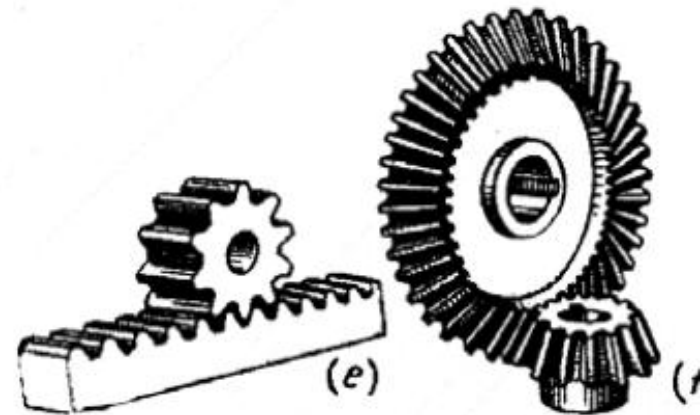
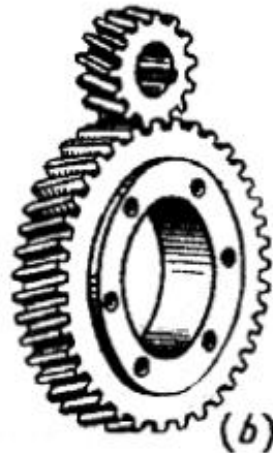


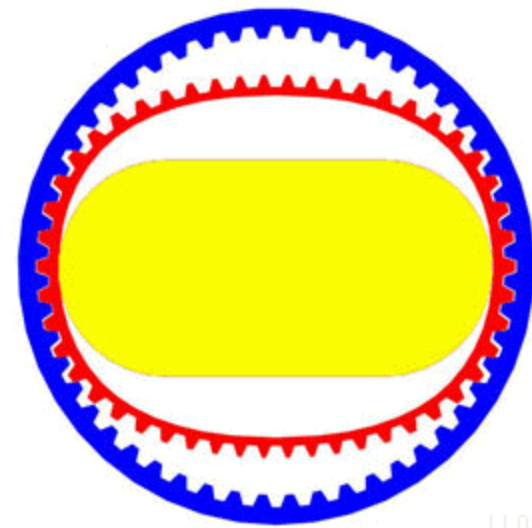
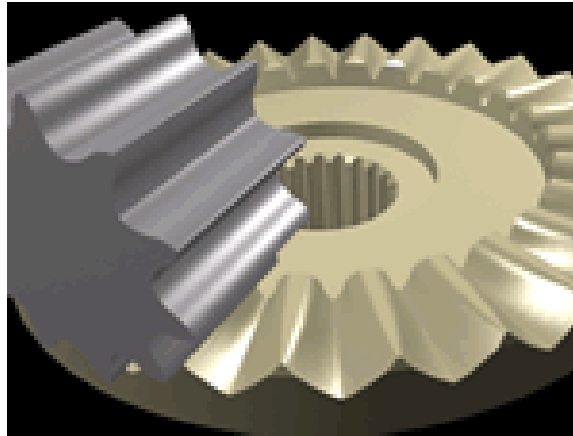
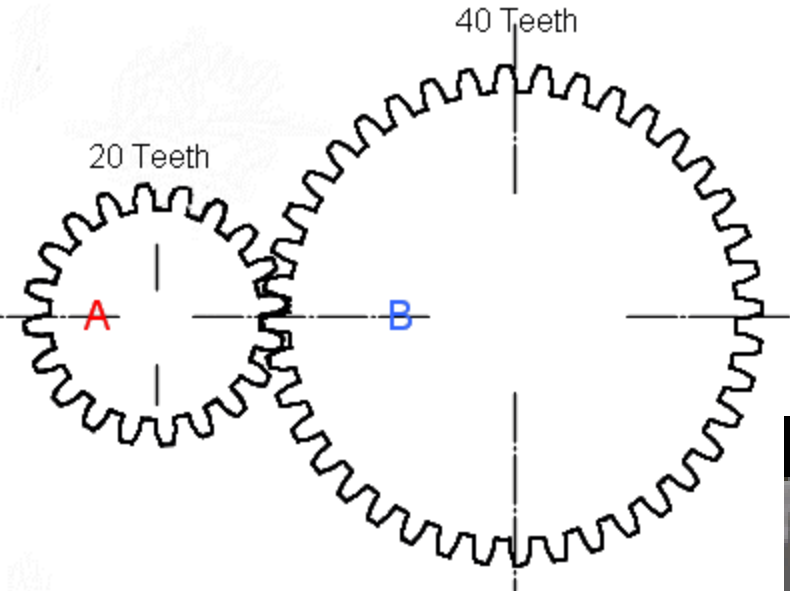
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MACHINE ELEMENTS II

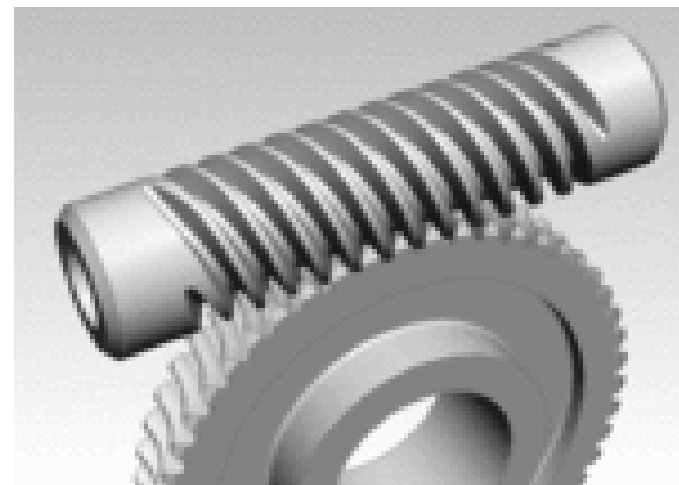
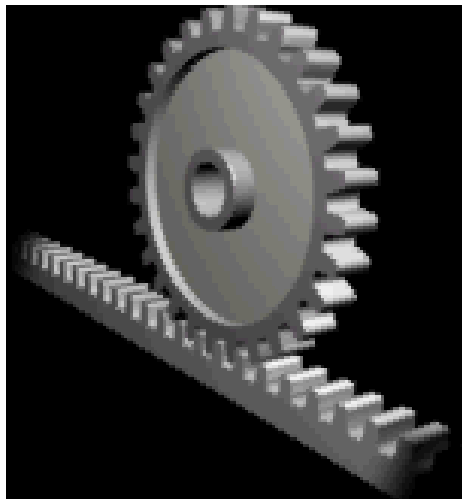
CHAPTER 5

GEARS_part1





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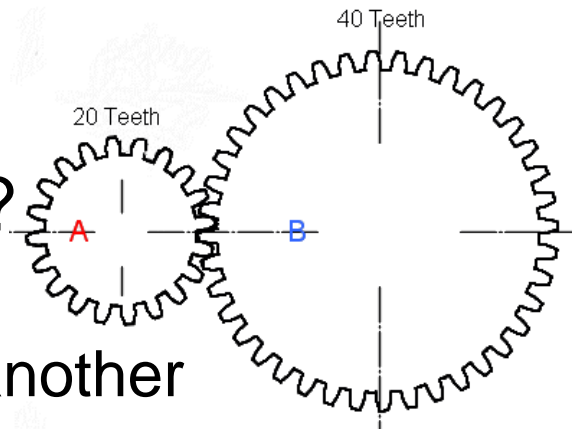
Why do we need to use gears?

- Think of the gears around you.
 - Gears in car transmissions
 - Gears in toys
 - Gears in turning machines (lathes)
 - Gears in watches
 - Gears in kitchen equipments (mixers etc)



- What are the main functions of gears?

- ?
- ? 1. Transmit motion from one shaft to another
- ? 2. Transmit motion by reducing rotational speed
- 3. Transmit torque from one shaft to another
- 4. Transmit motion by increasing the torque
- 5. Transmit ...?????.....(motion*torque=power)..



Gears are versatile mechanical components capable of performing many different kinds of power transmission or motion control.

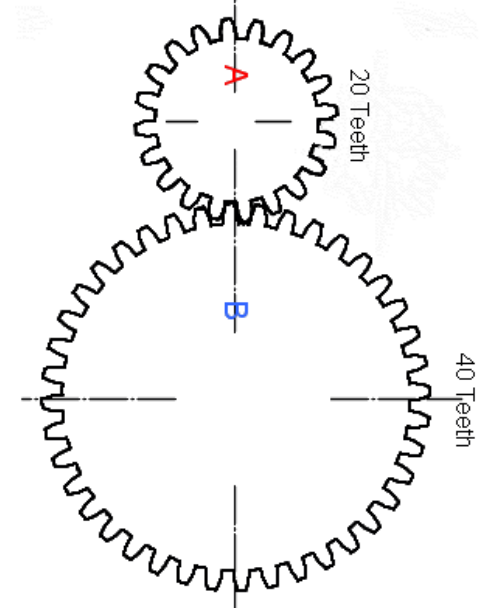
Examples of these are

- Changing rotational speed.
- Changing rotational direction.
- Changing the angular orientation of rotational motion.
- Multiplication or division of torque or magnitude of rotation.
- Converting rotational to linear motion and its reverse.
- Offsetting or changing the location of rotating motion.

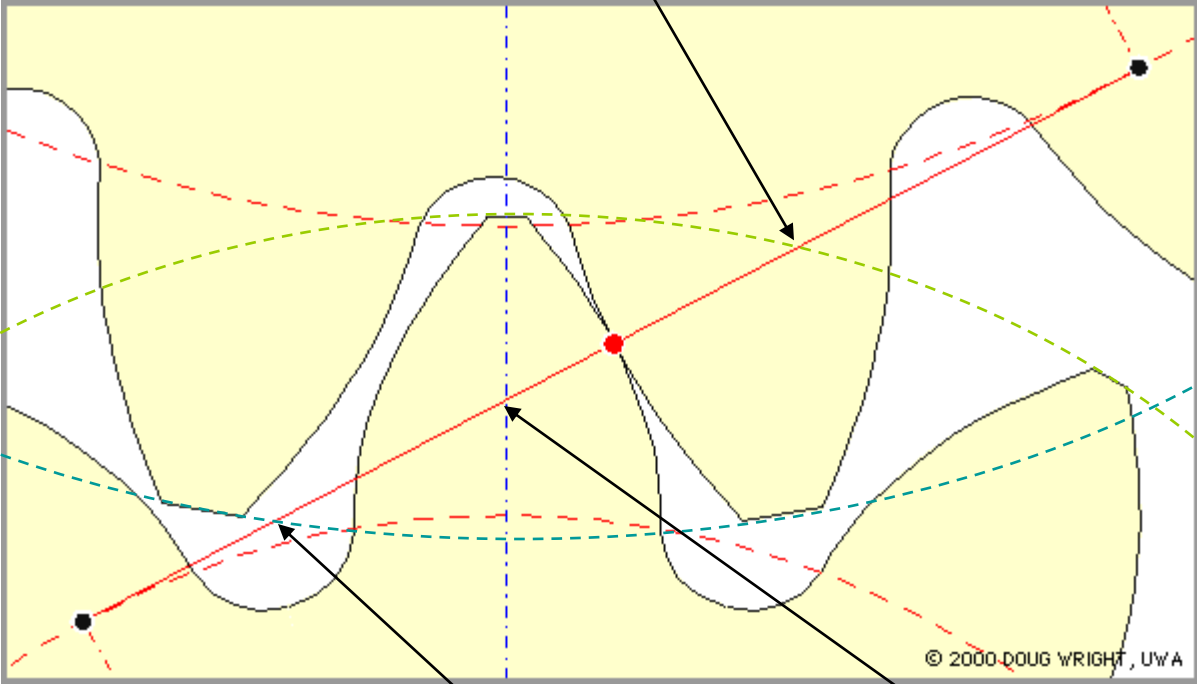
A gear is a wheel with teeth that mesh together with other gears.

Gears change the

- speed
- torque (rot. force)
- direction of rotating axes.



