MECH 344/M
Machine Element Design

Time: M _ _ _ _14:45 - 17:30

Lecture 12

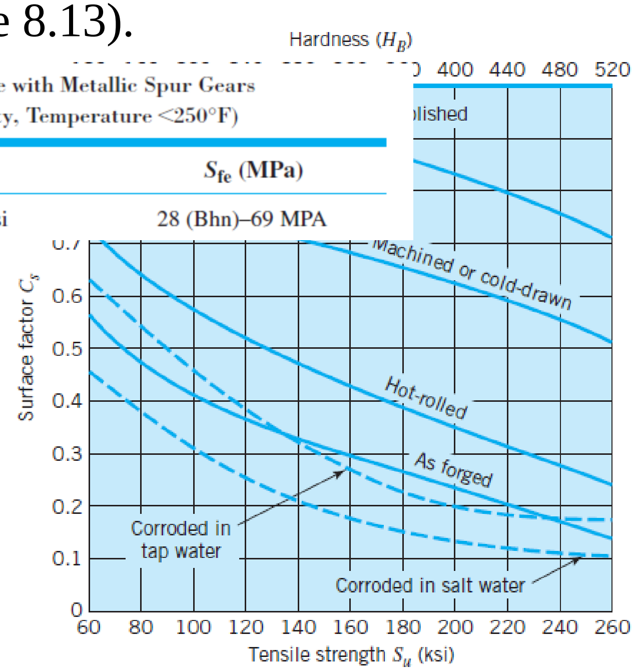
15.11 Spur Gear Design Procedures

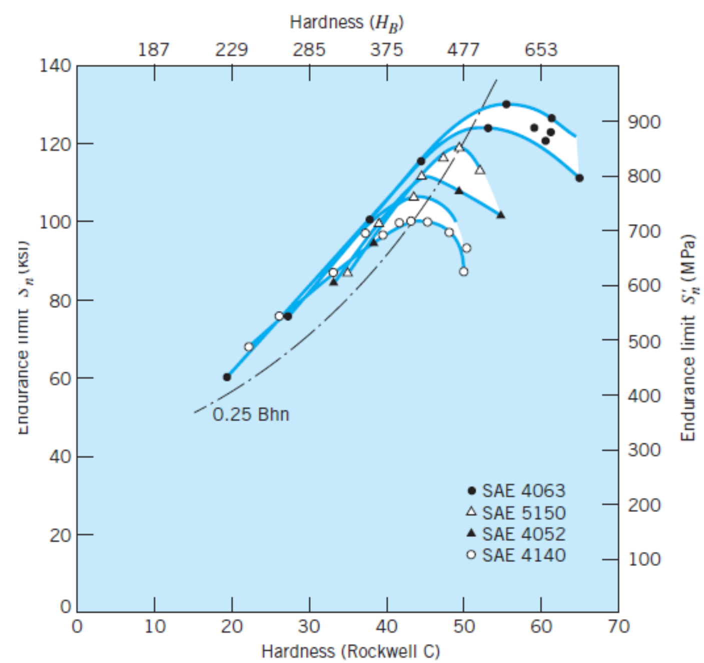
- Sample Problems so far shows the analysis of estimated capacity of a given pair of gears. As is generally the case with machine components, it is a more challenging task to design a suitable pair of gears for a given application. Few general observations.

1. Increasing the surface hardness of steel gears pays off handsomely in terms of surface endurance. Table 15.5 indicates that doubling the hardness more than doubles surface fatigue strength (allowable Hertz stress); Eq. 15.24 shows that doubling the allowable Hertz stress quadruples the load capacity F_t.
2. Increases in steel hardness also increase bending fatigue strength, but the increase is far less. For example, doubling the hardness will likely not double the basic endurance limit, (flattening of curves in Figure 8.6). Furthermore, doubling hardness substantially reduces C_S (see Figure 8.13).

Table 15.5 Surface Fatigue Strength S_{fe}, for Use with Metallic Spur Gears (10^7-Cycle Life, 99 Percent Reliability, Temperature <250°F)

Material	S_{fe} (ksi)	S_{fe} (MPa)
Steel	0.4 (Bhn)–10 ksi	28 (Bhn)–69 MPA





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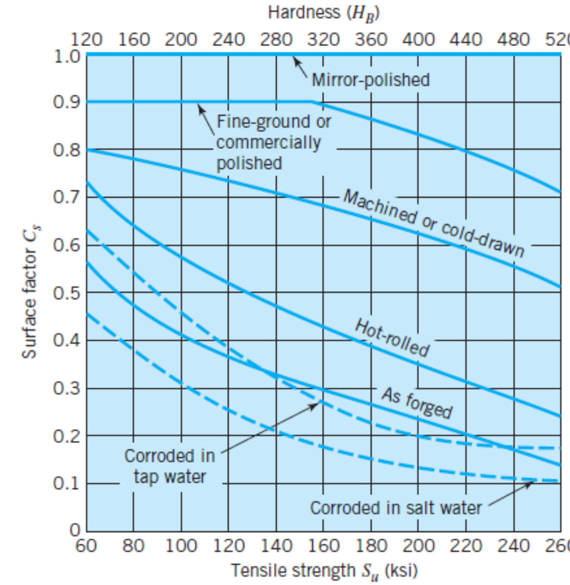

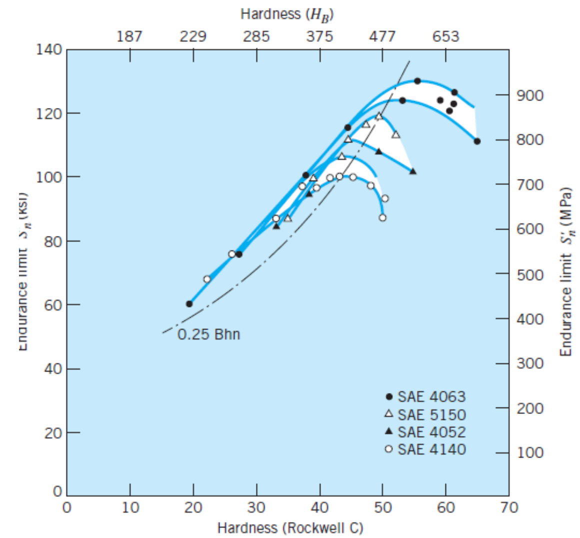

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$$\sigma_H = C_p \sqrt{\frac{F_t}{b d_p I} K_v K_o K_m}$$

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3. Increasing tooth size (using a coarser pitch) increases bending strength more than surface strength. This fact, together with points 1 and 2, correlates with two observations. (a) A balance between bending and surface strengths occurs typically in the region of P = 8 for high-hardness steel gears (above about 500 Bhn, or 50RC), with coarser teeth failing in surface fatigue and finer teeth failing in bending fatigue. (b) With progressively softer steel teeth, surface fatigue becomes critical at increasingly fine pitches. Other materials have properties giving different gear-tooth strength characteristics.

4. In general, the harder the gears, the more costly they are to manufacture. On the other hand, harder gears can be smaller and still do the same job. And if the gears are smaller, the housing and other associated parts may also be smaller and lighter. Furthermore, if the gears are smaller, pitch line velocities are lower, and this reduces the dynamic loading and rubbing velocities. Thus, overall cost can often be reduced by using harder gears.

5. If minimum-size gears are desired (for any given gear materials and application), it is best in general to start by choosing the minimum acceptable number of teeth for the pinion (usually 18 teeth for 20º pinions, and 12 teeth for 25º pinions), and then solving for the pitch (or module) required.

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Sample Problem 15.5D Design of a Single Reduction Spur Gear Train

Using a standard gear system, design a pair of spur gears to connect a 100-hp, 3600-rpm motor to a 900-rpm load shaft. Shock loading from the motor and driven machine is negligible. The center distance is to be as small as reasonably possible. A life of 5 years of 2000 hours/year operation is desired, but full power will be transmitted only about 10 percent of the time, with half power the other 90 percent. Likelihood of failure during the 5 years should not exceed 10 percent.

Solution

Known: A spur gear pair is to transmit power from a motor of known horsepower and speed to a driven machine shaft rotating at 900 rpm. Full power is transmitted 10 percent of the time, half power the other 90 percent. The likelihood of failure should not exceed 10 percent when the gears are operated at 2000 hours/year for 5 years. Center distance is to be as small as reasonably possible. (See Figure 15.28.)

Find: Determine the geometry of the gearset.

Schematic and Given Data:

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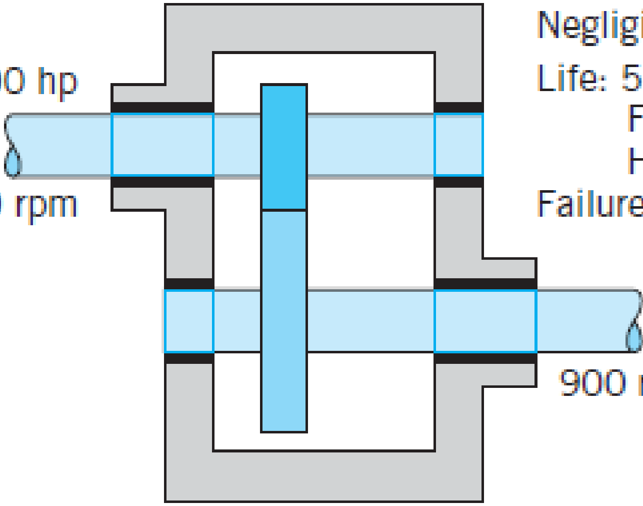

FIGURE 15.28
Single-reduction spur gear train.

Decisions:

1. Choose hardened-steel gears corresponding to the spur gear curve in Figure 9.21, which shows a 10 percent probability of failure. Steel gear material will be selected to provide relatively high strength at relatively low cost. The pinion and gear will be machined and then ground. In accordance with good practice, specify a case-hardening procedure that will leave compressive residual stresses in the gear-tooth surfaces.
2. Specify high surface hardness of 660 Bhn and 600 Bhn, respectively, for pinion and gear to obtain the minimum center distance and the pinion-tooth hardness that will exceed the gear-tooth hardness by 10 percent.
3. For these hardnesses (which are too hard for normal machining), specify a ground finish and precision manufacture corresponding to the average of curves *A* and *B* in Figure 15.24.

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4. Choose the more common 20° full-depth involute tooth form.
5. Choose 18 teeth, the minimum number of pinion teeth possible to avoid interference.
6. For minimum center distance (i.e., minimum gear diameters), tentatively choose width b at the maximum of the normal range, $14/P$.
7. Choose a safety factor of 1.25 for failure by surface fatigue.
8. A nominal value for face width will be used.
9. A standard diametral pitch will be selected.

Assumptions:

1. The Palmgren–Miner cumulative-damage rule applies.
2. The ground-surface finish will correspond to the average of curves A and B in Figure 15.24, and $K_v = 1.4$.
3. The characteristics of support are accurate mountings, small bearing clearances, minimum deflection, and precision gears.
4. The spur gear curve in Figure 9.21 represents about the highest contact strength that is obtainable for steel gears, and this curve is a plot of $S_H = S_{fe}C_{Li}C_R$ for a 10 percent probability of failure versus the number of cycles constituting the life of the spur gear.

