Chapter 6 Rolling Element Bearings

6-1 Advancing from mechanics

Figure 6-1 shows one of the mechanical models that we saw in *Mechanics of Materials*, where a shaft, or a beam, is simply supported by two bearings. The bearings were often expressed by some simple symbols, which is fine in *Mechanics of Materials* because the focus of that subject there is on forces and stresses. We will learn and design real structures of the bearing supports in this chapter. One of the possible structures for the model shown in **Figure 6-1** is given in **Figure 6-2**, which is a shaft-system assembly drawing. Keep in mind that the end caps are needed in order to secure the bearing positions along the shaft. The dash-dot lines across the end caps are for the bolts whose details are omitted here.

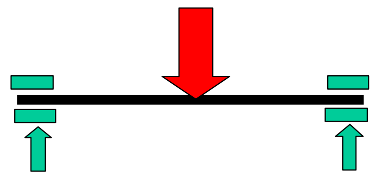


Figure 6-1 Mechanical model in Mechanics of Materials

Each mechanical model corresponds to a real mechanical structure. The system shown in **Figure 6-2** is for a simply supported beam (shaft) with two radial bearings, which are two ball bearings in this design. Their inner rings rotate with the shaft, but their outer rings are kept largely stationary in the housing. The balls connect the rotating and stationary parts.

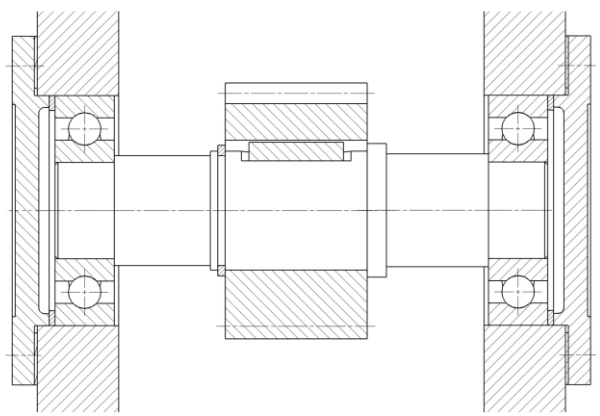


Figure 6-2 Real structure for the mechanical model shown in **Figure 6-1** (Plotted by Jie Lu).

We have already learned that if we have a mechanical structure, we can easily obtain the corresponding mechanical model for the structure, and usually the model is unique. However, if we are given a mechanical model, our design of the mechanical structure can hardly be unique. We can design the system shown in **Figure 6-2**, based on the model given in **Figure 6-1**, using ball bearings. We can also do so using two roller bearings, or even two sliding bearings on a different concept, to be discussed in the next chapter. Nevertheless, our designs should be reasonable, our products should be cost effective, with the considerations of convenience in manufacturing, assembly, and maintenance.

First, we would like to learn these bearings and to know how they work. We will then learn how to design bearing supports. The manufacturing of rolling-element bearings has been highly specialized. Therefore, this chapter mainly deals with bearing-selection and bearing-support-design problems.

6-2 Introduction to bearings, classification

Bearing are used to connect surfaces under different motions; it allows parts to contact and relatively move. The bearings in **Figure 2** are typical examples, where the bearing's outer rings are mounted inside the stationary housing, while the inner rings are on the rotating shaft. Thus, the housing frame surface under no motion is connected to the shaft surface that is doing a rotation motion. There are two kinds of bearings: rolling-element bearings and sliding bearings (or fluid-film bearings). A rolling-element bearing, by its definition, has elements that roll when the bearing works. Typical rolling elements are balls and rollers (straight or tapered). On the other hand, a sliding bearing simply consists of a shaft segment (called the journal) and a hollow cylinder (named the sleeve, bushing, or bearing). It does not include any rolling components between the surfaces under a relative motion, rather, it maintains a fluid layer to separate and lubricate the surfaces. That is why a sliding bearing is also called a fluid-film bearing. Typically, the relative motion between its journal and sleeve surfaces is sliding that shears the fluid film.

Figure 6-3 shows two examples of rolling-element bearings: a ball bearing (left), where the balls are the rolling elements, and a roller bearing (right), where the tapered rollers are the rolling elements. These two bearings look very different. The ball bearing is a radial bearing designed mainly for supporting a radial load, while the roller bearing here is a thrust bearing designed mainly for supporting an axial load.

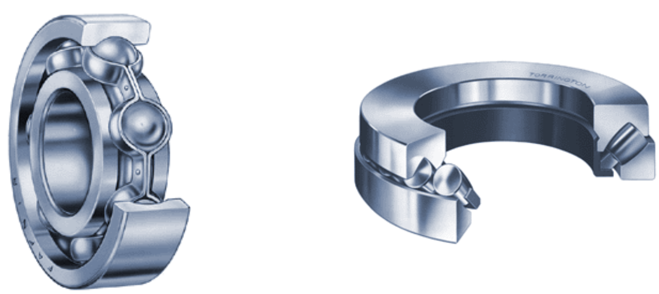


Figure 6-3 Rolling-element bearings: a ball bearing (left) and a roller bearing (right) Courtesy of the Timken Company.

An example of sliding bearings is shown in **Figure 6-4**, where a journal (which is usually a portion of a shaft) and a sleeve, or a bearing, are assembled (left) and are under a relative sliding. The right plot in this figure is the cross-sectional view of the bearing, which is nothing more than a hollow cylinder of a finite length. The structure looks very simple; however, the mechanics is quite complicated. Fluid dynamics is applied to ensure that there is a thin lubricant layer in between the surfaces to support the working load of the bearing and to separate the two surfaces, and thus to present rubbing and wear of the surfaces.

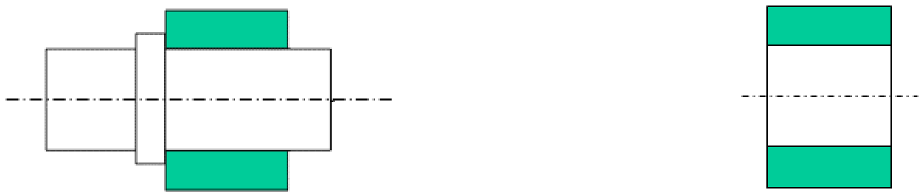


Figure 6-4 Sliding bearing set (left) and the sectional view of the sleeve (right). The green colored area means cross section.

6-3 Rolling-element bearings

Loads on shaft and bearing reactions. Generally, the loads on a shaft are three-dimensional. **Figure 6-5** shows the shaft for a helical gear, where the radial, tangential, and axial load components, W_{21}^r, W_{21}^t, and W_{21}^a, applied by gear 2, all exist. However, when these forces are transmitted to the bearings, the bearing reactions to the tangential and radial gear forces are A_z and A_y, and B_z and B_y. They can be combined into $\vec{A}_r$ and $\vec{B}_r$ as the total radial reaction forces at bearing A and B. In the bearing industries, the axial load is often called the thrust load. The thrust load here can be assigned to either bearing A or B. Let's assume bearing B takes care of W_{21}^a by B_a, which may also be named B_x. Sometimes we use $\boldsymbol{A}$ and $\boldsymbol{B}$, or $\boldsymbol{R}_A$ and $\boldsymbol{R}_B$, to express the total reactions at bearings A and B, and for the system shown in **Figure 6-5**, $\boldsymbol{A} = \boldsymbol{R}_A = [A_z, A_y]$, and $\boldsymbol{B} = \boldsymbol{R}_B = [B_x, B_y, B_z] = [B_a, B_y, B_z] = [B_a, B_r]$.

Bearings do not constrain torque because the shaft rotates. They confine forces and the in-plane bending moment. The radial forces and the bending moment may be supported by a radial bearing pair. The thrust force may be supported by a thrust bearing. However, radial-thrust bearings are commonly used instead of separated radial and thrust bearings.

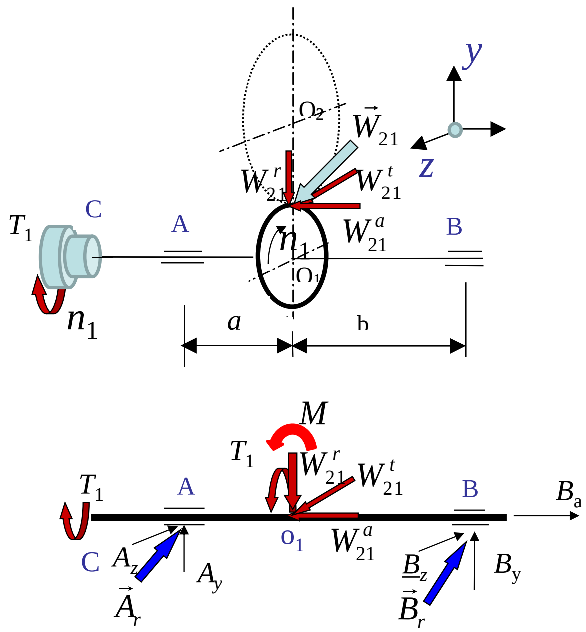


Figure 6-5 Forces on a shaft and bearing reactions.

Structure of rolling-element bearings. Rolling-element bearings, or rolling bearings for short, are assemblies. Three typical bearings made by major bearing manufacturers are shown in **Figures 6-6**, **6-7**, and **6-8**. **Figure 6-6** is a basic ball bearing, consisting of an inner ring, an outer ring, a ball retainer, and a number of balls. The bearing rings have bearing races, which are the designed tracks for the ball rotation. The balls are in contact with the inner race of the outer ring and the outer race of the inner ring, respectively. The bearing bore fit with the supporting journal and the outer surface of the bearing fits with the housing bore. The cross-sectional drawing of the radial ball bearing is also shown in this figure. Such a ball bearing is designed mainly for taking the radial load.

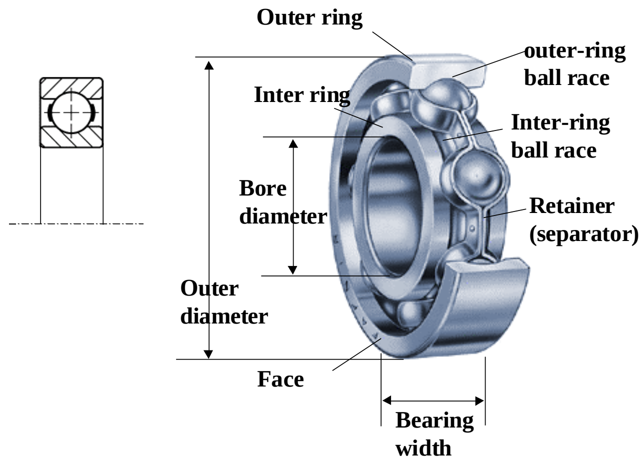


Figure 6-6 A deep-groove ball bearing (Courtesy of the Timken Company) and the cross-sectional drawing of such a ball bearing (left, one half is shown).

Figure 6-7 shows a Timken tapered-roller bearing. A cross-sectional drawing and bearing components are presented. It has a cone (the inner ring), a cup (the outer ring), a number of rollers, and a cage as the roller retainer. Note that the widths of the cup and the cone are different. This type of bearings is designed to support both the radial load and thrust load (axial load). It allows one direction of the thrust load because its inner ring and outer ring are usually separate. We will learn later why the tapered-roller bearings can provide both radial and thrust supports later.

