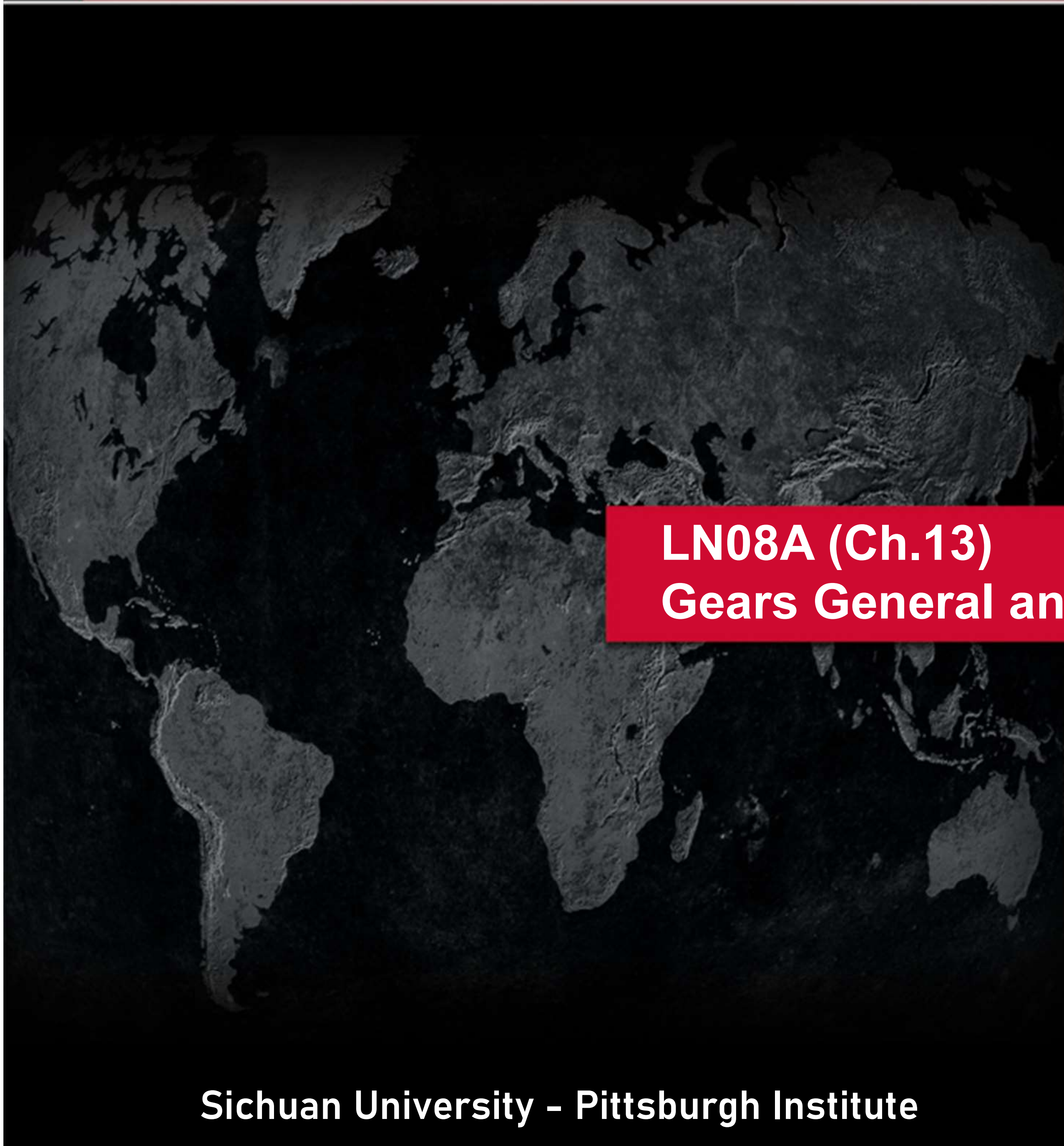
LN08A (Ch.13)
Gears General and Spur Gears



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ME 1029 Machine Design 2

Fall 2021

Gears

- Ch.13 Gears – General
- Ch.14 Spur and Helical Gears
- Ch.15 Bevel and Worm Gears

Focus Topics – Spur Gears

Gear - General

- 13–1 Types of Gears
- 13–2 Nomenclature

Gear Profile Definition

- 13–3 Conjugate Action
- 13–4 Involute Properties
- 13–5 Fundamentals
- 13–6 Contact Ratio
- 13–7 Interference
- 13–8 The Forming of Gear Teeth
- 13–10 Parallel Helical Gears
- 13–12 Tooth Systems

Gear Train Force Analysis

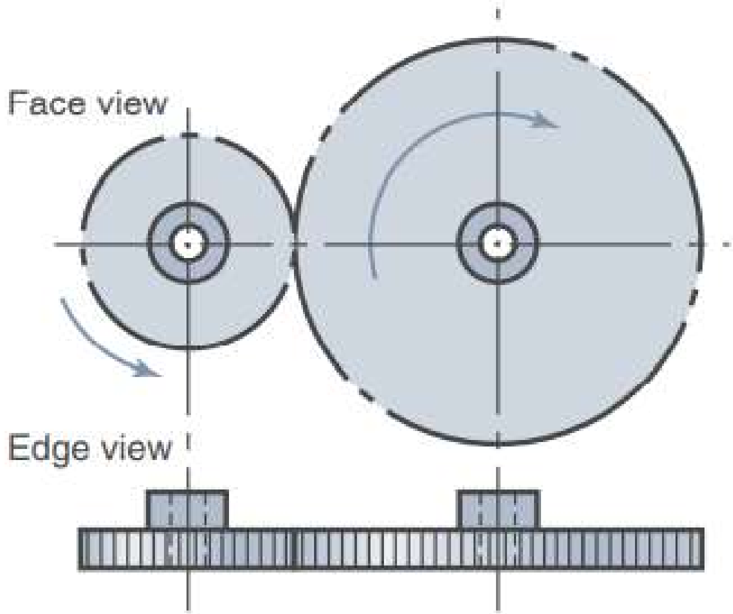
- 13–13 Gear Trains
- 13–14 Force Analysis - Spur Gearing
- 13–16 Force Analysis - Helical Gearing

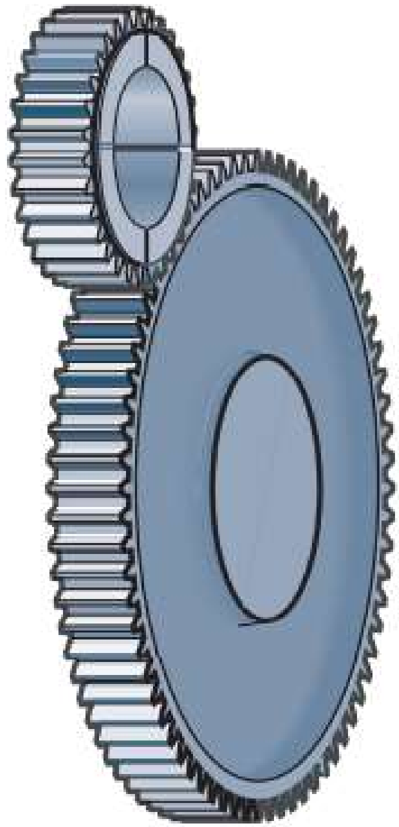
Gear Structural Design Analysis

- Ch.14 Spur and Helical Gears

Types of Gears: Spur Gear

- Spur gears have teeth parallel to the axis of rotation and are used to transmit motion from one shaft to another, parallel, shaft.
- Of all types, the spur gear is the simplest.





13-9 Straight Bevel Gears
13-10 Parallel Helical Gears
13-11 Worm Gears

Parallel Helical

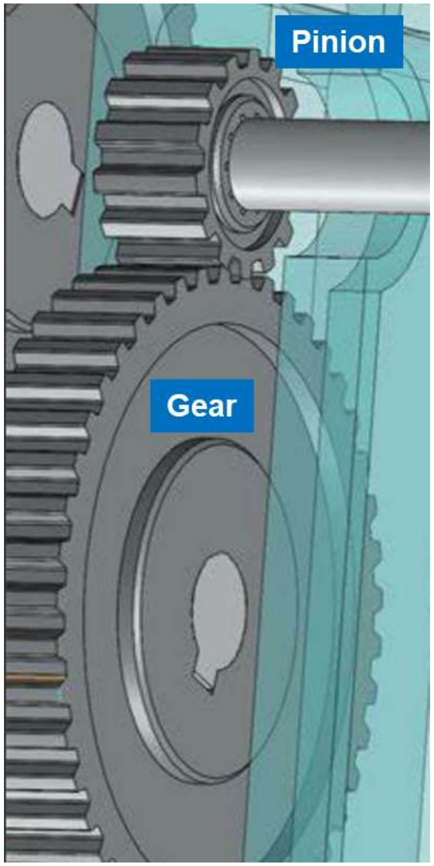
Cross Helical

Worm
Gears

13-2 **Nomenclature**

Gear Nomenclature

- **Pitch Circle**: a theoretical circle upon which all calculations are usually based;
  - its diameter is the **pitch diameter** (d).
  - Pinion: the **smaller** of the two mating gears
  - Gear: the **larger** of the two mating gears
- **Circular Pitch** $\left(p = \frac{\pi d}{N}\right)$: distance measured on the pitch circle, from a point on one tooth to a corresponding point on an adjacent tooth.
  - circular pitch = tooth thickness + width of space