5. There is no load sharing between gear teeth.
6. In the limiting case, the fatigue strength of the core material must be equal to the bending fatigue stresses at the surface. Under the surface C_s is 1.
7. For the steel core material, $S'_n = 250$ (Bhn).

Design Analysis:

1. Total life required = 3600 rev/min × 60 min/h × 2000 h/yr × 5 yr = 2.16×10^9 revolutions of the pinion. Only 2.16×10^8 cycles are at full power. Looking at the spur gear curve in Figure 9.21, we note that if the stresses for 2×10^8 cycles of full power are on the curve, stresses for 50 percent power would correspond to over 10^{10}-cycle life. Considering the Palmgren–Miner cumulative-damage rule (Section 8.12), and recognizing the approximate nature of our solution, we appear justified in designing for the full-load cycles only and in ignoring the half-load cycles.

2. Anticipating that surface fatigue will likely be more critical than bending fatigue, we solve for the value of P that will balance σ_H and S_H with a small safety factor, SF, of say 1.25:

$$\sigma_H \text{ (from Eq. 15.24)} = S_H \text{ (from Eq. 15.25)}$$

$$C_p\sqrt{\frac{F_t(SF)}{bd_pI}K_vK_oK_m} = S_{\mathrm{fe}}C_{\mathrm{Li}}C_R$$

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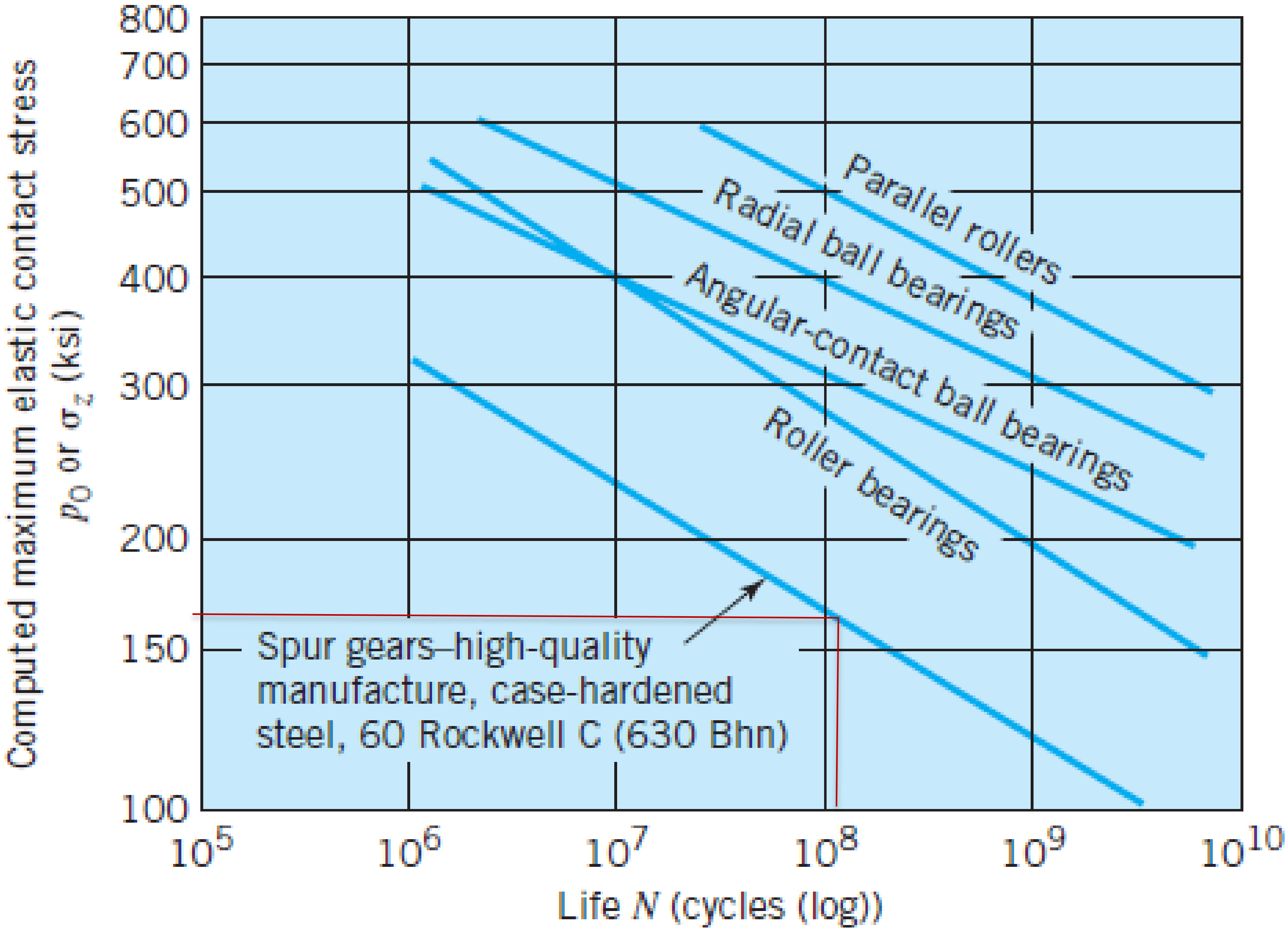


A few auxiliary calculations are required:

$$V = \pi d_p(3600 \text{ rpm})/12 = 942 d_p = 942(18/P) = 16{,}960/P$$

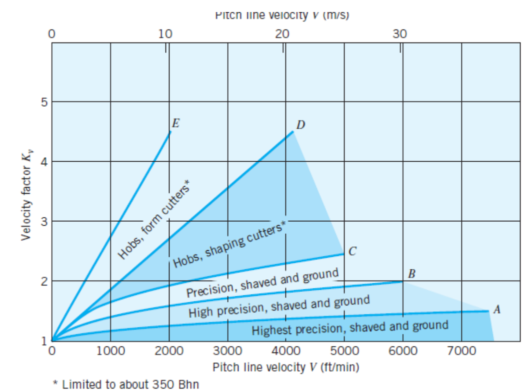
$$K_v \approx 1.4 \text{ (This value is a rough estimate from Figure 15.24, and must be confirmed or modified after } P \text{ is determined.)}$$

$$K_m = 1.3 \text{ (This value must be increased if } b > 2 \text{ in.)}$$

$$F_t = 100 \text{ hp } (33{,}000)/V = 195P$$

$$I = [(\sin 20^\circ \cos 20^\circ)/2](4/5) = 0.128$$

$$S_{fe} C_{Li} C_R = 165{,}000 \text{ psi} \quad \text{(directly from Figure 9.21)}$$



Substituting gives

$$2300\sqrt{\frac{(195P)(1.25)}{(14/P)(18/P)(0.128)}}(1.4)(1)(1.3) = 165{,}000$$

from which

$$P = 7.21 \text{ teeth/in.}$$

3. Tentatively choose a standard pitch of 7, compute the corresponding value of V, refine the estimate of K_v, and compute the value of b required to balance σ_H and S_H. (Note that if $P = 8$ were chosen, b would have to exceed $14/P$ to balance σ_H and S_H.)

$$V = \frac{\pi d_p n_p}{12} = \frac{\pi(18/7)(3600)}{12} = 2424 \text{ fpm}$$

From Figure 15.24, $K_v = 1.5$, and

$$2300\sqrt{\frac{(195 \times 7)(1.25)}{b(18/7)(0.128)}(1.5)(1)(1.3)} = 165{,}000$$

from which $b = 1.96$ in. Round off to $b = 2$ in. For this value of b, $K_m = 1.3$ is satisfactory. Also note that b remained at $14/P$ because decreasing P from 7.21 to 7 offsets increasing K_v from 1.4 to 1.5.

4. Check the contact ratio, using Eq. 15.9.

The pitch radii are $r_p = 9/7$ and $r_g = 36/7$.

The addendum, $a = 1/P$; and hence, $r_{ap} = 10/7$, $r_{ag} = 37/7$.

Center distance, $c = r_p + r_g = 45/7$.

From Eq. 15.11, $r_{bp} = (9/7)\cos 20°$, $r_{bg} = (36/7)\cos 20°$.

From Eq. 15.10, $p_b = \pi(18/7)(\cos 20°)/18 = 0.422$ in.

Substituting in Eq. 15.9 gives CR $= 1.67$.

This is satisfactory, but it means that a single pair of teeth carries the load in the vicinity of the pitch line, where pitting is most likely to occur. Thus there can be no sharing of the surface fatigue load, regardless of manufacturing precision. (Note that no sharing was assumed in the preceding calculations.)

5. We need to design the gears to provide adequate bending fatigue strength. Detailed consideration of gear-tooth-bending fatigue for case-hardened gears must include an analysis of stress and strength gradients, as represented in Figure 8.29. Since we anticipate no problem in satisfying this requirement, let us, as previously stated, make the conservative assumption that the fatigue strength of the *core* material (Eq. 15.18) must be equal to the bending fatigue stresses at the surface (Eq. 15.17):

$$S'_n C_L C_G C_S k_r k_t k_{\text{ms}} = \frac{F_t P}{bJ} K_v K_o K_m$$

The manufacturing accuracy is in a "gray area" with respect to load sharing. There will likely be at least a partial sharing, meriting a value of J at least intermediate between the "sharing" and "not sharing" curves (i.e., between $J = 0.235$ and 0.32). But since we conservatively assumed no sharing, there is no need to consider the matter further. In calculating a value for C_s, remember that we are considering fatigue strength *under* the surface, where surface roughness would not be involved:

$$S'_n(1)(1)(1)(0.897)(1)(1.4) = \frac{1365(7)}{2(0.235)}(1.5)(1)(1.3)$$

From this equation $S'_n = 31{,}600$ psi, which requires a (core) hardness of 126 Bhn, a value that will be satisfied or exceeded by any steel selected to meet the case-hardened surface requirement.

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6. In summary, our tentatively proposed design has 20° full-depth teeth, precision-manufactured with ground finish (between curves *A* and *B* of Figure 15.24) from case-hardening steel, surface-hardened to 660 Bhn and 600 Bhn, respectively, for pinion and gear, and with core hardness of at least 126 Bhn. The design also has $P = 7$, $N_p = 18$, $N_g = 72$, $b = 2$ in. ($D_p = 2.57$ in., $D_g = 10.29$ in., $c = 6.43$ in.). As decided, we will specify a case-hardening procedure leaving compressive residual stresses in the surfaces.

Comment: This sample problem represents but one of a great many situations and approaches encountered in the practical design of spur gears. The important thing for the student is to gain a clear understanding of the basic concepts and to understand how these may be brought to bear in handling specific situations. We have seen that a great amount of empirical data is needed in addition to the fundamentals. It is always important to seek out the best and most directly relevant empirical data for use in any given situation. Textbooks such as this can include only sample empirical information. Better values for actual use are often found in company files, contemporary specialized technical literature, and current publications of the AGMA.

