Ch# 11

Rolling Contact Bearings

- The terms rolling-contact bearings, antifriction bearings, and rolling bearings are all used to describe the class of bearing in which the main load is transferred through elements in rolling contact rather than in sliding contact
- In a rolling bearing the starting friction is about twice the running friction, but still it is negligible in comparison with the starting friction of a sleeve bearing
- Load, speed, and the operating viscosity of the lubricant do affect the frictional characteristics of a rolling bearing
- It is probably a mistake to describe a rolling bearing as "antifriction" but the term is generally used throughout the industry

Rolling Contact Bearings

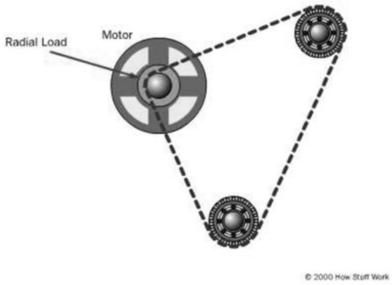
ME-305 (Machine Design II)

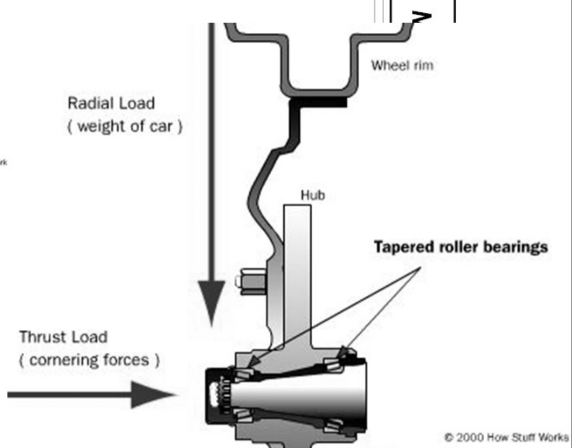
- Bearing specialist consider matters such as
  - Fatigue loading
  - Friction
  - Heat
  - Corrosion resistance
  - Kinematic problems
  - Material properties
  - Lubrication
  - Machining tolerances
  - Assembly and cost

Rolling Contact Bearings

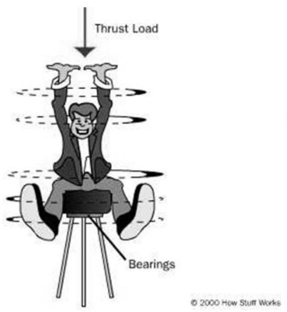
A device to permit constrained relative motion between two parts.

The bearings that support the shafts of motors and pulleys are subject to a radial load





The bearings in a car wheel are subject to both thrust and radial loads.



The bearings in this stool are subject to a thrust load

Roller Thrust Bearing

Ball Thrust Bearing

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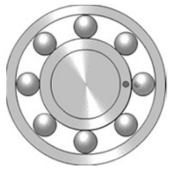
11-1 Bearing Types

- Anti-friction bearings are manufactured to take
  1. Pure radial load
  2. Pure thrust loads, or
  3. A combination of the two

Sliding contact , rolling contact → Ball bearings ; ↘ Roller bearings



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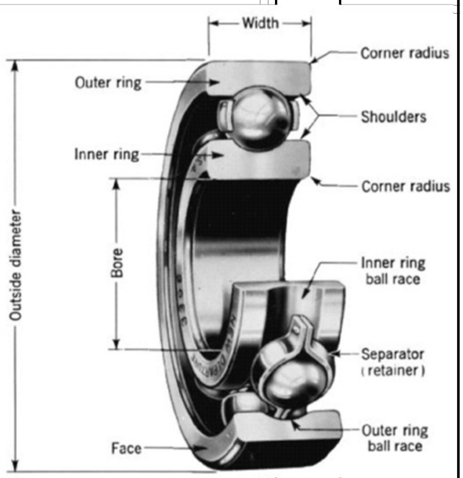
11-1 Bearing Types

Nomenclature

Single row (Ball)

- Four Essential Parts of Ball Bearings
  1. Outer Ring
  2. The Inner Ring
  3. Ball or Rolling Element
  4. Separator