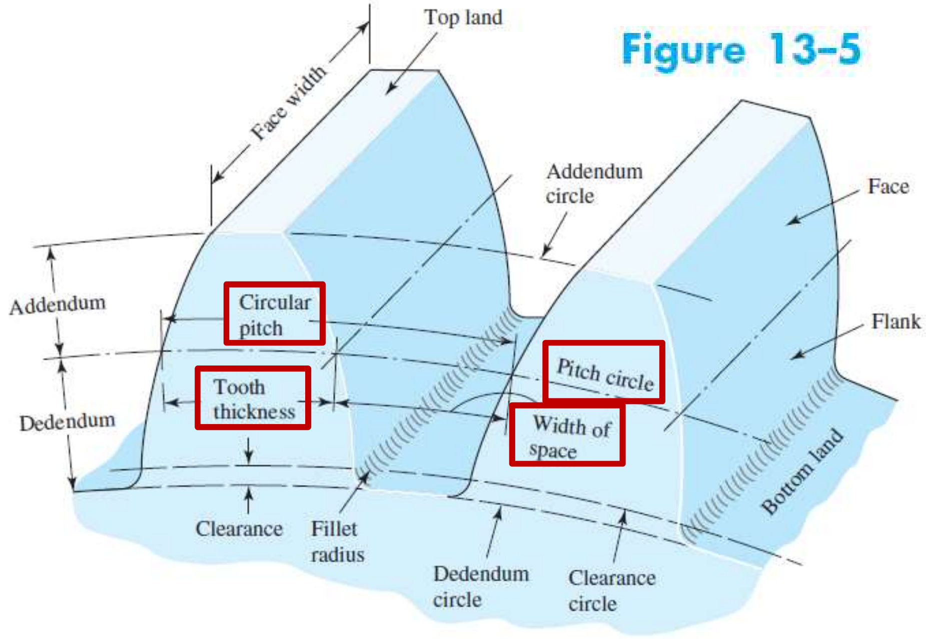
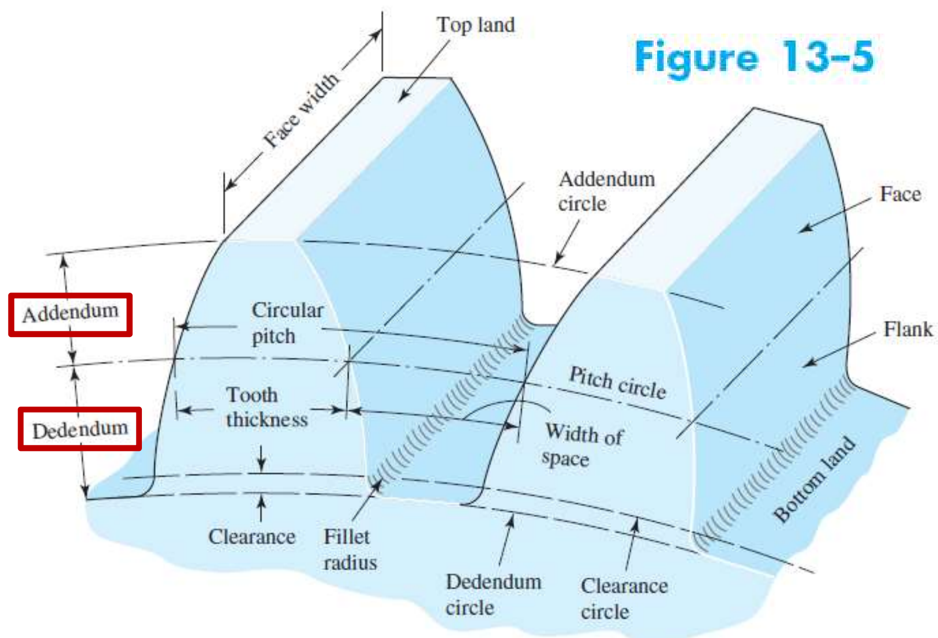


Figure 13-5

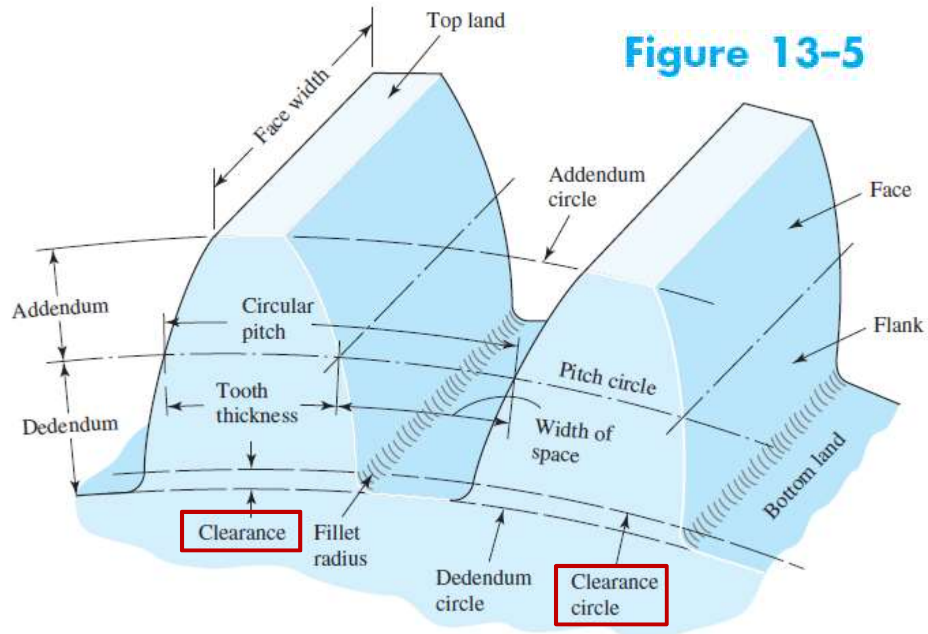
Gear Nomenclature

- **Addendum** (a): radial distance between the top land and the pitch circle
- **Dedendum** (b): radial distance from the bottom land to the pitch circle.
- **Whole Depth** $(h_t = a + b)$: sum of addendum and dedendum
- **Backlash**: the amount by which the width of a tooth space exceeds the thickness of the engaging tooth measured on the pitch circles
  - Backlash=circular pitch (p) -2*tooth thickness



Gear Nomenclature

- **Clearance Circle**: a circle that is tangent to the addendum circle of the mating gear
- **Clearance** $(c = b - a)$: is the amount by which the dedendum in a given gear exceeds the addendum of its mating gear



Frequently Used Gear Geometry Relationship

$$m = \frac{d}{N}$$

$$P = \frac{d}{N} = \frac{1}{m}$$

$$p = \frac{\pi d}{N} = \pi m$$

$$pP = \pi$$

- P : diametral pitch, teeth per inch
- N : number of teeth
- d : pitch diameter, in or mm
- m : module, mm
- p : circular pitch, in or mm

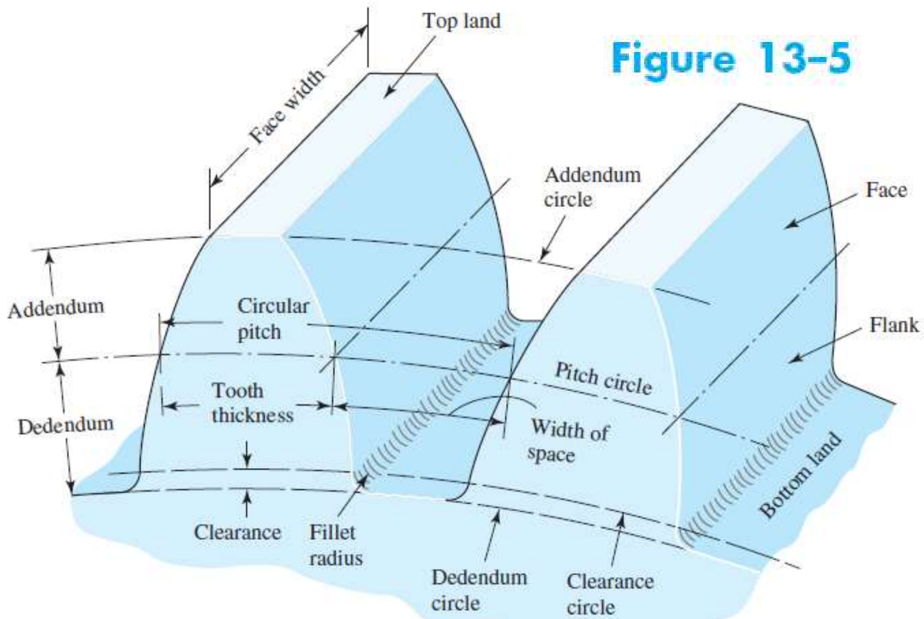


Figure 13-5

- Module (m) is the index of tooth size in SI (unit: mm)
- Diametral Pitch (P): Used only with U.S. units only (Unit: teeth per inch)
- For correctly meshing gears, **module (diametral pitch)** must be the same for the two gears.

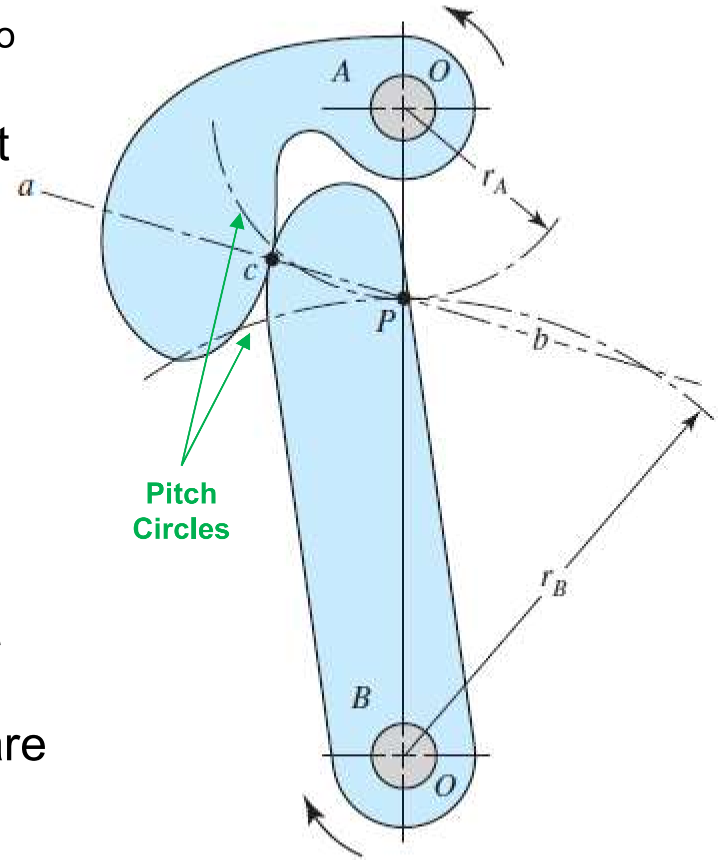
13-3 Conjugate Action

- Mating gear teeth acting against each other to produce rotary motion.
  - Point **C**: point of contact (tangent point between two profiles)
- For the bodies to remain in contact, there must be no component of relative motion along the common normal:

$$\frac{\omega_A}{\omega_B} = \frac{r_B}{r_A} = constant$$

  - r_A & r_B: pitch radius
  - Point **P**: pitch point
- When one tooth profile designed with constant angular velocity also produces a constant angular velocity in the meshing profile, these are said to have **conjugate action**.
- Involute profile is one of these solutions which, with few exceptions, is in universal use for gear teeth.

Figure 13–6



13-3 Conjugate Action

- With involute profile, all points of contact occur along the same line which is the **line of action** (Line a-b).
- To transmit rotation at constant angular velocity the pitch point must remain fixed, meaning that the line of action must pass through the same point “P”.

