Comparison of Stable Governors

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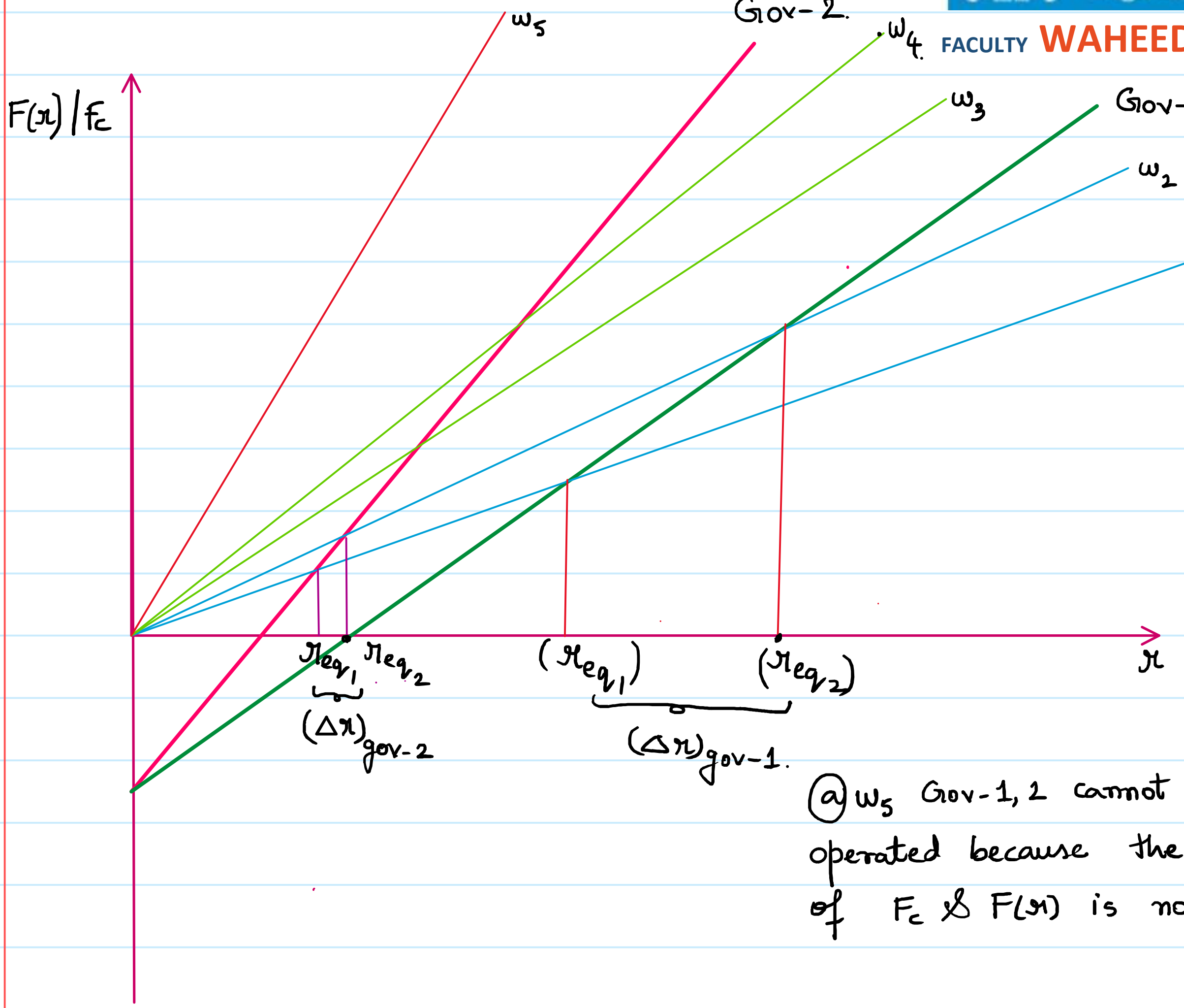

@ ω_5 Gov-1, 2 cannot be operated because the equilibrium of F_c & $F(r)$ is not obtained.

Range of speed.

Gov-1 $\longrightarrow$ ω_1 to ω_2. (Low speed governor)

Gov-2 $\longrightarrow$ ω_1 to ω_4 (High speed governor).

$$\boxed{\left(\frac{dF(r)}{dr}\right)_{gov-2} > \left(\frac{dF(r)}{dr}\right)_{gov-1}}$$

$$(\Delta r)_{gov-1} > (\Delta r)_{gov-2}$$

Gov-1 produced more change in radius when speed increases from ω_1 to ω_2 $\rightarrow$ Gov-1 produces more sleeve displacement.

Gov-1 is more sensitive then Gov-2.

Gov-2 is more stable then Gov-1

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11. Hartnell governors are specified by their C.F. curves

CF = 50 r – 2000 for governor 1
CF = 50 r – 1000 for governor 2] ||le line.
CF = 100 r – 1000 for governor 3