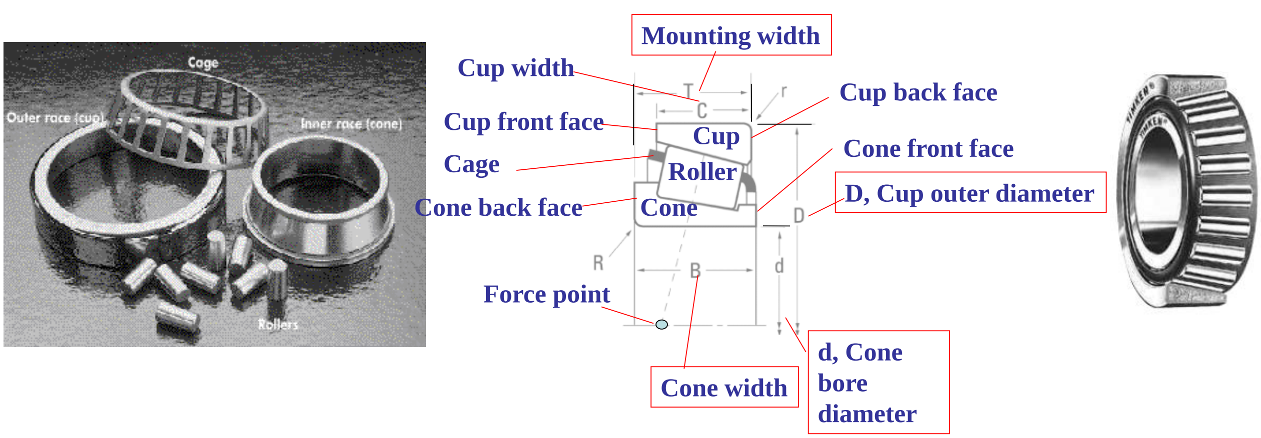


Figure 6-7 A Timken tapered-roller bearing (Courtesy of the Timken Company).

Figure 6-8 is an SKF double row tapered-roller bearing. It is for heavy-duty applications because double-row rollers allow a heavy load. This bearing allows thrust loads in both directions.

Figure 6-8 A SKF double-row tapered-roller bearing (Courtesy of SKF).

Major types of rolling-element bearings. Now, let's have a general view of rolling-element bearings. They may be simply classified as 1) radial bearings, which are mainly used for radial loads, 2) radial-thrust bearings, for combined radial (major) and thrust (minor) loads, 3) thrust-radial load, for combined thrust (major) and radial (minor) loads, and 4) thrust bearings, mainly for thrust loads. **Figure 6-9** shows several radial bearings together with their cross-sectional drawings. Note that the ball bearing, which is a deep-groove ball bearing here, may allow some light thrust loading. However, the roller and the needle bearings are for radial loading only. The inner ring and outer ring of a roller or needle bearing are usually separate. The needle bearing may have a very small radial dimension, and the needle rollers may be directly mounted in between a shaft and a bearing bore, without the use of an inner and/or outer ring, for radial-direction compact design.

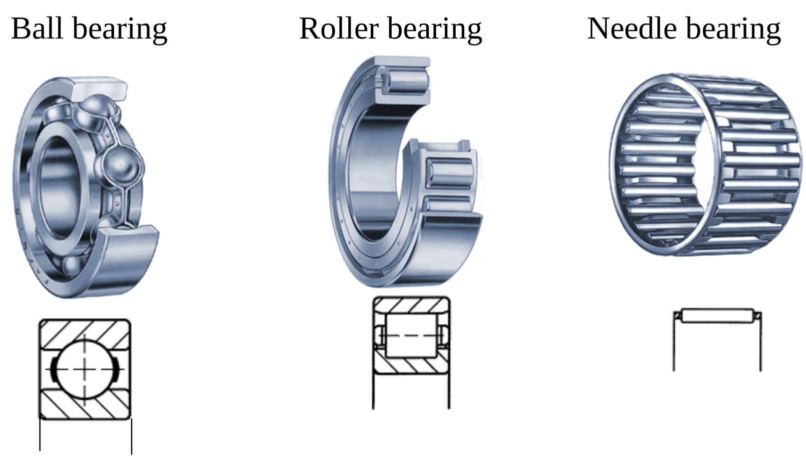


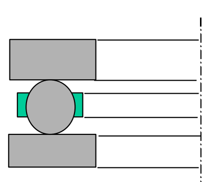
Figure 6-9. Several radial bearings, from left, deep-groove ball bearing, roller bearing, and needle bearing.

Figure 6-10 shows several radial-thrust bearings. In the angular-contact ball bearing and the tapered-roller bearing, angular contacts are created between the rolling elements and the races. Again, roller bearings have a higher load capacity than ball bearings. Double row bearings have a higher load capacity than single-row bearings.

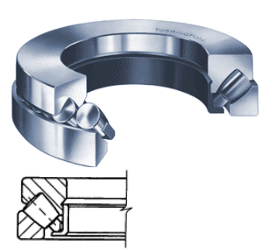
Generally, balls permit higher shaft (also the bearing inner ring) rotation speed as a result of lower friction due to smaller contact areas, which on the other hand, allow lower load.

Figure 6-10. Several radial-thrust bearings, from left, angular-contact ball bearing, tapered-roller beating, double-row roller bearing, and double-row spherical bearing. Note that in the tapered roller bearing drawing, the grey areas show cross sections.

Two thrust bearings are shown in **Figure 6-11**. Actually, the tapered-roller bearing (right) is a thrust-radial bearing because the tapered rollers permit a certain amount of radial loading. A rotating shaft is mounted on one of the bearing rings and rotates with this ring. Similarly, roller bearings have a higher load capacity.



Ball bearing



Tapered roller bearing

Figure 6-11 Ball thrust bearing and taper-roller thrust-radial bearing.

Which bearing should we select in our design? This depends on operation and environment requirements. **Table 6-1** lists several tips for bearing selection.

Table 6-1 Tips for bearing selection

Case or condition	First choice
•High speed	*Ball bearings*
•Single or combined radial and thrust load	*Radial, thrust, or radial-thrust bearings*
•Heavy load	*Roller bearings*
•Very heavy load	*Double-row bearings*
•Environment seal	*Bearings with seals*
•Misalignment	*Self-aligning bearings*
•Compact space	*Needle bearings*

6-4 Bearing life

The major failure mechanism of rolling-element bearings are pitting and large plastic deformation, and the former is a typical rolling contact fatigue (RCF) failure. **Figure 6-12** shows the contact between a rolling element (ball or roller) and a race surface (simplified it to a plane, as we did in gear contact problems), where a high contact pressure (the peak of the red pressure distribution) and large subsurface stresses (not shown) are expected. The contact may be a point contact (or a circular contact) between the surfaces of a ball and a race, or a line contact (or a rectangular contact) between the surfaces of a roller and a race. The high contact stress varies cyclically when the balls travel with the rotating ring, resulting in pits in the balls and the rings if the bearings are not properly designed or used, or run for a long time. Pits are then responsible for uneven rotation, vibration, noise, and even dynamic impact that is able to cause further bearing damage in a larger scale. In the bearing industry, the bearing design against pitting is to design the bearings with a proper service life in which no more than 10% of the apparently same bearings failure due to RCF.

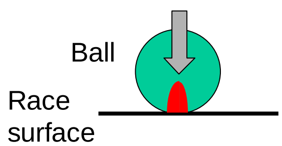


Figure 6-12 Contact between a rolling element and a race simplified to a plane.

Bearing life. Bearing life is defined by the total number of revolutions (or total hours at a constant speed) before the RCF failure occurs. A commonly used terminology for the bearing life is the rating life, or the L_{10} life.

Bearing rating life (the L_{10} life). The rating life of a group of apparently identical bearings is defined as the number of revolutions (or hours at a constant speed) at which 10% of the bearings will fail (or 90% will survive).

The life of a bearing is closely related to the load that the bearing supports. Research shows that bearings seem to have the same "capacity" in terms of (LF^m), as shown in **Figure 6-13**. The statement that LF^m is constant anywhere along the curve results in Equation (6-1), which is actually the exponential relationship shown in Equation (6-2).

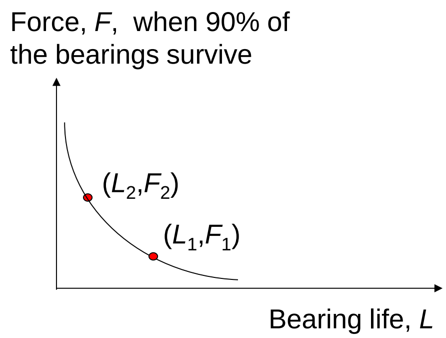


Figure 6-13 Relation between bearing life and load.

$$L_1 F_1^m = L_2 F_2^m = L_i F_i^m \tag{6-1}$$

$$\frac{L_1}{L_2} = \left(\frac{F_2}{F_1}\right)^m \tag{6-2}$$

Here, m = 3 is used for ball bearings, and m = 10/3 for roller bearings, with either straight or tapered rollers.

Basic load rating, *C*. The basic load rating of a bearing is a constant radial load that a group of apparently identical bearings can endure for one million revolutions of the inner ring.

If $F_2 = C$ corresponding to L_2 = one million revolutions of the inner rings of the bearings, Equation 6-2 becomes

$$L_1 = \left(\frac{C}{F_1}\right)^m L_2 \tag{6-3}$$

Or using $P = F_1$, where P is the equivalent bearing load, to be explained later.

$$L = \left(\frac{C}{P}\right)^m \quad \text{unit in one million revolutions} \tag{6-4}$$

There is one more term we should define: the static load rating, to prevent large plastic deformation.

Static load rating, C_0. Static load rating is the maximum radial load carried by the most heavily loaded rolling element before plastic deformation happens under a set of given operating conditions.

6-5 Ball bearing selection and bearing-support structure design

Ball bearing designation. Bearing companies have their own ways of bearing designation. We should look at their catalogues for the expressions of different bearings. **Figure 6-14** shows how SKF names its bearings by numbers.