- Point **C**: point of contact
  - tangent point between two profiles
- Point **P**: pitch point

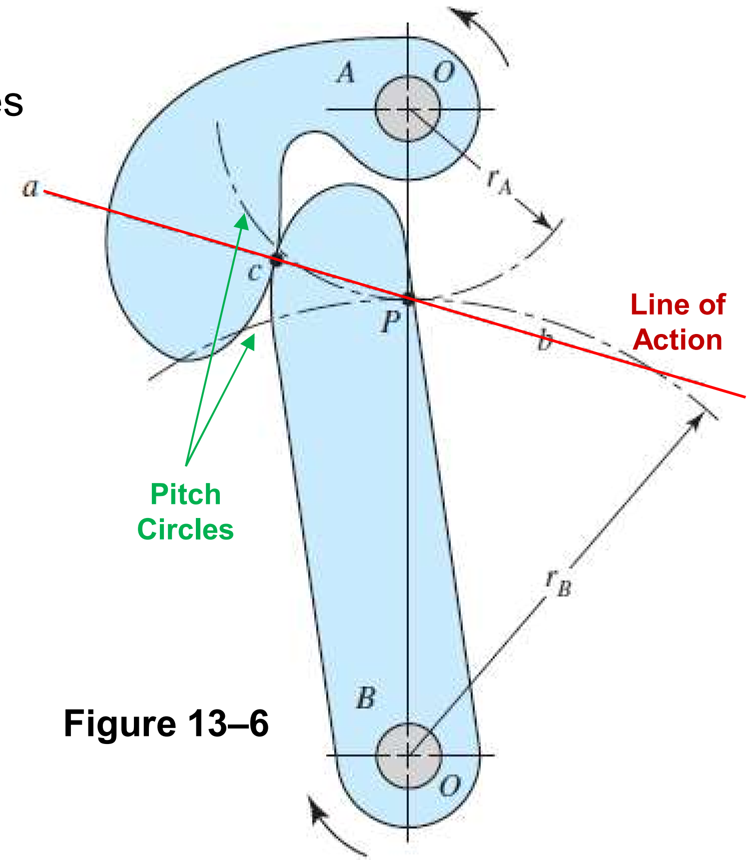


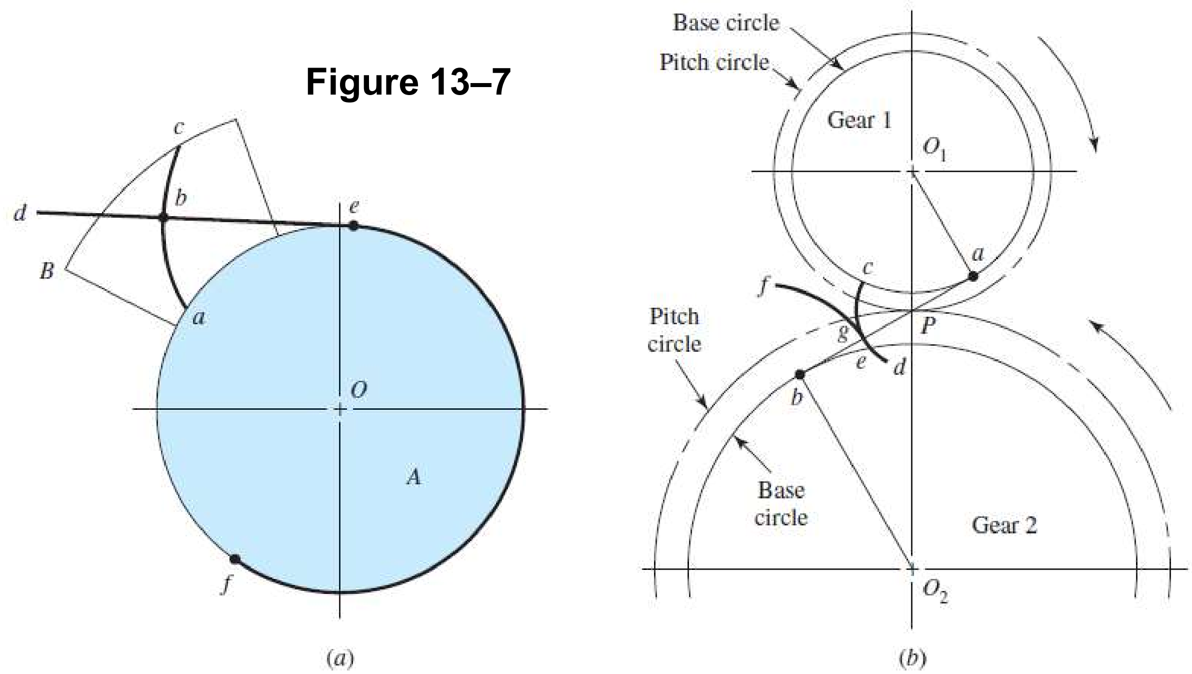
Figure 13–6

13-4 Involute Properties

To generate an involute curve:

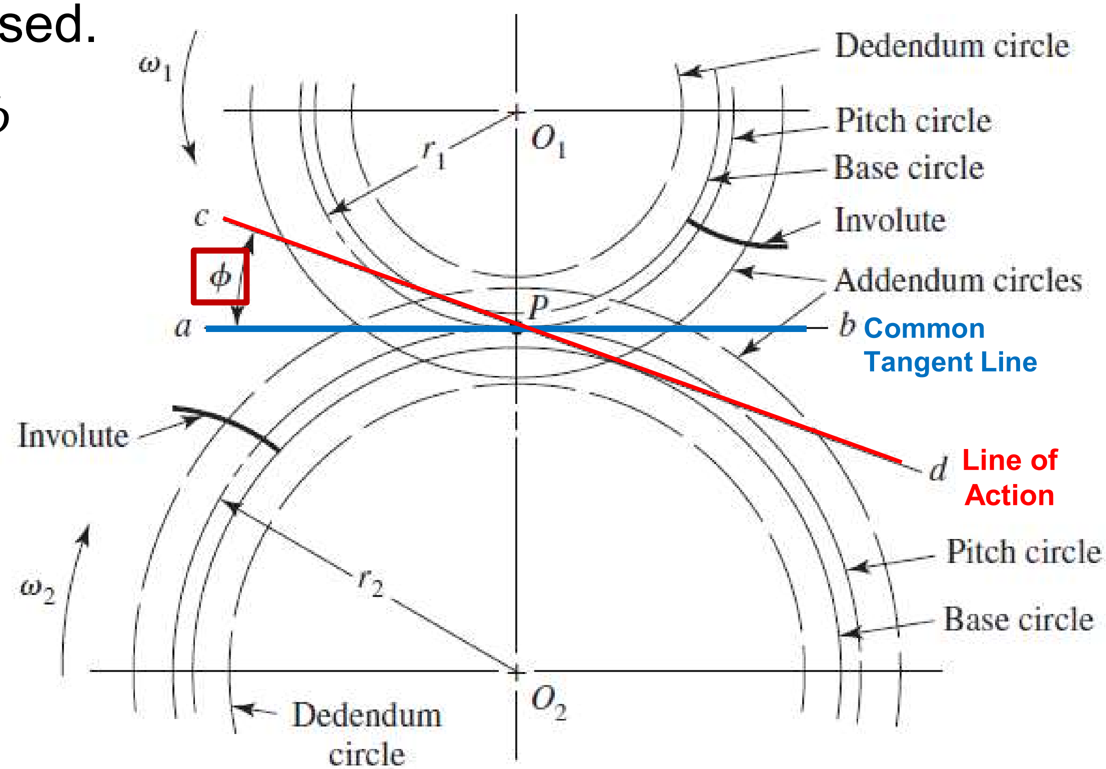
- A partial flange B is attached to the cylinder A
- Cord d-e-f is wrapped around cylinder A and held tight
- Point b on the cord represents the tracing point
- As cord is wrapped and unwrapped about the cylinder, point b will trace out the involute curve ac.
- The circle on which the involute is generated is called the base circle (cylinder A)

Figure 13–7



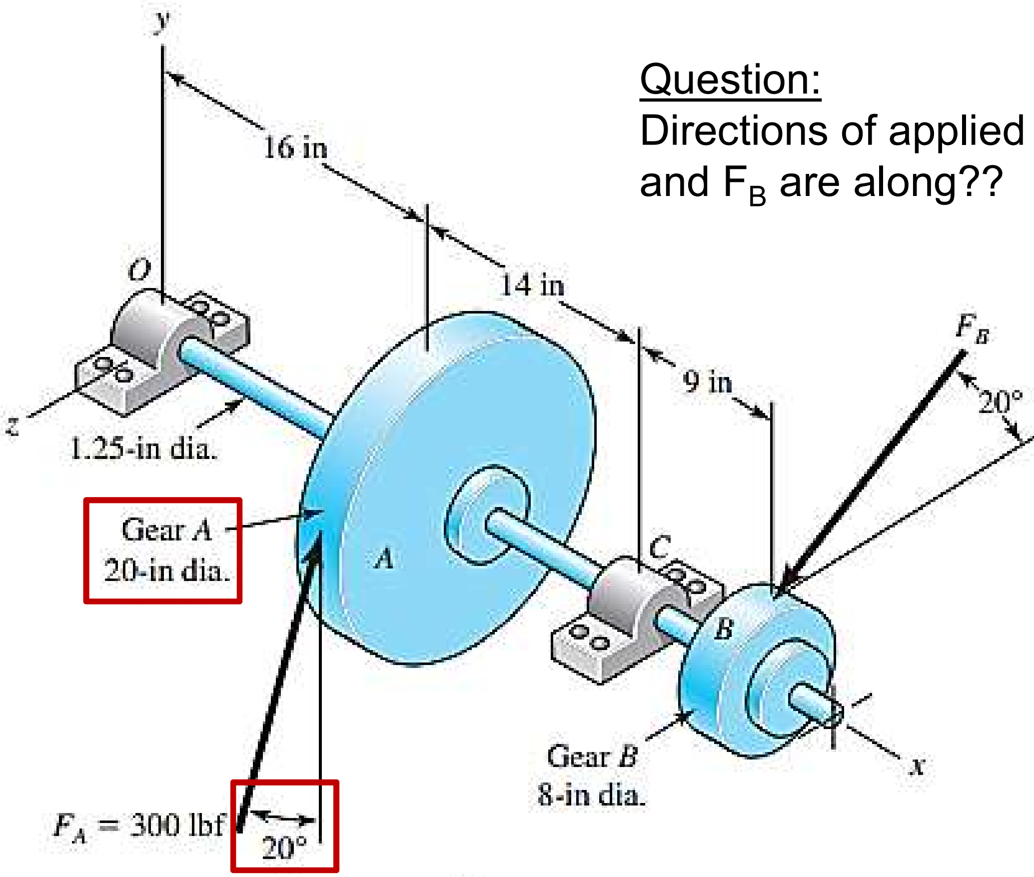
Tooth Action

- Line **ab**: common tangent line
- Line **cd**: known as the pressure line, generating line, and **line of action**, which represents the direction the resultant force acts between the gears.
- Angle ϕ : the **pressure angle**, and it usually has values of 20 or 25°, though $14\frac{1}{2}°$ was once used.
- Base Pitch $p_b = p\cos\phi$



Recall Ch.03 Problem 3-72

A gear reduction unit uses the countershaft shown in the figure. Gear A receives power from another gear with the transmitted force F_A applied at the 20° pressure angle as shown.



Problem 3–72*

Question:
Directions of applied forces F_A and F_B are along??

