The life factor adjusts the allowable stress for the required number of operation. Table 3.2 defines values of K_l. Beyond 10^7 cycles the allowable stress may have to be further derated. A value of K_l of 0.8 might be chosen for 10^{10} cycles.

K_t = temperature factor. The temperature factor is usually taken as 1.0 unless the oil or gear blank temperature exceeds 250°F.
K_r = reliability factor. The allowable bending stress values in Table 3.1 reflect a failure probability of fewer than 1 in 100 in 10^7 load cycles. If a lower statistical probability of failure is desired, K_r must be greater than 1.0. A K_r of 1.25 may reflect a failure probability of 1 in a 1000 and a K_r of 1.5 may reflect a failure probability of 1 in 10000.

Power Rating

Quite often, gears are rated on the basis of power. Equation (3.1) can be manipulated to the following:

$$P_{at} = \frac{n_p \cdot PD \cdot K_v}{126{,}000K_a} \quad \frac{F}{K_m} \quad \frac{J}{K_s P_d} \quad \frac{S_{at}K_l}{K_r K_t} \tag{3.2}$$

where

P_{at} = allowable transmitted power on the basis of bending strength, hp
n_p = pinion speed (high-speed member), rpm
PD = pinion pitch diameter, in.

Overloads

When a gear is subjected to infrequent momentary high overloads, the transient stress should be compared to the yield strength of the material. A factor of safety K_r, ranging from 1.33 for conventional industrial gearing to 3.0 for very high reliability, should be applied. Examples of infrequent overloads are equipment that experiences high starting loads once a day or construction machinery that occasionally stalls, incurring high power drain at low speed.

Reverse Bending

When gear teeth experience reverse bending (loading in both directions), such as in an idler or planet gear, 70% of the allowable fatigue strength should be used.

DURABILITY RATING

The durability rating of gear teeth concerns itself with fatigue pitting resistance. Equations based on work by Herz are used to calculate contact (compressive) stresses between the mating gear teeth (Figure 3.6). Gear tooth contact conditions are similar to those between two cylinders except that on gear teeth the radii of curvature are continuously changing. A specific mesh point, such as at the pitch diameters, is chosen at which to make the calculation. Although it is the surface stress which is calculated, subsurface shear stresses proportional to the surface compressive stress are the actual cause of crack initiation. The fundamental equation for compressive stress in a gear mesh is [4]

$$S_c = C_p \sqrt{\frac{W_T}{PD \cdot F} \quad \frac{1.0}{I}} \quad \text{psi}$$

$$C_p = \text{elastic coefficient} = \sqrt{\frac{1.0}{\pi\left[\left(\frac{1-\mu_p^2}{E_p}\right) + \left(\frac{1-\mu_g^2}{E_g}\right)\right]}}$$

where

E_p = pinion modulus of elasticity, psi
E_g = gear modulus of elasticity, psi
μ_p = pinion Poisson's ratio
μ_g = gear Poisson's ratio

For steel-on-steel gears the elastic coefficient C_p = 2300 psi.

I = geometry factor

The geometry factor deals with the radii of curvature at the point of contact and the load sharing between teeth:

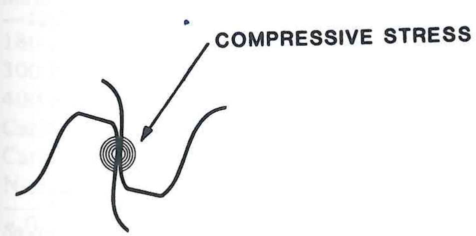


Figure 3.6 Compressive stress criterion for durability rating.

$$I = \frac{C_c}{M_n}$$

where

C_c = curvature factor
M_n = load sharing ratio

For helical gears the compressive stress is usually calculated at the pitch line, where

$$C_c = \frac{\cos\phi_t \sin\phi_t}{2.0}\left(\frac{M_g}{M_g \pm 1.0}\right)$$ + for external gear mesh, - for internal gear mesh

where

ϕ_t = transverse pressure angle, deg
M_g = gear ratio (number of gear teeth divided by number of pinion teeth, always greater than 1.0)

For spur gears the compressive stress is usually calculated at the start of single tooth contact. The curvature factor C_c must be multiplied by a modification factor C_x:

$$C_x = \frac{R_1 R_2}{R_P R_G}$$

where

R_1 = radius of curvature at the lowest point of single tooth contact on the pinion, in.
R_2 = radius of curvature at the highest point of single tooth contact on the gear, in.
R_P = radius of curvature at the pinion pitch diameter, in.
R_G = radius of curvature at the gear pitch diameter, in.

The method of calculation of these radii of curvature was discussed in Chapter 2.

M_n = load-sharing ratio, which depends on the profile and face contact ratios
M_n = 1.0 for spur gears
M_n = F/L_{min} for helical gears
L_{min} = total length of lines of contact (minimum), in.

A procedure for calculating L_{min} was presented in Chapter 2. An estimate for L_{min} which holds for most helical gears when the face contact ratio exceeds 2.0, or when the face contact ratio or transverse contact ratio is an integer of 1.0 or greater, is

$$M_n = \frac{p_n}{0.95Z}$$

where

p_n = normal base pitch, in.
Z = length of action in the transverse plane, in. (see Figure 2.15)

COMPRESSIVE STRESS RATING

In the AGMA rating system the basic compressive stress is modified by several factors that deal with characteristics of a specific application:

$$S_c = C_p \sqrt{\frac{W_t C_a}{C_v} \frac{C_s}{PD \cdot F} \frac{C_m C_f}{I}} \quad \text{psi} \tag{3.3}$$

where C_f is the surface condition factor. This factor depends on the tooth finish, residual stresses, and work-hardening effects. For good-quality gearing it is taken as 1.0. The C_a, C_v, C_s, and C_m factors have the same values as the corresponding K_a, K_v, K_s, and K_m factors discussed previously.

The relation of calculated compressive stress to the allowable stress of the material is

$$S_c \leqq S_{ac} \frac{C_l C_h}{C_t C_r}$$

where

S_{ac} = allowable contact stress, psi (see Table 3.3)
C_h = hardness ratio factor. When the pinion is significantly harder than the gear, there is a work-hardening effect on the gear and the gear

Table 3.3 Allowable Contact Stress S_{ac} for Gear Steels

Material hardness	S_{ac} (psi)
180 Bhn	85,000–95,000
300 Bhn	120,000–135,000
400 Bhn	155,000–170,000
Carburized R_c 55	180,000–200,000
Carburized R_c 60	200,000–225,000
Nitrided R_c 60	192,000–216,000

Source: Ref. 4.

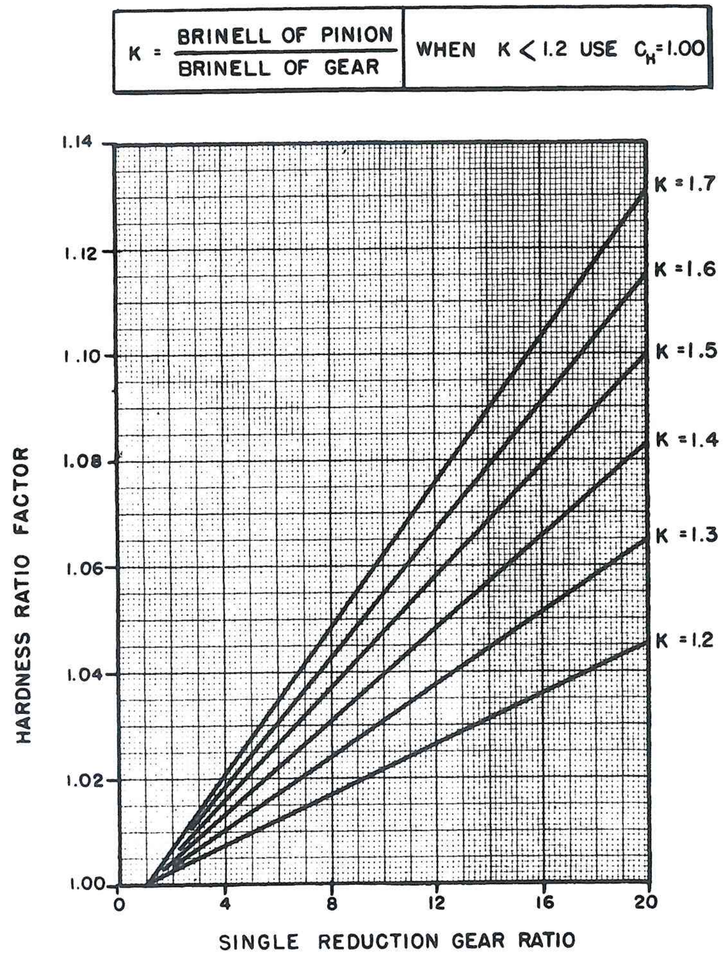


Figure 3.7 Hardness ratio factor. (From Ref. 4.)

durability rating is increased. This effect increases with the reduction ratio as shown in Figure 3.7.

C_1 = life factor. The life factor adjusts the allowable stress for the required number of cycles of operation. A factor of 1.0 is used for 10^7 cycles. For 10^4 cycles and less a factor of 1.5 can be used. Insufficient data are available to define C_1 beyond 10^7 cycles; however, it appears there is no well-defined endurance limit and allowable stress should be decreased as load cycles accumulate over 10^7. A life factor of 0.7 might be used for 10^{10} cycles.

The temperature factor C_t and the reliability factor C_r have the same values as the corresponding K_t and K_r factors discussed previously.

Power Rating

Quite often, gears are rated on the basis of power. Equation (3.3) can be manipulated to the following:

$$P_{ac} = \frac{n_p F}{126{,}000} \frac{IC_v}{C_s C_m C_f C_a} \left(\frac{S_{ac} \cdot PD}{C_p} \frac{C_l C_h}{C_t C_r}\right)^2 \quad \text{hp} \tag{3.4}$$

where P_{ac} is the allowable transmitted horsepower on the basis of compressive stress.

AGMA STANDARDS FOR ENCLOSED DRIVE RATINGS

AGMA Standard 420.04 [1] covers enclosed drives with pitch line velocities not exceeding 5000 fpm and pinion speeds not exceeding 3600 rpm. Higher-speed enclosed drives are covered by AGMA Standard 421.06 [2], which deals with helical and herringbone gear units. These Standards basically use Eqs. (3.2) and (3.4) for strength and durability rating but simplify the many modifying factors and arrive at a service factor based on transmitted power rather than a reliability factor based on stress.

For the strength rating Standard 420.04 states that

$$P_{at} = \frac{K_1 K_2 K_3 J}{P_d}$$

$$K_1 = \frac{n_p \cdot PD \cdot K_v}{126{,}000}$$

$$K_2 = \frac{F}{K_m}$$

$$K_3 = S_{at} K_1$$

where

P_{at} = horsepower rating based on tooth strength
J = geometry factor (Figures 3.4a, 3.4b, and 3.4c)
P_d = diametral pitch–transverse, in.$^{-1}$
n_p = pinion rpm
PD = pinion pitch diameter, in.
K_v = $50/(50 + \sqrt{PLV})$ for spur gears

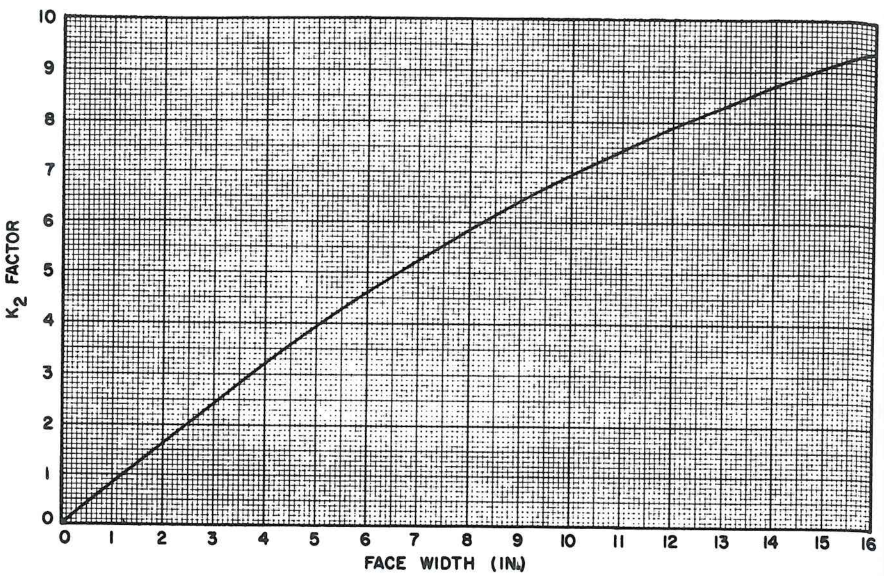


Figure 3.8 K_2 factor for strength rating of spur and helical gearings. (From Ref. 1.)

K_v = $\sqrt{78/(78 + \sqrt{PLV})}$ for helical gears
PLV = pitch line velocity, fpm
K_2 = face width alignment factor as specified in Figure 3.8
K_3 = allowable fatigue stress times life factor as specified in Figures 3.9 and 3.10

For the durability rating Standard 420.04 states:

$$P_{ac} = C_1 C_2 C_3 C_4 \tag{3.5}$$

$$C_1 = \frac{n_p \cdot PD^2 \cdot C_v}{126{,}000}$$

$$C_2 = \frac{F}{C_m}$$

$$C_3 = I(S_{ac}/C_p)^2$$

$$C_4 = (C_1)^2$$

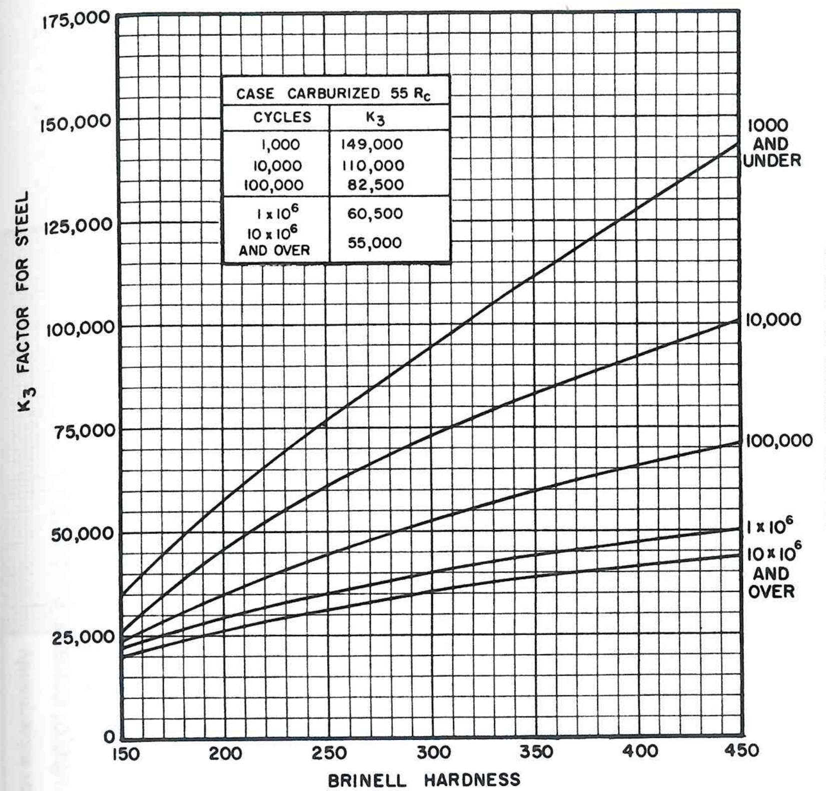


Figure 3.9 K_3 factor for strength rating of spur gears. (From Ref. 1.)

where

P_{ac} = horsepower rating based on tooth durability
n_p = pinion rpm
PD = pinion pitch diameter, in.
C_v = $50/(50 + \sqrt{PLV})$ for spur gears
C_v = $78/(78 + \sqrt{PLV})$ for helical gears
PLV = pitch line velocity, fpm
C_2 = face width/alignment factor as specified in Figure 3.11
I = geometry factor (see also "Durability" section)
$(S_{ac}/C_p)^2$ = allowable contact stress/elastic coefficient as specified in Figure 3.12
C_4 = life factor² as defined in Figure 3.13

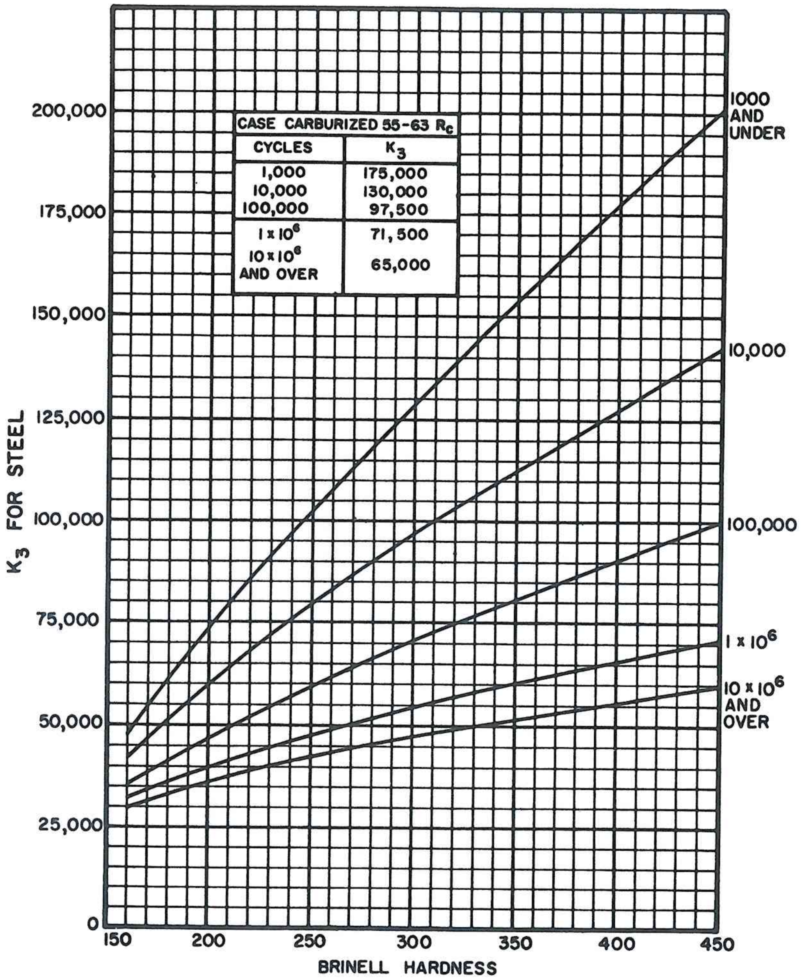


Figure 3.10 K_3 factor for strength rating of helical gears. (From Ref. 1.)

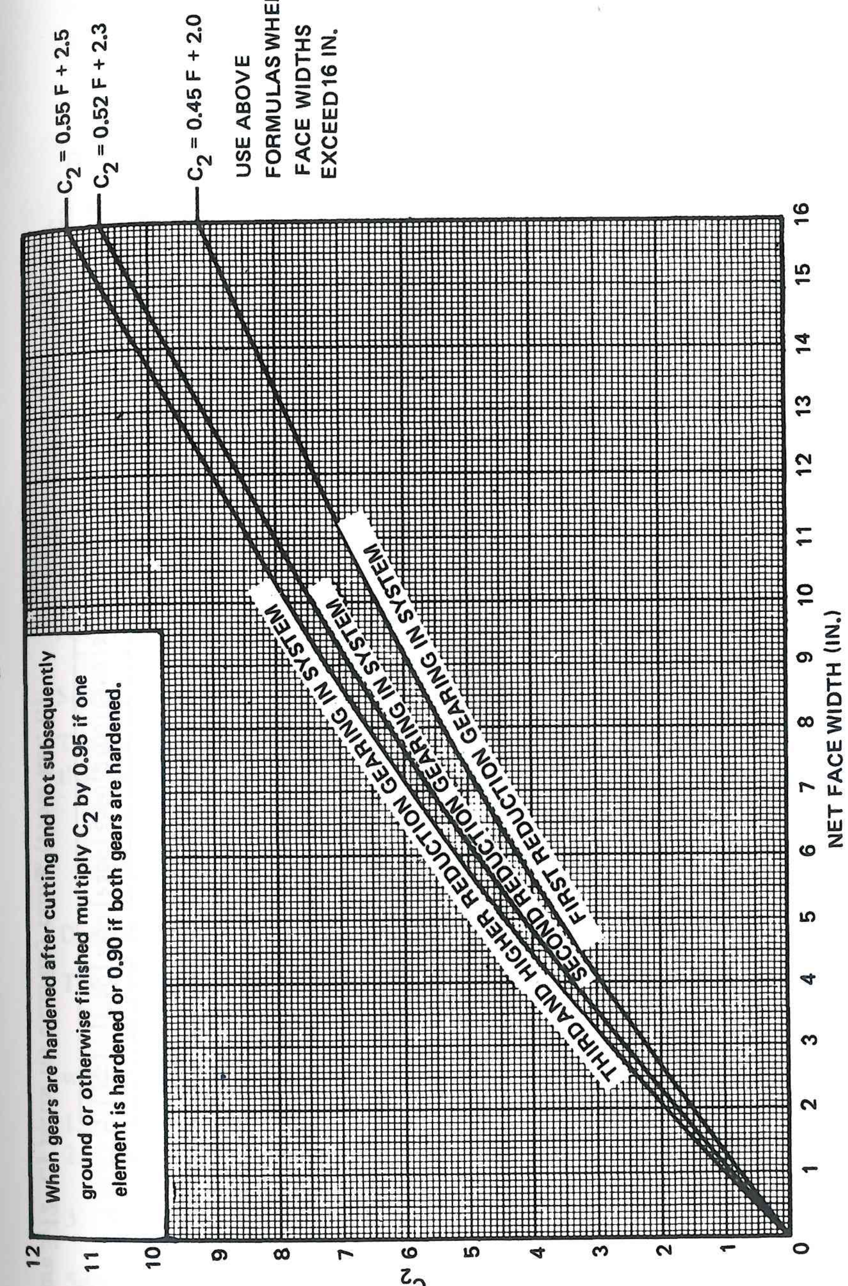
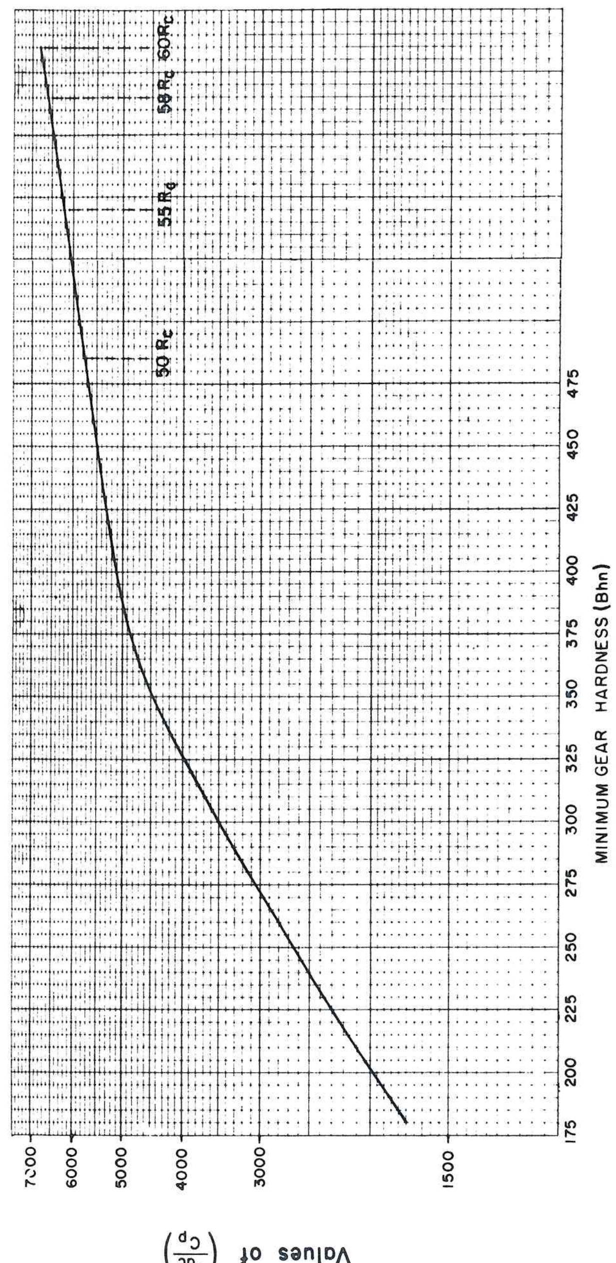


Figure 3.11 C_2 factor for durability rating. (From Ref. 1.)



VALUES ARE TO BE TAKEN FROM THE ABOVE CURVE FOR THE MINIMUM HARDNESS SPECIFIED FOR THE GEAR.
VALUES FOR SUGGESTED GEAR AND PINION HARDNESS COMBINATIONS ARE TABULATED BELOW FOR CONVENIENCE.

	MINIMUM BRINELL HARDNESS												
GEAR	180	210	225	245	255	270	285	300	335	350	375	55 Rc	58 Rc
PINION	210	245	265	285	295	310	325	340	375	390	415	55 Rc	58 Rc
$\left(\frac{s_{ac}}{C_p}\right)^2$	1750	2100	2300	2560	2700	2950	3200	3460	4200	4460	4820	6200	6600

Figure 3.12 C_3 factor for durability rating. (From Ref. 1.)

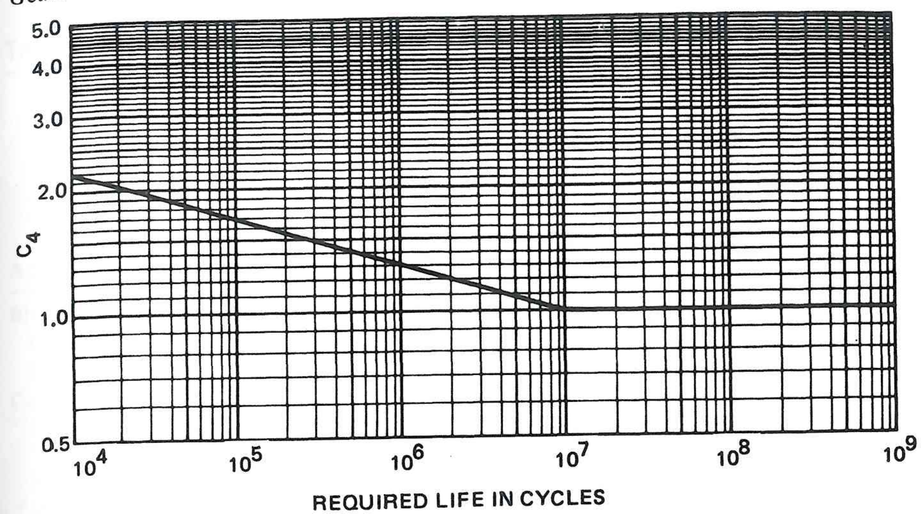


Figure 3.13 C_4 factor for durability rating. (From Ref. 1.)