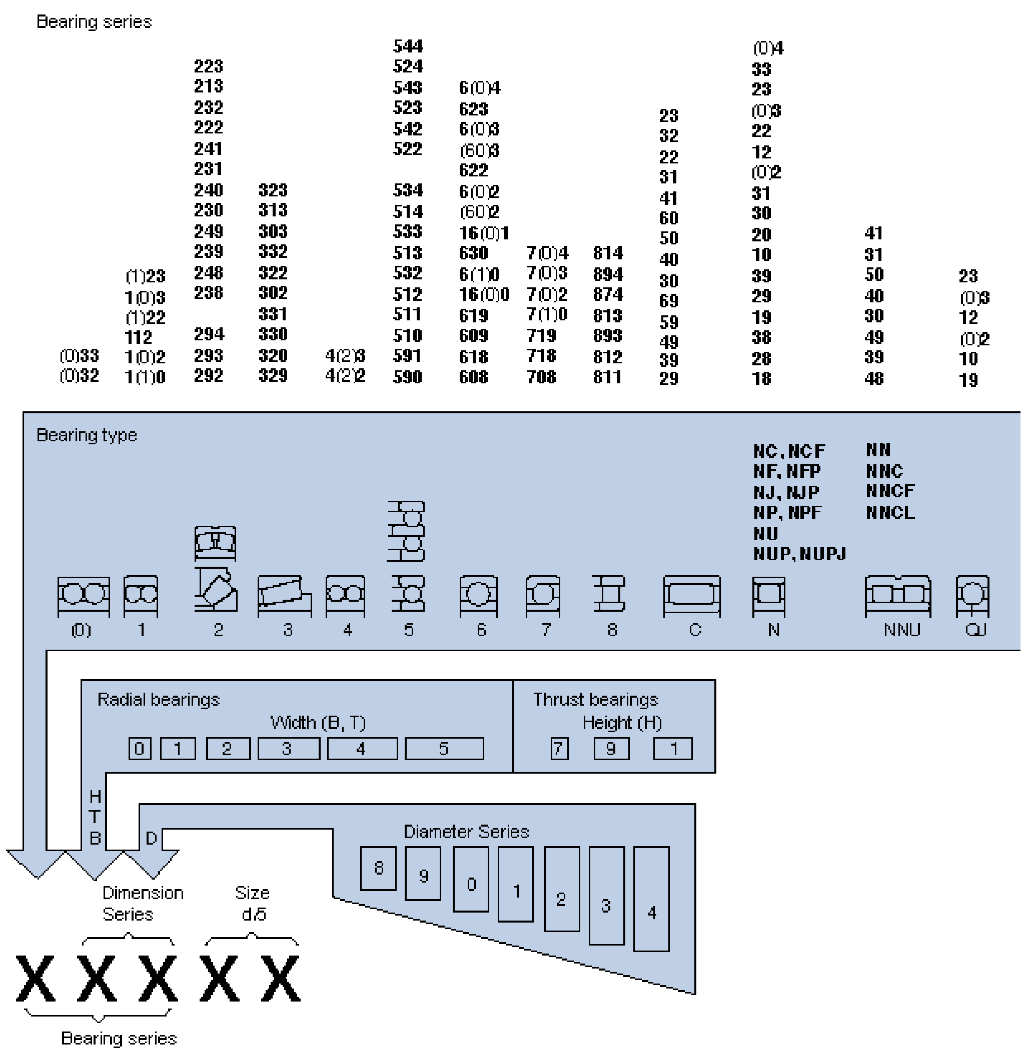


Figure 6-14 SKF bearing designations (Courtesy of SKF)

The series 6 SKF bearings are deep-grove bearings, which are mainly designed for radial-load supporting; however, they also allow a small amount of thrust loading. **Figure 6-15** is an example showing the designation of a deep-groove ball bearing with four digits.

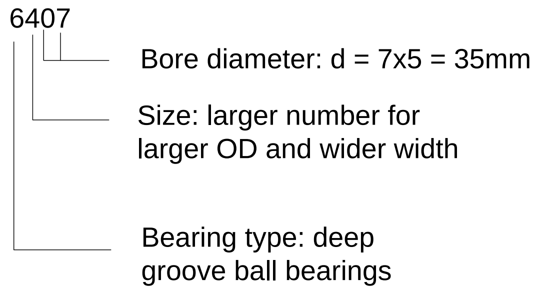


Figure 6-15 Deep-groove ball bearing designations.

As it was mentioned before, the rolling-element bearing design problem is actually a bearing selection issue. We will use the catalog data for bearing selection, for which we should do some simple calculations.

Ball bearing selection (also applicable to cylindrical-roller bearings). When we select a bearing, we should select the bearing type, the bore diameter (or inner diameter), d_b, the outer diameter, d_a, and the bearing width, b_w. The bearing type should be matched with the loads that the bearing should support. The radial and width dimensions should satisfy the needs for the bearing-support structure design.

Usually, the duration of the bearing service in hours is known, which can be easily converted into the L_{10} life by the following equation.

$$L_{10} = h(60)n/10^6 \qquad \text{(In million revolutions)} \qquad (6\text{-}5)$$

Rotation speed, rpm

Hours of life

Equivalent load P is defined in Equation (6-4) for the bearing selection calculation. This load considers the loading conditions of the bearing, for instance, whether there is a thrust load, P_a, involved or not, and if the operation is smooth or not as indicated by the application factor A_p. For a pure radial load, P_r, the equivalent load is this this radial load (which equals the bearing radial reaction force) multiplied by A_p, as shown in Equation (6-6). However, if the load has both radial and thrust components, the equivalent load should be calculated by means of Equation (6-7), where two factors, X and Y, are introduced for the load conversion.

Pure radial loading: $P = A_p P_r$ (6-6)

Radial-thrust loading $P = A_p(XP_r + YP_a)$ (6-7)

Equation (6-7) is applied only when the thrust load is significantly large, determined by parameter e, as indicated by Equations (6-8) and (6-9).

$P = A_p P_r$ when $P_a/P_r \le e$ (6-8)

$P = A_p(X P_r + Y P_a)$ when $P_a/P_r > e$ (6-9)

Note that the symbol for the application factor here is different from that used in gear analysis because gears and bearings are different components; however, their meanings are the same. For non-shock loading, A_p may simply be one.

The values of parameter e are listed in **Table 6-2** based on the ratio of the thrust load to the static load rating of the selected bearing, C_0. **Table 6-3** lists the basic load ratings and dimensions for a number of 6-series SKF single-row deep-groove ball bearings.

Table 6-2 Equivalent radial load factors X and Y for deep grove bearings

P_a/C_o	e	$P_a/P_r \le e$		$P_a/P_r > e$	
		X	Y	X	Y
< 0.025	0.22	1	0	0.56	2.0
0.025	0.22	1	0	0.56	2.0
0.04	0.24	1	0	0.56	1.8
0.07	0.27	1	0	0.56	1.6
0.13	0.31	1	0	0.56	1.4
0.25	0.37	1	0	0.56	1.2
0.50	0.44	1	0	0.56	1.0

Here, for $P_a/C_0 < 0.025$, use 0.025 for P_a/C_0.

Table 6-3 Basic load ratings and dimensions of SKF single-row deep-groove ball bearings (Courtesy of SKF)

Nominal diameter *d*	OD	Width	Basic load ratings Dynamic	Static limit	Allowable load	Speed ratings Grease	Oil	Mass	Designation
mm	mm	mm	C (N)	C_o(N)	(N)	rmp	rpm	Kg	
15	**32**	**8**	**5 590**	**2 850**	**120**	**22000**	**28000**	**0.025**	**16002**
	32	8	5 590	2 850	120	22000	28000	0.030	6002
	35	11	7 800	3 750	160	19000	24000	0.045	6202
	42	13	11400	5400	228	17000	20000	0.082	6302
20	**42**	**8**	**6890**	**4050**	**173**	**17000**	**20000**	**0.050**	**16004**
	42	12	9360	5000	212	17000	20000	0.090	6004
	47	14	12700	6550	280	15000	18000	0.11	6204
	52	15	15900	7800	335	13000	16000	0.14	6304
	72	19	30700	15000	640	10000	13000	0.40	6404
25	**47**	**12**	**11200**	**6 550**	**275**	**15 000**	**18000**	**0.080**	**6005**
	52	15	14000	7800	335	12000	15000	0.13	6205
	62	17	22500	11600	490	11000	14000	0.23	6305
	80	21	35800	19300	815	9000	11000	0.53	6405
30	**55**	**15**	**13300**	**8300**	**355**	**12000**	**15000**	**0.12**	**6006**
	62	16	19500	11200	475	10000	13000	0.20	6206
	72	19	28100	16000	670	9000	11000	0.35	6306
	90	23	43600	23600	1000	8500	10000	0.74	6406
35	**62**	**14**	**15900**	**10200**	**440**	**10000**	**13000**	**0.16**	**6007**
	72	17	25500	15300	655	9000	11 000	0.29	6207
	80	21	33200	19000	815	8500	10000	0.46	6307
	100	25	55300	31000	1290	7000	8500	0.95	6407

Table 6-3 (continued)

Nominal diameter	OD	Width	Basic load ratings Dynamic	Static limit	Allowable load	Speed ratings Grease	Oil	Mass	Designation
mm	mm	mm	C (N)	C_o(N)	(N)	rmp	rpm	Kg	
40	**68**	**15**	**16800**	**11600**	**490**	**9500**	**12000**	**0.19**	**6008**
	80	18	30700	19000	800	8 500	10000	0.37	6208
	90	23	41000	24000	1020	7500	9000	0.63	6308
	110	27	63700	36500	1530	6700	8000	1.25	6408
45	**75**	**16**	**20800**	**14600**	**640**	**9000**	**11000**	**0.25**	**6009**
	85	19	33200	21600	915	7500	9000	0.41	6209
	100	25	52700	31500	1340	6700	8000	0.83	6309
	120	29	76100	45000	1900	6000	7000	0.55	6409
50	**80**	**16**	**21600**	**16000**	**710**	**8500**	**10000**	**0.26**	**6010**
	90	20	35100	23200	980	7000	8500	0.46	6210
	110	27	61800	38000	1600	6300	7500	1.05	6310
	130	31	87100	52000	2200	5300	6300	1.90	6410

Example 6-1 Ball bearing selection (including straight-roller bearings)

Select a pair of SFK deep-grove ball bearings working in a non-shock application. Here, the radial load is P_r = 8KN and the thrust load is P_a = 4 KN. The required L_{10} life is 5000 h, and the shaft speed is 900 rpm.

Solution and solution steps

1. Force analysis, which has been done. P_r = 8KN, P_a = 4 KN

   $P = XP_r + YP_a$, No shock, and the application factor should be 1.

2. Initial selection

   Because $P = XP_r + YP_a$, with unknown X and Y, we must select a bearing first, so that we can get C_0, then we can determine X and Y from P_a/C_0.

   A Start from considering the radial load only, $P = P_r$ = 8KN

   $L_{10} = h(60)n/10^6 = 5000(60)(900)/10^6 = 270$ millions of cycles.

   From $L = \left(\frac{C}{P}\right)^m$, we get $C = PL^{1/m} = 8(270)^{1/3} = 52KN$.

B From Table 6-3, select: SKF 6309 based on C=52KN. The bearing diameter is d = 45 mm (bore diameter),

and C = 52700N, C_0 =31500N

3. Evaluation, to see if SKF 6309, d_b = 45 mm, C = 52700N, C_0 =31500N is a good choice or not.

A Determine X, Y, from P_a/C_0 =4/31.5=0.128 to determine e = 0.309, using Table 6-2.

Because P_a/P_r =4/8=0.5 > 0.309, we have, X = 0.56, and Y =1.407.

B Re-calculate P and C

$$P = XP_r + YP_a = 0.56(8) + 1.407(4) = 10.108KN$$

$C = PL^{1/m} = 10.108(270)^{1/3} = 65.38KN > 52.7KN$, we need a new bearing.

4. Re-select a new bearing based on C = 65.38KN.

Try SKF 6409, d = 45 mm, C = 76100N, C_0 =45000N

A Determine X, Y again for this new bearing based on P_a/C_0 =4/45=0.089, e = 0.283

Because P_a/P_r =4/8=0.5 > 0.283, we have X = 0.56, and Y =1.53.

B Re-calculate P and C one more time.

$$P = XP_r + YP_a = 0.56(8) + 1.537(4) = 10.63KN$$

$$C = PL^{1/m} = 10.63(270)^{1/3} = 68.7KN < 76.1KN .$$

Therefore, no further modification is needed,

Bearing support design for radial bearings. Bearing selection is only a portion of the design work. We need to mount the bearings onto their shafts and into their housing. **Figures 6-16**, **6-17**, and **6-18** are examples of bearing support designs. In the design example for ball-bearing support shown in **Figure 6-16**, two deep-groove ball bearings are used. Note that the inner and outer rings of a deep-groove ball bearing are not separable. Therefore, constraints on one side of the inner ring and one side of the outer ring, as indicated by the arrows in the top view of this figure, should be sufficient to hold the bearing at its desired location. The bearing inner ring may be assembled against a shaft shoulder, and the outer ring may be held by an end-cap ring. Note that the mounting dimensions, $D_{a,\,max}$ and $D_{b,\,min}$, given in **Table 6-**4, are usually from the bearing catalog. The shaft shoulder should not be too low and the thickness of the end-cap ring there cannot be too thin (a certain mounting strength is needed); they cannot be too tall or too thick either (they should not block the bearing ring and the rolling element for assembly and disassembly convenience). The assembly (only the top half is shown) in this figure presents the design for a simply supported shaft involving two deep-groove ball bearings. This assembly is largely the same as the one shown in **Figure 6-2**. The end cap is connected to the housing by several bolts, indicated by the centerlines, although detailed structures are not shown here for simplicity. We will learn bolted connections later.