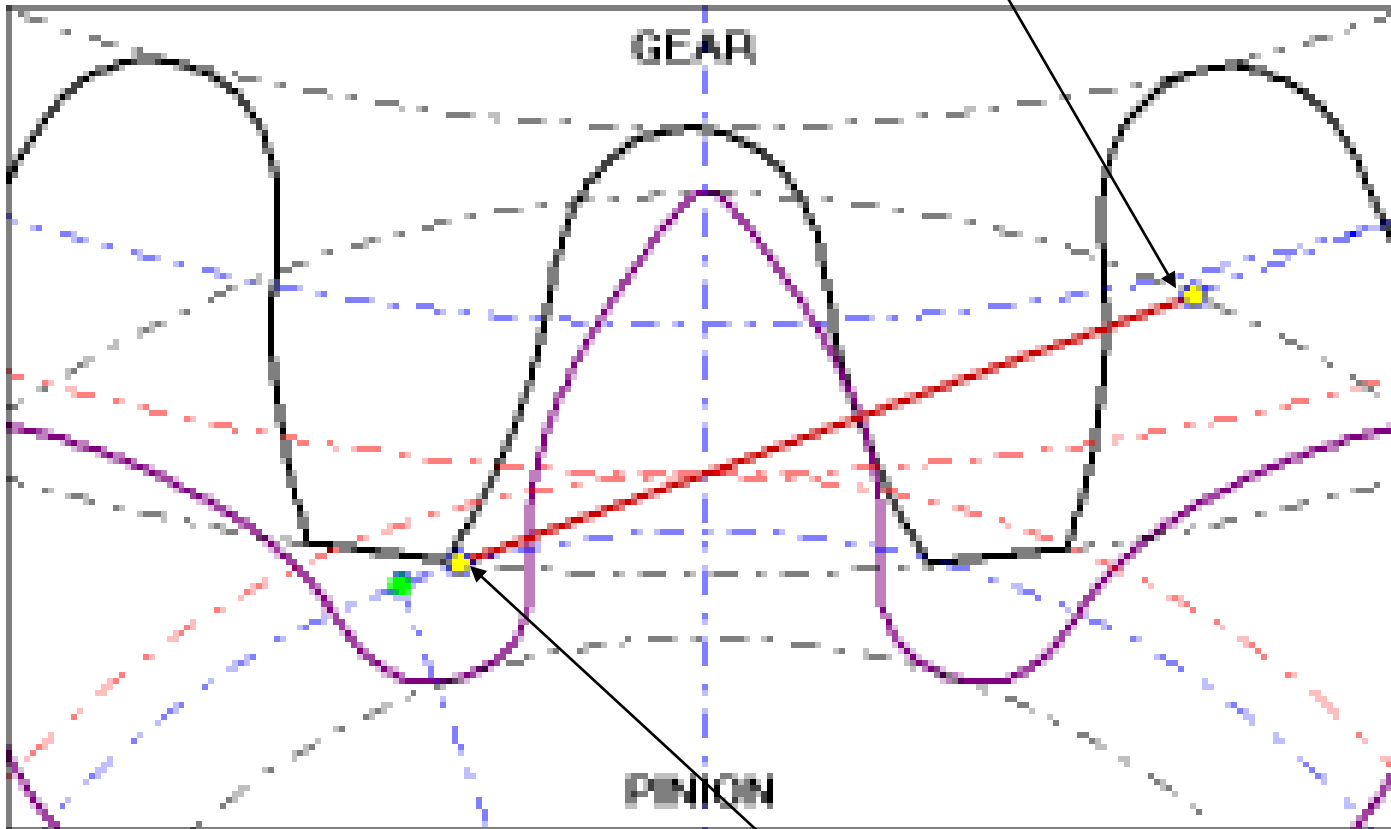
End point of mesh
(end of contact)



Pitch point

Start point of mesh
(start of contact)

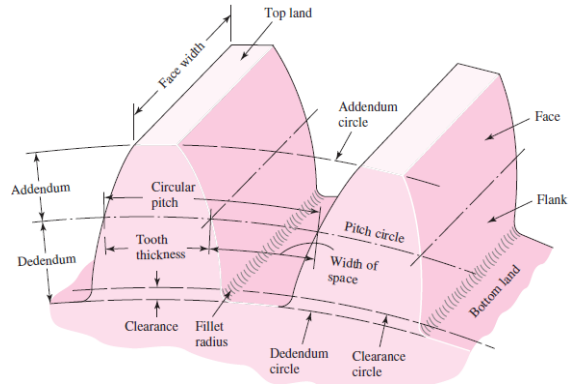
End point of mesh
(end of contact)



Start point of mesh
(start of contact)

This chapter addresses

- gear geometry,
- the kinematic relations, and
- the forces transmitted by the gears (spur, helical, bevel, and worm).



The forces transmitted between meshing gears supply torsional moments to shafts for motion and power transmission and create forces and moments that affect the shaft and its bearings.

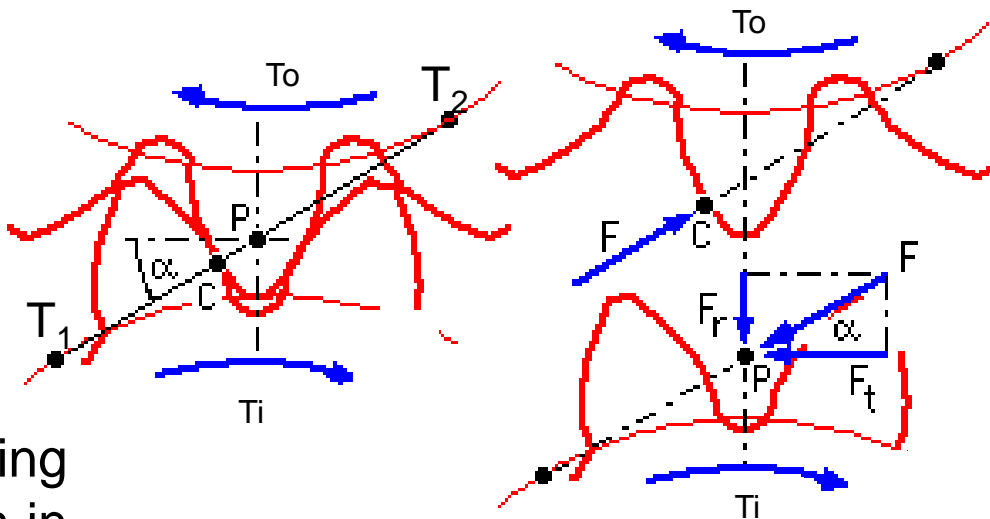
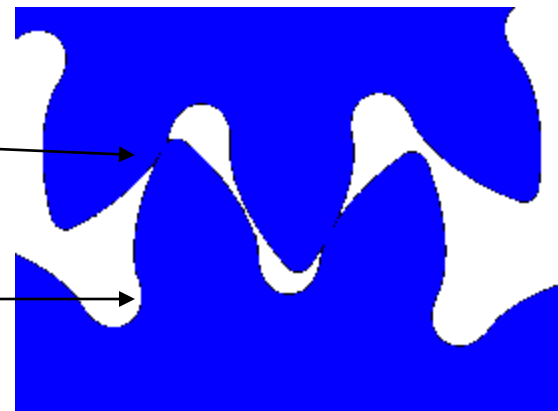


Fig. 3 Tooth forces

The forces transmitted between meshing gears do also effect the teeth of gears in two respects:

- teeth will be in contact under forces creating contact stress at teeth surfaces
- teeth will be bend by the forces creating bending stress at tooth root and



5.1 TYPES OF GEARS

Gears are usually cylindrical sometimes conical disks with some teeth on the circumference to ensure a continuous and positive transmission of rotary motion between the shafts on which they are mounted.

Some examples are:

- Spur gears - for parallel shaft
- Helical gears
- Bevel gears } these 3 can accommodate non-parallel shafts.
- Worm gears

Spur gears especially are used to transmit rotary motion between parallel shafts.

5.1 TYPES OF GEARS

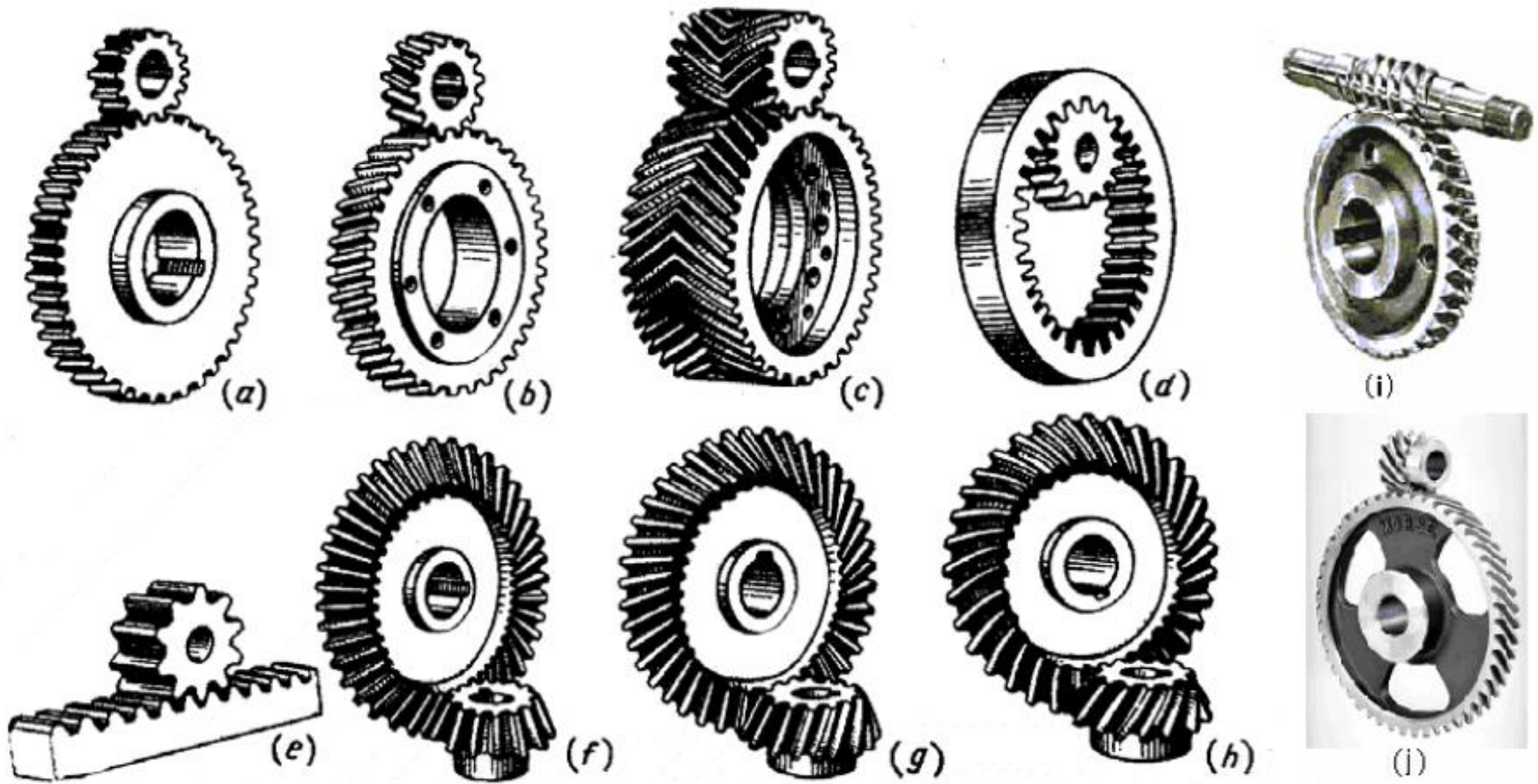


Fig. 5.1 Types of Gears (a) Spur gear, (b) Helical gear, (c) Double helical gear or herringbone gear, (d) Internal gear , (e) Rack and pinion, (f) Straight bevel gear, (g) Spiral bevel gear, (h) Hypoid bevel gear , (i) worm gear and (j) Crossed helical or spiral gear.