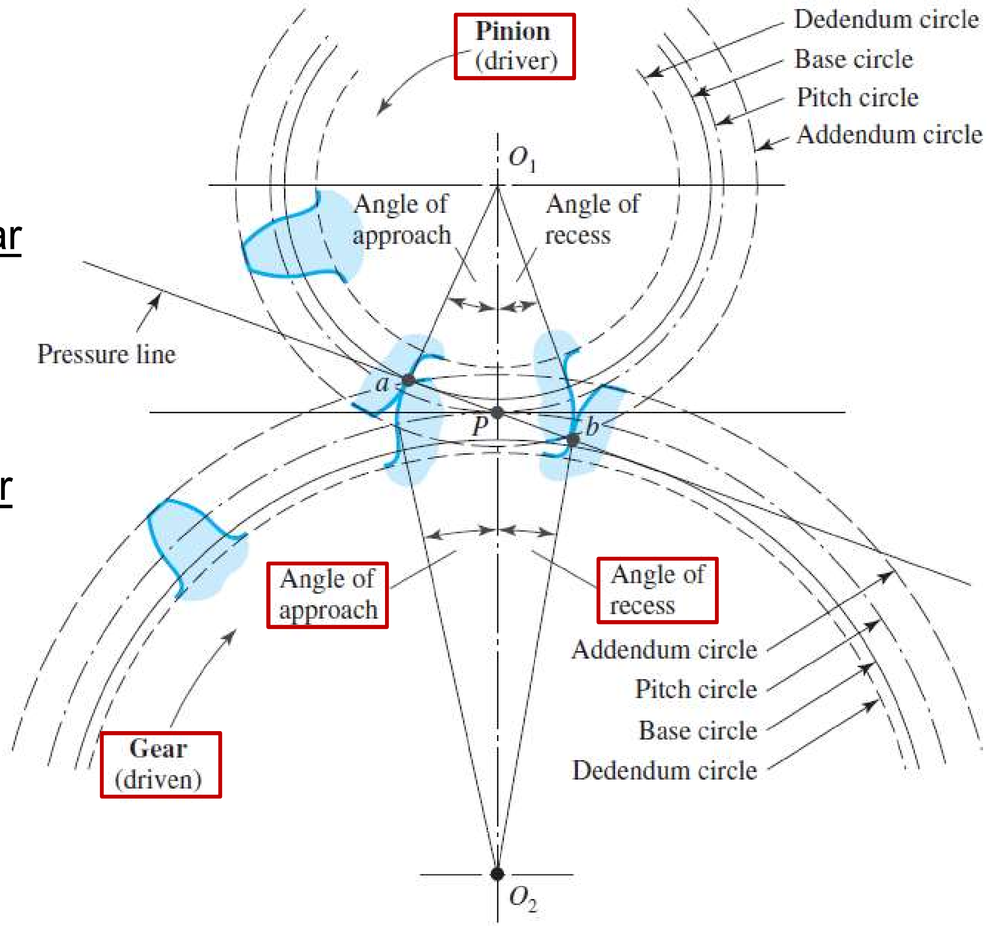
Tooth Action

Pinion: **Driver** (CCW)

Gear: **Driven** (CW)

- Point **a**: initial contact
  - addendum circle of the driven gear crosses the pressure line
- **Angle of Approach (a ► P)**
- Point **b**: Final contact
  - addendum circle of the driver gear crosses the pressure line
- **Angle of Recess (P ► b)**
- **Angle of Action**= Angle of Approach+ Angle of Recess
- Angle of Action on Pinion Side ≠ Angle of Action on Gear Side

Figure 13-12



13-6 Contact Ratio

- Defines the average number of pairs of teeth in contact at any time.

- Initial contact occurs at **a** and final contact at **b.**
- Tooth profiles drawn through these points intersect the pitch circle at A and B, respectively.
- Distance AP: arc of approach q_a
- Distance PB: arc of recess q_r
- Arc of action: $q_t = q_a + q_r$
- Contact Ratio $m_c = \frac{q_t}{p} = \frac{L_{ab}}{p \cos \phi} \quad (L_{ab} = q_t \cos \phi)$

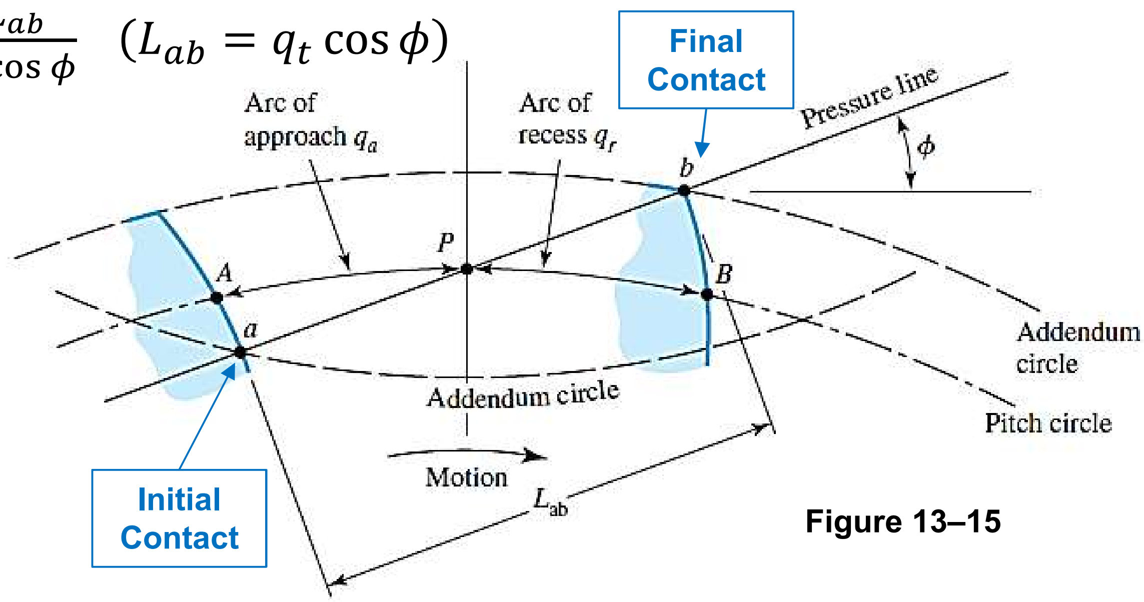
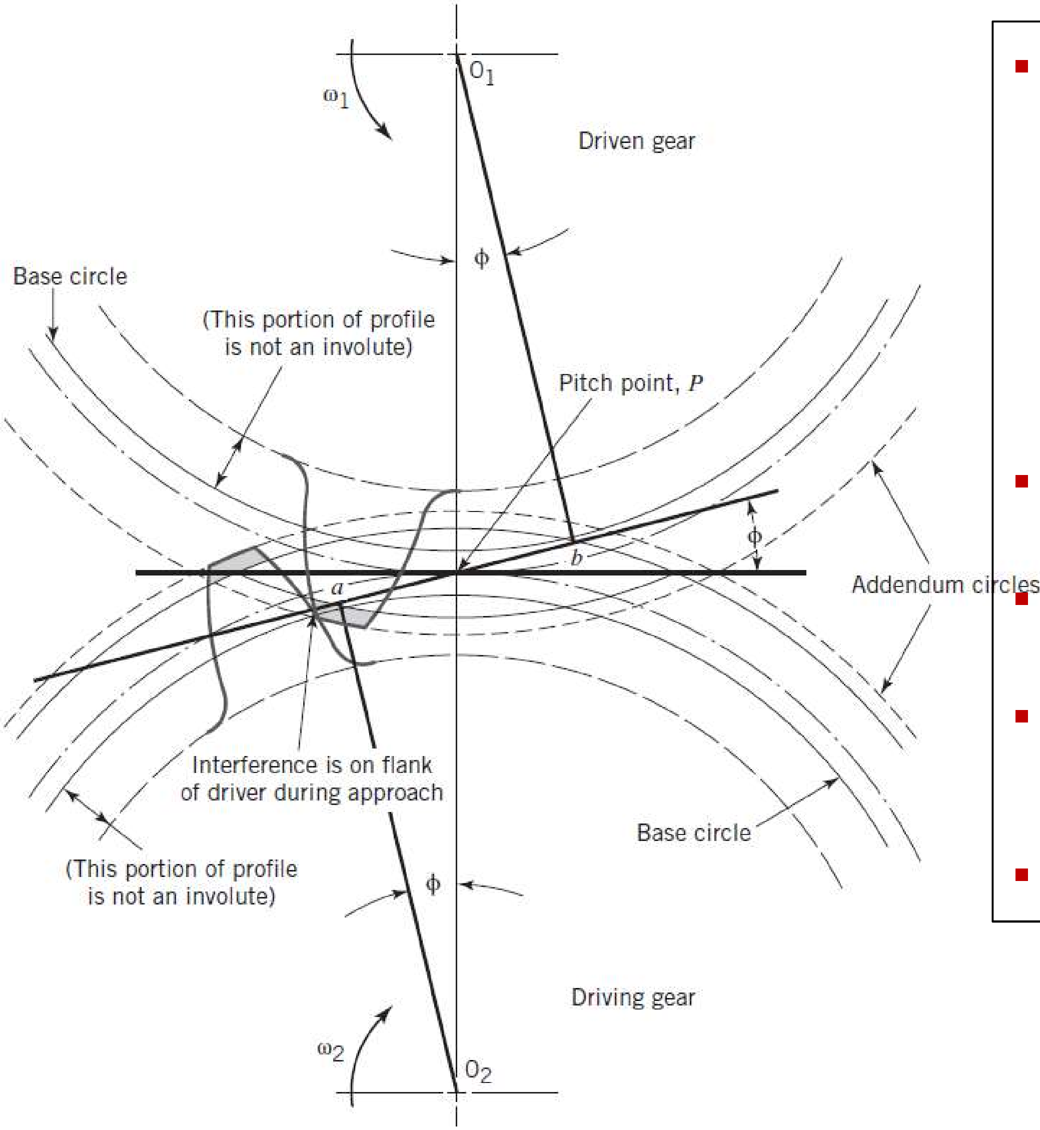


Figure 13–15

13-6 Contact Ratio



- Contact Ratio $m_c = \frac{q_t}{p} = \frac{L_{ab}}{p\cos\phi}$
  $(L_{ab} = q_t \cos\phi)$

$$m_c = \frac{\sqrt{r_{ap}^2 - r_{bp}^2} + \sqrt{r_{ag}^2 - r_{bg}^2} - C\sin\phi}{p_b}$$

- r_{ap} & r_{ag}: addendum radii of the mating pinion and gear
- r_{bp} & r_{bg}: base circle radii of the mating pinion and gear
- p_b: base pitch

$$p_b = p\cos\phi = \frac{\pi d_b}{N}$$

- C: Center Distance

13-6 Contact Ratio

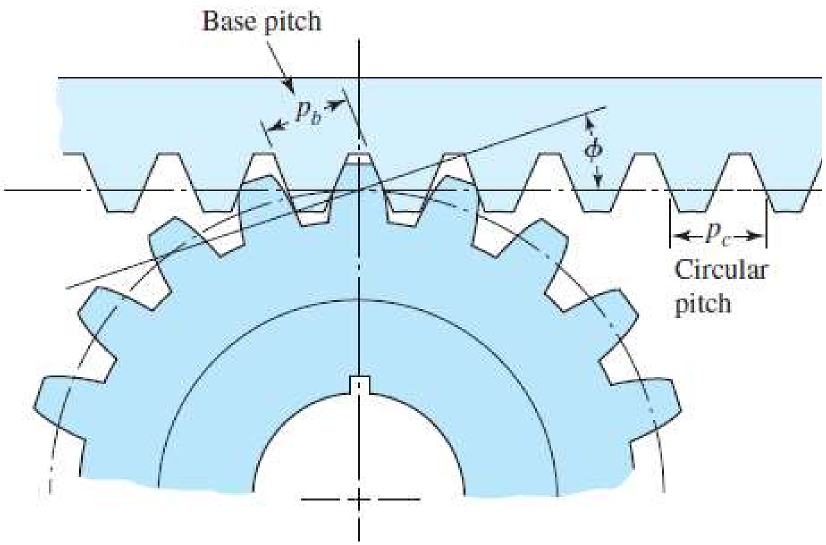
- Defines the average number of teeth in contact at any time.
- Contact ratio = 1.0
  - A new pair of teeth coming in contact just as the old pair is leaving contact, and there will be exactly one pair of teeth in contact at all times.
  - Seem acceptable, but leads to poor performance, since loads will be applied at tooth tips, and there will be additional vibration, noise, and backlash.
- Contact ratio = 1~2
  - part of the time two pairs of gear teeth are in contact and during the remaining time one pair is in contact.
- Contact ratio = 2~3
  - part of the time three pairs of teeth are in contact.
  - Rare with spur gear sets, but typical for helical and worm gears.