HIGH-SPEED GEARING

Strength

The high-speed standard, Standard 421.06, uses Eqs. (3.2) as it stands except that the dynamic factor K_v is defined as follows:

$$K_v = \sqrt{78/(78 + \sqrt{PLV})} \text{ for pitch line velocities below 7000 fpm}$$

$$K_v = 0.695 \text{ for pitch line velocities 7000 fpm and higher}$$

Durability

The 421.06 Standard uses Eq. (3.5):

$$P_{ac} = C_1 C_2 C_3 C_4$$

with three modifications:

1. $C_4 = 1.0$.
2. The dynamic factor C_v is 0.48 for pitch line velocities of 7000 fpm and higher.
3. The C_2 factor is defined in Figure 3.14.

Service Factors

When P_{at} and P_{ac} are calculated, the rating of the gearbox will be based on the smaller of the strength and durability ratings. A service factor must now be

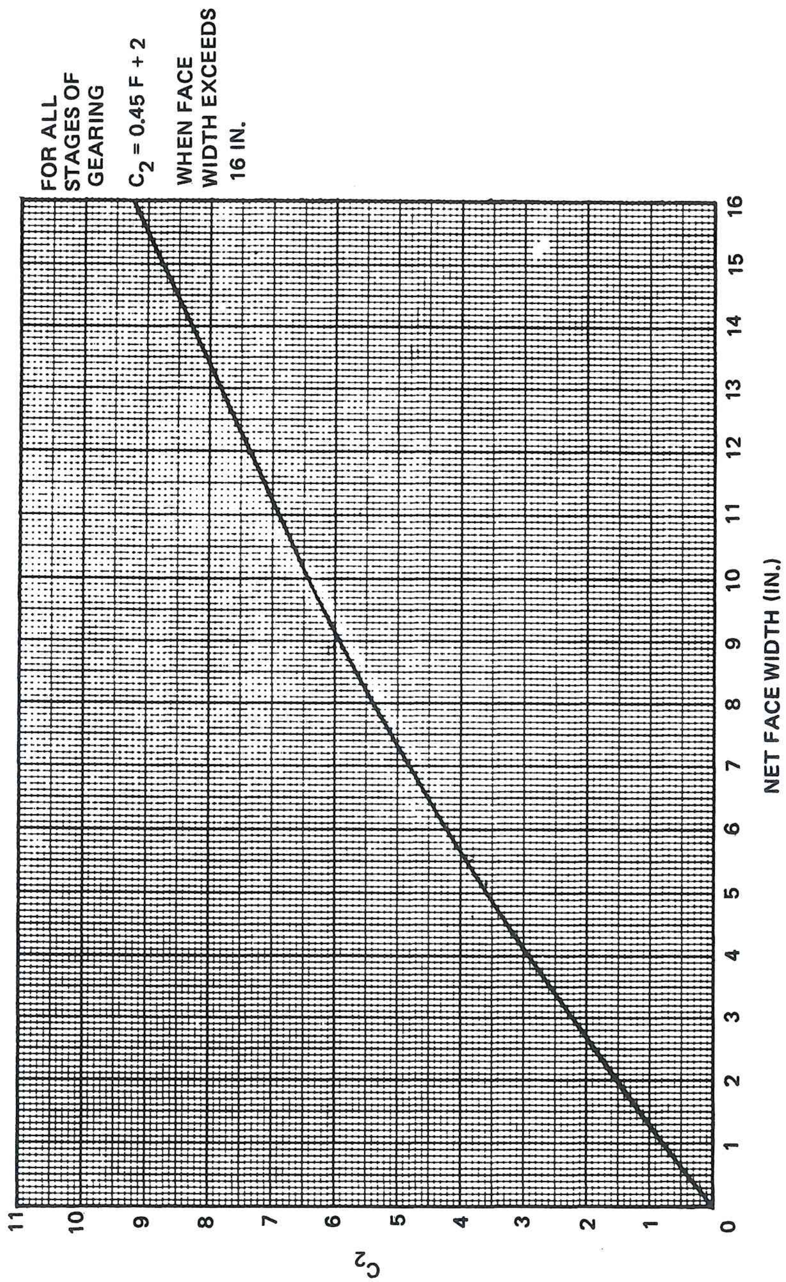


Figure 3.14 C_2 factor for high-speed-unit durability rating. (From Ref. 2.)

Table 3.4 Service Factors for High-Speed Units

Application	Service factor — Prime mover: Motor	Turbine	Internal combustion engine (multicylinder)
Blowers			
Centrifugal	1.4	1.6	1.7
Lobe	1.7	1.7	2.0
Compressors			
Centrifugal—process gas except air conditioning	1.3	1.5	1.6
Centrifugal—air conditioning service	1.2	1.4	1.5
Centrifugal—air or pipeline service	1.4	1.6	1.7
Rotary—axial flow, all types	1.4	1.6	1.7
Rotary—liquid piston (Nash)	1.7	1.7	2.0
Rotary—lobe-radial flow	1.7	1.7	2.0
Reciprocating—3 or more cylinders	1.7	1.7	2.0
Reciprocating—2 cylinders	2.0	2.0	2.3
Dynamometer—test stand	1.1	1.1	1.3
Fans			
Centrifugal	1.4	1.6	1.7
Forced draft	1.4	1.6	1.7
Induced draft	1.7	2.0	2.2
Industrial and mine (large with frequent start cycles)	1.7	2.0	2.2
Generators and Exciters			
Base load or continuous	1.1	1.1	1.3
Peak duty cycle	1.3	1.3	1.7
Pumps			
Centrifugal (all service except as listed below)	1.3	1.5	1.7
Centrifugal—boiler feed	1.7	2.0	
Centrifugal—descaling (with surge tank)	2.0	2.0	
Centrifugal—hot oil	1.5	1.7	
Centrifugal—pipeline	1.5	1.7	2.0
Centrifugal—water works	1.5	1.7	2.0
Dredge	2.0	2.4	2.5
Rotary—axial flow, all types	1.5	1.5	1.8
Rotary—gear	1.5	1.5	1.8
Rotary—liquid piston	1.7	1.7	2.0
Rotary—lobe	1.7	1.7	2.0

Table 3.4 (Continued)

Application	Service factor — Prime mover: Motor	Turbine	Internal combustion engine (multicylinder)
Rotary–lobe	1.7	1.7	2.0
Rotary–sliding vane	1.5	1.5	1.8
Reciprocating–3 cylinders or more	1.7	1.7	2.0
Reciprocating–2 cylinders	2.0	2.0	2.3
Paper industry			
Jordan or refiner	1.5	1.5	
Paper machine–line shaft	1.3	1.3	
Paper machine–sectional drive	1.5		
Pulp beater	1.5		
Sugar industry			
Cane knife	1.5	1.5	1.8
Centrifugal	1.5	1.7	2.0
Mill	1.7	1.7	2.0

Source: Ref. 2.

assigned to the application in order to determine the horsepower that the unit will be rated against. The specified or actual horsepower transmitted through the gearbox is multiplied by the service factor and called the equivalent or service horsepower. The smaller of P_{ac} and P_{at} must equal or exceed the equivalent horsepower:

$$\text{Service factor} = \frac{\text{smaller of } P_{ac} \text{ or } P_{at}}{\text{specified horsepower}}$$

The AGMA and API Standards [1-3] contain tables of recommended service factors for various applications. Table 3.4 illustrates the numerical values of service factors for high-speed units. The API Standard has very similar values. Lower-speed gearboxes covered by AGMA Standard 420.04 have service factors as shown in Table 3.5, which takes into account the duration of service, prime mover characteristics, and driven machine characteristics.

The AGMA and API standards are good basic guidelines which, when followed, result in gearboxes that perform successfully. They provide a baseline with which to judge competing designs. The enclosed drive standards [1-3] will give more conservative results than the general standard [4], but this is to be

Table 3.5 Service Factors for Low-Speed Units

Prime mover	Duration of service	Driven machine load classification: Uniform	Moderate shock	Heavy shock
Electric motor, steam turbine, or hydraulic motor	Occasional ½ hr per day	0.50	0.80	1.25
	Intermittent 3 hr per day	0.80	1.00	1.50
	Over 3 hr up to and incl. 10 hr per day	1.00	1.25	1.75
	Over 10 hr per day	1.25	1.50	2.00
Multicylinder internal combustion engine	Occasional ½ hr per day	0.80	1.00	1.50
	Intermittent 3 hr per day	1.00	1.25	1.75
	Over 3 hr up to and incl. 10 hr per day	1.25	1.50	2.00
	Over 10 hr per day	1.50	1.75	2.25
Single cylinder internal combustion engine	Occasional ½ hr per day	1.00	1.25	1.75
	Intermittent 3 hr per day	1.25	1.50	2.00
	Over 3 hr up to and incl. 10 hr per day	1.50	1.75	2.25
	Over 10 hr per day	1.75	2.00	2.50

Source: Ref. 1.

expected since the enclosed drive standards reflect gear manufacturer's field experience and are based on practical as well as theoretical considerations.

A drawback of the enclosed drive standards is that they do not quantify the effects of component quality or metallurgy in the rating procedures. A given gearbox may contain gears with better metallurgical characteristics or closer tolerances than another, yet both units could claim the same rating according to existing standards. There is work proceeding to improve the standards. If an application is considered critical, the user should determine the specific quality and metallurgical characteristics required and identify them in the procurement document rather than simply call for a unit complying with a particular standard.

Very high speed gearing operating above 20,000 fpm pitch line velocity requires analysis beyond that presented in industry standards. If a procedure such as that outlined in the API standard is used to size a very high speed unit, the gear diameters that result may become so large that centrifugal effects may cause stresses and deflections that override the conventional strength and durability considerations. In some high-speed applications it is preferable to minimize pitch line velocity at the expense of higher bending and compressive stresses. These types of units must be carefully designed and incorporate the highest-quality components with the best metallurgy available.

API STANDARD

The American Petroleum Institute standard [3], like the AGMA standards, is widely used to procure gear units. There are two editions, the first issued in 1968 [8] and the second in 1977 [3]. The first edition rating method conforms to the AGMA high-speed standard. The second edition rating method is more conservative. Gear units are sized on the basis of a tooth pitting index called the K factor. K-factor ratings are common in the industry and often used as a simple rating index:

$$K = \frac{W_t}{F \cdot PD}\left(\frac{M_g \pm 1}{M_g}\right) \quad \text{+ for external gear meshes, - for internal gear meshes}$$

where

- K = tooth pitting index
- W_t = transmitted tangential load, lb
- F = net face, width, in.
- PD = pinion pitch diameter, in.
- M_g = gear ratio

Table 3.6 Material Index Numbers and Maximum L/d Values[a]

Gear hardness minimum	Pinion hardness minimum	Material index number	Maximum pinion L/d: Double-helical	Maximum pinion L/d: Single-helical
223 Bhn	262 Bhn	130	2.4	1.7
262 Bhn	302 Bhn	160	2.3	1.6
302 Bhn	341 Bhn	200	2.2	1.5
352 Bhn	50 R_c (nitrided)	260	2.0	1.45
50 R_c (nitrided)	50 R_c (nitrided)	300	1.9	1.4
55 R_c (carburized)	55 R_c (carburized)	410	1.7	1.35
58 R_c (carburized)	58 R_c (carburized)	440	1.6	1.3

[a]L, net face width plus gap;
d, pinion pitch diameter.
Source: Ref. 3.

The allowable K is the material index number shown in Table 3.6, divided by the service factor:

$$K = \frac{\text{material index number}}{\text{service factor}}$$

Also shown in Table 3.6 are maximum allowable face width/pinion pitch diameter ratios. In the case of double helical gears the total face width L is the net face width plus the gap between the helices. The bending stress is calculated as follows:

$$S_t = \frac{W_t P_{nd}}{F} \text{ (SF) } \frac{1.8 \cos \psi}{J} \quad \text{psi}$$

where

- S_t = bending stress, psi
- P_{nd} = normal diametral pitch, in.$^{-1}$
- F = net face width, in.
- SF = service factor
- ψ = helix angle, deg
- J = geometry factor

The bending stress must not exceed the values shown in Figure 3.15.

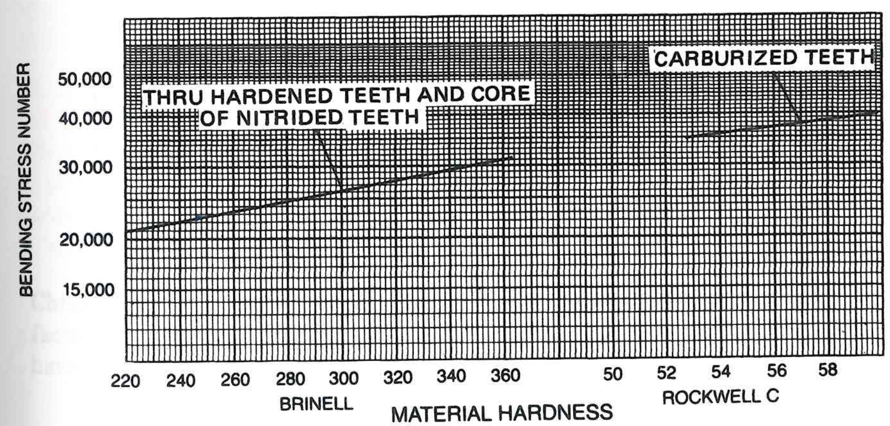


Figure 3.15 Bending stress allowables. (From Ref. 3.)

SAMPLE RATING CASE

Let us look at a specific application and see how the various rating procedures compare. Consider the first mesh of a two-stage high-speed reducer connecting a gas turbine with a generator having the following characteristics:

hp = 4500 hp

Pinion rpm = 14,500 rpm

Pinion pitch diameter = 4.625 in.

Number of pinion teeth = 37

Gear pitch diameter = 14.875 in.

Number of gear teeth = 119

Face width = 6 in.

Transverse pressure angle = 20.3439°

Helix angle = 11.0°

Transverse diametral pitch = 8.0

The gears are single helical, carburized, hardened to R_c 60 and ground to AGMA Quality Class 12. The tangential load is

$$W_T = \frac{63{,}025(4500)}{14{,}500} \times \frac{2}{4.625} = 8458 \text{ lb}$$

The pitch line velocity is

$$PLV = \frac{14{,}500(4.625\,\pi}{12} = 17{,}557 \text{ fpm}$$

Using Eq. (3.1), the bending stress is calculated:

K_a = 1.0; smooth operation is anticipated

K_v = 0.94; extrapolated from Figure 3.5

K_s = 1.0

K_m = 1.32, assuming accurate gears and rigid mounting

$$\left.\begin{array}{l} J = 0.54 \text{ for the pinion} \\ J = 0.56 \text{ for the gear} \end{array}\right\} \text{Figures 3.4b and 3.4c}$$

$$S_t = \frac{8458(1.0)}{0.94} \times \frac{8}{6} \times \frac{1.0(1.32)}{0.54} = 29{,}326 \text{ psi}$$

Choosing a life factor K_l = 0.8, a temperature factor K_t = 1.0, and an allowable material stress of 70,000 psi (Table 3.1), we have

$$K_r = \frac{70{,}000(0.8)}{(1.0)(29{,}326)} = 1.91$$

which is a very low probability of failure.

If a reliability factor of 1.0 (a failure probability of 1 in 100) is chosen, the allowable transmitted power on the basis of bending strength from Eq. (3.2) is

$$P_{at} = \frac{14{,}500(4.625)(0.94)}{126{,}000(1.0)} \times \frac{6}{1.32} \times \frac{0.54}{1.0(8)} \times \frac{70{,}000(0.8)}{1.0(1.0)}$$
$$= 8596 \text{ hp}$$

To calculate the compressive stress, use Eq. (3.3):

$$I = \frac{C_c}{M_n}$$

$$C_c = \frac{\cos 20.3439° \sin 20.3439°}{2} \times \frac{3.22}{4.22} = 0.1244$$

$$M_n = \frac{F}{L_{min}} = \frac{6}{10.75} = 0.558$$

I = 0.2229

C_a = 1.0

C_v = 0.94

C_s = 1.0

C_m = 1.32

C_f = 1.0

$$S_c = 2300\sqrt{\frac{8458(1.0)}{0.94} \times \frac{1.0}{4.625(6)} \times \frac{1.32(1.0)}{0.2229}} = 100{,}800 \text{ psi}$$

Choosing a life factor C_l = 0.8, a hardness factor C_h = 1.0, a temperature factor C_t = 1.0, and an allowable material stress of 225,000 psi (Table 3.3), we have

$$C_r = \frac{225{,}000(0.8)(1.0)}{100{,}800(1.0)} = 1.79$$

Since C_r is lower than K_r, this gear mesh must be rated on the basis of durability rather than tooth strength at the 4500-hp level.

If a reliability factor C_r of 1.0 (a failure probability of 1 in 100) is chosen, the allowable transmitted power on the basis of durability from Eq. (3.4) is

$$P_{ac} = \frac{14{,}500(6)}{126{,}000} \times \frac{0.2229(0.94)}{1.0(1.32)(1.0)(1.0)} \times \left[\frac{225{,}000(4.625)}{2300} \times \frac{0.8(1.0)}{1.0(1.0)}\right]^2 = 14{,}360 \text{ psi}$$

As the transmitted power increases, the bending stress increases linearly and the compressive stress increases as the square root of the horsepower; therefore, at a power point of approximately 6300 hp, the bending stress becomes critical. With a reliability factor C_r of 1.0 the bending stress is more critical than the compressive stress and the allowable transmitted power is 8596 hp. The definition of C_r as the reliability factor is relatively new. In the past it has been termed the factor of safety.

On the basis of AGMA Standard 421.06[2], the strength rating is

$$P_{at} = \frac{14{,}500(4.625)(0.695)}{126{,}000(1.0)} \times \frac{6}{1.32} \times \frac{0.54}{(1.0)(8)} \times \frac{70{,}000(0.8)}{(1.0)(1.0)}$$

$$= 6355 \text{ hp}$$

The durability rating as per AGMA 421.06 is

$$C_1 = \frac{14{,}500(4.625)^2(0.48)}{126{,}000} = 1.18$$

$$C_2 = 4.2 \quad \text{(Figure 3.14)}$$

$$C_3 = I(S_{ac}/C_p)^2$$

$$I = 0.2229$$

$$(S_{ac}/C_p)^2 = 6800 \quad \text{(Figure 3.12)}$$

$$P_{ac} = 7513 \text{ hp}$$

The service factor at 4500 hp is

$$SF = \frac{6355}{4500} = 1.41$$

On the basis of API Standard 613 [3], the strength rating is

$$S_t = \frac{8458(8.15)}{6} \text{ (SF) } \frac{1.8 \cos 11^\circ}{0.54}$$

From Figure 3.15 the allowable bending stress number is 40,000 psi and the service factor is

$$SF = \frac{40{,}000(6)(0.54)}{8458(8.15)(1.8)(\cos 11)} = 1.06$$

The API durability rating is

$$K = \frac{8458}{6(4.625)} \times \frac{4.22}{3.22} = 399$$

The allowable material index number from Table 3.4 is 440, and in this case the service factor is less than 1.0:

$$SF = \frac{399}{440} = 0.91$$

SCORING

High-speed gearing (above 5000 fpm or 3600 rpm) operating with low-viscosity lubricants is prone to a failure mode called scoring. In contrast to the classic failure modes pitting and breakage, which generally take time to develop, scoring occurs early in the operation of a gear set and can be the limiting factor in the gear's power capability. Scoring failures and the degree of scoring that may be accepted are described in Chapter 12.

FLASH TEMPERATURE INDEX

The critical total temperature hypothesis (flash temperature index) [9] appears to be the most reliable method of analysis presently used to predict scoring. It states that scoring will occur when a critical total temperature, which is characteristic of the particular combination of lubricant and gear material, is reached.

$$T_f = T_b + \Delta T$$

where

T_f = flash temperature index, °F
T_b = gear blank temperature, °F
ΔT = maximum rise of instantaneous surface temperature in the tooth mesh above the gear blank's surface temperature

The gear blank temperature is difficult to estimate. It may be significantly higher than the bulk oil temperature [10]. The heat transfer capability of the gear must be considered in attempting to estimate this parameter. Often, the blank temperature is approximated as the average of the oil temperature entering and leaving the gearbox.

One form of the fundamental flash temperature index formula is [11]

$$T_f = T_b + \frac{C_f f W (V_1 - V_2)}{(\sqrt{V_1} + \sqrt{V_2})\sqrt{B_c/2}}$$

where

C_f = material constant for conductivity, density, and specific heat
f = friction coefficient
V_1 = rolling velocity of pinion at point of contact, fps
V_2 = rolling velocity of gear at point of contact, fps
B_c = width of band of contact
W = specific loading, normal load divided by face width, lb/in.

For steel on steel gears, taking C_f as 0.0528, f as a constant 0.06, and adding a term taking surface finish into account [10], the following equation results [12]:

$$T_f = T_b + \left(\frac{W_{te}}{F_e}\right)^{3/4} \frac{50}{50 - S} \; Z_t (n_p)^{1/2}$$

where

W_{te} = effective tangential load, lb
F_e = effective face width (use minimum contact length for helical gears), in.
S = surface finish (after running in), rms
n_p = pinion rpm
Z_t = scoring geometry factor

$$Z_t = \frac{0.0175\left[\sqrt{e_p} - \sqrt{\frac{N_p}{N_g} e_g}\right]}{(\cos \phi_t)^{3/4}\left[\frac{e_p e_g}{(e_p + e_g)}\right]^{1/4}}$$

Note: Use absolute value of Z_t

where

e_p = pinion radius of curvature, in.
e_g = gear radius of curvature, in.
N_p = number of pinion teeth (smaller member)

N_g = number of gear teeth (larger member)
ϕ_t = pressure angle, transverse operating

The 50/(50 - s) term was developed by Kelly [10] in an experimental program using gears with surface finish in the range 20 to 32 rms. For gears with surface finish rougher than this range, if the computed value exceeds 3, a factor of 3 should be used. For gears with surface finishes finer than 20, the resulting computed factor may be conservative.

The term W_{te} must be adjusted to allow for the sharing of load by more than one pair of teeth. The following analysis, which modifies the tooth load depending on the position of the gear mesh along the line of action, was developed by Dudley using spur gears of standard proportions [11]. If a more accurate prediction of tooth load sharing is available to the reader, it would be appropriate to use that analysis.

$$W_{te} = KW_t$$

where W_t is the tangential tooth load in pounds.

1. Unmodified tooth profiles

$$K = \frac{1}{3} + \frac{1}{3}\,\frac{\theta - \theta_{LD}}{\theta_L - \theta_{LD}}$$

$$\theta_{LD} \lesssim \theta < \theta_L$$

$$K = 1.0$$

$$\theta_L \lesssim \theta \lesssim \theta_H$$

$$K = \frac{1}{3} + \frac{1}{3}\,\frac{\theta_o - \theta}{\theta_o - \theta_H}$$

$$\theta_H < \theta \lesssim \theta_o$$

2. Modified tooth profiles
   a. Pinion driving

$$K = \frac{6}{7}\,\frac{\theta - \theta_{LD}}{\theta_L - \theta_{LD}}$$

$$\theta_{LD} \lesssim \theta < \theta_L$$

$$K = 1.0$$

$$\theta_L \lesssim \theta \lesssim \theta_H$$

$$K = \frac{1}{7} + \frac{6}{7}\,\frac{\theta_o - \theta}{\theta_o - \theta_H}$$

$$\theta_H < \theta \leq \theta_o$$

b. Pinion driven by gear:

$$K = \frac{1}{7} + \frac{6}{7}\,\frac{\theta - \theta_{LD}}{\theta_L - \theta_{LD}}$$

$$\theta_{LD} \leq \theta < \theta_L$$

$$K = 1.0$$

$$\theta_L \leq \theta \leq \theta_H$$

$$K = \frac{6}{7}\,\frac{\theta_o - \theta}{\theta_o - \theta_H}$$

$$\theta_H < \theta \leq \theta_o$$

where

θ = any pinion roll angle, rad
θ_{LD} = roll angle at the pinion limit (form) diameter, rad
θ_L = roll angle at the lowest point of single tooth contact on the pinion, rad
θ_H = roll angle at the highest point of single tooth contact on the pinion, rad
θ_o = roll angle at the pinion outside diameter, rad

A modified tooth profile would be one that has tip and/or flank relief rather than a true involute form.

The flash temperature index should be calculated at five specific points on the line of action and at several additional points of contact. The five specific points are:

1. Outside diameter of pinion
2. Highest point of single tooth contact
3. Pitch point (flash temperature rise will be zero since there is no sliding)
4. Lowest point of single tooth contact
5. Form (contact diameter) of pinion

Figure 3.16 is a typical plot of flash temperature rise along the line of action.

