac{x_D}{\theta [C_{10}/(a_f F_D)]^a} \right\}^b = 1 - \left\{ \frac{x_D}{4.48 [C_{10}/(a_f F_D)]^{10/3}} \right\}^{3/2} \quad (11-24)$$

Example 11–10

In Ex. 11–8 bearings A and B (cone 15100 and cup 15245) have $C_{10} = 12\ 100\ N$. What is the reliability of the pair of bearings A and B ?

Solution

The desired life x_D was $5000(800)60/[90(10^6)] = 2.67$ rating lives. Using Eq. (11–24) for bearing A , where from Ex. 11–8, $F_D = F_{eA} = 4938\ N$, and $a_f = 1$, gives

$$R_A \approx 1 - \left\{ \frac{2.67}{4.48[12\ 100/(1 \times 4938)]^{10/3}} \right\}^{3/2} = 0.994\ 791$$

which is less than 0.995, as expected. Using Eq. (11–24) for bearing B with $F_D = F_{eB} = 2654\ N$ gives

$$R_B \approx 1 - \left\{ \frac{2.67}{4.48[12\ 100/(1 \times 2654)]^{10/3}} \right\}^{3/2} = 0.999\ 766$$

Example 11–10 (continued)

The reliability of the bearing pair is

Answer

$$R = R_A R_B = 0.994\ 791(0.999\ 766) = 0.994\ 558$$

which is greater than the overall reliability goal of 0.99. When two bearings are made identical for simplicity, or reducing the number of spares, or other stipulation, and the loading is not the same, both can be made smaller and still meet a reliability goal. If the loading is disparate, then the more heavily loaded bearing can be chosen for a reliability goal just slightly larger than the overall goal.

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 .