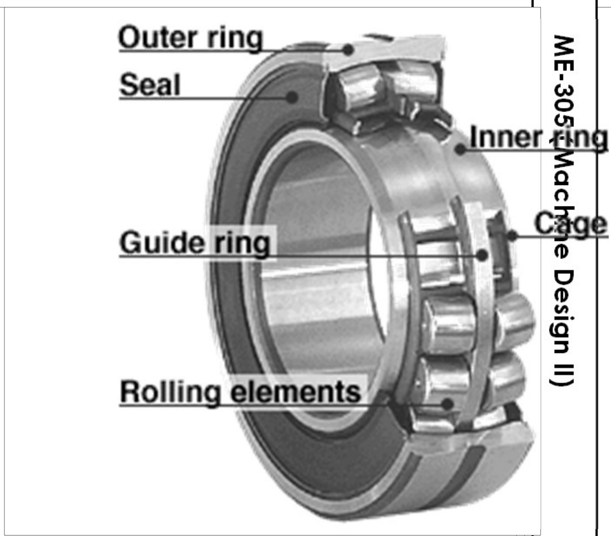


Low cost bearings have no separator, It has very important Function (rubbing contact will not occur)

11-1 Bearing Types

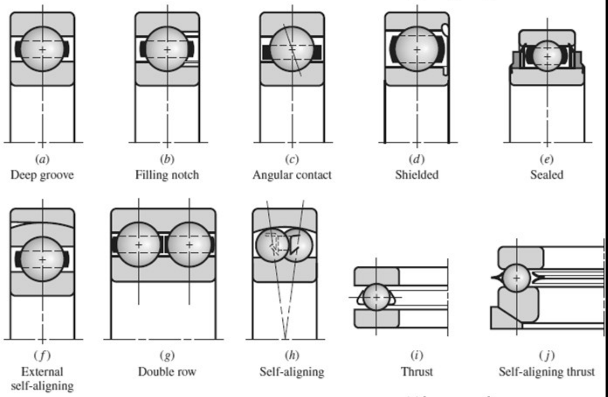
Nomenclature

Double row (Roller)



11-1 Bearing Types

1. Deep groove
2. Filling notch
3. Angular contact
4. Shielded
5. Sealed
6. Self-aligning
7. Thrust
8. Self-aligning thrust
9. Double row



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Conrad or Deep Groove

The balls rotate in a deep groove machined into the inner and outer races. A cage maintains Ball space. This type of bearing exhibits a good radial load capacity, a fair axial load capacity and a fair capability to resist misalignment.

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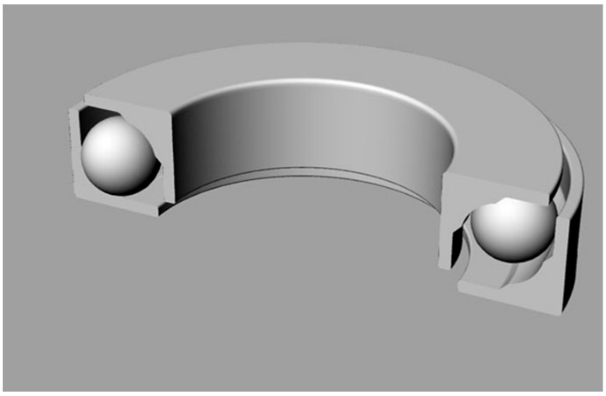
Maximum Capacity (filling Notch)

Maximum capacity bearings have a higher load capacity than the Conrad type but diminished thrust capacity. These bearings have no cage to separate the balls.
A filling slot is used to load as many balls as possible.

Angular contact Bearings



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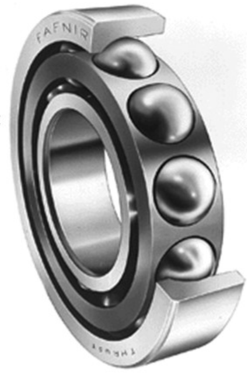
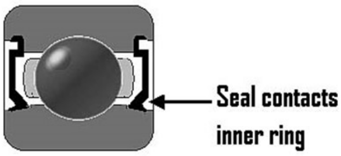
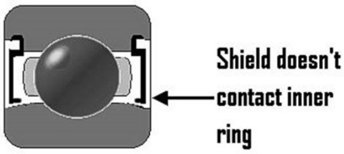
Sealed Bearings



Sealed Angular





Seals and shields provide protection from contamination and as a retainer for lubricant. Seals provide better protection and lubricant containment than shields, but have lower maximum speed capabilities.

Open bearings offer no protection from contamination or lubricant containment, but have higher maximum speed capabilities than bearings with seals or shields.