The most convenient way to generate a plot such as shown in Figure 3.16 is by the use of a computer program. By stepping through successive roll angles, the flash temperature index can be calculated at many points. From Figure 3.16 it can be seen that there will be two peaks, one during the arc of approach (pinion form diameter to pitch diameter) and one during the arc of recess (pinion pitch diameter to outside diameter). To achieve the minimum flash

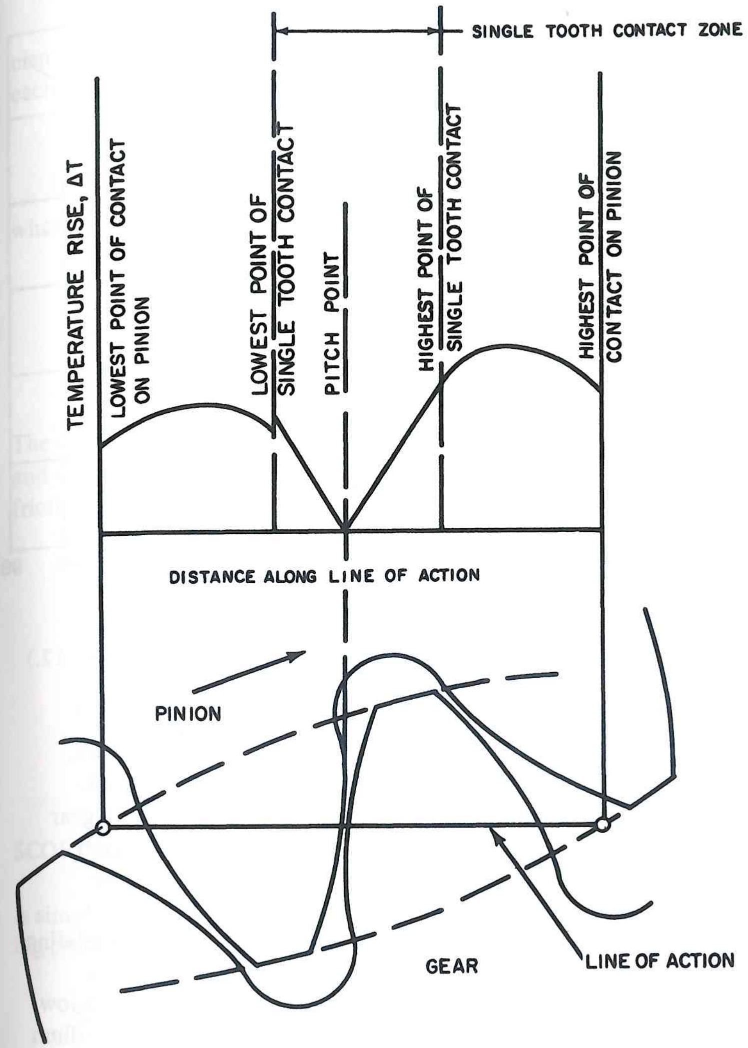


Figure 3.16 Flash temperature rise along the line of action.

temperature index, the flash temperature rise in the arc of approach should be equal to the rise in the arc of recess. An optimum tooth design can be achieved by the use of long and short addendums. The computer program, starting with standard addendums (1/diametral pitch), automatically varies the pinion and gear addendums in defined increments until the optimum flash temperature is

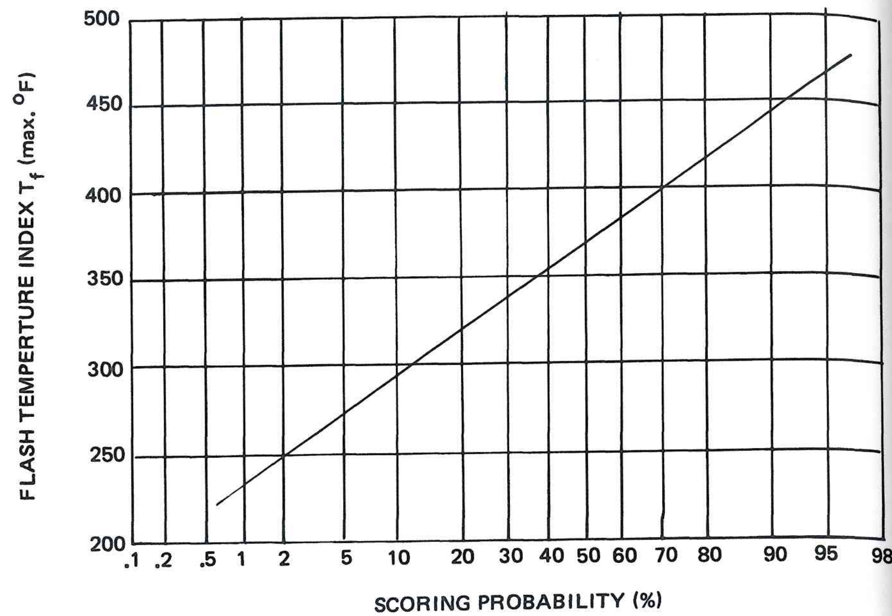


Figure 3.17 Scoring probability versus flash temperature index. (From Ref. 12.)

obtained. With the resulting long and short addendum designs of this nature, standard tooth thicknesses are no longer applicable. If standard tooth thicknesses were utilized, an unbalance of bending stresses between pinion and gear would result. To optimize the bending stresses, the program enters a second iteration procedure, which varies tooth thickness until bending stresses are balanced.

Figure 3.17 presents the results of an aerospace industry survey correlating scoring to flash temperature index. From the data used in the study, it was assumed that a gearset with a calculated index of 276°F or less represents a low scoring risk. A calculated index ranging from 277 to 338°F represents a medium scoring risk where scoring may or may not occur, and a calculated index of 339°F and higher represents a high scoring risk. The data presented in Figure 3.17 reflects cases using Society of Automotive Engineers (SAE) 9310 steel operating with military standard Mil-L-7808 or Mil-L-23699 synthetic oil. The viscosity of these oils is approximately 4 to 6 cSt at 200°F and 18 to 30 cSt at 100°F. Mineral oils such as light turbine oils, used in high-speed industrial applications, are more viscous and therefore may possibly tolerate a higher flash temperature index.

The equation for the flash temperature index assumes a constant coefficient of friction of 0.06. If it is desired to calculate the coefficient of friction at each point on the line of action, the following equation can be used [13]:

$$f = 0.0127 \log_{10} \frac{3.17 \times 10^8}{\mu_0 V_s V_t^2 / W}$$

where

f = coefficient of friction
μ_0 = absolute viscosity, cP
V_s = sliding velocity, ips
V_t = sum velocity, ips
W = specific loading, lb/in.

The equation breaks down at the pitch point, where the sliding velocity is 0.0 and the friction coefficient goes to infinity. Using a variable coefficient of friction the flash temperature index formula becomes

$$T_f = T_b + f \left(\frac{W_{te}}{F_e}\right)^{3/4} \frac{50}{50 - S} \left[Z_t (n_p)^{1/2}\right]$$

$$Z_t = \frac{0.2917 \left[\sqrt{e_p} - \sqrt{\frac{N_p}{N_g}}\, e_g\right]}{(\cos \theta_t)^{3/4} \left[\frac{e_p e_g}{(e_p + e_g)}\right]^{1/4}}$$

SCORING CRITERION NUMBER

A simplified form of the flash temperature index is presented in Ref. 14. A scoring criterion number is defined.

$$\text{Scoring criterion number} = \left(\frac{W_t}{F_e}\right)^{3/4} \frac{n_p^{1/2}}{P_d^{1/4}}$$

where

W_t = tangential driving load, lb
F_e = contacting face width, in.
n_p = pinion rpm
P_d = diametral pitch

Table 3.7 gives scoring criterion numbers for various oils at various gear blank temperatures. If the scoring criterion number is above the values shown in the table, a possibility exists that scoring will be encountered. The gear blank temperature can be taken as the average of the oil-in and oil-out temperatures.

Table 3.7 Critical Scoring Criterion Numbers[a]

Blank temperature (°F):	100	150	200	250	300
Kind of oil	Critical scoring index numbers				
AGMA 1	9,000	6,000	3,000		
AGMA 3	11,000	8,000	5,000	2,000	
AGMA 5	13,000	10,000	7,000	4,000	
AGMA 7	15,000	12,000	9,000	6,000	
AGMA 8A	17,000	14,000	11,000	8,000	
Grade 1065 Mil-0-6082B	15,000	12,000	9,000	6,000	
Grade 1010 Mil-0-6082B	12,000	9,000	6,000	2,000	
Synthetic (Turbo 35)	17,000	14,000	11,000	8,000	5,000
Synthetic (Mil-L-7808D)	15,000	12,000	9,000	6,000	3,000

[a]Scoring number = $\left(\frac{W_t}{F_e}\right)^{3/4} \frac{n_p^{1/2}}{P_d^{1/4}}$

Source: Ref. 14.

MINIMUM FILM THICKNESS CRITERION

Scoring is a phenomenon that will occur when gears are operating in the boundary lubrication regime [15] rather than with a hydrodynamic or elastohydrodynamic oil film separating the gear teeth. The film thickness can be calculated [16-18] and compared to the combined surface roughness of the contacting elements to determine if metal-to-metal contact is likely to occur. A criterion used to determine the possibility of surface distress is the ratio of film thickness to composite surface roughness:

$$\lambda = \frac{h_{min}}{\sigma}$$

$$\sigma = \sqrt{\sigma_p^2 + \sigma_g^2}$$

where

λ = film parameter
h_{min} = minimum oil film thickness, in.
σ_p = pinion average roughness, rms
σ_g = gear average roughness, rms

The "partial elastohydrodynamic" or "mixed" lubrication regime occurs if the film parameter λ is between approximately 1 and 4. At higher values, full hydrodynamic lubrication is established and asperity contact is negligible. Below a λ of 1.0 there is a risk of surface distress.

The minimum elastohydrodynamic film thickness is calculated as follows [16,17]:

$$H = \frac{2.65\, G^{0.54}\, U^{0.70}}{W^{0.13}}$$

$H = \frac{h_0}{R}$ (film thickness parameter)

h_0 = minimum film thickness, in.

R = equivalent radius, in.

$R = \frac{e_p e_g}{e_p \pm e_g}$ + external, - internal

e_p = pinion radius of curvature, in.

e_g = gear radius of curvature, in.

$G = \alpha E'$ (materials parameter)

α = pressure viscosity coefficient, in.^{2}/lb (Figure 3.18)

$$\frac{1}{E'} = \frac{1}{2}\left(\frac{1-\delta_1^2}{E_1} + \frac{1-\delta_2^2}{E_2}\right)$$

$E' = 33 \times 10^6$ for steel on steel

δ_1 = pinion Poisson's ratio

δ_2 = gear Poisson's ratio

E_1 = pinion Young's modulus

E_2 = gear Young's modulus

$W = \frac{W'}{E\,R}$ (load parameter)

W' = Specific loading, lb/in.

$$U = \frac{\frac{1}{2}(v_1 + v_2)\mu_0}{E'R}$$

$v_1 = w_p e_p$ = rolling velocity of pinion at point of contact, ips

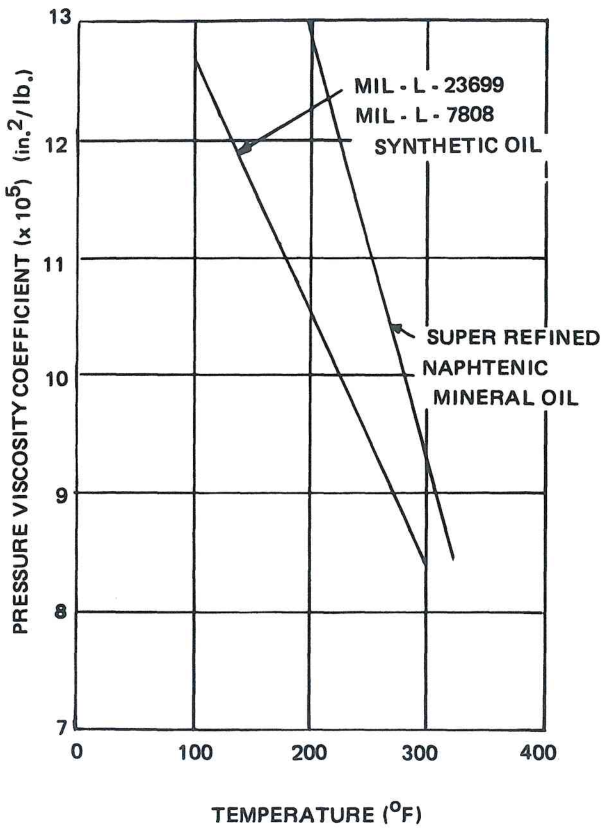


Figure 3.18 Pressure viscosity coefficient versus temperature.

$v_2 \;=\; w_g e_g \;=$ rolling velocity of gear at point of contact, ips

$w_p \;=$ pinion angular velocity, rad/sec

$w_g \;=$ gear angular velocity, rad/sec

$\mu_0 \;=$ absolute viscosity, Reyns (lb-sec/in.^{2})

$$\mu_0 = \frac{eZ_k}{6.9(10)^6}$$

$e \;=$ specific gravity (Figure 3.19)

$Z_k \;=$ kinematic viscosity, cSt (Figure 3.20)

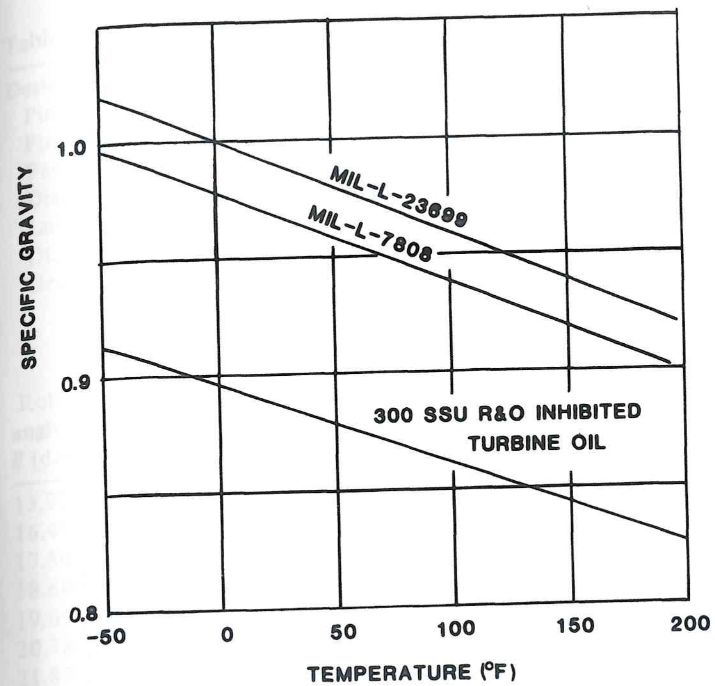


Figure 3.19 Specific gravity versus temperature.

An extensive survey of aerospace power gears operating with synthetic lubricants at high temperature revealed that calculated oil films were on the order of 0.000010 to 0.000020 in. With surface roughness on the order of 20 rms it can be seen that these gears are operating with λ less than 1.0, are in the boundary lubrication regime, and are therefore prone to scoring problems.

Table 3.8 shows the results of a computer analysis of a high-speed gear set with standard addendums. The flash temperature index is the maximum flash temperature rise, 71°F, plus the gear bulk temperature, 160°F. The index of 231°F presents a low scoring risk (Figure 3.17), which could be slightly reduced by optimizing tooth proportions. The calculated coefficient of friction is significantly lower than the assumed valued of 0.06, with a corresponding lower flash temperature rise. The calculated minimum film thickness is 0.000020 with a λ term of 0.71, indicating operation in the boundary lubrication regime.

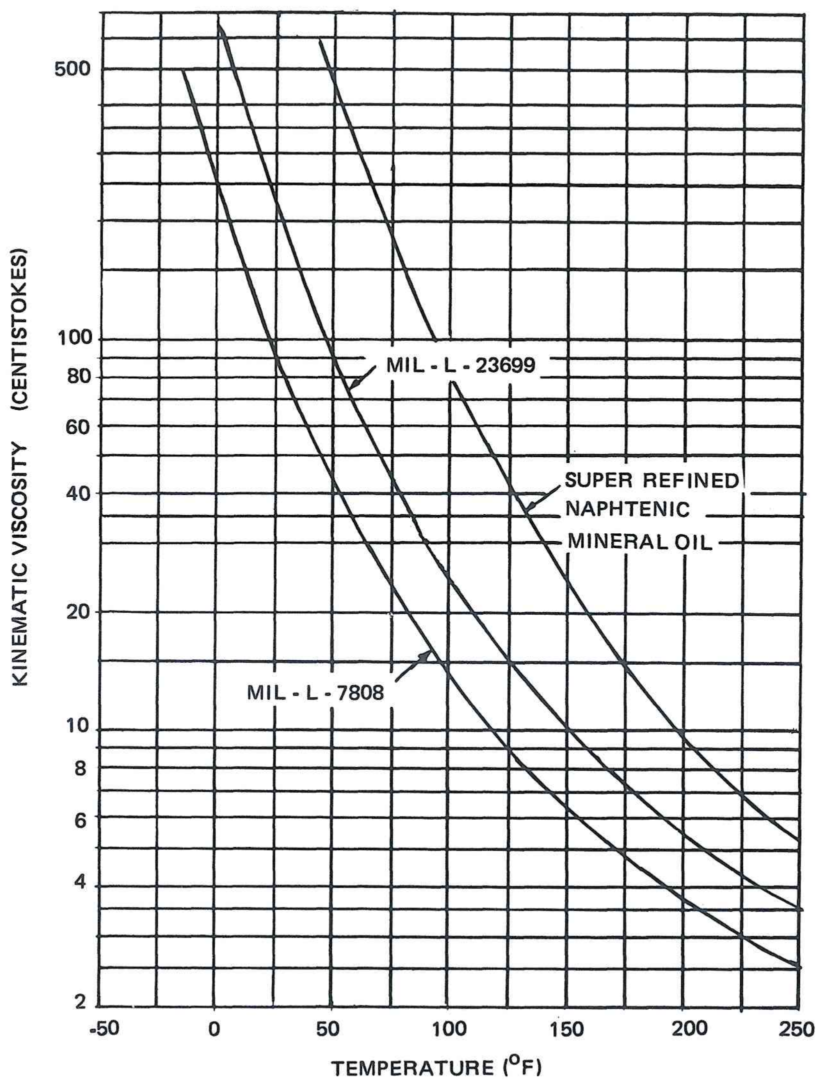


Figure 3.20 Kinematic viscosity versus temperature.

Table 3.8 Scoring Analysis of a High-Speed Gearset

Design parameters

Pinion teeth: 25
Pinion speed: 12,223 rpm
Face width: 1.0 in.
Diametral pitch: 10
Lubricant type: Mil-L-23699 oil
Oil viscosity: 1.22 (10^{-6}) lb-sec/in.^{2}
Gear material: SAE 9310 steel

Gear teeth: 85
Horsepower: 281
Helix angle: 0
Pressure angle: 25°
Gear blank temperature: 160°F
Pressure viscosity coefficient: 11.4 (10^{-5}) in.^{2}/lb
Surface finish: 20 μin. rms

Roll angle, θ (deg)	Friction coefficient, f	Flash temperature rise, ΔT (°F)[a]	Flash temperature rise, ΔT (°F)[b]	EHD film thickness, h_{min} (μin.)	Film parameter, λ
15.32	0.0	0	0	0	0.0
16.41	0.014	0	40	22	0.77
17.50	0.018	18	58	21	0.73
18.60	0.021	23	68	20	0.72
19.69	0.023	28	71	20	0.71
20.78	0.025	30	70	20	0.71
21.87	0.027	30	64	20	0.72
22.96	0.030	30	60	20	0.71
24.05	0.032	22	42	21	0.73
25.14	0.035	13	25	21	0.75
26.23	0.041	5	7	22	0.76
27.32	0.040	7	9	22	0.78
28.41	0.034	13	25	23	0.80
29.50	0.031	22	41	23	0.81
30.59	0.028	25	52	24	0.84
31.68	0.026	25	59	25	0.88
32.77	0.024	25	62	26	0.91
33.86	0.021	22	61	27	0.96
34.95	0.019	18	57	29	1.01
36.04	0.016	13	48	30	1.08
37.13	0.012	7	33	34	1.19

[a] f is variable.
[b] f held constant at 0.06.
Source: Preventing Gear Tooth Scoring, *Machine Design*, March 20, 1980, Penton/IPC.

SHAFT RATING

Three elements of shaft design are covered in this section:

Shaft stresses
Keyways
Splines

SHAFT STRESSES

Figure 3.21 gives maximum allowable stresses in a shaft due to torsion and bending. These allowable stresses provide for a stress concentration not exceeding 3.0

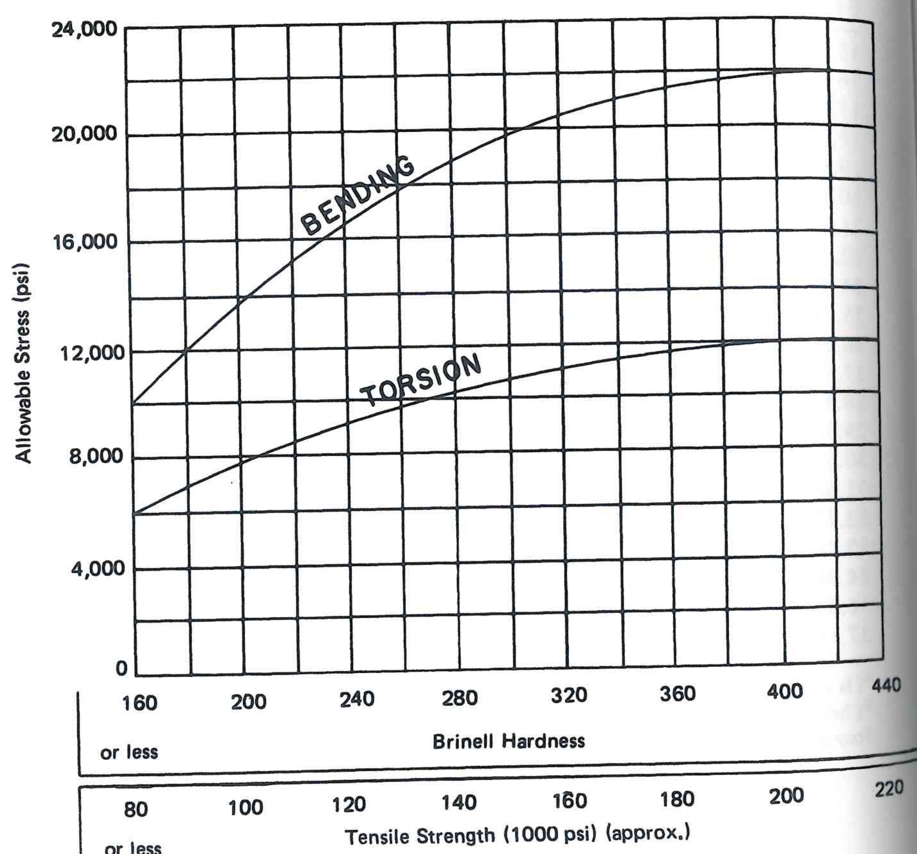


Figure 3.21 Allowable shaft stresses. (From Ref. 1.)

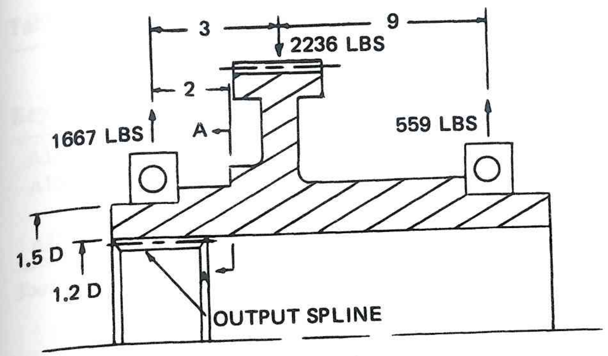


Figure 3.22 Calculation of shaft stress.

and are based on a service factor of 1.0. It is assumed that over 1 million operating cycles will be accumulated. If the shaft is designed to a finite life of less than 1 million cycles, higher allowables can be used.

Let us review a simple example to demonstrate shaft design. The spur gear shaft shown on Figure 3.22 transmits 500 hp at 10,000 rpm. The torque is therefore 3151 in.-lb and if the pitch diameter is 3.0 in., the tangential load is 2101 lb. For a 20° pressure angle the separating load is 765 lb and the resultant radial load is 2236 lb. The bearing reactions are 559 and 1677 lb, respectively.

To find the bending stress at section AA, use

$$S_b = \frac{Mc}{I}$$

where

M = bending moment, in.-lb
c = radius where stress is calculated, in.
I = moment of inertia of section, in.^{4}, for a hollow shaft
$I = (\pi/64)(\text{O.D.}^4 - \text{I.D.}^4)$

The bending moment at section AA is

$$M = 1677(2.0) = 3354 \text{ in.-lb}$$

$$I = \frac{\pi}{64}(1.5^4 - 1.2^4) = 0.147 \text{ in.}^4$$

and

$$S_b = \frac{3354(0.75)}{0.147} = 17{,}112 \text{ psi}$$

To find the torsional stress, use

$$S_s = \frac{Tr}{J}$$

where

T = torque, in.-lb
r = radius, in.
J = polar moment of inertia, in.^{4}; for a hollow shaft $J = (\pi/32)(\text{O.D.}^4 - \text{I.D.}^4)$

At section AA:

$$J = \frac{\pi}{32}(1.5^4 - 1.2^4) = 0.293 \text{ in.}^4$$

and

$$S_s = \frac{3151(0.75)}{0.293} = 8065 \text{ psi}$$

From Figure 3.21 it can be seen that the shaft hardness must be over 240 Bhn for the stress levels calculated. The analysis given above for shaft bending and torsion will yield satisfactory results for most applications. If it is desired to minimize shaft weight or if a shaft has an unusual configuration, a detailed analysis combining tensile, shear, and compressive stresses must be conducted. Also, stress concentrations must be calculated accurately. When refining a shaft analysis in order to minimize shaft weight, deflections must also be considered, since this factor may be more critical than stress.

KEYWAYS

Keyways must be analyzed for shear and compressive stresses as follows [6]:

$$S_s = \frac{2T}{dwL}$$

$$S_c = \frac{2T}{dh_1 L}$$

where

S_s = shear stress, psi
S_c = compressive stress, psi
T = shaft torque, lb-in.

Table 3.9 Allowable Keyway Stresses

Key material	Hardness Bhn	Allowable stress (psi) Shear	Comp.
AISI 1018	None specified	7,500	15,000
AISI 1045	225-265	10,000	20,000
	225-300	15,000	30,000
AISI 4140	310-360	20,000	40,000

Source: Ref. 1.

d = shaft diameter, in. (for taper shaft use mean diameter)
w = width of key, in.
L = length of key, in.
h_1 = height of key in the shaft or hub that bears against the keyway; for designs where unequal portions of the keyway are in the hub or shaft, h_1 must be the minimum portion

Allowable keyway stresses are presented in Table 3.9.

SPLINE RATING

Standard SAE splines are rated on the basis of tooth shear and compressive stress:

$$S_s = \frac{8T}{D^2 \pi F_e}$$

$$S_c = \frac{2TK_m}{D^2 F_e}$$

where

T = torque, in.-lb
D = spline pitch diameter, in.
F_e = effective face width, in.
K_m = load distribution factor, Table 3.10

The compressive stress equation simply divides the tangential load per tooth by the tooth bearing area:

$$\text{Tangential load per tooth} = \frac{2T}{D(\text{number of teeth})}$$

Table 3.10 Spline Load Distribution Factor (K_m)

Misalignment (in./in.)	Face width of splines			
	0.5	1.0	2.0	4.0
0.001	1.0	1.0	1.0	1.5
0.002	1.0	1.0	1.5	2.0
0.004	1.0	1.5	2.0	2.5
0.008	1.5	2.0	2.5	3.0

Source: Ref. 19.

$$\text{Bearing area} = F_e(\text{tooth height})$$

$$\text{Tooth height} = 2(\text{spline addendum})$$

$$= \frac{2}{\text{denominator of diametral pitch}}$$

$$= \frac{1}{\text{numerator of diametral pitch}}$$

$$= \frac{\text{pitch diameter}}{\text{number of spline teeth}}$$

For instance, a 20-tooth, 20/40-diametral pitch spline has a tooth height of 0.5 in.

The shear stress equation divides the tangential load by the shear area at the pitch line, which is

$$\text{Shear area} = \frac{F_e \pi}{2(\text{numerator of diametral pitch})}$$

$$= \frac{F_e \pi(\text{pitch diameter})}{2(\text{number of teeth})}$$

In the shear stress equation the assumption is made that only half the spline teeth are in contact. Allowable shear stresses for splines are 50,000 psi for hardened splines (Rc 60) and 40,000 psi for splines in the range Rc 33 to 38.