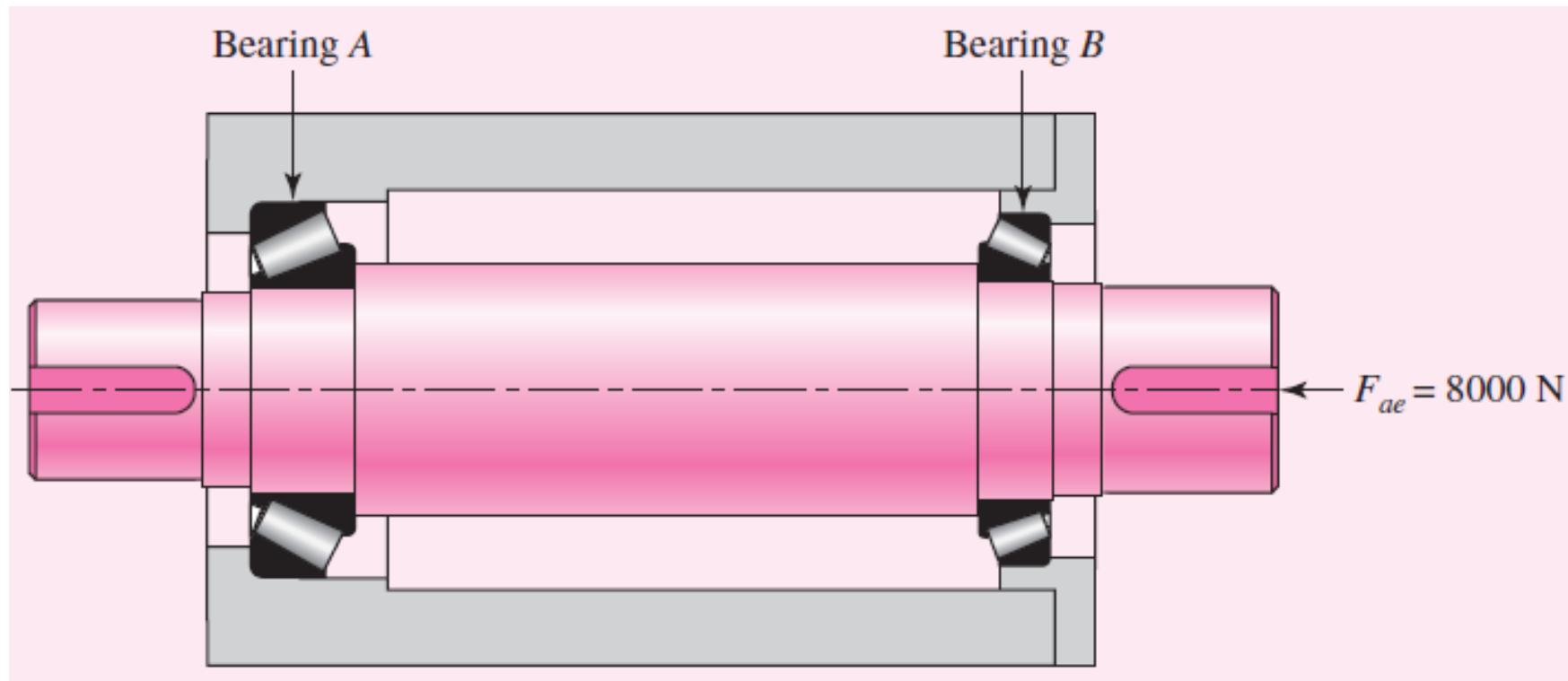


Fig. 11–19

Example 11–11 (continued)

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. (11–19) 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:

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

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

Example 11–11 (continued)

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. (11–10), for bearing *A*

$$\begin{aligned} C_{10} &= a_f F_{eA} \left[\frac{x_D}{4.48(1 - R_D)^{2/3}} \right]^{3/10} \\ &= (1)8000 \left[\frac{6.33}{4.48(1 - 0.95)^{2/3}} \right]^{3/10} = 16\,159 \text{ N} \end{aligned}$$

Answer

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

Example 11–11 (continued)

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

Answer

(c) The actual reliability of bearing A , from Eq. (11–24), is

$$\begin{aligned} R_A &\approx 1 - \left\{ \frac{x_D}{4.48[C_{10}/(a_f F_D)]^{10/3}} \right\}^{3/2} \\ &\approx 1 - \left\{ \frac{6.333}{4.48[17\,200/(1 \times 8000)]^{10/3}} \right\}^{3/2} = 0.963 \end{aligned}$$

Answer

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

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

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

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

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

Answer

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

Bearing Lubrication

- The purposes of bearing lubrication
 - To provide a film of lubricant between the sliding and rolling surfaces
 - To help distribute and dissipate heat
 - To prevent corrosion of the bearing surfaces
 - To protect the parts from the entrance of foreign matter

Bearing Lubrication

- Either oil or grease may be used, with each having advantages in certain situations.

Use Grease When	Use Oil When
<ol style="list-style-type: none">1. The temperature is not over 200°F.2. The speed is low.3. Unusual protection is required from the entrance of foreign matter.4. Simple bearing enclosures are desired.5. Operation for long periods without attention is desired.	<ol style="list-style-type: none">1. Speeds are high.2. Temperatures are high.3. Oiltight seals are readily employed.4. Bearing type is not suitable for grease lubrication.5. The bearing is lubricated from a central supply which is also used for other machine parts.

