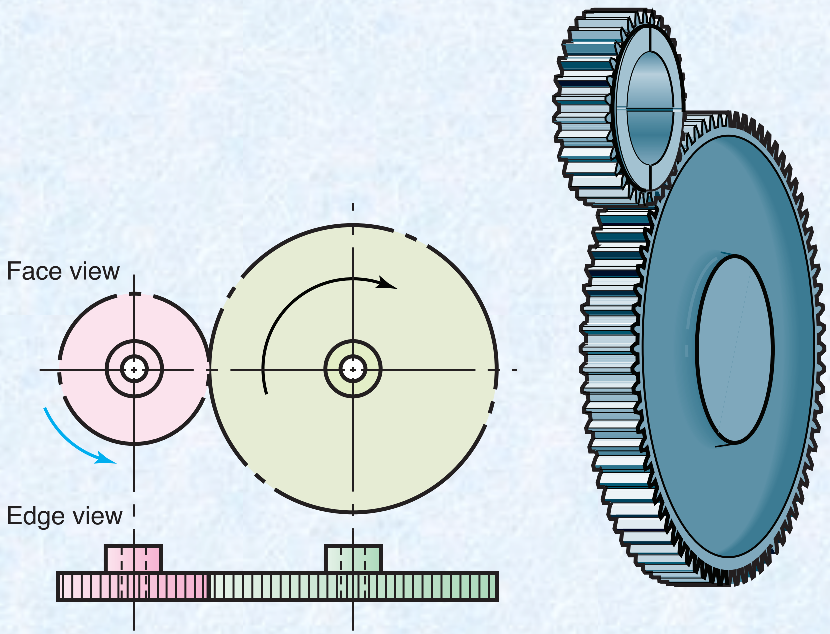
Chapter 14: General Gear Theory; Spur Gears

An assortment of gears. *Source:* Courtesy of Quality Transmission Components.

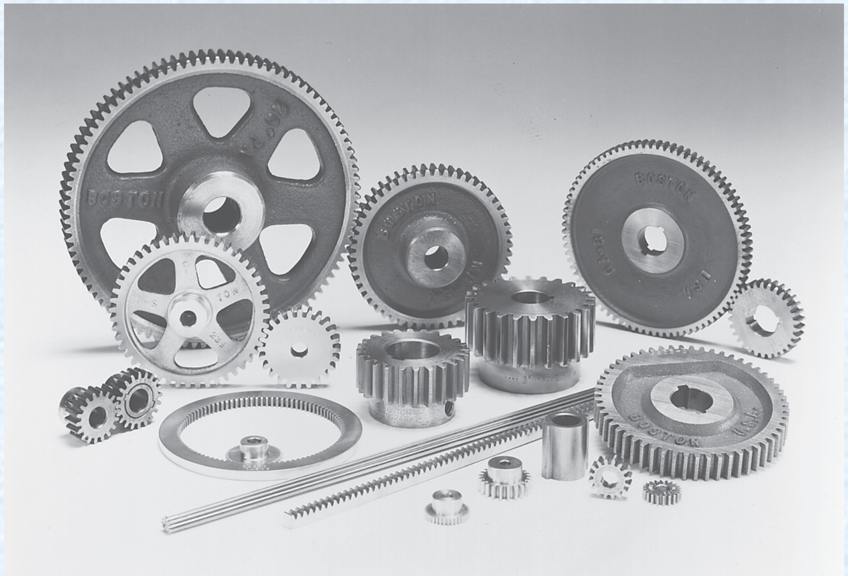
We are what we repeatedly do. Excellence, therefore, is not an act, but a habit.

Aristotle

Spur Gears



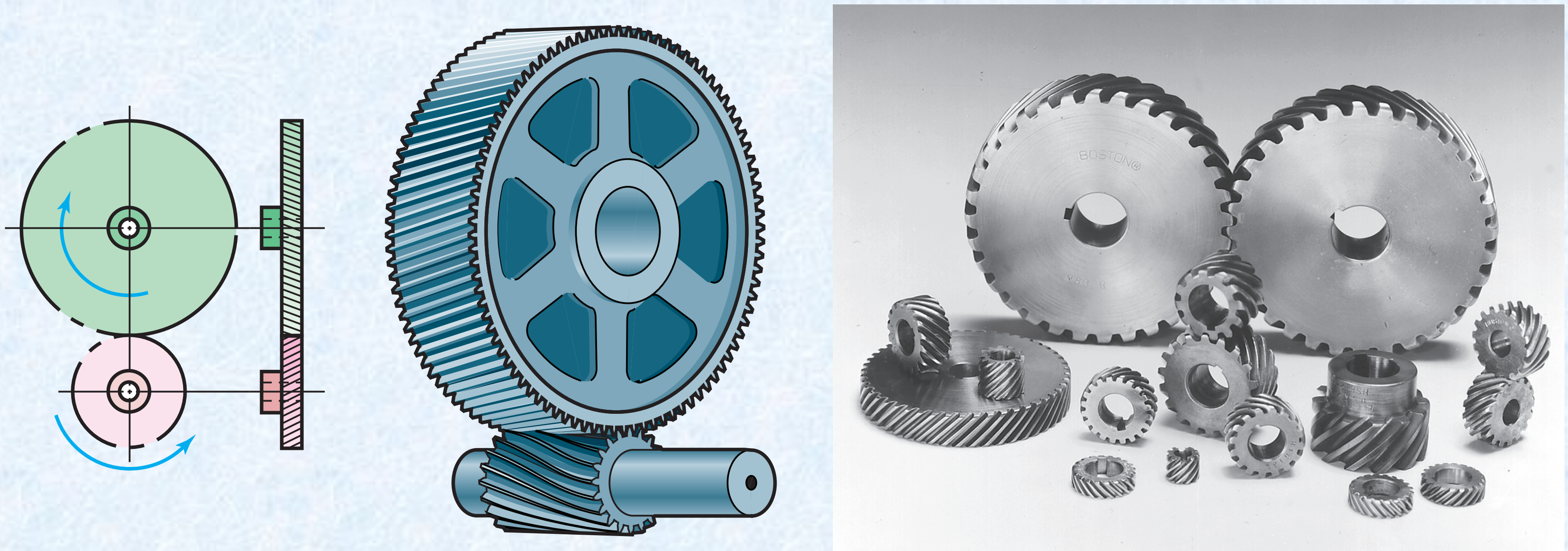
(a)



(b)

Figure 14.1: Spur gear drive. (a) Schematic illustration of meshing spur gears; (b) a collection of spur gears. *Source:* Courtesy of Boston Gear Works, Inc.

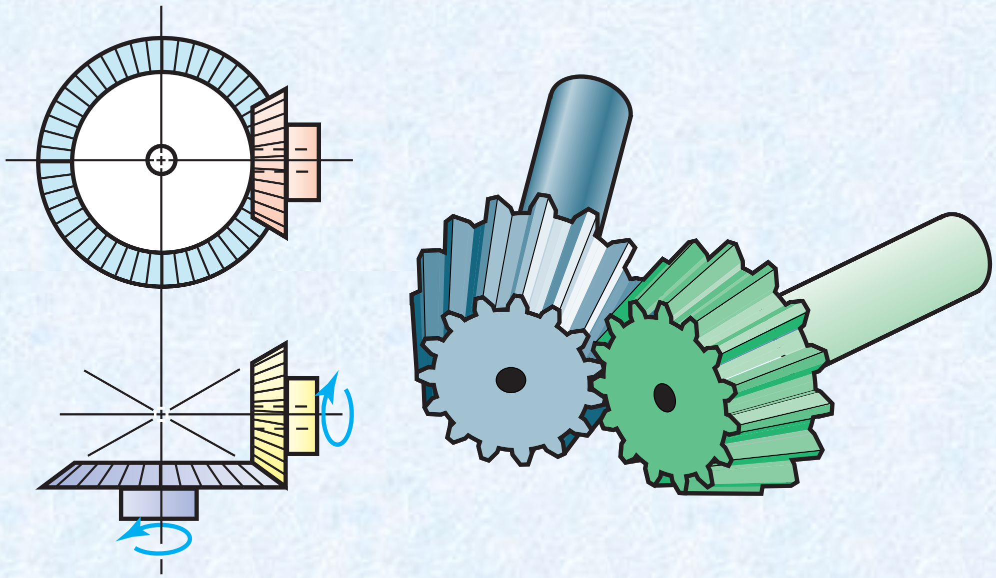
Helical Gears



(a) (b)

Figure 14.2: Helical gear drive. (a) Schematic illustration of meshing helical gears; (b) a collection of helical gears. *Source:* Courtesy of Boston Gear Works, Inc.

Bevel Gears

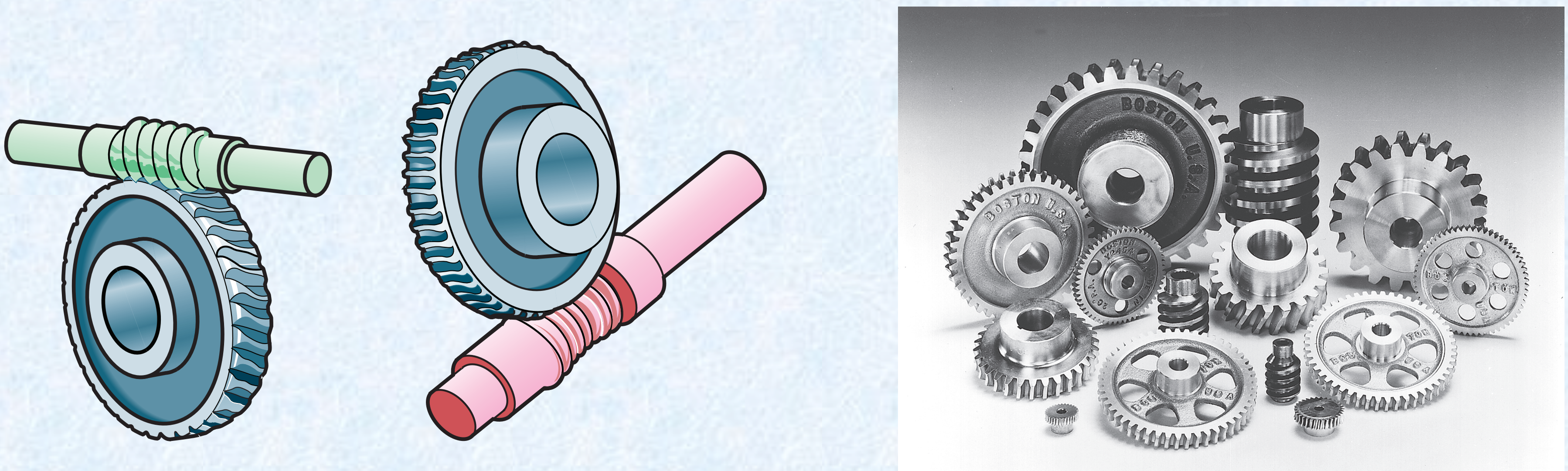


(a)

(b)

Figure 14.3: Bevel gear drive. (a) Schematic illustration of meshing bevel gears with straight teeth; (b) a collection of bevel gears. *Source:* Courtesy of Boston Gear Works, Inc.

Worm Gears



(a) (b) (c)

Figure 14.4: Worm gear drive. (a) Cylindrical teeth; (b) double enveloping; (c) a collection of worm gears. *Source:* Courtesy of Boston Gear Works, Inc.

Spur Gear Geometry

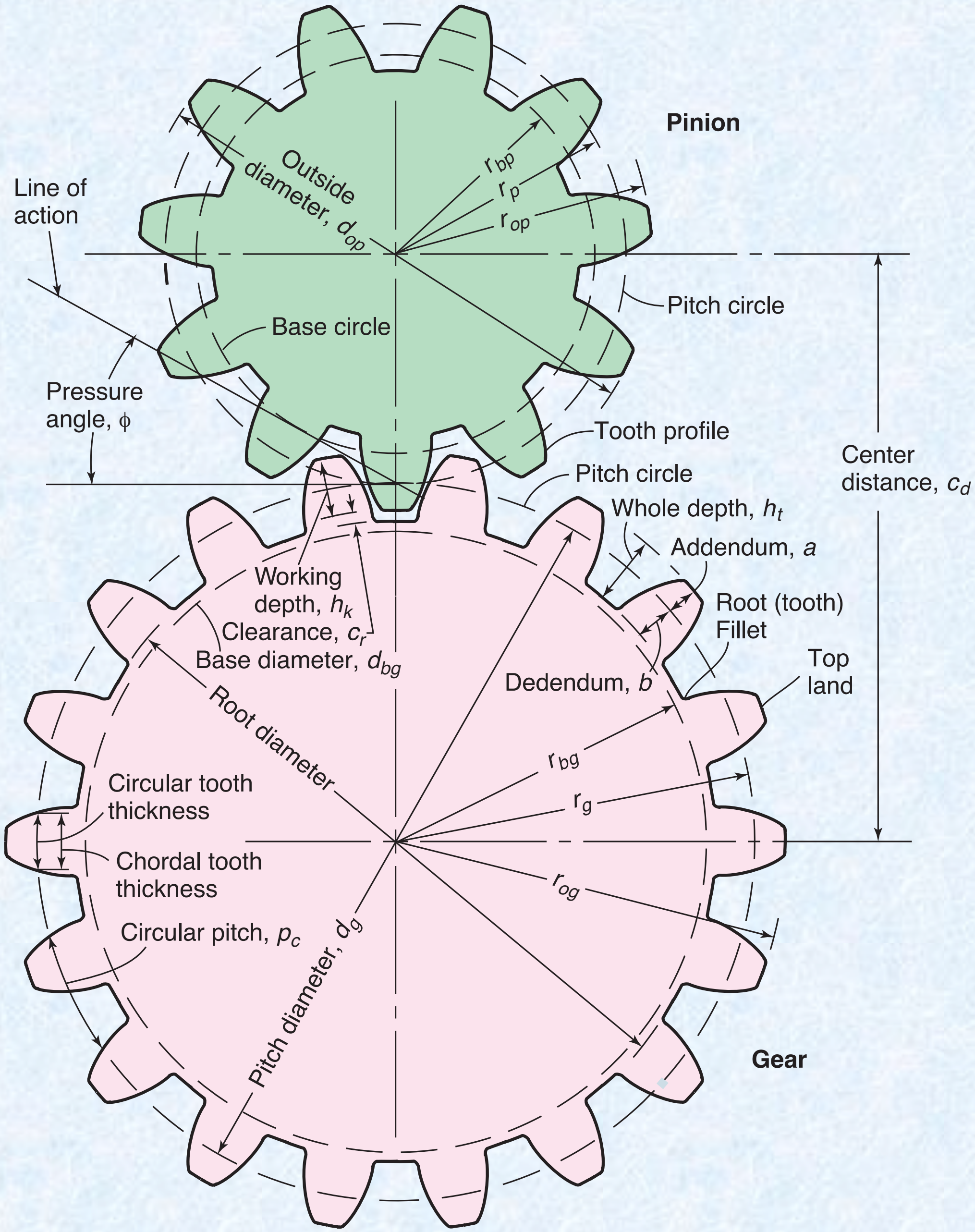


Figure 14.5: Basic spur gear geometry.