To determine the compressive stress allowable, the S_c calculated above is modified by the following factors [19]:

$$\text{Sliding splines} - S_c' = \frac{S_c K_a}{L_w}$$

Table 3.11 Spline Application Factor (K_a)

Power source	Type of load: Uniform	Light shock	Intermittent shock	Heavy shock
Uniform	1.0	1.2	1.5	1.8
Light shock	1.2	1.5	1.8	2.1
Medium shock	2.0	2.2	2.4	2.8

Source: Ref. 19.

Table 3.12 Spline Wear Factor (L_w)

Number of shaft revolutions	Wear factor, L_w
10,000	4.0
100,000	2.8
1 million	2.0
100 million	1.0
1 billion	0.7
10 billion	0.5

Source: Ref. 19.

Table 3.13 Spline Fatigue Life Factor (L_f)

Number of torque cycles	Life factor, L_f: Unidirectional	Fully reversed
10^3	1.8	1.8
10^4	1.0	1.0
10^5	0.5	0.4
10^6	0.4	0.3
10^7	0.3	0.2

Source: Ref. 19.

Table 3.14 Allowable Compressive Stress for Splines

Material	Hardness	Allowable stress (psi) Straight	Crowned
Steel	160–200 Bhn	1,500	6,000
Steel	230–260 Bhn	2,000	8,000
Steel	33-38 R_c	3,000	12,000
Surface hardened	48-53 R_c	4,000	16,000
Case hardened	58-63 R_c	5,000	20,000

Source: Ref. 19.

$$\text{Fixed or locked splines} - S'_c = \frac{S_c K_a}{9 L_f}$$

where

K_a = application factor (Table 3.11)
L_w = wear factor (Table 3.12)
L_f = life factor (Table 3.13)

The life factor is based on the number of times the torque is applied and removed in the expected life of the machine (number of starts). The wear factor is based on the revolutions of the splines. Each revolution there is a back-and-forth rubbing of the teeth, which causes wear. The value of S'_c should not exceed the allowable compressive stress numbers given in Table 3.14.

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4
BEARINGS AND SEALS

BEARINGS

One of the first questions to be asked when reviewing a gear box design is: "How are the shafts supported"? Each shaft must be radially and axially located by bearings and proper bearing design and application is at least as important to the operation of the unit as the gears themselves. In fact, bearing problems are more common in gear boxes than are gear failures. A basic choice when designing a unit is which type of bearing to use. There are two general classes, one being journal-type bearings (sometimes called fluid film or slider), the other being rolling-element-type bearings (sometimes called antifriction).

Journal bearings are characterized by operating with a relatively thick oil film between the rotating and stationary elements, the oil film being sheared by relative sliding. Rolling element bearings include ball, roller, and needle bearings with many design variations in each of these classes. The rolling elements are in intimate contact with one another, the oil film being relatively thin compared to journal bearings. In many applications it is not clear which configuration to use. Let us look at various parameters as they relate to journal and rolling element bearings.

1. *Load capacity*. As will be seen later in the chapter, rolling element bearing fatigue life can be calculated on a statistical basis; therefore, in critical applications where load is significant and/or speeds are high, finite lives are predicted. Journal bearings are thought of as having infinite lives provided that the loading is kept below a predetermined level. Therefore, in many applications requiring long life, journal bearings are chosen since the calculated rolling element life is considered insufficient. It is true that a journal bearing when

operating well can last indefinitely while a rolling element bearing eventually is prone to fatigue failure; however, the L_{10} bearing life concept is probably conservative and rolling element bearings quite often operate far longer than predicted lives.

In many applications load is proportional to speed; however, if load is applied at startup, rolling element bearings have an advantage over journal bearings since a journal bearing must attain some speed before developing a hydrodynamic oil film. In some cases journal bearings are externally pressurized at startup so that the load is carried by this pressure rather than a hydrodynamic oil film. This is often done with heavy rotors such as large generator or turbine shafts.

2. *Speed.* Journal and rolling element bearings are both speed limited but for different reasons. As journal bearing speeds increase, the conjunction zone becomes more turbulent and lubricant shearing increases the temperatures, thereby decreasing the oil film thickness. Surface velocities at the bearing bore of 15,000 ft/min are considered high, although operation to 30,000 ft/min has been attained. Rolling element bearing speeds are limited by the centrifugal forces generated between the balls or rollers and outer races, which can exceed the capacity of the bearing. Rolling element bearing speeds are characterized by the DN value (shaft diameter in millimeters times speed in rpm). DN values of over 1×10^6 are considered high. Some gas turbine advanced applications incorporate bearings operating at 3×10^6 DN; however, these bearings require extensive development in such areas as lubrication and retainer design for proper operation.

3. *Lubrication.* Rolling element bearings require significantly less oil flow than journal bearings. As a result, they exhibit less power consumption and heat generation. They also have lower starting torque requirements, particularly at low temperatures where lubricant viscosity is high. Because of their large operating oil films, journal bearings provide some damping in the mechanical system, whereas rolling element bearings are stiff. Journal bearings are very sensitive to dirt in the lubricant since the operating surface is much softer than the hardened contact surfaces of rolling element bearings. Sometimes rolling element bearings are chosen over journal bearings because pre- or postlubrication can be eliminated. It is not necessarily true that journal bearings need to be lubricated prior to startup, and many applications depend on a soft external bearing layer such as babbitt to survive the initial cycles until an oil film forms. For a given application, testing is the only positive way to determine if prelubrication is necessary. Postlubrication is rarely necessary for either rolling element or journal bearings, being used only if some high soak-back temperature will exceed the stabilization temperatures of the bearing materials or cause oil left in the bearing cavity to coke up.

4. *Cost.* In very large quantities journal bearings can be produced at lower cost than rolling element bearings. An example of this is crank shaft bearings in the auto industry. In small quantities for special designs, journal bearings are more expensive. The rolling element bearing industry has standardized on a series of bearing types and manufactures these in large quantities as off-the-shelf items. Equivalent journal bearings for gearboxes, although sometimes listed in catalogs, are not mass produced and in the quantities purchased are relatively expensive.

In many cases the cost of replacing a bearing and the downtime involved is far more significant than the original cost. An example of such a case is a gearbox in a critical process in a petroleum plant. In such an instance journal bearings are preferred, for three reasons:

As discussed before, journal bearings, because the moving parts are separated by a large oil film, can last indefinitely, whereas rolling element bearings are subject to fatigue failures.

The types of failure encountered with journal bearings are less catastrophic and easier to detect. For instance, a journal bearing that is wearing can be monitored with proximity probes and replaced at a convenient time. A spalling rolling element bearing is harder to detect, is introducing abrasive debris into the lubrication system, and may fail abruptly as surface deterioration progresses.

Journal bearings can be split, which enables the user to replace a defective bearing without removing the shafts from the gearbox. This way the couplings remain in place and a time-consuming realignment is not required.

In a broad overview of when journal or rolling element bearings are used: Journal bearings tend to be chosen for high-speed operation, where long life and extreme reliability are desired and the bearing cost is small compared to the cost of system downtime. Lower speed, less critical gearboxes tend to incorporate rolling element bearings. In some sophisticated applications such as aircraft turbine engines, rolling element bearings are used to take advantage of low oil flow requirements, high efficiency, ability to start without oil and at low ambient temperatures.

Once the choice between journal or rolling element bearings has been made, a more detailed analysis of bearing selection and application is in order. The next section will be a review of rolling element bearing fundamentals followed by a consideration of journal bearings.

Rolling Element Bearings

Rolling element bearings are composed of four elements: an inner race, an outer race, a complement of balls or rollers, and a ball or roller separator (sometimes

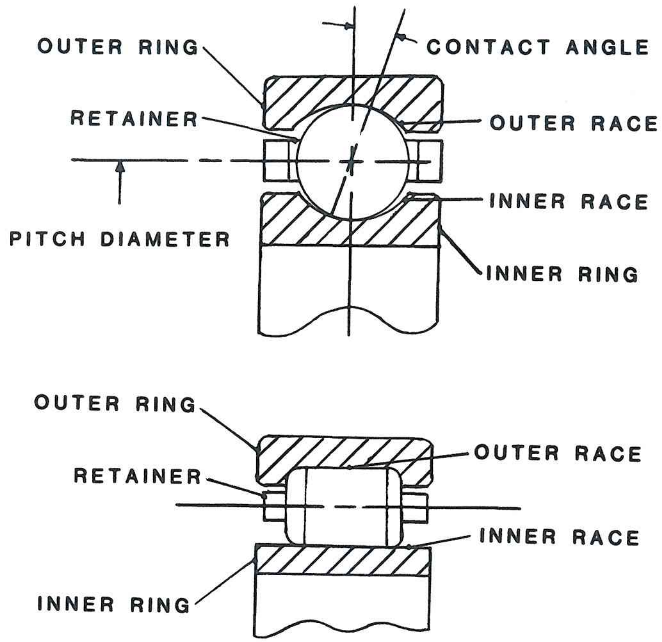


Figure 4.1 Bearing nomenclature.

called a retainer or cage). Figure 4.1 illustrates bearing nomenclature. Usually, the inner race rotates and the outer race is stationary, but there are applications where the reverse is true or both races rotate. Ball bearings in addition to reacting radial loads can carry significant thrust loads, whereas roller bearings though having greater radial load capacity than ball bearings can react only small thrust loads. The greater roller bearing radial capacity is due to the fact that rollers are in line contact with the races as compared to the point contact of ball bearings; therefore, the radial load is spread over a greater area. The following paragraphs describe the most widely used rolling element bearing configurations.

Deep Groove Ball Bearings This configuration, sometimes called the Conrad type, is illustrated in Figure 4.2. This type of bearing is commonly used in a wide variety of applications where either thrust or radial capacity or both are required. The bearing is assembled by bunching the balls in one arc of the circumference, which permits the inner and outer races to be located in their proper positions. The balls are then separated around the circumference and their retainer fitted in place. A number of retainer configurations are used, the most common being a pressed steel design with the two halves riveted together.

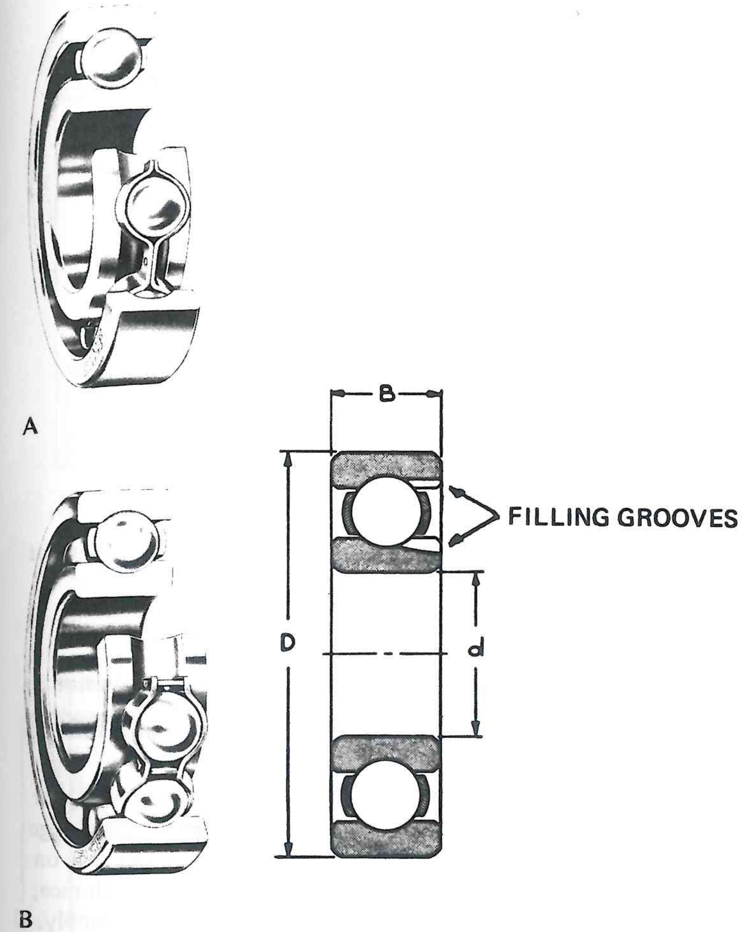


Figure 4.2 Deep groove ball bearing. A. Conrad type. B. Filling groove type. (Courtesy of SKF Industries, Inc., King of Prussia, Pa.)

Because the load capacity is dependent on the number of balls, several design variations are available which increase the ball complement. One such design is the filling notch bearing, where a single notch is ground into both the inner and outer race through which additional balls can be fed. For maximum strength and high-speed operation a split inner ring bearing can be used where the inner race is made of two sections. In addition to allowing the maximum complement of balls to be assembled, a strong one-piece machined steel retainer can be used in this configuration. It is also possible to split the outer race axially in order to

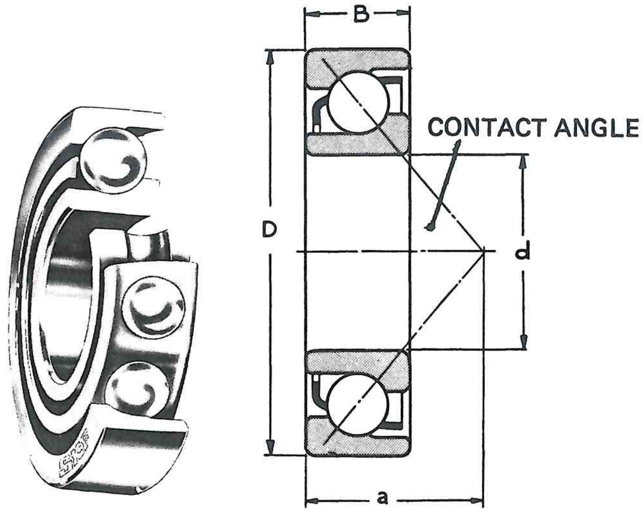


Figure 4.3 Angular contact bearing. (Courtesy of SKF Industries, Inc., King of Prussia, Pa.)

assemble more balls. In the case of the split inner race design the balls do not touch the split, so no reduction in fatigue life occurs. This is not the case with the split outer race bearings.

Angular Contact Ball Bearings Angular contact bearings are designed for applications with high thrust loads in one direction. Figure 4.3 shows the design features, which include one high thrust shoulder on the inner and outer race on opposite sides. There is a low shoulder opposite the high shoulder on each race; thus the maximum number of balls can be snapped into place during assembly. The contact angle between the balls and raceways is designed to be higher than a deep groove bearing giving more thrust capacity. Angular contact bearings can be used in pairs to provide rigid axial location and high thrust capacity in either direction. These bearings can react a combination of thrust and radial loads but are usually used when the thrust load is predominant.

Duplex Ball Bearings Duplex ball bearings are a pair that can be mounted in four ways, as shown in Figure 4.4, to accommodate different loading conditions and stiffness requirements. The inner and outer races are machined such that there is a controlled axial relationship between the two. When the bearings are mounted in the tandem configuration, the precise machining will enable the bearings to share the thrust load equally. In this case the contact angles are parallel. In the face-to-face configuration the contact angle lines converge

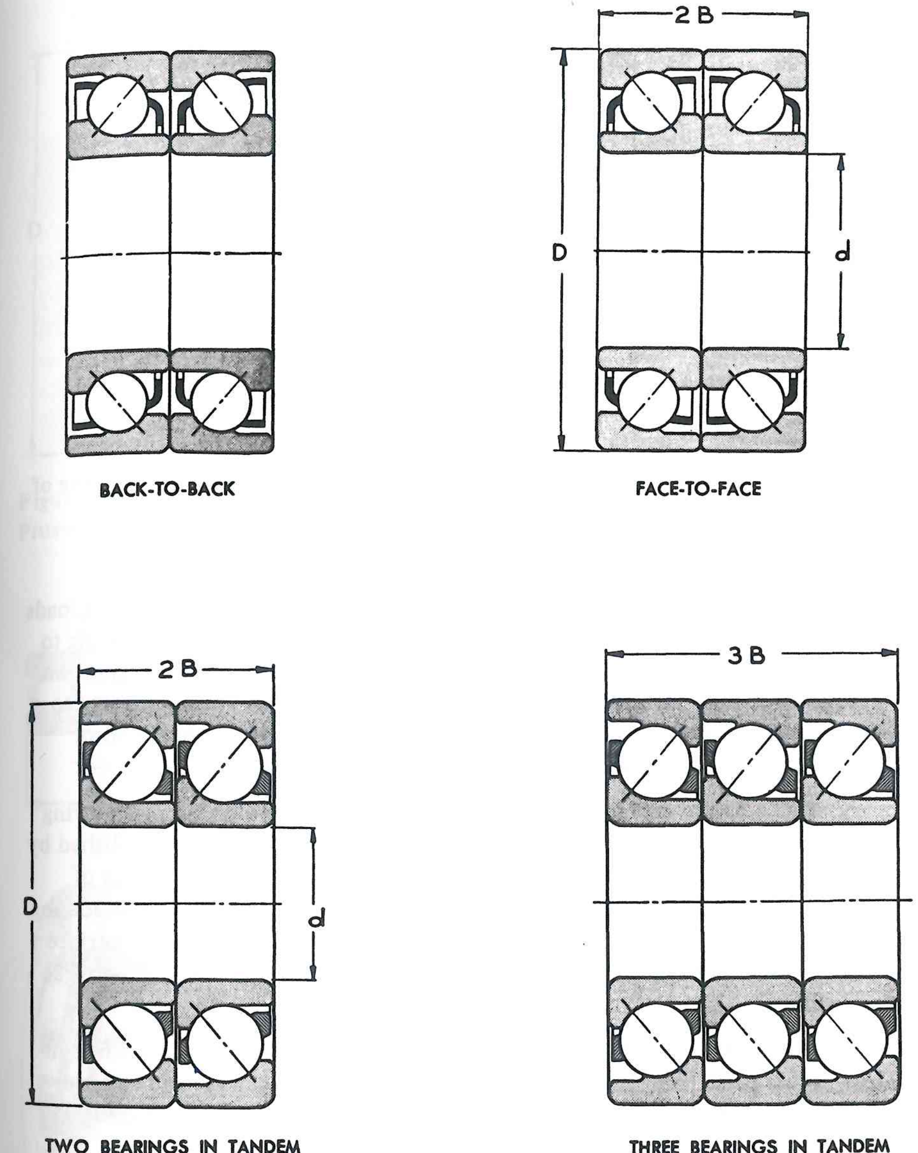


Figure 4.4 Duplex bearing arrangements. (Courtesy of SKF Industries, Inc., King of Prussia, Pa.)

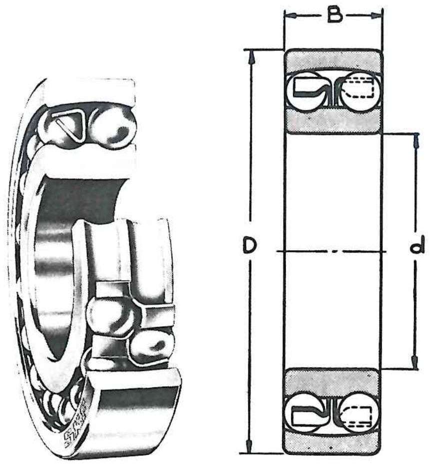


Figure 4.5 Self-aligning ball bearing. (Courtesy of SKF Industries, Inc., King of Prussia, Pa.)

inwardly. This design is used for heavy radial or combined radial and thrust loads or for reversing thrust loads. The face-to-face arrangement allows the bearing to accommodate a small amount of misalignment. The back-to-back configuration has the same large load capacity as the face to face but adds angular rigidity to the system and may be used where it is necessary to restrict misalignment or shaft deflection. The contact angle lines diverge outwardly.

Self-Aligning Ball Bearings Figure 4.5 shows a self-aligning ball bearing which can accommodate large amounts of misalignment. This is accomplished by machining the outer race to a spherical contour; however, the large radius of curvature reduces the bearing-load capacity. Another method of accommodating misalignment is to make the outside diameter of the bearing a spherical surface which then mounts into a spherical seat in the housing. The bearing is then free to position itself.

Double Row Ball Bearings The double row ball bearing (Figure 4.6) consists of one-piece inner and outer races each having two raceways and two ball complements. This design provides heavy radial and thrust capacity in either direction in an envelope somewhat smaller than two single-row bearings. The contact angles can be either face to face or back to back depending on whether it is desired to accommodate misalignment or provide a stiff mounting.

Ball Thrust Bearings Figure 4.7 shows a typical ball thrust bearing, which is similar to a radial ball bearing except that the raceways are axial rather than radial.

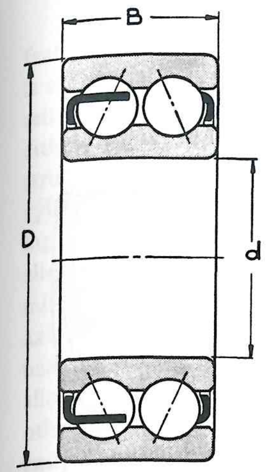


Figure 4.6 Double row ball bearing. (Courtesy of SKF Industries, Inc., King of Prussia, Pa.)

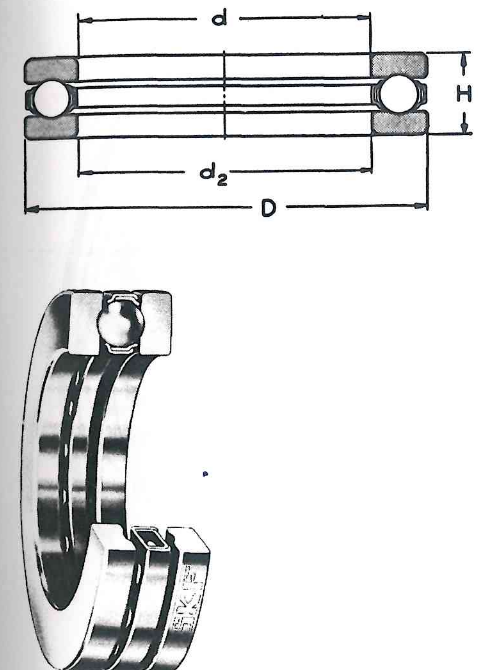


Figure 4.7 Ball thrust bearing. (Courtesy of SKF Industries, Inc., King of Prussia, Pa.)

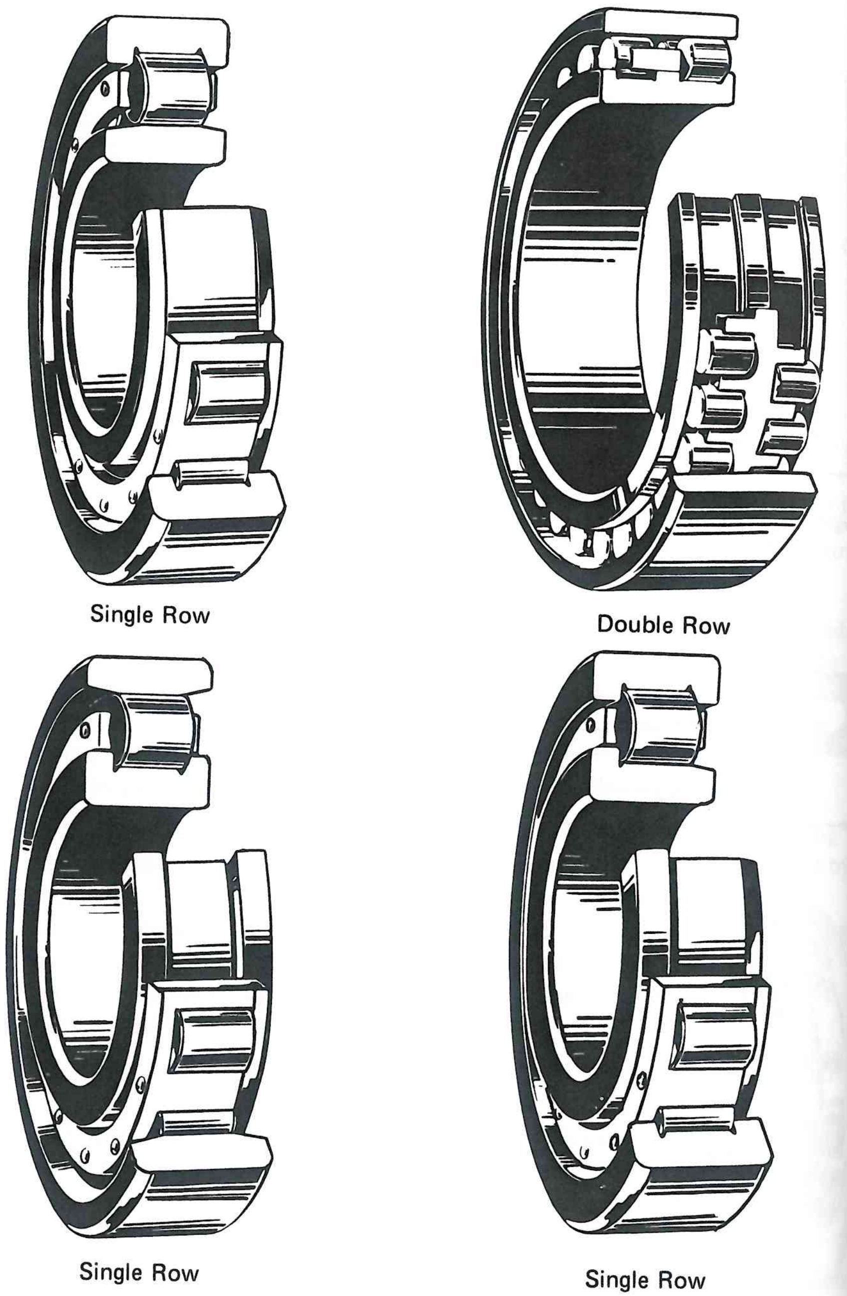


Figure 4.8 Cylindrical roller bearings. (Courtesy of SKF Industries, Inc., King of Prussia, Pa.)