Some Common Bearing Mounting Configurations

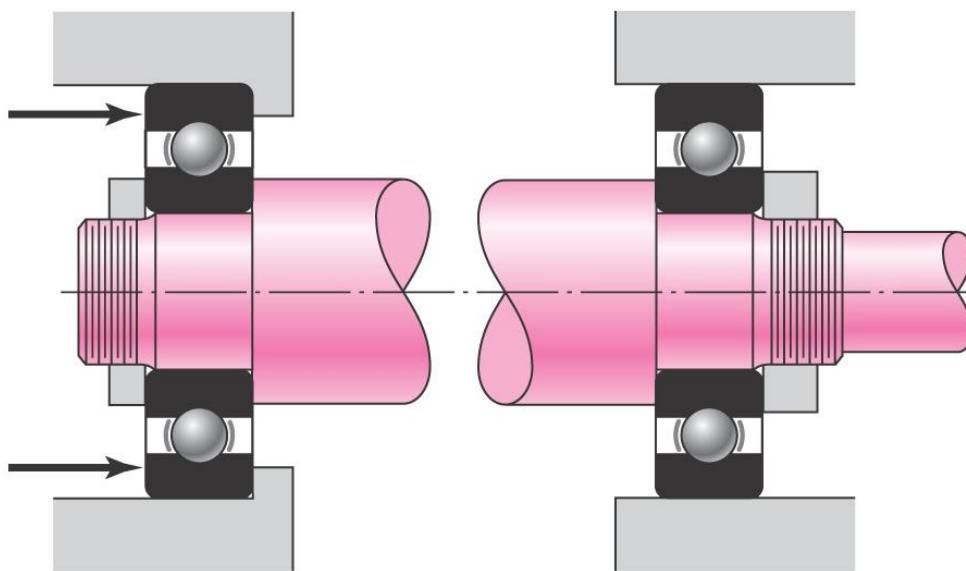


Fig. 11-20

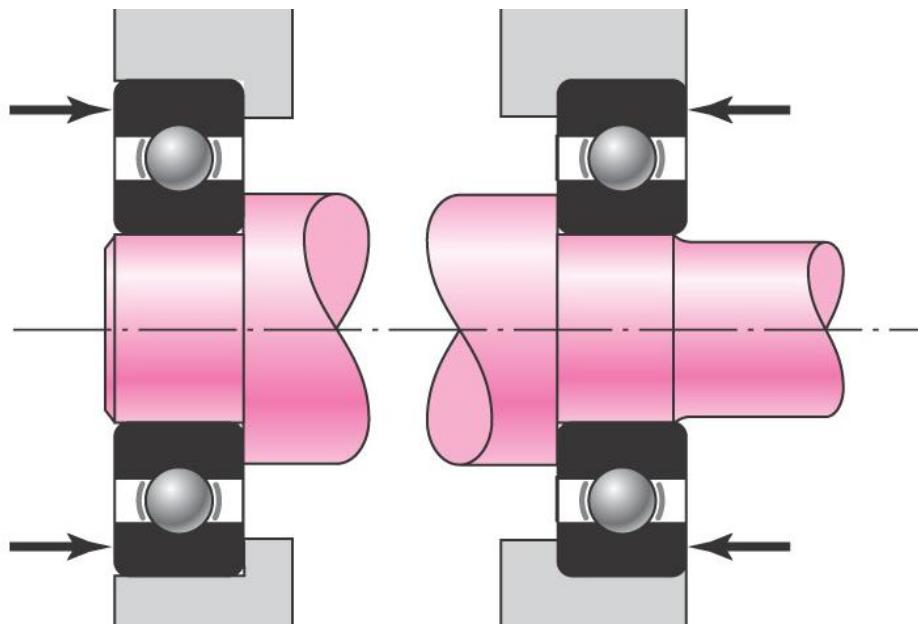
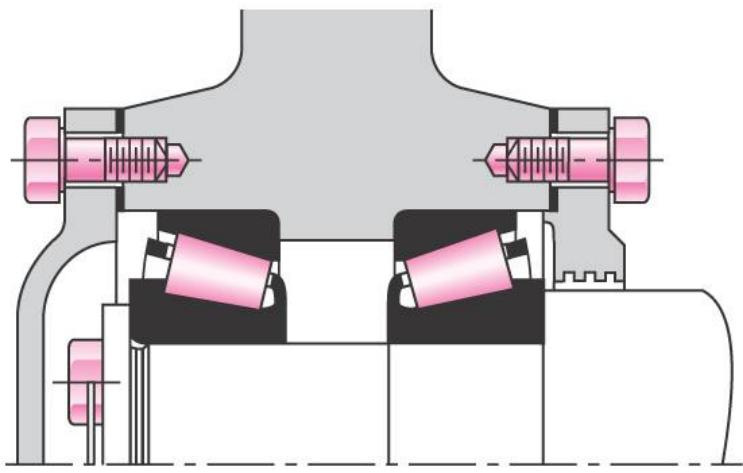
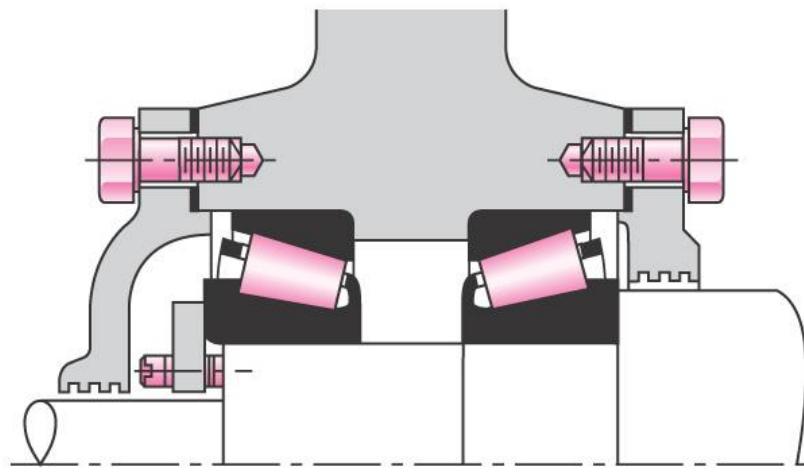