5.1.1 SPUR GEARS

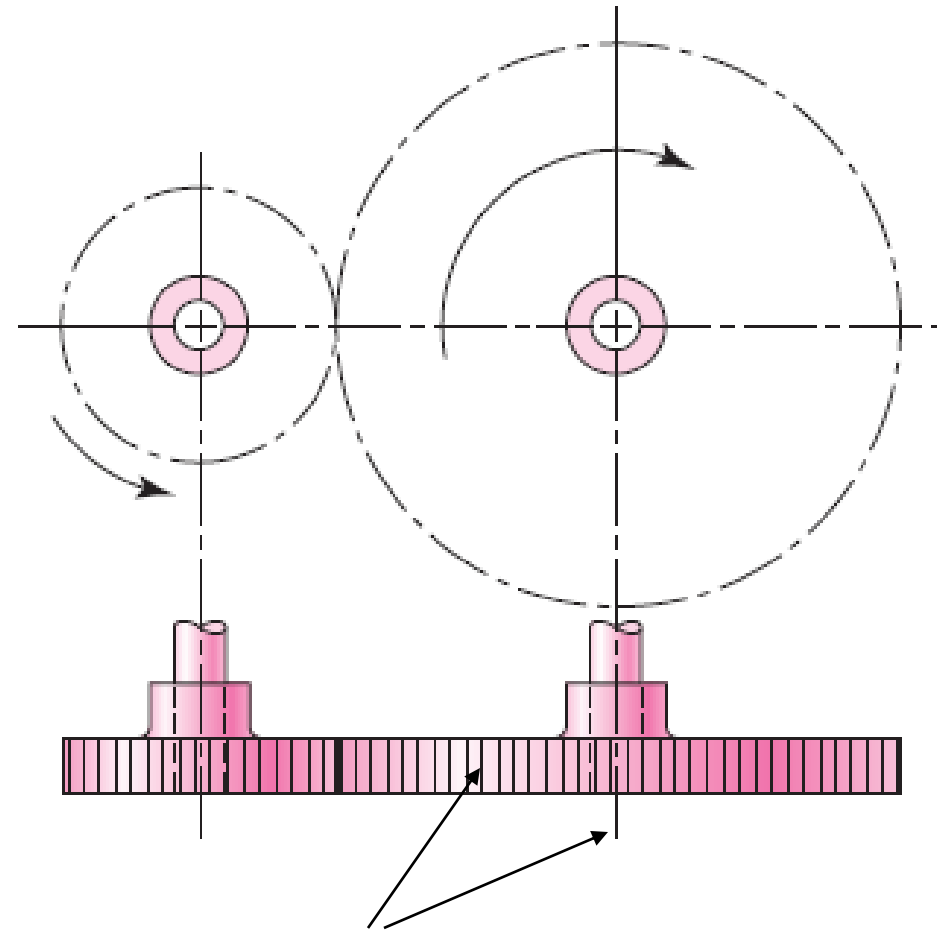


Fig.5.2 Spur gears are used to transmit rotary motion between parallel shafts

Spur gears, illustrated in Fig. 5.2, 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 and, for this reason, will be used to develop the primary kinematic relationships of the tooth form.

5.1.2 HELICAL GEARS

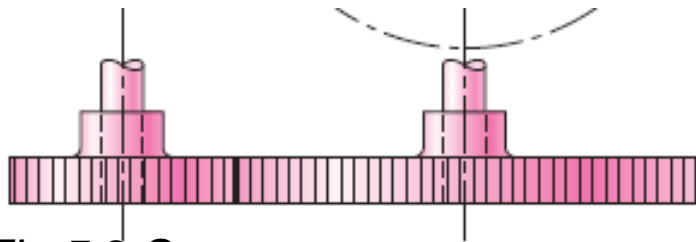


Fig.5.2 Spur gears

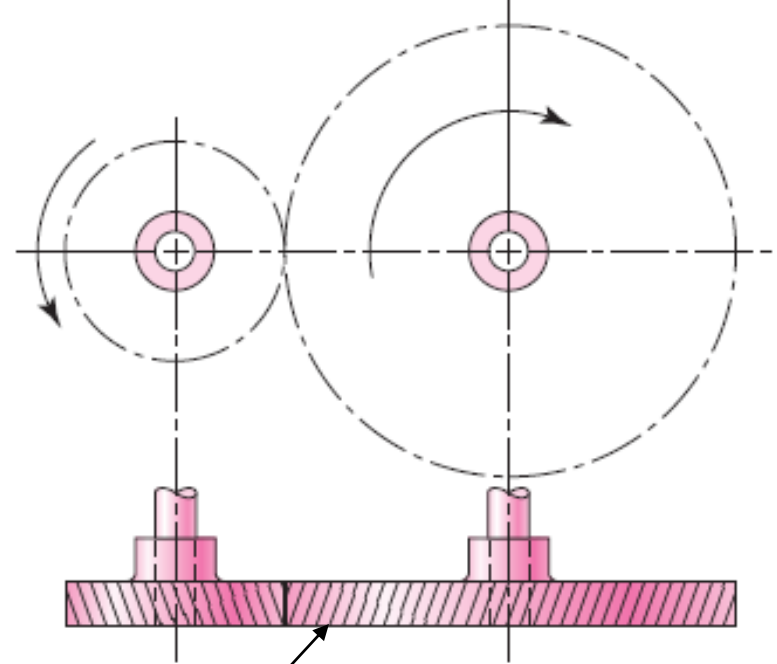


Fig.5.3 Helical gears are used to transmit motion between parallel or nonparallel shafts.

Helical gears, shown in Fig. 5.3, have teeth inclined to the axis of rotation. Helical gears can be used for the same applications as spur gears and, when so used, are not as noisy, because of the more gradual engagement of the teeth during meshing.

The inclined tooth also develops thrust loads and bending couples, which are not present with spur gearing.

5.1.3 BEVEL GEARS

*Bevel gears, shown in Fig. 5.4, have teeth formed on conical surfaces and are used mostly for transmitting motion between intersecting shafts. The figure actually illustrates *straight-tooth bevel gears*.*

Spiral bevel gears are cut so the tooth is no longer straight, but forms a circular arc.

Hypoid gears are quite similar to spiral bevel gears except that the shafts are offset and nonintersecting.

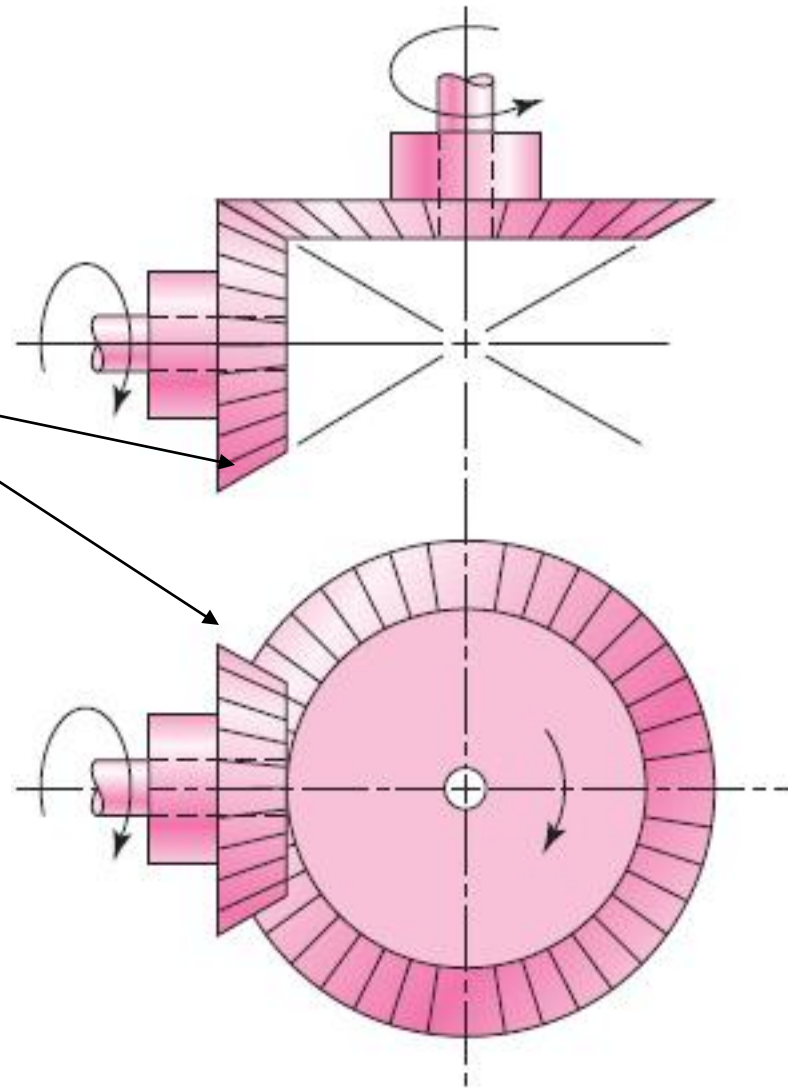
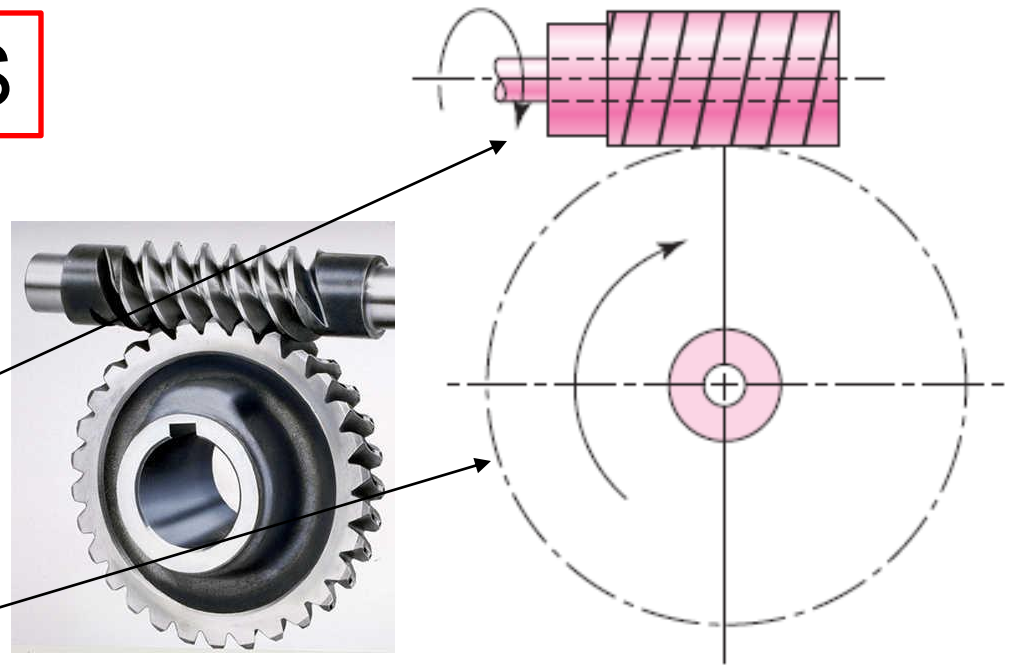


Fig.5.4 Bevel gears are used to transmit rotary motion between intersecting shafts.

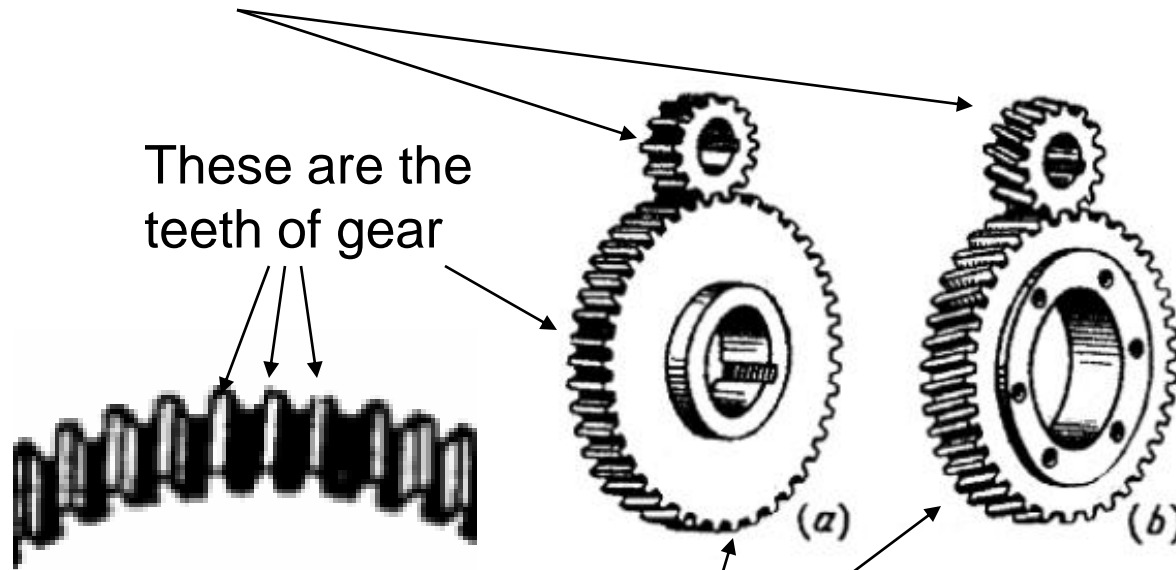
5.1.4 WORM GEARS

Fig. 5.5 Worm gear sets are used to transmit rotary motion between nonparallel and nonintersecting shafts.



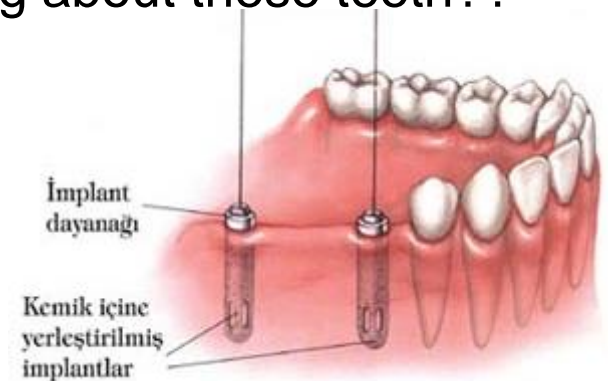
Worms and worm gears, shown in Fig. 5.5, represent the fourth basic gear type. As shown, the worm resembles a screw. The direction of rotation of the worm gear, also called the worm wheel, depends upon the direction of rotation of the worm and upon whether the worm teeth are cut right-hand or left-hand. Worm-gear sets are also made so that the teeth of one or both wrap partly around the other. Such sets are called *single enveloping* and *double-enveloping worm-gear sets*. *Worm-gear sets are mostly used* when the speed ratios of the two shafts are quite high, say, 3 or more.