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Self aligning

Double Row Deep Groove

This bearing style is a modified version of the single row deep groove ball bearing through the addition of a second raw of balls. This design enlarges its capability to carry radial and thrust loads, due to the increased number of balls. Though using the same radial mounting space as the single-row style, its increased width lowers the capability of this bearing to accommodate misalignment

Manufacturing of Deep Groove bearings

Video 1

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Advantages of Rolling Bearing

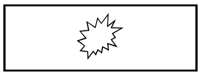
- Low starting moment.
- Low friction at all speeds.
- Low energy consumption.
- High reliability.
- Small width.
- Low consumption of lubricant.
- Long re-lubrication intervals.
- Easy to mount and dismount.
- Standardized dimensions

ME-305 (Machine Design II)

11-2 Bearing Life

If a bearing is clean, properly lubricated and mounted and is operating at reasonable temp., failure is due to fatigue caused by ***repeated contact stresses*** (Hertzian stress)

Fatigue failure consists of a spalling or pitting of the curved surfaces



Spalling – crack initiates below the curved surface at the location of maximum shear stress, propagates to the surface causing surface damage.

Failure criterion – spalling or pitting of an area of 0.01 in^2, Timken company (tapered bearings)

ME-305 (Machine Design II)

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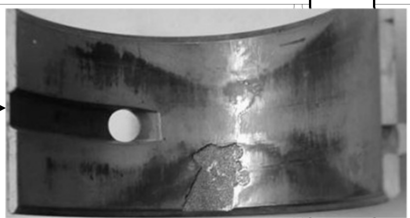
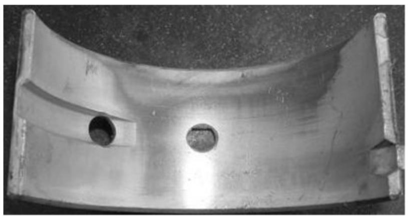
11-2 Bearing Life



Fatigue failed Lycoming 320 main bearing. Contact pressure from the crankshaft seems to be more concentrated



Main bearing wear. Contact occurs only on upper half of bearing. Crankshaft is not running true in bearing. Either crankshaft is bent or crankcase journals are mis-aligned

11-3 Bearing load Life

Life – number of revolution or hours of operation, at constant speed, required for the failure criterion to develop.

Rating Life – defines the number of revolution or hours of operation, at constant speed, in such a way that ***90%*** of the bearings tested (from the same group) will complete or exceed before the first evidence of failure develops. This is known as L_{10} life.

For ball bearings and spherical bearings;

L_{10} = 500 (hours) x 33.33 (rpm) x 60 = 10^6 = 1 million revolutions

For tapered bearings manufactured by Timken:

L_{10} = 3000 (hours) x 500 (rpm) x 60 = 90 x 10^6 = 90 million revolutions

Basic Dynamic Load Rating, C – constant radial load that a group of bearings can carry for L_{10} life.

ME-305 (Machine Design II)

11-3 Bearing load Life

- For a bearing with 90% reliability and given load and life, we can write a regression equation in the form

$$FL^{1/a} = constant$$ (As per the ISO 281:2007-02 standard)

- Where *F* is the applied radial load, *a* = 3 for ball bearings and *a* = 10/3 for roller bearings
- We can write

$$F_1 L_1^{1/a} = F_2 L_2^{1/a}$$
$$F_R L_R^{1/a} = F_D L_D^{1/a}$$
$$C_{10} = F_D \left(\frac{\mathcal{L}_D n_D 60}{\mathcal{L}_R n_R 60}\right)^{1/a}$$

- *F* is in kN, $\mathcal{L}$ is in hours, *n* is in rpm

ME-305 (Machine Design II)

Example 11.1

Select a deep groove ball bearing for a desired life of 5000 hours at 1725 rpm with 90% reliability. The bearing radial load is 2kN.

$$C_{10} = 16.1\ kN$$

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	d_H	Load Ratings, kN Deep Groove C_{10}	C_0	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

Read topics 11-4 & 11-5

- These topics are important if reliability is not equal to 90%.
- Weibull distribution is used to determine relationship between the load, life, and reliability.

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

- $x_D = \frac{L_D}{L_{10}}$
- x_o, $(\theta - x_o)$ and b are Weibull parameters.
- R_D is the desired reliability
- a_f is the application factor serves as fos.

ME-305 (Machine Design II)

Example 11.3

The design load on a ball bearing is 413 lbf 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 $b = 1.483$.