Fig. 11-21

Some Common Bearing Mounting Configurations



(a)



(b)

Fig. 11-22

Some Common Bearing Mounting Configurations

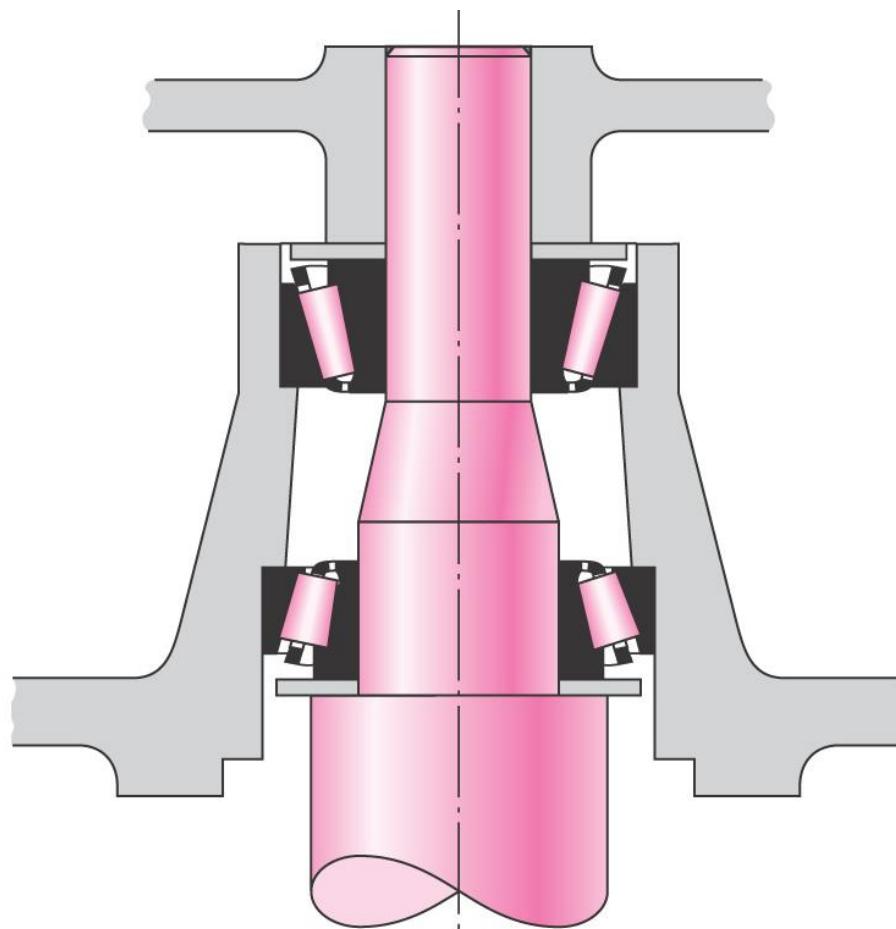
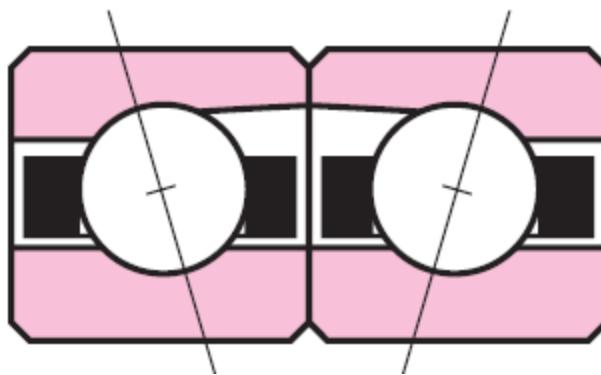