A pinion is the smaller of two mating gears.

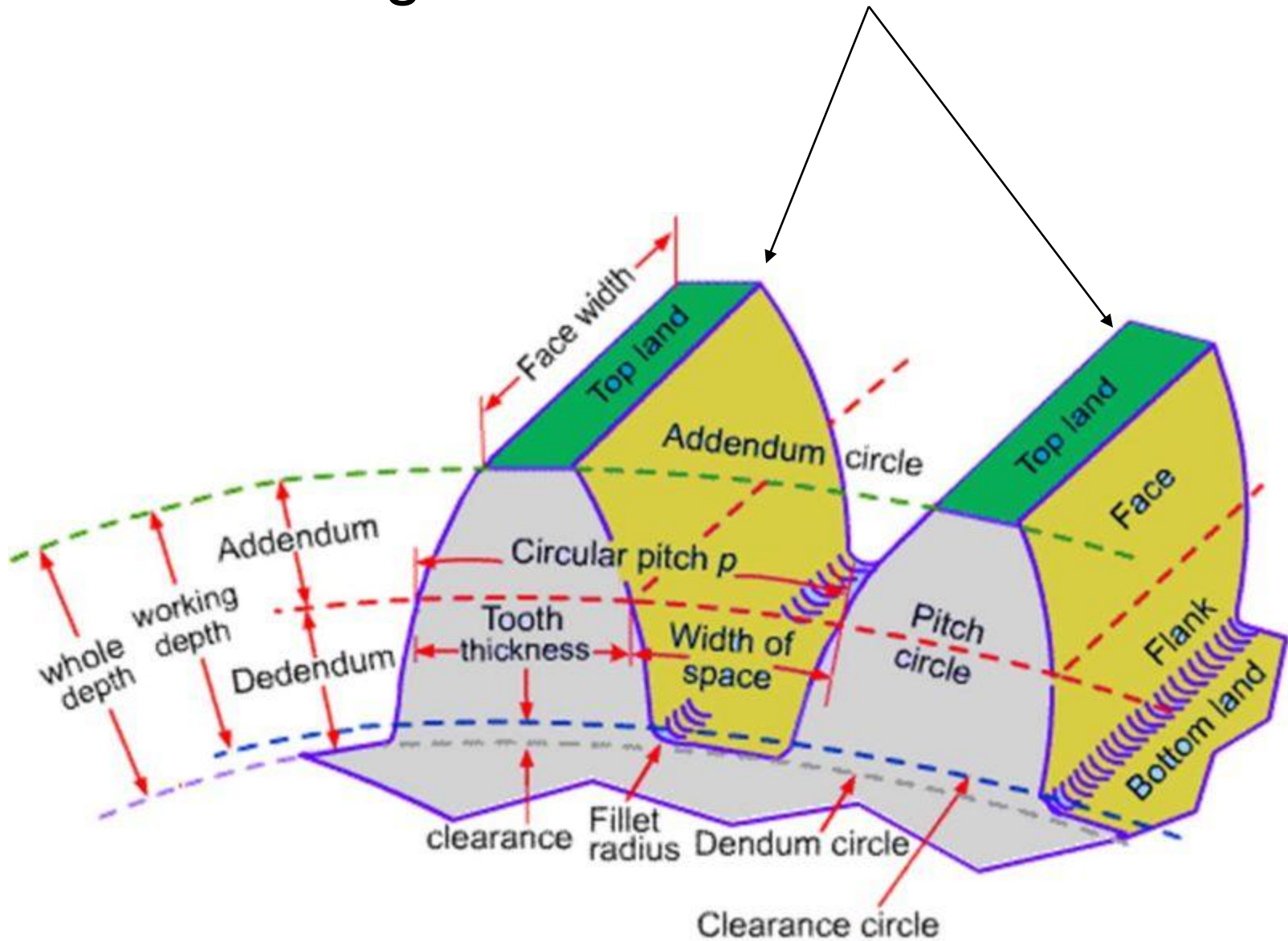


The larger is often called the gear

But we are not talking about these teeth??



We are talking about those teeth



5.2 NOMENCLATURE OF SPUR GEARS

The terminology of spur-gear teeth is illustrated in Fig. 5.6. The pitch circle is a theoretical circle upon which all calculations are usually based; its diameter is the pitch diameter. The pitch circles of a pair of mating gears are tangent to each other at pitch point.

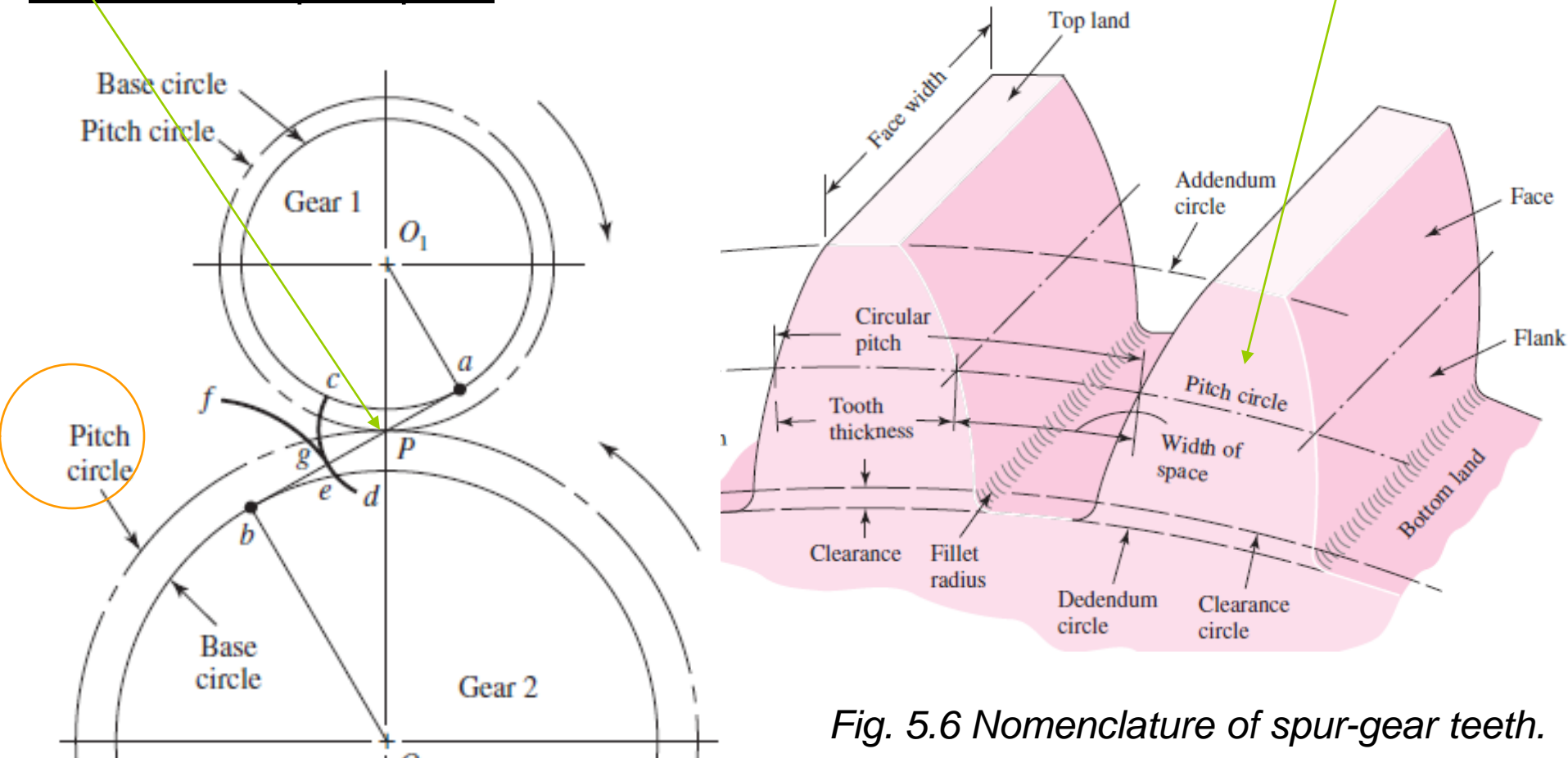
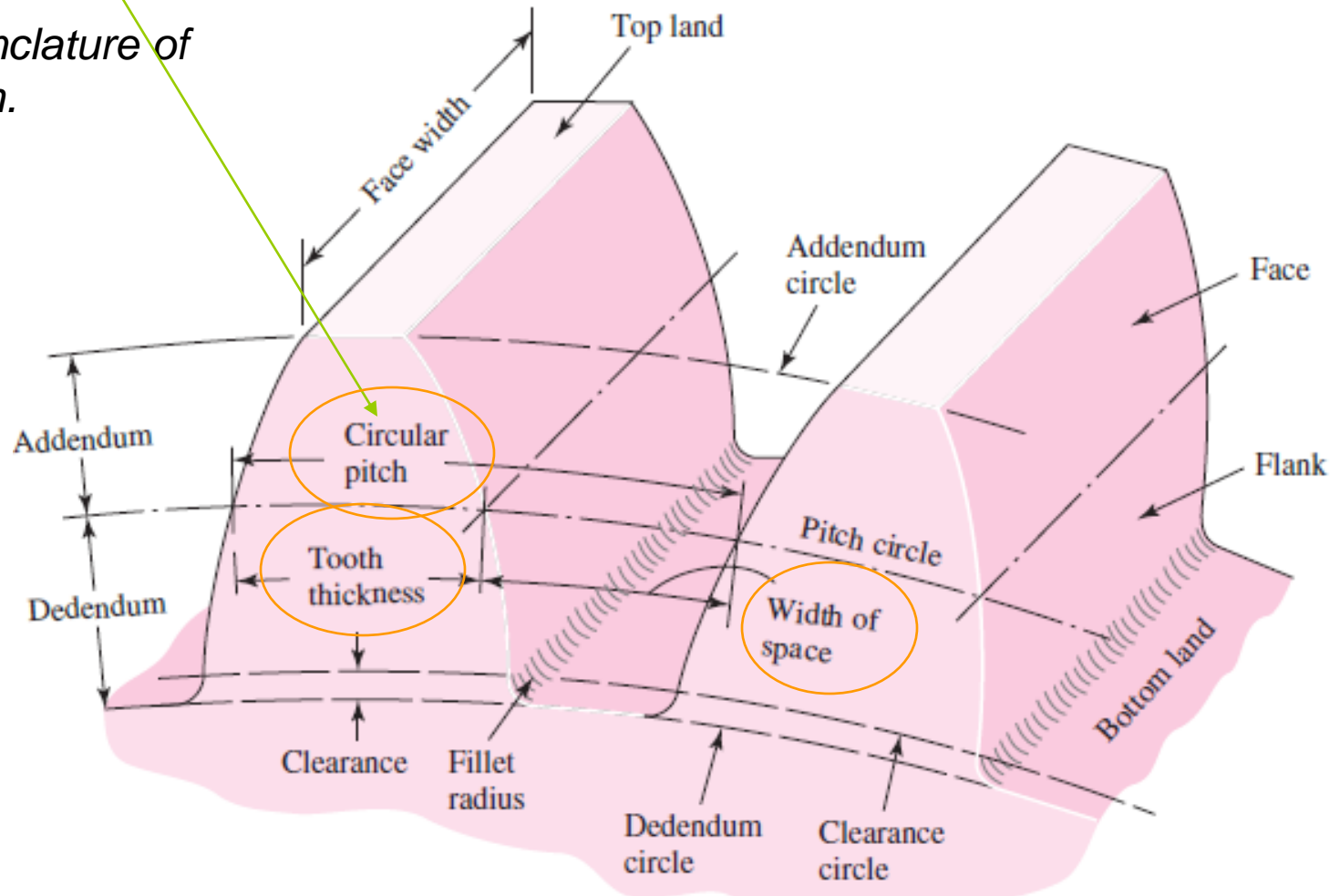


Fig. 5.6 Nomenclature of spur-gear teeth.

