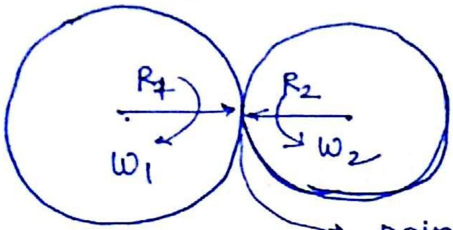
"Gear's"

Drive which is used in power transmission

Body ① Body ②



point of contact (Q)

At Q

By body① By Body②

$$\boxed{R_1\omega_1 = R_2\omega_2}$$

No slipping (Pure rolling)

- $\boxed{\frac{\omega_1}{\omega_2} = \frac{R_2}{R_1}}$ → Constant
- Friction will be static (F_s)
  Variable friction $\{0 \le f_s \le \mu N\}$

All those drive in which slip is possible called Negative drive eg:- Belt, Rope, chain

Driver

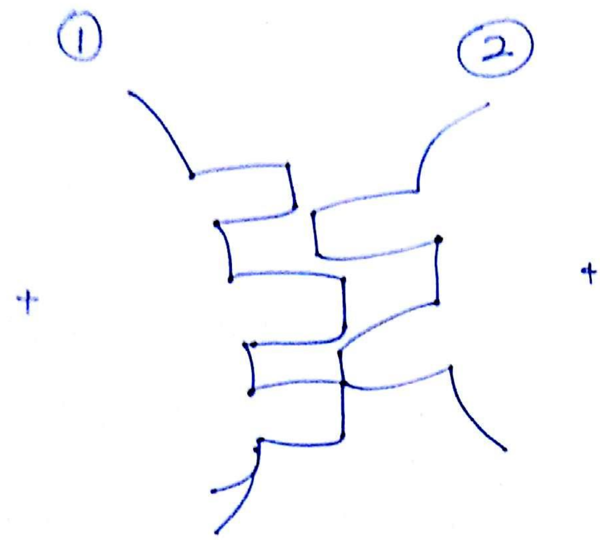
In slipping condition

$R_1\omega_1 \neq R_2\omega_2$

- $\frac{\omega_1}{\omega_2} \neq \frac{R_2}{R_1} \neq$ Const.
- kinetic friction is there (some power loss)

rolling with some amount of sliding

* In some systems a very high level of accuracy is demanded for the Velocity Ratio to be Constant in power transmission $\frac{\omega_1}{\omega_2} = \text{Constant}$



Slip is impossible

⇓

Positive Drive

⇓

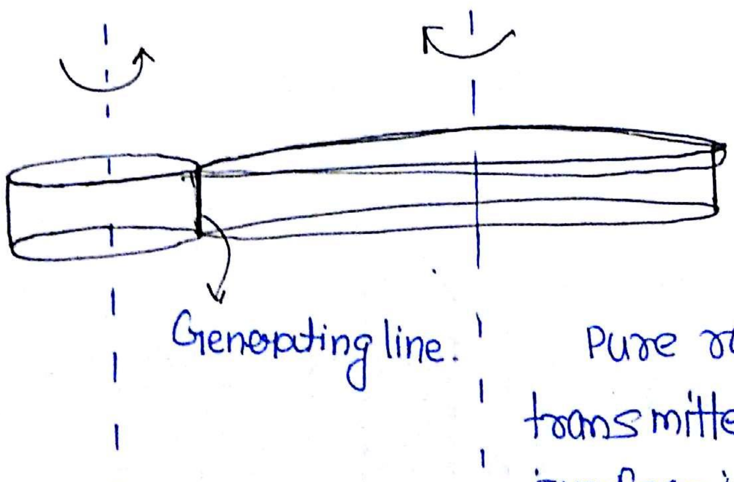
"Gear Drive"

$$\frac{\text{Manf. Cost}}{\text{Mat. Cost}} > 100$$

General classification of Gears :-

A] According to the axes of shaft connected :-

(i) Both Axes are parallel :-



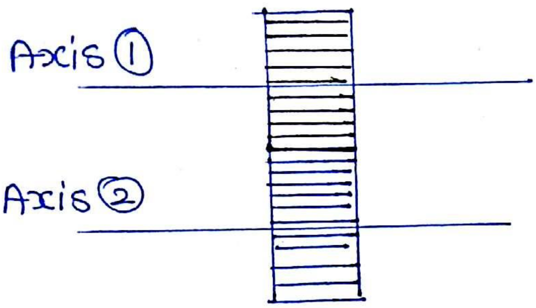
Generating line.