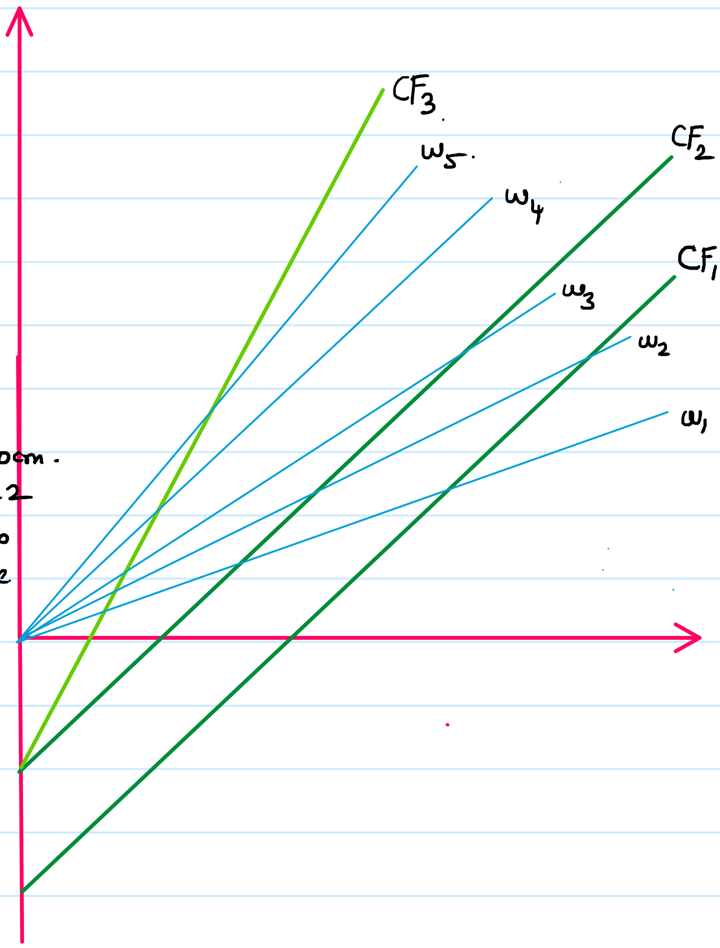
where CF in Newtons and r is in cm.

Identify the correct statements about them

1) Governor 1 is useful for very low speeds where as Governor 2 is useful at high speeds and 3 at still higher speeds. ✓
2) The minimum radius at which the governor 1 is active is greater than 40 cm where as it is greater than 20 cm for governor 2. ✓
3) In the range of 50 cm to 60 cm radii of rotation. The Governor 3 is less sensitive than governor 2 ✓
4) At any radius above 20 cm governor 3 has higher speed compared to governor 2 ✓

(a) all are correct
(b) 2, 3, 4 are correct
(c) 1, 3, 4 are correct
(d) 1, 2, 3 are correct

@ $r = 10$ cm. Gov-3 is active., @ $r = 20$ cm. Gov-3 is operating, Gov-2 is about to get active. Gov-1 is not active.



	Range of speed
Gov-1	ω_1 to ω_2
Gov-2	ω_1 to ω_3
Gov-3	ω_1 to ω_5

$$CF_1 = 50r - 2000 \geq 0$$

$$(r_{min})_{gov-1} \geq \frac{2000}{50} = 40\text{cm}.$$

$\longrightarrow$ Gov-1 becomes active. $r \geq 40$cm.

$$CF_2 = 50r - 1000 \geq 0$$

$(r_{min})_{gov-2} \geq 20$cm. $\longrightarrow$ Gov-2 become active $r \geq 20$cm.

$$CF_3 = 100r - 1000 \geq 0$$

$(r_{min})_{gov-3} \geq 10$cm. $\longrightarrow$ Gov-3 becomes active $r \geq 10$cm.

$$\left(\frac{dF(r)}{dr}\right)_{gov-3} > \left(\frac{dF(r)}{dr}\right)_{gov-2,1}$$

Gov-3 is more stable and less sensitive then Gov-2 & Gov-1.

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GEARS:
Gears are the example of higher pair mechanism they are used to transfer constant velocity from one shaft to another shaft.

Gear: The wheel which is larger in size is known as gear. Due to its bigger size mass moment of inertia is more, therefore it is preferred as **driven (o/p)** element.

Pinion: The wheel which is smaller in size is known as pinion. Due to smaller size it's mass moment of inertia is less and hence preferred for **driver(i /p)** element.

Classification of Gears:
Gears may be classified according to the relative position of the axes of shafts to be connected. It may be

1. Parallel axes
   a. Spur Gear
   b. Helical Gear
   c. Double Helical / Herringbone Gear

2. Intersecting axes
   a. Straight Bevel Gear
   b. Spiral Bevel Gear
   c. Zerol Bevel Gear
   d. Crown Gear

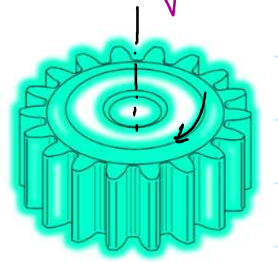
3. Neither parallel nor intersecting i.e. skew axes.
   a. Hypoid Gear
   b. Worm and Wheel
   c. Crossed Helical Gears

Gears for connecting parallel shafts
1. Spur gear → Teeth are cut on the cylinderical surface.

- Straight Spur gear is the simplest form of gears having teeth parallel to the gear axis.

sudden type of contact b/w. teeth.
cause shock loading.

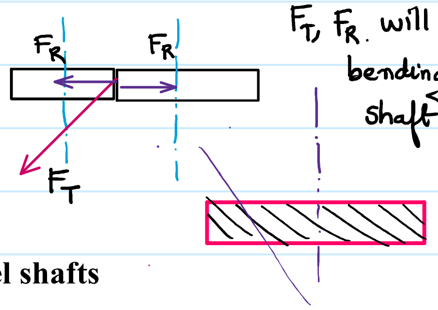
- The contact of two teeth takes place over the entire width along a line parallel to the axes of rotation.



- As gear rotate, the line of contact goes on shifting parallel to the shaft.
- Spur gears generate noise in high speed applications due to sudden contact over the entire face width between two meshing teeth. → shock loading.
- Spur gears are cheapest.

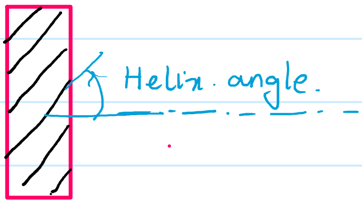
Use:

- Automobiles.