15.12 Gear Materials

- The least expensive gear material is CI, ASTM grade 20. Grades 30, 40, 50, and 60 are progressively stronger and more expensive.
- CI gears typically have greater surface fatigue strength than bending fatigue strength. Their internal damping tends to make them quieter than steel gears.
- Nodular CI gears have substantially greater bending strength, together with good surface durability. A good combination is often a steel pinion mated to a CI gear.
- Steel gears that are not heat-treated are inexpensive, but have low surface endurance capacity. Heat-treated steel gears must be designed to resist warpage; hence, alloy steels and oil quenching are usually preferred.
- For hardnesses > 250 to 350 Bhn, machining must usually be done before hardening.
- Greater profile accuracy is obtained if the surfaces are finished after heat treating, as
- by grinding. (But if grinding is done, avoid residual tensile stresses at surface.)
- Through-hardened gears generally have 0.35 to 0.6 % carbon. Surface or case-hardened gears are usually processed by flame hardening, induction hardening, carburizing, or nitriding.
- Of the nonferrous metals, bronzes are most often used for making gears.

15.12 Gear Materials

- Nonmetallic gears made of acetal, nylon, and other plastics are generally quiet, durable, reasonably priced, and can often operate under light loads without lubrication.
- Their teeth deflect more easily than those of corresponding metal gears. This promotes effective load sharing among teeth in simultaneous contact, but results in substantial hysteresis heating if the gears are rotating at high speed.
- Since non-metallic materials have low thermal conductivity, special cooling provisions may be required.
- Also, these materials have relatively high coefficients of thermal expansion, and thus may require installation with greater backlash than metal gears.
- Often the base plastics used for gears are formulated with fillers, such as glass fibers, for strength, and with lubricants such as Teflon for reduced friction and wear.
- Nonmetallic gears are usually mated with CI or steel pinions.
- For best wear resistance, hardness of mating metal pinion should be at least 300 Bhn.
- Design procedures for gears made of plastics are similar to those for gears made of metals, but are not yet as reliable.
- Hence, prototype testing is even more important than for metal gears.

Contents of today's lecture

14

Rolling-Element Bearings

14.1 Comparison of Alternative Means for Sup

- simplest bearings are unlubricated sliding bearing
- Lower friction-lubricant-Oil or grease is used in low-speed applications
- sliding bearings in engine crankshafts - hydrodynamic lubrication where oil film completely separates the surfaces.
- In rolling-element bearings the shaft and outer members are separated by balls or rollers
- Hard material cos of high contact stresses
- inner and outer rings and balls /rollers, retainer
- advantage of R-E bearings is low starting friction.
- Sliding bearings can achieve comparably low friction only with full-film lubrication-hydrostatic
- Rolling-element aka “antifriction” bearings. not in all cases provide lower friction than fluid-film bearings.

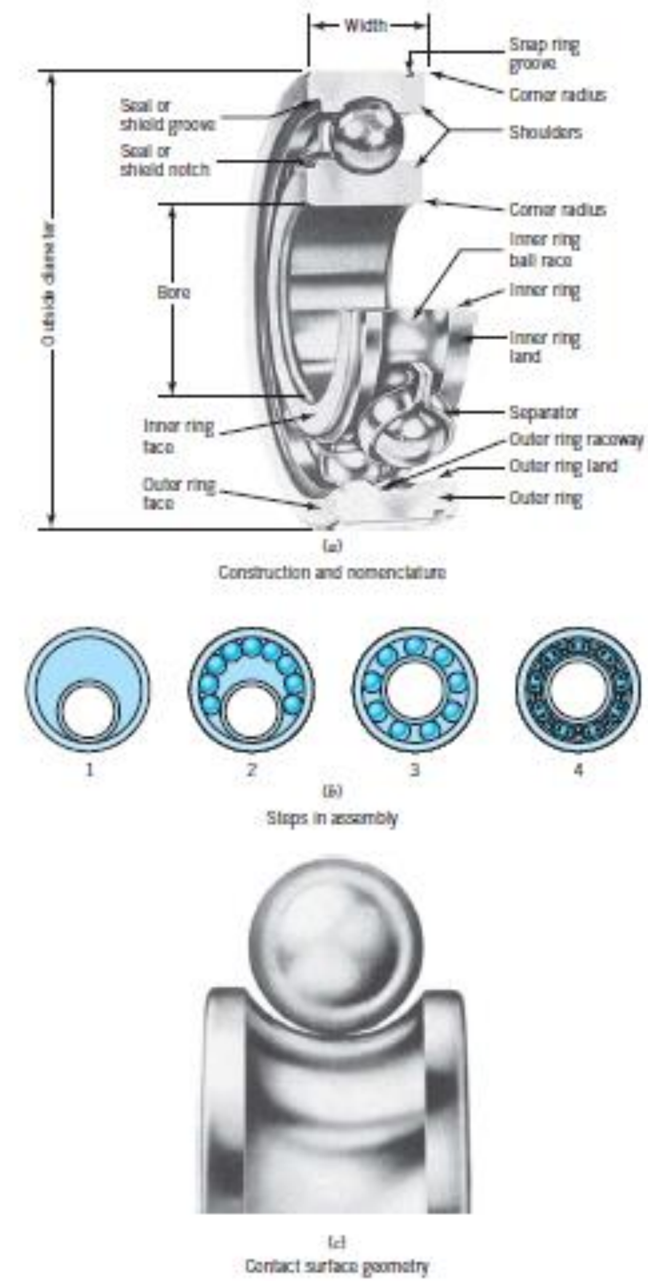
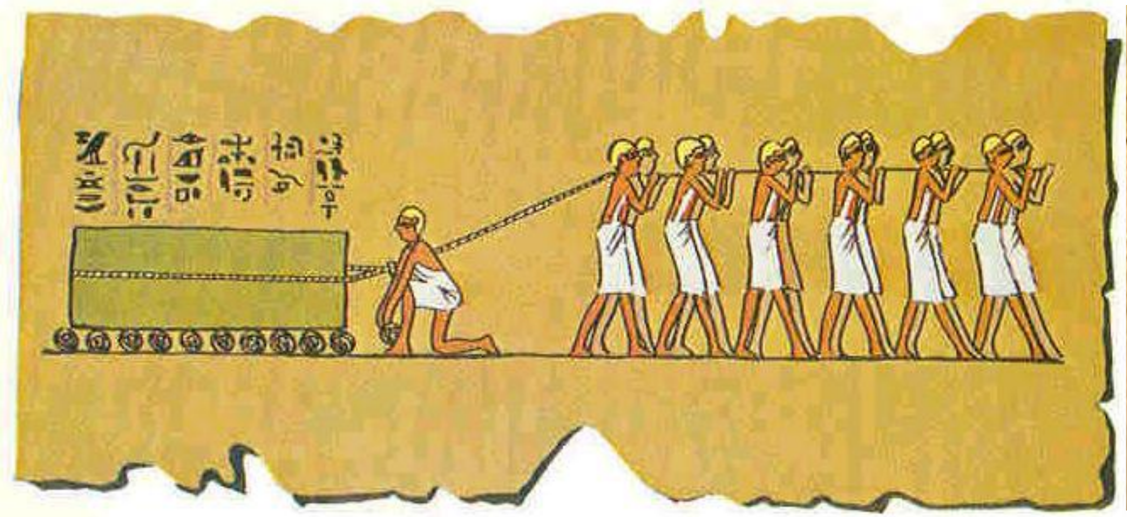


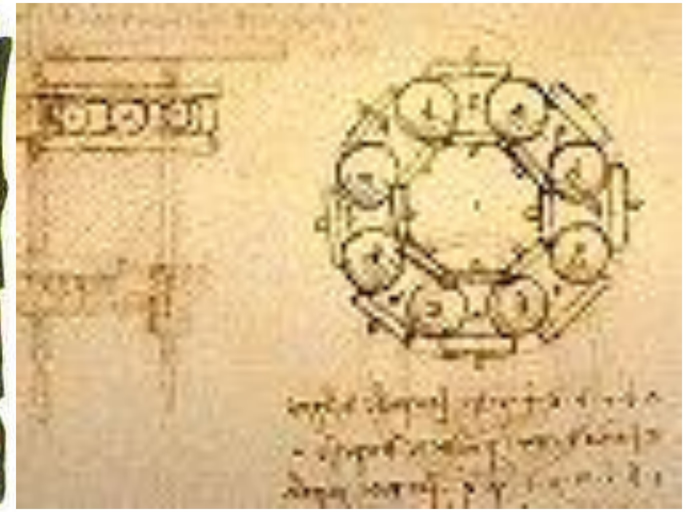
Figure 14.1
Radial ball bearing (deep-groove or “Conrad” type). (Courtesy New Departure–Hyatt Bearing Division, General Motors Corporation.)

comparison	Journal bearing	Roller bearing
Speed	operate efficiently in high speed	operate efficiently in low speed
Friction Please refer to Slides 3:Bearings page 15	high friction at lower speeds but lower friction in high speed power losses due to friction will increase again at high speeds which requires proper oil flow rate for heat rejection.	lower friction in lower speed but higher friction in high speed
Reliability	more reliable	relatively less reliable
Maintenance	need maintenance Permanent care about lubrication	doesn't need maintenance Only occasional attention
Repairing	Possible re-repair	we must replace it if failure occurred
Accuracy of rotation	higher rpm precision	relatively lower rpm precision
Rigidity	more rigid	less rigid
Shocks	Bear the shocks and high loads	Doesn’t withstand the shocks and high loads
Axial length(width)	very long to bear the shocks and high loads	relatively less length
Application	used in higher speed application like turbines and compressors….etc	used in lower speed application like gear box
Cost	Relatively inexpensive	expensive
Bearing Life	Long life	Short life
failure	Lower possibility to failure occur	Higher possibility to failure occur
Examples	Journal bearing(sleeve bearing)	Ball bearing, Cylindrical roller bearing, Taper roller bearing and Needle bearing.

- The inner and outer rings, balls and cage; also known as a retainer, are assembled.
- Lubrication such as oil or grease is applied based on the application requirement of the bearing.
- After lubrication has been applied, and based on the customer's requirements, the bearing is assembled with either one or two shields.
- This is the process where exactly the right parts are assembled to meet the customer's specific requirements.