5.2 NOMENCLATURE

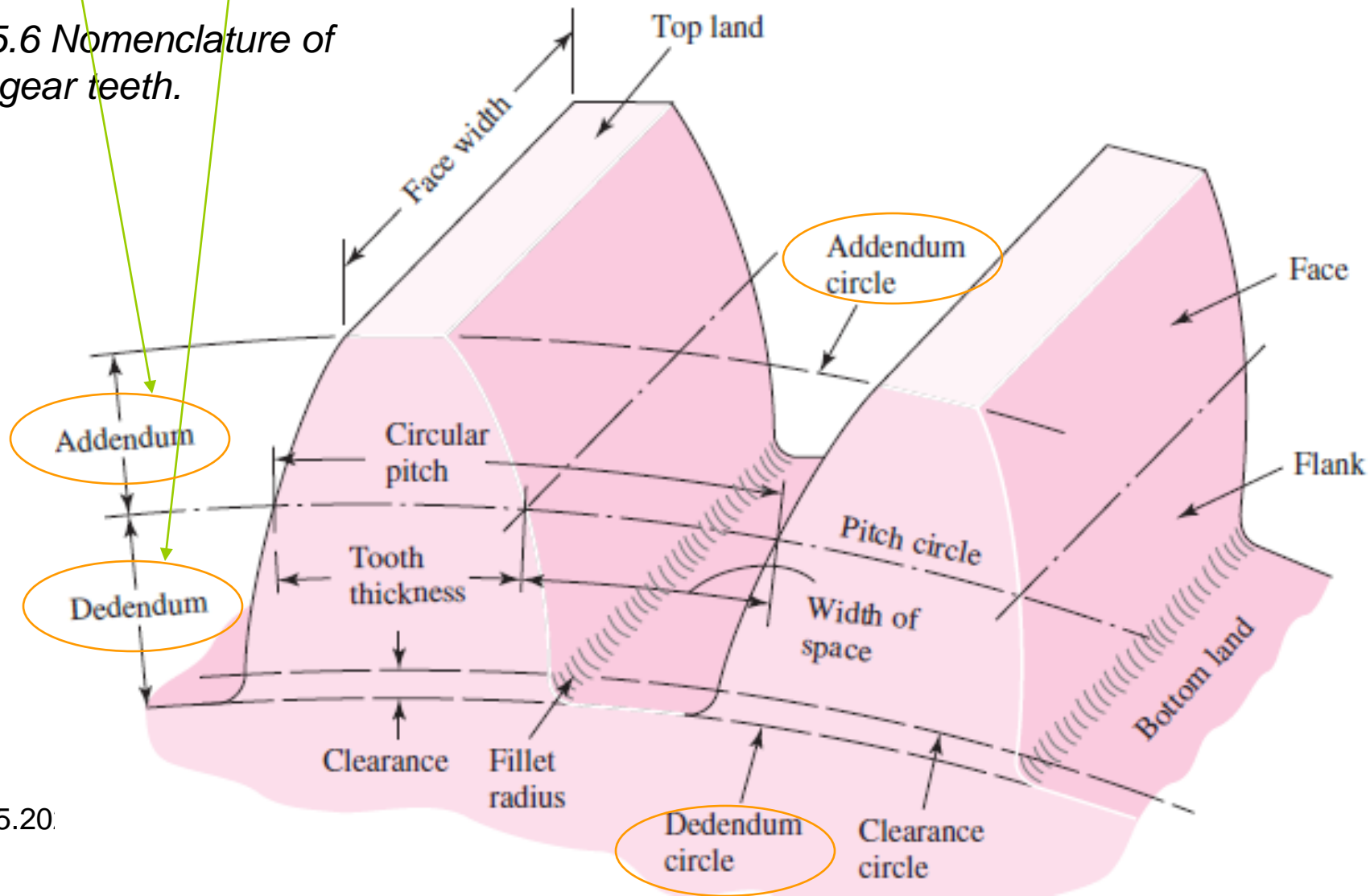
The circular pitch p is the distance, measured on the pitch circle, from a point on one tooth to a corresponding point on an adjacent tooth. Thus the circular pitch is equal to the sum of the tooth thickness and the width of space.

Fig. 5.6 Nomenclature of spur-gear teeth.



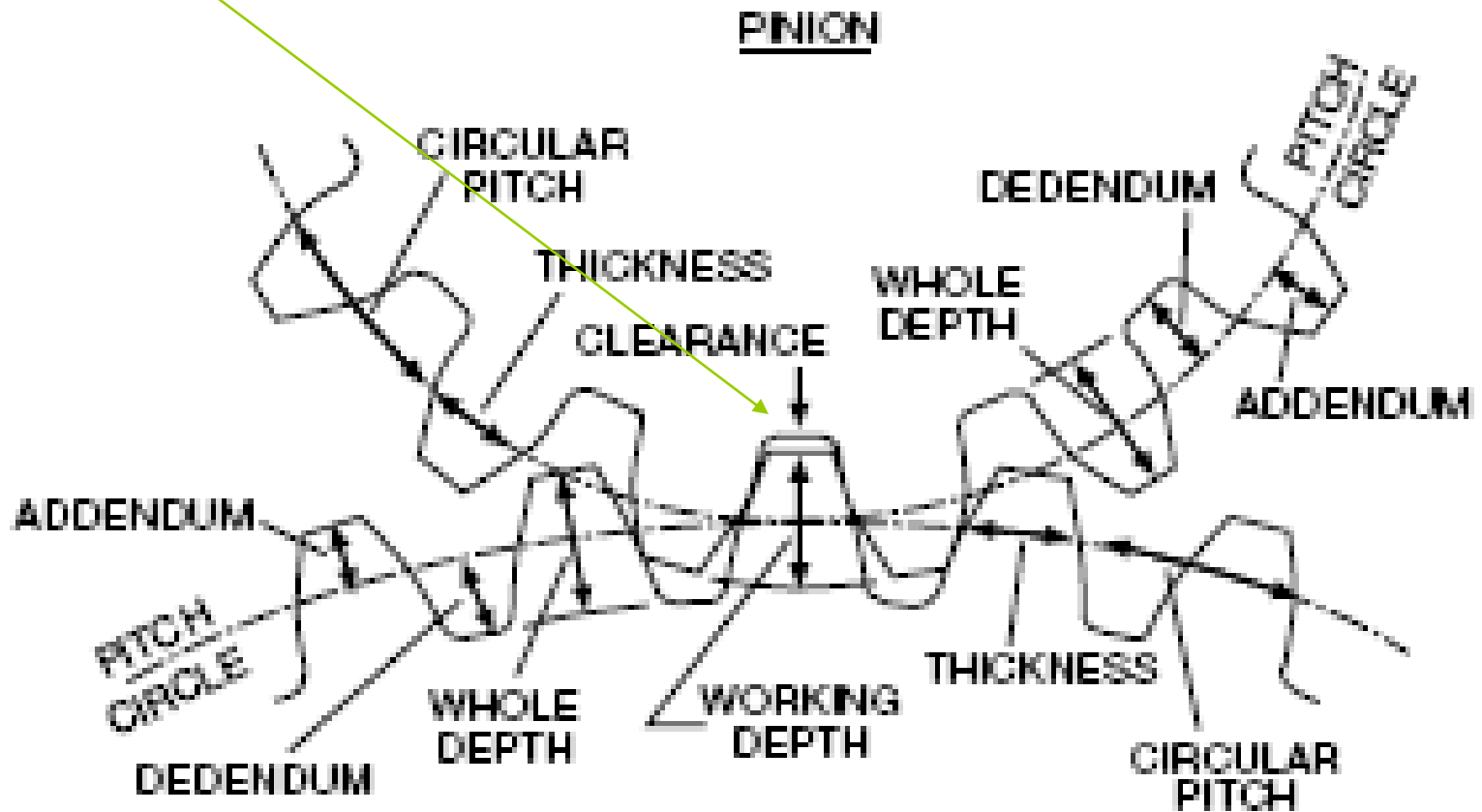
The addendum a is the radial distance between the *top land* and the *pitch circle*. The dedendum b is the radial distance from the *bottom land* to the pitch circle. The *whole depth* h_t is the sum of the addendum and the dedendum.

Fig. 5.6 Nomenclature of spur-gear teeth.

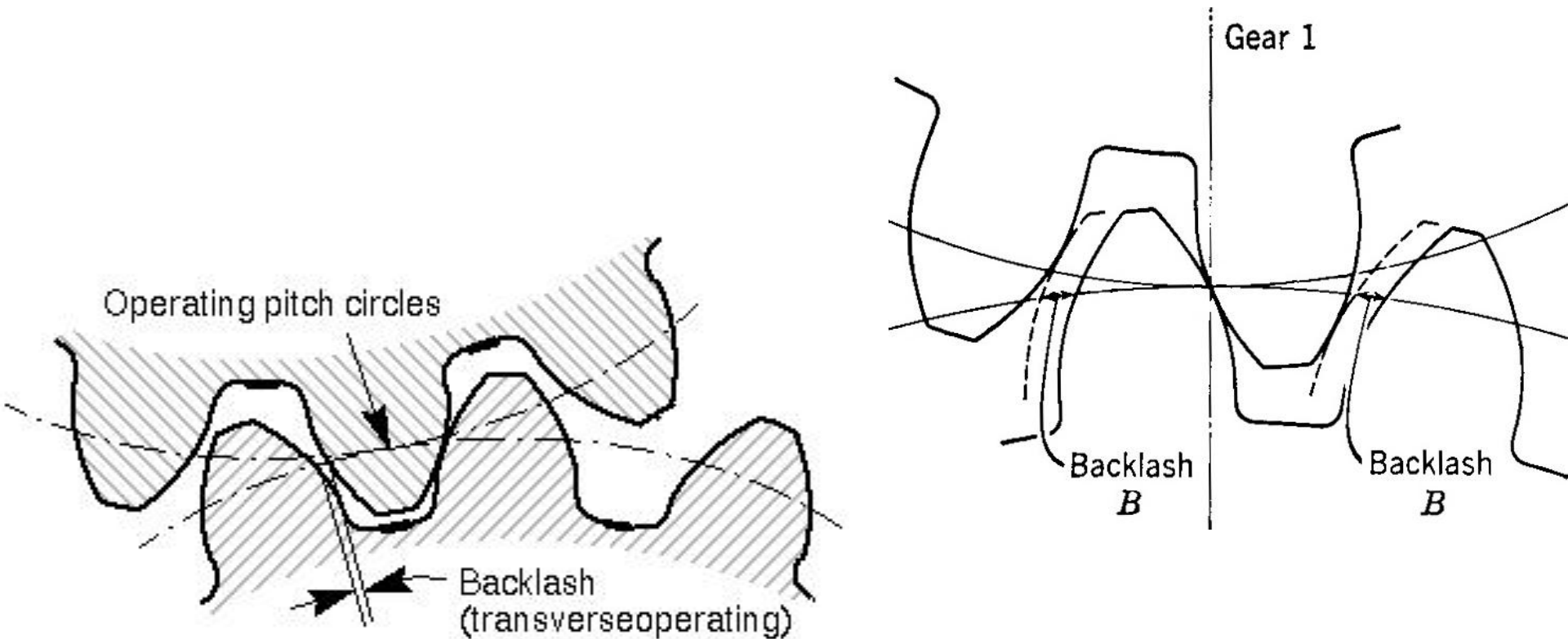


The *clearance circle* is a circle that is tangent to the addendum circle of the mating gear.

The clearance c is the amount by which the dedendum in a given gear exceeds the addendum of its mating gear.



The *backlash* is the amount by which the width of a tooth space exceeds the thickness of the engaging tooth measured on the pitch circles.



Gear teeth size are standardized as the other mechanical elements (bolts, nuts, bearings, etc.) too.

You should prove for yourself the validity of the following useful relations:

In SI Units

Size is in *mm*

N_P # of teeth on pinion

N_G # of teeth on gear

d_P dia. of pinion pitch circle, mm

d_G dia. of gear pitch circle, mm

m : module (index of tooth size, mm)

$$m = \frac{d}{N}; \quad \frac{d_P}{N_P} \text{ or } \frac{d_G}{N_G}$$

p_c : circular pitch (not much used in SI)

$$p_c = \frac{\pi d}{N} = \pi \left(\frac{d}{N} \right) = \pi m; \quad \text{in mm.}$$

In American System

Size is in *inches*

N_P # of teeth on pinion

N_G # of teeth on gear

d_P dia. of pinion pitch circle, *in*

d_G dia. of gear pitch circle, *in*

P : diametral pitch (index of tooth size 1/inches)

$$P = \frac{N}{d}; \quad \frac{N_P}{d_P} \text{ or } \frac{N_G}{d_G}$$

p_c : circular pitch used in many calculations

$$p_c = \pi \frac{d}{N} = \frac{\pi}{N/d} = \frac{\pi}{P} \text{ or}$$

$$p_c \times P = \pi \quad \text{in inches.}$$

$$p_c = \frac{\pi d}{N} = \pi \left(\frac{d}{N} \right)$$

$$p_c = \pi m; \quad \text{in mm.}$$

$$p_c = \pi \frac{d}{N} = \frac{\pi}{N/d} = \frac{\pi}{P} \text{ or}$$

$$p_c \times P = \pi \quad \text{in inches.}$$

$$p_c = \frac{\pi}{P} \text{ in} \times \frac{25.4 \text{ mm}}{\text{in}}$$

$$\pi m = \frac{\pi}{P} 25.4$$

$$m = \frac{25.4}{P}$$

This is the relation between SI module and Imperial(American) diametral pitch

A gear with 8(teeth/inch) diametral pitch will have a corresponding module of

$$m = \frac{25.4}{8} = 3.175 \text{ mm}$$

5.3 CONJUGATE ACTION

Mating gear teeth acting against each other to produce rotary motion are similar to cams. When the tooth profiles (or cams) are designed so as to produce a constant angular velocity ratio during meshing, these are said to have conjugate action.