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

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

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

$$C_{10} = (1.2)(413)\left[\frac{540}{0.02 + 4.439(1 - 0.99)^{1/1.483}}\right]^{1/3} = 6696 \text{ lbf}$$

ME-305 (Machine Design II)

11-6 Combined radial and thrust loading

- Ball bearings can take both radial and thrust loading
- Both load can be expressed as

$$F_e = X_i V F_r + Y_i F_a$$

- F_e = equivalent radial load that does the same damage as the combined radial and thrust loads
- *V=1* when inner ring rotates and *1.2* when outer ring rotates (Rotation factor).
- *X* and *Y* are radial and thrust factors respectively (given)

ME-305 (Machine Design II)

11-6 Combined radial and thrust loading

ME-305 (Machine Design II)

- i = a factor whose value is 1 if $\frac{F_a}{VF_r} \leq e$ and 2 if $\frac{F_a}{VF_r} > e$
- Table 11-1 gives values of X_1, Y_1, X_2 and Y_2.
- e = minimum ratio b/w axial and radial loading bellow which axial load can be ignored
- e is a function of $\frac{F_a}{C_0}$ (Table 11-1)
- C_0 is the *basic static load rating* is the load that will produce a total permanent deformation in the raceway and rolling element at any contact point of 0.0001 times the diameter of the rolling element and is given by the manufacturer (Table 11-2).

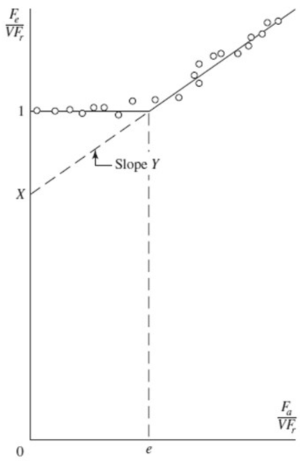
11-6 Combined radial and thrust loading

- $\frac{F_e}{VF_r} = 1$ when $\frac{F_a}{VF_r} \leq e$
- $\frac{F_e}{VF_r} = X + Y\frac{F_a}{VF_r}$ when $\frac{F_a}{VF_r} > e$

Table 11-1

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



Example 11-4

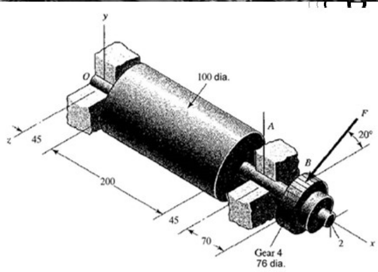
An SKF 6210 angular contact ball bearing has an axial load F_a of 1.8 kN and a radial load F_r of 2.2 kN applied with the outer ring stationary. The basic static load rating C_0 is 19.8 kN and the basic load rating is 35 kN. Estimate the L_{10} life at a speed of 720 rpm.

ME-305 (Machine Design II)

11-8 Selection of Ball and Cylindrical Bearings…

Example 11-7 (No thrust)

Shown in Figure 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 300rpm, and a design life of 30kh is desired. Use an application factor of 1.2 and select a pair of angular-contact *O2-series* ball bearings from Table 11-2 to be mounted at *O* and *A*. Use the same size bearings at both locations with 90% reliability.



11-8 Selection of Ball and Cylindrical Bearings…

Problem 11-23 (with thrust load)

An 02-series single-row deep-groove ball bearing is to be selected from Table 11–2 for the application conditions specified bellow.

F_r	F_a	Life	Ring rotating	Reliability
8 kN	2 kN	10 kh	Inner, 400 rpm	99%

Assume Table 11–1 is applicable if needed. Specify the smallest bore size from Table 11–2 that can satisfy these conditions.

Solution:

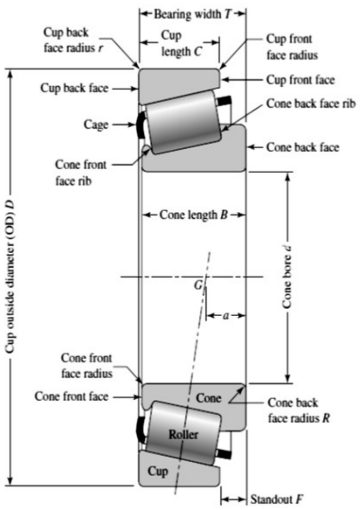
- Since bearing is not known, *"e"* can not be determined to check weather i =1 or 2.
- Start analysis from the middle of Table 11-1 with X = 0.56, Y = 1.63
- Iterate until getting the result.