- Contact ratio is important for performance; for example, if two pairs of teeth share the transmitted load, the stresses are lower.

- Good practice to maintain a contact ratio of 1.2 or greater.
- Most spur gearsets have contact ratios between 1.4 and 2.0.

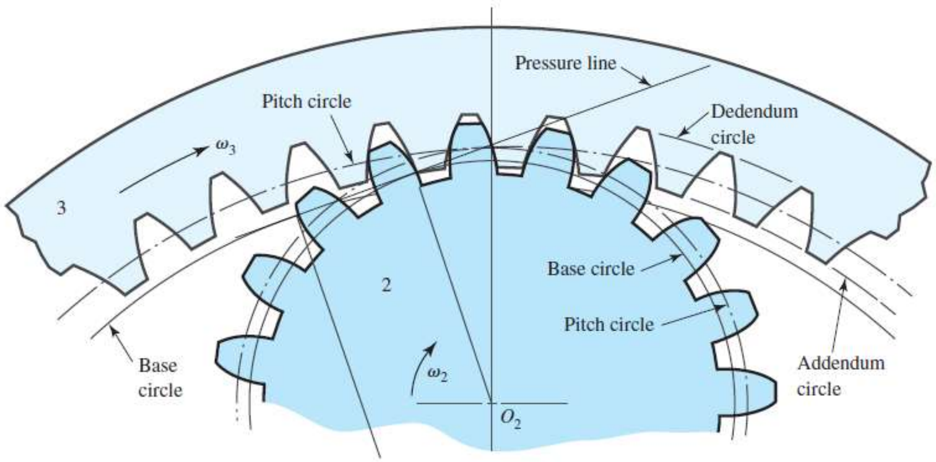
Alternative Gear Meshing

Figure 13–13
Involute-toothed pinion and rack.

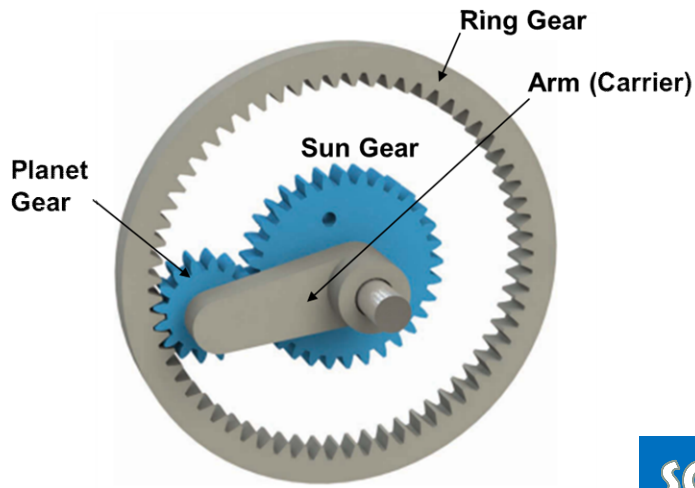


Rotation ⇔ Translation

Figure 13–14
Internal gear and pinion



Achieve high speed reduction
in a very small volume.



EXAMPLE 13–1

A gear set consists of a 16-tooth pinion driving a 40-tooth gear. The diametral pitch (P) is 2, and the addendum and dedendum are 1/P and 1.25/P, respectively. The gears are cut using a pressure angle of 20°.

a) Compute the circular pitch, the center distance, and the radii of the base circles.

Tooth Number, Pinion	$N_P := 16$	Tooth Number, Gear	$N_G := 40$
Diametral Pitch	$P := 2$	Circular Pitch	$p := \frac{\pi}{P} \cdot in = 1.571\ in$
Pitch Diameter, Pinion	$d_P := \frac{N_P}{P} \cdot in = 8\ in$	Pitch Diameter, Gear	$d_G := \frac{N_G}{P} \cdot in = 20\ in$
Center Distance	$\frac{d_P + d_G}{2} = 14\ in$		
Pressure Angle	$\phi := 20\ °$		
Radius of Base Circle, Pinion	$r_{bP} := \frac{d_P}{2} \cdot \cos(\phi) = 3.759\ in$		
Radius of Base Circle, Gear	$r_{bG} := \frac{d_G}{2} \cdot \cos(\phi) = 9.397\ in$		

EXAMPLE 13–1

The gears are cut using a pressure angle of 20°.

b) In mounting these gears, the center distance was incorrectly made ¼-in larger. Compute the new values of the pressure angle and the pitch-circle diameters.

$$\frac{d_P + d_G}{2} = 14 + 0.25 = 14.25\ in$$

$$\frac{d_P}{d_G} = \frac{N_P}{N_G} = \frac{16}{40}$$

The above relation gives: $d_P = 8.143\ in$ $\qquad d_G = 20.357\ in$

Since $r_b = d\cos\phi$; therefore, the new pressure angle

$$\phi = \cos^{-1}\left(\frac{r_{bP}}{d_P}\right) = \cos^{-1}\left(\frac{3.759}{8.143}\right) = 22.59°$$

13-12 Tooth Systems

- A standard that specifies the relationships involving addendum, dedendum, working depth, tooth thickness, and pressure angle.
- Standardized by the American Gear Manufacturers Association (AGMA).

Commonly Used Tooth Systems for Spur Gears

Table 13-1 Spur Gears

Tooth System	Pressure Angle ϕ, deg	Addendum a	Dedendum b
Full depth	20	$1/P$ or m	$1.25/P$ or $1.25m$
			$1.35/P$ or $1.35m$
	$22\frac{1}{2}$	$1/P$ or m	$1.25/P$ or $1.25m$
			$1.35/P$ or $1.35m$
	25	$1/P$ or m	$1.25/P$ or $1.25m$
			$1.35/P$ or $1.35m$
Stub	20	$0.8/P$ or $0.8m$	$1/P$ or m

$$P = \frac{N}{d}$$

$$m = \frac{d}{N}$$

$$p = \frac{\pi d}{N} = \pi m$$

$$pP = \pi$$

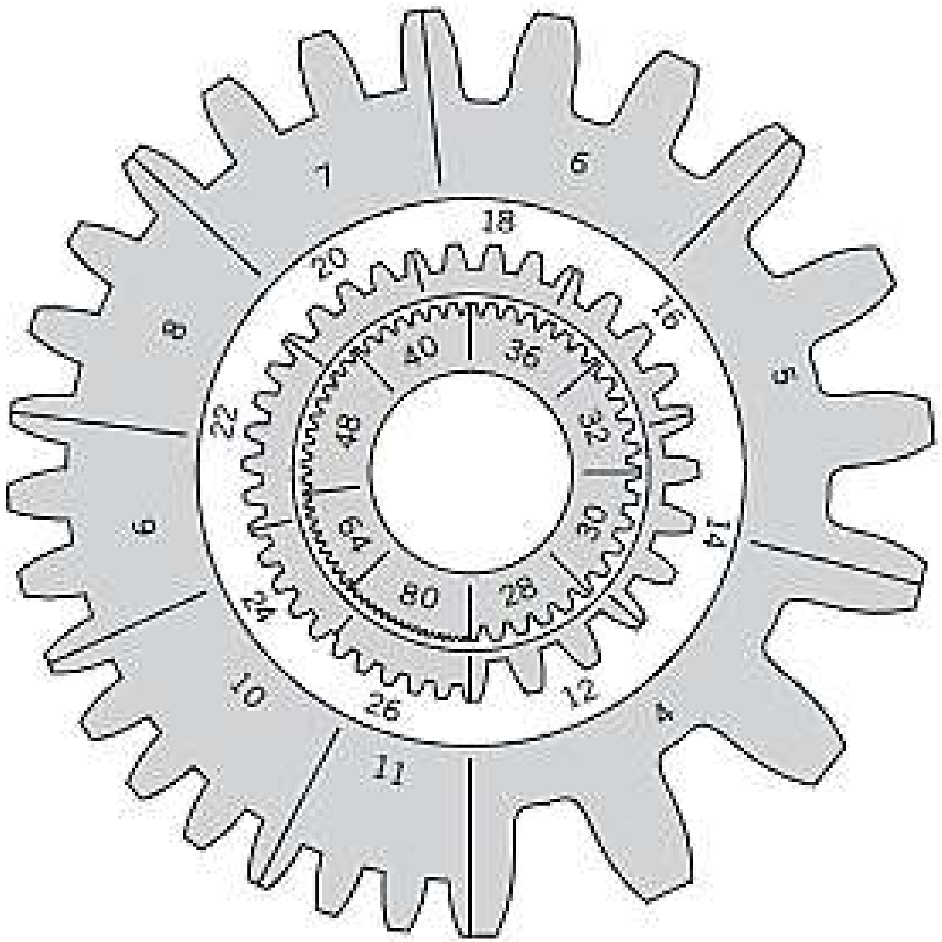
Table 13-2 Tooth Size in General Uses

Diametral Pitch P(teeth/in)	
Coarse	2, $2\frac{1}{4}$, $2\frac{1}{2}$, 3, 4, 6, 8, 10, 12, 16
Fine	20, 24, 32, 40, 48, 64, 80, 96, 120, 150, 200

Module m(mm/tooth)	
Preferred	1, 1.25, 1.5, 2, 2.5, 3, 4, 5, 6, 8, 10, 12, 16, 20, 25, 32, 40, 50
Next Choice	1.125, 1.375, 1.75, 2.25, 2.75, 3.5, 4.5, 5.5, 7, 9, 11, 14, 18, 22, 28, 36, 45

Commonly Used Tooth Systems for Spur Gears

Visual comparison of actual sizes of gear teeth of various diametral pitches (teeth per inch).