In theory, at least, it is possible arbitrarily to select any profile for one tooth and then to find a profile for the meshing tooth that will give conjugate action.

One of these tooth profile solutions is the involute profile, which, with few exceptions, is in universal use for gear teeth and is the only one with which we should be concerned.

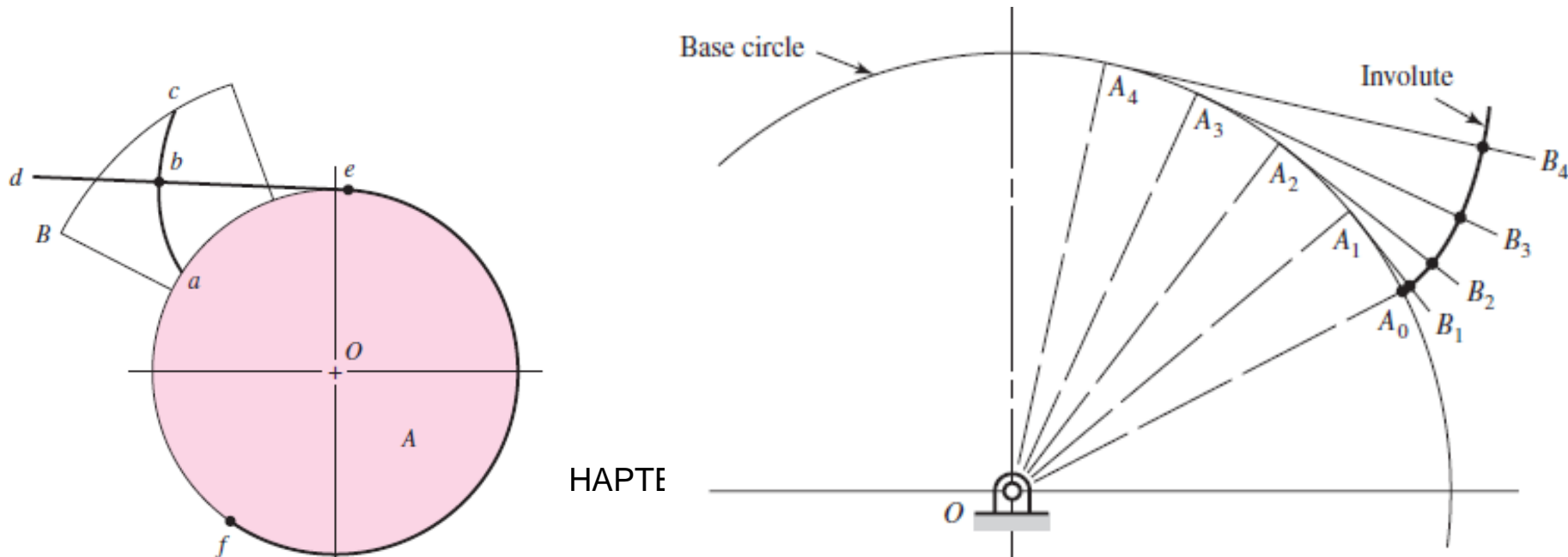
Tooth profile shapes: 1)-involute -----> most widely used shape
2)-cycloidal }
3)-circular } Not much used anymore.

5.4 INVOLUTE PROPERTIES

An involute curve may be generated as shown in Fig. 5.8. A partial flange B is attached to the cylinder A , around which is wrapped a cord def , which is held tight.

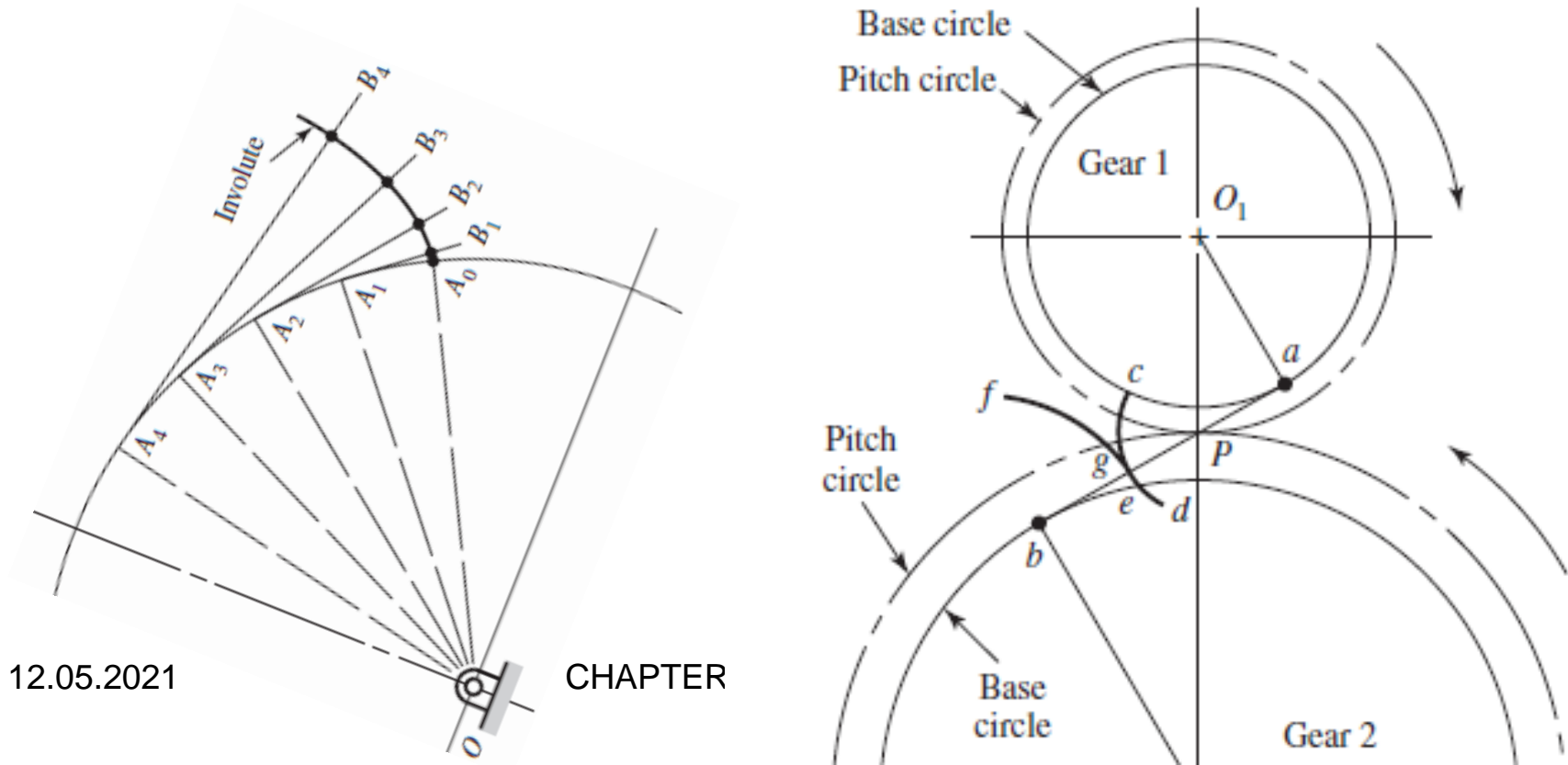
Point b on the cord represents the tracing point, and as the cord is wrapped and unwrapped about the cylinder, point b will trace out the involute curve ac .

The radius of the curvature of the involute varies continuously, being zero at point a and a maximum at point c . At point b the radius is equal to the distance be , since point b is instantaneously rotating about point e . Thus the generating line de is normal to the involute at all points of intersection and, at the same time, is always tangent to the cylinder A . The circle on which the involute is generated is called the *base circle*



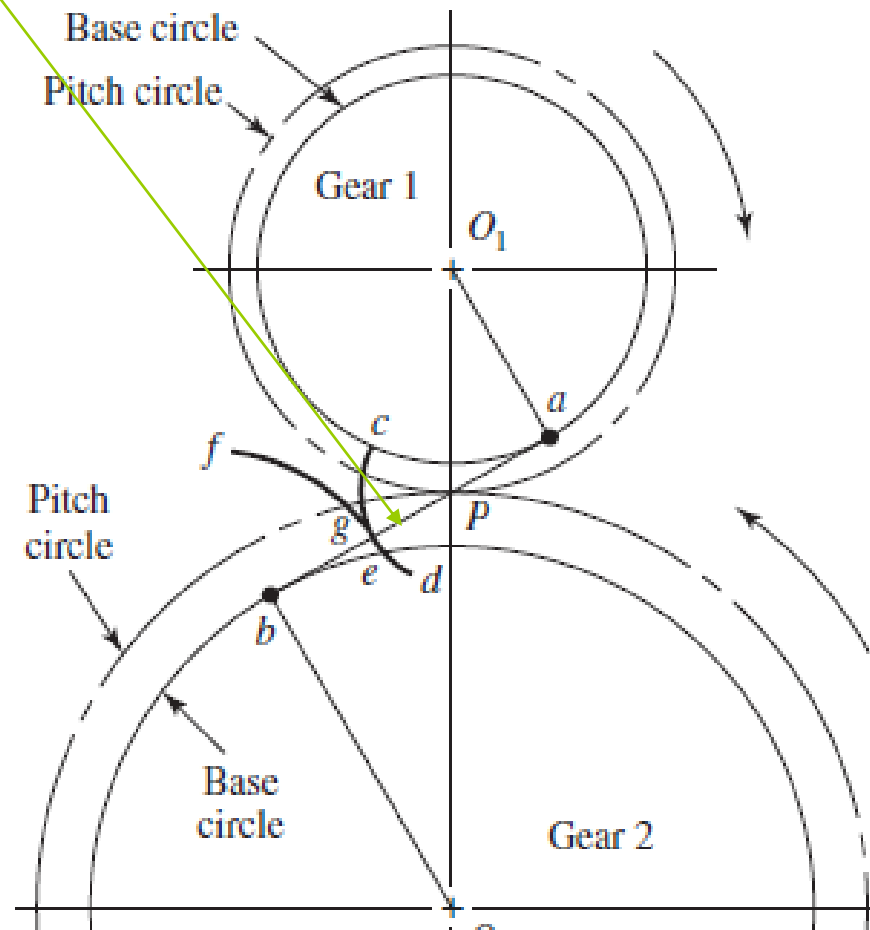
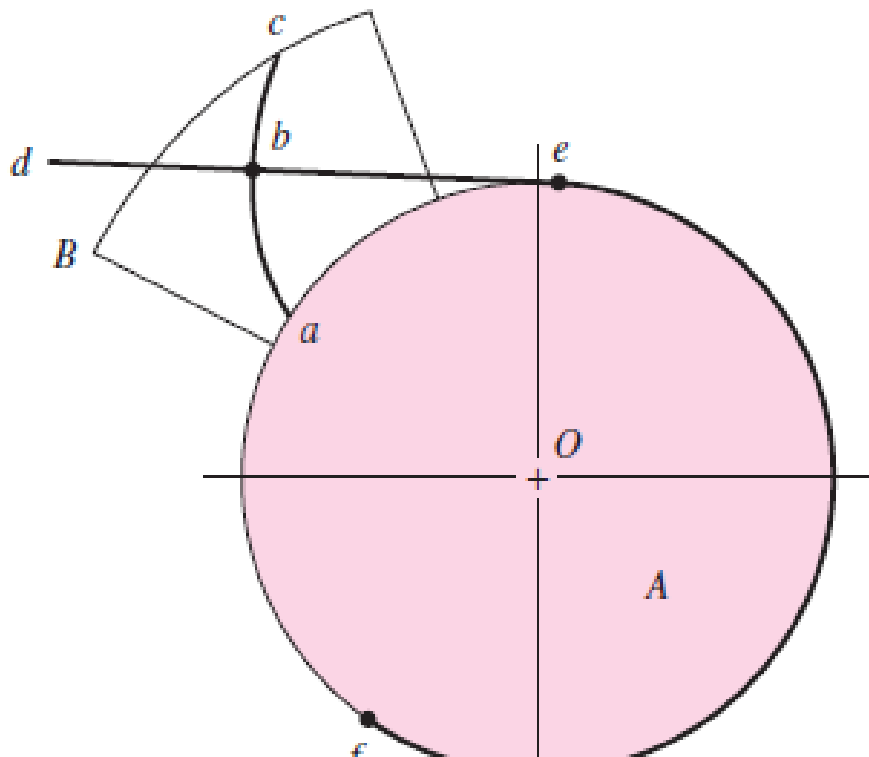
5.4 INVOLUTE PROPERTIES