Cylindrical Roller Bearings These bearings, illustrated in Figure 4.8, feature cylindrical rollers which run on cylindrical raceways. The roller length is approximately equal to the roller diameter and the rollers are crowned to relieve potentially high stresses at their ends. Roller retainers are positioned either by the rollers or the inner or outer races and are usually two-piece construction either riveted or screwed together. Figure 4.8 shows some of the cylindrical roller bearing configurations available. On the left side are two floating designs where either the inner or outer race has no shoulders and the shaft is allowed to float axially in relation to the housing. Only radial loads can be transmitted. This type of bearing is useful when the shaft must be allowed to move axially due to thermal expansion or deflection due to loading. Cylindrical roller bearings with one shoulder on one race and two shoulders on the other race allow axial movement in one direction and can sustain some thrust load in the other direction. Designs are also available with two shoulders on both the inner and outer races and can sustain axial loads in both directions. In this case one of the races is machined in two pieces for ease of assembly and inspection. It is difficult to determine the amount of thrust load a cylindrical roller bearing is capable of transmitting. The load is carried by sliding contact between the roller

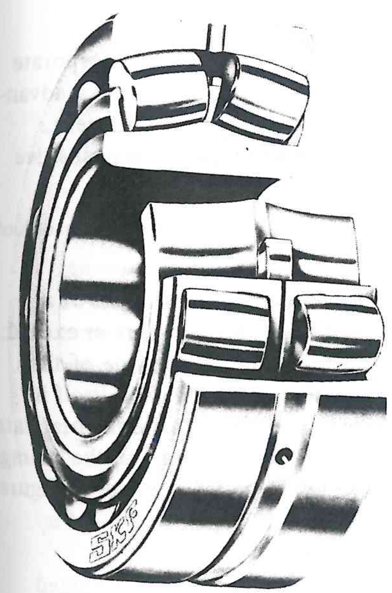


Figure 4.9 Spherical roller bearing. (Courtesy of SKF Industries, Inc., King of Prussia, Pa.)

ends and the race lips such that the action is more like a journal thrust bearing than a rolling element bearing. The important parameters, therefore, are the contact area, surface finish, surface geometry, sliding velocity, lubricant, and operating temperature. In general, a roller bearing can be used to axially locate a shaft when no large thrust loads are anticipated, but should not be expected to react significant amounts of thrust.

Spherical Roller Bearings Figure 4.9 illustrates a typical spherical roller bearing. There are two rows of rollers which run on a common raceway in the outer ring, which has been ground to a spherical contour on the inner diameter. The inner ring has two raceways ground at an angle to the axis of rotation; thus the bearing is capable of reacting moderate thrust loads in either direction, in addition to heavy radial loads. The spherical outer race enables the bearing to accommodate some shaft misalignment. This type of bearing is suitable for relatively low speed operation.

Tapered Roller Bearings The tapered roller bearing (Figure 4.10) is designed such that lines extended from each tapered surface intersect at a common point on the bearing axis. Because of the tapered races heavy loads in both the radial and axial directions can be handled. As shown in Figure 4.10, both single row and double row configurations are available. Tapered roller bearings are usually limited to low-speed operation.

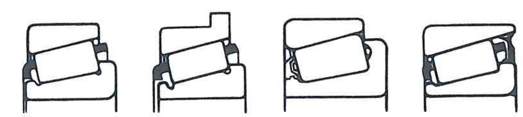
Needle Bearings As shown in Figure 4.11, needle bearings incorporate rollers which are relatively long in relation to their diameter. Their major advantage is high radial load capacity with a small radial envelope requirement. Because of the large length-to-diameter ratio, needle bearings are very sensitive to shaft misalignment.

Bearing Life Rating

Bearings are conventionally rated in terms of L_{10} life, the life that 90% of a group of bearings operating at a given set of conditions will complete or exceed. A bearing failure is defined as the first occurrence of fatigue on one of the rolling elements or on one of the raceways. Fatigue usually manifests itself in the form of a spall. The L_{10} life concept is based on extensive experimental data and as with all fatigue data, there is considerable scatter; however, on the average a plot of life versus percentage of bearings failed takes the form shown in Figure 4.12. In the figure it can be seen that the median life, the life that 50% of a group of bearings will achieve, is five times L_{10} life.

The term "rating life" or just "life" of a bearing has been standardized on as the L_{10} life by organizations such as the Anti-Friction Bearing Manufacturers Association (AFBMA), the American National Standards Institute (ANSI), and the International Standards Organization (ISO). It is a

single row, tapered roller bearings



double row, tapered roller bearings

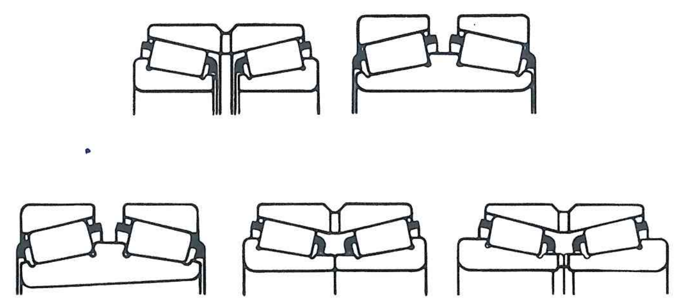


Figure 4.10 Tapered roller bearings. (Courtesy of SKF Industries, Inc., King of Prussia, Pa.)

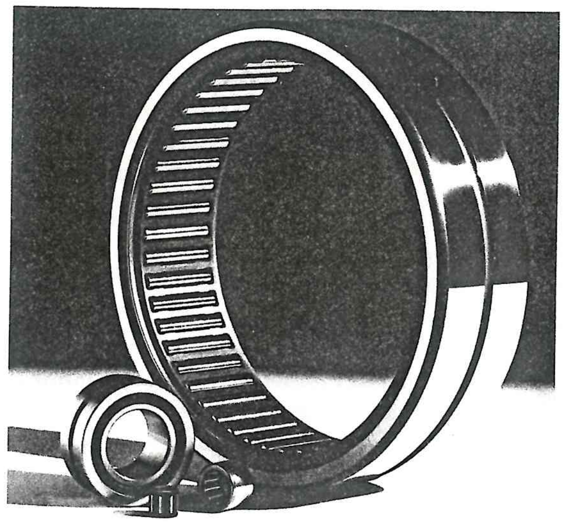


Figure 4.11 Needle bearing. (Courtesy of SKF Industries, Inc., King of Prussia, Pa.)

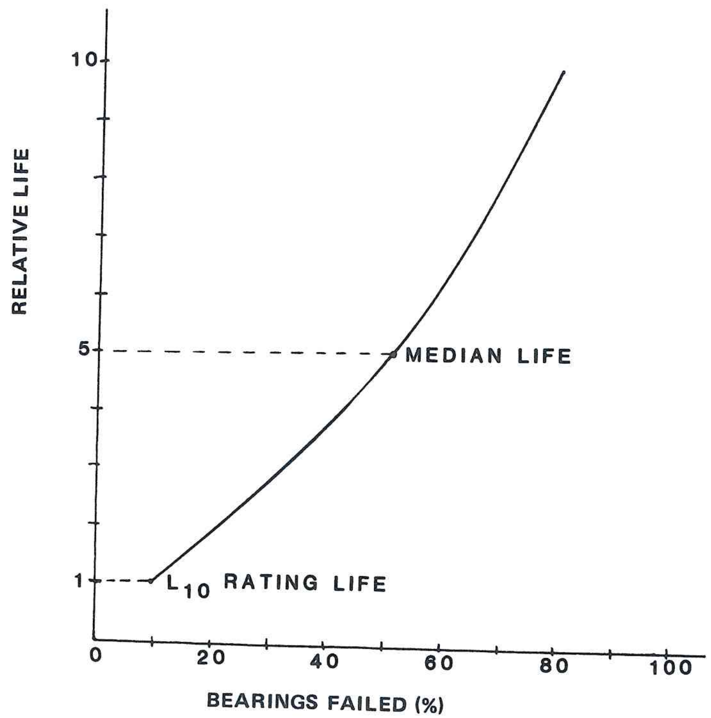


Figure 4.12 Relative bearing life versus percentage failed.

reasonable compromise that provides reliable bearing service while meeting practical economic requirements. In other words, bearings chosen using this criterion will provide reasonably long service, yet not be overly expensive. It must be remembered that the L_{10} life concept pertains only to fatigue of the rolling elements and raceways. Failures due to other causes, such as wear, excessive heat generation, retainer failure, and so on, are not covered by the fatigue model and in many cases limit bearing life.

Extensive testing by the bearing manufacturers, together with analytical studies, has established that the fatigue life of ball bearings is inversely proportional to the third power of the load and the fatigue life of roller bearings is inversely proportional to the 3.33 power of the load:

$$\frac{L_A}{L_B} = \left(\frac{F_B}{F_A}\right)^3 \qquad \text{for ball bearings}$$

$$\frac{L_A}{L_B} = \left(\frac{F_B}{F_A}\right)^{3.33} \qquad \text{for roller bearings}$$

where

L_A = bearing life at condition A, cycles
L_B = bearing life at condition B, cycles
F_A = applied load at condition A, lb
F_B = applied load at condition B, lb

Let us assume the following:

L_B = 1 million cycles
F_B = load (lb) that will give a life of 1 million cycles = C
L_A = L_{10} = rating life, millions of cycles
F_A = P = applied load, lb

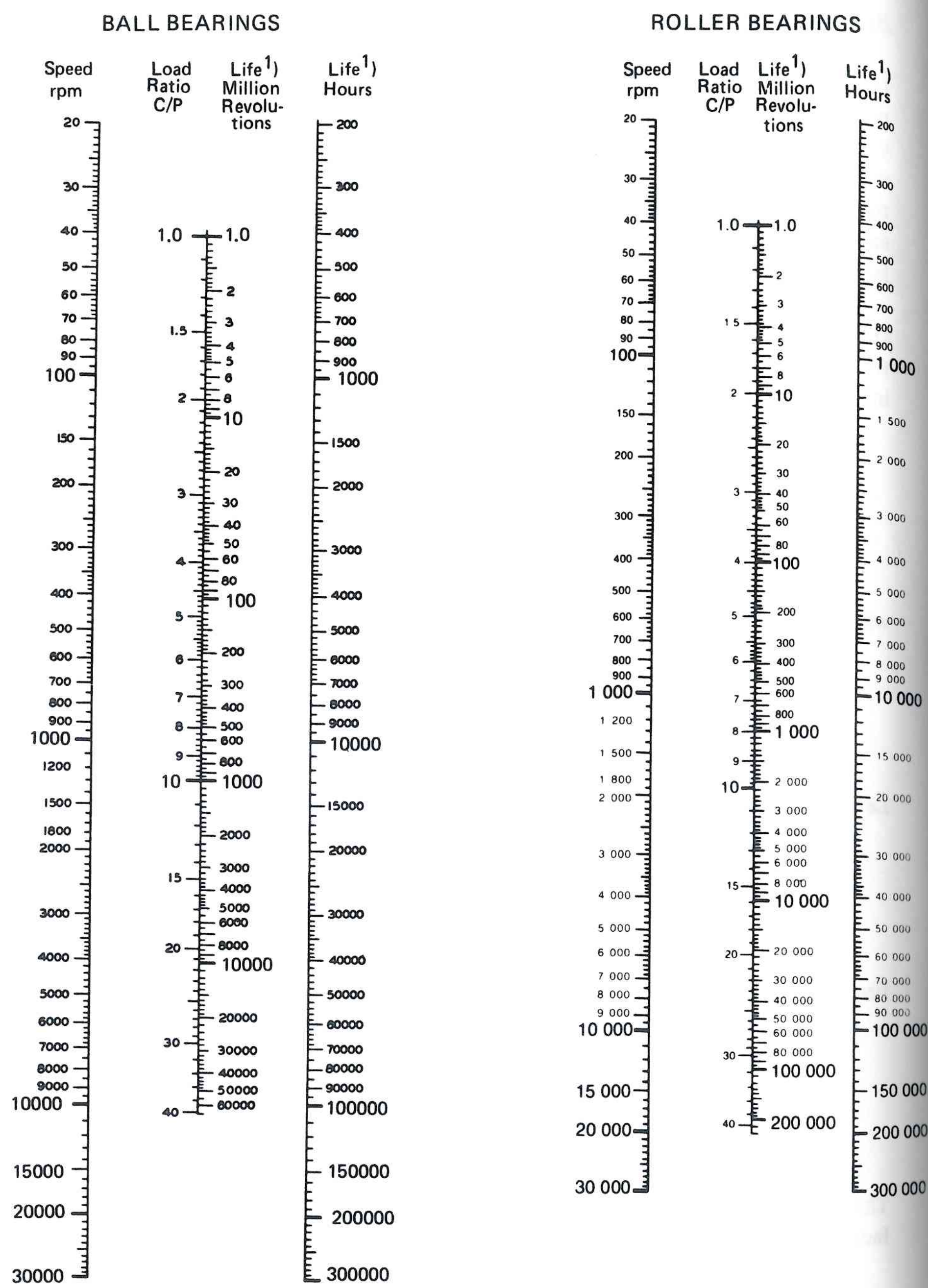
From the above are generated the familiar equations

$$L_{10} = \left(\frac{C}{P}\right)^3 \times 10^6 \quad \text{for ball bearings}$$

$$L_{10} = \left(\frac{C}{P}\right)^{3.33} \times 10^6 \quad \text{for roller bearings}$$

The basic load rating C for any bearing can be found tabulated in bearing manufacturers' catalogs. If the L_{10} life is desired in hours:

$$L_{10} = \frac{16{,}667}{N} \left(\frac{C}{P}\right)^3 \qquad \text{hours for ball bearings}$$



1)This Life is expected to be exceeded by 90% of the bearings.

Figure 4.13 Bearing life nomogram. (From Ref. 1.)

$$L_{10} = \frac{16{,}667}{N}\left(\frac{C}{P}\right)^{3.33} \quad \text{hours for roller bearings}$$

where N is the inner or outer race net speed in rpm. The nomograms in Figure 4.13 represent the equations above and are often used to calculate bearing lives. If the operating load P, operating rpm N, and bearing capacity C are known, a straight line can be drawn that intersects the left scale at the operating speed and the C/P scale at the calculated value of C/P. Where this line intersects the hours scale is the rating life in hours. If the desired rating life, rpm, and applied load are known, a straight line drawn between the speed and life axes will intersect the C/P axis and the required bearing capacity can be calculated.

The L_{10} life concept implies a reliability of 90% with regard to surviving the rating life. In some critical applications reliability numbers greater than 90% are desired. For instance, to achieve a 97% reliability, an L_3 life, the life that 97% of a group of bearings will reach or exceed at a given set of operating conditions, must be calculated. Figure 4.14 gives an estimate of a life adjustment factor corresponding to increased reliability. The factor is based on experimental data and is used as follows:

$$L_n = \alpha_1 L_{10}$$

where

α_1 = life adjustment factor
L_n = fatigue life expectancy for other than 90% reliability

For example, L_3 corresponds to 97% reliability and the factor α_1, from Figure 4.14, is 0.425. If the calculated L_{10} life of a bearing is 10,000 hr, the L_3 life is

$$L_3 = 0.425(10{,}000) = 4250 \text{ hr}$$

Figure 4.14 indicates that there is no probability of fatigue failure at lives below 5% of the L_{10} rating life. In other words, in the example above the bearing is 100% reliable up to 500 hr of operation.

Rating Life Adjustments

Over the past years substantial increases in the fatigue endurance of rolling element bearings have been made possible by improvements in bearing design, materials, processing, and manufacturing techniques. Also, investigation into effects of high-speed operation and misalignment has generated new insights into their effect on load rating. Using the AFBMA L_{10} life method for determining fatigue life and the bearing load ratings in the various manufacturers' catalogs to calculate the basic L_{10} life, an expected bearing life L_A can be calculated as follows:

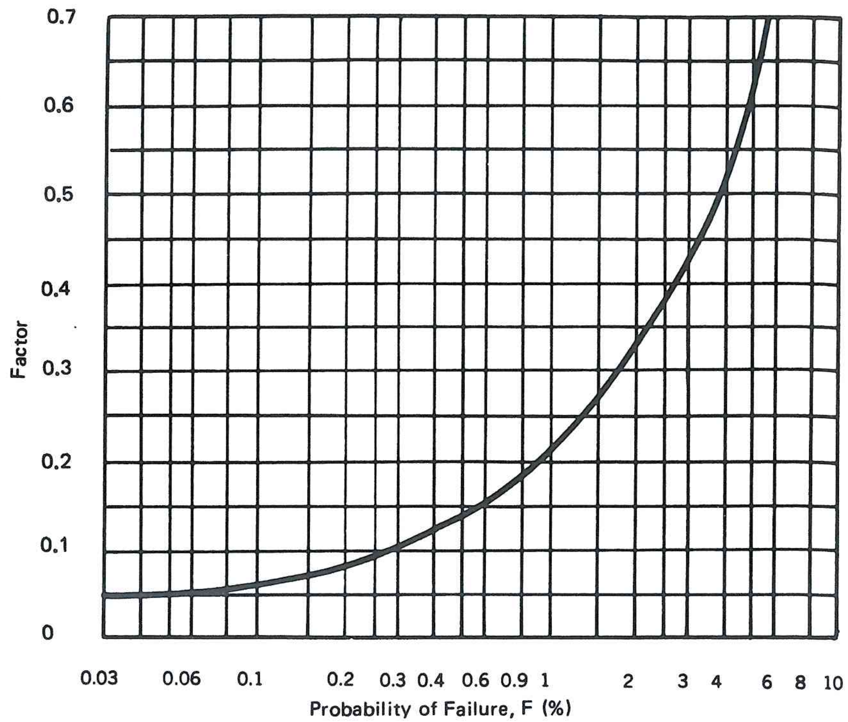


Figure 4.14 Life adjustment factor for reliability. (From Ref. 1.)

$$L_A = (D)(E)(F)(G)(H)L_{10}$$

where D through H are life adjustment factors. This equation, developed by the Rolling Elements Committee of the Lubrication Division of the American Society of Mechanical Engineers and published as Ref. 2, explains the life adjustment factors in detail. The following paragraphs summarize the work.

Material Factor D The predominant material for rolling element bearings is AISI 52100. AISI is the American Iron and Steel Institute designation for steels of various specific chemistry. The basic bearing dynamic capacities which are presented in catalogs are based on air-melted 52100 steel through-hardened to Rc 58 minimum. The mathematical model to define bearing load capacity evolved in 1949 and since then the bearing steels have been improved such that they are more homogeneous with fewer impurities. A materials factor D of 2 is suggested for currently available steels. Case-hardened materials, used in tapered roller bearings and other applications, have also improved over the years, but insufficient data are available to recommend a materials factor.

Processing Factor E The processing considered here is concerned mainly with the melting practice. Air-melt material is considered the baseline and is assigned an E factor of 1. Vacuum-melted material has an E factor of 3. Although standard catalog bearings are not necessarily vacuum melt, quite often this processing is used and the factor can be taken advantage of.

Lubrication Factor F Bearing life has been found to be significantly affected by the thickness of the lubricant film developed between the contacting elements. Film thickness is affected mainly by speed and lubricant properties at the operating temperature with higher fatigue lives obtained at high speeds or with higher-viscosity lubricants. Conversely, if a bearing operates with poor lubrication film formation due to low speed or insufficient lubricant viscosity, the life predicted from the catalog rating may not be achieved. The oil film between rolling elements of a bearing has been found to be in the elastohydrodynamic regime and can be calculated according to the methods outlined in Ref. 3 for roller bearings and Ref. 4 for ball bearings. The calculated oil film is compared to the composite roughness of the contacting elements to form what is referred to as the lambda ratio:

$$\lambda = \frac{h}{\sigma}$$

where

λ = lambda ratio
h = oil film thickness, in.
σ = composite surface roughness of the rolling element surfaces, in.

$$\sigma = \sqrt{\sigma_1^2 + \sigma_2^2}$$

where

σ_1 = rms surface finish of body 1, in.
σ_2 = rms surface finish of body 2, in.

The lubrication factor F is proportional to the lambda ratio and varies from approximately 0.2 to 2.8. As an example, let us assume that the balls and raceway of a bearing have rms surface textures of 0.000010 in. The composite surface roughness will be 0.000014 in. If the calculated lubricant film thickness is also 0.000014 in., the lambda ratio is 1. The lubrication factor, F in this case is approximately 1.

Speed Effect Factor G At high speeds the load at the outer race is increased because the balls or rollers are subject to centrifugal forces and therefore the fatigue life is reduced. Also, the load distribution on the rolling elements and the zone of loading can change because of centrifugal growth and thermal

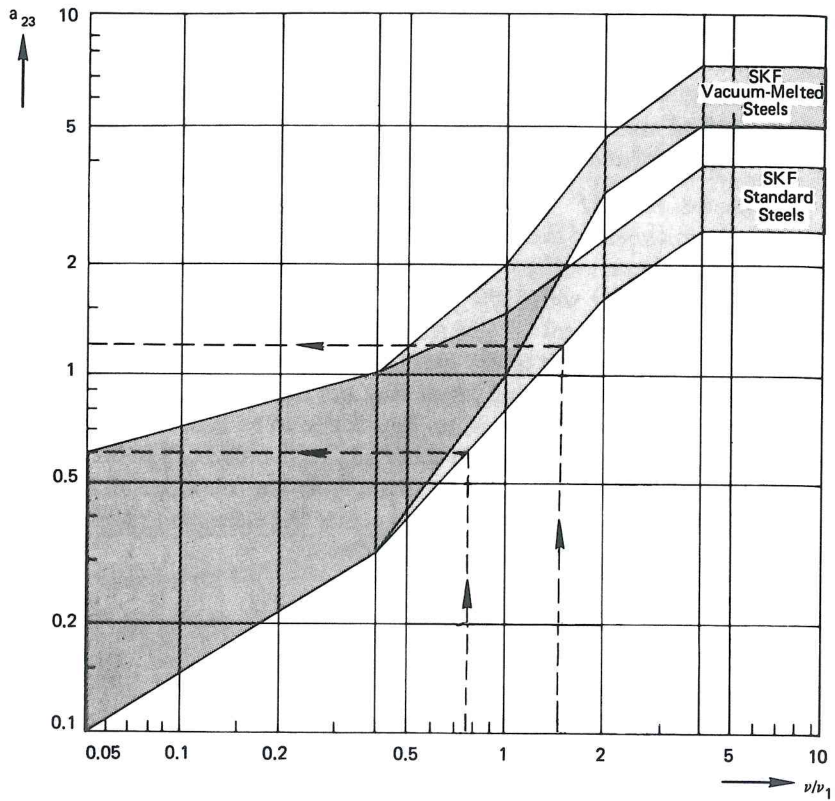


Figure 4.15a Fatigue life adjustment factor a_{23} for bearing materials and lubrication. Lubricant viscosity in application = ν; required lubricant viscosity = ν_1. (From Ref. 5.)

expansion. High-speed bearings must therefore be analyzed using sophisticated computer models which can take all the varying parameters into account, and bearing manufacturers have such programs available for users. In general, centrifugal effects will become significant at DN (bore in millimeters times speed in rpm) values above approximately 0.5×10^6 and certainly bearings operating above 1.0×10^6 DN should be carefully analyzed.

Misalignment Factor H The effect of misalignment on roller bearings is to concentrate the load on one end and therefore reduce the fatigue life. Misalignment can occur due to deflection under load or manufacturing errors in the bearing housing or shaft. The following table provides some misalignment limits beyond which reduction in bearing life may be expected.

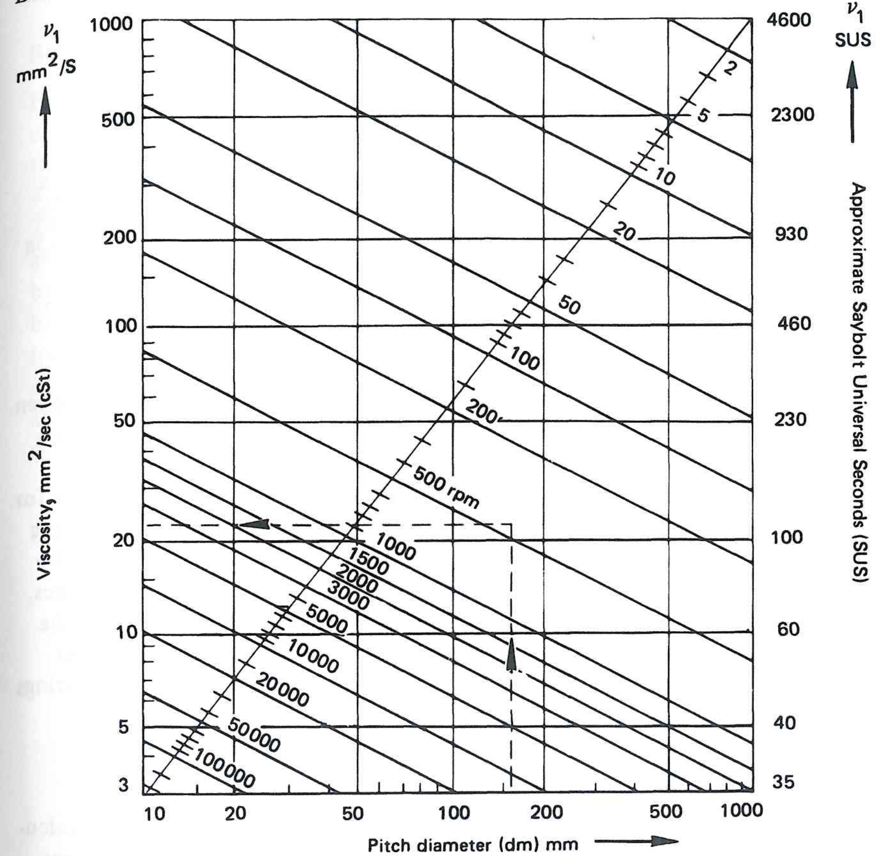


Figure 4.15b Minimum required lubricant viscosity. (Bearing bore + OD) ÷ 2 = dm; required lubricant viscosity for adequate lubrication at the operating temperature = ν_1. (From Ref. 5.)

Allowable Misalignment[a]

	Radians	Minutes
Cylindrical and tapered roller bearings	0.001	3–4
Spherical bearings	0.0087	30
Deep groove ball bearings	0.0035–0.0047	12–16

[a]Based on general experience as expressed in manufacturers' catalogs.
Source: Ref. 2.

When using roller bearings on flexible shafts the misalignment due to load should always be calculated, since this factor may be life limiting.

We can summarize the section on rating life adjustments by noting that today's bearings, assuming good lubrication, can achieve fatigue lives significantly greater than those calculated from catalog ratings. Three- to sixfold life improvement ratings are possible.

One bearing manufacturer's method of presenting these improvements is outlined in [5]. A combined life adjustment factor designated a_{23} replaces the material and lubrication factors described in the preceding section. Figure 4.15a is a plot of the a_{23} factor that can be expected for a particular viscosity ratio (ν/ν_1) and bearing material combination. The viscosity ratio is the ratio of the actual lubricant viscosity (ν), at the operating temperature to the minimum value required (ν_1), which can be found from Figure 4.15b. Figure 4.15b presents the minimum viscosity at the operating temperature required for a given bearing pitch diameter/speed combination. Some common lubricant viscosities at various temperatures can be found in Figures 3.20 and 5.2.

Figure 4.15a is divided into two bands, one for vacuum-melted and one for standard bearing steels. Under comparable operating conditions, certain bearing types (e.g., spherical roller bearings, tapered roller bearings, and spherical roller thrust bearings) normally have a higher operating temperature than other types, such as deep groove ball bearings and cylindrical roller bearings. Therefore, the upper boundary of either band is generally used for ball and cylindrical roller bearings and the lower boundary is used for tapered and spherical roller bearings [5].