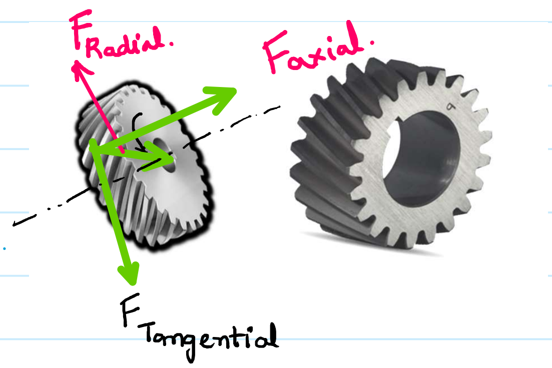
F_T, F_R will cause. bending of shaft.

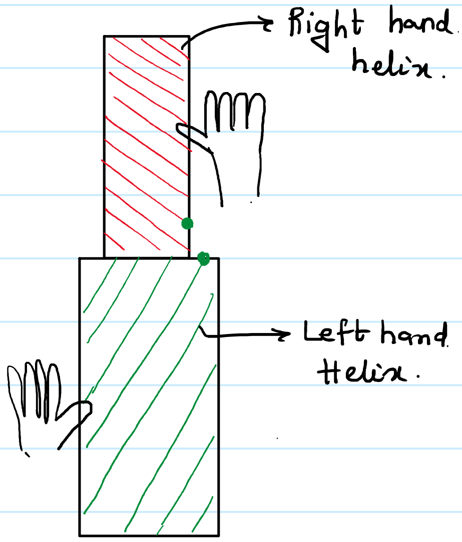
F_R – Radial force.
F_T – Tangential force.



Gears for connecting parallel shafts
2. Helical Gears:

- In helical gears teeth are part of helix instead of straight across the gear parallel to the axis.
- The mating gears will have same helix angle but in opposite direction for proper mating.
- As the gear rotates, the contact shifts along the line of contact in involute helicoid across the teeth.





Gears for connecting parallel shafts
2. Helical Gears:

- There is gradual engagement at the beginning of any individual tooth, starting at the point on leading edge and progressing across the face of the gear as it rotates. → Reduces the effects of shock loading.
- This results in reduced dynamic effects and less noise.
- The inclined teeth develops thrust loads and bending couples, which are not present with spur gear.

Gears for connecting parallel shafts
3. Herringbone Gears:

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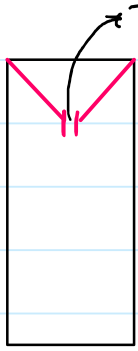
- Single Helical gear will produce an axial thrust on the shaft bearing due to the inclination of the teeth it can be eliminated by employing double helical gears with two sets of teeth back to back, each set cut to opposite hand.

Herring bone.



Double Helical gear.

Gears for connecting parallel shafts
3. Herringbone Gears:

- Herringbone gears are also known as Double Helical Gears.
- Herringbone gears are made of two helical gears with opposite helix angles, which can be up to 45 degrees.
- Helical gears use to obtain herringbone have **same module, number of teeth and pitch circle diameter,** but with teeth having **opposite hand of helix.**
- In Double helical gear there is a groove known as tool run out where as in Herringbone gear this groove is not present.

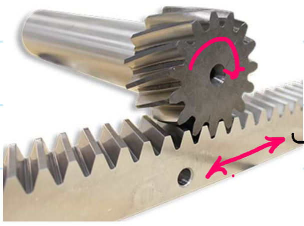
Use:

- High power applications such as ship drives and large turbines.

Gears for connecting parallel shafts

4. Rack and Pinion:

- In these gears the spur rack can be considered to be spur gear of infinite pitch radius with its axis of rotation placed at infinity parallel to that of pinion.
- The pinion rotates while the rack translates.



Gear of infinite Radius of Pitch circle.

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Gears for Intersecting Shafts:

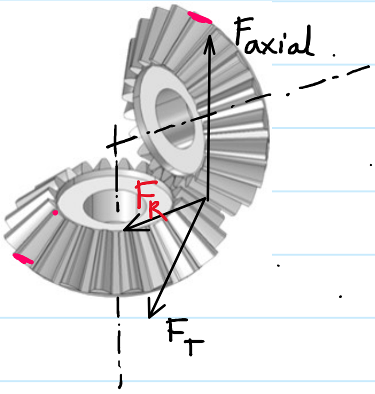
1. Straight Bevel Gears: – Teeth are cut on the conical surface

- Straight bevel gears are provided with straight teeth, radial to the point of intersection of the shaft axes and vary in cross section through the length inside generator of the cone.

C/s of teeth is uniform in case of spur, Helical gear.

- Straight Bevel Gears can be seen as modified version of straight spur gears in which teeth are made in conical direction instead of parallel to axis.
- Produce noise in high speed applications.

$F_{Tangential}$, F_{Radial}, F_{axial} are all present.

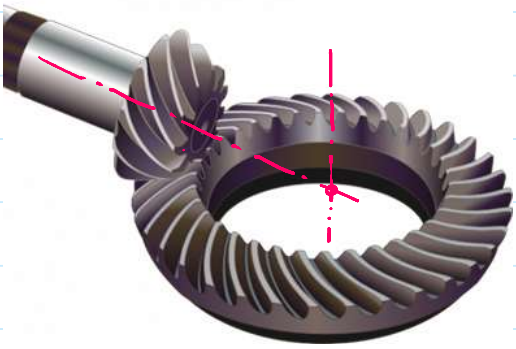


Gears for Intersecting Shafts:

2. Spiral Bevel Gears: → Teeth are cut on conical surface.

- Bevel gears are made with their teeth are inclined at an angle to face of the bevel and forms a circular arc.
- Spiral gears are also known as helical bevels.
- It gives the same advantage of helical teeth.
- Spiral bevels are difficult to design and costly to manufacture.
- They have smooth teeth engagement which results in quite application even at high speeds.
- Spiral bevel gears have better strength so use for high power transmission applications.