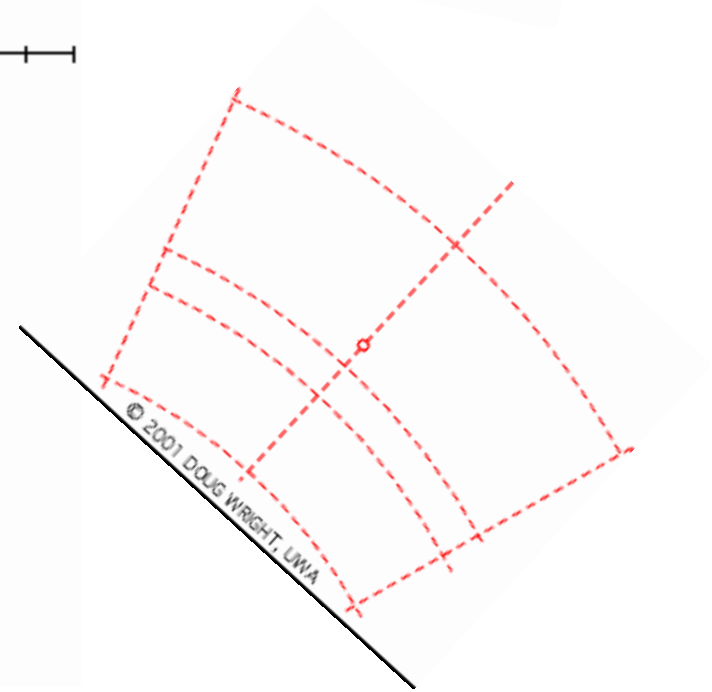
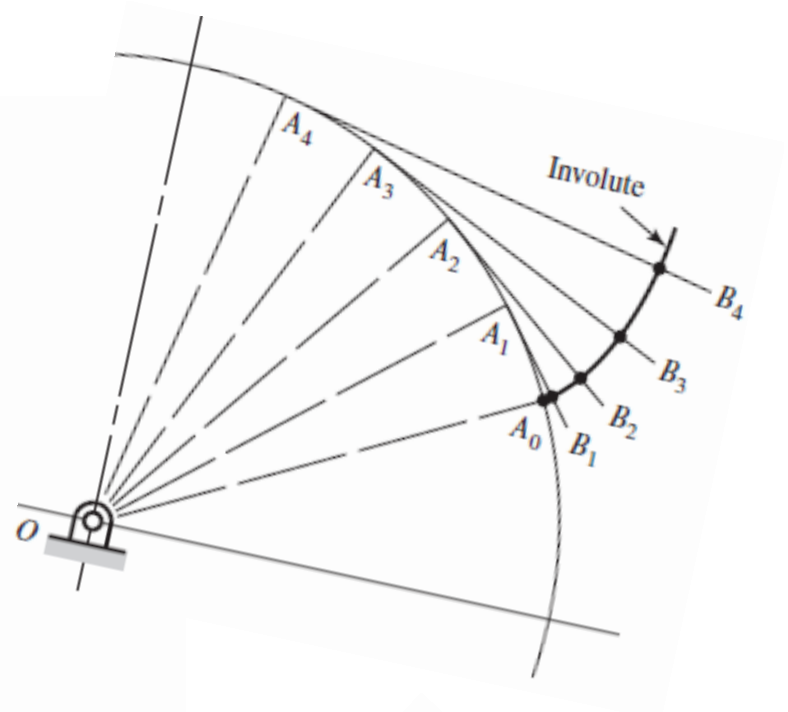
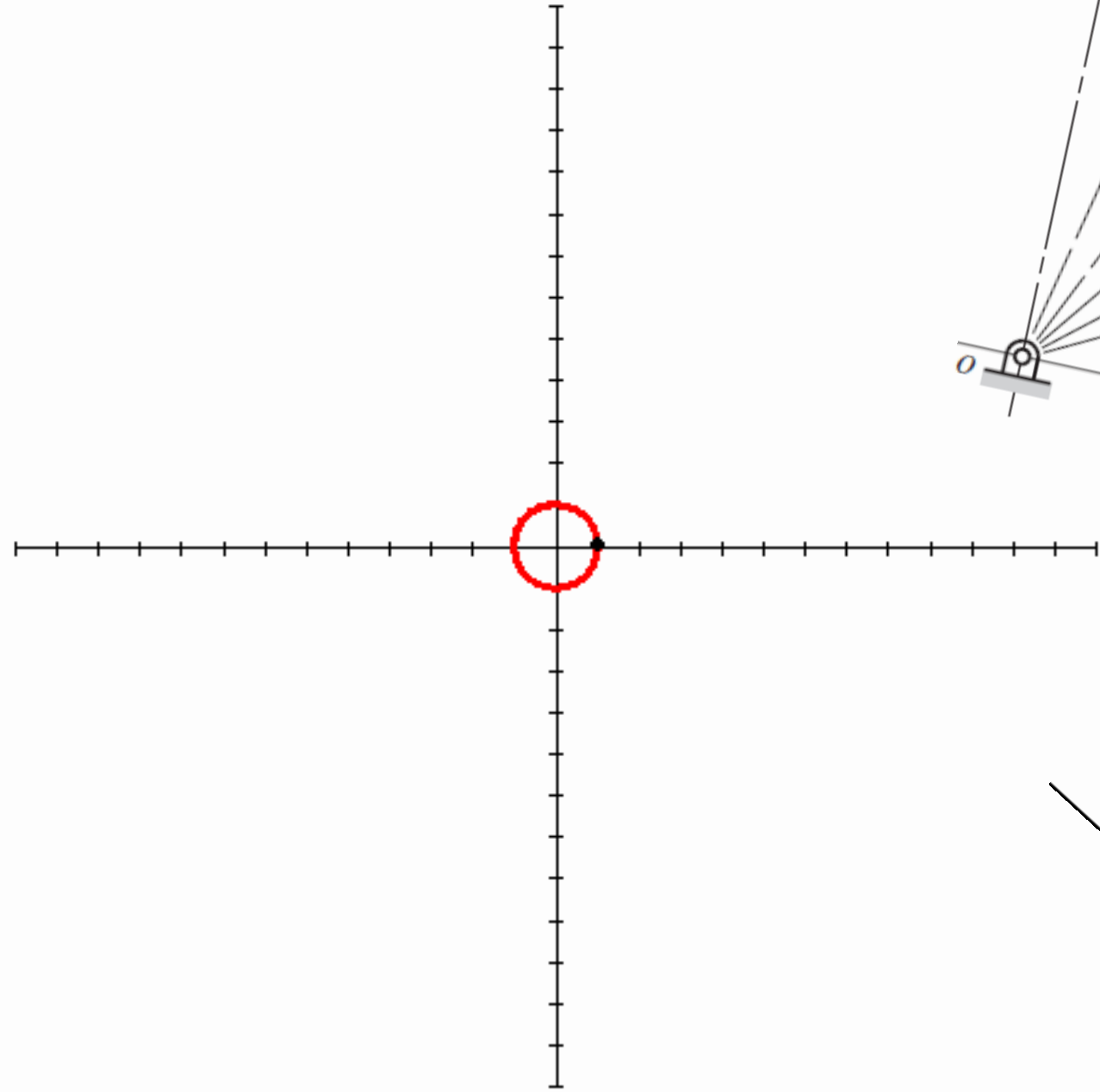
Let us now examine the involute profile to see how it satisfies the requirement for the transmission of uniform motion. In Fig. 5.8b, two gear blanks with fixed centers at O_1 and O_2 are shown having base circles whose respective radii are O_1a and O_2b . We now imagine that a cord is wound clockwise around the base circle of gear 1, pulled tight between points a and b , and wound counterclockwise around the base circle of gear 2. If, now, the base circles are rotated in different directions so as to keep the cord tight, a point g on the cord will trace out the involutes cd on gear 1 and ef on gear 2.

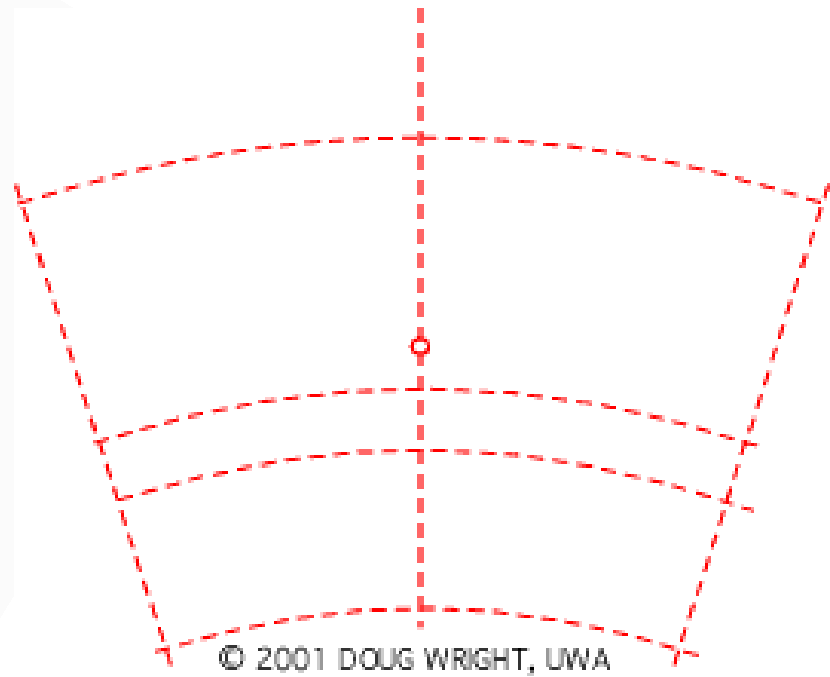
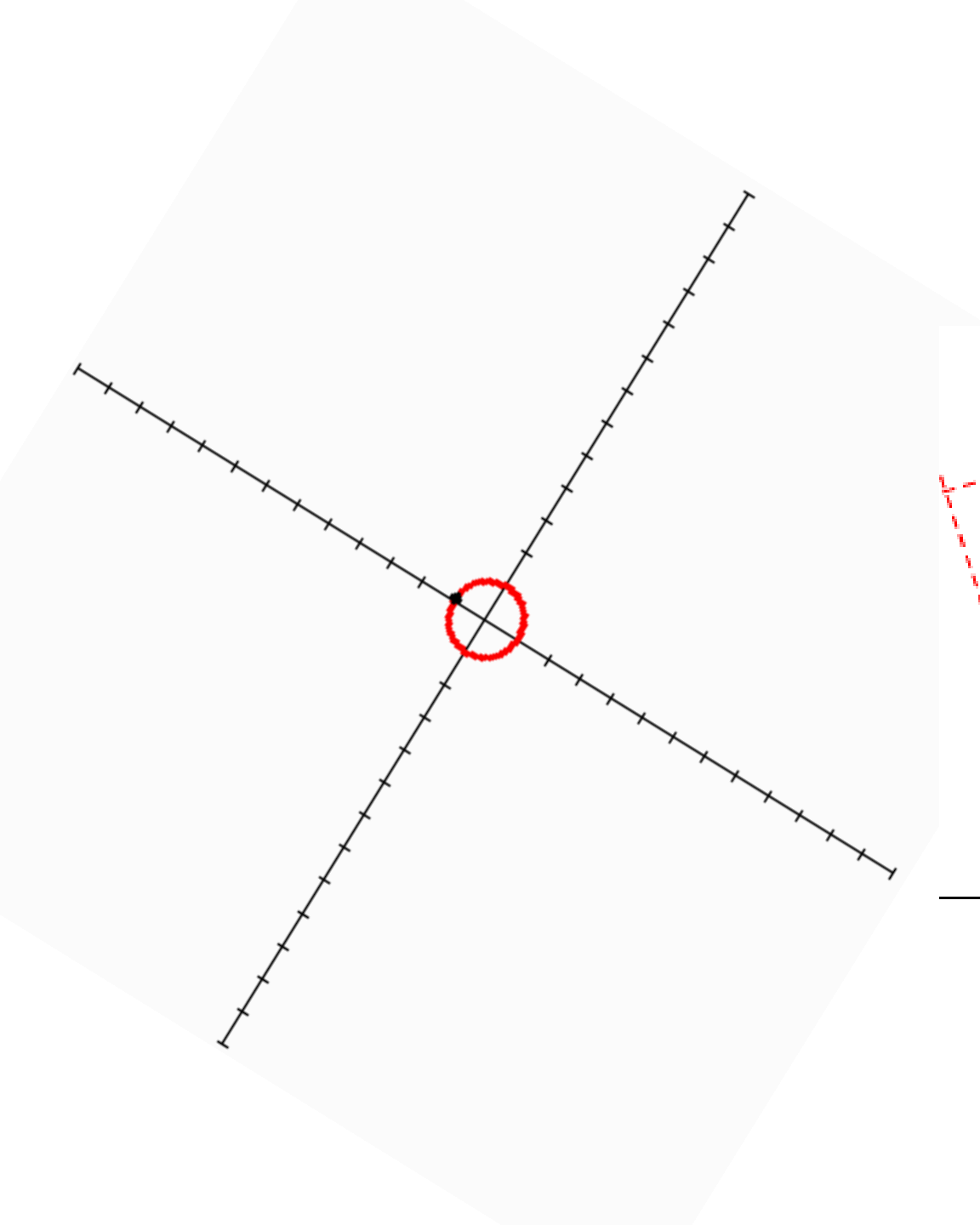


5.4 INVOLUTE PROPERTIES

The involutes are thus generated simultaneously by the tracing point. The tracing point, therefore, represents the point of contact, while the portion of the cord ab is the generating line.