Simultaneous Radial and Thrust Loads

Bearings frequently must carry a combination of radial and axial loads. To calculate an L_{10} life, the loading combination must be converted into an equivalent load which would be the constant stationary radial load which if applied to a bearing with rotating inner ring and stationary outer ring would give the same life as that which the bearing would attain under the actual conditions of load and rotation. The general equation for the conversion applying to rotating bearings is

$$P = XF_r + YF_a$$

where

P = equivalent load, lb
F_r = actual constant radial load, lb
F_a = actual constant thrust load, lb
X = radial factor
Y = thrust factor

The information required to calculate the X and Y factors for a specific bearing is tabulated in bearing manufacturers' catalogs.

Bearing Selection Example

Figure 4.16 shows a typical page from a bearing catalog. The AFBMA has standardized on bore size, outside diameter, and width of ball and roller bearings and these dimensions are commonly given in millimeters. For instance, in Figure 4.16 a 6020 bearing has a 100-mm bore, a 150-mm outside diameter, and a 24-mm width. This size bearing will be found in any manufacturer's catalog and although the internal geometry will vary, externally all 100 × 150 × 24 mm bearings will be geometrically interchangeable. Let us assume that the 6020 bearing carries a radial load of 1000 lb and a thrust load of 500 lb at a constant inner ring speed of 2500 rpm. Lubrication is oil jet with a 200 SUS viscosity mineral oil. The bearing material is 52100 air melt. The radial and thrust load are combined into an equivalent load using the procedure outlined in Figure 4.16:

F_r = 1000 lb
F_a = 500 lb
V = 1.0 (inner ring rotating)
C_o = 9410 lb (basic static load rating)

$$\frac{F_a}{C_o} = 0.053$$

e = 0.26
Y = 1.71

$$\frac{F_a}{V(F_r)} = 0.5 \text{ (larger than e)}$$

X = 0.56
P = 0.56(1)(1000) + 1.71(500) = 1415 lb
C = 10,400 lb (basic dynamic load rating)

$$\frac{C}{P} = 7.35$$

$$L_{10} = \frac{16{,}667}{2500}\left(\frac{C}{P}\right)^3 = 2647 \text{ hr}$$

The bearing pitch diameter is

$$PD = \frac{OD + bore}{2} = \frac{150 + 100}{2} = 125 \text{ mm}$$

BEARING NUMBER	Nominal Bearing Dimensions d		D		B		F	M	S	H	T	Max. Fillet Radius[1])	Preferred Shoulder Diameter in.		Brg. Wt. lb	Balls		Basic Static Load Rating C_o	Basic Dynamic Load Rating C	Approx. Speed Limit[2]) rpm	Basic Brg. No.
	mm	in.	mm	in.	mm	in.	in.	in.	in.	in.	in.	in.	Shaft	Housing		No.	Diam. in.	lb	lb		
6000	10	.3937	26	1.0236	8	.3150						.012	.472	.945	.044	7	3/16	440	790	28000	6000
6001	12	.4724	28	1.1024	8	.3150						.012	.551	1.024	.049	8	3/16	500	880	25000	6001
6002	15	.5906	32	1.2598	9	.3543	1,187	1 7/16	.120	.078	.042	.012	.669	1.181	.068	9	3/16	565	965	21000	6002
6003	17	.6693	35	1.3780	10	.3937						.012	.748	1.299	.089	10	3/16	625	1040	19000	6003
6004	20	.7874	42	1.6535	12	.4724						.024	.890	1.457	.16	9	1/4	1000	1620	16000	6004
6005	25	.9843	47	1.8504	12	.4724						.024	1.087	1.654	.18	10	1/4	1110	1740	14000	6005
6006	30	1.1811	55	2.1654	13	.5118						.039	1.346	1.929	.26	11	9/32	1550	2290	12000	6006
6007	35	1.3780	62	2.4409	14	.5512	2.347	2 21/32	.143	.078	.065	.039	1.543	2.205	.35	11	5/16	1910	2760	10000	6007
6008	40	1.5748	68	2.6772	15	.5906	2.552	2 59/64	.159	.094	.065	.039	1.748	2.441	.43	12	5/16	2090	2900	9200	6008
6009	45	1.7717	75	2.9528	16	.6299						.039	2.008	2.717	.54	13	11/32	2730	3630	8400	6009
6010	50	1.9685	80	3.1496	16	.6299	3.024	3 13/32	.159	.094	.065	.039	2.205	2.913	.58	14	11/32	2940	3770	7700	6010
6011	55	2.1654	90	3.5433	18	.7087	3.417	3 51/64	.204	.109	.095	.039	2.441	3.268	.85	13	13/32	3820	4890	6900	6011
6012	60	2.3622	95	3.7402	18	.7087						.039	2.638	3.465	.92	14	13/32	4110	5090	6500	6012
6013	65	2.5591	100	3.9370	18	.7087	3.811	4 3/16	.204	.109	.095	.039	2.835	3.661	.96	15	13/32	4410	5280	6100	6013
6014	70	2.7559	110	4.3307	20	.7874						.039	3.031	4.055	1.33	14	15/32	5480	6580	5500	6014
6015	75	2.9528	115	4.5276	20	.7874	4.402	4 25/32	.204	.109	.095	.039	3.228	4.252	1.41	15	15/32	5870	6830	5200	6015
6016	80	3.1496	125	4.9213	22	.8661						.039	3.425	4.646	1.87	14	17/32	7030	8240	4800	6016
6017	85	3.3465	130	5.1181	22	.8661	4.930	5½	.218	.109	.109	.039	3.622	4.843	1.96	15	17/32	7540	8560	4600	6017
6018	90	3.5433	140	5.5118	24	.9449						.059	3.898	5.157	2.56	14	19/32	8790	10000	4300	6018
6019	95	3.7402	145	5.7087	24	.9449						.059	4.094	5.354	2.67	15	19/32	9410	10500	4100	6019
6020	100	3.9370	150	5.9055	24	.9449	5.718	6 9/32	.250	.141	.109	.059	4.291	5.551	2.76	15	19/32	9410	10400	3900	6020
6021	105	4.1339	160	6.2992	26	1.0236						.079	4.528	5.906	3.51	15	21/32	11500	12500	3700	6021
6022	110	4.3307	170	6.6929	28	1.1024	6.443	7 3/16	.261	.141	.120	.079	4.724	6.299	4.32	14	23/32	12900	14200	3500	6022
6024	120	4.7244	180	7.0866	28	1.1024	6.837	7 19/32	.261	.141	.120	.079	5.118	6.693	5.34	15	23/32	13800	14700	3200	6024
6026	130	5.1181	200	7.8740	33	1.2992						.079	5.512	7.480	8.16	15	13/16	17600	18400	2900	6026
6028	140	5.5118	210	8.2677	33	1.2992						.079	5.906	7.874	8.62	16	13/16	18800	19000	2700	6028
6030	150	5.9055	225	8.8583	35	1.3780						.079	6.378	8.386	10.6	16	7/8	21800	21800	2500	6030

$$\frac{C}{P} = \frac{\text{basic dynamic load rating from table}}{\text{equivalent load (from formula below)}}$$

$P = XVF_r + YF_a$

where X = a radial factor given below
V = a rotation factor = 1.0 for inner ring / = 1.2 for outer ring $\Big\}$ rotating in relation to the load

Y = a thrust factor given below
F_r = the radial load, calculated
F_a = the thrust load, calculated

e is a reference value given in the table.

When $\dfrac{F_a}{VF_r}$ $\begin{cases}\text{is smaller than or equal to e use } X = 1 \text{ and } Y = 0 \\ \text{is greater than e use } X = 0.56 \text{ and Y from table}\end{cases}$

$\frac{F_a}{C_o}$ =	0.014	0.028	0.056	0.084	0.11	0.17	0.28	0.42	0.56
e =	0.19	0.22	0.26	0.28	0.30	0.34	0.38	0.42	0.44
Y =	2.30	1.99	1.71	1.55	1.45	1.31	1.15	1.04	1.00

C_o = basic static load rating from table.

[2])This refers to oil lubrication and moderate load. With grease lubrication it is generally not practical to use speeds higher than ⅔ of those shown.

Figure 4.16 Typical bearing catalog data; series 60 single row, deep groove ball bearings. (*Courtesy of SKF Industries, Inc., King of Prussia, Pa.*)

From Figure 4.15b the minimum required viscosity ν_1 is 8 cSt. The 200 SUS oil viscosity ν is converted to 40 cSt using the scales on Figure 4.15b. Note that the relationship of SUS to centistokes is not linear; therefore, the ν/ν_1 ratio used in Figure 4.15a must be calculated in centistokes. The ν/ν_1 ratio is 40/8 = 5 and from Figure 4.15a the life adjustment factor a_{23} is 4 for a standard steel. The L_{10} life, therefore, is four times the calculated catalog life of 2647 hr. If a vacuum-melted steel is used, the life adjustment factor can be as high as 7. The bearing DN value is 0.25×10^6; therefore, speed effects will not significantly downgrade the life rating.

Permissible Speeds

Note that the catalog listing includes a speed limit for the 6020 bearing, 3900 rpm. This limit is based on permissible lubricant operating temperature and is set somewhat below 250°F. Speed limits listed in bearing catalogs are not absolute limits and by using proper design practices significantly higher operating speeds are possible. The areas critical in high-speed operation are lubrication, cooling, retainer design, and centrifugal effects.

Temperature Effects

When operating bearings at high temperatures, two potential problem areas are phenolic retainers and steel stability. Phenolic retainers are temperature limited and should not be used when operating temperatures approach 250°F. The standard bearing steels are stabilized at 250°F and will change dimensionally if operated at higher temperatures. It is possible to procure bearings stabilized at temperatures above 250°F; however, this will reduce hardness and fatigue life. It is also possible to procure high-hot-hardness steels such as M50, which have excellent fatigue properties at temperatures up to 500°F.

Static Load Rating

The life load relationship would indicate that at zero speed a bearing has infinite load capacity. Of course, this is not so. One could consider the limiting static load, the load at which the bearing will fracture; however, at much lower levels of loading, permanent deformations develop in the surfaces of the contacting elements. The static load rating C_0 is dependent on the magnitude of permanent deformation that can be allowed. Experience has shown that when deformations become as large as 1/10,000 of the diameter of the rolling element, objectionable noise and vibration occur during subsequent rotation. The static load rating given in Figure 4.16 is the load that corresponds to this magnitude of permanent deformation. The fracture load is approximately eight times the C_0 value.

Stationary bearings subject to vibration may exhibit pitting at very light loads. This is a form of fretting corrosion and can affect the fatigue life of bearings that alternately rotate and are stationary, such as those used on over-running clutches.

Prorating Bearing Loads

To calculate bearing L_{10} life an equivalent load is required; however, bearings often operate under a varying load schedule. For instance, a typical operating schedule for a bearing application follows:

Load point	Time (min)	Load (lb)	rpm
1	7	300	1000
2	2	500	2000
3	1	700	3000

To find an equivalent load for the C/P term, the schedule can be prorated as follows:

$$P_E = \left[\sum \frac{N_i(P_i)^3}{N_T}\right]^{1/3}$$

where

P_E = equivalent load, lb
N_i = number of cycles at each load point
P_i = load at each point, lb
N_T = total number of cycles

For the example:

$$P_E = \left[\frac{7(1000)(300)^3 + 2(2000)(500)^3 + 1(3000)(700)^3}{7(1000) + 2(2000) + 1(3000)}\right]^{1/3}$$

$$= 497 \text{ lb}$$

Each 10-min cycle consists of 14,000 revolutions. If the C/P value is calculated to be 2, the life according to Figure 4.13 will be 8 million revolutions or 571 ten-minute cycles. The equivalent load is sometimes called the cubic mean load and the exponent used is the same as for the life formula: 3 for ball bearings and 3.3 for roller bearings. When using this method of prorating, the difference between the equivalent loads obtained with the two exponents is usually negligible; therefore, the exponent 3 is used for both ball and roller bearing calculations.

Bearing Dimensions and Tolerances

The Antifriction Bearing Manufacturers Association has arrived at standardized dimensions for rolling element bearings. These dimensions are the bore size, the outside diameter, and the width, conventionally given in millimeters. The variety of different sizes is limited so that bearings can be produced in large quantities, thereby satisfying the user's need for economical yet high-quality bearings. The basic dimension in the system is the bearing bore, which defines the shaft size upon which it is assembled. For a given bore a variety of outside diameters and widths are available in increments such that requirements for load capacity or envelope dimensions are satisfied. Through the dimensioning system, bearings from different manufacturers or of different types can be interchanged. It must be remembered, however, that although the bore, outside diameter, and width of two bearings are the same, the internal geometry may be quite different. This can affect the bearing capacity and operation. Manufacturers also offer a variety of options, such as integral seals for grease-packed bearings, snap ring grooves, and so on.

The Antifriction Bearing Manufacturers Association has set up a series of quality classes known as grades ABEC-1, 3, 5, and 7. ABEC-1 is the standard quality to which catalog bearings conform. The higher numbers correspond to better quality: that is, smaller tolerances on the bore, outside diameter, and width dimensions, as well as closer control of eccentricities, parallelism, and squareness. Also, the higher classes will have better raceway surface texture and the variation in ball or roller size within a bearing will be progressively less. Grade 3 bearings are selected from the standard grade 1 production by inspection. Grades 5 and 7 are manufactured separately with higher-precision processing. The higher-accuracy grades are used when there are requirements for especially smooth operation with low noise and vibration. An example of such an application is a machine tool spindle. Also, high-speed bearing applications should incorporate more precise bearings. Gearbox shafts operating above 3600 rpm or 5000 fpm peripheral speed should be mounted on ABEC-5 bearings.

Internal Clearance

The internal clearance of a bearing is defined as the total distance one bearing ring can move in relation to the other ring in a radial direction under no load. Ideally, bearings should operate with little or no radial clearance; however, this condition is difficult to attain. The designed bearing clearance must take into account the reduction in clearance if one or both of the raceways is mounted with a press fit. Pressing on an inner bearing ring will reduce the internal clearance 50 to 80% of the amount of the interference fit. Also, in operation the inner ring will tend to be at a higher temperature than the outer ring. The outer ring runs cooler since heat is more easily dissipated through the bearing housing to the atmosphere than from the inner ring to the shaft. This temperature differential means the inner ring expansion will reduce the bearing clearance. Catalog bearings have standard clearances which are set such that the bearing will operate properly when one of the raceways is mounted with a standard press fit. For unusual conditions bearings are available with internal clearances smaller and greater than standard. Bearing clearance will increase with bearing size and the clearances, together with recommended press fits, are tabulated in bearing catalogs.

Shaft and Housing Tolerances

Possible fits of inner races on shafts and outer races in housings range from loose to transition to tight. One of the most important factors in determining suitable fits at the bore and outside diameter is the relationship of the load to the bearing rings. The load can be either stationary or rotating with respect to the rings. For instance, in a gear shaft system using bearings with rotating inner rings the load will always act in the same direction; therefore, the inner ring rotates in relation to the load and all points on the inner race come under load each revolution. If the inner ring were loose on the shaft, the relative motion of the load would cause the ring to creep around the shaft, creating wear, fretting corrosion, and the possibility of crack initiation. In the case of a stationary load and rotating inner ring, therefore, the ring must have a tight fit on the shaft such that no clearance exists and none can be developed by the action of the load. The higher the load, the greater the press fit required. The outer ring, which in this case is stationary with respect to the load, can be slightly loose in the housing to ease assembly and disassembly. Clearance should be minimal, for the following reasons:

1. A large clearance would allow the ring to cock at assembly.
2. A large clearance would reduce the centering ability of the bearing.
3. The outer ring can deform under load if not securely assembled in the housing.

If the outer ring rotates and the load is stationary, the outer ring should be assembled with a press fit to keep it from spinning. Quite often in shaft systems there is a rotating load superimposed on stationary loads due to unbalance. This rotating load tends to make loose-fitting outer races creep. A solution to this problem is to restrain the outer ring from rotating with a pin or other locking device.

The bearing shaft ideally should have a minimum hardness of Rc40 and be ground to size. If these requirements are not practical in a particular application, a tighter fit than normal should be used. In many applications bearing inner rings are clamped axially. The axial restraint should not be counted on to restrain the bearing from creeping and should not be used in place of a press fit.

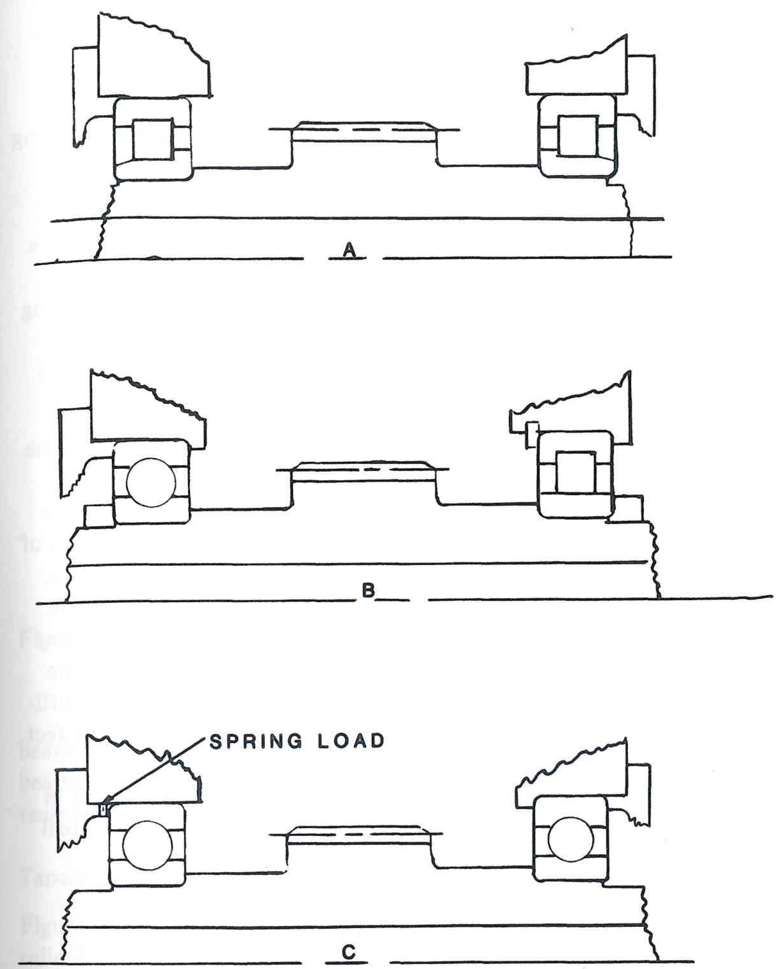


Figure 4.17 Typical bearing mounting configurations.

Bearing Mounting

The bearing arrangement must offer radial support and axial location. Also, it must accommodate thermal expansion and deflections due to load. Figure 4.17 shows three common design solutions for the support of a gear shaft. Figure 4.17A illustrates a spur gear that generates no axial loads supported by two roller bearings. The shaft is axially located by the bearing outer race shoulders and one shoulder on each inner race. It is extremely important in this type of design to allow sufficient endplay to accommodate thermal expansion of the shaft. A detailed tolerance stack-up must be made to ensure endplay under the worst dimensional conditions. A potential problem with this type of design at high speeds is bearing skidding. This condition occurs when the bearings are unloaded and do not have sufficient tractive force to keep them rolling. Skidding will result in scoring and wear and may progress to complete bearing failure. Some solutions for skidding problems are:

1. Reduction in bearing clearance such that the bearing always operates with a slight preload giving sufficient tractive force.
2. Use of out-of-round outer races which impart a constant load on the bearing in the zone where the rollers are pinched.
3. Reduction of forces that tend to retard rolling. In some cases lubrication is excessive. Retainer design can alleviate the problem.

Figure 4.17B has a ball bearing which can react thrust in either direction on one end of the shaft and a roller bearing on the other. The roller bearing has no shoulders on the inner race and is free to accommodate thermal distortions. In this case the gear might be helical. Care should be taken that the combination of thrust and radial load on the ball bearing does not result in insufficient life. Figure 4.17C illustrates a two-ball-bearing arrangement where the bearings are preloaded by a spring or wave washer. This arrangement has the advantage that at full speed and light load the spring ensures some bearing preload and the chance of skidding is minimized. For some axial distance the spring rate is sufficiently linear such that thermal expansion is accommodated and does not affect bearing loading.

For heavy combined radial and thrust loads the arrangement shown in Figure 4.18 can be used. The relief around the outside the diameter of the ball

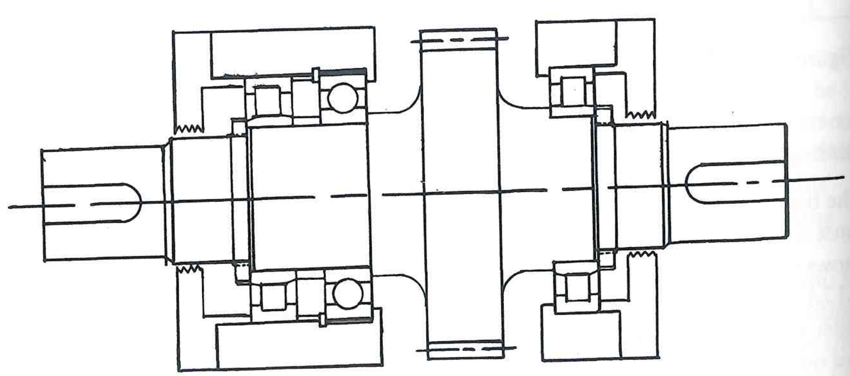


Figure 4.18 Mounting configuration for combined radial and thrust loads.

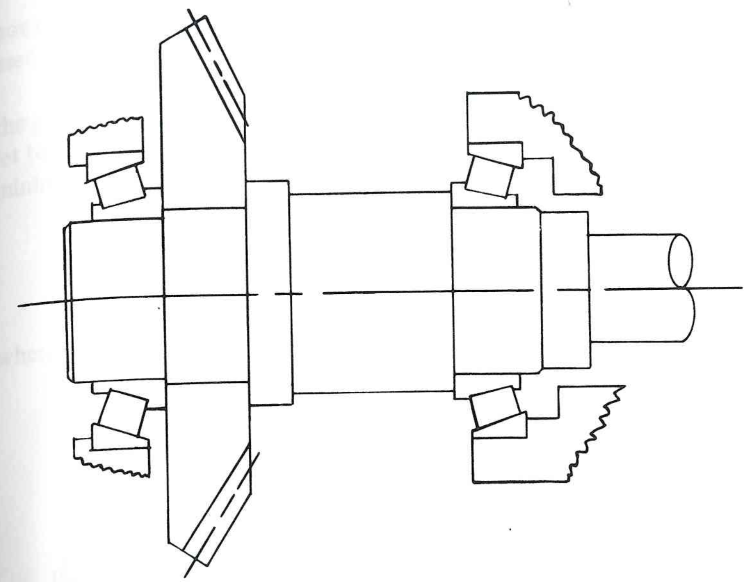


Figure 4.19 Tapered roller bearings in straddle-mounted configuration.

bearing ensures that all radial loads are reacted by the roller bearings and the ball bearing reacts only thrust. The rollers are free to move axially to compensate for temperature effects.

Tapered Roller Bearing Mounting

Figures 4.19 and 4.20 show two common ways to mount bevel gears in tapered roller bearings. Figure 4.19 illustrates a straddle mount where the gear is supported on either side. In the case shown the bearings are set up in a face-to-face configuration. In Figure 4.20 the gear is overhung mounted. A pair of tapered roller bearings takes thrust in one direction while a single bearing reacts thrust in the opposite direction. The bearings are set up in a back-to-back configuration. Care must be taken in the assembly to provide sufficient endplay for thermal expansion of the shaft. In calculating thrust loads on a shaft supported by tapered roller bearings it must be remembered that due to the angle of contact, a radial load on one bearing induces an axial load that must be reacted either by an external load or the other bearing. The induced axial load depends on the bearing internal geometry, and methods of calculation are included in catalogs of tapered roller bearings.

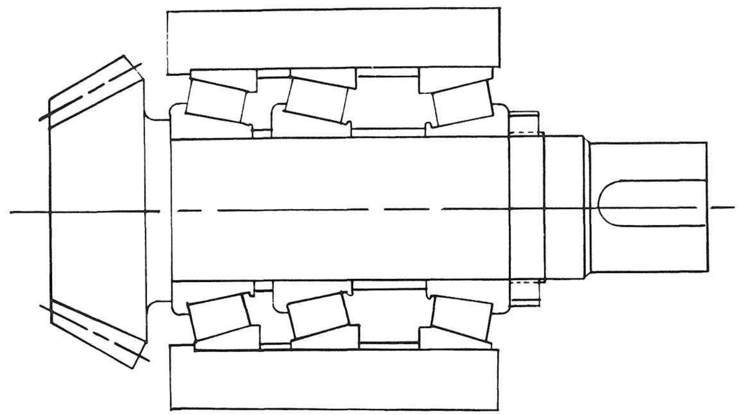


Figure 4.20 Tapered roller bearings in overhung-mounted configuration.

Bearing Lubrication

The type of lubrication to which rolling element bearings are exposed in gear units is dictated by the gear oil requirements. Gear boxes discussed in this book typically incorporate oil splash or jet lubrication systems and the bearing lubrication discussion will be limited to these areas. The types of oil used are discussed in Chapter 5.

The functions of a rolling element bearing lubricant are:

1. Provide an oil film between contacting elements.
2. Provide a cooling medium.
3. Protect bearing surfaces against corrosion.

Very little oil is required to maintain a satisfactory oil film; however, for high-speed highly loaded bearings significant oil flow must be supplied to perform the cooling function.

In a splash lubrication system the gearbox is filled with oil to a level that the gears and bearings are dipping into the lubricant as they rotate. At low speeds the heat generated can be dissipated through the casing and the unit reaches a satisfactory equilibrium temperature. Typical of such applications are electric or hydraulic motor speed reduction units. If heat generation becomes excessive the casing can be air or water cooled and an integral splash lubrication system may still be satisfactory. As speeds and loads increase a circulating system may be necessary with an external oil cooler.

In high-speed boxes, gear pitch line velocities of 5000 fpm or more, jet lubrication is used. At high speeds it is impractical to allow the components to rotate through an oil bath since the churning would create excessive heat and

power loss. In some high-speed, light-load applications oil mist lubrication can be used to avoid churning losses in the bearing.

With jet lubrication one or more jet streams are directed on the bearing at the gap between the retainer and race. It is good practice to use more than one jet to ensure against oil starvation if a jet clogs. The jet diameter should be a minimum of 0.030 in. also to avoid clogging.