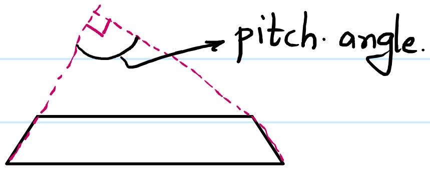
Note: When the axes are at right angles the larger gear is called a crown wheel and the smaller pinion.



Gears for Intersecting Shafts:
2. Spiral Bevel Gears – Types:

(a) Mitre gears: $\boxed{V.R. = 1.}$

- Two identical bevel gears mounted on shafts, which are intersecting at right angles. (Modifying the direction of angular velocity without changing magnitude)
- The pinion and gear have same dimensions namely addendum, dedendum, pitch circle diameter, number of teeth and module.
- The pinion and gear rotate at same speed.



Gears for Intersecting Shafts:
2. Spiral Bevel Gears – Types:

(b) Crown gear:

- When one of the gear has pitch angle 90° then that gear is called crown gear.
- Such gears are mounted on shafts which are intersecting at an angle that is more than 90°.

(c) Hypoid gears:

- Similar to spiral bevel gears that are mounted on shafts which are non-parallel, non-intersecting.
- Hypoid gears are based upon pitch surfaces which are hyperboloids of revolution.

Use:

- Automobile differentials.

Gears for Intersecting Shafts:

2. Spiral Bevel Gears – Types:

(d) Zerol gear:

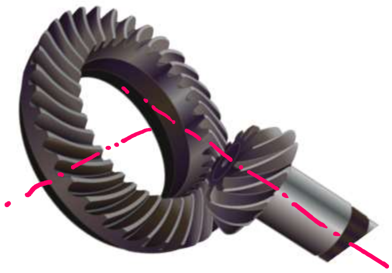
- Spiral bevel gears with zero spiral angle.
- These gears theoretically give more gradual contact and a slightly larger contact ratio.

NOTE: Bevel gears are always designed in pairs.

Gears for Skew Shafts:

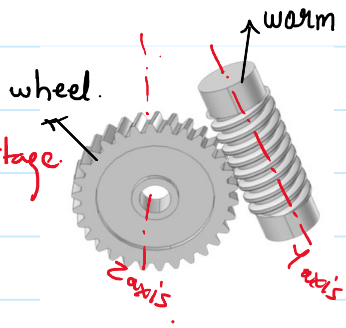
1. Hypoid Gears:

- The Hypoid Gears are made of the frusta of hyperboloids of revolution.
- Two matching hypoid gears are made by revolving the same line of contact, these gears are not interchangeable. → Mfg in pairs.
- Similar to bevel gears except that the shafts are offset and non-intersecting.



Gears for Skew Shafts:
2. Worm Gears:

- A special form of skew gears which has line contact is the worm and wheel pair.
- Usually, though not necessarily, the axes are at right angles.
- Teeth on worm gear are cut continuously like the threads on a screw.
- Worm resembles a screw.
- Direction of rotation of the worm gear (or worm wheel) depends upon the direction of rotation of the worm.



Reduction Ratio of 20:1 to 100:1 can obtained in single stage.

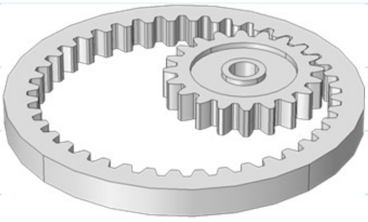
Advantages of Worm gear drives:
(i) High speed reduction such as 100:1 can be obtained with a single pair of worm gears.
(ii) Compact with small overall dimensions.
(iii) Smooth and silent operation.

$\eta \leq 50\%$

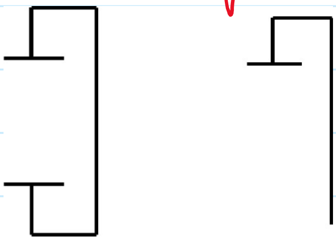
4. Self-locking provision can be made where the motion is transmitted only from worm to worm wheel, this is advantageous in applications like cranes and lifting devices.

Drawbacks of Worm gear drives:
(i) Efficiency is low compared to other gear drives.
(ii) Worm wheel is generally made of phosphor bronze, which increases the cost.

Ring gear or Annulus: The gear which is having teeth in inner side is known as ring gear and annulus.



Representation of Internal gear.



Performance Parmeter.

→ Gear Ratio - G = $\frac{\text{No of teeth on the wheel (driven element)}}{\text{No of teeth on Pinion(driving element)}}$ $G > 1$.

$$G = \frac{\omega_{pinion}}{\omega_{Gear}} > 1$$

→ Velocity Ratio $V.R/\lambda = \frac{\text{No. of teeth on driving element}}{\text{No. of teeth on driven element}}$

$$\lambda < 1.$$

$$\boxed{G = \frac{1}{\lambda}}$$

→ Train value.

$$T.V. = \frac{\text{Product of No. of teeth on driving element}}{\text{Product of No. of teeth on driven element}}$$

→ Speed Ratio $= \frac{1}{\text{Train Value}}$.