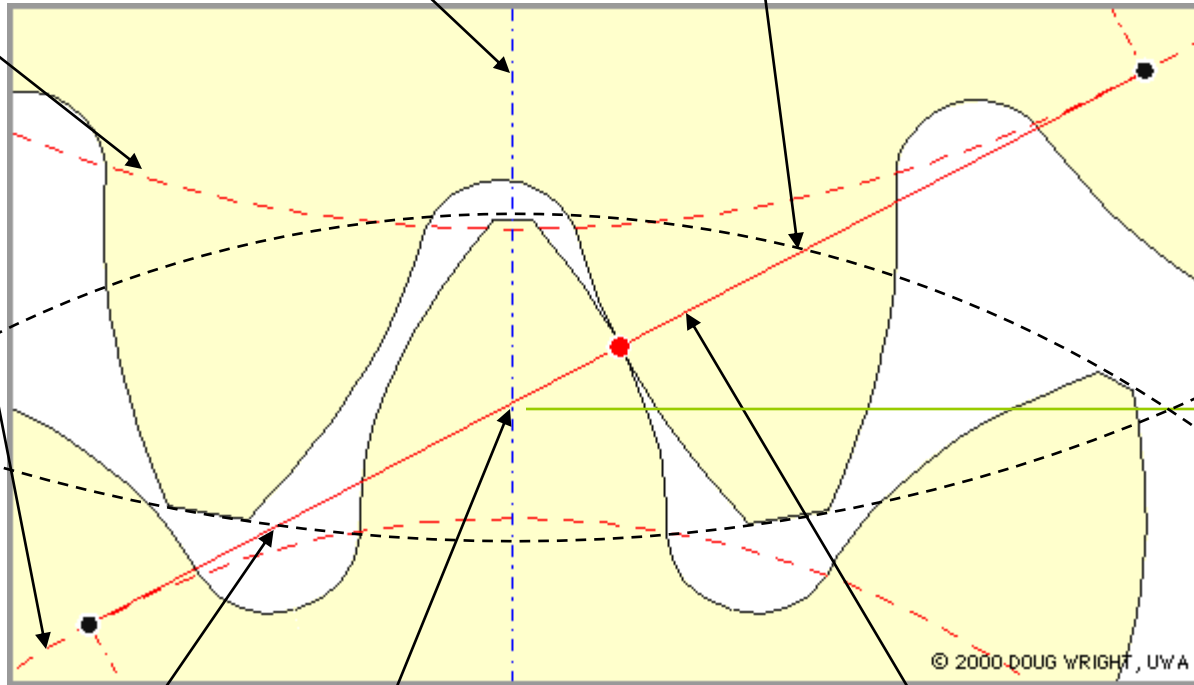
© 2001 DOUG WRIGHT, UWA

Center line
of gear
centers

End of tooth contact

Tip (addendum)
circles

Base circles



Start of tooth contact

Pitch point

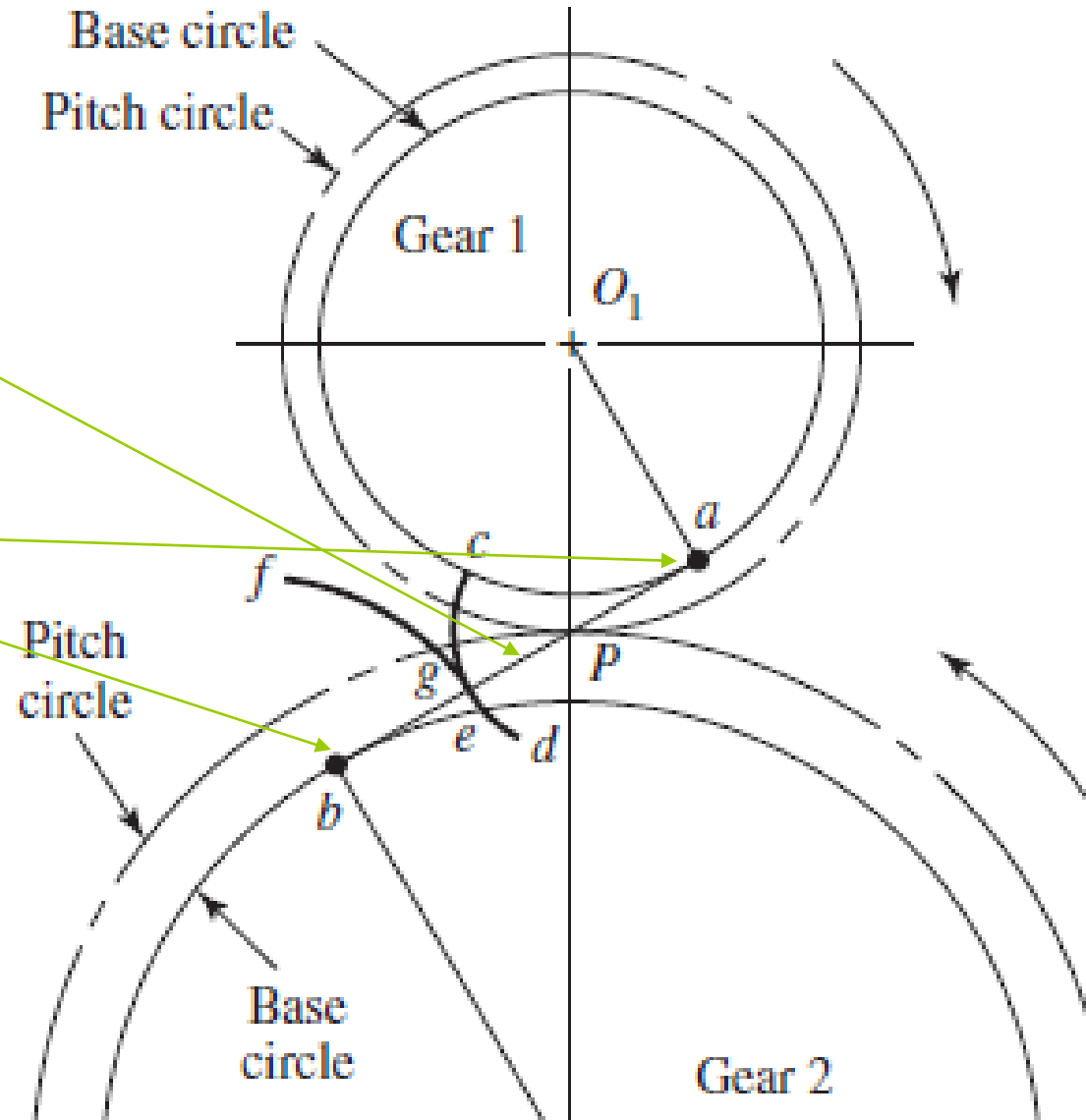
Generating line, Line of action,
Pressure line (tangent to base circles
at two ends)

5.5 FUNDAMENTALS

The point of contact moves along the generating line;

the generating line does not change position, because it is always tangent to the base circles;

and since the generating line is always normal to the involutes at the point of contact, the requirement for uniform motion is satisfied.



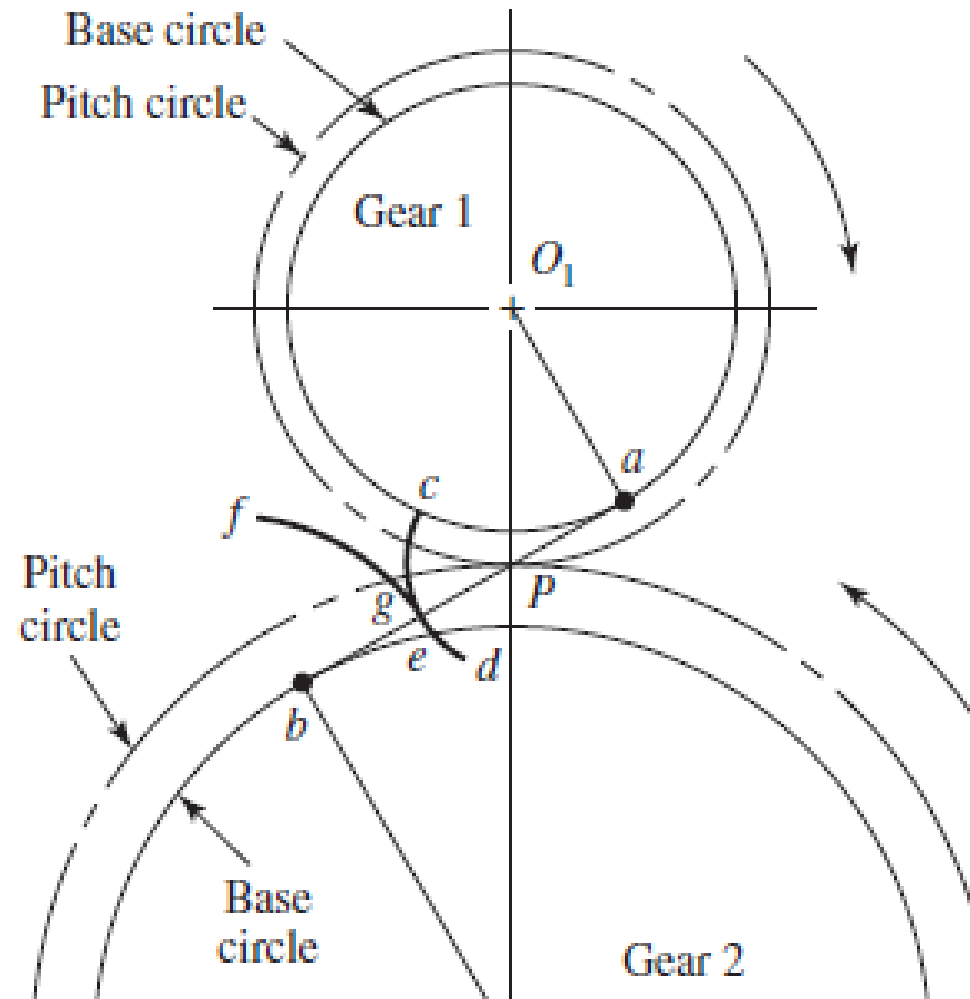
5.5 FUNDAMENTALS

When two gears are in mesh, their pitch circles roll on one another without slipping. Designate the pitch radii as r_1 and r_2 and the angular velocities as ω_1 and ω_2 , respectively. Then the pitch-line velocity is

$$V = |r_1 \omega_1| = |r_2 \omega_2|$$

Thus the relation between the radii on the angular velocities is

$$\left| \frac{\omega_1}{\omega_2} \right| = \frac{r_2}{r_1}$$



5.5 FUNDAMENTALS

Suppose now we wish to design a speed reducer such that the input speed is 1800 rev/min and the output speed is 1200 rev/min.

This is a speed decreasing ratio of 3:2; the pitch diameters would be in the same ratio, for example, a 4-in pinion driving a 6-in gear.

Or we could start with tooth numbers and diametral pitch and then find diameters of gears. The various dimensions found in gearing are always based on the pitch circles.

Suppose we specify that an 18-tooth pinion is to mesh with a 27-tooth gear and that the diametral pitch of the gear set is to be 6 teeth per inch.

Then, from Eq. ($P = \frac{N}{d}$), the pitch diameters of the pinion and gear are, respectively,

$$d_1 = \frac{N_1}{P} = \frac{18}{6} = 3 \text{ in}$$

$$d_2 = \frac{N_2}{P} = \frac{27}{6} = 4.5 \text{ in}$$