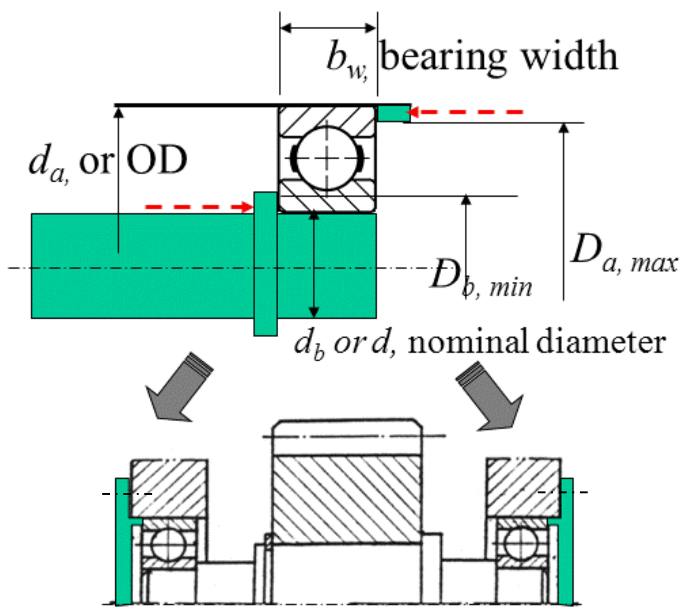


Figure 6-16 Ball bearing mounting.

Figure 6-17 shows roller-bearing mounting considerations. Here, the inner and outer rings of the bearing are separable. Therefore, both inner and outer rings should be properly contained as indicated by the arrows. A nut is applied to the shaft (threaded there) to lock the bearing inner ring in its place, as shown in the upper diagram, but a snap ring is used for the shaft in the lower diagram to hold the bearing inner ring. The nut looks stronger than the snap ring. However, an anti-loosening measure should be taken to ensure that the nut is properly tightened during machine operation. **Figure 6-18** shows the design of a fixed end of a "cantilever beam" under a radial load, where two roller bearings are used. This "beam" is a rotating shaft, very different from the fixed beams in civil engineering. Nevertheless, the mechanical model of this design is a cantilever beam, also shown in **Figure 6-18**, which is also shown in the same figure. The moment at the "wall" caused by the force applied at the free end is balanced by the moment produced by the radial reactions of the bearings. Apparently, this structure does not permit any thrust loading due to the use of the straight roller bearings.

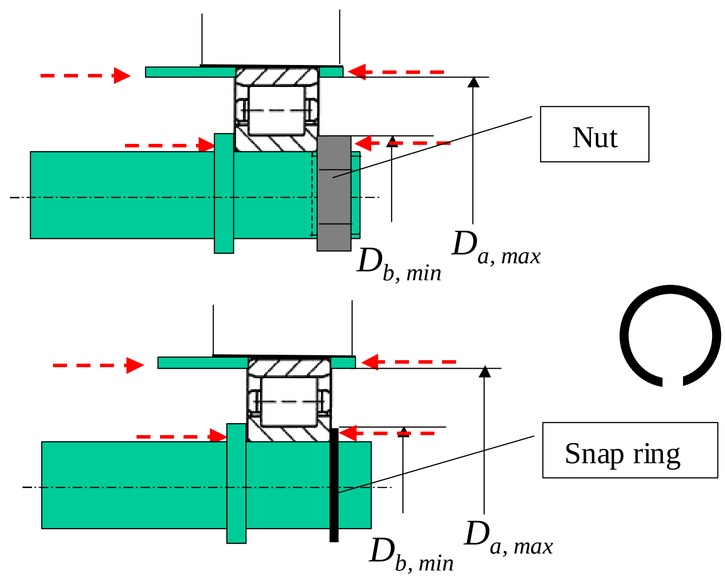


Figure 6-17 Roller bearing mounting structures.

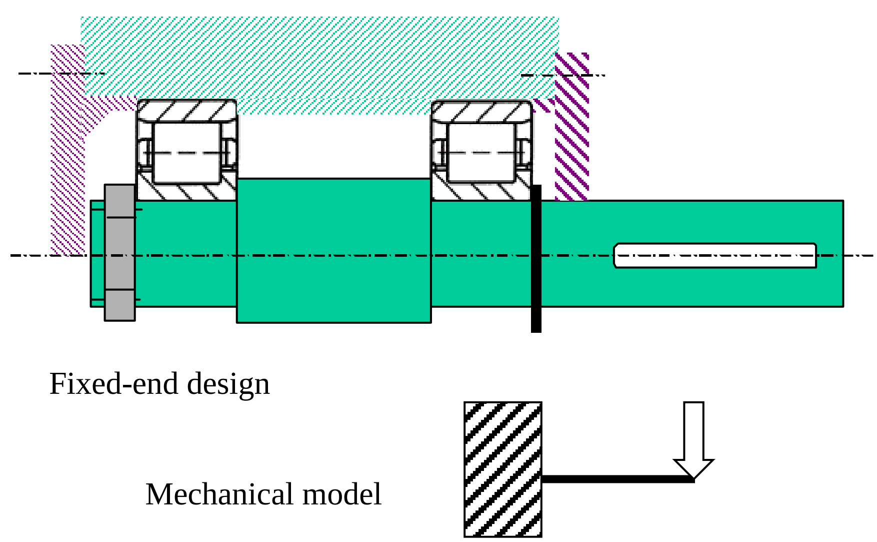


Figure 6-18 Design of the fixed end of a rotating cantilever beam.

Table 6-4 Bearing size and mounting dimension (Hamrock et al 1999)

Principal Dimensions			Basic load ratings		Allowable load limit	Speed ratings		Abutment and fillet Dimensions		
d	d_a	b_w	dynamic C	static C_0	w_{al}	grease	oil	$D_{b\,min}$	$D_{a\,max}$	$r_{a\,max}$
mm	mm	mm	N			rpm		mm	mm	mm
in	in	in	lbf					in	in	in
20	**32**	**7**	**2700**	**1500**	**63**	19000	24000	**22**	**30**	**0.3**
0.7874	1.2598	0.2756	607	337	14.2			0.866	1.181	0.012
	37	**9**	**6370**	**3650**	**156**	18000	22000	**22**	**35**	**0.3**
	1.4567	0.3543	1430	821	35.1			0.866	1.378	0.012
	42	**8**	**6890**	**4050**	**173**	17000	20000	**22**	**40**	**0.3**
	1.6535	0.3150	1550	910	38.9			0.866	1.575	0.012
	42	**12**	**9360**	**5000**	**212**	17000	20000	**24**	**38**	**0.6**
	1.6535	0.4724	2100	1120	47.7			0.945	1.496	0.024
	47	**14**	**12700**	**6550**	**280**	15000	18000	**25**	**42**	**1**
	1.8504	0.5512	2860	1470	62.9			0.984	1.654	0.039
	52	**15**	**15900**	**7800**	**335**	13000	16000	**26.5**	**45.5**	**1**
	2.0472	0.5906	3570	1750	75.3			1.043	1.791	0.039
	72	**19**	**30700**	**15000**	**640**	10000	13000	**26.5**	**65.5**	**1**
	2.8346	0.7480	6900	3370	144			1.043	2.579	0.039
25	**37**	**7**	**4360**	**2600**	**125**	17000	20000	**27**	**35**	**0.3**
0.9843	1.4567	0.2756	980	585	28.1			1.063	1.378	0.012
	47	**12**	**11200**	**6550**	**275**	15000	18000	**29**	**43**	**0.6**
	1.8504	0.4724	2520	1470	61.8			1.142	1.693	0.024
	62	**17**	**22500**	**11600**	**490**	11000	14000	**31.5**	**55.5**	**1**
	2.4409	0.6693	5060	2610	110			1.240	2.185	0.039
	80	**21**	**35800**	**19300**	**815**	9000	11000	**33**	**72**	**1.5**
	3.1496	0.8268	8050	4340	183			1.299	2.835	0.059
30	**42**	**7**	**4490**	**2900**	**146**	15000	18000	**32**	**40**	**0.3**
1.1811	1.6535	0.2756	1010	652	32.8			1.260	1.575	0.012
	55	**13**	**13300**	**8300**	**355**	12000	15000	**35**	**50**	**1**
	2.1654	0.5118	2990	1870	79.8			1.378	1.969	0.039
	72	**19**	**28100**	**16000**	**670**	9000	11000	**36.5**	**65.5**	**1**
	2.8346	0.7480	6320	3600	151			1.437	2.579	0.039
	90	**23**	**43600**	**23600**	**1000**	8500	10000	**38**	**82**	**1.5**
	3.5433	0.9055	9800	5310	225			1.496	3.228	0.059

Principal Dimensions			Basic load ratings		Allowable load limit	Speed ratings		Abutment and fillet Dimensions		
d	d_a	b_w	dynamic C	static C_0	w_{al}	grease	oil	$D_{b\,min}$	$D_{a\,max}$	$r_{a\,max}$
mm	mm	mm	N			rpm		mm	mm	mm
in	in	in	lbf					in	in	in
35	**47**	**7**	**4750**	**3200**	**166**	13000	16000	**37**	**45**	**0.3**
1.3780	1.8504	0.2756	1070	719	37.3			1.457	1.772	0.012
	62	**14**	**15900**	**10200**	**440**	10000	13000	**40**	**57**	**1**
	2.4409	0.5512	3570	2290	98.9			1.575	2.244	0.039
	80	**21**	**33200**	**19000**	**815**	8500	10000	**43**	**72**	**1.5**
	3.1496	0.8268	7460	4270	183			1.693	2.835	0.059
	100	**25**	**55300**	**31000**	**1290**	7000	8500	**43**	**92**	**1.5**
	3.3970	0.9843	12400	6970	290			1.693	3.622	0.059
40	**52**	**7**	**4940**	**3450**	**186**	11000	14000	**42**	**50**	**0.3**
1.5748	2.0472	0.2756	1110	776	41.8			1.654	1.969	0.012
	80	**18**	**30700**	**19000**	**800**	8500	10000	**46.5**	**73.5**	**1**
	3.1496	0.7087	6900	4270	180			1.831	2.894	0.039
	110	**27**	**63700**	**36500**	**1530**	6700	8000	**49**	**101**	**2**
	4.3307	1.0630	14300	8210	344			1.929	3.976	0.079
45	**58**	**7**	**6050**	**4300**	**228**	9500	12000	**47**	**56**	**0.3**
1.7717	2.2835	0.2756	1360	967	51.3			1.850	2.205	0.012
	75	**16**	**20800**	**14600**	**640**	9000	11000	**50**	**70**	**1**
	2.9528	0.6299	4680	3280	144			1.969	2.756	0.039
	120	**29**	**76100**	**45000**	**1900**	6000	7000	**54**	**111**	**2**
	4.7244	1.1417	17100	10100	427			2.126	4.370	0.079
50	**65**	**7**	**6240**	**4750**	**250**	9000	11000	**52**	**63**	0.3
1.9685	2.5591	0.2756	1400	1070	56.2			2.047	2.480	0.012
	80	**16**	**21600**	**16000**	**710**	8500	10000	**55**	**75**	**1**
	3.1496	0.6299	4860	3600	160			2.165	2.953	0.039
	130	**31**	**87100**	**52000**	**2200**	5300	6300	**61**	**119**	**2**
	5.1181	1.2205	19600	11700	495			2.402	4.685	0.079
60	**78**	**10**	**8710**	**6700**	**365**	7500	9000	**62**	**76**	**0.3**
2.3622	3.0709	0.3937	1960	1510	82.1			2.441	2.992	0.012
	95	**18**	**29600**	**23200**	**980**	6700	8000	**66.5**	**88.5**	**1**
	3.7402	0.7087	6650	5220	220			2.618	3.484	0.039
	150	**35**	**108000**	**69500**	**2900**	4800	5600	**71**	**139**	**2**
	5.9055	1.3780	24300	15600	652			2.795	5.472	0.079