Gear Terminology:

Important Circle:

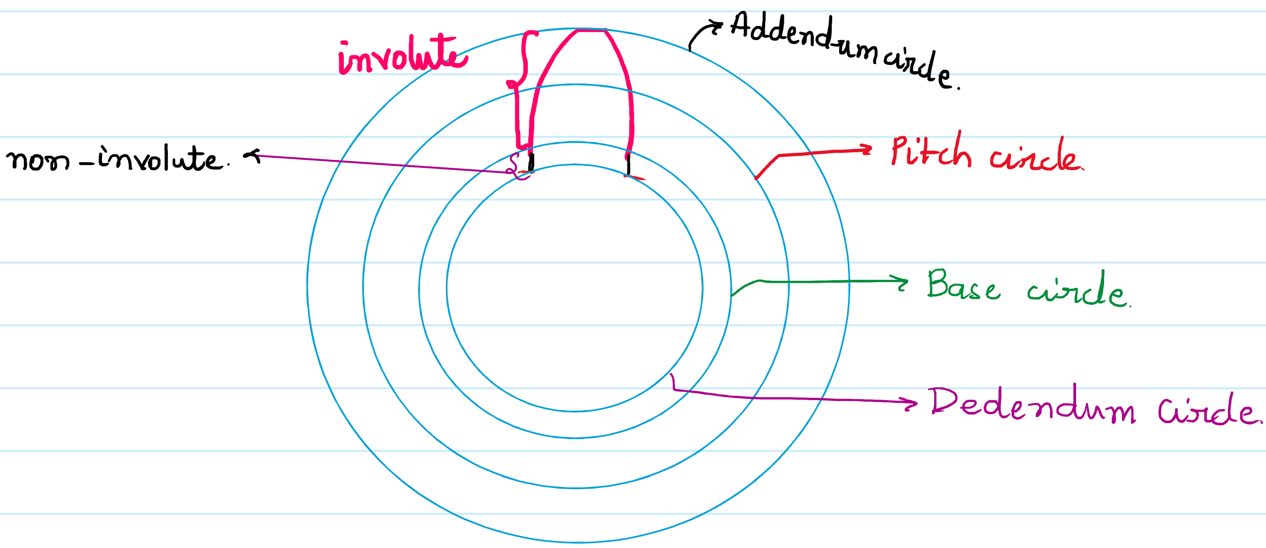
(a)**Pitch Circle**: It denotes the size of the gears, all the important directions such as tooth thickness, tooth space, arc of contact etc. are measured on the pitch circle circumference. Pitch circle corresponds to Pure Rolling action.

- It is an imaginary circle i.e., it's radius can be changed.
- Two gears which are in mesh can be represented by their pitch circle and the point of the contact of the pitch circle is known as pitch point.

(b) **Base Circle**: It is a real circle i.e., it's radius cannot be changed. It is the circle from where the involute profile begins.

(c) **Addendum Circle**: The circle which passes through top of the gear teeth is known as addendum circle.

(d) **Dedendum** Circle: Circle from where gear tooth begins.



Units of Gears:

(a) Circular Pitch (p_c): It is the distance between two similar points an adjacent tooth measured along pitch circle circumference.

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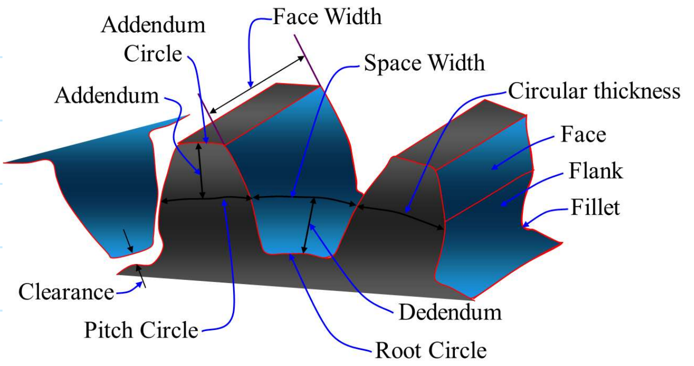
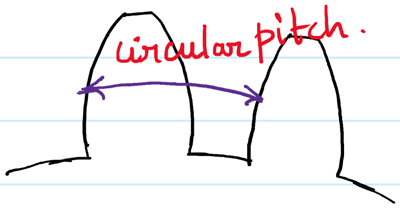


$$p_c = \pi m. \quad = \frac{\pi D}{T}.$$



module – $m = \dfrac{\text{Diameter of pitch circle}}{\text{No. of teeth.}}$

module signifies the thickness of teeth.

Diametrical pitch – $p_D = \dfrac{T}{D}$

No. of teeth per inch of diameter.

F.P.S.

Product of Diametrical Pitch and circular pitch.

$$p_D \times p_c = \frac{T}{D} \cdot \frac{\pi D}{T} = \pi.$$

Product of Diametrical pitch and Module.

$$p_D \times m = T/_D \cdot \times D/_T = 1.$$

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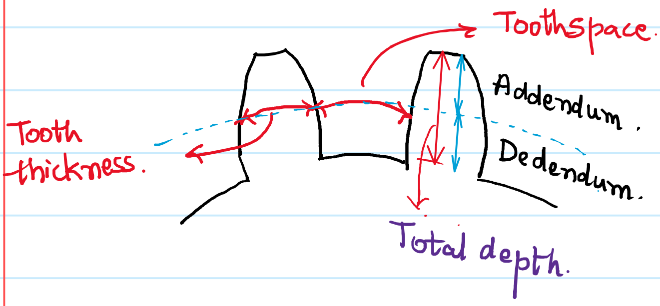
Important Dimensions:

Set (A): Dimensions which are visible in a single gear (without meshing).

1. **Tooth thickness**: the thickness of the tooth measured along pitch circle circumference.

2. **Tooth Space**: The distance between adjacent tooth is known as tooth space, during meshing tooth thickness of the gear will entre in to tooth space of pinion and vice-versa.

3. **Dedendum**: The radial distance between pitch circle and dedendum circle is known as addendum.