Fig. 11-23

Duplexing

- When maximum stiffness and resistance to shaft misalignment is desired, pairs of angular-contact bearings can be used in an arrangement called *duplexing*.
- Duplex bearings have rings ground with an offset.
- When pairs are clamped together, a preload is established.



Duplexing Arrangements

- Three common duplexing arrangements:
 - (a) DF mounting – Face to face, good for radial and thrust loads from either direction
 - (b) DB mounting – Back to back, same as DF, but with greater alignment stiffness
 - (c) DT mounting – Tandem, good for thrust only in one direction

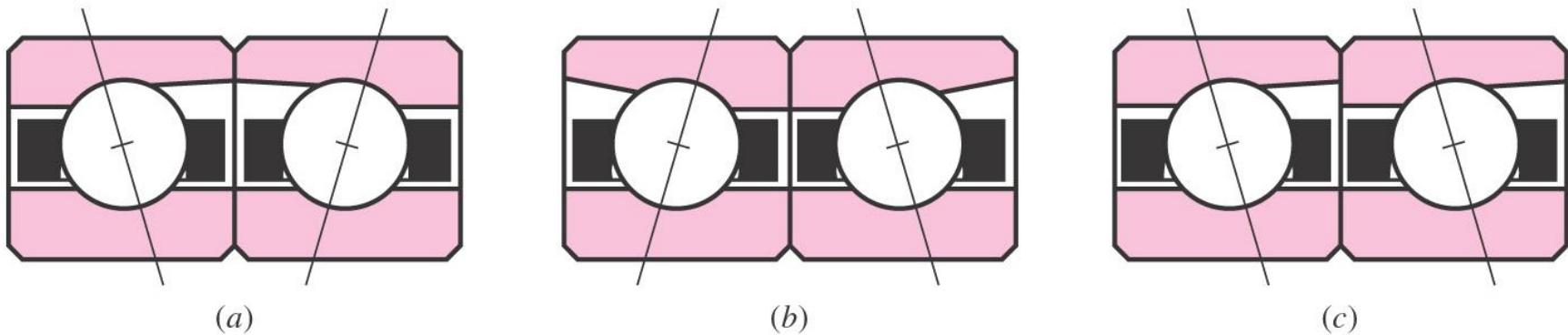


Fig. 11–24

Preferred Fits

- Rotating ring usually requires a press fit
- Stationary ring usually best with a push fit
- Allows stationary ring to creep, bringing new portions into the load-bearing zone to equalize wear

Preloading

- Object of preloading
 - Remove internal clearance
 - Increase fatigue life
 - Decrease shaft slope at bearing