14.2 History of Rolling-Element Bearings

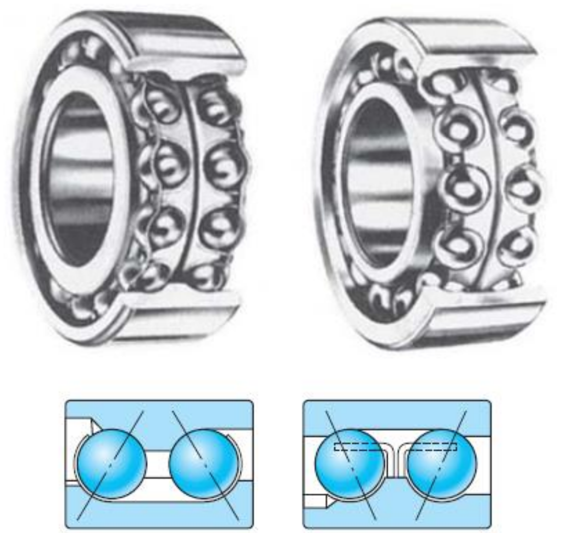




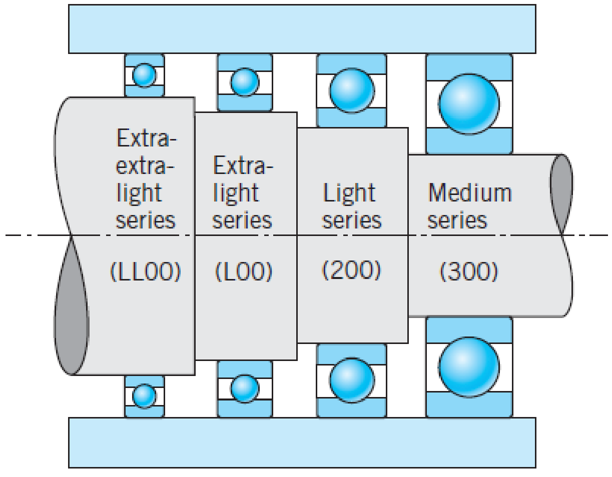
- The first recorded use of rolling elements to overcome sliding friction was by Egyptians, probably before 200 B.C.
- Some early chariot wheels used crude roller bearings.
- Around A.D. 1500 Leonardo da Vinci is considered to have invented and partially developed modern ball and roller bearings.
- Invention of Bessemer steel process in 1856 made economic possibility of suitable bearing material.
- Then, ball bearings were rapidly developed in Europe for use in bicycles.

14.3 Rolling-Element Bearing Types

- With normal operating loads, R E bearings (without seals) typically provide $0.001 < \mu < 0.002$
- Roller bearings - (1) cylindrical, (2) spherical, (3) tapered, and (4) needle.
- Figure 14.1a shows the construction and nomenclature of a typical radial ball bearing, Figure 14.1c illustrates the contact between a ball and raceway.
- Ball bearings are made in various proportions (fig)
- intended for radial loads, but also some thrust
- Angular-contact bearings, have substantial thrust capacity in one direction only.
- The double-row ball bearing incorporates a pair of angular-contact bearings into a single unit. made to tolerate substantial angular misalignment of the shaft.



(c) Double row



(b)

Relative proportions of bearings with same outside diameter

FIGURE 14.2
Relative proportions of bearings of different series.

14.4 Design of Rolling-Element Bearings

- technology of rolling-element bearings is curved-surface contact stresses and related fatigue failures - Figure 14.1c geometry of ball-bearing surfaces
- The selection of bearing race curvature (104 % of ball radius for the inner race, slightly more for the outer race is a compromise between providing load-supporting area and accepting sliding friction.
- Selection of the material is also critical. ball-bearing rings and balls have been made of high-carbon chrome steel(58-65 Rockwell C).
- Roller-bearing components are more often made of carburized alloy steel. Surface compressive residual stresses are inherent with carburization.
- Cleanliness of the steel is of extreme importance
- Manufacturing tolerances are extremely critical
- For example, tolerances on bearing bores between 35 and 50 mm range from +0.0000 in. to -0.0005 in. for ABEC grade 1 to +0.00000 in. to -0.00010 in. for ABEC grade 9. Tolerances on other dimensions are comparable.

14.6 "Catalogue Information" for Rolling-Element Bearings

- Bearing manufacturers' catalogues identify bearings by number, complete dimensional information, rated load capacities, details concerning mounting, lubrication, and operation.
- Dimensions of the more common bearings given in Table 14.1 and in Figure 14.11.
- For bearings bores > 20 mm, the bore dia is 5X the last two digits in Bearing #
- L08 is an extra-light series bearing with a 40-mm bore. bearing numbers include additional letters and # to give more info.
- Table 14.2 lists rated load capacities, C – constant radial load 90 % of a group of identical bearings can endure for 9 * 10^7 revs without surface fatigue failures (3000 hours at 500rpm)
- The basis for ratings must always be checked.

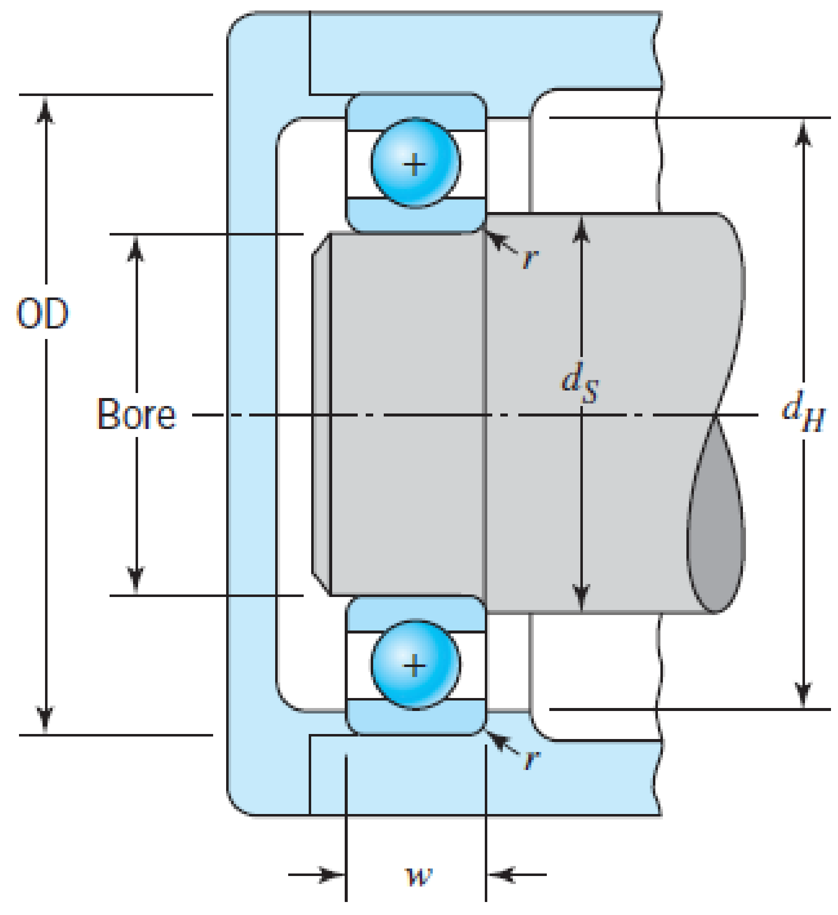


FIGURE 14.11
Shaft and housing shoulder dimensions.

TABLE 14.1 Bearing Dimensions

Bearing Basic Number	Bore (mm)	Ball Bearings OD (mm)	Ball Bearings w (mm)	Ball Bearings r^a (mm)	Ball Bearings d_S (mm)	Ball Bearings d_H (mm)	Roller Bearings OD (mm)	Roller Bearings w (mm)	Roller Bearings r^a (mm)	Roller Bearings d_S (mm)	Roller Bearings d_H (mm)
L00	10	26	8	0.30	12.7	23.4					
200	10	30	9	0.64	13.8	26.7					
300	10	35	11	0.64	14.8	31.2					
L01	12	28	8	0.30	14.5	25.4					
201	12	32	10	0.64	16.2	28.4					
301	12	37	12	1.02	17.7	32.0					
L02	15	32	9	0.30	17.5	29.2					
202	15	35	11	0.64	19.0	31.2					
302	15	42	13	1.02	21.2	36.6					
L03	17	35	10	0.30	19.8	32.3	35	10	0.64	20.8	32.0
203	17	40	12	0.64	22.4	34.8	40	12	0.64	20.8	36.3
303	17	47	14	1.02	23.6	41.1	47	14	1.02	22.9	41.4
L04	20	42	12	0.64	23.9	38.1	42	12	0.64	24.4	36.8
204	20	47	14	1.02	25.9	41.7	47	14	1.02	25.9	42.7
304	20	52	15	1.02	27.7	45.2	52	15	1.02	25.9	46.2
L05	25	47	12	0.64	29.0	42.9	47	12	0.64	29.2	43.4
205	25	52	15	1.02	30.5	46.7	52	15	1.02	30.5	47.0
305	25	62	17	1.02	33.0	54.9	62	17	1.02	31.5	55.9
L06	30	55	13	1.02	34.8	49.3	47	9	0.38	33.3	43.9

(continued)

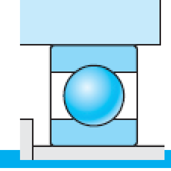
TABLE 14.1 Bearing Dimensions

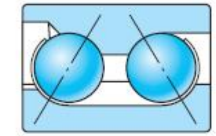
Bearing Basic Number	Bore (mm)	Ball Bearings OD (mm)	Ball Bearings w (mm)	Ball Bearings r^a (mm)	Ball Bearings d_S (mm)	Ball Bearings d_H (mm)	Roller Bearings OD (mm)	Roller Bearings w (mm)	Roller Bearings r^a (mm)	Roller Bearings d_S (mm)	Roller Bearings d_H (mm)
206	30	62	16	1.02	36.8	55.4	62	16	1.02	36.1	56.4
306	30	72	19	1.02	38.4	64.8	72	19	1.52	37.8	64.0
L07	35	62	14	1.02	40.1	56.1	55	10	0.64	39.4	50.8
207	35	72	17	1.02	42.4	65.0	72	17	1.02	41.7	65.3
307	35	80	21	1.52	45.2	70.4	80	21	1.52	43.7	71.4
L08	40	68	15	1.02	45.2	62.0	68	15	1.02	45.7	62.7
208	40	80	18	1.02	48.0	72.4	80	18	1.52	47.2	72.9
308	40	90	23	1.52	50.8	80.0	90	23	1.52	49.0	81.3
L09	45	75	16	1.02	50.8	68.6	75	16	1.02	50.8	69.3
209	45	85	19	1.02	52.8	77.5	85	19	1.52	52.8	78.2
309	45	100	25	1.52	57.2	88.9	100	25	2.03	55.9	90.4
L10	50	80	16	1.02	55.6	73.7	72	12	0.64	54.1	68.1
210	50	90	20	1.02	57.7	82.3	90	20	1.52	57.7	82.8
310	50	110	27	2.03	64.3	96.5	110	27	2.03	61.0	99.1
L11	55	90	18	1.02	61.7	83.1	90	18	1.52	62.0	83.6
211	55	100	21	1.52	65.0	90.2	100	21	2.03	64.0	91.4
311	55	120	29	2.03	69.8	106.2	120	29	2.03	66.5	108.7
L12	60	95	18	1.02	66.8	87.9	95	18	1.52	67.1	88.6
212	60	110	22	1.52	70.6	99.3	110	22	2.03	69.3	101.3
312	60	130	31	2.03	75.4	115.6	130	31	2.54	72.9	117.9
L13	65	100	18	1.02	71.9	92.7	100	18	1.52	72.1	93.7
213	65	120	23	1.52	76.5	108.7	120	23	2.54	77.0	110.0
313	65	140	33	2.03	81.3	125.0	140	33	2.54	78.7	127.0
L14	70	110	20	1.02	77.7	102.1	110	20	Not Available		
214	70	125	24	1.52	81.0	114.0	125	24	2.54	81.8	115.6
314	70	150	35	2.03	86.9	134.4	150	35	3.18	84.3	135.6
L15	75	115	20	1.02	82.3	107.2	115	20	Not Available		
215	75	130	25	1.52	86.1	118.9	130	25	2.54	85.6	120.1
315	75	160	37	2.03	92.7	143.8	160	37	3.18	90.4	145.8