Figure 6-19 compares several improper bearing-mounting structures with proper mounting designs. We need to pay attention to the shoulder height and the shaft fillet radius. If the fillet radius is too large, the bearing mounting position on the shaft is not secured because the chamfer of the inner ring of the bearing interferes with the shaft fillet there. The fillet radius should be smaller than the ring chamfer dimension. Similarly, the shaft shoulder should not be too low; otherwise, the shoulder constraint can only be applied to the chamfer of the bearing inner ring, again, the bearing positioning is not secured. On the other hand, the shoulder cannot be too tall either due to the assembly and disassembly considerations, as mentioned before. No matter what, there is no need to make the shoulder that high.

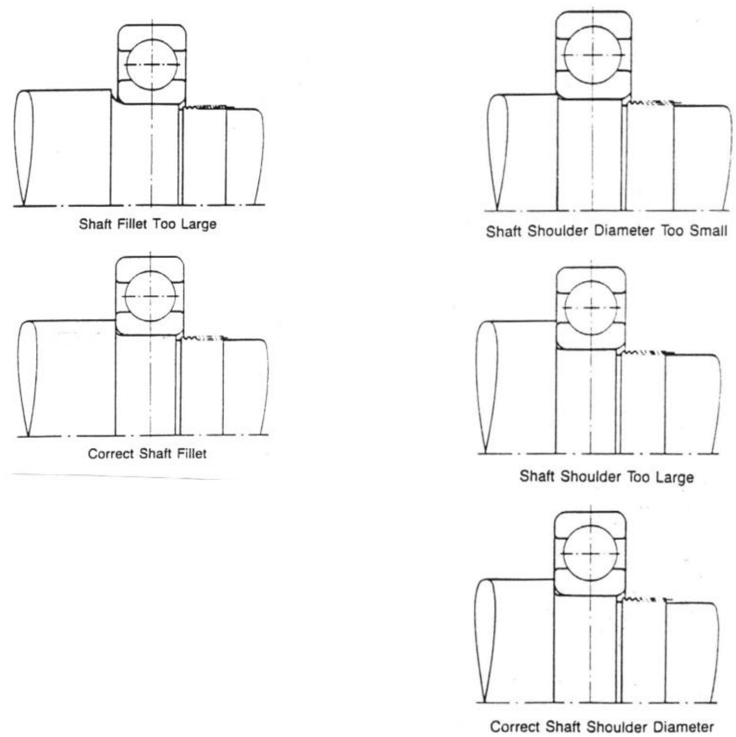


Figure 6-19 **Incorrect and correct bearing mounting (Courtesy of SKF).**

Usually, the bearing mounting structure should be sealed in the housing of the machine, to prevent the lubricant inside the housing from leaking out, and at the same time, to protect the components and the lubricant inside from be contaminated by outside dust.

Let's now summarize the procedure for a bearing design problem.

1. Draft the shaft system design, so that we know the structural dimensions. Then, make the FBD of the shaft.
2. Conduct the force analysis to get bearing reactions.
3. Know what type of bearings to be used: ball or roller, radial or radial-thrust bearings.
4. Calculate the life and bearing load rating, select a proper bearing from a bearing catalog.
5. Check if the selection is proper or not. We may have to iterate between 4 and 5 a couple of times.
6. Refine the structural design of the shaft system.

6-6 Tapered-roller bearing selection

Tapered roller bearings. Tapered roller bearings combine the advantages of ball and roller bearings. Generally speaking, a tapered-roller bearing can take either radial or thrust loads, or their combinations. However, it should be mentioned that compared with the radial-thrust type of the tapered roller bearings (**Figure 6-7**, which is re-plotted in **Figure 6-20**), the thrust bearings, or the thrust-radial bearings, shown in **Figure 6-11**, are superior in taking a pure thrust load. Here we will mainly discuss the radial-thrust tapered roller bearings shown in **Figure 6-20**. Because the inner and outer rings (or the cone and the cup) of these bearings are separable, traditionally, the selection of a tapered roller bearing means the determination of the bearing type, the cone and cup, the bore diameter, the outer diameter, and the bearing width. However, the Timken ISO numbering system uses the cone and cup combinations in its bearing designation. Therefore, we only need to select the bearing with a given cone-cup combination. More information for Timken bearings can be found on the company website, one is given below.
http://www.timken.com/EN-US/PRODUCTS/Pages/Catalogs.aspx.

Selection for

Type

Bore diameter, d
Outer diameter, D
Width, T

Figure 6-20 Tapered-roller bearing for radial-thrust loading (Courtesy of the Timken Company).

Life of tapered-roller bearings. Recall the bearing life-load relationship, which is given in Equation (6-10) with m=10/3 for roller bearings. Here, *C* is the basic load rating.

$$L_1 = \left(\frac{C}{F_1}\right)^{10/3} \quad L_2 = \left(\frac{C}{P}\right)^{10/3} \qquad (6\text{-}10)$$

The Timken tapered-roller bearings have two versions, ISO and traditional tapered roller bearings. The basic load rating for the ISO bearings is $C=C_1$ (or C_r) defined for the life of 1-million cycles of inner-ring revolutions. The basic load rating for Timken's traditional bearings is $C=C_{90}$ corresponding to 3000 hours of life at 500 rpm, or 3000hours(60)(500rpm)=90million cycles of the inner-ring rotations.

The L_{10} life for traditional Timken tapered-roller bearings may also be calculated in service hours based on the bearing rating corresponding to 3000 hours of life at 500 rpm of the inner-ring rotation. The expected life in hours is

$$L_{10} = \left(\frac{C_{90}}{P}\right)^{10/3} \left(\frac{3000(500)}{n}\right) a \qquad \text{(6-11)}$$

rpm

Life adjustment factor
$a \geq 1$
$a = 1$, if environmental conditions are not considered

Timken bearing designation (ISO). We will study one type of the tapered-roller bearings, the ISO 355 bearings. Similar to the bearing designation, the numbering system for the 355 bearings consists of the type (355), the width series, the diameter series, and the bore diameter (again, the designated number times 5 in millimeters). Here, a symbol, M, representing bearing surface heat treatment, may also appear. **Figure 6-21** shows the ISO 355 bearing designation.

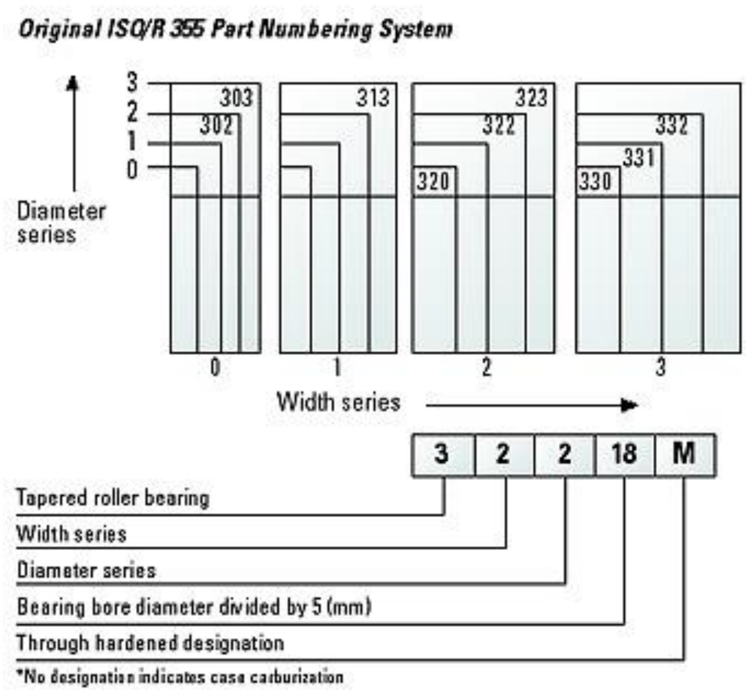


Figure 6-21 Timken ISO tapered-roller bearing system (Courtesy of the Timken Company).

Table 6-5 lists a number of the 355 bearings, where the meanings of *d, D, T, B, C, R*, and *r* are given in **Figure 6-7**. Here, the load rating is C_1, which has the same meaning as *C*. Factors *e* and *Y* may be used for the calculation of the equivalent bearing load. *R* and *r* are round-corner radii.

Table 6-5

Timken Standard IsoClass Bearings

Base Part Number	ISO 355 Reference	Dimensions (mm) d	D	T	B	C	R	r	Load rating C1(N)	Factors e	Y	Weight (kg)
30203	2DB	17	40	13.25	12.0	11.0	1.0	1.0	20 000	0.35	1.74	0.08
30204	2DB	20	47	15.25	14.0	12.0	1.0	1.0	28 300	0.35	1.74	0.13
30205	3CC	25	52	16.25	15.0	13.0	1.0	1.0	31 400	0.37	1.60	0.15
30206	3DB	30	62	17.25	16.0	14.0	1.0	1.0	41 800	0.37	1.60	0.23
30207	3DB	35	72	18.25	17.0	15.0	1.5	1.5	51 100	0.37	1.60	0.32
30208	3DB	40	80	19.75	18.0	16.0	1.5	1.5	60 100	0.37	1.60	0.41
30209	3DB	45	85	20.75	19.0	16.0	1.5	1.5	69 400	0.40	1.48	0.46
30210	3DB	50	90	21.75	20.0	17.0	1.5	1.5	72 100	0.42	1.43	0.52
30211	3DB	55	100	22.75	21.0	18.0	2.0	1.5	90 400	0.40	1.48	0.68
30212	3EB	60	110	23.75	22.0	19.0	2.0	1.5	96 700	0.40	1.48	0.87
30213	3EB	65	120	24.75	23.0	20.0	2.0	1.5	115 200	0.40	1.48	1.10
30214	3EB	70	125	26.25	24.0	21.0	2.0	1.5	126 000	0.42	1.43	1.28
30215	4DB	75	130	27.25	25.0	22.0	2.0	1.5	134 000	0.44	1.38	1.33
30216	3EB	80	140	28.25	26.0	22.0	2.5	2.0	152 000	0.42	1.43	1.59
30217	3EB	85	150	30.50	28.0	24.0	2.5	2.0	176 000	0.42	1.43	2.02
30218	3FB	90	160	32.50	30.0	26.0	2.5	2.0	202 000	0.42	1.43	2.49
30219	3FB	95	170	34.50	32.0	27.0	3.0	2.5	221 000	0.42	1.43	2.95
30220	3FB	100	180	37.00	34.0	29.0	3.0	2.5	252 000	0.42	1.43	3.77
30221	3FB	105	190	39.00	36.0	30.0	3.0	2.5	274 000	0.42	1.43	4.48
30222	3FB	110	200	41.00	38.0	32.0	3.0	2.5	311 000	0.42	1.43	5.19
30224	4FB	120	215	43.50	40.0	34.0	3.0	2.5	331 000	0.44	1.38	6.15
30226	4FB	130	230	43.75	40.0	34.0	4.0	3.0	360 000	0.44	1.38	7.60
30228	4FB	140	250	45.75	42.0	36.0	4.0	3.0	418 000	0.44	1.38	8.65
30230	4GB	150	270	49.00	45.0	38.0	4.0	3.0	469 000	0.44	1.38	11.00

(Courtesy of the Timken Company).