11-9 Selection of Tapered Roller Bearings…

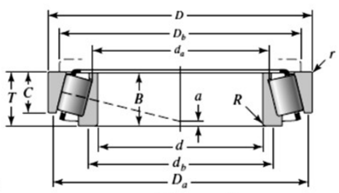
4 parts

1. Cup
2. Cone
3. Tapered rollers
4. Cage

- Two separable parts
- G is the location of the effective load center. Use this point to estimate radial bearing load.

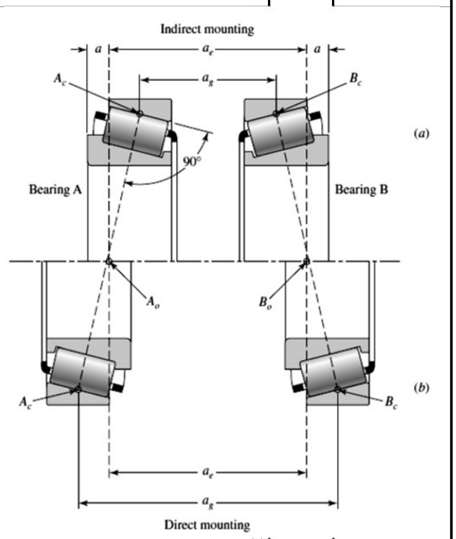


SINGLE-ROW STRAIGHT BORE



11-9 Selection of Tapered Roller Bearings…

- Can carry both radial and axial load, or any combination
- Even if no axial load, due to configuration will have thrust reaction.
- *DIRECT or INDIRECT* mounting arrangements can be used to eliminate this reaction force *(section 11-12 for mounting details)*
- A_o and B_o are the shaft reaction locations
- For the shaft as a beam, a_e is the effective spread
- a_g is greater for direct mounting than indirect

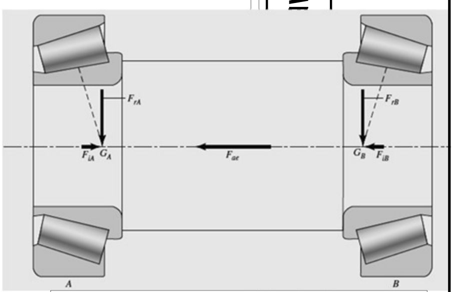


11-9 Selection of Tapered Roller Bearings…

- If a_e is same, the bearing are closer in Indirect mounting
- Due geometric configuration, even radial load will produce thrust reaction F_i which is (by Timken)

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

- *K* is the ratio of the radial load rating to the thrust load rating
- As first estimate, *K* = 1.5 for radial and 0.75 for steep angle bearings



Force vectors in Direct mounting

11-9 Selection of Tapered Roller Bearings…

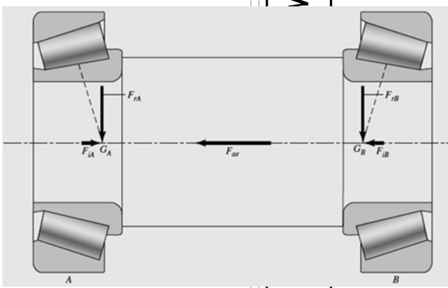
ME-305 (M

- Since bearings experience both axial and radial forces, then;

$$F_e = XVF_r + YF_a.$$

- Timken suggests (for all cases)

$$F_e = 0.4F_r + KF_a$$

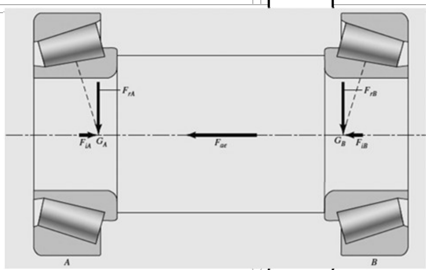


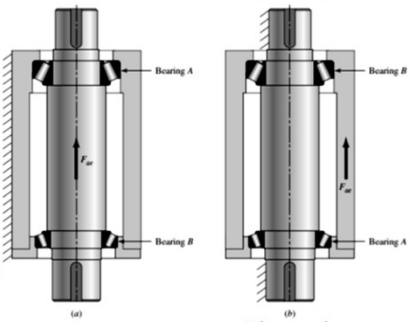
- F_r is the radial load and F_a is the net axial load induced by the other pair bearing and external load
- Only one bearing will carry the net axial load, which one? depends on
  - Mounting direction
  - External load direction
  - Weather Shaft or housing is moving

11-9 Selection of Tapered Roller Bearings…

Selection methodology

1. Visually determine the bearing which is squeezed by the external load F_{ae} and label it *A*, other is *B*
2. Determine which bearing carries the net axial load. Generally *A* will carry.
3. If the induced thrust F_{iA} from bearing *A* happens to be larger than the combination of the external thrust and the thrust induced by bearing *B*, then bearing *B* will carry the net thrust load.