TABLE 14.1 Bearing Dimensions

Bearing Basic Number	Bore (mm)	Ball Bearings OD (mm)	Ball Bearings w (mm)	Ball Bearings r[a] (mm)	Ball Bearings d_S (mm)	Ball Bearings d_H (mm)	Roller Bearings OD (mm)	Roller Bearings w (mm)	Roller Bearings r[a] (mm)	Roller Bearings d_S (mm)	Roller Bearings d_H (mm)
L16	80	125	22	1.02	88.1	116.3	125	22	2.03	88.4	117.6
216	80	140	26	2.03	93.2	126.7	140	26	2.54	91.2	129.3
316	80	170	39	2.03	98.6	152.9	170	39	3.18	96.0	154.4
L17	85	130	22	1.02	93.2	121.4	130	22	2.03	93.5	122.7
217	85	150	28	2.03	99.1	135.6	150	28	3.18	98.0	139.2
317	85	180	41	2.54	105.7	160.8	180	41	3.96	102.9	164.3
L18	90	140	24	1.52	99.6	129.0	140	24		Not Available	
218	90	160	30	2.03	104.4	145.5	160	30	3.18	103.1	147.6
318	90	190	43	2.54	111.3	170.2	190	43	3.96	108.2	172.7
L19	95	145	24	1.52	104.4	134.1	145	24		Not Available	
219	95	170	32	2.03	110.2	154.9	170	32	3.18	109.0	157.0
319	95	200	45	2.54	117.3	179.3	200	45	3.96	115.1	181.9
L20	100	150	24	1.52	109.5	139.2	150	24	2.54	109.5	141.7
220	100	180	34	2.03	116.1	164.1	180	34	3.96	116.1	167.1
320	100	215	47	2.54	122.9	194.1	215	47	4.75	122.4	194.6
L21	105	160	26	2.03	116.1	146.8	160	26		Not Available	
221	105	190	36	2.03	121.9	173.5	190	36	3.96	121.4	175.3
321	105	225	49	2.54	128.8	203.5	225	49	4.75	128.0	203.5
L22	110	170	28	2.03	122.7	156.5	170	28	2.54	121.9	159.3
222	110	200	38	2.03	127.8	182.6	200	38	3.96	127.3	183.9
322	110	240	50	2.54	134.4	218.2	240	50	4.75	135.9	217.2
L24	120	180	28	2.03	132.6	166.6	180	28		Not Available	
224	120	215	40	2.03	138.2	197.1	215	40	4.75	139.2	198.9
324	120		Not Available				260	55	6.35	147.8	235.2
L26	130	200	33	2.03	143.8	185.4	200	33	3.18	143.0	188.2
226	130	230	40	2.54	149.9	210.1	230	40	4.75	149.1	213.9
326	130	280	58	3.05	160.0	253.0	280	58	6.35	160.3	254.5
L28	140	210	33	2.03	153.7	195.3	210	33		Not Available	
228	140	250	42	2.54	161.5	228.6	250	42	4.75	161.5	232.4
328	140		Not Available				300	62	7.92	172.0	271.3
L30	150	225	35	2.03	164.3	209.8	225	35	3.96	164.3	212.3
230	150	270	45	2.54	173.0	247.6	270	45	6.35	174.2	251.0
L32	160	240	38	2.03	175.8	223.0	240	38		Not Available	
232	160		Not Available				290	48	6.35	185.7	269.5
L36	180	280	46	2.03	196.8	261.6	280	46	4.75	199.6	262.9
236	180		Not Available				320	52	6.35	207.5	298.2
L40	200						310	51		Not Available	
240	200		Not Available				360	58	7.92	232.4	334.5
L44	220						340	56		Not Available	
244	220		Not Available				400	65	9.52	256.0	372.1
L48	240						360	56		Not Available	
248	240		Not Available				440	72	9.52	279.4	408.4

TABLE 14.2 Bearing Rated Capacities, C, for $L_R = 90 \times 10^6$ Revolution Life with 90 Percent Reliability





	Radial Ball, $\alpha = 0°$			Angular Ball, $\alpha = 25°$			Roller		
Bore (mm)	L00 Xlt (kN)	200 lt (kN)	300 med (kN)	L00 Xlt (kN)	200 lt (kN)	300 med (kN)	1000 Xlt (kN)	1200 lt (kN)	1300 med (kN)
10	1.02	1.42	1.90	1.02	1.10	1.88			
12	1.12	1.42	2.46	1.10	1.54	2.05			
15	1.22	1.56	3.05	1.28	1.66	2.85			
17	1.32	2.70	3.75	1.36	2.20	3.55	2.12	3.80	4.90
20	2.25	3.35	5.30	2.20	3.05	5.80	3.30	4.40	6.20
25	2.45	3.65	5.90	2.65	3.25	7.20	3.70	5.50	8.50
30	3.35	5.40	8.80	3.60	6.00	8.80	2.40[a]	8.30	10.0
35	4.20	8.50	10.6	4.75	8.20	11.0	3.10[a]	9.30	13.1
40	4.50	9.40	12.6	4.95	9.90	13.2	7.20	11.1	16.5
45	5.80	9.10	14.8	6.30	10.4	16.4	7.40	12.2	20.9
50	6.10	9.70	15.8	6.60	11.0	19.2	5.10[a]	12.5	24.5
55	8.20	12.0	18.0	9.00	13.6	21.5	11.3	14.9	27.1
60	8.70	13.6	20.0	9.70	16.4	24.0	12.0	18.9	32.5

60	8.70	13.6	20.0	9.70	16.4	24.0	12.0	18.9	32.5
65	9.10	16.0	22.0	10.2	19.2	26.5	12.2	21.1	38.3
70	11.6	17.0	24.5	13.4	19.2	29.5		23.6	44.0
75	12.2	17.0	25.5	13.8	20.0	32.5		23.6	45.4
80	14.2	18.4	28.0	16.6	22.5	35.5	17.3	26.2	51.6
85	15.0	22.5	30.0	17.2	26.5	38.5	18.0	30.7	55.2
90	17.2	25.0	32.5	20.0	28.0	41.5		37.4	65.8
95	18.0	27.5	38.0	21.0	31.0	45.5		44.0	65.8
100	18.0	30.5	40.5	21.5	34.5		20.9	48.0	72.9
105	21.0	32.0	43.5	24.5	37.5			49.8	84.5
110	23.5	35.0	46.0	27.5	41.0	55.0	29.4	54.3	85.4
120	24.5	37.5		28.5	44.5			61.4	100.1
130	29.5	41.0		33.5	48.0	71.0	48.9	69.4	120.1
140	30.5	47.5		35.0	56.0			77.4	131.2
150	34.5			39.0	62.0		58.7	83.6	
160								113.4	
180	47.0			54.0			97.9	140.1	
200								162.4	
220								211.3	
240								258.0	

[a] 1000 (Xlt) series bearings are not available in these sizes. Capacities shown are for the 1900 (XXlt) series.

14.7 Bearing Selection

- For specific application, we select the bearing type, grade of precision, lubricant, seal, and basic load rating.
- special circumstances like subjected to a heavy load when not rotating, its static load capacity should not be exceeded. Else brinnelling
- Another consideration is speed. The limitation is linear surface speed rather than rotating speed; so, small bearings can operate at higher rpm than large bearings.
- Lubrication is important in high speed applications, the best being a fine oil mist or spray. This provides lubricant film and carries away friction permitting surface speeds up to 75 m/s and have a life of 3000 hours carrying 1/3rd of the rated load
- In selecting bearings, attention should be given to possible misalignment and to sealing and lubrication. The size of bearing selected for an application is usually influenced by the size of shaft (for strength and rigidity considerations) and space.
- In addition, the bearing must have a high enough load rating to provide an acceptable combination of life and reliability. The major factors influencing the load rating requirement are discussed.

14.7.1 Life Requirement

- Bearing applications require lives different from that used for the catalogue rating.
- Palmgren determined that ball-bearing life varies inversely with approximately the third power of the load. Later studies have indicated that this exponent ranges between 3 and 4 for various rolling-element bearings.
- Many manufacturers retain Palmgren's exponent of 3 for ball bearings and use 10/3 for roller bearings. We use 10/3 for all bearings

$$L = L_R(C/F_r)^{3.33} \qquad C_{\text{req}} = F_r(L/L_R)^{0.3}$$

C = rated capacity (as from Table 14.2) and C_{req} = the required value of C for the application

L_R = life corresponding to rated capacity (i.e., 9×10^7 revolutions)

F_r = radial load involved in the application

L = life corresponding to radial load F_r, or life required by the application

- Doubling the load on a bearing reduces its life by a factor of about 10. Different manufacturers' catalogues use different values of L_R.
- Some use $L_R = 10^6$ revolutions. Values in Table 14.2 must be multiplied by 3.86 to be comparable with ratings based on a life of 10^6 revolutions.

14.7.2 Reliability Requirement

- Tests show that the median life of rolling-element bearings is about five times the standard 10 % failure fatigue life.

- Standard life is designated as the L_{10} life (B_{10} life). corresponds to 10% failures

- it also means that this is the life for which 90% have not failed, and corresponds to 90% reliability. Thus, the life for 50% R is about 5X the life for 90% R.

- Many designs require > 90%R. Fatigue lives characteristically have a skewed distribution, as in Figure 14.12.

- Using the Weibull equation with experimental data, the AFBMA has formulated recommended life adjustment reliability factors, K_r , plotted in Figure 14.13.