Fundamentals of Machine Elements, 3rd ed.
Schmid, Hamrock and Jacobson

Gear Tooth Nomenclature

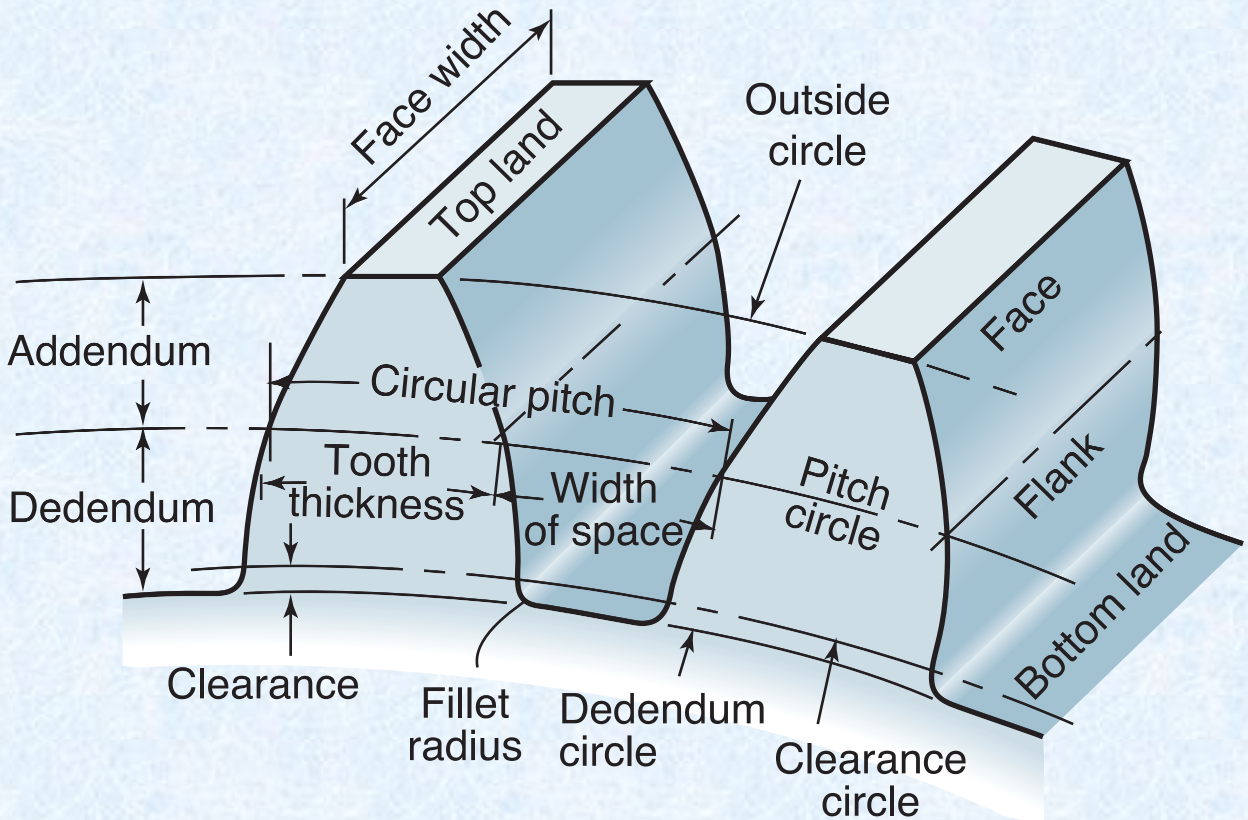


Figure 14.6: Nomenclature of gear teeth.

Tooth Size

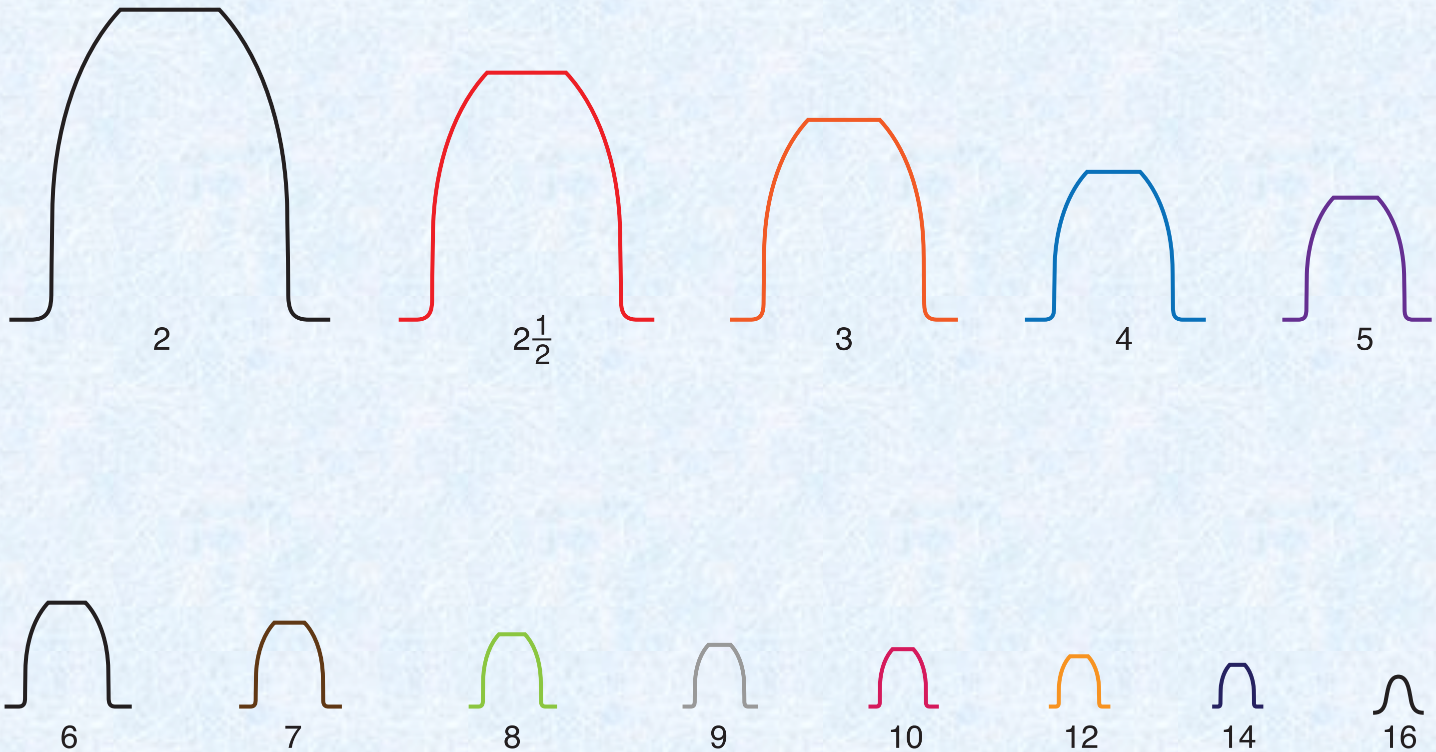


Figure 14.7: Standard diametral pitches compared with tooth size. Full size is assumed.

CRC
Fundamentals of Machine Elements, 3rd ed.
Schmid, Hamrock and Jacobson

Preferred Pitches

Class	Diametral pitch, p_d, in^{-1}
Coarse	1/2, 1, 2, 4, 6, 8, 10
Medium coarse	12, 14, 16, 18
Fine	20, 24, 32, 48, 64
	72, 80, 96, 120, 128
Ultrafine	150, 180, 200

Table 14.1: Preferred diametral pitches for four tooth classes.

Power Capacity

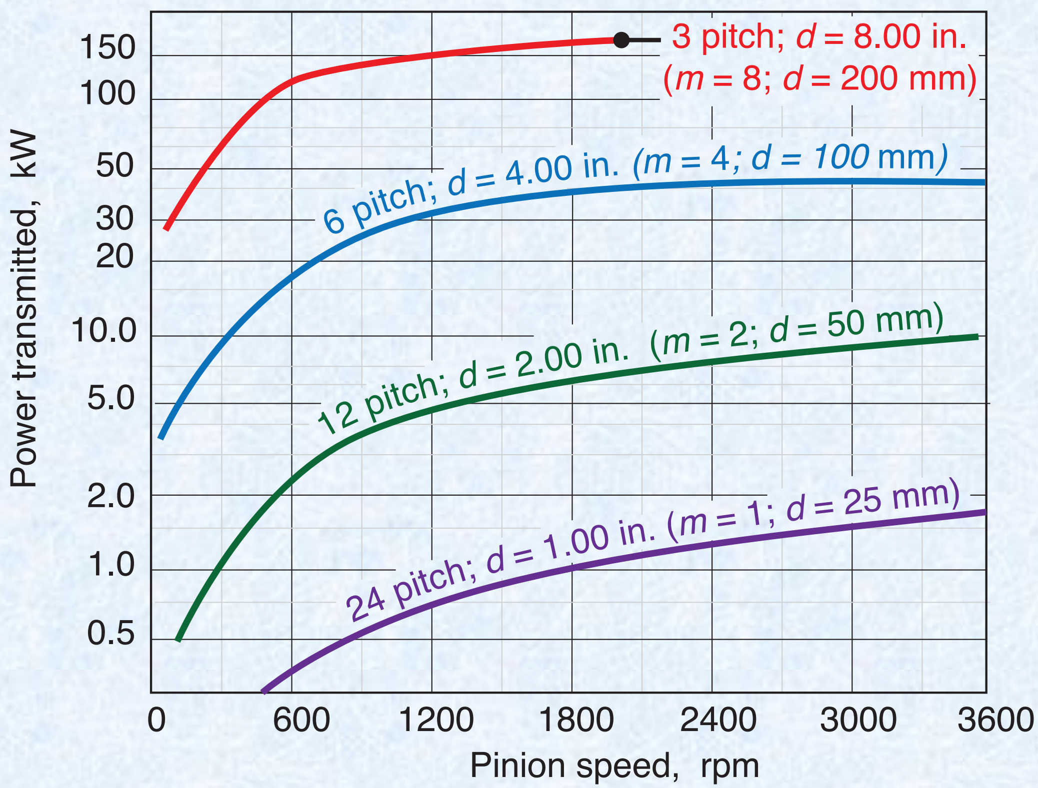


Figure 14.8: Transmitted power as a function of pinion speed for a number of diametral pitches. For all cases shown, g_r=4, N_p=24, K_o=1.0, φ=20°. *Source:* From Mott [2003].

Pitch and Base Circles; Line of Action

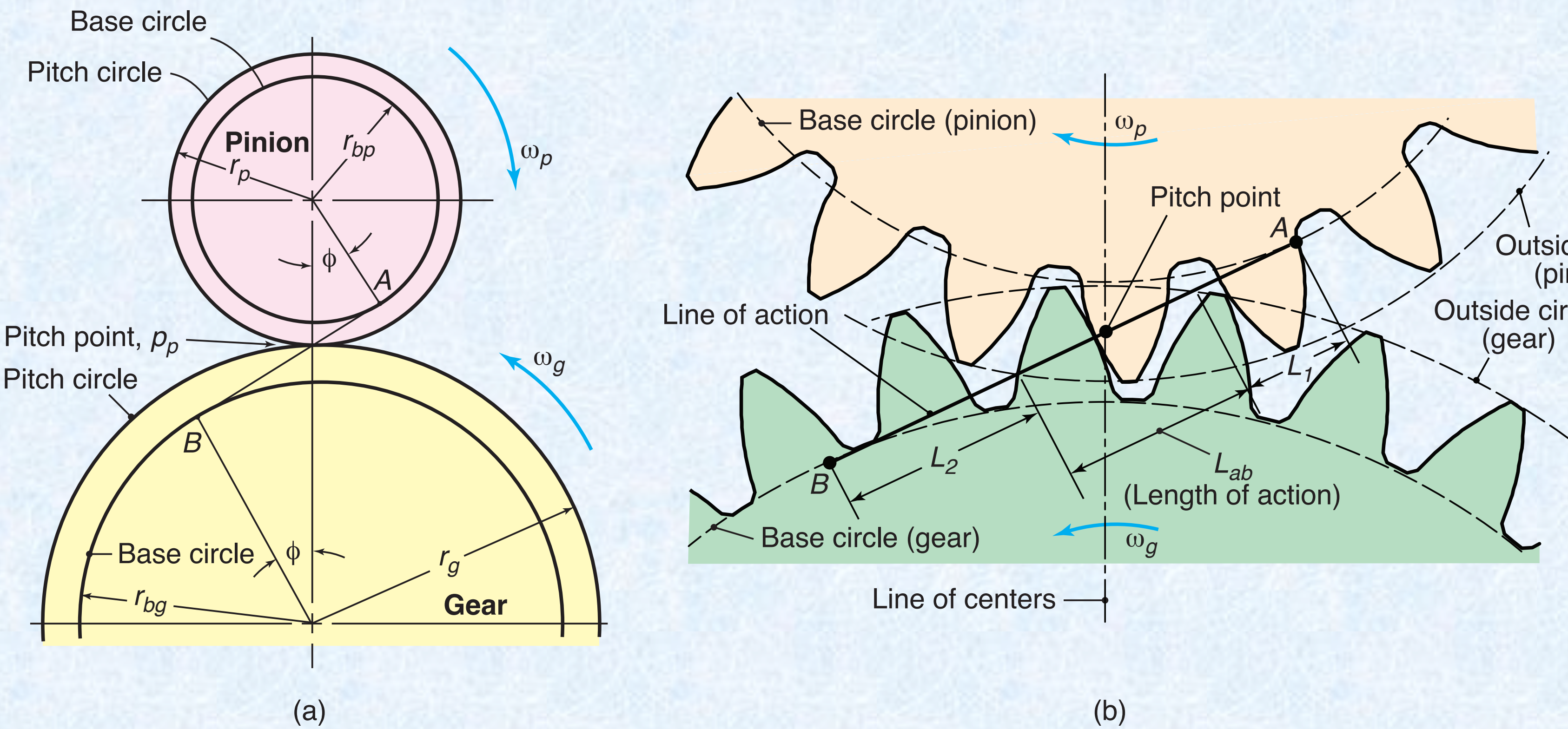


Figure 14.9: (a) Pitch and base circles for pinion and gear as well as line of action and pressure angle. (b) Detail of active profile, showing detail of line of action and length of action, L_{ab}.

Addendum, Dedendum & Clearance

Parameter	Symbol	Coarse pitch ($p_d <$ 20 in.$^{-1}$)	Fine pitch ($p_d \geq$ 20 in.$^{-1}$)	Metric module system
Addendum	a	$1/p_d$	$1/p_d$	1.00 m
Dedendum	b	$1.25/p_d$	$1.200/p_d + 0.002$	1.25 m
Clearance	c	$0.25/p_d$	$0.200/p_d + 0.002$	0.25 m

Table 14.2: Formulas for addendum, dedendum and clearance (φ=20°; full-depth involute).

Involute Curve

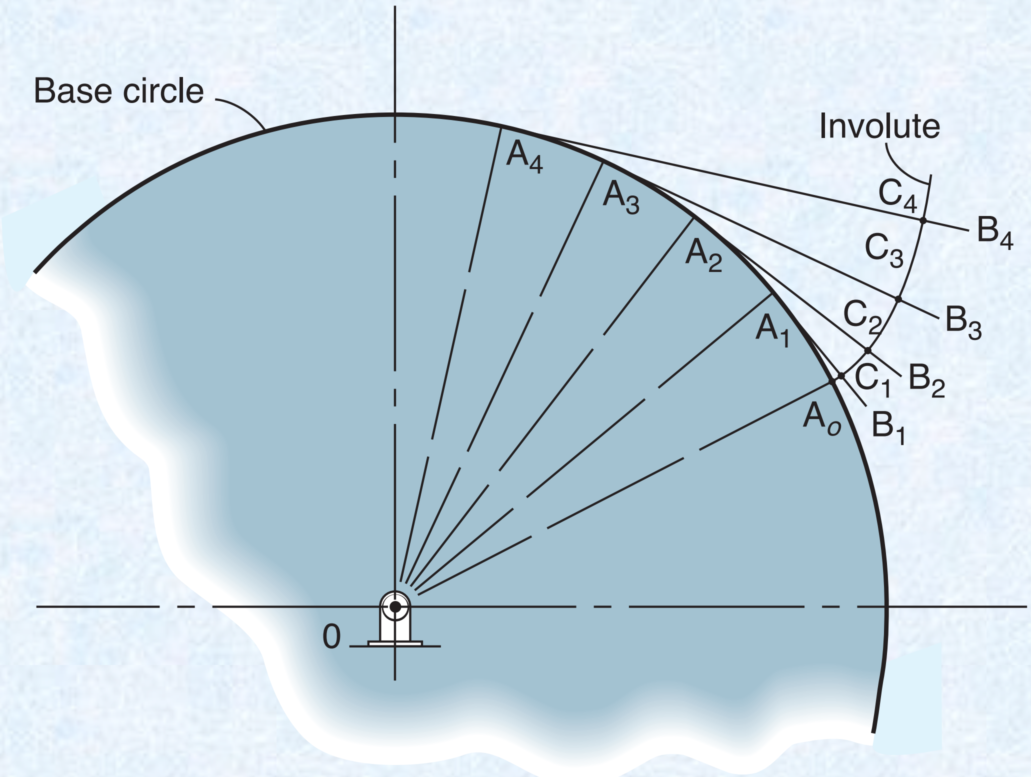


Figure 14.10: Construction of involute curve.