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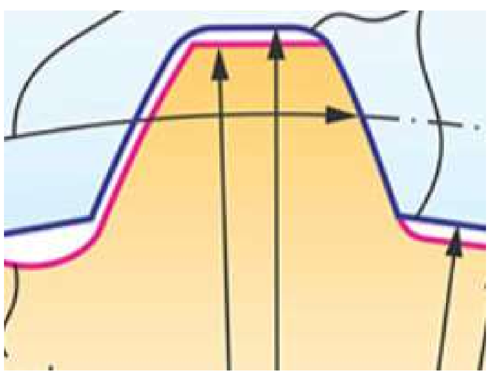
13-7 Interference

Gear Mesh Interference

- Gear teeth cut completely per involute profile does not have interference problem.
- Sometimes this could results in pronounced <u>undercutting</u> to considerably weaken the tooth.
- To restore the structure strength, tooth flank profile occasionally deviates from involute during cutting.
- Interference can occur on non-involute portion of the flan



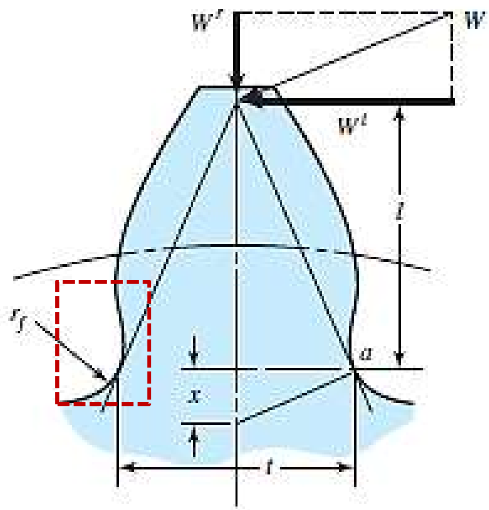
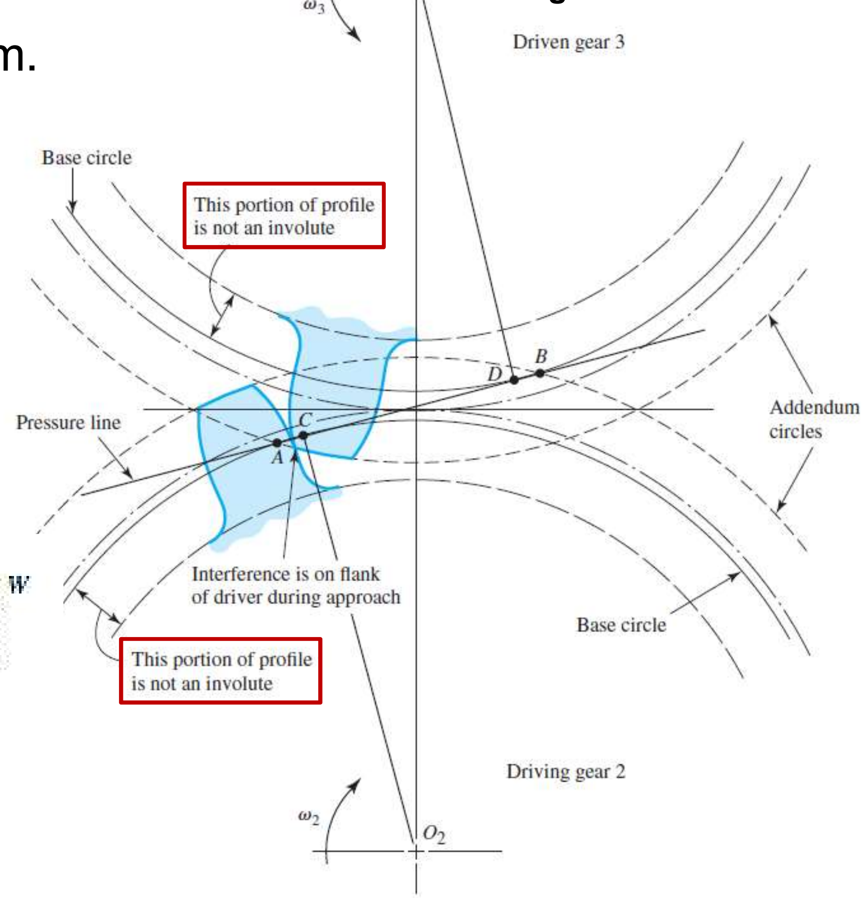


Figure 13–15



Discussions on Gear Interference

Problem of teeth weakened by undercutting is a serious design concern.

Alternative means to alleviate interference concern:

- Interference can be eliminated by using <u>more teeth</u> on the pinion.
  - However, if the pinion is to transmit a given amount of power, more teeth can be used only by increasing the pitch diameter.

- Interference can also be reduced by using a <u>larger pressure angle</u>.
  - This results in a smaller base circle, so that more of the tooth profile becomes involute. The demand for smaller pinions with fewer teeth thus favors the use of a 25° pressure angle
  - Frictional forces and bearing loads are increased and the contact ratio decreased.

Checking for Interference

N_p : Number of teeth on spur pinion

N_G : Number of teeth on spur gear

Case 1: $m = \frac{N_G}{N_P} = 1$, the **smallest** number of N_P which can exist without interference:

$$N_P = \frac{2k}{3sin^2\phi}\left(1 + \sqrt{1 + 3sin^2\phi}\right)$$

- k=1 for full-depth teeth, k=0.8 for stub teeth and ϕ = pressure angle

Example: For a 20° pressure angle, with k=1,

$$N_P = \frac{2 \cdot 1}{3sin^2 20°}\left(1 + \sqrt{1 + 3sin^2 20°}\right) = 12.3 \approx 13\ teeth$$

Thus 13 teeth on pinion and gear are interference-free.

Case 1A: $m = \frac{N_G}{N_P} \approx 1$, the **largest** gear N_G with a specified N_P that is interference-free:

$$N_G = \frac{N_p^2 \cdot sin^2\phi - 4k^2}{4k - 2N_p sin^2\phi}$$

Under the same circumstance as the example: m=1, 20° pressure angle, N_p=13

$$N_G = \frac{13^2 \cdot sin^2 20° - 4 \cdot 1^2}{4 \cdot 1 - 2 \cdot 13 \cdot sin^2\ 20°} = 16.25 \approx 16$$

For a 13-tooth spur pinion, the maximum number of gear teeth possible without interference is 16.

Checking for Interference (Cont'd)

N_p : Number of teeth on spur pinion

N_G : Number of teeth on spur gear

Case 2: $m = \frac{N_G}{N_p} > 1$, the **smallest** number of N_P which can exist without interference:

$$N_p = \frac{2k}{(1+2m)\cdot sin^2\phi}\left(m + \sqrt{m^2 + (1+2m)sin^2\phi}\right)$$

Example: m=4, 20° pressure angle

$$N_p = \frac{2\cdot 1}{(1+2\cdot 4)\cdot sin^2 20°}\left(4 + \sqrt{m^2 + (1+2\cdot 4)sin^2 20°}\right) = 15.4 \approx 16$$

A 16-tooth pinion will mesh with a 64-tooth gear without interference

Selection Criteria of Pinion Number of Teeth (N_p)

- Generally speaking, gears with more teeth tend to run more smoothly and quietly than gears with fewer teeth.
- Number of pinion teeth should be as small as possible to keep a transmission system compact.
- Possibility of interference or undercut is great with fewer teeth.

- For enclosed gearings, N_p = 20~40 initially
- For open gearings, N_p = 17~20

Selection Criteria of Face Width (b)

- Increasing face width reduces both bending stress and contact stresses and improves surface durability.
- Normally not greater than the pitch diameter of pinion, as a wide face width increases possibility of misalignment and uneven load distribution along the gear teeth.
- The wider the face width, the more difficult it is to manufacture and mount the gears so that contact is uniform across the full face width.
- Face width, b, is not standardized, but generally,

$$\frac{9}{P} < b < \frac{14}{P} \quad or \quad 9m < b < 14m$$

- Or depending from type of bearing support:

$$b_{gear} = \varphi_d \cdot d_p$$

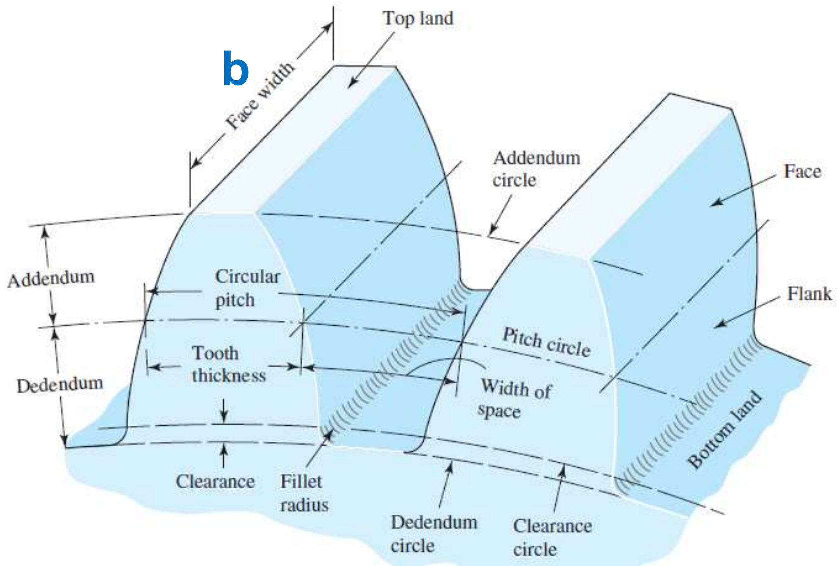
$$b_{pinion} = b_{gear} + (5\sim10)mm$$

, where

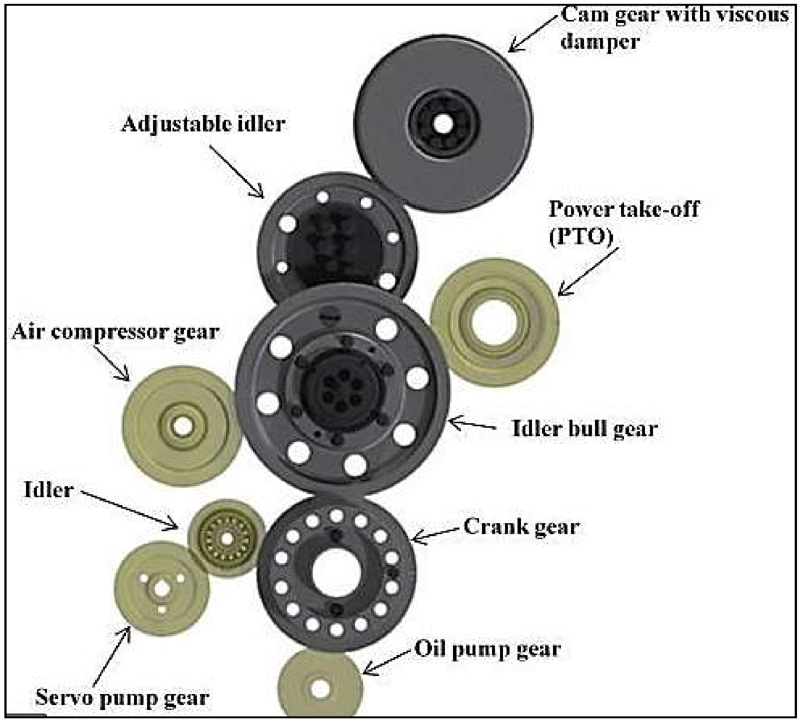
$$\varphi_d = 0.9\sim1.4 \text{ Symmetric Support}$$

$$\varphi_d = 0.7\sim1.15 \text{ Asymmetric Support}$$

$$\varphi_d = 0.4\sim0.6 \text{ Cantilever Support}$$



13-13 Gear Trains



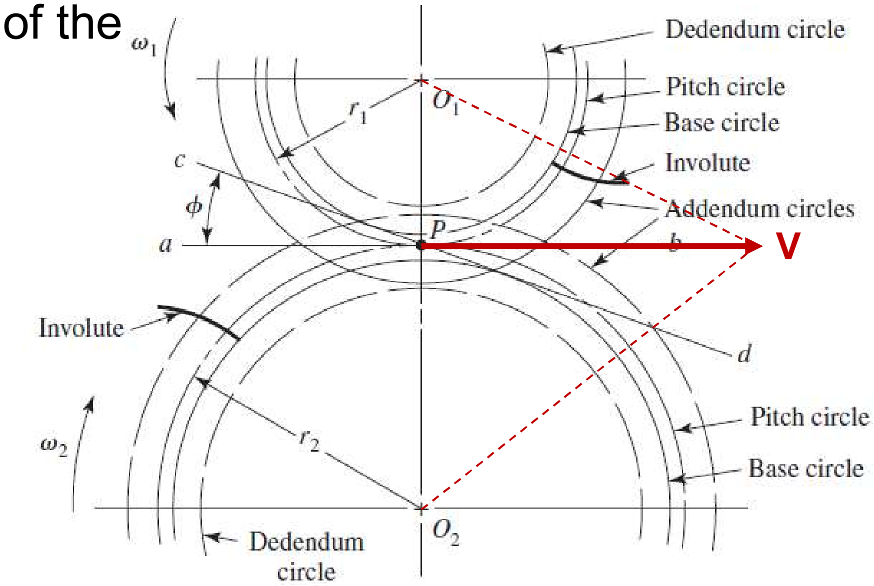
Relationship of Gear Train Angular Speed