- The rated bearing life for R > 90% is, $K_r L_R$. Putting it in Eq. 14.1 gives

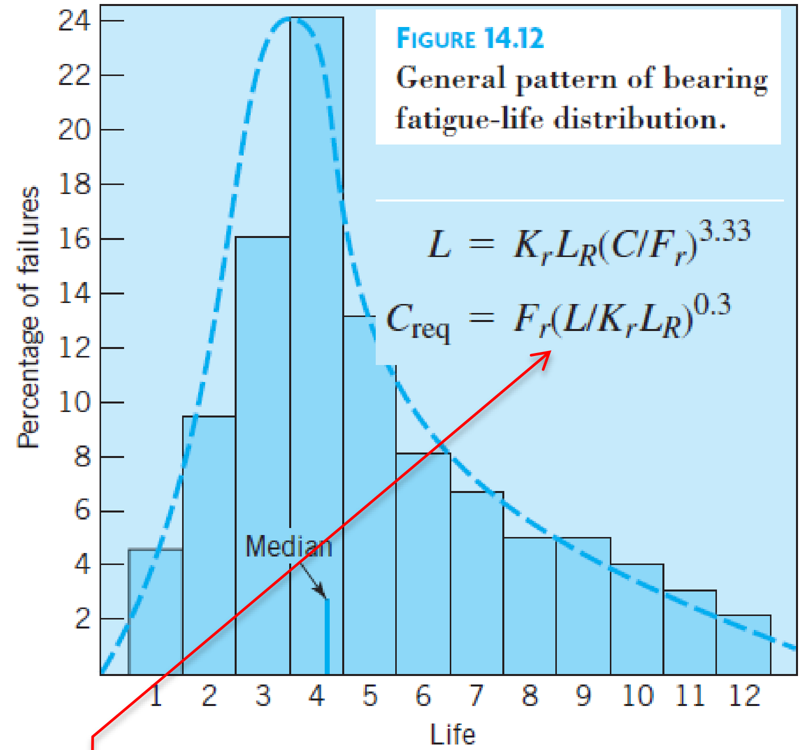


FIGURE 14.12
General pattern of bearing fatigue-life distribution.

$$L = K_r L_R (C/F_r)^{3.33}$$

$$C_{req} = F_r (L/K_r L_R)^{0.3}$$

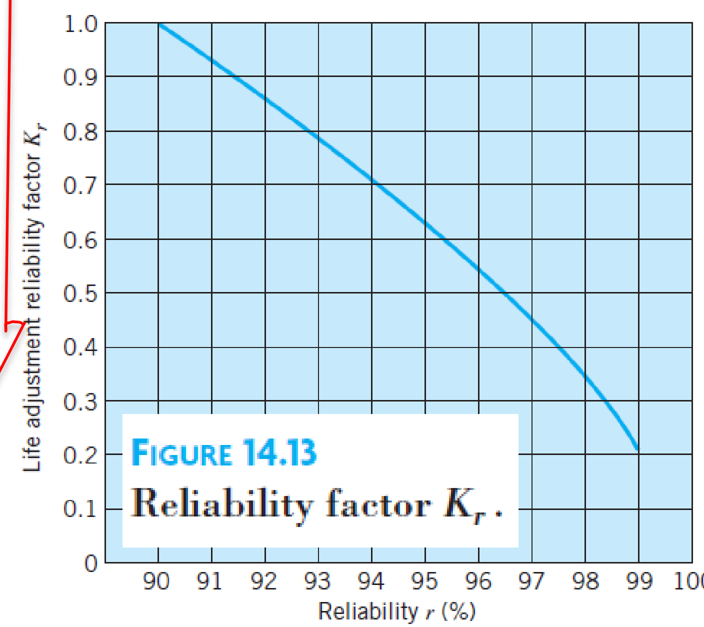
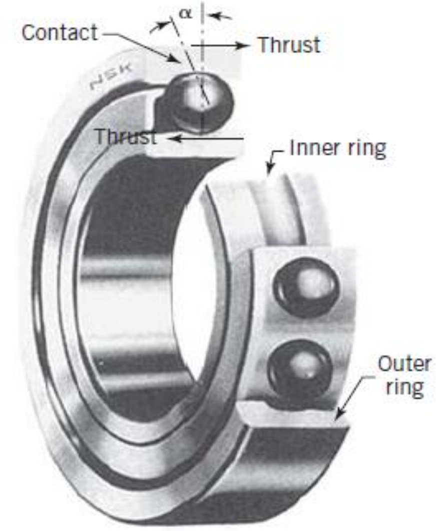


FIGURE 14.13
Reliability factor K_r.

14.7.3 Influence of Axial Loading



(b) Angular-contact type

- Cylindrical roller bearings are limited in F_t because F_r produce sliding friction at the roller ends. If bearings are properly aligned, radially loaded, and oil-lubricated, their F_t is 20% F_r.
- So pair of cylindrical roller bearings support shafts having light F_t by gears or sprockets. Tapered roller bearings carry substantial F_t and F_r.
- For ball bearings, any combination F_r and F_t results in ≅ same life as does a pure radial equivalent load, F_e, calculated from equns. Radial bearings have a 0° α,

$\alpha = 0°$ (radial ball bearings)

$$\left.\begin{aligned}
&\text{For } 0 < F_t/F_r < 0.35, \quad F_e = F_r \\
&\text{For } 0.35 < F_t/F_r < 10, \quad F_e = F_r\left[1 + 1.115\left(\frac{F_t}{F_r} - 0.35\right)\right] \\
&\text{For } F_t/F_r > 10, \quad F_e = 1.176F_t
\end{aligned}\right\} \qquad (14.3)^2$$

angular ball bearings 15°, 25°, & 35° α. We look at angular ball bearings with α = 25°.

$\alpha = 25°$ (angular ball bearings)

$$\left.\begin{aligned}
&\text{For } 0 < F_t/F_r < 0.68, \quad F_e = F_r \\
&\text{For } 0.68 < F_t/F_r < 10, \quad F_e = F_r\left[1 + 0.870\left(\frac{F_t}{F_r} - 0.68\right)\right] \\
&\text{For } F_t/F_r > 10, \quad F_e = 0.911F_t
\end{aligned}\right\} \qquad (14.4)^2$$

14.7.4 Shock Loading

- The standard bearing rated capacity is for condition of uniform load w/o shock.
- This desirable condition may prevail for some applications (such as bearings on the motor and rotor shafts of a belt-driven electric blower), but other applications have various degrees of shock loading.
- This has the effect of increasing the nominal load by an application factor K_a. Table 14.3 gives representative sample values.

TABLE 14.3 Application Factors K_a

Type of Application	Ball Bearing	Roller Bearing
Uniform load, no impact	1.0	1.0
Gearing	1.0–1.3	1.0
Light impact	1.2–1.5	1.0–1.1
Moderate impact	1.5–2.0	1.1–1.5
Heavy impact	2.0–3.0	1.5–2.0

14.7.5 Summary

$$L = K_r L_R (C/F_e K_a)^{3.33}$$

$$C_{req} = F_e K_a (L/K_r L_R)^{0.3}$$

- Substituting F_e for F_r and adding K_a modifies Eq. 14.2
- When these equations are used, what L should be required
- Bearing manufacturers formerly reduced life ratings when the outer ring rotated relative to the load (as with a trailer wheel, rotating around a fixed spindle).
- As a result of more recent evidence, If both rings rotate, the relative rotation between the two is used in making life calculations.
- Table 14.4 may be used as a guide - specific info not present
- When loads vary with time, cumulative-damage rule is applicable.

Table 14.4 Representative Bearing Design Lives

Type of Application	Design Life (thousands of hours)
Instruments and apparatus for infrequent use	0.1–0.5
Machines used intermittently, where service interruption is of minor importance	4–8
Machines intermittently used, where reliability is of great importance	8–14
Machines for 8-hour service, but not every day	14–20
Machines for 8-hour service, every working day	20–30
Machines for continuous 24-hour service	50–60
Machines for continuous 24-hour service where reliability is of extreme importance	100–200

Sample Problem 14.1D Ball Bearing Selection

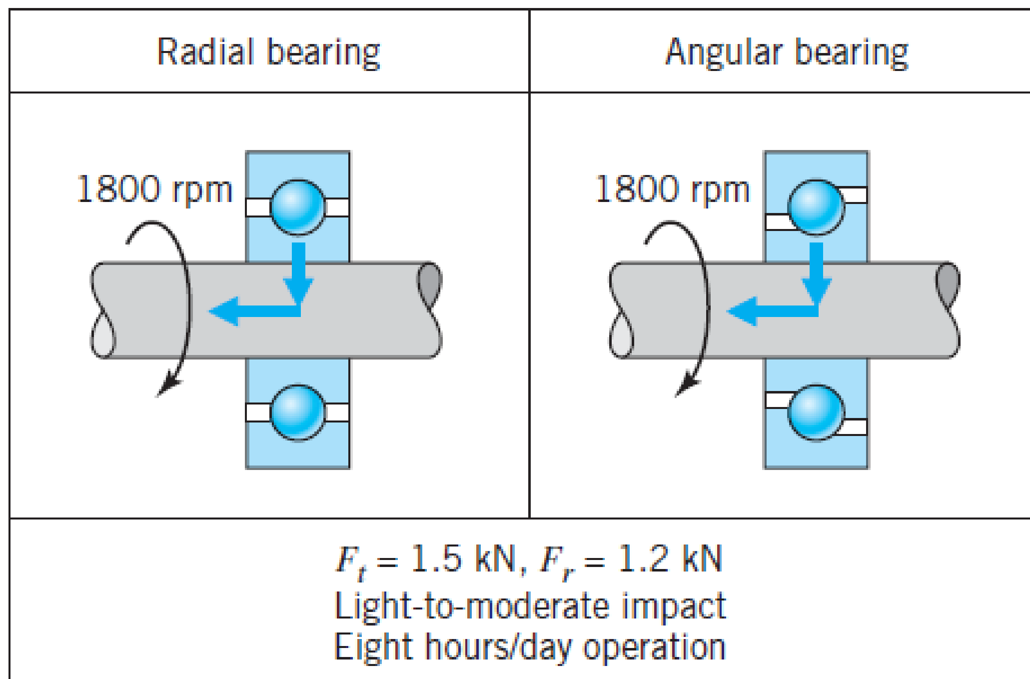
Select a ball bearing for an industrial machine press fit onto a shaft and intended for continuous one-shift (8-hour day) operation at 1800 rpm. Radial and thrust loads are 1.2 and 1.5 kN, respectively, with light-to-moderate impact.

Solution

Known: A ball bearing operates 8 hours per day, 5 days per week, and carries constant radial and thrust loads.

Find: Select a suitable ball bearing.

Schematic and Given Data:



Decisions and Assumptions:

1. A conservative design for light-to-moderate impact is required.
2. A conservative design life for 8 hours per day continuous service is required.
3. A 90 percent reliability is required.
4. Both a radial ($\alpha = 0°$) and an angular ball bearing ($\alpha = 25°$) should be chosen. (See Figure 14.14.)
5. Ball-bearing life varies inversely with the $\frac{10}{3}$ power of the load (Eq. 14.5b is accurate).
6. The press fit does not affect the bearing life.