Arc of Approach

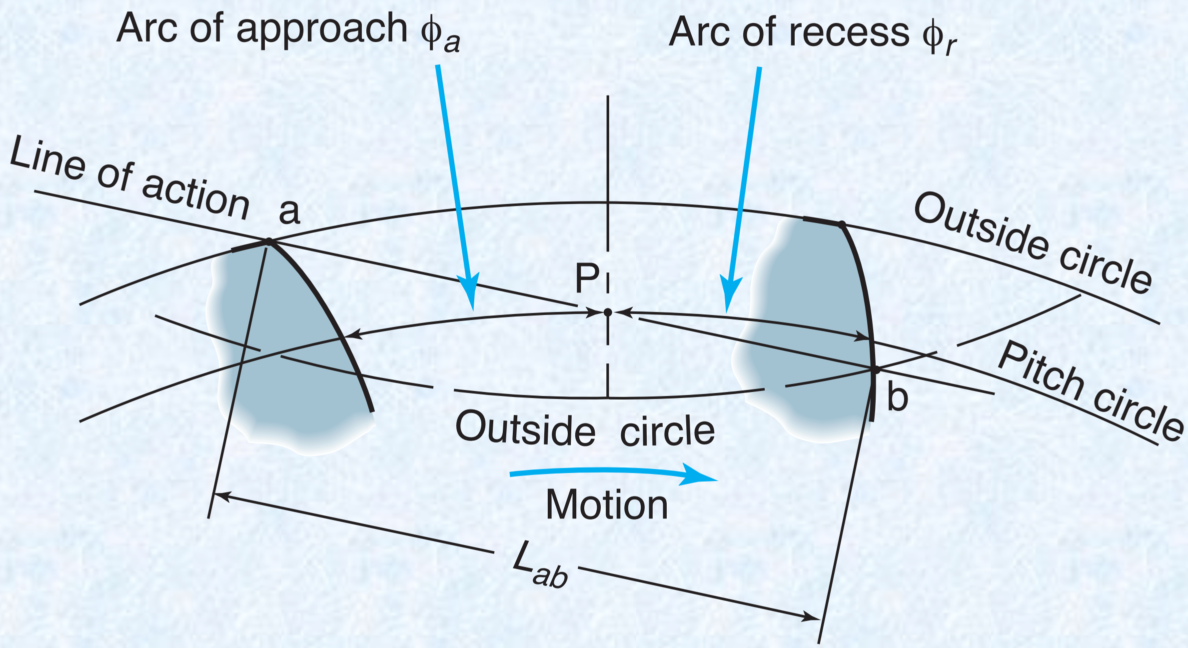


Figure 14.11: Illustration of arc of approach, arc of recess and length of action.

Backlash

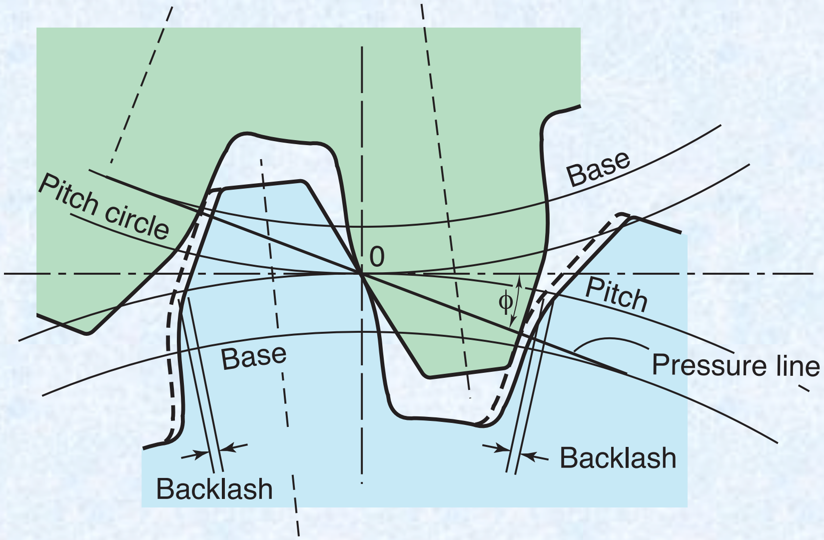


Figure 14.12: Illustration of backlash in gears.

Diametral pitch p_d, in.$^{-1}$	Center distance, c_d, in. 2	4	8	16	32
18	0.005	0.006	—	—	—
12	0.006	0.007	0.009	—	—
8	0.007	0.008	0.010	0.014	—
5	—	0.010	0.012	0.016	—
3	—	0.014	0.016	0.020	0.028
2	—	—	0.021	0.025	0.033
1.25	—	—	—	0.034	0.042

Table 14.3: Recommended minimum backlash for coarse-pitched gears.

Meshing Gears

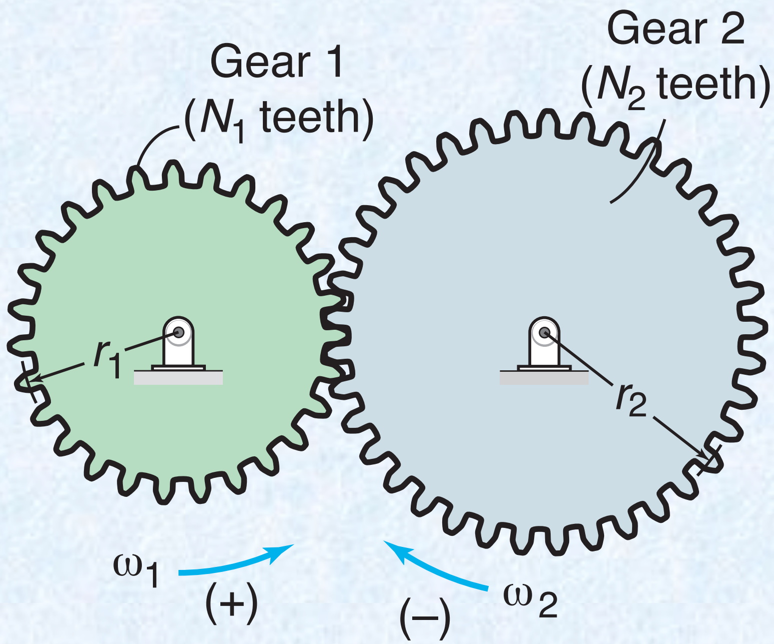


Figure 14.13: Externally meshing spur gears.

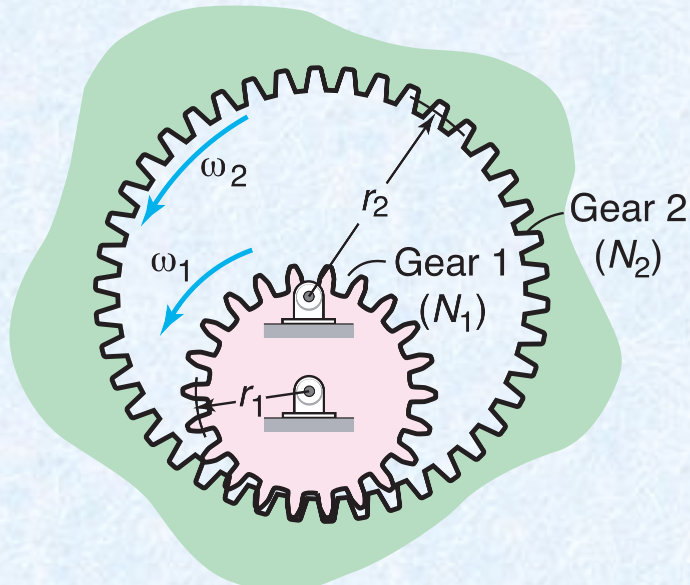


Figure 14.14: Internally meshing spur gears.

Gear Trains

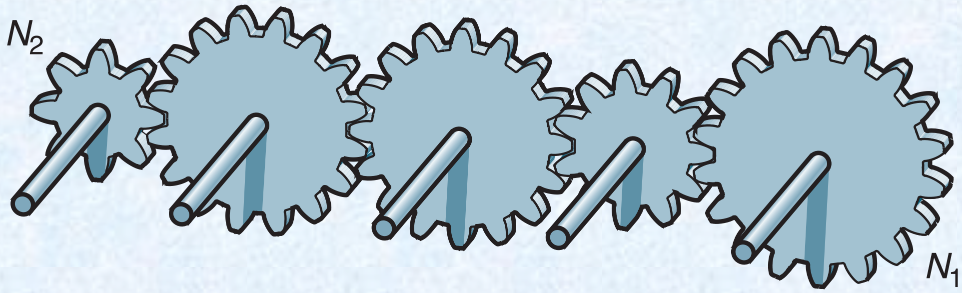


Figure 14.15: Simple gear train.

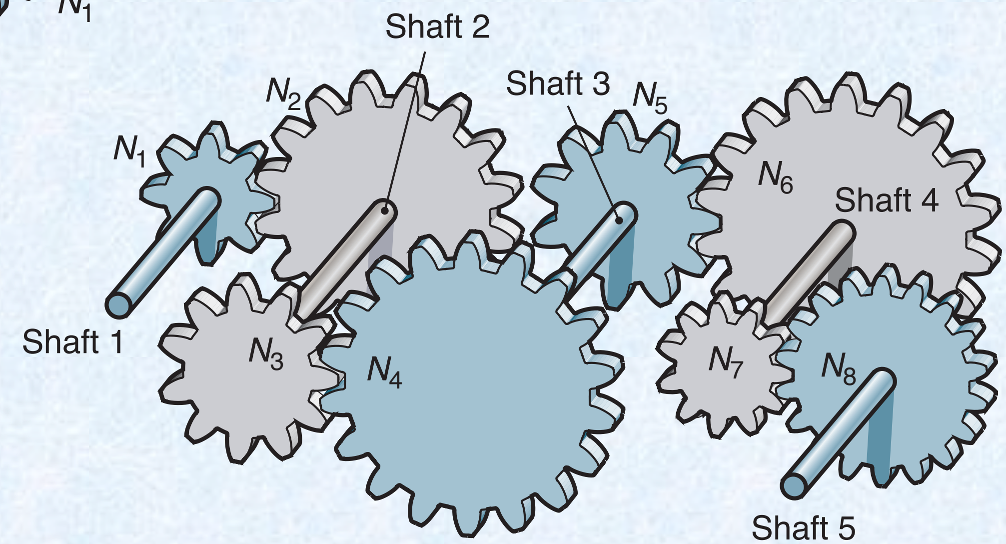


Figure 14.16: Compound gear train.

Example 14.6

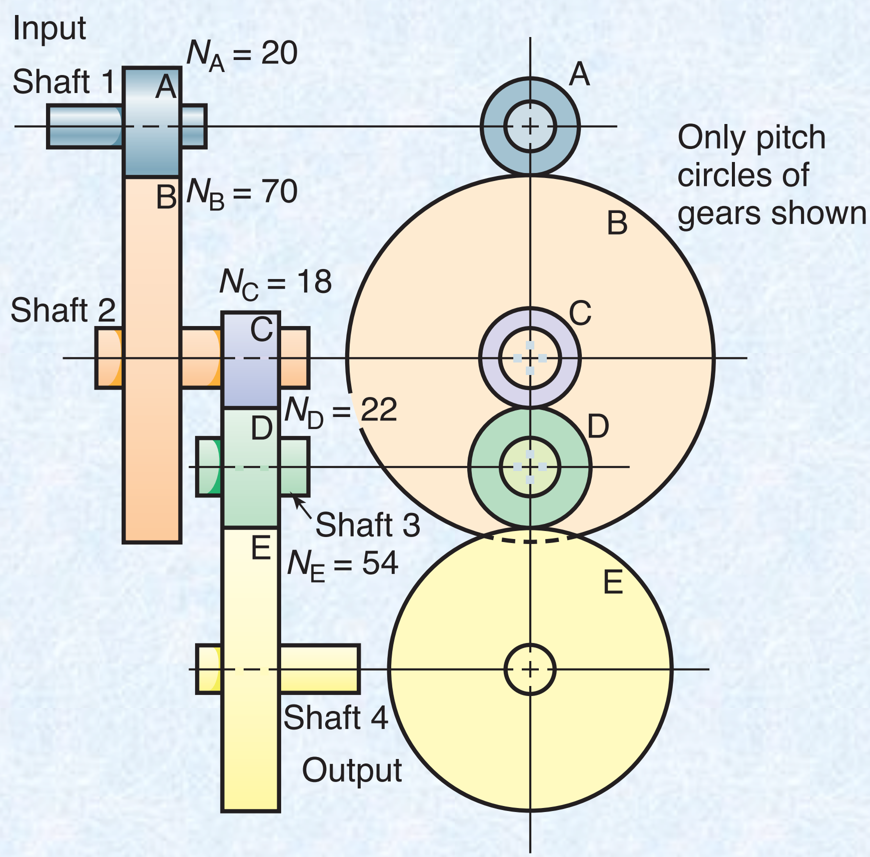


Figure 14.17: Gear train used in Example 14.6.