11-9 Selection of Tapered Roller Bearings…

Selection methodology

4. Timken recommends

If $F_{iA} \leq (F_{iB} + F_{ae})$ $\begin{cases} F_{eA} = 0.4F_{rA} + K_A(F_{iB} + F_{ae}) & (11\text{–}16a) \\ F_{eB} = F_{rB} & (11\text{–}16b) \end{cases}$

If $F_{iA} > (F_{iB} + F_{ae})$ $\begin{cases} F_{eB} = 0.4F_{rB} + K_B(F_{iA} - F_{ae}) & (11\text{–}17a) \\ F_{eA} = F_{rA} & (11\text{–}17b) \end{cases}$

5. Once the equivalent radial loads are determined, they should be used to find the catalog rating load using any of Eqns. discussed earlier.
6. Timken uses 2 parameters Weibell parameters with x_o= 0 for Reliability other than 0.90

ME-305 (Machine Design II)

11-9 Selection of Tapered Roller Bearings…

Bearing Reliability

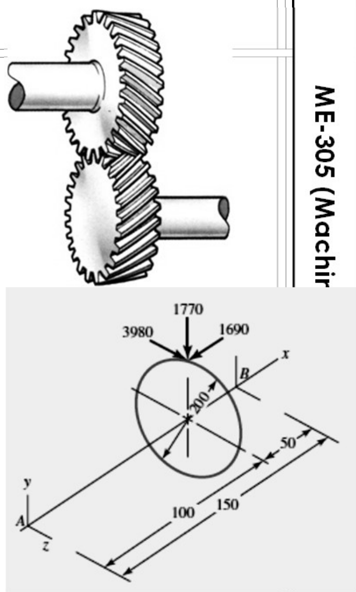
- Bearing Reliability greater than 0.90 is given by;

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

ME-305 (Machine Design II)

Example 11-8

- The shaft depicted in Fig. carries a helical gear with a tangential force of *3980N*, a radial force of *1770N*, and a thrust force of *1690N* at the pitch cylinder with directions shown. The pitch diameter of the gear is *200mm*. The shaft runs at a speed of *800rpm* , and the span (effective spread) between the direct-mount bearings is *150mm*. The design life is to be *5000h* 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.

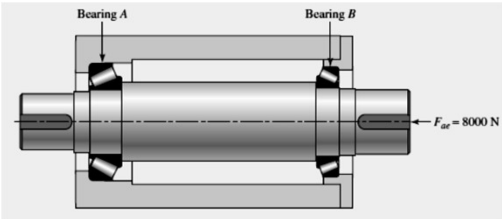


Example 11-8

									cone				cup			
bore	outside diameter	width	rating at 500 rpm for 3000 hours L_{10}		factor	eff. load center	part numbers		max shaft fillet radius	width	backing shoulder diameters		max housing fillet radius	width	backing shoulder diameters	
			one-row radial	thrust			cone	cup								
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	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
25.400 1.0000	57.150 2.2500	19.431 0.7650	11700 2620	10900 2450	1.07	−3.0 −0.12	M84548	M84510	1.5 0.06	19.431 0.7650	36.0 1.42	33.0 1.30	1.5 0.06	14.732 0.5800	48.5 1.91	54.0 2.13
25.400 1.0000	58.738 2.3125	19.050 0.7500	11600 2610	6560 1470	1.77	−5.8 −0.23	1986	1932	1.3 0.05	19.355 0.7620	32.5 1.28	30.5 1.20	1.3 0.05	15.080 0.5937	52.0 2.05	54.0 2.13
25.400 1.0000	59.530 2.3437	23.368 0.9200	13900 3140	13000 2930	1.07	−5.1 −0.20	M84249	M84210	0.8 0.03	23.114 0.9100	36.0 1.42	32.5 1.27	1.5 0.06	18.288 0.7200	49.5 1.95	56.0 2.20
25.400 1.0000	60.325 2.3750	19.842 0.7812	11000 2480	6550 1470	1.69	−5.1 −0.20	15578	15523	1.3 0.05	17.462 0.6875	32.5 1.28	30.5 1.20	1.5 0.06	15.875 0.6250	51.0 2.01	54.0 2.13
25.400 1.0000	61.912 2.4375	19.050 0.7500	12100 2730	7280 1640	1.67	−5.8 −0.23	15101	15243	0.8 0.03	20.638 0.8125	32.5 1.28	31.5 1.24	2.0 0.08	14.288 0.5625	54.0 2.13	58.0 2.28
25.400 1.0000	62.000 2.4409	19.050 0.7500	12100 2730	7280 1640	1.67	−5.8 −0.23	15100	15245	3.5 0.14	20.638 0.8125	38.0 1.50	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	19.050 0.7500	12100 2730	7280 1640	1.67	−5.8 −0.23	15101	15245	0.8 0.03	20.638 0.8125	32.5 1.28	31.5 1.24	1.3 0.05	14.288 0.5625	55.0 2.17	58.0 2.28