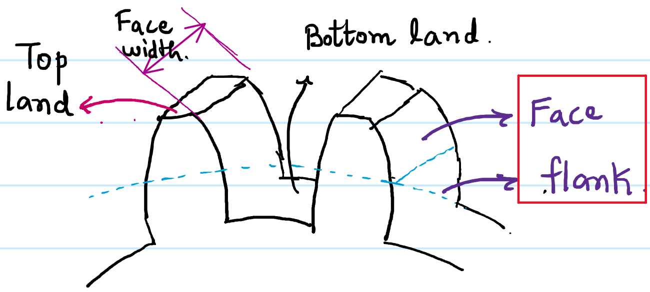
circular pitch = (Tooth space + Tooth thickness)

Total depth = Addendum + Dedendum.

Dedendum = 1.125m.

4. **Addendum** is the radial height above the pitch circle

	Addendum.
Full depth.	1module.
Stub.	(0.85 to 0.95) module.



Face - It is the portion of tooth above the pitch circle along the axis of gear.

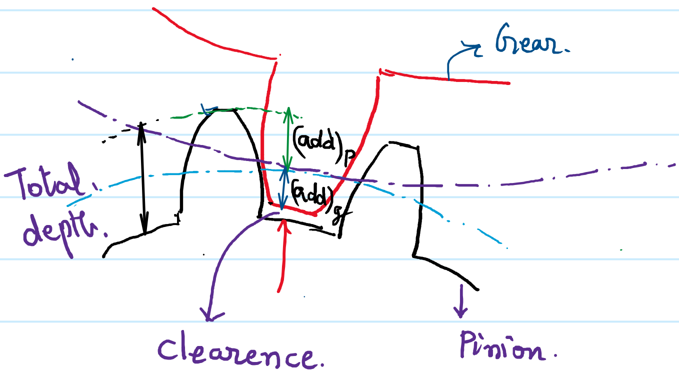
Flank - It is the portion of tooth below the pitch circle along the axis of gear.

Set (B):

01. **Working Depth** : Summation (addition) of addendum of gear and addendum of pinion is known as working depth.

02. **Clearance**: The distance between addendum circle of gear and dedendum circle of pinion is known as clearance.

- If clearance is absent then it will result in **interference.**



Working depth $= (add)_{pinion} + (add)_{gear}$.

$(add)_p$ – addendum of pinion

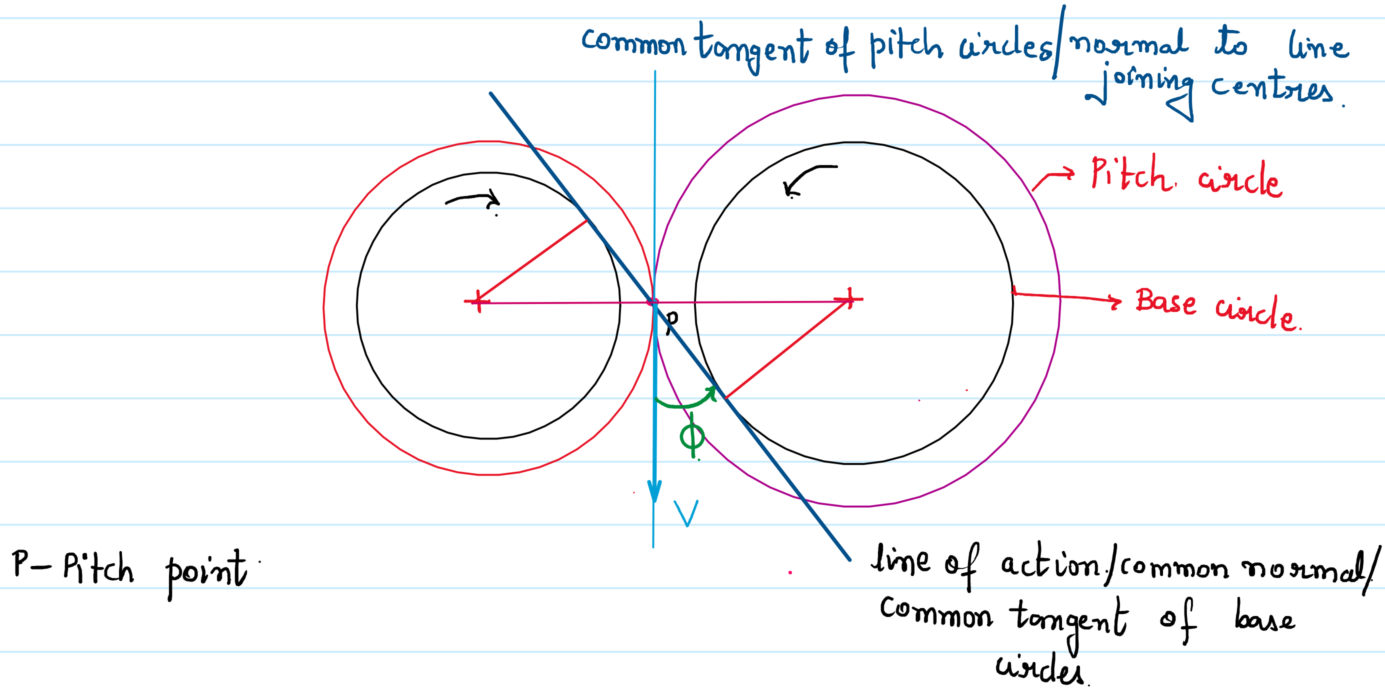
$(add)_g$ – addendum of gear

$$\text{Clearence} = \text{Total depth} - \text{Working depth}.$$
$$= (\text{Addendum} + \text{Dedendum})_{pinion} - \left((\text{Addendum})_{pinion} + (\text{Addendum})_{gear}\right)$$
$$= (\text{Dedendum})_{pinion} - (\text{Addendum})_{gear}.$$

3. **Backlash**:
The amount by which the tooth space of a gear is larger than tooth thickness of mating element is known as backlash.