Planetary Gear Train

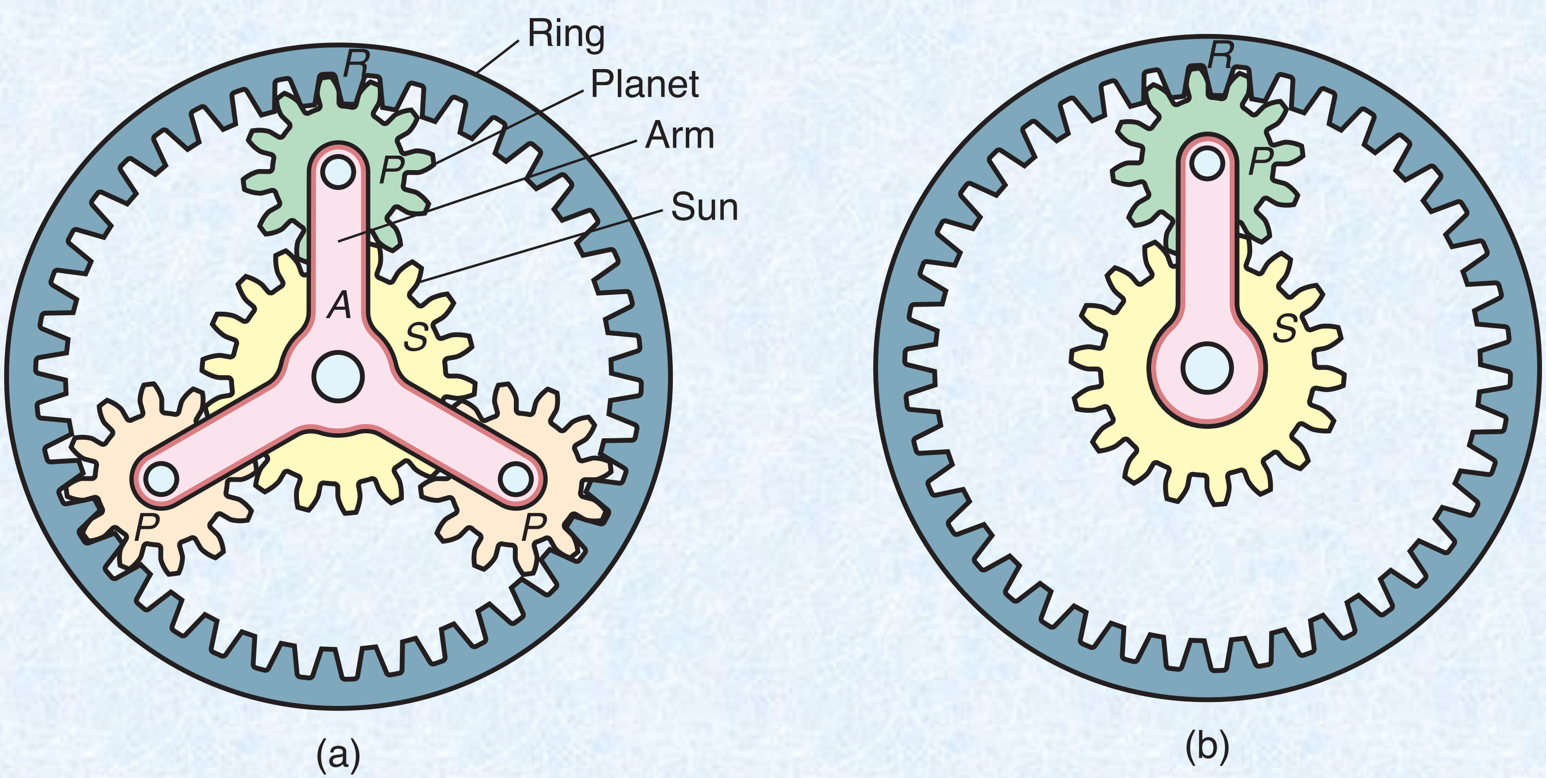


Figure 14.18: Illustration of planetary gear train. (a) With three planets; (b) with one planet (for analysis only).

Gear Cost vs. Quality

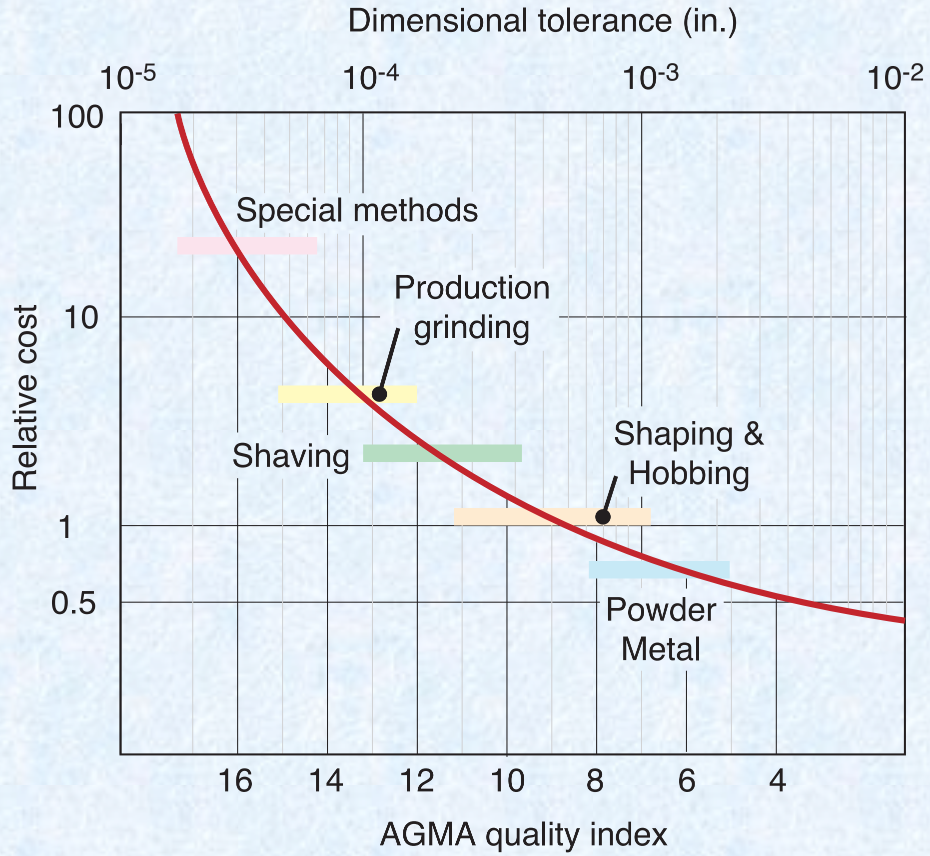


Figure 14.19: Gear cost as a function of gear quality. Note that the powder metallurgy approaches of pressing and sintering and metal injection molding can produce gears up to a quality index of 8 without additional machining. Recent research has suggested that similar quality levels can be achieved from cold forging as well.

Quality Index

Application	Quality index, Q_v
Cement mixer drum driver	3-5
Cement kiln	5-6
Steel mill drives	5-6
Corn pickers	5-7
Punch press	5-7
Mining conveyor	5-7
Clothes washing machine	8-10
Printing press	9-11
Automotive transmission	10-11
Marine propulsion drive	10-12
Aircraft engine drive	10-13
Gyroscope	12-14

Pitch velocity		Quality index, Q_v
ft/min	m/s	
0-800	0-4	6-8
800-2000	4-10	8-10
2000-4000	10-20	10-12
$>$ 4000	$>$ 20	12-14

Table 14.4: Quality index Q_v for various applications. *Source:* Courtesy of the American Gear Manufacturers Association.

Form Cutting of Gears

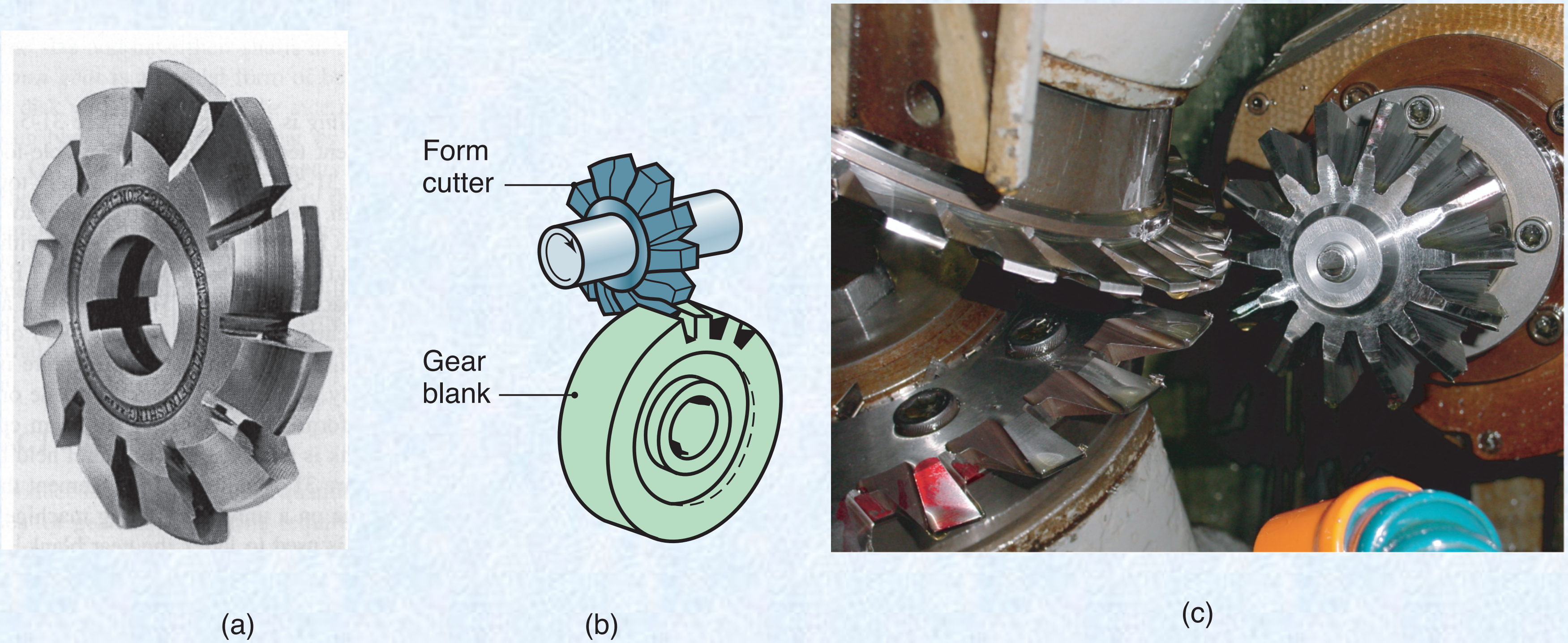
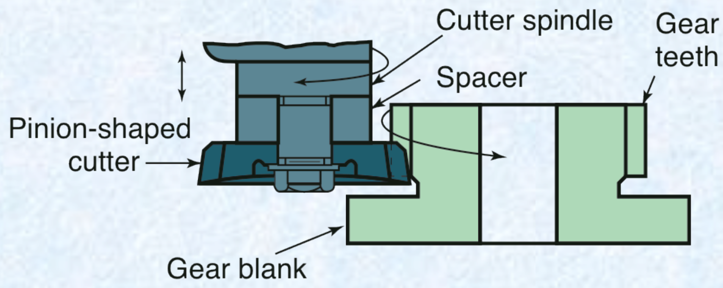
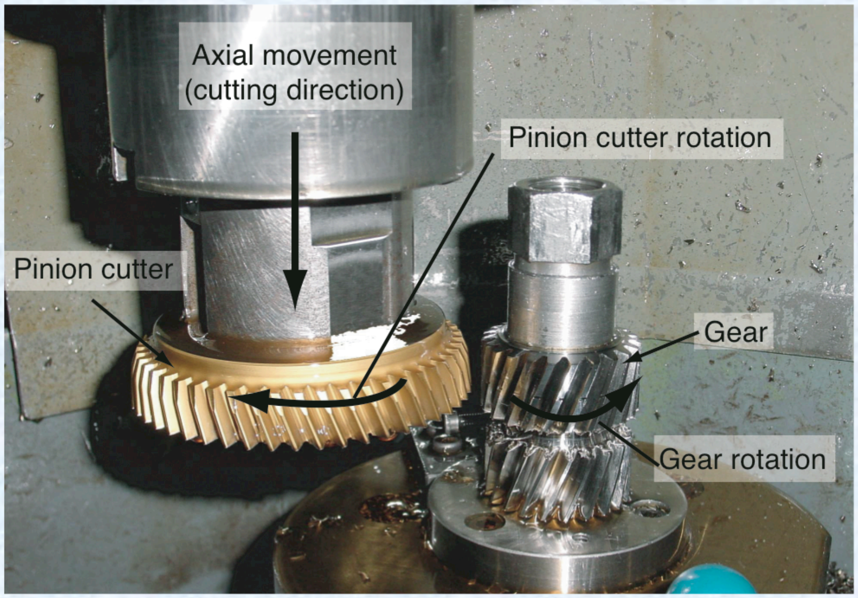


Figure 14.20: Form cutting of gear teeth. (a) A form cutter. Notice that the tooth profile is defined by the cutter profile; (b) schematic illustration of the form cutting process; (c) form cutting of teeth on a bevel gear. *Source:* (a) and (b) From Kalpakjian and Schmid [2010]; (c) Courtesy Schafer Gear Works, Inc.

Shaping



(a)



(b)

Figure 14.21: Production of gear teeth with a pinion-shaped cutter. (a) Schematic illustration of the process. *Source:* From Kalpakjian and Schmid [2010]; (b) photograph of the process with gear and cutter motions indicated. *Source:* Courtesy Schafer Gear Works, Inc.

Hobbing

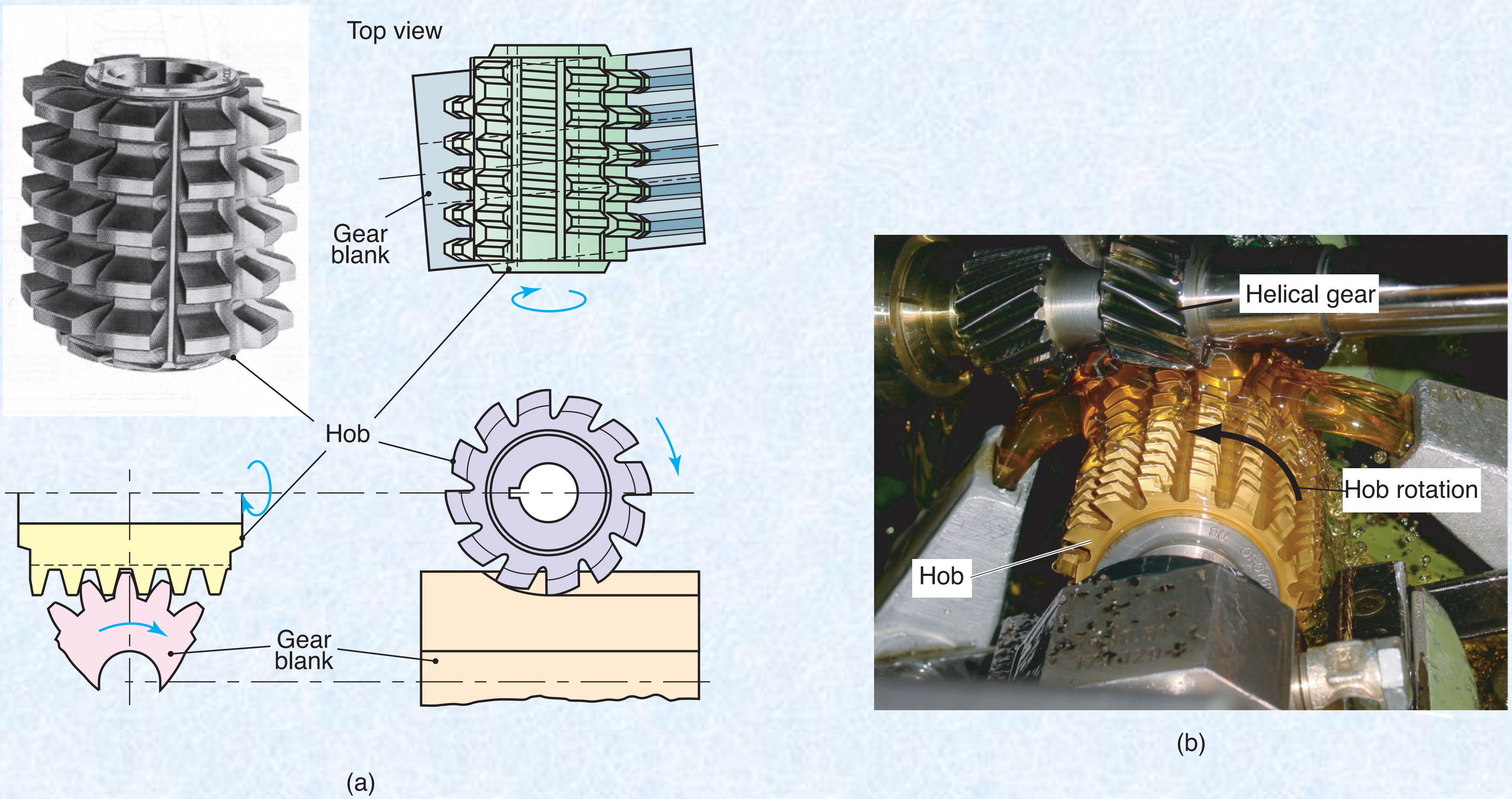


Figure 14.22: Production of gears through the hobbing process. (a) A hob, along with a schematic illustration of the process. *Source:* From Kalpakjian and Schmid [2010]; (b) production of a worm gear through hobbing. *Source:* Courtesy Schafer Gear Works, Inc.