Flow through an oil jet can be calculated as follows:

$$Q = KA\sqrt{\frac{2g\,\Delta p}{w}}$$

where

Q = oil flow, in.^{3}/sec
A = jet area, in.^{2}
K = discharge coefficient (assume 0.65)
g = acceleration due to gravity, 386 in./sec^2
Δp = pressure drop across orifice, lb/in^2
w = specific weight, lb/in.^{3}

In order to calculate the amount of flow required to cool the bearing an estimate of the heat generated is needed. Through experience, empirical methods have been generated which approximate the losses due to friction and oil churning. Two such methods will be presented. The first method [6] estimates losses as

$$T = fRW$$

where

T = frictional torque, in.-lb
f = friction coefficient
R = shaft radius, in.
W = load, lb

The following table lists friction coefficients for various bearing configurations.

Bearing type	Friction coefficient
Radial ball bearings	0.0015
Self-aligning ball bearings	0.0010
Angular-contact bearings	0.0013
Pure thrust ball bearings	0.0013
Cylindrical roller bearings	0.0011
Spherical roller bearings	0.0018

From the friction torque the horsepower or heat generation can be calculated:

$$HP = \frac{TN}{63{,}025}$$

where N is the bearing rpm.

$$Q = HP(42.44)$$

where Q is the heat generated in Btu/min. This method does not consider the amount of oil flow or the oil viscosity directly. Another empirical procedure that includes these parameters is [7]:

$$Q = B[(DN)^{1.5}W^{0.07}M^{0.42}\mu^{0.25}]$$

where

Q = heat generated, Btu/min
D = bearing bore, mm
N = bearing rpm
W = bearing load, lb
μ = dynamic viscosity, Reyns (lb-sec/in.^{2})
M = oil flow, lb/hr

The following table lists values for the coefficient B:

Type of bearing	B coefficient
Angular contact	10.1×10^{-7}
Radial ball	4.46×10^{-7}
Cylindrical roller	6.46×10^{-7}

For a given flow and heat loss the oil temperature rise across the bearing can be calculated:

$$\Delta T = \frac{Q/M(c_p)}{60}$$

where

ΔT = temperature rise, °F
M = oil flow, lb/hr
c_p = specific heat, approximately 0.5 Btu/lb-°F

The conversion from GPM to lb/hr for oil is approximately

$$1 \text{ GPM} \cong 450 \text{ lb/hr}$$

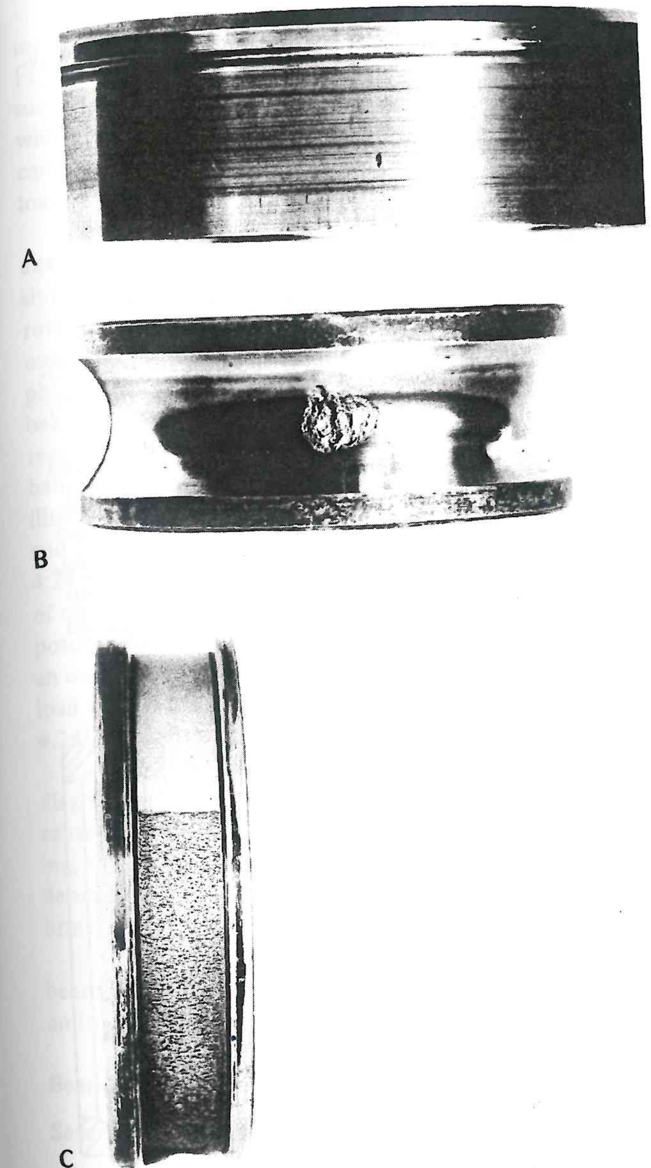


Figure 4.21 Progression of fatigue spalling. A. Incipient fatigue spalling. B. More advanced spalling. C. Greatly advanced spalling. (Courtesy of SKF Industries, Inc., King of Prussia, Pa.)

Bearing Failures

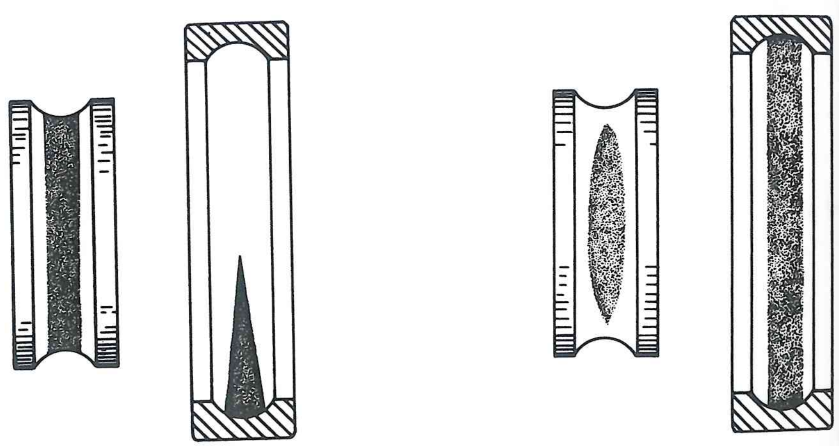
Rolling element bearing failures are characterized by one or more of the following operating conditions:

1. Excessive vibration
2. Excessive noise
3. Overheating
4. Chip generation
5. Hard turning or excessively loose shafts

These problems can be a result of a great variety of errors but can usually be categorized as either load or lubrication related.

Lubrication-related failures occur when the supply of lubricant is insufficient or misdirected. The mating bearing surfaces, without a proper oil film, come into intimate contact, resulting in wear, heat generation, and thermal expansion. The thermal expansion reduces the bearing internal clearance and therefore increases the bearing load and the failure becomes self-perpetuating. Quite often a failure that appears to be due to overload masks a lubrication problem that led to loss of internal clearance and internally generated loading.

Load-related failures can be a result not only of the operating loads but also of forces applied during assembly. Also, the effect of operating loads can be magnified due to defective bearing seats on shafts or in housings or misalignments.



A B

Figure 4.22 Normal radial load-bearing patterns. (Courtesy of SKF Industries, Inc., King of Prussia, Pa.)

The mode of failure related to load that is most common is fatigue spalling. Figure 4.21 shows the progression of fatigue spalling. The small spall in Figure 4.21A began with a crack which probably originated below the race surface. This initial spall will advance to the size shown in Figure 4.21B, at which point the vibration and noise level of the unit will have increased significantly. Left unattended, the spalling will progress in proportion to speed and load as shown in Figure 4.21C.

Sometimes a clue to the cause of bearing problems can be found by examination of the pattern of the load path on the bearing races. Figure 4.22 shows the normal load pattern of a radial bearing. Figure 4.22A illustrates a rotating inner ring operating with a stationary load. The pattern of a rotating outer ring with a stationary load or a rotating inner ring with a load rotating in phase is shown in Figure 4.22B. Figure 4.23 illustrates load patterns on a thrust bearing. Figure 4.23A is a nornal pattern which stays within the raceway, not reaching out to the edge. In Figure 4.23B the thrust load is excessive and the balls contact the race edge, resulting in a load concentration. Figure 4.23C illustrates a normal contact pattern for a bearing experiencing combined radial and thrust load. Figure 4.24 shows load patterns that reflect problems. Figure 4.24A illustrates a bearing that is internally preloaded. This may be due to loss of clearance because of excessive press fits on the shaft or in the housing or possibly thermal expansion. Figure 4.24B shows the load pattern produced by an out-of-round housing pinching the bearing outer ring. Figure 4.24C shows the load zone when the outer ring is misaligned relative to the shaft, and Figure 4.24D the pattern when the inner ring is misaligned with respect to the housing.

Two types of pitting failures that occasionally occur and are hard to diagnose are false brinelling and electrical pitting. False brinelling is a condition caused when the gear unit is subject to vibration while the shafts are not rotating, such as in transit during shipping. It is usually characterized by polished depressions spaced equal to the distance between rolling elements. False brinelling is a form of fretting corrosion.

Electrical pitting occurs when a current seeks ground by passing through a bearing. The current is broken where the balls or rollers contact the raceway and arcing results, causing high temperatures and pitting damage.

Bearing Costs

Several variables influence the cost of gearbox bearings:

1. Obviously, when buying bearings in production quantities rather than prototype, lower prices prevail. The quantity can also determine whether the bearings are bought from a manufacturer, distributor, or supply house, which can affect the price significantly.

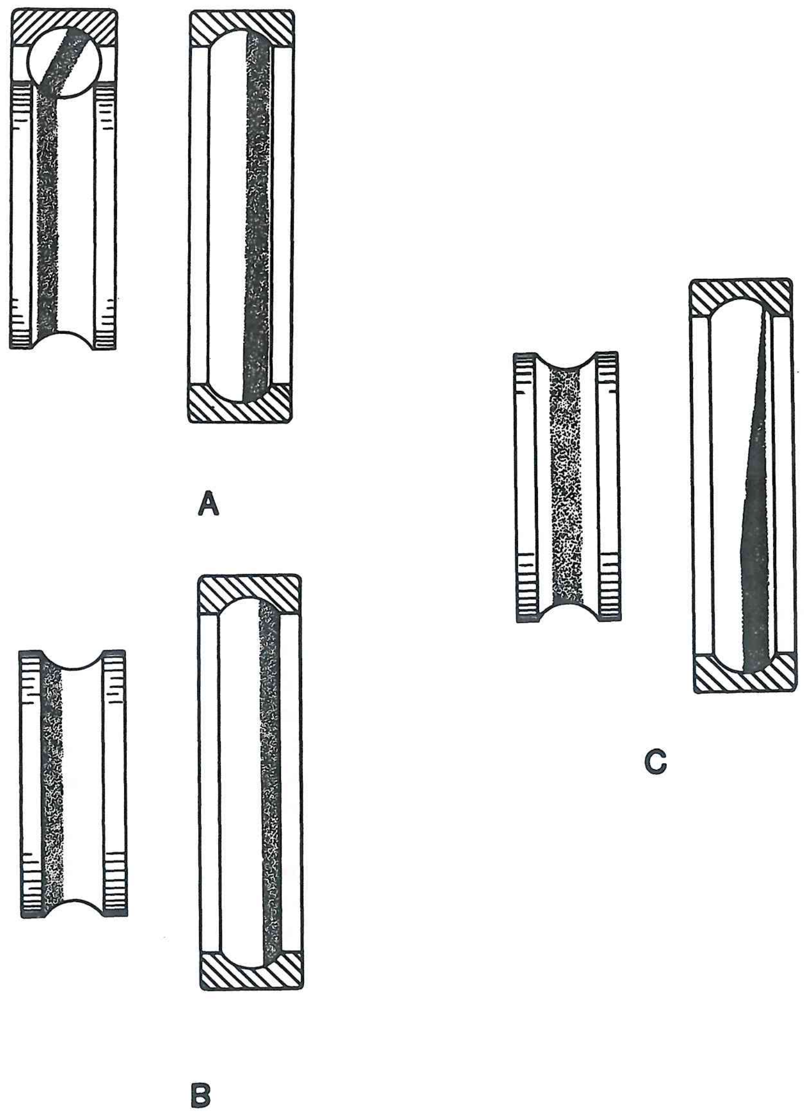


Figure 4.23 Normal thrust load-bearing patterns. (Courtesy of SKF Industries, Inc., King of Prussia, Pa.)

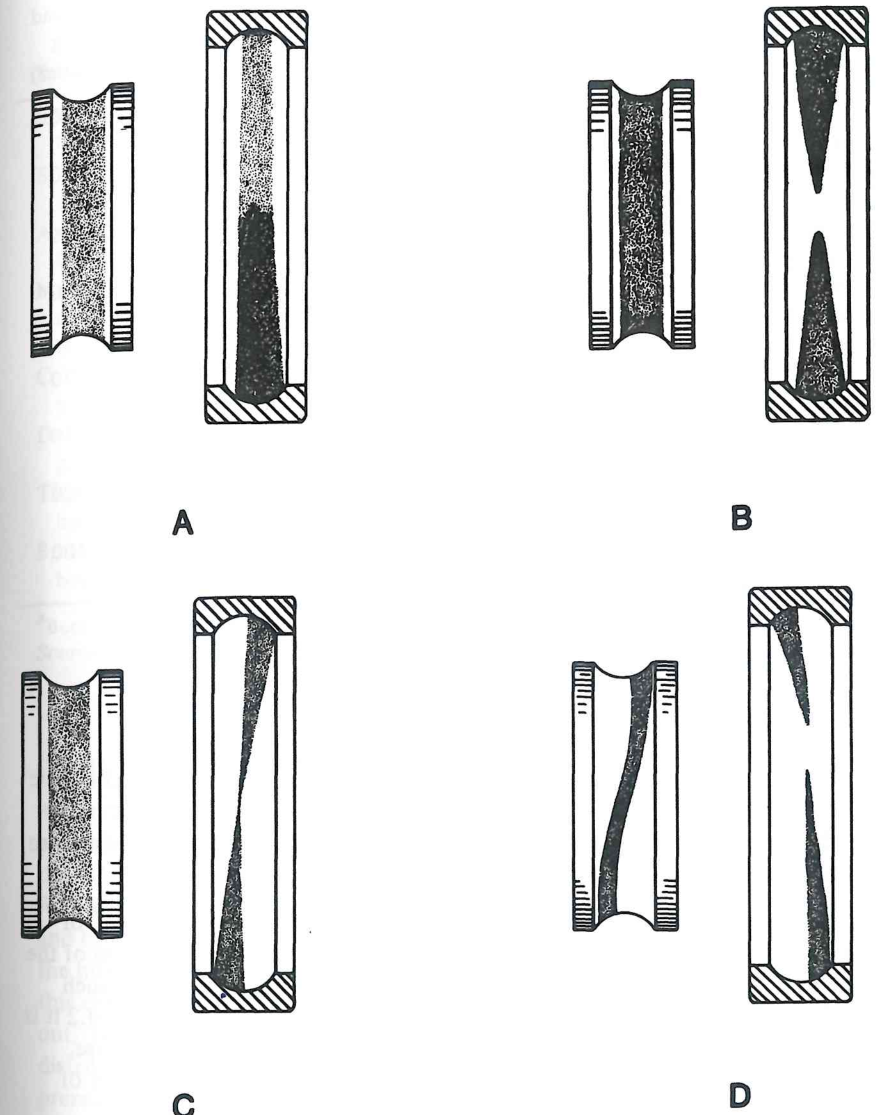


Figure 4.24 Abnormal bearing load zone patterns. (Courtesy of SKF Industries, Inc., King of Prussia, Pa.)

Table 4.1 Cost Comparison, 75-mm Bearing

Bearing type	Basic load rating (lb)	Limiting speed (rpm)	Relative cost	Cost of basic load rating (lb/dollar)
Self-aligning ball bearing	13,700	4,200	1.69	810
Single-row, deep groove ball bearing	19,600	4,200	1.32	1.480
Angular contact (40°) ball bearing	21,500	4,200	2.10	1,030
Maximum capacity, deep groove ball bearing	25,700	4,200	1.52	1,700
Cylindrical roller bearing	36,500	4,200	3.53	1,030
Double row, deep groove ball bearing	30,200	2,900	2.87	1,050
Tapered roller bearing[a]	50,100	2,460	1.00	5,010
Spherical roller bearing	68,000	2,500	4.00	1,700

[a]Bore: 3.000 in., O.D.: 5.909 in.
Source: Ref. 8.

2. Design options such as special material or processing add greatly to the basic cost.
3. Higher-than-standard precision requirements or internal clearances which are nonstandard have a significant price effect.

An interesting study was conducted investigating the comparative cost of bearings on a basic load rating per dollar basis [8]. Various configurations of the same two sizes, 55- and 75-mm bore, were evaluated to determine how much load capacity each bearing offers per dollar of cost. From Tables 4.1 and 4.2 it is seen that tapered roller bearings offer the best value on this basis. Of course, many applications have design conditions which dictate that other types of bearings are required.

It must be remembered that the initial bearing cost is not the only expenditure to be considered. If failure occurs, the expense of disassembly and reassembly must be considered, and also the downtime cost.

Table 4.2 Cost Comparison, 55-mm Bearing

Bearing type	Basic load rating (lb)	Limiting speed (rpm)	Relative cost	Cost of basic load rating (lb/dollar)
Self-aligning ball bearing	4,630	6,500	1.99	982
Single-row, deep groove ball bearing	7,500	6,500	1.42	2,220
Angular contact (40°) ball bearing	8,010	6,500	2.39	1,411
Maximum capacity, deep groove ball bearing	9,830	6,500	1.70	2,430
Cylindrical roller bearing	10,300	6,500	5.70	762
Double row, deep groove ball bearing	11,400	4,500	2.92	1,650
Tapered roller bearing[a]	15,000	3,720	1.00	6,330
Spherical roller bearing	19,300	4,000	6.02	1,350

[a]Bore: 2.1653 in.
Source: Ref. 8.

Journal Bearings

A simple radial bearing is shown in Figure 4.25. At rest, the shaft (journal) will lay on the bottom of the bearing with a load W, the shaft weight. At this point the lubricant is squeezed out and metal-to-metal contact occurs (Figure 4.26A). As the shaft begins to rotate (Figure 4.26B) it will climb up the bearing wall and be slightly off-center. The rotation of the shaft will tend to pull oil into the interface between the shaft and bearing, a wedge-shaped zone. The inlet to this converging region would like to take more oil in than the outlet will allow out. The jamming of the fluid into the converging region creates a pressure distribution between the journal and bearing as shown in Figure 4.27. The high pressure in the center forces the fluid to slow down at the inlet and speed up at the outlet, so that the flow coming in equals the flow going out. The pressure generated creates a lifting force which separates the journal and bearing with an oil film (Figure 4.26C). This pressure is the basis of hydrodynamic lubrication and the load capacity of the bearing depends on the hydrodynamic pressure

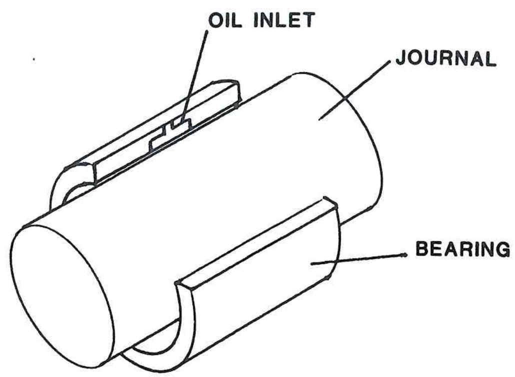


Figure 4.25 Simple radial journal bearing.

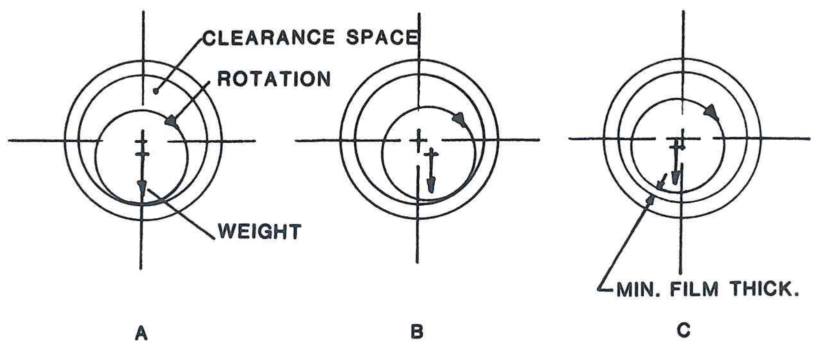


Figure 4.26 Journal bearing operation.

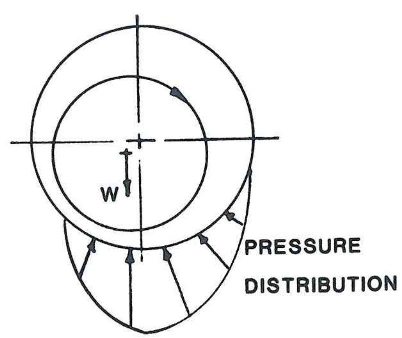


Figure 4.27 Hydrodynamic pressure distribution.

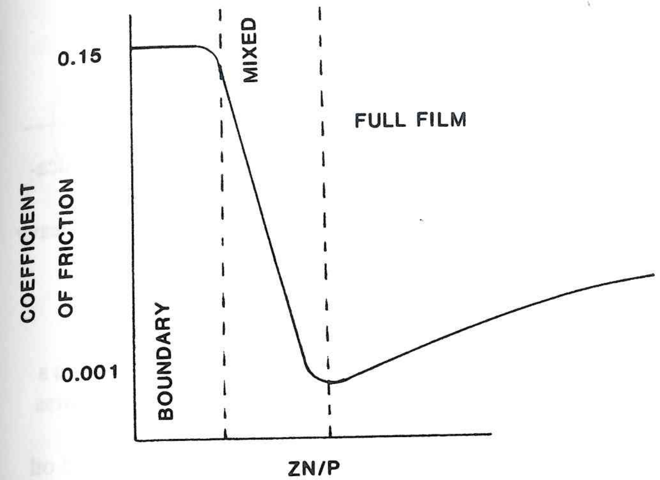


Figure 4.28 Regimes of lubrication.

generated. Increasing the fluid viscosity or the shaft velocity will increase the hydrodynamic pressure and therefore increase the film thickness. Typical hydrodynamic bearings operate with film thicknesses in the range 0.0005 to 0.002 in.

As a bearing accelerates from rest it passes through three regimes of lubrication. The first regime is boundary lubrication, where there is metal-to-metal contact. The second is mixed lubrication, where there is a transition from boundary, to hydrodynamic lubrication, where the bearing operates with a full fluid film. These regimes are illustrated in Figure 4.28, where the coefficient of friction is plotted against a bearing parameter:

$$\text{Bearing parameter} = \frac{ZN}{P}$$

where

Z = viscosity, cP
N = shaft rpm
P = unit loading, psi

The unit loading is defined as

$$P = \frac{W}{LD}$$

where

W = radial load, lb
L = bearing length, in.
D = bearing diameter, in.

The minimum ZN/P value to assure full film lubrication varies with the application. As a rule of thumb, for lightly loaded high-speed bearings with low-viscosity oils it should exceed 150. Low-speed, high-load applications with heavy oils may have ZN/P values as low as 10.

Pressure-Fed Bearings

Gearbox journal bearings are usually supplied oil under pressure. This assures a reliable source of lubricant and also provides a cooling medium. Feed pressures are typically in the range 20 to 50 psig. The oil flow to the bearing must be sufficient to replace oil lost by leakage at the bearing ends. Usually, the feed oil passages are sized sufficiently large such that the flow is metered by the bearing itself. There are various designs of pressure-fed radial bearings in use, the difference being in the shape and location of the oil feed grooves and the shape of the bearing bore.

Oil distribution grooves should not be located in the load zone since this will disrupt the hydrodynamic pressure distribution and reduce the load-carrying capacity.

A problem that arises with high speed bearings when they are lightly loaded is oil whip or whirl. It is a vibration phenomenon which occurs at either half the rotational speed or the natural frequency of the shaft when the shaft is operating well above the natural frequency. One explanation of whirl is that a wedge of oil is traveling around the bearing at the average oil velocity (one-half shaft velocity) or at the shaft critical velocity. Bearing bore designs other than the simple full round bearing (Figure 4.25) are meant to minimize this whirl tendency.

Following are descriptions of some commonly used gearbox radial bearings:

1. *Axial groove* (Figure 4.29A). This type of bearing has a cylindrical bore and one or more axial oil spreader grooves extending almost the full length of the bearing. It has high load capacity but at light loads, usually under 150 psi, is quite susceptible to oil whip.
2. *Elliptical* (Figure 4.29B). The elliptical bearing has a large clearance in one direction and a smaller clearance 90° away. It is manufactured by placing shims at the split between two halves of a cylindrical bearing and then machining the bore. The shims are removed, producing a variation in clearance around the bore, which is not truly elliptical. This type of bearing

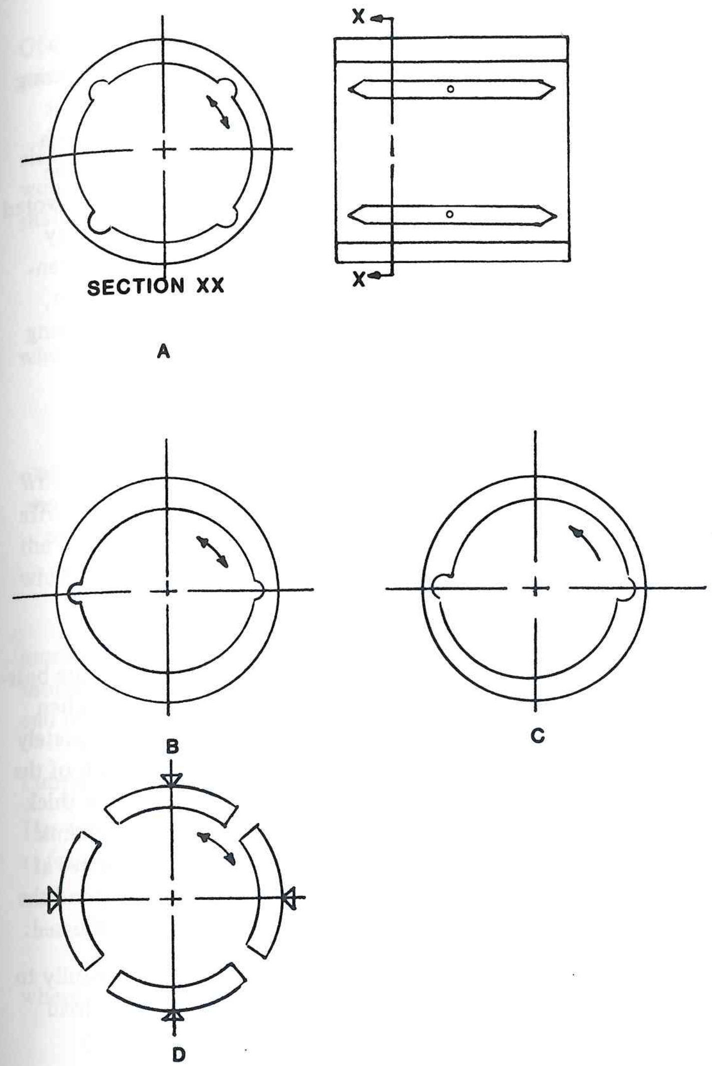


Figure 4.29 Types of pressure-fed radial bearings.

allows larger oil flows than a cylindrical axial groove design and therefore affords more cooling. It is also somewhat more stable as regards oil whip.
3. *Offset half* (Figure 4.29C). This is a modification of an axial groove bearing where the upper and lower halves are offset. It has proved successful in eliminating oil whip. A disadvantage of this design is that it can accommodate shaft rotation in only one direction.