Design Analysis:

1. From Eqs. 14.3 and 14.4, the equivalent radial load for radial and angular ball bearings, respectively, are for $F_t/F_r = 1.25$,

$$F_e = F_r\left[1 + 1.115\left(\frac{F_t}{F_r} - 0.35\right)\right]$$

$$F_e = F_r\left[1 + 1.115\left(\frac{F_t}{F_r} - 0.35\right)\right]$$

$$= 1.2\left[1 + 1.115\left(\frac{1.5}{1.2} - 0.35\right)\right] = 2.4 \text{ kN} \quad \text{(radial bearing)}$$

$$F_e = F_r\left[1 + 0.870\left(\frac{F_t}{F_r} - 0.68\right)\right]$$

$$F_e = F_r\left[1 + 0.870\left(\frac{F_t}{F_r} - 0.68\right)\right]$$

$$= 1.2\left[1 + 0.87\left(\frac{1.5}{1.2} - 0.68\right)\right] = 1.8 \text{ kN} \quad \text{(angular bearing)}$$

TABLE 14.3 Application Factors K_a

Type of Application	Ball Bearing	Roller Bearing
Uniform load, no impact	1.0	1.0
Gearing	1.0–1.3	1.0
Light impact	1.2–1.5	1.0–1.1
Moderate impact	1.5–2.0	1.1–1.5
Heavy impact	2.0–3.0	1.5–2.0

TABLE 14.4 Representative Bearing Design Lives

Type of Application	Design Life (thousands of hours)
Instruments and apparatus for infrequent use	0.1–0.5
Machines used intermittently, where service interruption is of minor importance	4–8
Machines intermittently used, where reliability is of great importance	8–14
Machines for 8-hour service, but not every day	14–20
Machines for 8-hour service, every working day	20–30
Machines for continuous 24-hour service	50–60
Machines for continuous 24-hour service where reliability is of extreme importance	100–200

2. From Table 14.3 choose $K_a = 1.5$. From Table 14.4 choose (conservatively) 30,000-hour life. Life in revolutions is $L = 1800 \text{ rpm} \times 30{,}000 \text{ h} \times 60 \text{ min/h} = 3240 \times 10^6$ rev.

3. For standard 90 percent reliability ($K_r = 1$), and for $L_R = 90 \times 10^6$ rev (for use with Table 14.2), Eq. 14.5b gives

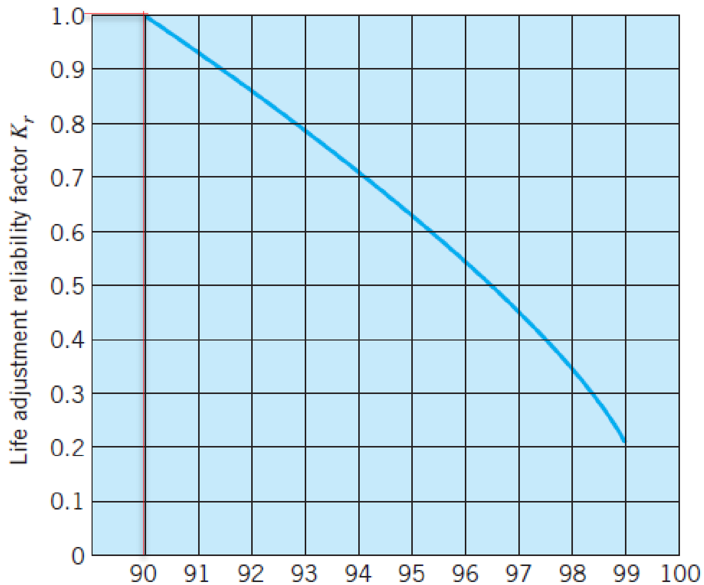
$$
\begin{aligned}
C_{\text{req}} &= (2.4)(1.5)(3240/90)^{0.3} = 10.55 \text{ kN} \quad \text{(radial bearing)} \\
&= (1.8)(1.5)(3240/90)^{0.3} = 7.91 \text{ kN} \quad \text{(angular bearing)}
\end{aligned}
$$

4. From Table 14.2 (with bearing number for a given bore and series obtained from Table 14.1), appropriate choices would be radial bearings L14, 211, and 307, and angular-contact bearings L11, 207, and 306.

Comment: Other factors being equal, the final selection would be made on the basis of cost of the total installation, including shaft and housing. Shaft size should be sufficient to limit bearing misalignment to no more than 15′.

$$L = K_r L_R (C/F_e K_a)^{3.33}$$

$$C_{\text{req}} = F_e K_a (L/K_r L_R)^{0.3}$$



radial bearings L14, 211, and 307,

angular-contact bearings L11, 207, and 306.

TABLE 14.2 Bearing Rated Capacities, C, for $L_R = 90 \times 10^6$ Revolution Life with 90 Percent Reliability

	Radial Ball, $\alpha = 0°$			Angular Ball, $\alpha = 25°$			Roller		
Bore (mm)	L00 Xlt (kN)	200 lt (kN)	300 med (kN)	L00 Xlt (kN)	200 lt (kN)	300 med (kN)	1000 Xlt (kN)	1200 lt (kN)	1300 med (kN)
30	3.35	5.40	8.80	3.60	6.00	8.80	2.40[a]	8.30	10.0
35	4.20	8.50	10.6	4.75	8.20	11.0	3.10[a]	9.30	13.1
40	4.50	9.40	12.6	4.95	9.90	13.2	7.20	11.1	16.5
45	5.80	9.10	14.8	6.30	10.4	16.4	7.40	12.2	20.9
50	6.10	9.70	15.8	6.60	11.0	19.2	5.10[a]	12.5	24.5
55	8.20	12.0	18.0	9.00	13.6	21.5	11.3	14.9	27.1
60	8.70	13.6	20.0	9.70	16.4	24.0	12.0	18.9	32.5
65	9.10	16.0	22.0	10.2	19.2	26.5	12.2	21.1	38.3
70	11.6	17.0	24.5	13.4	19.2	29.5		23.6	44.0

Table 14.1 Bearing Dimensions

Bearing Basic Number	Bore (mm)	Ball Bearings OD (mm)	Ball Bearings w (mm)	Ball Bearings r^a (mm)	Ball Bearings d_S (mm)	Ball Bearings d_H (mm)	Roller Bearings OD (mm)	Roller Bearings w (mm)	Roller Bearings r^a (mm)	Roller Bearings d_S (mm)	Roller Bearings d_H (mm)
206	30	62	16	1.02	36.8	55.4	62	16	1.02	36.1	56.4
306	30	72	19	1.02	38.4	64.8	72	19	1.52	37.8	64.0
L07	35	62	14	1.02	40.1	56.1	55	10	0.64	39.4	50.8
207	35	72	17	1.02	42.4	65.0	72	17	1.02	41.7	65.3
307	35	80	21	1.52	45.2	70.4	80	21	1.52	43.7	71.4
L08	40	68	15	1.02	45.2	62.0	68	15	1.02	45.7	62.7
208	40	80	18	1.02	48.0	72.4	80	18	1.52	47.2	72.9
308	40	90	23	1.52	50.8	80.0	90	23	1.52	49.0	81.3
L09	45	75	16	1.02	50.8	68.6	75	16	1.02	50.8	69.3
209	45	85	19	1.02	52.8	77.5	85	19	1.52	52.8	78.2
309	45	100	25	1.52	57.2	88.9	100	25	2.03	55.9	90.4
L10	50	80	16	1.02	55.6	73.7	72	12	0.64	54.1	68.1
210	50	90	20	1.02	57.7	82.3	90	20	1.52	57.7	82.8
310	50	110	27	2.03	64.3	96.5	110	27	2.03	61.0	99.1
L11	55	90	18	1.02	61.7	83.1	90	18	1.52	62.0	83.6
211	55	100	21	1.52	65.0	90.2	100	21	2.03	64.0	91.4
311	55	120	29	2.03	69.8	106.2	120	29	2.03	66.5	108.7
L12	60	95	18	1.02	66.8	87.9	95	18	1.52	67.1	88.6
212	60	110	22	1.52	70.6	99.3	110	22	2.03	69.3	101.3
312	60	130	31	2.03	75.4	115.6	130	31	2.54	72.9	117.9
L13	65	100	18	1.02	71.9	92.7	100	18	1.52	72.1	93.7
213	65	120	23	1.52	76.5	108.7	120	23	2.54	77.0	110.0
313	65	140	33	2.03	81.3	125.0	140	33	2.54	78.7	127.0
L14	70	110	20	1.02	77.7	102.1	110	20	Not Available		
214	70	125	24	1.52	81.0	114.0	125	24	2.54	81.8	115.6
314	70	150	35	2.03	86.9	134.4	150	35	3.18	84.3	135.6
L15	75	115	20	1.02	82.3	107.2	115	20	Not Available		
215	75	130	25	1.52	86.1	118.9	130	25	2.54	85.6	120.1
315	75	160	37	2.03	92.7	143.8	160	37	3.18	90.4	145.8

Sample Problem 14.2 Ball Bearing Life and Reliability

Suppose that radial-contact bearing 211 ($C = 12.0$ kN) is selected for the application in Sample Problem 14.1. (a) Estimate the life of this bearing, with 90 percent reliability. (b) Estimate its reliability for 30,000-hour life. (See Figure 14.15.)

Solution

Known: The radial-contact bearing 211 is selected for the application in Sample Problem 14.1.

Find: Determine (a) the bearing life for 90 percent reliability and (b) the bearing reliability for 30,000-hour life.

Schematic and Given Data:

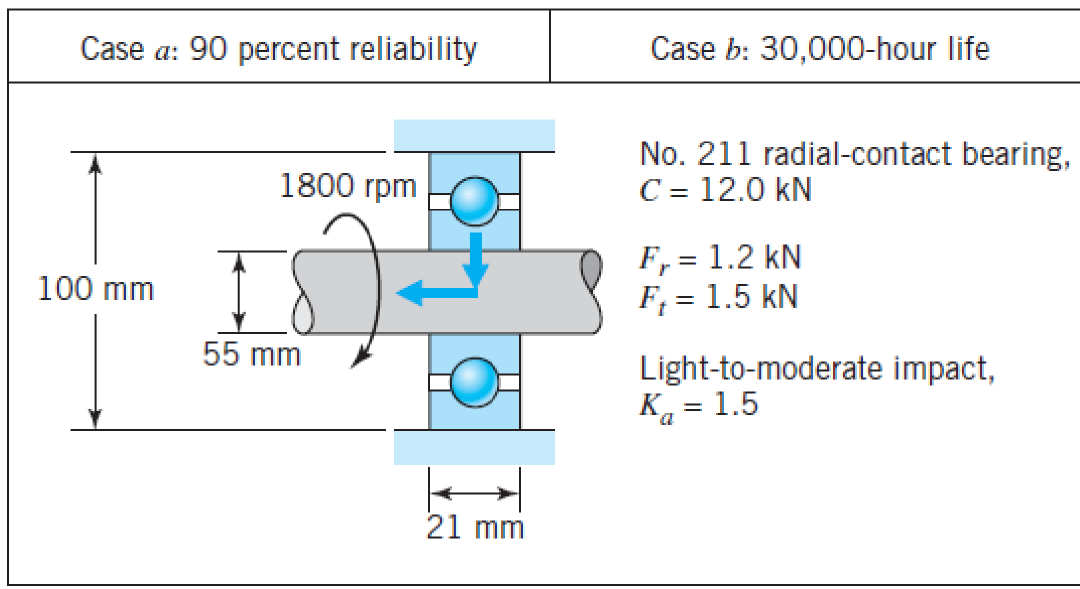


Figure 14.15
Radial-contact bearing.