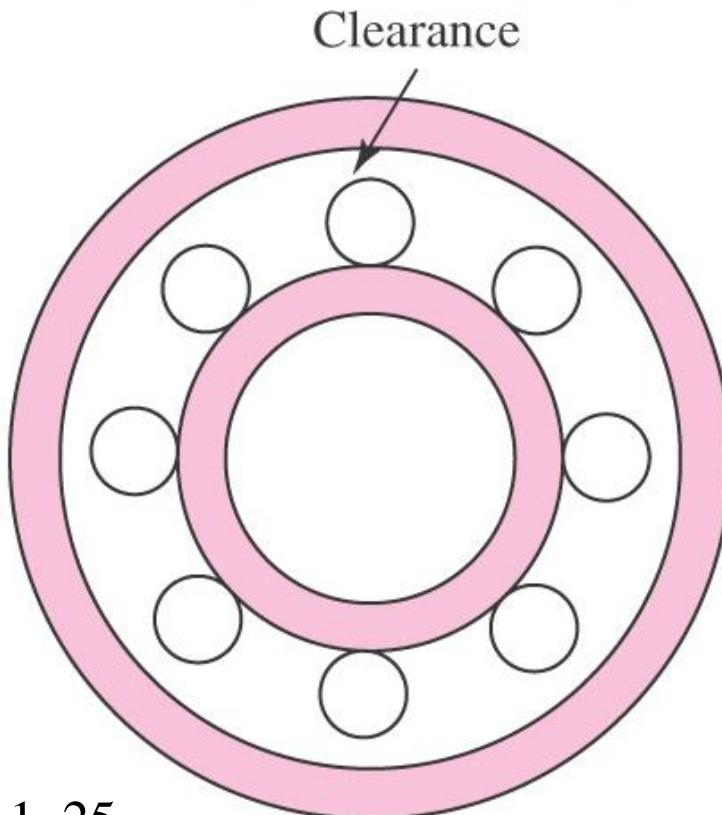


Fig. 11–25

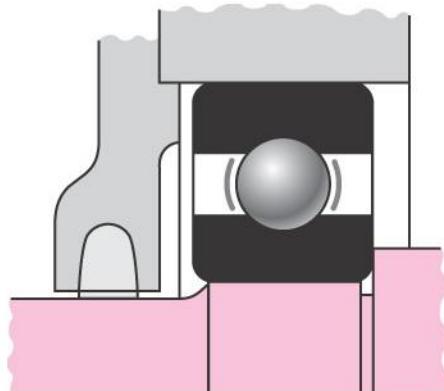
Alignment

- Catalogs will specify alignment requirements for specific bearings
- Typical maximum ranges for shaft slopes at bearing locations

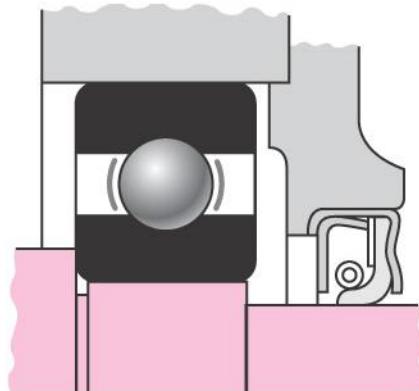
Tapered roller	0.0005–0.0012 rad
Cylindrical roller	0.0008–0.0012 rad
Deep-groove ball	0.001–0.003 rad
Spherical ball	0.026–0.052 rad
Self-align ball	0.026–0.052 rad

Enclosures

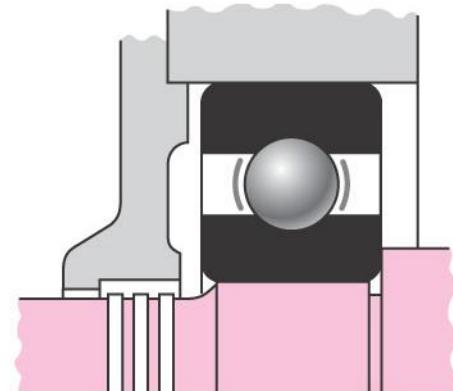
- Common shaft seals to exclude dirt and retain lubricant



(a) Felt seal



(b) Commercial seal



(c) Labyrinth seal

Fig. 11–26



THANK YOU