Gear Finishing

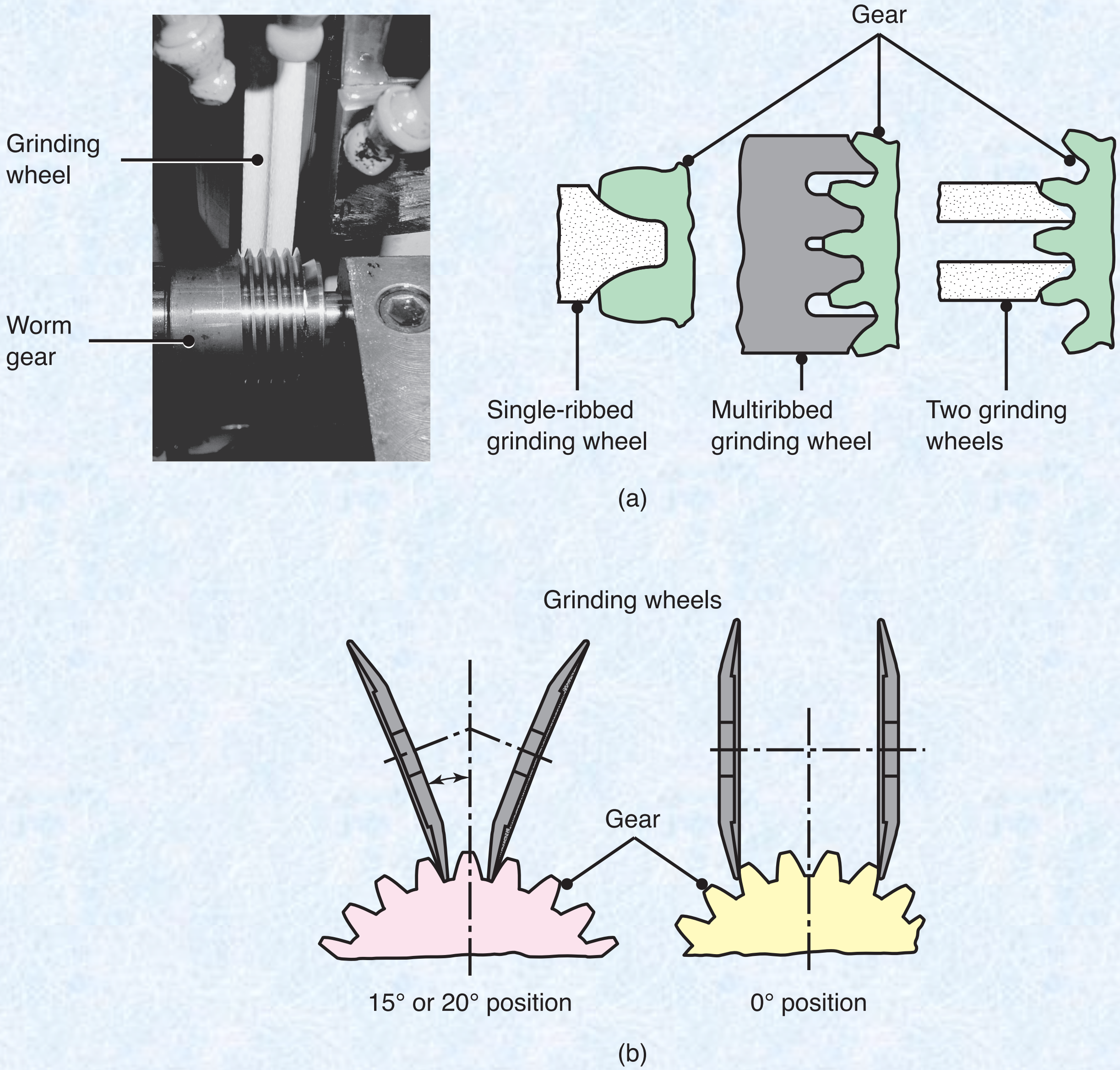


Figure 14.23: Finishing gears by grinding: (a) form grinding with shaped grinding wheels; (b) grinding by generating, using two wheels. *Source:* From Kalpakjian and Schmid [2010].

Hardness Effects on Bending Strength

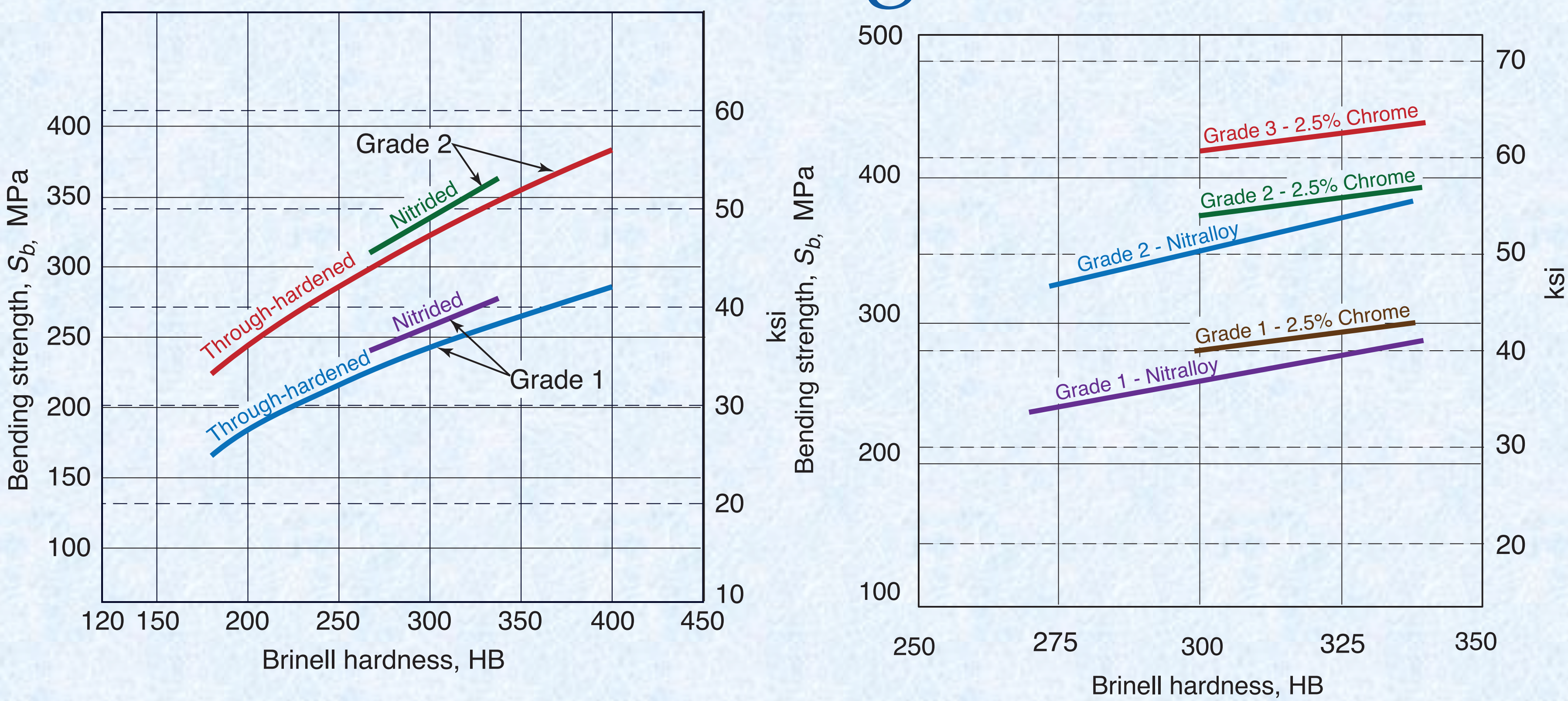


Figure 14.24: Effect of Brinell hardness on allowable bending stress for steel gears. (a) Through-hardened steels; (b) Flame or induction hardened nitriding steels. Note that Brinell hardness refers to case hardness for these gears. *Source:* ANSI/AGMA Standard 2101-D04 [2004].

Hardness Effects on Contact Strength

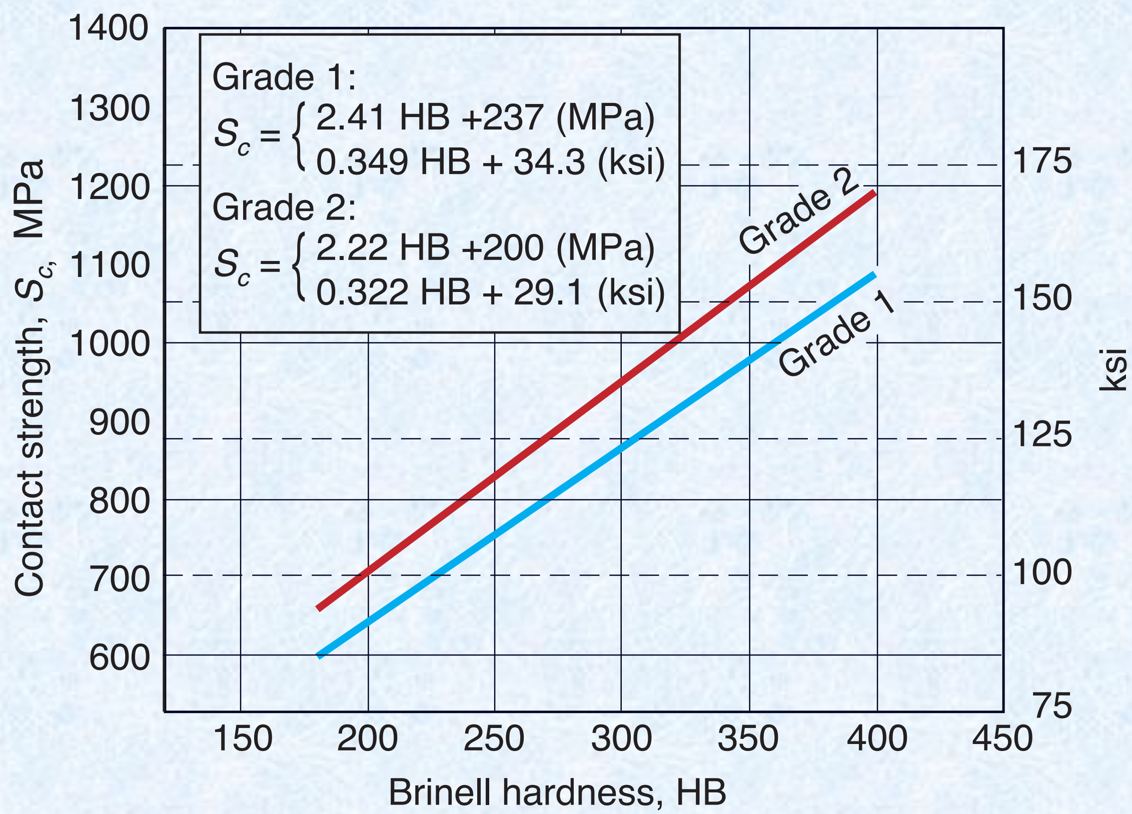


Figure 14.25: Effect of Brinell hardness on allowable contact stress number for two grades of through-hardened steel. *Source:* ANSI/AGMA Standard 2101-D04 [2004]

Strength of Gear Materials

Material designation	Grade	Typical Hardness[a]	Bending strength, S_b lb/in.^{2}	Bending strength, S_b MPa	Contact strength, S_c ksi	Contact strength, S_c MPa
Steel						
Through-hardened[b]	1	180-400 HB	0.0773 HB + 12.8	0.533 HB + 88.3	0.349 HB + 34.3	2.41 HB + 237
	2	180-400 HB	0.102 HB + 16.4	0.703 HB + 11.3	0.322 HB + 29.1	2.22 HB + 200
Carburized & hardened	1	55-64 HRC	55.0	380	180.0	1240
	2	58-64 HRC	65.0[c]	450[c]	225.0	1550
	3	58-64 HRC	75.0	515	275.0	1895
Nitrided and through-hardened[b]	1	83.5 HR15N	0.0823 HB +12.15	0.568 HB + 83.8	150,000	1035
	2	—	0.1086 HB + 15.89	0.749 HB + 110	163,000	1125
Nitralloy 135M and Nitralloy N, nitrided[b]	1	87.5 HR15N	0.0862 HB + 12.73	0.594 HB + 87.76	170,000	1170
	2	87.5 HR15N	0.1138 HB + 16.65	0.784 HB + 114.81	183,000	1260
2.5% Chrome, nitrided[b]	1	87.5 HR15N	0.1052 HB + 9.28	0.7255 HB + 63.89	155,000	1070
	2	87.5 HR15N	0.1052 HB + 22.28	0.7255 HB + 153.63	172,000	1185
	3	87.5 HR15N	0.1052 HB + 29.28	0.7255 HB + 201.81	189,000	1305
Cast Iron						
ASTM A48 gray cast iron, as-cast	Class 20	—	5.00	34.5	50.0-60.0	345-415
	Class 30	174 HB	8.50	59	65.0-75.0	450-520
	Class 40	201 HB	13.0	90	75.0-85.0	520-585
ASTM A536 ductile (nodular) iron	60-40-18	140 HB	22.0-33.	150-230	77.0-92.0	530-635
	80-55-06	179 HB	22.0-33.0	150-230	77.0-92.0	530-635
	100-70-03	229 HB	27.0-40.0	185-275	92.0-112.0	635-770
	120-90-02	269 HB	31.0-44.0	215-305	103.0-126.0	710-870
Bronze						
$S_{ut} > 40,000$ psi ($S_{ut} > 275$ GPa)			5.70	39.5	30.0	205
$S_{ut} > 90,000$ psi ($S_{ut} > 620$ GPa)			23.6	165	65.0	450
Powder Metal						
FL-4405, $\rho = 7.30$ g/cm^3		80 HRB	49.0	340	282.0	1945
FLN2-4405, $\rho = 7.35$ g/cm^3		90 HRB	60.0	410	180.0	1240
FLC-4608, $\rho = 7.30$ g/cm^3		65 HRB	95.72	660	210.0	1450
FN-0205, $\rho = 7.10$ g/cm^3		69 HRB	30.0	210	180.0	1240

[a] Hardness refers to case hardness unless through-hardened.
[b] See Figs. 14.24 and/or 14.25.
[c] 70,000 psi (485 MPa) may be used if bainite and microcracks are limited to grade 3 levels.

Table 14.5: Bending and contact strength for selected gear materials. *Source:* Adapted from ANSI/AGMA 2101-D04 [2004] and MPIF Standard 35 [2009].

Gear Strength Modification Factors and Reliability Factor

Allowable bending stress:

$$\sigma_{b,\mathrm{all}} = \frac{S_b}{n_s}\frac{Y_N}{K_t K_r}$$

Allowable contact stress:

$$\sigma_{c,\mathrm{all}} = \frac{S_c}{n_s}\frac{Z_N C_H}{K_t K_r}$$

Probability of survival, percent	Reliability factor[a] K_r
50	0.70[b]
90	0.85[b]
99	1.00
99.9	1.25
99.99	1.50

[a] Based on surface pitting. If tooth breakage is considered a greater hazard, a larger value may be required.
[b] At this value, plastic flow may occur rather than pitting.

Table 14.6: Reliability factor, K_r. *Source:* From ANSI/AGMA 2101-D04 [2004].