Consider a pinion driving a gear, the speed of the driven gear is

$$\frac{n_G}{n_p} = \left|\frac{N_p}{N_G}\right| = \left|\frac{d_p}{d_G}\right|$$

$$n_G = \left|\frac{N_p}{N_G} n_p\right| = \left|\frac{d_p}{d_G} n_p\right|$$

- n: revolutions or rpm
- N: number of teeth
- d: pitch diameter



Above relationship applies to any gear set no matter whether the gears are spur, helical, bevel, or worm.

(Angular speed is inversely proportional to number of teeth and pitch diameter)

For spur and parallel helical gears, directions in the viewing plane correspond to the right-hand rule:

- positive: CCW rotation
- negative: CW rotation

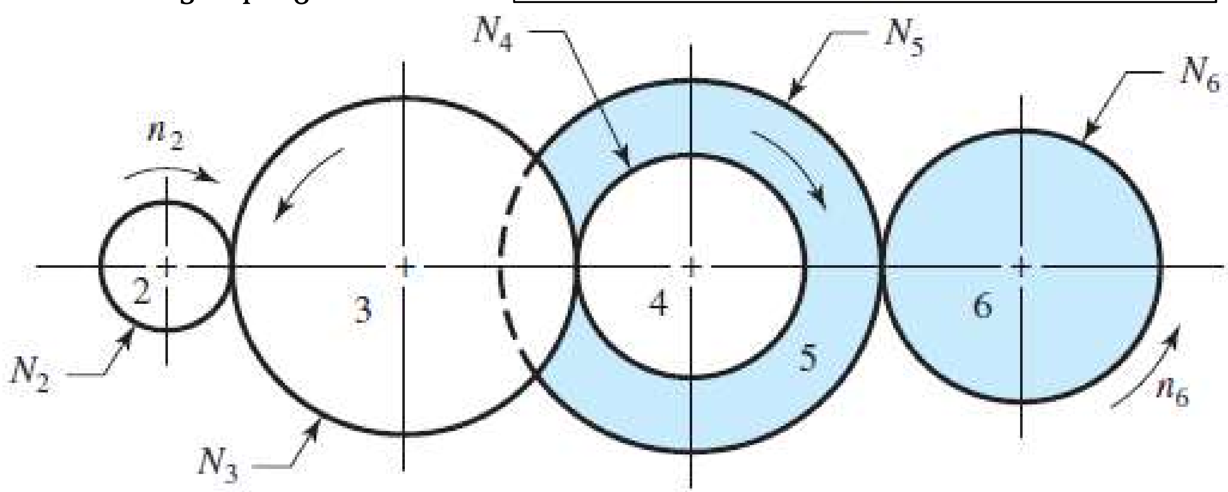
$$n_G = -\frac{N_p}{N_G} n_p = -\frac{d_p}{d_G} n_p$$

Relationship of Gear Train Angular Speed

- Consider gear 2 to be the primary driving gear.
- Gears 2, 3, and 5 are drivers, while 3, 4, and 6 are driven members
- Gear 6 speed is

$$n_6 = -\frac{N_2}{N_3}\frac{N_3}{N_4}\frac{N_5}{N_6}n_2$$

$$n_6 = n_2\left(-\frac{N_2}{N_3}\right)\left(-\frac{N_3}{N_4}\right)\left(-\frac{N_5}{N_6}\right)$$



- No $\frac{N_4}{N_5}$ in equation
- Gear 3 is an idler, that its tooth numbers cancel in equation and hence that it affects only the direction of rotation of gear 6.

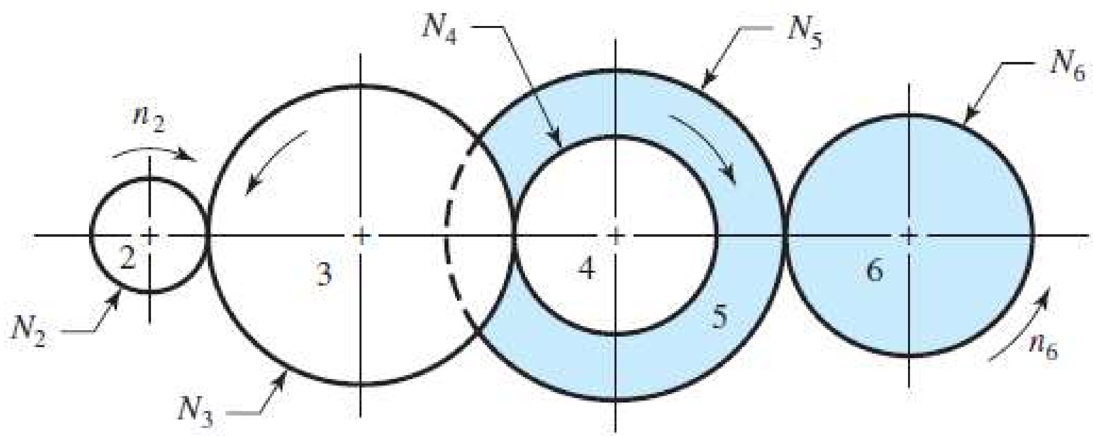
Relationship of Gear Train Angular Speed

Gear 2 to be the primary driving gear, gear 6 speed is: $n_6 = -\frac{N_2}{N_3}\frac{N_3}{N_4}\frac{N_5}{N_6}n_2$

- Gears 2, 3, and 5 are drivers, while 3, 4, and 6 are driven members
- Define Train Value: $e = \frac{\text{product of driving tooth numbers}}{\text{product of driven tooth numbers}} = \frac{N_2}{N_3}\frac{N_3}{N_4}\frac{N_5}{N_6}$
  - e is positive if the last gear rotates in the same sense as the first,
  - e is negative if the last rotates in the opposite sense

$$n_L = e \cdot n_F$$

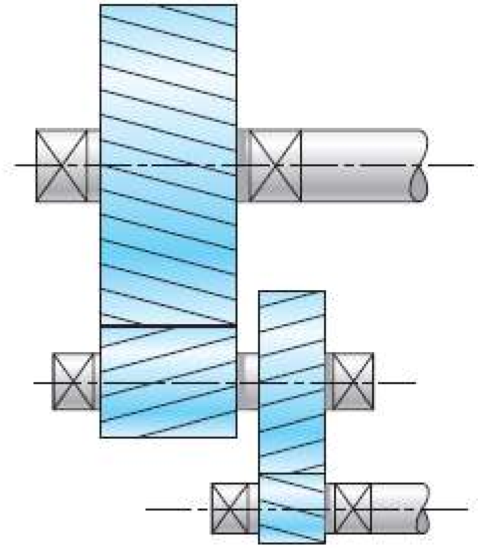
n_L is the speed of the last gear in the train and n_F is the speed of the first.



- A train value of up to **10** can be obtained with one pair of gears.
- Greater ratios can be obtained in less space and with fewer dynamic problems by compounding additional pairs of gears.

Example 13-3

A gearbox is needed to provide a 30:1 (±1 percent) increase in speed, while minimizing the overall gearbox size. Specify appropriate tooth numbers.



Since the ratio is greater than 10:1, but less than 100:1, a two-stage compound gear train is needed. The portion to be accomplished in each stage is $\sqrt{30} = 5.4772$.

For this ratio, assuming a typical 20° pressure angle, the minimum number of teeth to avoid interference is

$m = 5.4772$ and $k = 1$

$$N_p = \frac{2k}{(1+2m)\cdot sin^2\phi}\left(m + \sqrt{m^2 + (1+2m)sin^2\phi}\right) = 15.947 \approx 16$$

The number of teeth necessary for the mating gears is:

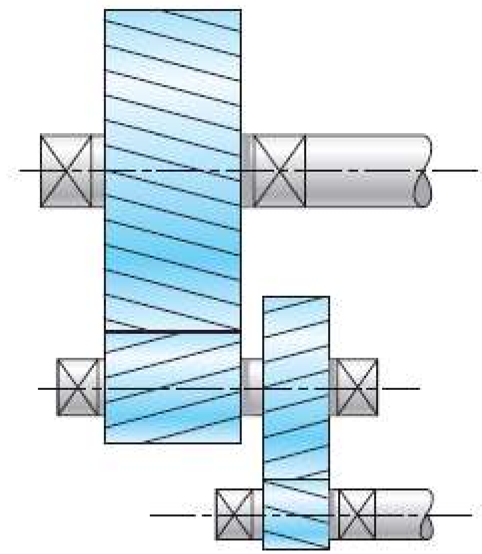
$16\sqrt{30} = 87.64 \approx 88$

The overall train value: $e = \frac{88}{16}\frac{88}{16} = 30.25$

, which is within 1 percent tolerance.

Example 13-4

A gearbox is needed to provide an exact 30:1 increase in speed, while minimizing the overall gearbox size. Specify appropriate teeth numbers.



In order to obtain integers, factor the overall ratio into two integer stages: $e = 30 = 6 \cdot 5$

$$\frac{N_2}{N_3} = 6, \qquad \frac{N_4}{N_5} = 5, \qquad e = \frac{N_2}{N_3}\frac{N_4}{N_5}$$

Choose N_3 and N_5 to be as small as possible without interference. Assuming a 20° pressure angle:

$m = 5,\ k = 1$

$$N_p = \frac{2k}{(1+2m)\cdot sin^2\phi}\left(m + \sqrt{m^2 + (1+2m)sin^2\phi}\right) = 15.74 \approx 16$$

Eq. (13–11) gives the minimum as 16.