- If backlash is absent then **rubbing** will take place in both the surface which will result in **more friction, more wear** , due to rubbing there will be more **heat generation** and as thermal expansion is prevented as no space is available it will further result in **thermal stress and strain concentration** due to which the tooth become week.
- If backlash increases there will be more noise and more vibrations.

1. **Pressure Angle** (f):
It is the angle between **common normal** and **velocity vector** at pitch point **rotated** in a direction similar to **driven element**.



Pressure Angle - It is the angle subtended between the common normal/common tangent of base circles and normal to line joining centres/common tangent of pitch circle measured at the pitch point.

- When a straight line rolls without slipping on a circle locus of any point on it will be **involute**, the circle is known as base circle and the line which generates involute is known as **common normal**.
- **For constant velocity ratio there must be meshing between conjugate profile therefore the common normal should be same St. line.**
- The entire action during meshing such as **beginning of engagement, ending of engagement, force transmission** or everything will take place along the **common normal**, therefore it is also called as line of action.
- The angle between common normal and common tangent to both the pitch point circles at pitch point is known as pressure angle.

Law of Gearing

Locating I - centres in Gear and Pinion

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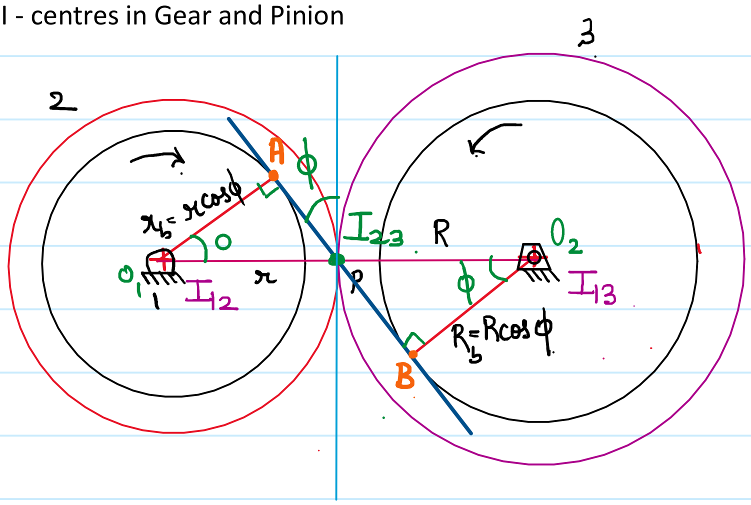

I_{23} must lie on common normal and colinear with I_{12} & I_{13}.

$O_1P - r -$ Pitch circle radius of Pinion.

$O_2P - R -$ Pitch circle radius of gear.

$AP = r\sin\phi$

$BP = R\sin\phi$

$$V_{I_{23}} = I_{12}I_{23} \cdot \omega_2 = I_{13} \cdot I_{23} \cdot \omega_3$$

$$\frac{\omega_2}{\omega_3} = \frac{I_{13} \cdot I_{23}}{I_{12} \cdot I_{23}} = \frac{R}{r} \neq \text{constant.}$$

Δ^{le} $I_{12} \cdot A\, I_{23}$ and Δ^{le} $I_{13} \cdot B \cdot I_{23}$ are similar Δ^{les}.

$$\frac{I_{13} \cdot I_{23}}{I_{12} \cdot I_{23}} = \frac{I_{13} \cdot B}{I_{12} \cdot A} = \frac{R_b}{r_b} = \frac{R\cos\phi}{r\cos\phi}$$

$$\frac{\omega_2}{\omega_3} = \frac{R_b}{r_b} = \frac{R\cos\phi}{r\cos\phi} = \text{constant.}$$

Law of Gearing:

- The ratio of first kinematic ratio .i.e., angular velocity must be constant.
- **Since the pitch circle is an imaginary circle that is radius can be changed therefore from here we cannot conclude that angular velocity will be constant.**
- The pitch point 'P' will be fixed point and it will divide the common normal in a constant ratio.
- All the elements satisfying law of gearing are known as gears presence of teeth is not a necessary condition.
- To transfer the relative motion at constant ratio the common normal must always pass through the pitch point.

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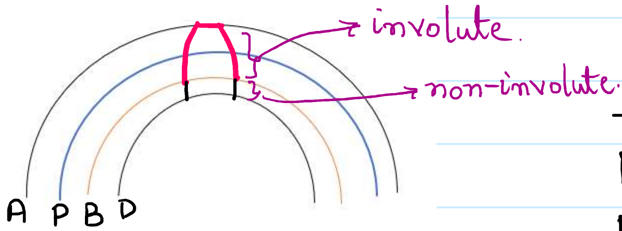
Case(i)

Pinion –



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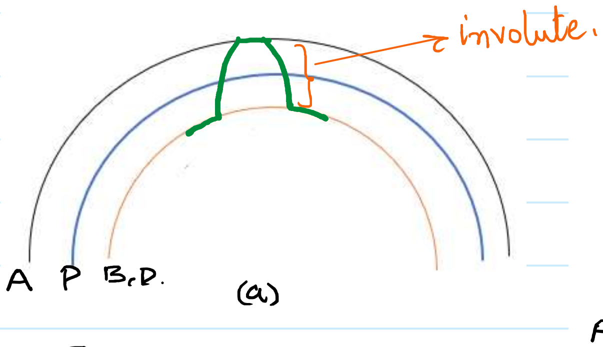
A – Addendum circle.