Bending Stress Cycle Factor

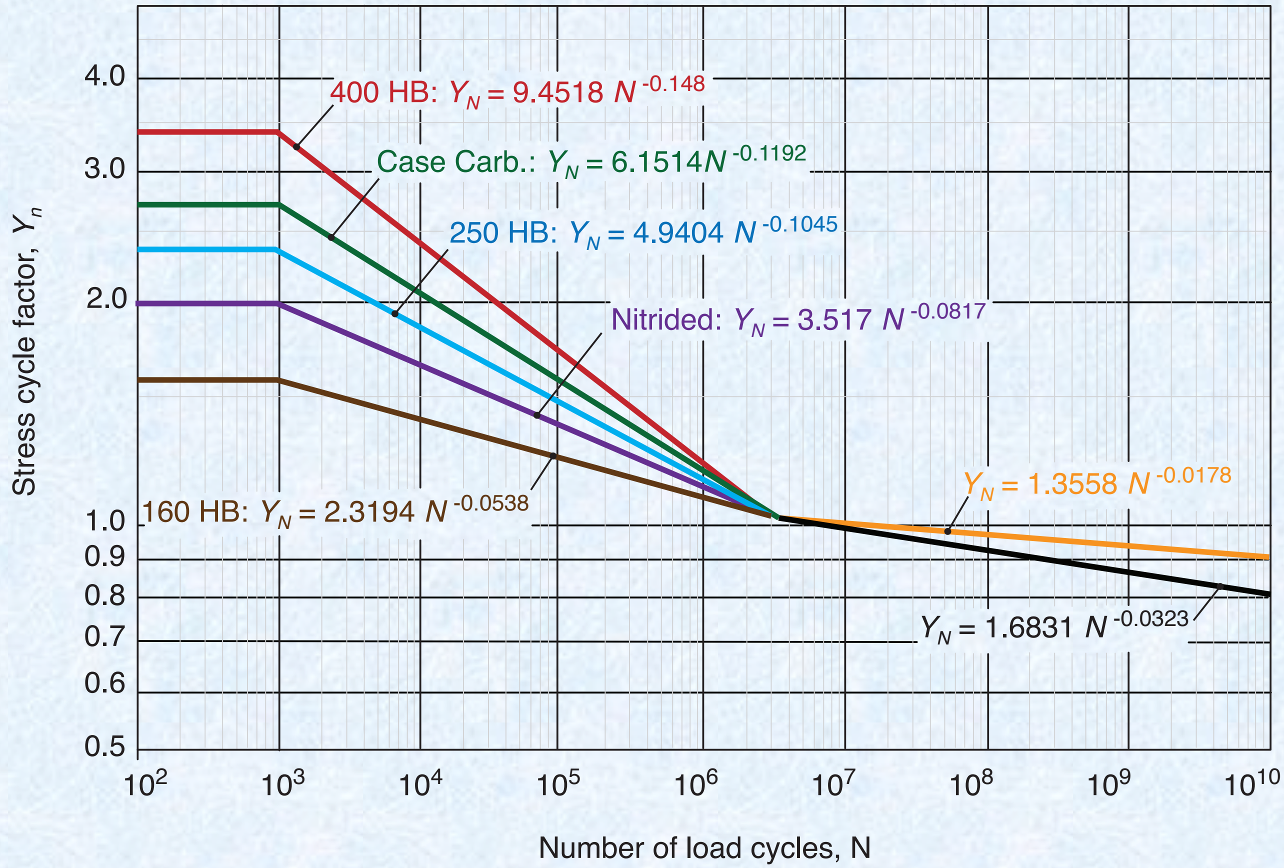


Figure 14.26: Stress cycle factor. (a) Bending strength stress cycle factor Y_N; *Source:* ANSI/AGMA Standard 2101-D04 [2004].

Pitting Resistance Cycle Factor

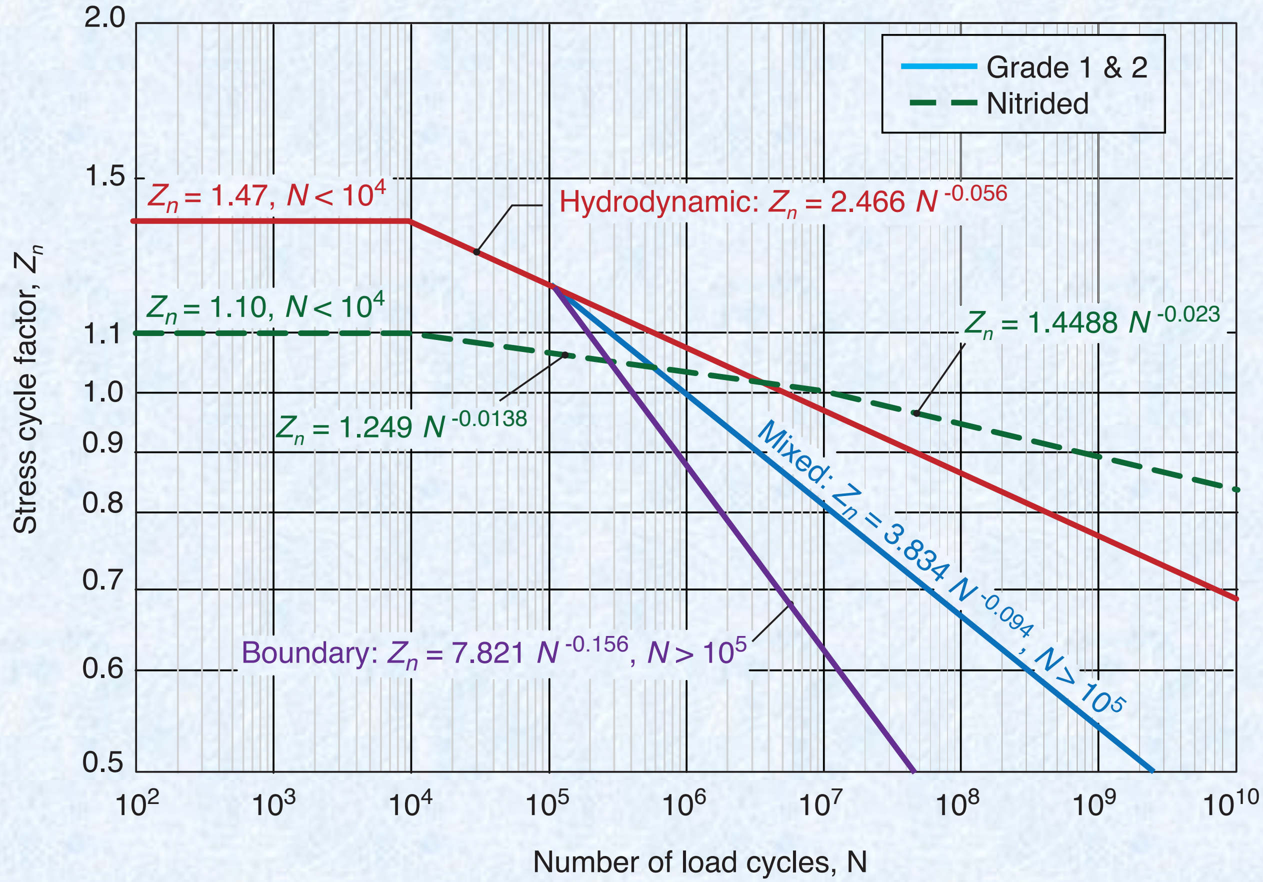


Figure 14.26: Stress cycle factor. (b) Pitting resistance stress cycle factor Z_N. *Source:* ANSI/AGMA Standard 2101-D04 [2004].

Loads on Gear Tooth

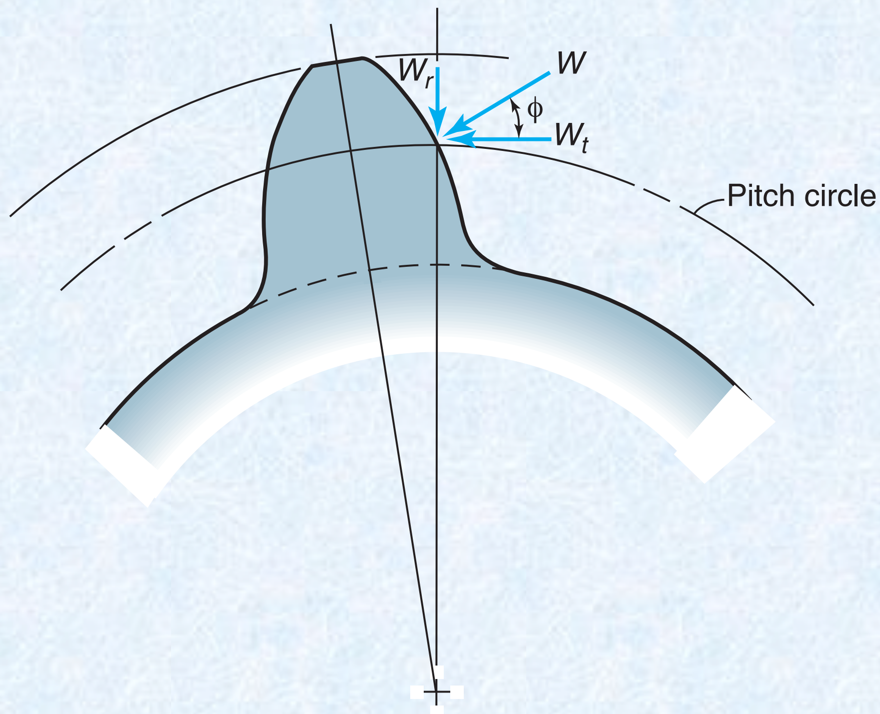


Figure 14.27: Loads acting on an individual gear tooth.

Tangential load:

$$W_t = \frac{h_p}{u} = \frac{60h_p}{\pi d N_a}$$

If h_p is in horsepower:

$$W_t = \frac{126,050h_p}{dN_a}$$

Normal load:

$$W = \frac{W_t}{\cos\phi}$$

Radial load:

$$W_r = W_t \tan\phi$$

Bending of Gear Teeth

Figure 14.28: A crack that has developed at the root of a gear tooth due to excessive bending stresses. *Source:* Courtesy of the American Gear Manufacturers Association.

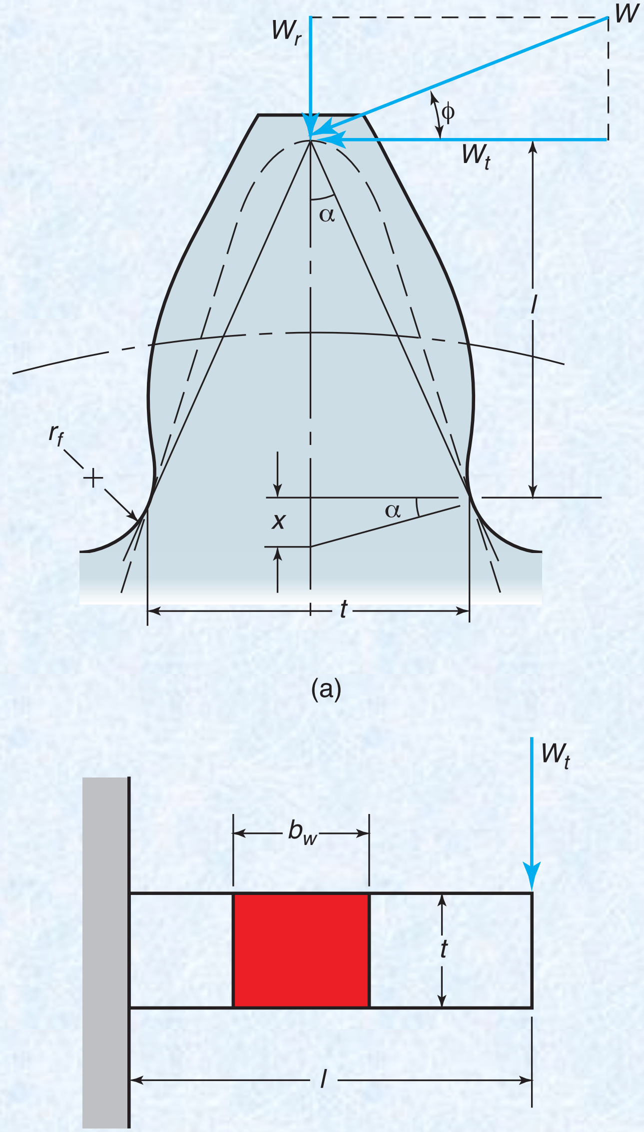


Figure 14.29: Loads and length dimensions used in determining tooth bending stress. (a) Tooth; (b) cantilevered beam.

Lewis Form Factor

Number of teeth	Lewis form factor	Number of teeth	Lewis form factor
10	0.176	34	0.325
11	0.192	36	0.329
12	0.210	38	0.332
13	0.223	40	0.336
14	0.236	45	0.340
15	0.245	50	0.346
16	0.256	55	0.352
17	0.264	60	0.355
18	0.270	65	0.358
19	0.277	70	0.360
20	0.283	75	0.361
22	0.292	80	0.363
24	0.302	90	0.366
26	0.308	100	0.368
28	0.314	150	0.375
30	0.318	200	0.378
32	0.322	300	0.382

Table 14.7: Lewis form factor for various numbers of teeth (pressure angle, 14.5°; full-depth involute).

Spur Gear Geometry Factors

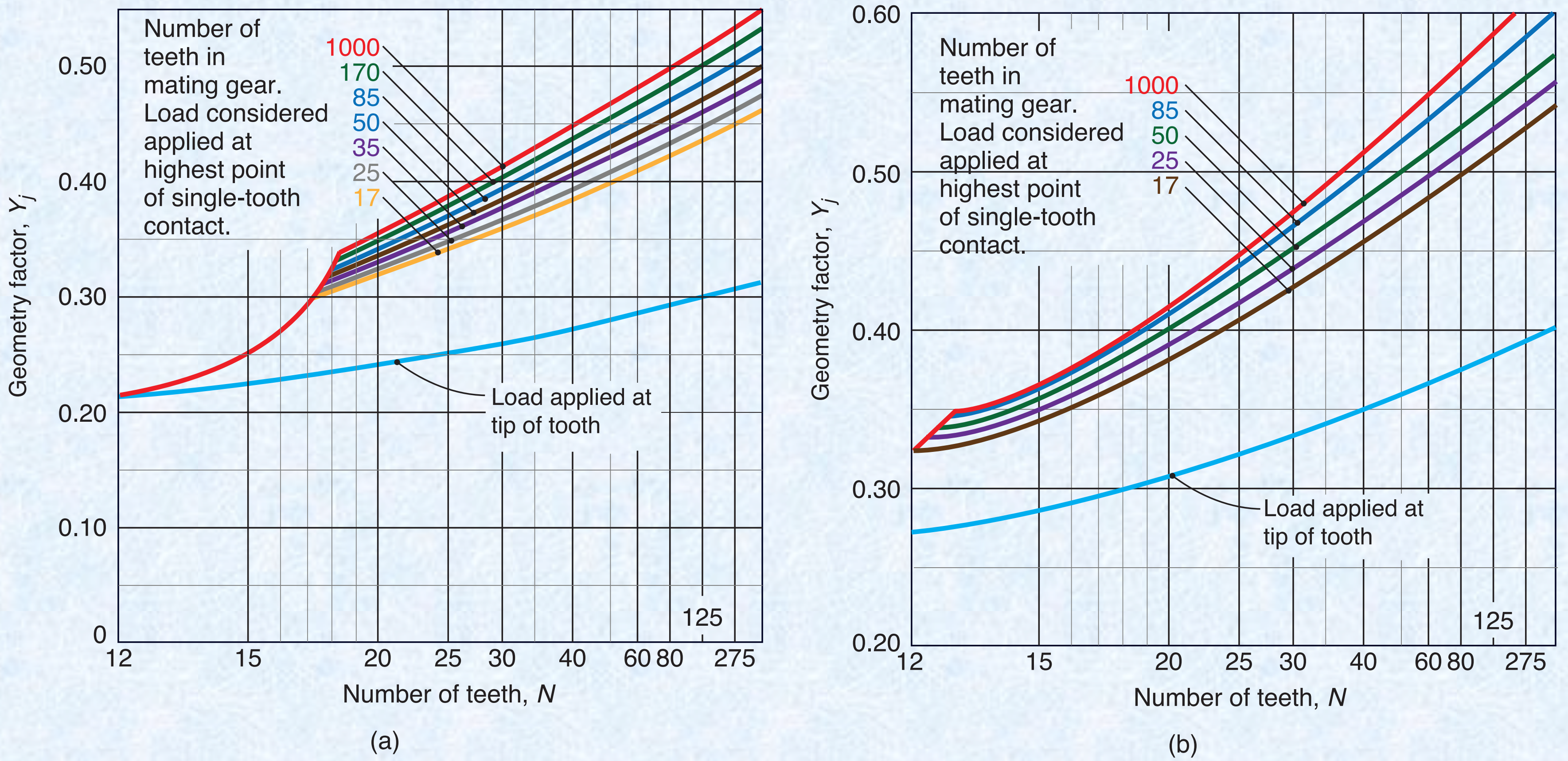


Figure 14.30: Spur gear geometry factors for full-depth involute profile. (a) φ=20°; (b) φ=25°. *Source:* AGMA Standard 908-B89 [1989].