4. *Tilting pad* (Figure 4.29D). The individual pads of this design are free to pivot and follow shaft excursions; therefore, forces produced in the bearing are incapable of driving the shaft into an unstable mode and operation is smooth. Each pad tilts such that a wedge-shaped oil film is formed which tends to center the shaft. The pivot can be located anywhere from the center of the pad to near the trailing edge. In one design the pads are pivoted near the trailing edge and the other edge is forced toward the shaft by springs. This preload adds significantly to high-speed stability. A disadvantage to this type of bearing is that it must be flooded with oil; therefore, required flows are high with accompanying power losses. Also, the bearing is mechanically complex and the parts are prone to fretting.

Relative Bearing Costs

Of the pressure-fed bearing designs the axial groove configuration is easiest to fabricate and has the lowest cost. Elliptical bore and offset half bearings are somewhat more expensive and tilting pad bearings are significantly more expensive.

Design and Rating of Radial Journal Bearings

The gearbox designer tends to be concerned with simple criteria such as the bearing unit loading, its length-to-diameter ratio, and the clearance required when defining radial journal bearings. These concerns are not sufficient to adequately design a bearing and the bearing specialists are turned to for optimization of the final design. The significant parameters to be analyzed are minimum film thickness, maximum bearing surface temperature, and maximum hydrodynamic pressure. These parameters must be traded off against one another to arrive at the optimum design. The mathematical design of journal bearings is beyond the scope of this book, but the more commonly used parameters will be discussed:

Unit Loading Typical gearbox radial bearings can operate successfully to unit loadings of approximately 550 psi. By unit load is meant the radial load divided by the projected bearing area:

$$P = \frac{W}{LD}$$

where

L = bearing length, in.
D = bearing diameter, in.

More stringent criteria are used in some cases where extreme reliability is desired. For instance, some company specifications limit unit loading to 350 psi.

Of course, the loading allowed is dependent on the bearing material. This is discussed in a later section.

Clearance Many designers as a rule of thumb use 0.001 in. of diametral clearance per inch of shaft diameter. In other words, a 6-in.-diameter journal would operate in a 6.006-in.-diameter bearing. This rule of thumb does not take shaft speed into account and a better estimate of minimum clearance is [9].

$$\text{Minimum diametral clearance} = \frac{N^{0.25}D}{6} + \frac{1.0}{D} \times 10^{-3}$$

where

N = rpm
D = shaft diameter, in.

When a bearing is pressed into a housing the bore will tend to collapse and thus affect the clearance. Clearance, therefore, should be specified after assembly or the bore made sufficiently oversize to accommodate the collapse after press fit, which may be as much as 75% of the press.

Length-to-Diameter Ratio The length-to-diameter ratio of gearbox journal bearings is usually around 1.0, although it can vary from 0.3 to 2.0. In shorter bearings there is a reduction in load-carrying capacity due to excessive end leakage. Longer bearings are susceptible to end loading due to misalignment.

Lubrication

The amount of oil leakage from both ends of a journal bearing can be estimated as follows [10]:

$$Q = 0.816\,\frac{rc^3 p_s}{\mu L}$$

where

Q = total flow, gal/min
r = shaft radius, in.
μ = oil viscosity, Reyns
p_s = oil supply pressure, psi
L = uninterrupted bearing land length, in.
c = radial clearance, in.

The viscosity μ should be taken at the bearing operating temperature, which can be estimated as the average of oil temperature into and out of the gearbox. If the bearing has a full circular oil feed groove, the length L will be approximately half of the overall bearing length.

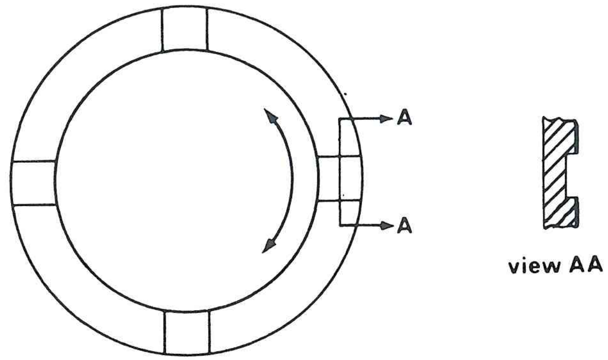


Figure 4.30 Simple thrust washer.

The oil chosen will depend on the unit speed and loading conditions. High-speed gear units typically use light turbine oils of approximately 150 SUS viscosity at 100°F. Lower-speed, highly loaded units will use heavier oils and occasionally EP additives.

After adequate oil flow, oil cleanliness is a prime factor controlling bearing life. The need for cleanliness is obvious when one considers that the magnitude of film thickness is sometimes as low as 0.0005 in. Proper cleaning of the gear unit prior to operation and continuing filtration of the lubricant cannot be too highly stressed.

Thrust Bearings

Thrust Washers Figure 4.30 illustrates the simplest type of journal thrust bearing, a flat stationary surface which reacts thrust from a flat runner which is either integral or attached to the rotating shaft. Because there is no taper to promote hydrodynamic action, this type of bearing normally should not be designed for unit loadings above 75 psi. Higher loads are possible but require detailed design and development.

The bearing is lubricated by directing oil to the inside diameter, the oil passing through radial oil feed grooves to the outside. To simplify manufacture an even number of grooves is usually specified. Various groove designs are used, which are usually as deep as they are wide. The load-carrying lands should be approximately square.

Tapered Land A tapered land thrust bearing is illustrated in Figure 4.31. The pad lands between oil grooves are tapered such that the runner will carry oil into a wedge-shaped region and build up a load-carrying hydrodynamic pressure. The taper may be simple or compounded, being larger at the inside

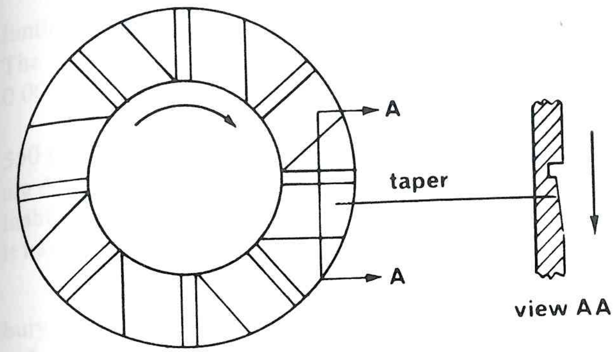


Figure 4.31 Tapered land thrust bearing.

diameter of the pad than at the outside diameter. This is done in an attempt to equalize oil flow across the pad and maintain an even temperature distribution to reduce distortion. To maintain some load-carrying ability when the bearing is starting up, the taper only extends for approximately 80% of the pad. The width-to-length ratio of the pads is approximately 1.0. Oil is fed to the inside diameter of the bearing and distributed outward through radial oil grooves. The grooves are usually dammed at the ends with a small chamfer cut to pass a

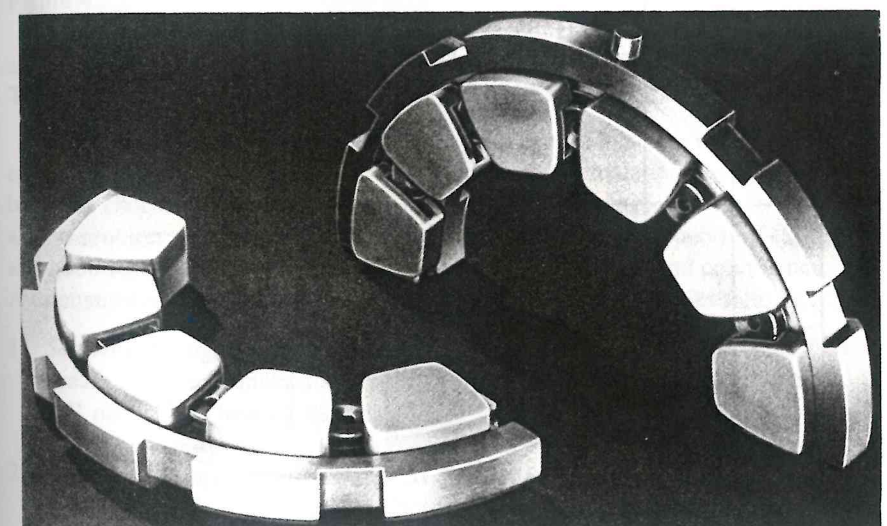
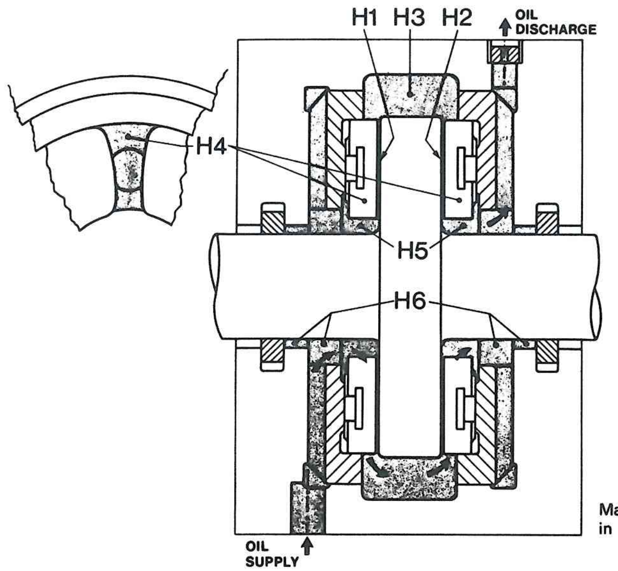


Figure 4.32 Tilting pad thrust bearing. (Courtesy of Glacier Metal Company Ltd., Wembley, England.)



Main sources of power loss in flooded double thrust bearing.

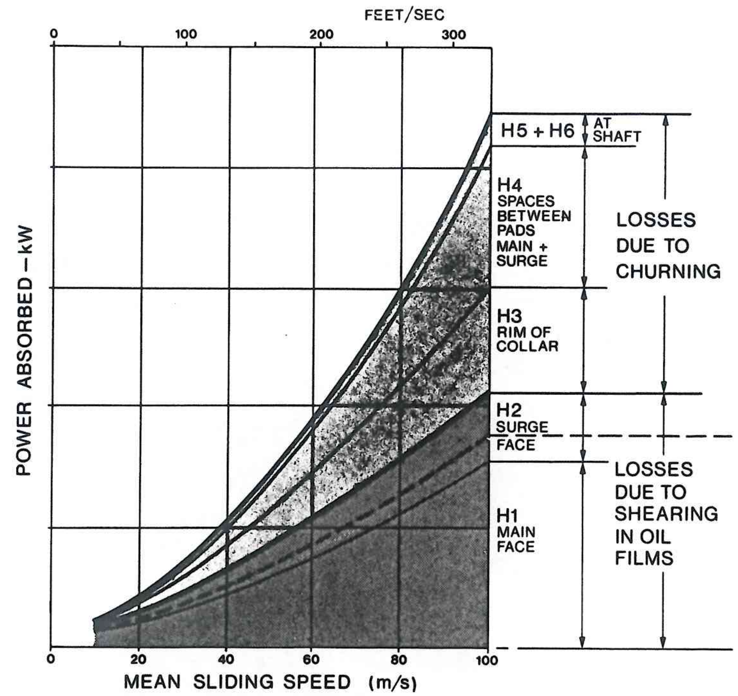


Figure 4.33 Tilting pad thrust bearing losses. (Courtesy of Glacier Metal Company Ltd., Wembley, England.)

limited amount of oil and prevent the accumulation of dirt at the groove ends. The amount of taper varies with the bearing size. A 1-in.^{2} pad may have a 0.004-in. taper, while a 7-in.^{2} pad may have a 0.008-in. taper.

Tapered land thrust bearings are used at unit loadings up to approximately 550 psi. In some critical applications where extreme reliability is required, they are derated to as low as 150 psi. Disadvantages of this type of bearing are its inability to accommodate significant misalignment and that because of the taper it has high load-carrying capacity in one direction of rotation only.

Tilting Pad This type of bearing, sometimes called pivoted shoe or Kingsbury, consists of a number of pads which are independently free to pivot in order to provide a tapered oil film (Figure 4.32). At rest the pads are parallel to the shaft face. As the shaft rotates, an oil film is generated and each pad tilts to an angle such that the oil pressure is evenly distributed. The oil is usually fed into the center of the bearing near the shaft and exits at the bearing outside diameter. As shown in Figure 4.33, the bearing area is sealed to ensure that the bearing operates in a flooded condition. Because of this, tilted pad bearings tend to generate more heat than other types and Figure 4.33 illustrates the losses in the bearing due to shear of the oil film and also due to churning. The churning losses can be significantly reduced by directing the lubrication to the pad face only (Figure 4.34).

Tilting pad bearings can accommodate shaft rotation in both directions if the pad pivots are located in the middle of the pads. To obtain even greater tolerance for misalignment pads are sometimes mounted on leveling plates, Figure 4.35.

Tilting pad bearings can withstand unit loadings of 550 psi or more and are often specified in applications where long life and reliability are paramount. The disadvantages of this type of bearing are high cost, oil flow, and power loss.

Bearing Materials Because a journal bearing experiences a variety of lubrication regimes from startup to full load and speed, the material must have a blend of characteristics to accommodate all the demands placed on it. This always requires a compromise between a high strength and temperature capacity and good surface characteristics, such as resistance to seizure and compliance. A discussion of the most important bearing material characteristics follows:

1. *Seizure resistance* At startup and on occasion during operation the oil film breaks down and metal-to-metal contact between the journal and bearing will occur. The bearing material must be capable of withstanding this contact without welding, scoring, or tearing.
2. *Ability to absorb foreign objects* If dirt particles enter the clearance space, it is desirable for the particles to become embedded in the bearing material, where they cannot score or wear the shaft.

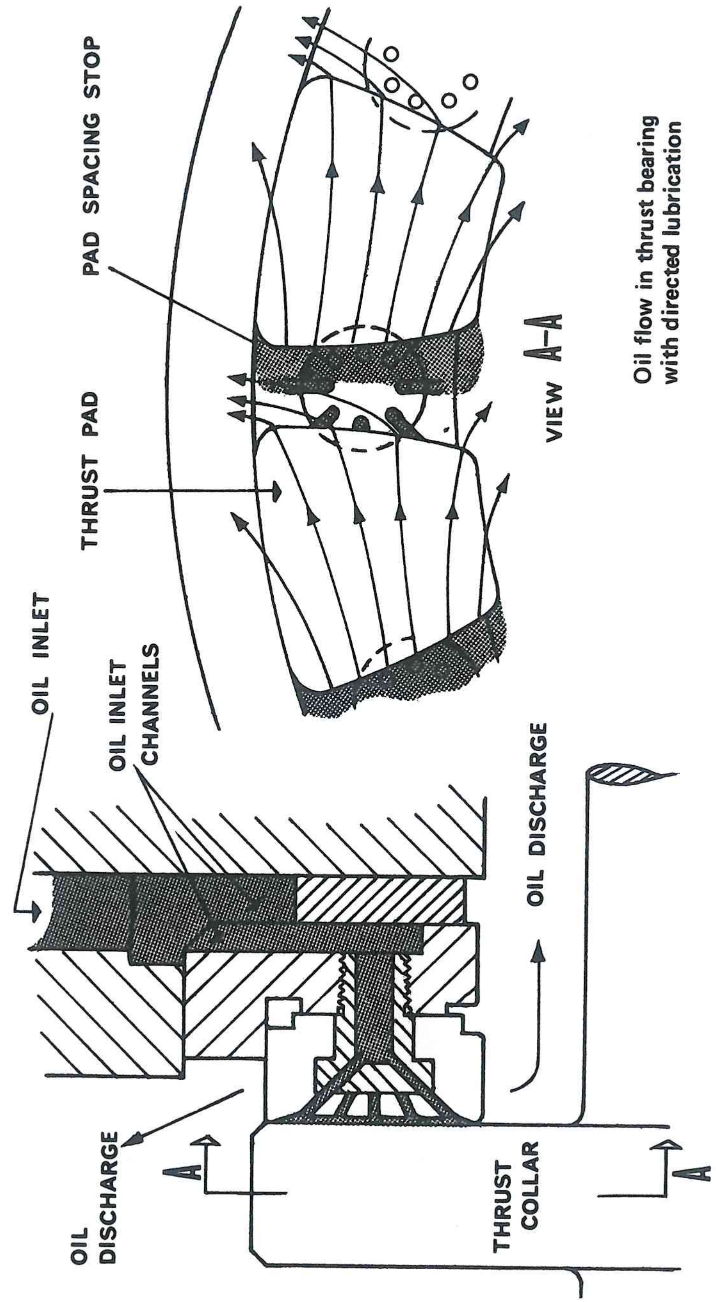


Figure 4.34 Directed lubrication of tilting pad thrust bearing. (Courtesy of Glacier Metal Company Ltd., Wembley, England.)

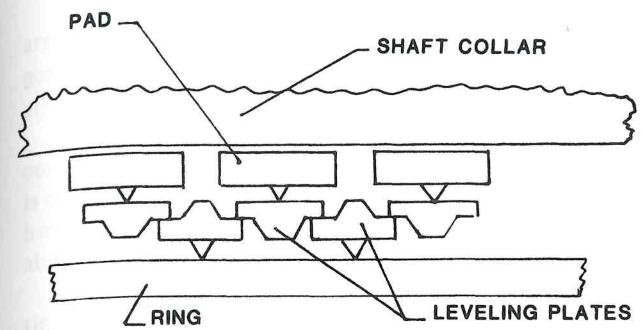


Figure 4.35 Leveling plate design for greater accommodation of misalignment.

3. *Compliability* The bearing material modulus of elasticity should be low such that if the shaft tends to touch the bearing ends due to misalignment the bearing will conform locally and the deformation will prevent severe rubbing and excessive temperature rise.
4. *Fatigue Resistance* The bearing must have the ability to withstand repeated stress cycles. In the case of gearbox bearings the load is generally steady; however, unbalance can lead to a rotating load with fatigue potential.
5. *Corrosion Resistance* If the lubricating oil contains acid or becomes acidic due to oxidation, the bearing material must be capable of withstanding the acidic attack.

The requirements of seizure resistance, embeddability, and compliability lead toward a soft bearing material such as babbitt, while fatigue strength requires a harder material. In the majority of gearbox bearings the bearing construction consists of a thin surface layer of babbitt bonded to a mild steel backing. In some cases a three-layer or trimetal construction is used with a higher-strength bearing material bonded to a steel backing and overlaid with a thin babbitt. A discussion of the most widely used materials follows:

1. *Babbitt.* Babbitts have the best seizure, embeddability, and compliability characteristics of all bearing materials. Two types are in common use; tin base and lead base. Table 4.3 lists the composition of these materials. The tin-base material is slightly more desirable in terms of seizure and corrosion resistance; however, the lead base is less expensive. A lead-base babbitt with a small percentage of tin is used in many applications as an economic compromise.

Babbitts are inherently weak and also temperature limited. They should not be used at operating temperatures over 250°F. Babbitt bearings in gearboxes are usually bonded to a mild steel backing material and the babbitt thickness is on the order of 0.030 to 0.060 in. The strength of a babbitt surface decreases with its thickness and in very high load applications such as crankshafts, which

Table 4.3 Composition of Bearing Materials

		Nominal composition (%)						
Material	Specification	Copper	Tin	Lead	Zinc	Antimony	Nickel	Arsenic
Lead-base babbitt	SAE 14	½	10	74		15		½
Tin-base babbitt	SAE 12	3½	89			7½		
Copper-lead	SAE 480	65		35				
Leaded bronze	SAE 792	80	10	10				
Leaded bronze	SAE 794	72	3	23	2			
Aluminum		1	1	6			1	

are prone to fatigue, babbitt thickness may be as low as 0.001 in.; however, in gearboxes conformability and embeddability are important and thicker layers are used.

2. *Copper-Lead*. This is the simplest of the copper-based materials and contains from 20 to 40% lead. The material has excellent fatigue strength and is capable of carrying heavy loads at high temperatures. Compared to babbitt, however, copper lead has poor seizure resistance, embeddability, and compliability. The surface behavior can be improved by using hardened journals.

3. *Bronze*. There are three categories of bearing bronze; lead bronze, tin bronze, and high-strength bronze. High-strength bronze may contain aluminum, iron, manganese, silicon, nickel, or zinc in varying percentages. Strength improves from lead to tin to high-strength bronze, and shaft compatibility characteristics deteriorate. Bronzes are cost effective since they can be easily cast and machined and do not require a steel backing.

4. *Aluminum*. Aluminum has good fatigue resistance and corrosion resistance but does not have the surface properties of babbitt. Hardened shafts should be used in conjunction with aluminum bearings.

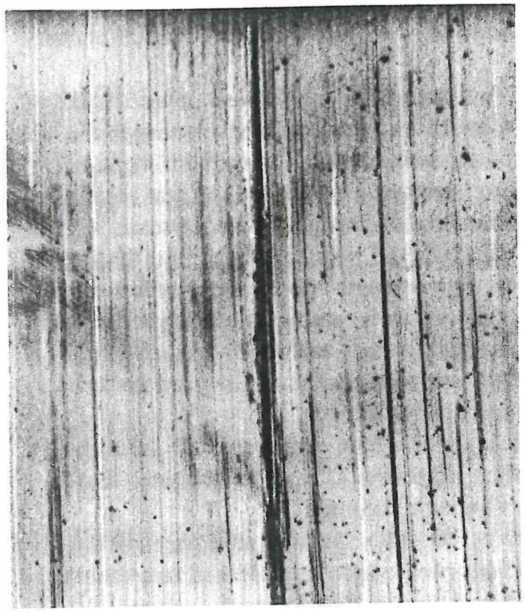
5. *Trimetal composite bearings*. In order to take advantage of the surface characteristics of babbitt yet improve bearing strength and temperature resistance, trimetal bearings have come into use. An intermediate layer of approximately 0.020 in. of copper-lead or leaded bronze is sandwiched between a mild steel backing and an overlay of babbitt approximately 0.002 in. thick.

Shaft Definition

Many journal and thrust bearing problems are a result of shaft discrepancies such as improper geometry or poor surface finish. The important geometric characteristics are taper, out of roundness, and grinding discrepancies such as waviness, chatter, or lobing, which must be closely controlled.

Shaft taper in the bearing area should be limited to 0.0002 in. per inch of length. Out of roundness should be limited to 0.0005 in. for shaft diameters up to 5 in. and 0.001 in. for shafts above 5 in. in diameter. Shaft surface finish should be a maximum of 20 rms with a preferred finish of approximately 10 rms.

Journals operating with babbitted surfaces can be made of soft steel with a minimum hardness of 200 bhn. Aluminum and copper-lead materials can be operated with shafts of 300 bhn minimum hardness. Where loads are high a journal hardness of Rc50 is recommended. In general, it is desirable to harden shaft surfaces in any case, in order to obtain longer life and improved wear and abrasion characteristics. In gearboxes the bearing journals quite often are integral with a pinion and therefore made of the gear material and hardened to the gear tooth specification and ground. In the case where gears are assembled on a shaft, the shaft might be SAE 4140 or 4340 steel and the journals nitrided and ground.



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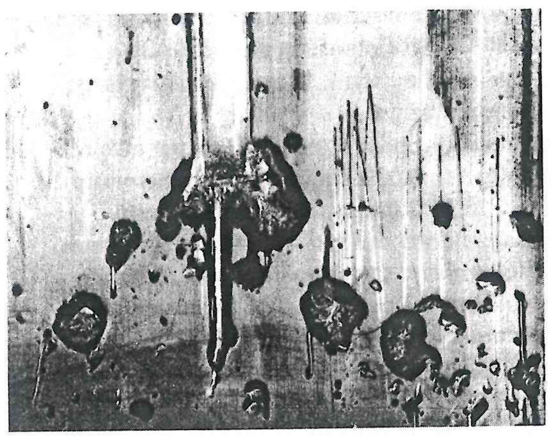

B

C

Figure 4.36 Effect of lubricant contamination. (Courtesy of Glacier Metal Company Ltd., Wembley, England.)

Journal Bearing Failures

Unlike rolling element bearings, fatigue failures are not common in gearbox journal bearings. Failures that occur are more likely to be associated with contamination of the lubricant, insufficient lubrication, dynamic excursion of the shaft, or faulty assembly.

Lubricant contamination. Abrasive materials in the lubricant may be a result of insufficient cleaning of the machined components at assembly, dirt entering the unit through breathers or bypassed oil filters, or wear particles generated inside the unit. Figure 4.36A shows a babbitted surface which has been scored and pitted by dirt, and Figure 4.36B shows surface distress to a greater extent caused by dirt particles. Figure 4.36C illustrates a thrust pad scored circumferentially by dirt particles.

The bearing in Figure 4.36A can be cleaned and reused provided that the wear experienced is not excessive and the bearing clearance remains within specification. Of course, the source of contamination must be eliminated and the lubrication system flushed out before continuing operation.

Lubricant cleanliness cannot be too greatly emphasized when operating journal bearings. From the assembly area, where all oil passages should be mechanically cleaned to eliminate machining chips, to the test stand, where the unit should be extensively flushed, and in the field, where oil must be carefully filtered, continuous care must be taken not to contaminate the oil. Filtration should be a maximum of 40 μm, and lower filtration levels to 10 μm are beneficial. When servicing the unit in the field care must be taken not to introduce dirt through filler ports or inspection covers.

Wiping of Bearing Surfaces. When the rotating journal and the bearing metal touch during operation the rubbing causes melting and smearing, as shown in Figure 4.37. Wiping may be due to insufficient clearance, overheating which closes down the clearance, high transient loads, shaft vibration due to unbalance, or dynamic journal instabilities. Figure 4.37A shows both halves of a wiped babbitted bearing and Figure 4.37B illustrates an overlay-plated copper-lead bearing wiped on half the circumference. When wiping occurs if the surface distress is light, the bearing can be cleaned and reused provided that the wear does not result in excessive clearance. The cause of the wipe should be identified and corrected. For instance, if a vibration survey shows synchronous vibration, the shaft should be balanced. If oil whip is detected, a different profile bore is indicated.

Corrosion. Corrosion of a bearing is a result of chemical attack by reactive materials in the oil stream. The most common problem is oxidation products formed in the oil, which corrode materials such as lead, copper, cadmium, and zinc. Figure 4.38 shows the severely corroded surface of a