Example 11-11

- Consider a constrained housing as depicted in Fig. 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 10000 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.



ME-305 (Machine Design II)

Solution

- Reactions

$$F_{rA} = F_{rB} = 0$$
$$F_{aA} = F_{ae} = 8000 \text{ N}$$

- Bearing *B* is unloaded, $R = R_A = 0.95$
- With no radial load, $F_r^B = 0$, and using $K = 1$, we get

$$F_{eA} = (1)8000 = 8000 \text{ N}$$
$$F_{eB} = F_{rB} = 0$$

- Use K = 1 to get $F_e^A = F_{ae} = 8000$ N and $F_e^B = F_r^B = 0$, also

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

- Use equation to calculate C_{10}

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

ME-305 (Machine Design II)

Solution…

a) Select cup HM88610 and cone HM88630 with thrust load of 17200 N

b) Bearing B has no load so use a simple ball or roller bearing

- The actual reliability of bearing *A* will be

$$R_A \doteq 1 - \left\{ \frac{x_D}{4.48[C_{10}/(a_f F_D)]^{10/3}} \right\}^{3/2} = 0.963$$

- The actual reliability of Bearing *B* will be ($F_D = 0$)

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

c) The combined reliability is

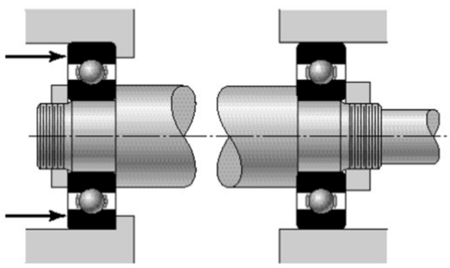
$$R = R_A R_B = 0.963(1) = 0.963$$

- Which is greater than 0.95 and is OK

ME-305 (Machine Design II)

Installation of Bearing
11-12 Mounting and enclosures

There are so many methods of mounting antifriction bearings that each new design is a real challenge to the ingenuity of the designer. The housing bore and shaft outside diameter must be held to very close limits, which of course is expensive. There are usually one or more counterboring operations, several facing operations and drilling, tapping, and threading operations, all of which must be performed on the shaft, housing, or cover plate. Each of these operations contributes to the cost of production, so that the designer, in ferreting out a trouble-free and low-cost mounting, is faced with a difficult and important problem.

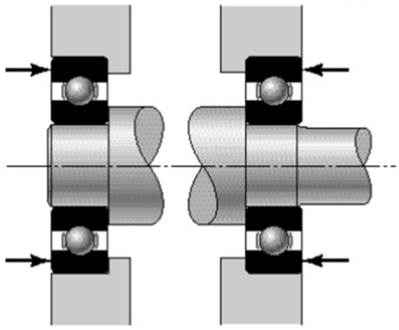


A common bearing mounting.

11-12 Mounting and enclosures…

ME-

Bearing bracket

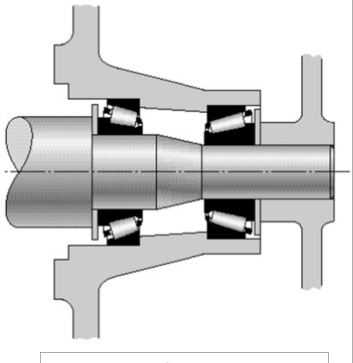


Bearing Housings

11-12 Mounting and enclosures…

ME-305 (Machine Design II)