TABLE 14.1 Bearing Dimensions

Bearing Basic Number	Bore (mm)	Ball Bearings OD (mm)	w (mm)	r^a (mm)	d_S (mm)	d_H (mm)	Roller Bearings OD (mm)	w (mm)	r^a (mm)	d_S (mm)	d_H (mm)
L11	55	90	18	1.02	61.7	83.1	90	18	1.52	62.0	83.6
211	55	100	21	1.52	65.0	90.2	100	21	2.03	64.0	91.4
311	55	120	29	2.03	69.8	106.2	120	29	2.03	66.5	108.7

TABLE 14.2 Bearing Rated Capacities, C, for $L_R = 90 \times 10^6$ Revolution Life with 90 Percent Reliability

	Radial Ball, $\alpha = 0°$			Angular Ball, $\alpha = 25°$			Roller		
Bore (mm)	L00 Xlt (kN)	200 lt (kN)	300 med (kN)	L00 Xlt (kN)	200 lt (kN)	300 med (kN)	1000 Xlt (kN)	1200 lt (kN)	1300 med (kN)
30	3.35	5.40	8.80	3.60	6.00	8.80	2.40[a]	8.30	10.0
35	4.20	8.50	10.6	4.75	8.20	11.0	3.10[a]	9.30	13.1
40	4.50	9.40	12.6	4.95	9.90	13.2	7.20	11.1	16.5
45	5.80	9.10	14.8	6.30	10.4	16.4	7.40	12.2	20.9
50	6.10	9.70	15.8	6.60	11.0	19.2	5.10[a]	12.5	24.5
55	8.20	12.0	18.0	9.00	13.6	21.5	11.3	14.9	27.1
60	8.70	13.6	20.0	9.70	16.4	24.0	12.0	18.9	32.5
65	9.10	16.0	22.0	10.2	19.2	26.5	12.2	21.1	38.3
70	11.6	17.0	24.5	13.4	19.2	29.5		23.6	44.0

Assumptions:

1. Ball-bearing life varies inversely with the $\frac{10}{3}$ power of the load (Eq. 14.5a is accurate).
2. The application factor is $K_a = 1.5$ for light-to-moderate impact.
3. The design life is 30,000 hours.

Analysis:

a. From Eq. 14.5a,

$F_e = 2.4\text{KN}$

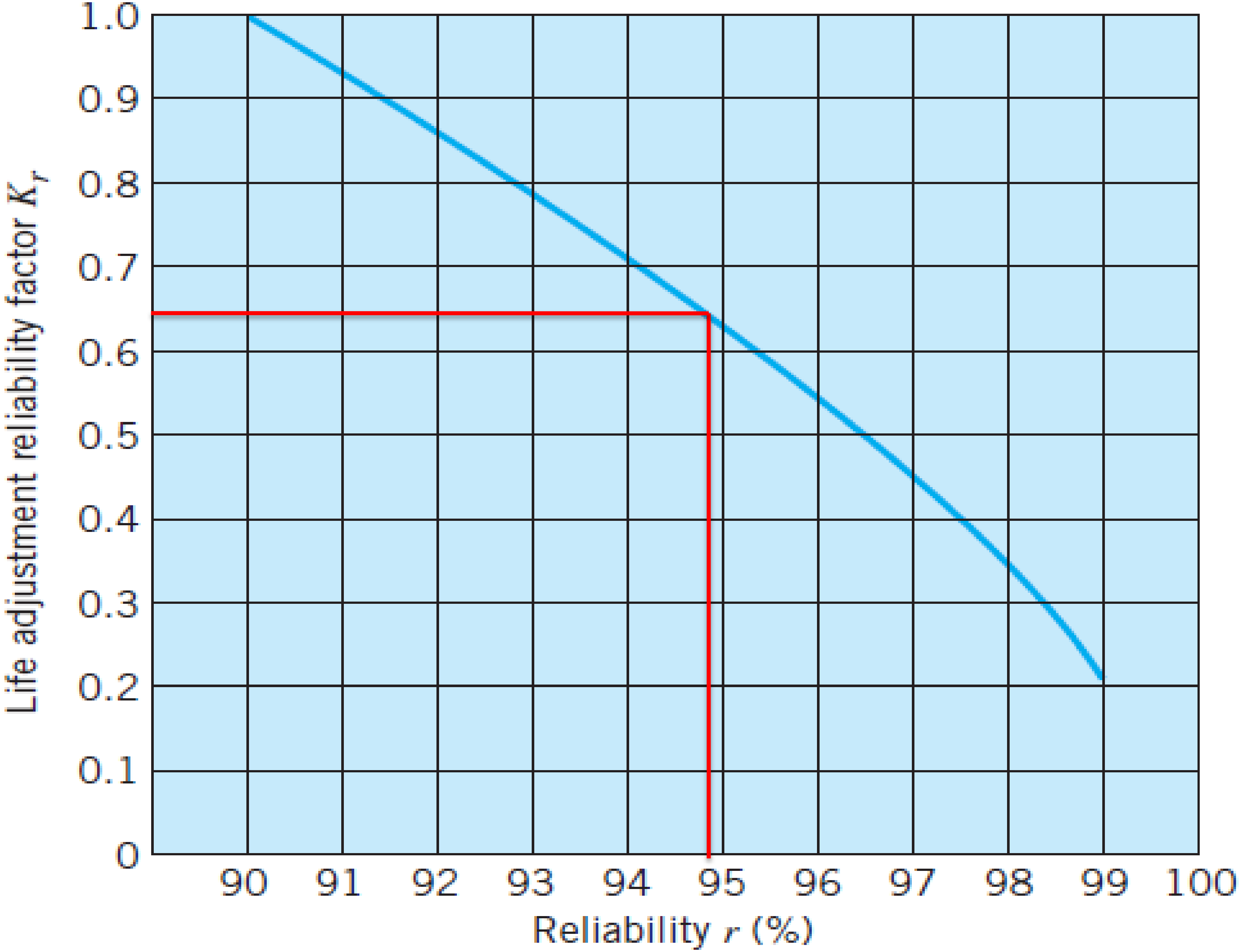
$$L = K_r L_R (C/F_e K_a)^{3.33}$$
$$= (1)(90 \times 10^6)(12.0/3.6)^{3.33} = 4959 \times 10^6 \text{ rev} = 45{,}920 \text{ h}$$

b. From Eq. 14.5a,

$$3240 \times 10^6 = K_r (90 \times 10^6)(12.0/3.6)^{3.33}$$
$$K_r = 0.65$$

From Figure 14.13, reliability is estimated as close to 95 percent.

Comment: For a 90 percent reliability, the bearing life is 45,920 hours. But for a 95 percent reliability, the bearing life is 30,000 hours.


Life adjustment reliability factor K_r
1.0
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0
90
91
92
93
94
95
96
97
98
99
100
Reliability r (%)

MECH 344/M
Machine Element Design

Time: M _ _ _ _14:45 - 17:30

Review

Contact details

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Outline of the course

8-Jan	**week 1**	**Introduction to Design:** An overview of the subject, Machine Design Process	**-**
15-Jan	week 2	Fundamental Topics from Mechanics of Materials: Stresses due to Axial, Bending, Direct Shear, Transverse Shear and Torsional Loadings; Curved Beams; Combined Stresses-Mohr Circle; Stress Concentration Factors; Residual Stresses; Thermal Stresses	4 (must be reviewed by students)
22-Jan	week 3	Static Failure Theories: Failure of Ductile Materials under Static Loading (Maximum Shear Stress Theory, Maximum Distortion Energy Theory); Failure of Brittle Materials under Static Loading (Modified Mohr Theory)	6 (6.5-6.12)
29-Jan	week 4	Fatigue Failure Theories: Basic Concepts and Standard fatigue Test; Fatigue Strengths for Reversed Bending, Reversed Axial Loading and Reversed Torsional Loading; Fatigue Strength for Reversed Biaxial Loading; Influence of Surface and Size on Fatigue Strength; Effect of Mean Stress on Fatigue Strength; Effect of Stress Concentration; Fatigue Life Prediction with Randomly Varying Loads	8 (8.1-8.12)
5-Feb	week 5		
12-Feb	week 6	Design of Screws and Fasteners: Thread Forms, Terminology and Standards; Power Screws; Screw Stresses; Threaded Fasteners; Fasteners Materials and Methods of Manufacture; Bolt Tightening and Initial Tension; Bolt Tension with External Joint-Separating Force; Bolt Selection for Static Loading; Bolt Selection for Fatigue Loading	10
26-Feb	week 7		
5-Mar	week 8*	Design of Springs: Coil Spring Stress and Deflection; Stress and Strength Analysis for Helical Compression Springs-Static Loading; End Designs of Helical Compression Springs; Bucking Analysis of Helical Compression Springs; Design Procedure for Helical Compression Springs-Static Loading; Design of Helical Compression Springs for Fatigue Loading	12 (12.1-12.8)
12-Mar	week 9	Design of Shafts and Keys: Shaft Loads; Attachments and Stress Concentrations; Shaft Stresses; Rotating-Shaft Dynamics; Overall Shaft Design; Keys	17 (17.1-17.6)
19-Mar	week 10	Design of Spur Gears: Geometry and Nomenclature; Interference and Contact Ratio; Gear Force Analysis; Gear-Tooth Strength; Gear-Tooth Bending Fatigue Analysis- Basic Concepts and Recommended Procedure; Gear Tooth Surface Fatigue Analysis-Basic Concepts and Recommended Procedure	15 (15.1-15.12)
26-Mar	week 11		
9-Apr	week 12	Design of Journal and Rolling-Element Bearings: Rolling-Element Bearing Types; Fitting of Rolling-Element Bearings; Catalogue Information for Rolling-Element Bearings; Bearing Selection based on Fatigue Life Requirement	14
16-Apr	week 13	Review	
5^{th} March, Term Test 1 will be during lecture, and 5^{th} April Term Test 2 will be during Tutorial			

Grading Scheme

- Grade composition:
  - Two Term Tests : 40%
  - Final: 60%

- To pass the course you have to
  - *Pass the final*
  - *Attended the term tests and got good marks*
    - *Students that have medical note, for not attending term tests, the weightage will be shifted towards final*

General Notes

- In order to pass the course you have to obtain at least 50% of mark from the Final Exam.
- Electronic communication devices (including cell phones) are not allowed in examination rooms.
- Only “Faculty Approved Calculators” will be allowed in examination rooms.

Final Test

- The final exam will have problems similar to the ones in tutorials/suggested problems
- There will be 5 problems to solve in 3 hours
- Date 25th April 2018, Wednesday (9:00 to 12:00 hrs)
- Write the final exam with confidence that you will do very well
- It is IMPERATIVE to pass the final to pass the course