P – Pitch circle

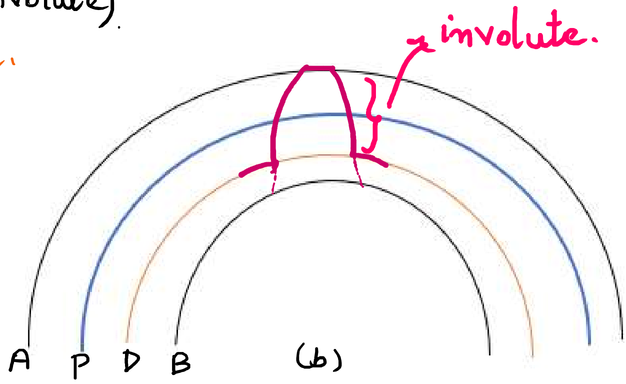
B – Base circle

D – Dedendum circle.

Gear.– $R_{Dedendum} \geq R_{base}$ → (Profile of Gear tooth is completely involute).



$R_{Base} = R_{Dedendum}$.



$R_{dedendum} > R_{base}$.

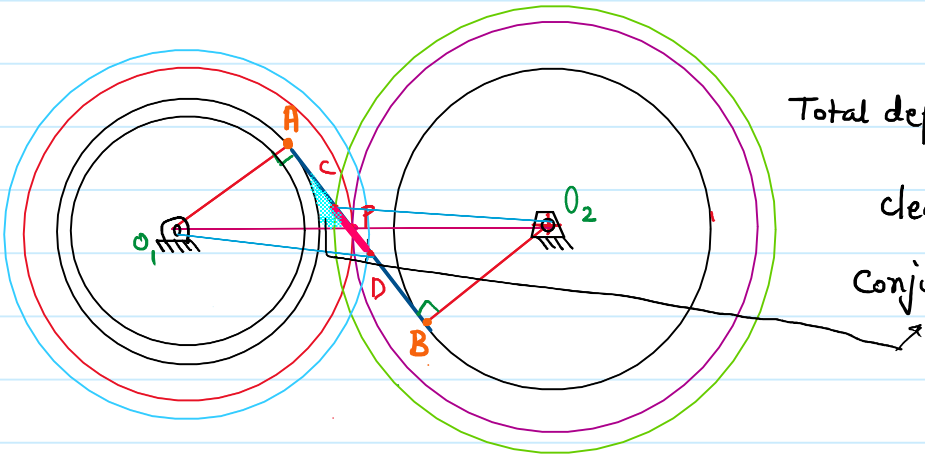
A, B are called points of tangency.

$O_1P - r$ $O_1A - r\cos\phi$

$O_2P - R$ $O_2B - R\cos\phi$

$AP - r\sin\phi$ $BP - R\sin\phi$

Case(ii)



Total depth > Working depth

clearence ≠ 0

Conjugate action occurs.

@ point C, D addendum of gear, pinion cut the line of actions.

AB – Max. lenght of line of action / Path of contact

CD – Actual lenght of line of action / Path of contact.

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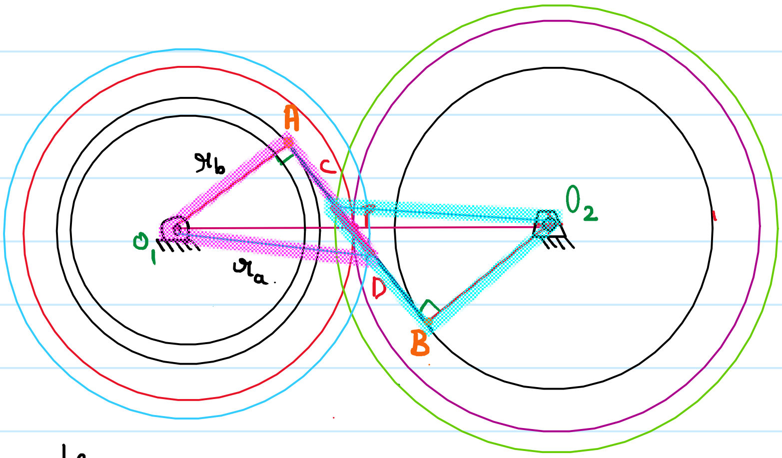

In Δ^{le} O_1AD

$$AD = \sqrt{O_1D^2 - O_1A^2} = \sqrt{r_A^2 - r_b^2} = \sqrt{r_A^2 - (r\cos\phi)^2}$$

In Δ^{le} O_2BC

$$BC = \sqrt{O_2C^2 - O_2B^2} = \sqrt{R_A^2 - R_b^2} = \sqrt{R_A^2 - (R\cos\phi)^2}$$

$$PD = AD - AP = \sqrt{r_A^2 - r_b^2} - (r\sin\phi)$$

$$CP = BC - BD = \sqrt{R_A^2 - R_b^2} - (R\sin\phi)$$

Actual lenght of Path of contact $= CP + PD$

$$= \underbrace{\sqrt{R_A^2 - R_b^2} - (R\sin\phi)}_{\text{Actual lenght of Path of Approach}} + \underbrace{\sqrt{r_A^2 - r_b^2} - (r\sin\phi)}_{\text{Actual lenght of Path of Recess}}$$

Max. lenght of Path of Contact $= AP + BP$

$$= \underbrace{r\sin\phi}_{\text{Max. Lenght of Path of Approach}} + \underbrace{R\sin\phi}_{\text{Max. lenght of Path of recess}}$$

Observations

Max. Length of Path of Approach is dependent on Pinion dimensions.
Actual Length of Path of Approach is dependent on Gear dimensions.
Max. length of Path of recess is dependent on Gear dimensions.
Actual length of Path of recess is dependent on Pinion dimensions.