Timken taper-roller bearing selection. The procedure for tapered-roller bearing selection is similar to that for the ball and straight roller bearings, including the calculation of the equivalent load, *P*, from the applied loads, the calculations of the needed bearing rating from *P* and the expected life by means of the life-load relationship, and then the bearing selection and evaluation. The equivalent bearing load, *P*, is given in Equation (6-12), where the radial load factor, *X*, is always 0.4 while the *Y* value is from **Table 6-4**. A_T is called the total axial force, or the total axial load; it includes the externally applied axial load and the induced axial force because the tapered rollers make angular contacts with the bearing race. An axial force is induced in the bearing even by the application of a pure radial load.

$$P = A_p(XF_r + YA_T) \tag{6-12}$$

Total axial force on the bearing
Axial force factor, bearing catalog
Radial force on the bearing
Radial force factor, X= 0.4
Application factor

In Eq. (6-12), the application factor, A_p, can be ignored for smooth loading cases.

It is important to know how the induced axial force, F_a, and the total bearing axial force A are determined. **Figure 6-22** shows a bearing on a shaft (right) and the free-body diagram of the shaft with the bearing inner ring (left). This figure illustrates the mechanism of the induced axial force. When a radial load, R, is applied onto the bearing, a force pair, F_r and F_a on the inner ring, must be produced by the bearing rollers in order to balance R. Obviously, $R = F_r$, and they are self-balanced. F_a is the induced (not the applied) axial force, which is on the inner ring of the bearing, and should be calculated from Equation (6-13) given by the bearing industry, and here, the Timken Company.

$$F_a = \frac{\lambda F_r}{Y} \tag{6-13}$$

where, λ is a coefficient determined by the bearing structure, and λ = 0.47 for Timken's traditional bearings and λ = 0.5 for Timken's ISO bearings.

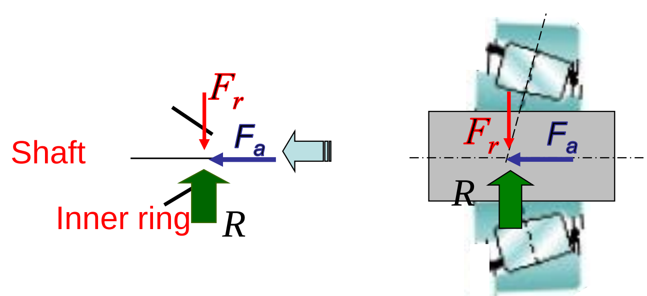


Figure 6-22 Bearing on a shaft (right, the shoulder is ignored for simplicity), and the free-body diagram of the shaft with the bearing inner ring (left). Here, F_a is the induced axial force. The wedge opening direction in the FBD must be the same as that in the bearing.

Note that, 1) in **Figure 6-22**, the location of the force application is no longer at the center of the bearing width, and 2) in Equation (6-12), A_T is the total axial force on the bearing, not simply the axial reaction force, or the induced axial force, F_a, on this bearing; it will be analyzed below together with bearing arrangements.

In the following, the application factor, A_p, is ignored for clarity.

Face-to-face arrangement of two bearings. Because of the asymmetry of the structure of the tapered roller bearings and the feature of inner-outer ring separation, two typical bearing arrangements, face-to-face and back-to-back, are usually used. **Figure 6-23** shows two bearings in a face-to-face arrangement subjected to external loading. The forces in the radial direction are self-balanced, but the bearing axial forces are unknown. The induced axial forces, F_{aM}, F_{aN}, at bearings M and N, are marked; however, they do not construct the equilibrium with the applied axial load, W_{ae}. Here is the list of what we know.

Applied forces, here *e* means external to bearings. Note here, we do not use the superscripts that we used for gear forces because the externally applied load may come from other elements.

> Applied radial force: W_{re}
> Applied axial force: W_{ae}

Reaction forces at bearings M and N: R_M, R_N

The bearings should produce:

> Radial forces on the bearing inner rings by the outer rings: $F_{rM} = R_M$, $F_{rN} = R_N$
>
> Induced bearing axial forces (on the inner rings): $F_{aM} = \dfrac{\lambda F_{rM}}{Y_M}$, $F_{aN} = \dfrac{\lambda F_{rN}}{Y_N}$

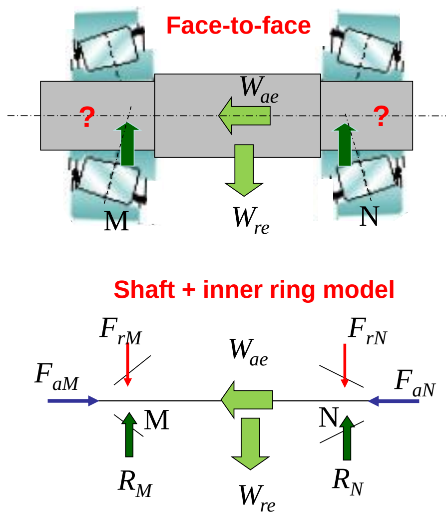


Figure 6-23 Two tapered roller bearings in a face-to-face arrangement (top) and the shaft + inner ring mechanical model (bottom). The forces in the radial direction are self-balanced, but the bearing axial forces are unknown. We only know the induced axial forces, F_{aM}, F_{aN}, at bearings M and N. Note that F_{rM}, F_{rN} are equivalent to the applied load on the bearings.

Let's consider the bearings one by one.

Bearing M

In order to see what bearing M is taking in the axial direction, we need to "remove" the bearing and replace it with an axial force. This is what we do to create a FBD. Consider the FBD of the shaft and the inner ring of bearing N, shown in **Figure 6-24**, where W_{ae} and F_{aN} are external to bearing M. The induced axial force, F_{aM}, which is internal to bearing M, is "removed" with the bearing; instead, all the axial force at M is now presented by A_{TM}, the resultant axial force on bearing M, including the effect of F_{aM}. The direction of A_{TM} is along the direction of the induced axial force in the same bearing, M.

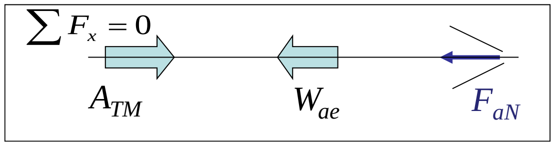


Figure 6-24 Analysis of the total axial force, A_{TM}, for bearing M, face to face.

The total axial force, A_{TM}, on bearing M, should balance the combined action of the applied and induced axial forces:

$$A_{TM} = F_{aN} + W_{ae} = \frac{\lambda F_{rN}}{Y_N} + W_{ae} \tag{6-14}$$

The total axial load, A_{TM}, is then substituted into the axial force term in Equation (6-12). The equivalent force, P_M, on bearing M can be determined by the larger of the radial reaction, F_{rM}, or the result of Equation (6-14), and this comparison is shown in Equation (6-15). This method avoids the use of e in **Table 6-4** for the equivalent force calculation.

The equivalent load on bearing M, P_M, is the larger of the two, expressed as follows:

$$P_M = \max\begin{cases} XF_{rM} + YA_{TM} = XF_{rM} + Y_M\left(\dfrac{\lambda F_{rN}}{Y_N} + W_{ae}\right) \\ F_{rM} \end{cases} \tag{6-15}$$

Bearing N

Likewise, consider the FBD of the shaft and the inner ring of bearing M, **Figure 6-25**, where F_{ae} and F_{aM} are external to bearing N. Therefore, A_{TN}, is the resultant axial force on bearing N, including the effect of F_{aN}. The induced axial force to bearing N. The direction of A_{TN}, is along the direction of the induced axial force in the same bearing, N.

The total axial force, A_{TN}, on bearing N is

$$A_{TN} = F_{aM} - F_{ae} = \frac{\lambda F_{rM}}{Y_M} - W_{ae} \tag{6-16}$$

Similarly, the equivalent force, P_N, on bearing N, can be determined from the following.

$$P_N = \max \begin{cases} XF_{rN} + YA_{TN} = XF_{rN} + Y_N\left(\dfrac{\lambda F_{rM}}{Y_M} - W_{ae}\right) \\ F_{rN} \end{cases} \qquad (6\text{-}17)$$

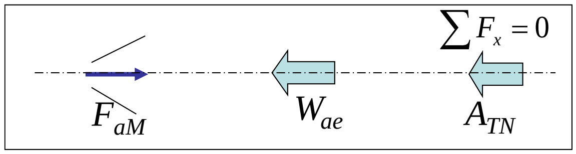


Figure 6-25 Analysis of the total axial force, A_{TN}, for bearing N, face to face.

Back-to-back arrangement of two bearings. Figure 6-26 shows that two bearings are in a back-to-back arrangement. Again, we list what we know first.

Applied forces, here e means external:

Applied radial force: W_{re}
Applied axial force: W_{ae}

Reaction forces at bearings M and N: R_M, R_N

The bearings should produce:

Radial forces on the bearing inner rings by the outer rings: $F_{rM} = R_M$, $F_{rN} = R_N$

Induced bearing axial forces (on the inner rings): $F_{aM} = \dfrac{\lambda F_{rM}}{Y_M}$, $F_{aN} = \dfrac{\lambda F_{rN}}{Y_N}$

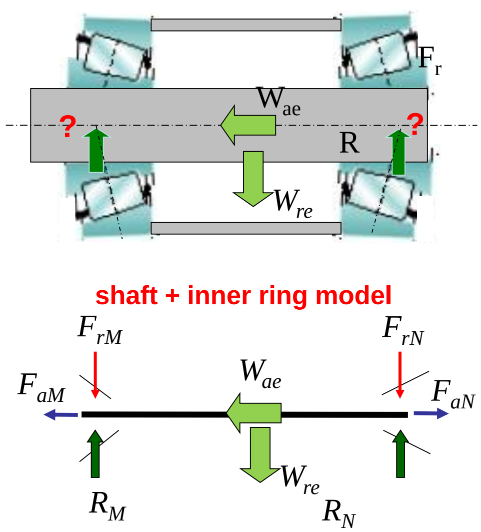


Figure 6-26. Two bearings in a back-to-back arrangement (top), and the shaft + inner ring mechanical model (bottom).

Bearing M

Similar to what is done for the face-to-face arrangement, consider the FBD of the shaft and the inner ring of bearing N, **Figure 6-27**, where W_{ae} and F_{aN} are external to bearing M. Therefore, A_{TM} is the resultant axial force on bearing M, including the effect of F_{aM}. The direction of A_{TM} is along the direction of the induced axial force in the same bearing, M.