AGMA Bending Stress Equation

$$\sigma_b = \frac{W_t p_d K_o K_s K_m K_v K_b}{b_w Y_j}$$

where

W_t = transmitted load, N
p_d = diametral pitch, m^{-1}
K_o = overload factor
K_s = size factor
K_m = load distribution factor
K_v = dynamic factor
K_b = rim thickness factor

Overload and Size Factors

Power source	Driven Machines: Uniform	Light shock	Moderate shock	Heavy shock
Uniform	1.00	1.25	1.50	1.75
Light shock	1.20	1.40	1.75	2.25
Moderate shock	1.30	1.70	2.00	2.75

Figure 14.8: Overload factor, K_o, as function of driving power source and driven machine.

Table 14.9: Recommended values of size factor, K_s.

Diametral pitch, p_d, in.$^{-1}$	Module, m, mm	Size factor, K_s
≥ 5	≤ 5	1.00
4	6	1.05
3	8	1.15
3	12	1.25
1.25	20	1.40

Load Distribution Factor

Load distribution factor:

$$K_m = 1.0 + C_{mc}\left(C_{pf}C_{pm} + C_{ma}C_e\right)$$

where

C_{mc} = lead correction factor
C_{pf} = pinion proportion factor
C_{pm} = pinion proportion modifier
C_{ma} = mesh alignment factor
C_e = mesh alignment correction factor

Lead correction factor:

$$C_{mc} = \begin{cases} 1.0 & \text{for uncrowned teeth} \\ 0.8 & \text{for crowned teeth} \end{cases}$$

Pinion Proportion Factor

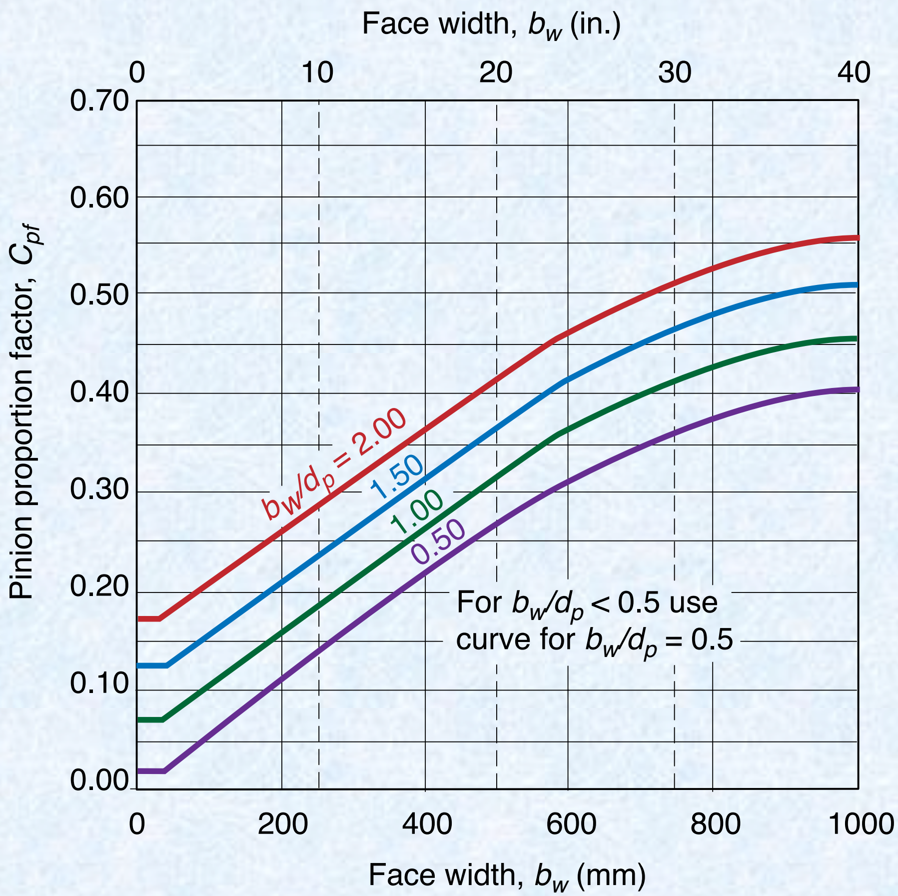


Figure 14.31: Pinion proportion factor, C_{pf}.
Source: ANSI/AGMA Standard 2101-D04 [2004].

Pinion proportion factor:

If $b_w < 25$ mm,

$$C_{pf} = \frac{b_w}{10d_p} - 0.025$$

For 25 mm < bw < 432 mm,

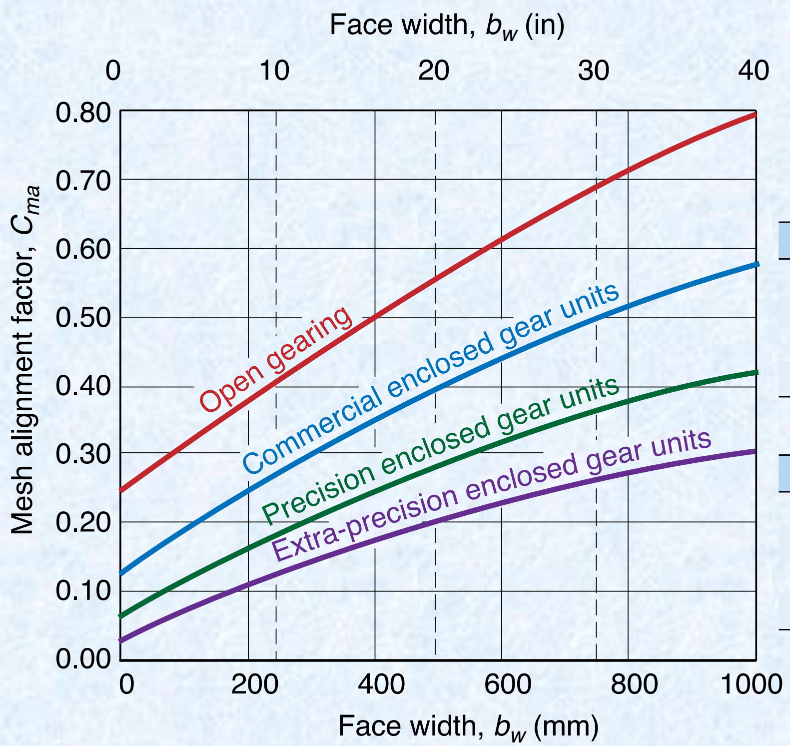
$$C_{pf} = \frac{b_w}{10d_p} - 0.0375 + 0.000492 b_w$$

For 432 mm < bw ≤ 1020 mm,

$$C_{pf} = \frac{b_w}{10d_p} - 0.1109 + 0.000815 b_w - (3.53 \times 10^{-7}) b_w^2$$

(see text if b_w is in inches)

Mesh Alignment Factor



$$C_{ma} = A + Bb_w + Cb_w^2$$

If b_w is in inches:

Condition	A	B	C
Open gearing	0.247	0.0167	-0.765×10^{-4}
Commercial enclosed gears	0.127	0.158	-1.095×10^{-4}
Precision enclosed gears	0.0675	0.0128	-0.926×10^{-4}
Extraprecision enclosed gears	0.000380	0.0102	-0.822×10^{-4}

If b_w is in mm:

Condition	A	B	C
Open gearing	0.247	6.57×10^{-4}	-1.186×10^{-7}
Commercial enclosed gears	0.127	6.22×10^{-4}	-1.69×10^{-7}
Precision enclosed gears	0.0675	5.04×10^{-4}	-1.44×10^{-7}
Extraprecision enclosed gears	0.000380	4.02×10^{-4}	-1.27×10^{-7}

Figure 14.33: Mesh alignment factor. *Source:* ANSI/AGMA Standard 2101-D04 [2004].

Pinion Proportion and Mesh Alignment Modifiers

Pinion proportion modifier:

$$C_{pm} = \begin{cases} 1.0, & (S_1/S) < 0.175 \\ 1.1, & (S_1/S) \geq 0.175 \end{cases}$$

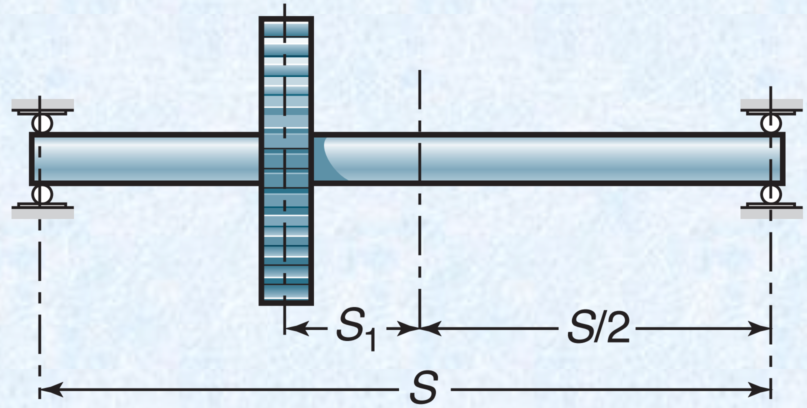


Figure 14.32: Evaluation of S and S_1. *Source:* ANSI/AGMA Standard 2101-D04 [2004].

Mesh alignment correction factor:

$$C_e = \begin{cases} 0.80 & \text{when gearing is adjusted at assembly} \\ 0.80 & \text{when compatibility between gear teeth is improved by lapping} \\ 1.0 & \text{for all other conditions} \end{cases}$$

Dynamic Factor

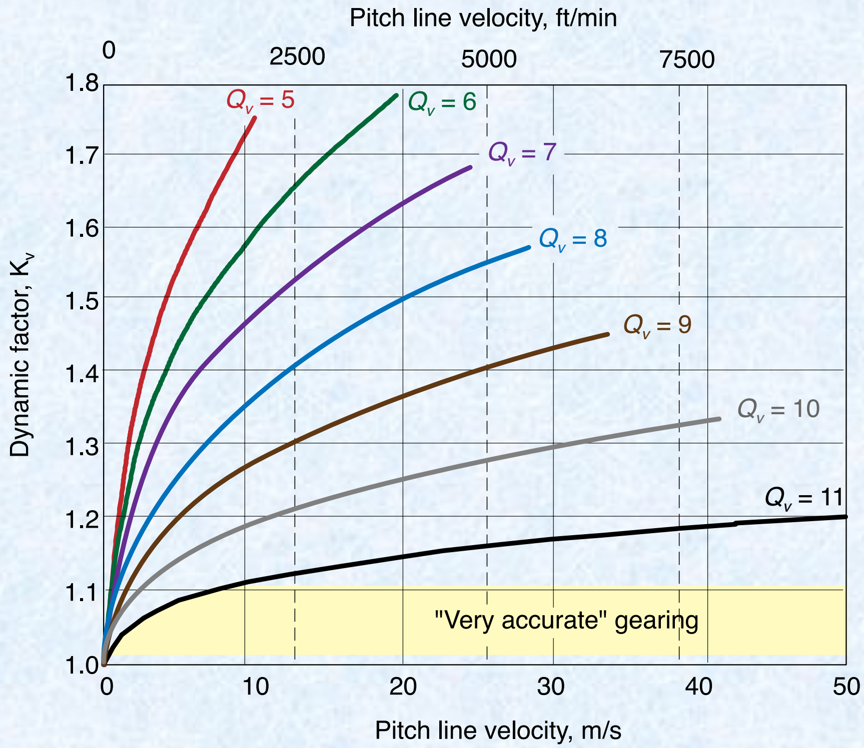


Figure 14.34: Dynamic factor as a function of pitch-line velocity and transmission accuracy level number. *Source:* ANSI/AGMA Standard 2101-D04 [2004].

Dynamic factor:

$$K_v = \left(\frac{A + C\sqrt{v_t}}{A} \right)^B$$

Where

$$A = 50 + 56(1.0 - B)$$

$$B = 0.25(12 - Q_v)^{0.667}$$

$$C = 1 \text{ for } v_t \text{ in ft/min}$$

$$C = \sqrt{200} = 14.14 \text{ for } v_t \text{ in m/s}$$

Maximum recommended pitch line velocity:

$$v_{t,\max} = \frac{1}{C^2} \left[A + (Q_v - 3) \right]^2$$

Design Procedure 14.2: Methods to Increase Bending Performance of Gears

If a gear does not produce a satisfactory design based on bending requirements, a design alteration may be needed. This is not always straightforward, since such alterations may help in one area and hurt in another, and may affect associated machine elements such as bearings. However, some factors that improve bending performance are the following:

1. Reduction in the load, such as by increasing contact ratio, or altering other aspects of the system.
2. Increase the center distance.
3. Apply gears with a coarser pitch.
4. Use a higher pressure angle.
5. Use a helical gear instead of a spur gear.
6. Use a carburized material.
7. Improve the gear accuracy through manufacturing process selection.
8. Select a better (stronger) gear material.
9. Use a wider effective face width.
10. Apply shot peening to the teeth.