The number of teeth necessary for the mating gears is:

$N_2 = 6N_3 = 6 \cdot 16 = 96$

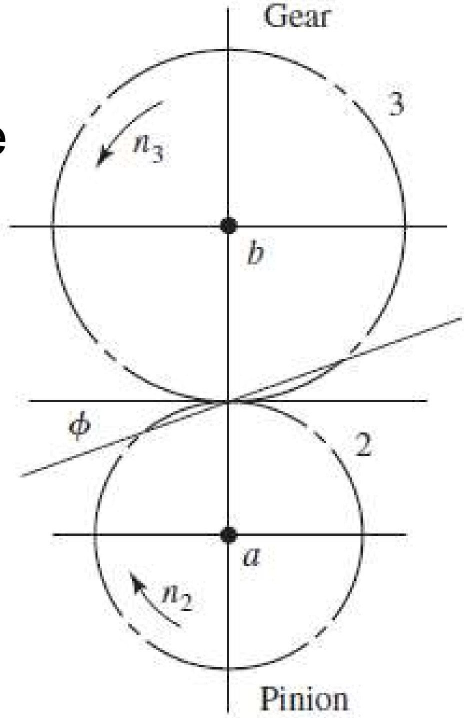
$N_4 = 5N_5 = 5 \cdot 16 = 80$

13-14 Force Analysis - Spur Gearing

Force Analysis Notation

- Numeral **1** for the frame of the machine
- Input gear as gear **2**, and then number the gears successively 3, 4, etc., until we arrive at the last gear in the train
- Designate the **shafts**, using lowercase letters of the alphabet, a, b, c, etc.
- Use superscripts to indicate directions
  - **x**, **y**, and **z** coordinates, and
  - **r** : radial direction
  - **t** : tangential direction

- Example F_{43}^{t}: tangential component of the force of gear 4 acting against gear 3.



(a)

Force Analysis of a Simple Gear Train

- A pinion mounted on shaft a rotating CW at n_2 rpm and driving a gear on shaft b at n_3 rpm
- Free-Body Diagram (FBD) of pinion
  - F_{a2} and T_{a2} are the force and torque, respectively, exerted by shaft **a** against pinion **2**.
  - F_{32} is the force exerted by gear **3** against the pinion **2**
- Similarly for FBD of gear

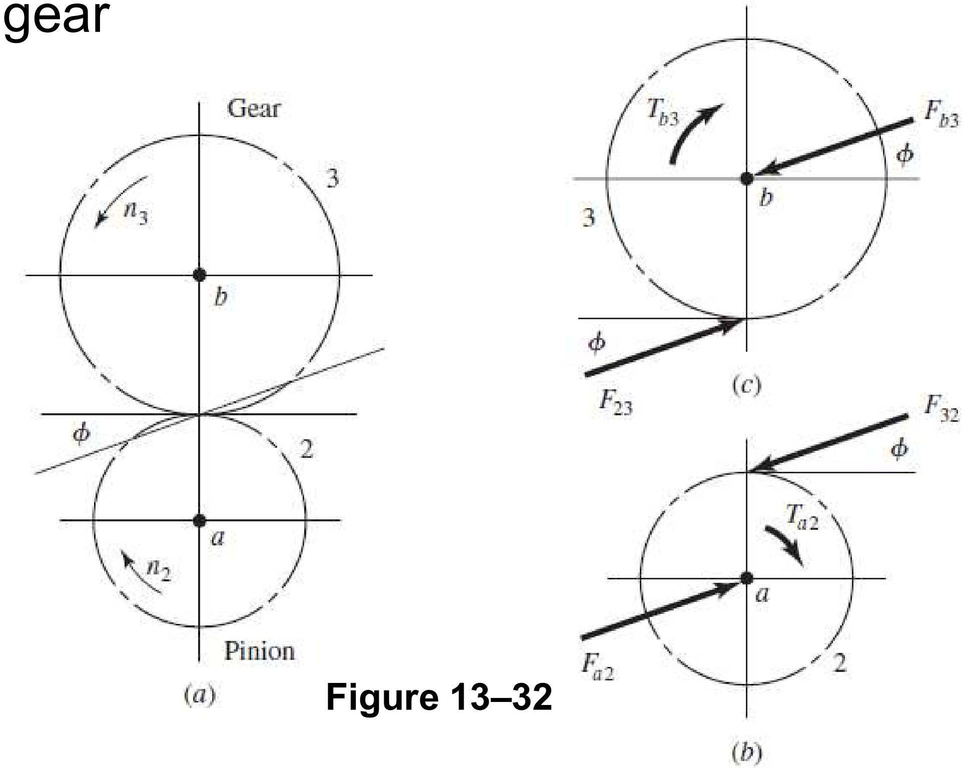


Figure 13–32

Force Analysis of a Simple Gear Train

Resolution of pinion gear forces

- Transmitted Load $W_t = F_{32}^t$
- Applied torque $T = \frac{d}{2} W_t$

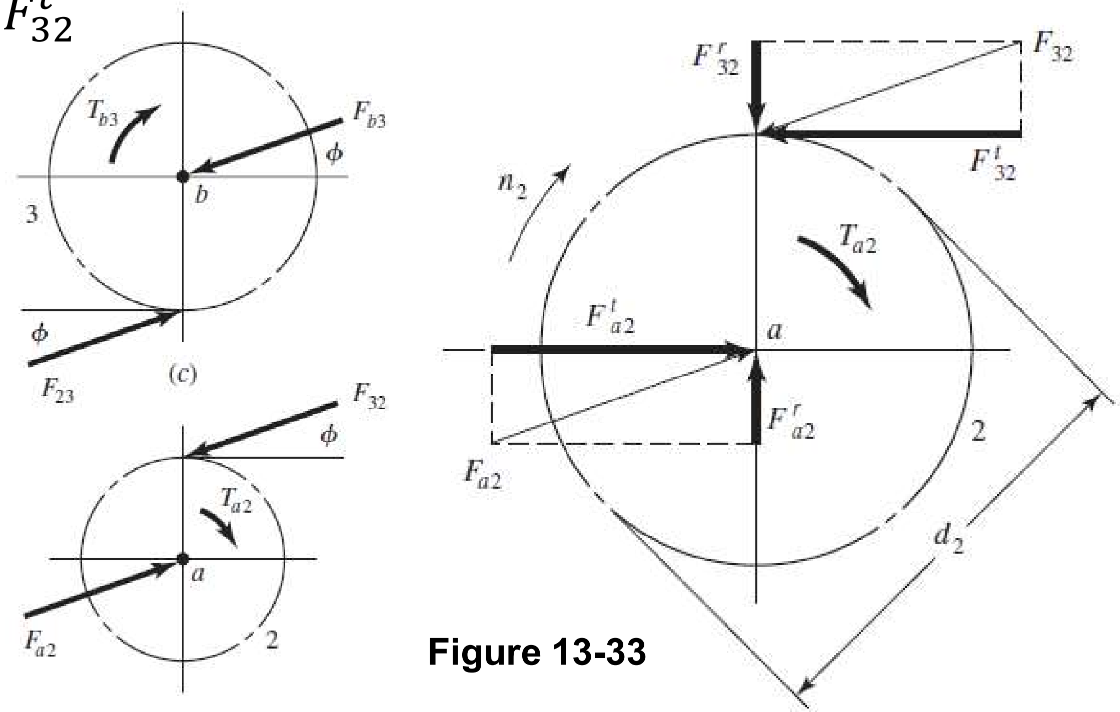


Figure 13-33

- Transmitted power $H = T\omega = \frac{W_t \cdot d}{2}\omega$
  - If meshed gears are reasonably efficient, the power is generally treated as constant through the mesh

Force Analysis of a Simple Gear Train

Pitch-Line Velocity: linear velocity of a point on the gear at the radius of the pitch circle

US Customary Unit

$$V = \frac{\pi d n}{12}$$

$$W_t = 33000\frac{H}{V}$$

- V = Pitch-Line Velocity (ft/min)
- W_t = transmitted load (lbf)
- H = power (hp)
- d = gear diameter (in)
- n = gear speed (rpm)

$$\left(1\ hp = 550\frac{ft \cdot lbf}{sec}\right)$$

SI Unit

$$V = \pi d n$$

$$W_t = 60000\frac{H}{V}$$

- V = Pitch-Line Velocity (mm/min)
- W_t = transmitted load, kN
- H = power (kW)
- d = gear diameter (mm)
- n = speed (rpm)

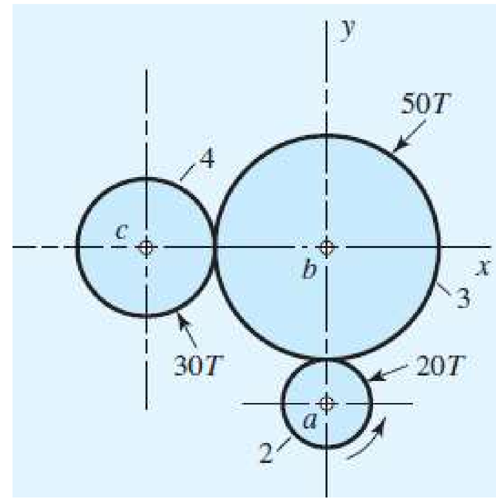
$$\left(1\ Watt = 1\frac{N \cdot m}{sec}\right)$$

$$(1\ hp = 745.7\ Watts\ )$$

$$H = Torque \cdot \omega$$

Example 13-7

Pinion 2 runs at 1750 rev/min and transmits 2.5 kW to idler gear 3. The teeth are cut on the 20° full-depth system and have a module of m=2.5 mm. Draw a free-body diagram of gear 3 and show all the forces that act upon it.



The pitch diameters of gears 2 and 3 are

$d_2 = N_2 m = 20 \cdot 2.5 = 50\, mm$

$d_3 = N_3 m = 50 \cdot 2.5 = 125\, mm$

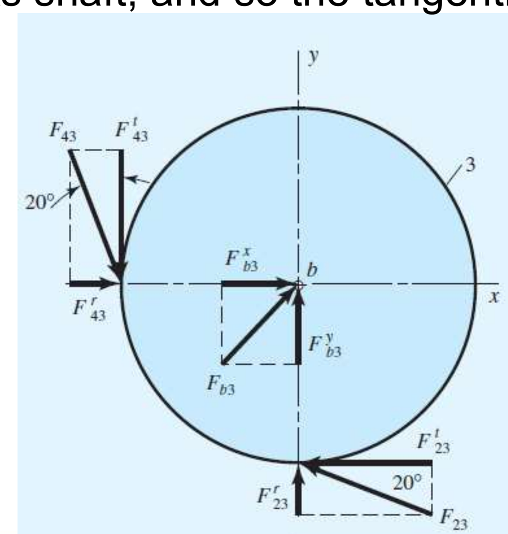
The transmitted load is

$$W_t = 60000 \frac{H}{V} = 60000 \frac{2.5}{\pi \cdot 50 \cdot 1750} = 0.546\, kN = F_{32}^t$$

$$F_{32}^r = F_{32}^t \tan 20° = 0.199\, kN; \quad F_{32} = \frac{F_{32}^t}{\cos 20°} = 0.581\, kN$$

Since gear 3 is an idler, it transmits no power (torque) to its shaft, and so the tangential reaction of gear 4 on gear 3 is also equal to W_t. Therefore

$F_{43}^t = 0.546\, kN;\ \ F_{43}^r = 0.199\, kN;\ \ F_{43} = 0.581\, kN$



The shaft reactions in the *x* and *y* directions are

$$F_{b3}^x = -(F_{23}^t + F_{43}^r) = -(-0.546 + 0.199) = 0.347 \text{ kN}$$

$$F_{b3}^y = -(F_{23}^r + F_{43}^t) = -(0.199 - 0.546) = 0.347 \text{ kN}$$