The total axial force, A_{TM}, on bearing M is

$$A_{TM} = F_{aN} - W_{ae} = \frac{\lambda F_{rN}}{Y_N} - W_{ae} \tag{6-18}$$

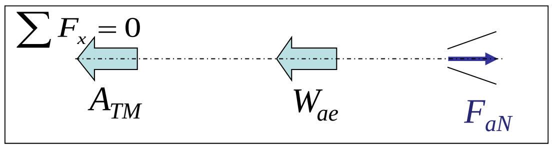


Figure 6-27 Analysis of the total axial force, A_{TM}, for bearing M, back to back.

The equivalent force, P_M, on bearing M is the larger from the equation group below.

$$P_M = \max \begin{cases} XF_{rM} + YA_{TM} = XF_{rM} + Y_M\left(\dfrac{\lambda F_{rN}}{Y_N} - W_{ae}\right) \\ F_{rM} \end{cases} \tag{6-19}$$

Bearing N

Likewise, consider the FBD of the shaft and the inner ring of bearing M, **Figure 6-28**, where W_{ae} and F_{aM} are external to bearing N. Therefore, A_{TN} is the resultant axial force on bearing N, including the effect of F_{aN}. The direction of A_{TN} is along the direction of the induced axial force in the same bearing, N.

The total axial force, A_N, on bearing N is

$$A_N = F_{aM} + W_{ae} = \frac{\lambda F_{rM}}{Y_M} + W_{ae} \tag{6-20}$$

The equivalent force, P_N, on bearing N is the larger of the following two.

$$P_N = \max \begin{cases} XF_{rN} + YA_N = XF_{rN} + Y_N\left(\dfrac{\lambda F_{rM}}{Y_M} + W_{ae}\right) \\ F_{rN} \end{cases} \tag{6-21}$$

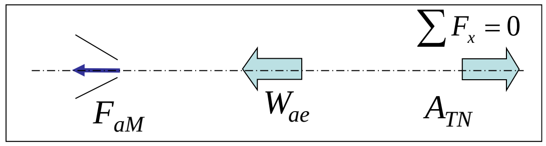


Figure 6-28 Analysis of the total axial force, A_N, for bearing N, back to back mounting.

Example 6-2 Tapered-roller bearing selection

Select bearings A and B, **Figure 6-29**, to support the shaft with a helical gear. The shaft speed is 150 rpm and the L_{10} life should be at least 90,000 hours. The loads and shaft span information are:

$W^a = 1100\text{N},\ W^r = 2300\text{N},\ W^t = 6200\text{N}$

$L_1 = 100\text{mm}$, $L_2 = 140\text{mm}$ (from the points of load application)

The gear pitch circle radius is $r = 50\text{mm}$

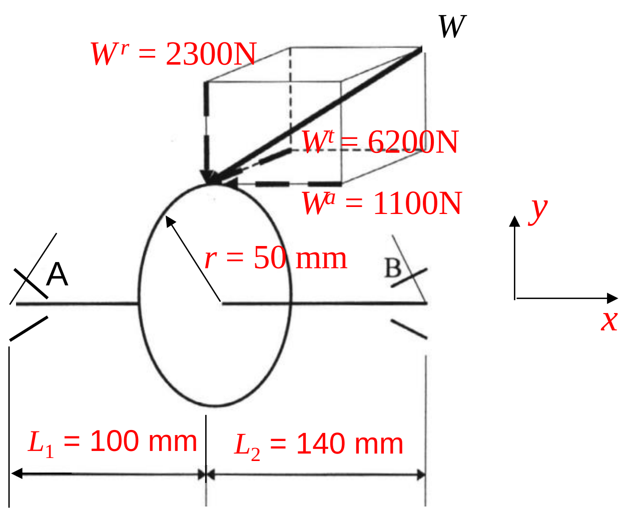


Figure 6-29. Diagram for **Example 6-2**.

Solution

(1) Radial forces on the bearings, $A_r = (A_y, A_z)$, $B_r = (B_y, B_z)$

In the x-y plane: $\sum M_A = 0$, $W^a r - W^r l_1 + B_y(l_1 + l_2) = 0$

$$B_y = \frac{-W^a r + W^r l_1}{l_1 + l_2} = \frac{-1100(50) + 2300(100)}{100 + 140} = 729N$$

$$A_y = W_r - B_y = 2300 - 729 = 1571N$$

In the x-z plane: $B_z = \dfrac{W^r l_1}{l_1 + l_2} = \dfrac{6200(100)}{100+140} = 2583N$

$$A_z = W^t - B_z = 6200 - 2583 = 3846N$$

Radial force on A: $F_{rA} = A_r = \sqrt{A_y^2 + A_z^2} = \sqrt{1571^2 + 3846^2} = 4154N$

Radial force on B: $F_{rB} = B_r = \sqrt{B_y^2 + B_z^2} = \sqrt{729^2 + 2583^2} = 2684N$

(2) Back-to-Back support

Using equations (6-19) and (6-21)

(3) Calculations for the selection

Bearing A

For initial bearing selection, let's use $Y_A = Y_B = 1.48$ as our initial guess.

$$P_A = XF_{rA} + Y_A(\frac{\lambda F_{rB}}{Y_B} - W^a) = 0.4(4154) + 1.48(\frac{0.5(2684)}{1.48} - 1100) = 1376N$$

Or $P_A = F_{rA} = 4154N$, whichever the lager.

Therefore, the equivalent load is $P_A = F_{rA} = 4154N$

Life in millions of cycles is $L_{10} = h(60)n/10^6 = 90{,}000(60)(150)/10^6 = 810$

Bearing rating for bearing A, calculated $C_1 = C_{1A}$

$$C_{1A} = P_A L^{1/m} = F_{rA} L^{1/m} = 4154(810)^{3/10} = 30975N$$

Selection for Bearing A is 30205: C_1 = 31,400N, Y = 1.6

Bearing B

Now, $Y_A = 1.6$

$$P_B = XF_{rB} + Y_B(\frac{\lambda F_{rB}}{Y_A} + W^a) = 0.4(2684) + 1.48(\frac{0.5(4154)}{1.6} + 1100) = 4623N$$

Or $P_B = F_{rB} = 2684N$, whichever the lager.

Therefore, the equivalent load is $P_B = 4623N$

The life is 810 million cycles.

The bearing rating $C_1 = C_{1B}$ is $C_{1B} = P_B L^{1/m} = 4623(810)^{3/10} = 34,472N$

Selection for Bearing B is 30206: C_1 = 41,800N, $Y = 1.6$

(4) Check. We need to examine our results because now we have Y for both bearings.

Now, let's use $Y_A = Y_B = 1.6$

Bearing A

$$P_A = \max \begin{cases} XF_{rA} + Y_A(\dfrac{\lambda F_{rB}}{Y_B} - W^a) = \\ = 0.4(4154) + 1.6(\dfrac{0.5(2684)}{1.6} - 1100) = 1244N \\ \\ F_{rA} = 4154N \end{cases}$$

The same equivalent load applies. $P_A = F_{rA} = 4154N$.

and $C_{1A} = P_A L^{1/m} = F_{rA} L^{1/m} = 4154(810)^{3/10} = 30975N$.

No change is made to C_1 and no need to do the selection again, and Bearing A is Timken ISO 355 30205: C_1 = 31,400N, Y = 1.6.

Bearing B

Now, let's use $Y_A = Y_B = 1.6$

$$P_B = \max \begin{cases} XF_{rB} + Y_B(\dfrac{\lambda F_{rA}}{Y_A} + W^a) = \\ = 0.4(2684) + 1.6(\dfrac{0.5(4154)}{1.6} + 1100) = 4911N \\ \\ F_{rB} = 2684N \end{cases}$$

A New equivalent load should apply, $P_B = 4911N$. The bearing rating is now

$$C_{1B} = P_B L^{1/m} = 4911(810)^{3/10} = 36620N .$$

We see a new load rating, but it does not affect the bearing selection, which is still Timken ISO 355 30206 with C_1 = 41,800N and Y = 1.6, good for bearing B.

Do we really have to choose two different bearings? No, not necessary. We can use 30306 for both. What are the advantages of using the same bearings? As usual, think about manufacturing (this time, not the bearing, but the supporting bores), assembly, and maintenance.

6-7 More on bearing mounting design

Mounting face-to-face tapered roller bearings is similar to what we have seen for ball bearing mounting. **Figure 6-30** shows the case of a shaft supported by two face-to-face tapered roller bearings. As revealed by the top figure, the axial locations of the bearings should be confined, and cup separation should be prevented. The bottom figure shows a possible structure, where the shaft shoulders are designed for bearing positioning and the edges of the end-cap rings are used to prevent the bearing cup from sliding out. The shoulders and the end-cap rings also constrain the lateral (axial) motions of the bearings. Therefore, an axial load in either direction on the shaft is securely transmitted to the housing by means of the bolts (indicated by the centerlines) that connect the end caps to the housing.

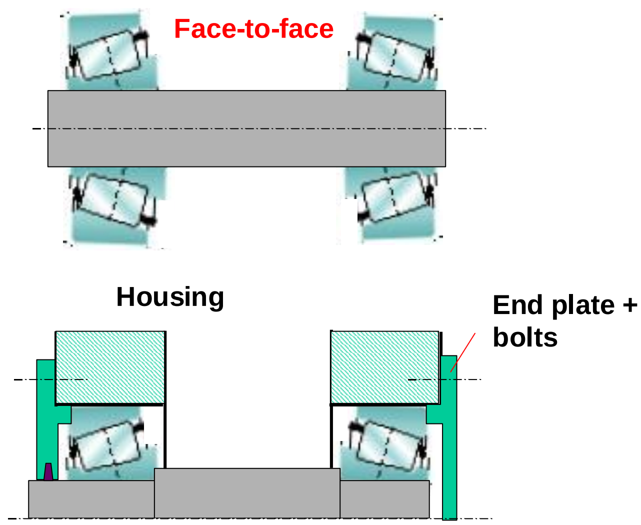


Figure 6-30. Face-to-face bearings (top) and a mounting structure (bottom).

Mounting the back-to-back tapered roller bearings is slightly more complicated, which is, however, similar to mounting straight-roller bearings. The top plot of **Figure 6-31** shows two back-to-back bearings on a shaft. Similarly, we need to confine their locations along the shaft and provide means for axial-force transmission. Like what we have done for two straight-roller bearings, both the inner and outer rings of the bearings need to be fixed in order to prevent ring separation and secure load transmission to the housing. The inner rings (cones) are positioned and confined by a shaft shoulder and an end plate connected to the shaft by a bolt. Unlike the straight-roller bearings, there is no need to confine both front faces of the outer rings (smaller ends of the cups) here because the outer rings can only separate along the wedge slope. The bottom plot of **Figure 6-31** shows a suggested design. An end cap is still used, but that is for the purpose of enclosure. There must be a clearance between the end face of the shaft and the end plate, and there must be a clearance between the surfaces of the bolt head and the end cap. Why? Note that the uses of these two clearances are different.