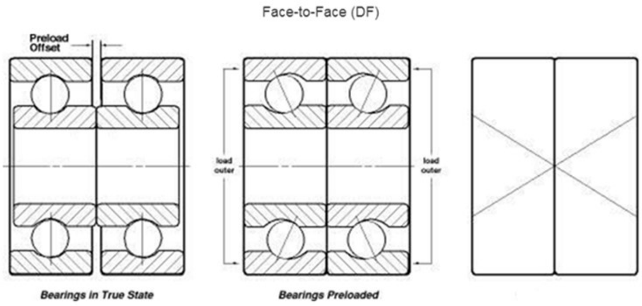
Tapered bearing in a car front wheel



Mounting for a washing-machine spindle (Indirect mount)

Mounting of Angular Ball Bearing

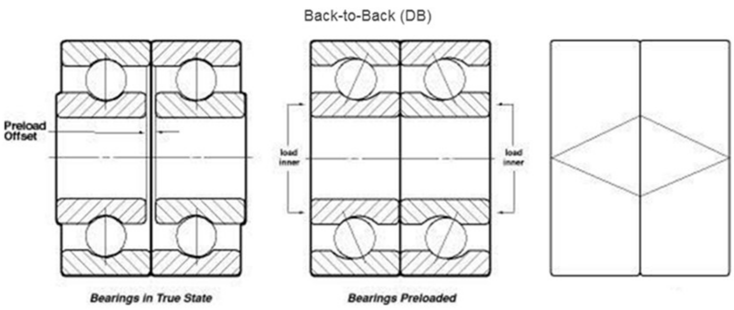
When maximum stiffness and resistance to shaft misalignment is desired, pairs of angular-contact ball bearings (Fig. 11–2) are often used in an arrangement called *duplexing*.



When face-to-face (DF) duplex pairs are mounted, the inner rings abut and the outer rings are drawn together, providing a higher radial and axial stiffness and accommodation of misalignment.

05 (Machine Design II)

Mounting of Angular Ball Bearing

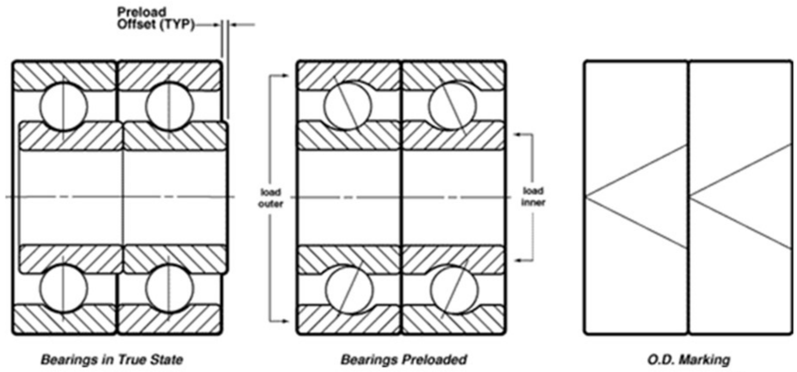


When a back-to-back (DB) duplex pair is mounted, the outer rings abut and the inner rings are drawn together, providing maximum stiffness.

ME-305 (Machine Design II)

Mounting of Angular Ball Bearing

Tandem (DT)



With tandem (DT) pairs, both inner and outer rings abut and are capable of sharing a thrust load, providing increased thrust capacity.

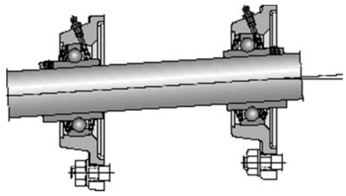
Mounting of Angular Ball Bearing

- DF
  - Face-to-face; will take heavy loads and thrust leading from either direction.
- DB
  - Back-to-back; greatest aligning stiffness and is also good for heavy radial loads and thrust loads from either direction.
- DT
  - Tandem arrangement; is used where thrust is always in the same direction.

Show video 2

Alignment

- As general, the permissible misalignment is
  - 0.0035 to 0.0047 radians for deep-groove ball bearings
  - 0.0087 for spherical ball bearings
  - 0.001 for tapered roller bearings
- The life decreases by 20% for every 0.001 radians after the given limit.



ME-305 (Machine Design II)

Sealing for Enclosures and Bearings

- Read

ME-305 (Machine Design II)

Problems…

Problems: 11-1, 11-2, 11-3, 11-7, 11-8, 11-10, 11-13 11-16, 11-20, 11-21, 11-22, 11-24, 11-25, 11-31, 40, 43

ME-305 (Machine Design II)