Pitting

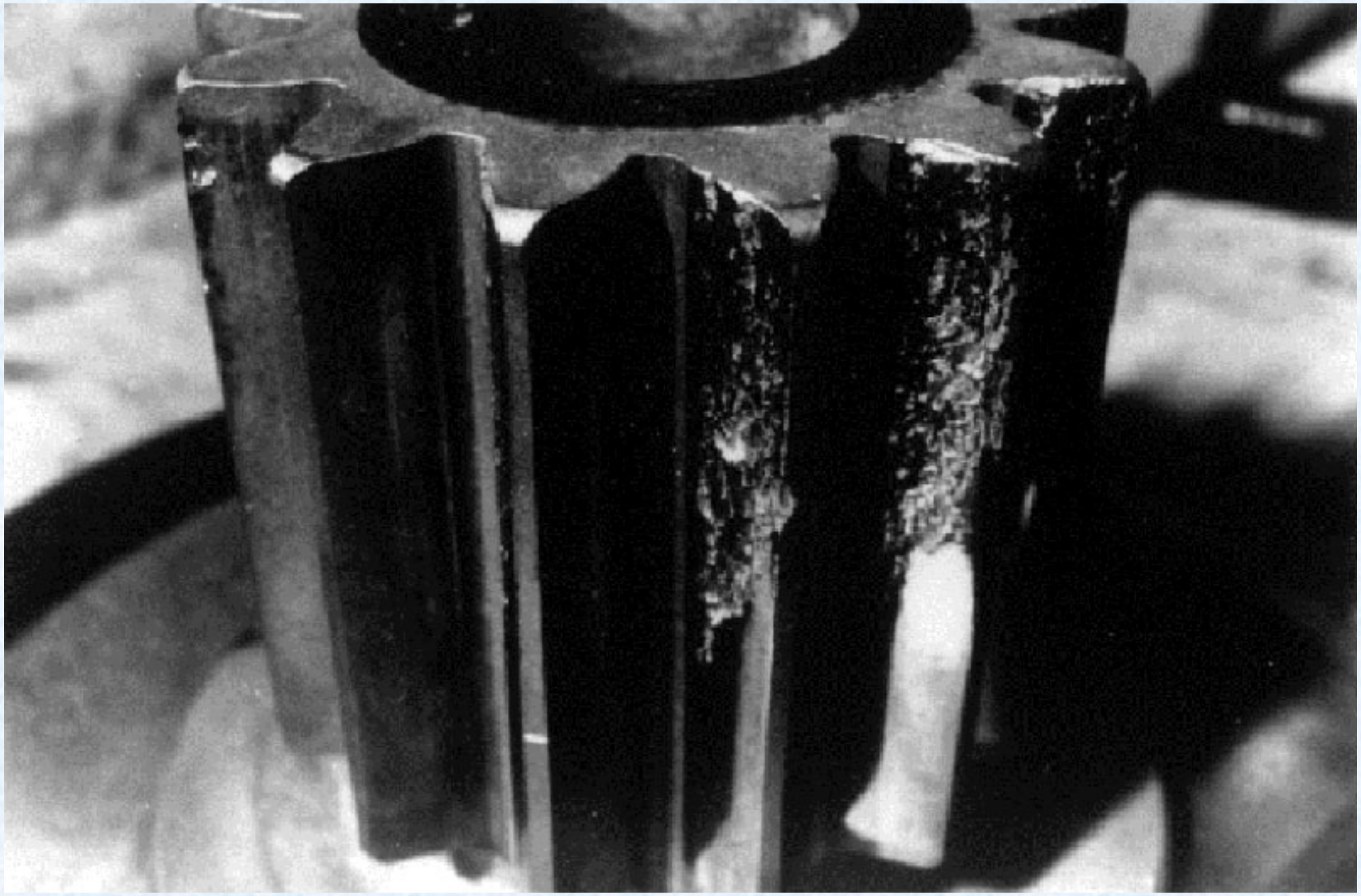


Figure 14.35: A gear showing extreme pitting or spalling. *Source:* Courtesy of the American Gear Manufacturing Association.

AGMA Contact Stress Equation

$$\sigma_c = E'\left(\frac{W'K_oK_sK_mK_v}{2\pi}\right)^{\frac{1}{2}} = p_H\left(K_oK_sK_mK_v\right)^{\frac{1}{2}}$$

Maximum Hertz pressure:

$$p_H = E'\left(\frac{W'}{2\pi}\right)^{\frac{1}{2}}$$

Where

$$E' = \text{effective elastic modulus} = \frac{2}{\dfrac{1-\nu_a^2}{E_a}+\dfrac{1-\nu_b^2}{E_b}}$$

$$W' = \text{dimensionless load} = \frac{w'}{E'R_x} = \frac{W}{E'R_xb_w}$$

$$\frac{1}{R_x} = \left(\frac{1}{r_p}+\frac{1}{r_g}\right)\frac{1}{\sin\phi} = \left(\frac{1}{d_p}+\frac{1}{d_g}\right)\frac{2}{\sin\phi}$$

$$b^* = R_x\left(\frac{8W'}{\pi}\right)^{1/2}$$

Ko, Ks, Km, and Kv are defined as with the AGMA Bending Stress Equation.

Design Procedure 14.3: Methods to Increase Pitting Resistance

If a gear does not produce a satisfactory design based on surface pitting requirements, a design alteration may be needed. This is not always straightforward, since such alterations may help in one area and hurt in another, and may affect associated machine elements such as bearings. However, some factors that improve pitting performance are the following:

1. Reduction in the load, such as by increasing contact ratio, or altering other aspects of the system.
2. Increase the center distance.
3. Apply gears with a finer pitch.
4. Use a higher pressure angle.
5. Use a helical gear instead of a spur gear.
6. Use a carburized material.
7. Increase the surface hardness by material selection, or by performing a surface hardening operation.
8. Improve the gear accuracy through manufacturing process selection.
9. Use a wider effective face width.
10. Increase the lubricant film thickness.

Design Procedure 14.4: Lubricant Film Thickness

It will be assumed that the power transmitted, gear and pinion angular velocity, number of teeth in pinion and gear, gear materials, lubricant properties, and geometry of the pinion and gear are known or can be determined from design constraints. Of these, the lubricant properties, especially the pressure exponent of viscosity, are most likely to be unknown or will have the largest uncertainty. Regardless, these properties should be attainable from the lubricant supplier.

The normal force acting on the gear teeth can be obtained from Eq. (14.50) as:

$$W = \frac{W_t}{\cos\phi}$$

The lubrication velocity for the pinion and gears is obtained from Eqs. (14.29) and (14.30) as

$$\tilde{u} = \frac{u_p + u_g}{2} = \frac{\omega_p r_p \sin\phi + \omega_g r_g \sin\phi}{2}$$

The effective radius is obtained from Eq. (14.74) as

$$\frac{1}{R_x} = \left(\frac{1}{r_p} + \frac{1}{r_g}\right)\frac{1}{\sin\phi}$$

Design Procedure 14.4 (concluded)

5. The effective modulus of elasticity is obtained from Eq. (8.16):

$$E' = \frac{2}{\dfrac{(1-\nu_a^2)}{E_a} + \dfrac{(1-\nu_b^2)}{E_b}}$$

6. For steel-on-steel, E'= 227.5$ GPa (32.97 Mpsi).
7. Equations (13.68), (13.69), and (13.72) yield the dimensionless load, speed, and materials parameters as:

$$W' = \frac{W}{b_w E' R_z}; \qquad U = \frac{\eta_o \tilde{u}}{E' R_x}; \qquad G = \xi E'$$

8. The lubricant film thickness is then obtained from the Hamrock-Dowson equation for rectangular contacts given by Eq.~(13.71):

$$H_{\min} = \frac{h_{\min}}{R_x} = 1.714 (W')^{-0.128} U^{0.694} G^{0.568}$$

Design Procedure 14.5: Gear Design Synthesis

1. From design requirements, determine the power transfer, total life required, and pinion and gear speed.
2. The gear ratio, g_r, must be known from design requirements. If this value is near or in excess of 6:1, a second stage is advisable.
3. If the pinion face-to-diameter ratio is not a design requirement, it can be estimated from the following equation for spur gears:

$$\frac{b_w}{d_p} = \frac{g_r}{g_r + 1},$$

   Where $g_r = N_g/N_p$ is the gear ratio.

4. Estimate K_o from Table 14.8.

Design Procedure 14.5 (continued)

5. The load distribution factor, K_m, cannot be determined until knowledge of the design, manufacturing approach, and mounting is established. A rough approximation based on pinion torque can be made according to

$$K_m = 1 + \frac{b_w}{d_p}\left[0.2 + 0.0112\left(\frac{T_p K_o}{b_w/d_p}\right)^{0.33}\right]$$

   However, if the pitch diameter is known (from design requirements or selected based on experience), then the load distribution factor can be more accurately estimated from

$$K_m = 1 + \frac{b_w}{d_p}(0.2 + 0.0012 d_p)$$

   Recall that the torque is related to the power by $h_p = T\omega$.

6. The dynamic factor, K_v, depends on gear speed and quality. For simplicity, a value of $K_v = 1.43$ can be assumed, which is conservative for most applications.

Design Procedure 14.5 (continued)

7. The pressure angle, ϕ, is generally chosen as 20°, since these gears are widely available. However, the pressure angle can be reduced to obtain higher contact ratios, or increased when precision and noise are not issues. Regardless, it is recommended that standard values of 14.5°, 17.5°, 22.5°, or 25° be used.
8. A geometry factor for spur gears is given by:

$$I = \frac{\sin\phi\cos\phi}{2}\frac{g_r}{g_r \pm 1}$$

   Where g_r is the gear ratio. Note that the plus sign in the denominator of Eq.~(14.79) applies for external gearsets, the negative sign for internal gearsets.
9. Y_j can be conservatively estimated as approximately 0.45 (see Fig. 14.30).
10. The life factors Y_n and Z_n can be estimated from Fig. 14.26.
11. Obtain the bending (σ_b) and contact (σ_c) strengths for the gear material according to Eqs.~(14.45) or (14.46), respectively.

Design Procedure 14.5 (continued)

12. An estimate for the preferred number of pinion teeth is then:

$$N_p = \frac{1}{K_b}\frac{Y_j}{I}\frac{\sigma_b}{\sigma_c^2}\frac{E'}{2\pi}\frac{n_{sc}^2}{n_{sb}}$$

For the special case of steel gears (E′ = 227 GPa) a pressure angle of ϕ=20°, K_b=1, n_{sc}=n_{sb}=n_s, and Y_j=0.45, the preferred number of pinion teeth becomes

$$N_p = (101.17\ \text{GPa})\frac{g_r+1}{g_r}\frac{\sigma_b}{\sigma_c^2}n_s$$

13. Obtain the pitting resistance constant:

$$K_c = \frac{0.3183E'h_pK_oK_mK_v}{I\omega_p}\left(\frac{n_{sc}}{\sigma_c}\right)^2$$

14. The pinion pitch diameter is then estimated as

$$d_p = \left(\frac{K_c}{b_w/d_p}\right)^{1/3}$$

Design Procedure 14.5 (concluded)

15. The face width can be calculated from the ratio of b_w/d_p established in step 3 if a specific value was not specified as a design requirement.
16. When the gear profile is selected from this design procedure, it is necessary to analyze the design more closely in accordance with the approaches described in Sections 14.11 and 14.12. This is clearly required since a number of approximate values were used.
17. It should also be noted that an essential part of the gear design synthesis is calculating lubricant film thickness as described in Section 14.13.

PM Spur Gear Manufacture

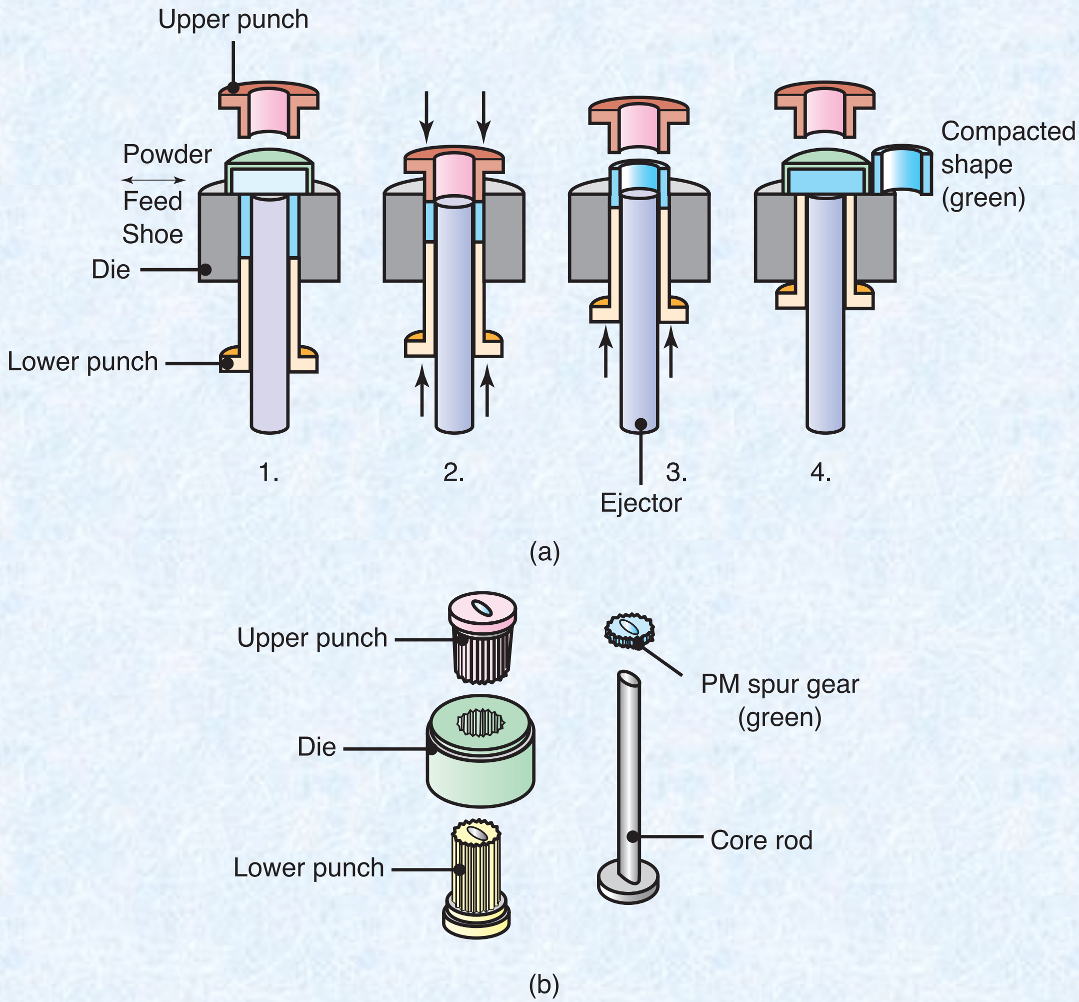


Figure 14.36: Production of gears through the powder compaction process. (a) Steps required to produce a part; (b) illustration of tooling required for a simple spur gear. *Source:* After Kalpakjian and Schmid [2010].

Surface Densified PM Gears

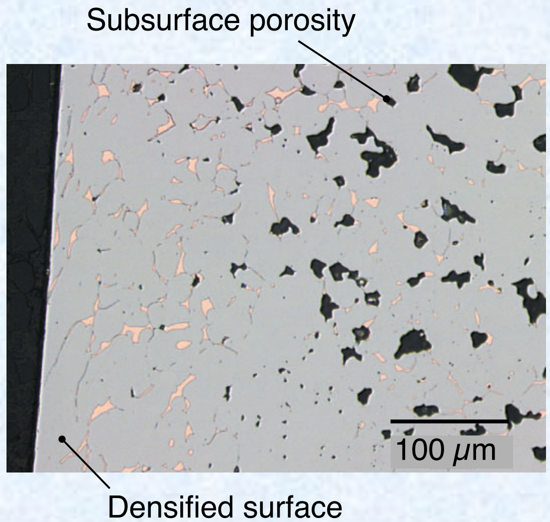


Figure 14.37: Image of a PM material after roll densification. Note the low porosity near the surface, resulting in high allowable contact stresses. *Source:* Courtesy Capstan Atlantic Corp.

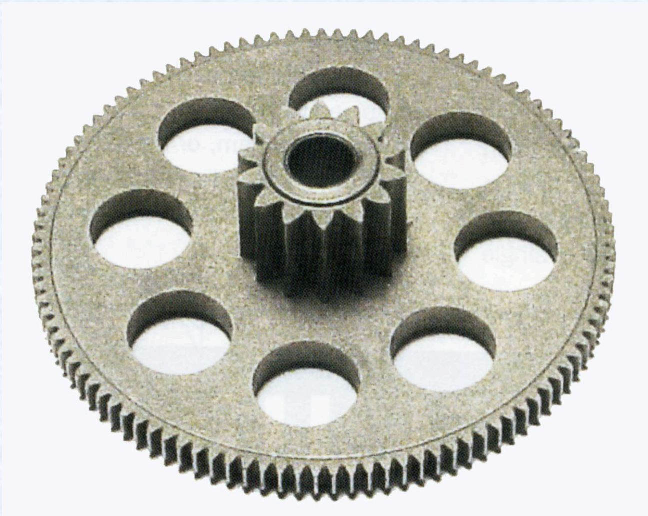


Figure 14.38: A stepped gear produced through the metal injection molding process. *Source:* Courtesy Perry Tool & Research, Inc.

Figure 14.39: A collection of PM gears used in automotive applications. *Source:* Courtesy Capstan Atlantic Corp.