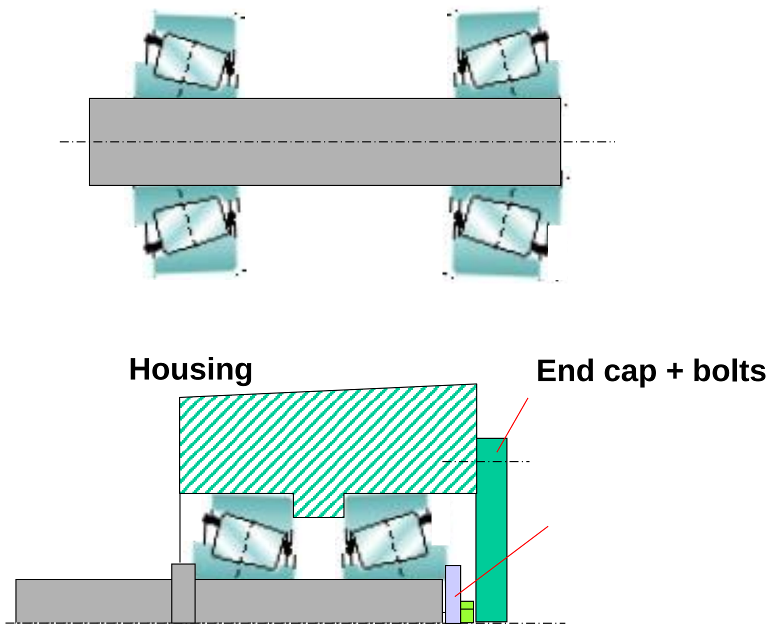
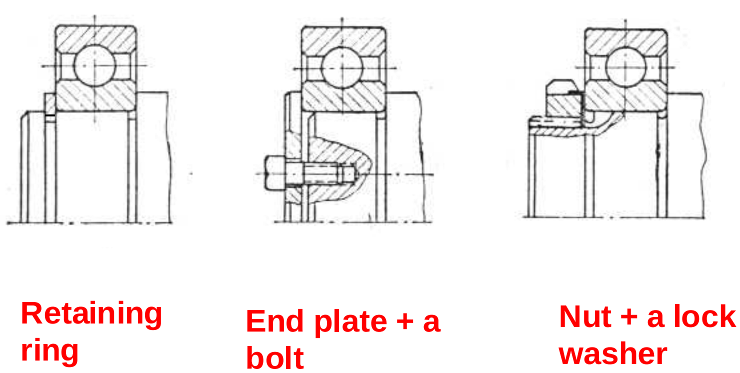


Figure 6-31 Back-to-back bearings (top) and mounting (bottom).

So far, we have seen three methods for inner ring mounting: by a snap ring (**Figure 6-32** (a)), by an end plate and a bolt (**Figure 6-32** (b)), and by a nut (**Figure 6-32** (c), where a lock washer is used to prevent the nut from loosening). Each of these structures has its own pros and cons and should be used based on detailed considerations of loading, structural strength, and locking security.



(a) (b) (c)

Figure 6-32 Methods for inner ring mounting (Pu et al).

Now, we are ready to study more complicated structures. **Figures 6-33** and **6-34** are two designs for a bevel gear shaft. We know that one of the shafts of a bevel-gear pair has to be a mechanical cantilever beam; that is, the support should balance the applied moment while permitting rotation. Both structures in these two figures work. However, they are different in several things, as we can see. We should be able to tell how the bearings are mounted and how the forces, as well as the moment, are balanced. Think about these structures in the following aspects: functions of each element, forces on the bearings, manufacturing-assembly cost/convenience, and pros and cons in assembly and disassembly, and in maintenance.

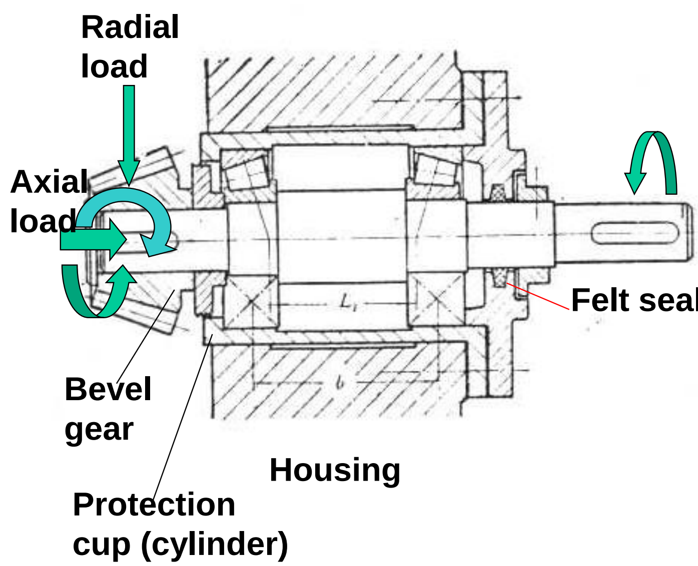


Figure 6-33 Face-to-face support of a bevel gear shaft (modified from Pu et al).

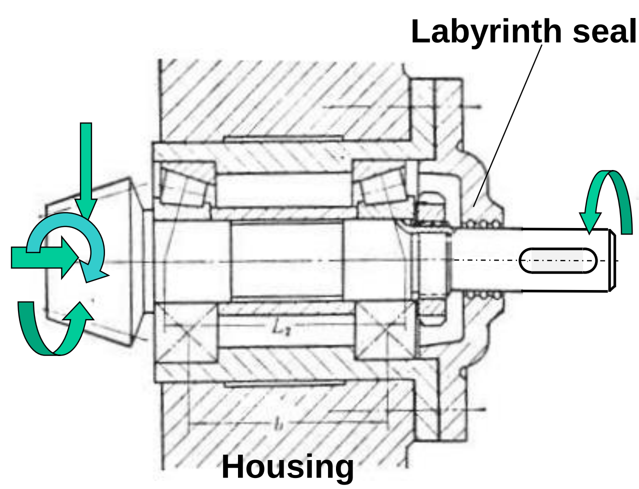


Figure 6-34 Back-to-back support of a bevel gear shaft (modified from Pu et al.).

6-8 Lubrication of rolling element bearings (gears as well)

Bearings are usually lubricated when they are in use, either by enclosed grease or by a lubricant supplied on site. Gears, too, need lubrication. The purposes of lubrication are, at least, to separate contacting surfaces with a thin fluid film and to transmit load through this film. This fluid layer represents a magic mechanism for load supporting, lubrication, and failure prevention. It is very thin, in microns, or even in nanometers, but capable of supporting a very high pressure, which may be as high as several GPa. Here are a few more functions of the lubricating film:

- Reducing friction and vibration
- Reducing peak stress
- Transferring heat
- Transporting wear debris
- Protecting surfaces from corrosion
- Improving bearing life

The lubrication of rolling-element bearings and gears is in the category of the Elasto-hydrodynamic lubrication, or EHL for short. Here, Elasto is for the elasticity of materials. H. M. Martin studied gear lubrication in 1916 by means of rollers, which we know were representations of tooth bodies at contact based on the radii of curvature there. Guess what he found? The predicted film thickness by means of the rigid-cylinder lubrication theory was much thinner than the measurement of surface roughness of the best-machined gear surfaces (Dowson 1998). If it were true, the gears tooth under such film thickness should have been worn out rapidly. However, in Martin's experiments, these gears worked fine, and the machining marks were still visible on the roller surfaces after they were tested. This observation suggested that the real film thickness was thicker than what was predicted by the rigid-cylinder lubrication model.

Note that the cylinder-cylinder, cylinder-flat, sphere-sphere, or sphere-flat contact forms a high pair. We know that when two elements make a high pair, they are in a concentrated contact over a small contact area (such as the contact of gear teeth and rollers with bearing races, where the two surfaces are convex, or if one is concave, its radius is much larger than that of the convex one), the contact pressure is highly localized, causing surfaces to deform. Such deformation opens the gap, or increases the film thickness. Elastic deformations of surfaces contribute to the lubricating film thickness.

6-9 Introduction to sliding bearings, bearing comparison

We may now proceed to the study of the other family of bearings, the sliding bearings, or the fluid-film bearings. We know that a journal bearing (for radial-load support) consists of a shaft (journal), a bearing (sleeve, bushing), and a fluid (oil, grease, or even gas/air). The elements of a fluid-film bearing form a low pair; their contact is highly conforming, meaning that the radii of the inner surface of the bearing and the outer surface of the journal are nominally the same, and the clearance between them is from the tolerance. Here, a continuous lubricant film is desired, and a secured lubricant supply should be guaranteed. **Figure 6-35** shows the axial cross section and front views of such a bearing and note that the gap (or the film thickness) is exaggerated in the right plot for clarity.

Table 6-6 compares the rolling element bearings with different types of sliding bearings. In this table, three types of sliding bearings are mentioned: bearings with a full-film lubrication (the lubricant film can completely separate the two surfaces), those with a mixed lubrication (the lubricant film is broken by surface contact spots), and hydrostatic bearings using pressurized lubricant by a supply system. The full-film lubrication bearings and the hydrostatic bearings are designed with a sufficiently thick lubricant film to separate the bearing surfaces; however, the mixed-lubrication film in a bearing is insufficient to do complete surface separation. Surface asperity contacts co-exist with lubrication. These will be discussed in the next chapter.

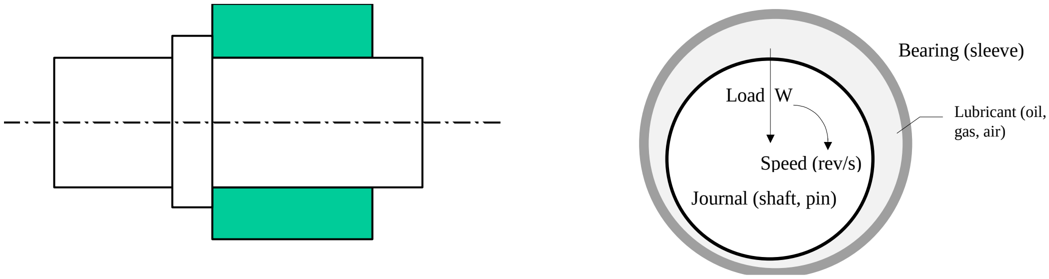


Figure 6-35 Structure (left) and interface of a fluid-film bearing.

Table 6-6 Bearing comparison

Performance	Rolling bearings	Sliding bearings		
		with mixed lubrication	with full-film lubrication	Hydrostatic bearings
Load (W)-speed (N) relation	N/A	$W\Downarrow$ as $N\Uparrow$	$W\Uparrow$ as $N\Uparrow$	N/A
Structural flexibility	Not flexible	Very flexible	Very flexible	Can be flexible
Shock buffering ability	low	low	high	high
High-speed operation	fair	poor	good	good
Starting resistance	low	high	May be high	low
Power loss	may be low	high	low	may be low
Life	fair	low	can by long	can be very long
Noise	high	low	can be very low	can be very low
Stiffness	high	fair	fair	can be very high
Rotation accuracy	high	low	can be high	very high
Radial dimension	large	small	small	small
Axial dimension	(0.2-0.5)d	(0.5-4)d	(0.5-0.4)d	fair
Lubricant	oil or grease	oil, grease, solid	oil, gas	oil, gas
Lubricant consumption	very low	low	high	very high
Maintenance	easy	change sleeves, shaft re-polishing	change sleeves, shaft re-polishing	change sleeves, shaft re-polishing
Cost	fair	may be low	high	may be very high

Chapter Summary. This chapter studies rolling element bearings, their structures, principles for bearing selection calculations, and bearing mounting. The concepts of L_{10} life, bearing equivalent load, and induced axial force are introduced. Typical bearing support designs are explained, and common design mistakes are discussed. This chapter also briefly introduced the concepts of lubrication and sliding bearings as preparation for the next chapter.

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Pu, L., Chen, D., and Wu, L., 2013, *Machine Design*, China Higher Education Press.

Industrial catalog data sets.

Media: 315 students should watch the short U-Tube movies, if any posted, before the class.