Pure rolling which can be transmitted b/w two Cy. ~~Sha~~ surface in Contact

Spur Gear :- teeth are straight and parallel to the axis of rotation.

99% failed

(1% use) → in very low power transmission at very low speed.



- Instantaneous engangement and disengangement (Impact stress on the profile)

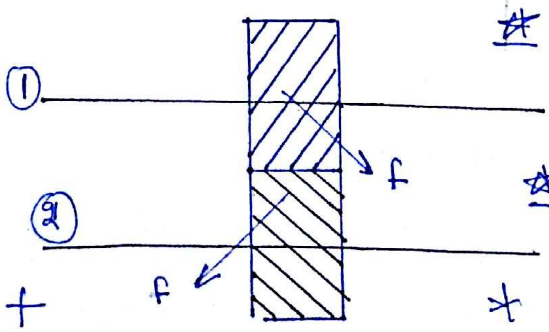
* No axial thrust.

Helical Gear :- teeth are straight but inclined to the axis of rotation.

Helical - Gear
→ Right handed
→ left handed

* Always opposite hand helical gears must come in contact.

* Gradual engangment
* No Impact stres
* 98% Used



~~Power transmission~~

* Axial thrust more.

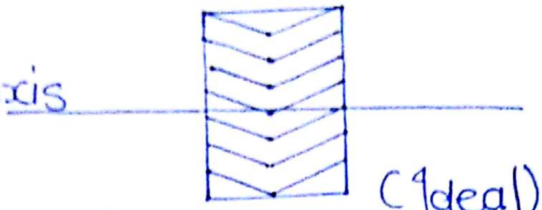
Double Helical Gear :- (Herringbone Gears)

Remember the name

→ to minimise axial thrust.

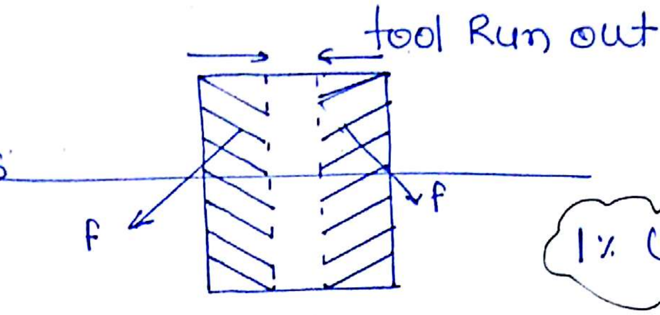
Costly

Axis



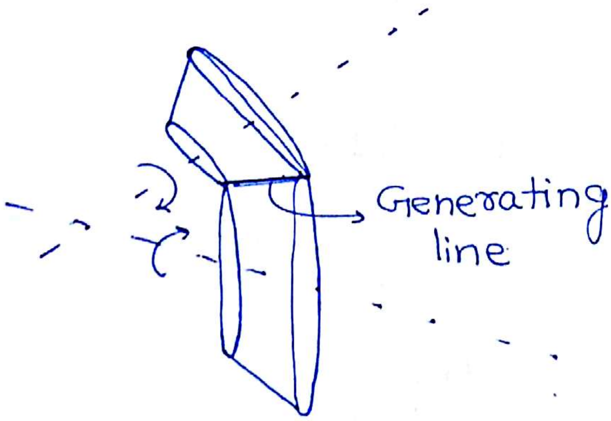
(Ideal)

In Real →



1% Use

3) Both Axes are intersecting :-



Pure Rolling motion can be transmitted b/w two conical surfaces in Contact

Bevel Gears :-

St. Bevel Gear — Zerol Bevel Gear — (Mainly used) Helical bevel Gear

St. Bevel Gear, Zerol Bevel Gear: Almost Same

Helical bevel Gear → R.H., L.H.

Plan

Plan

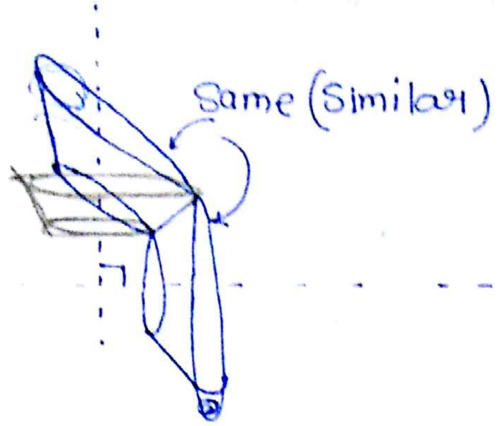
Axis

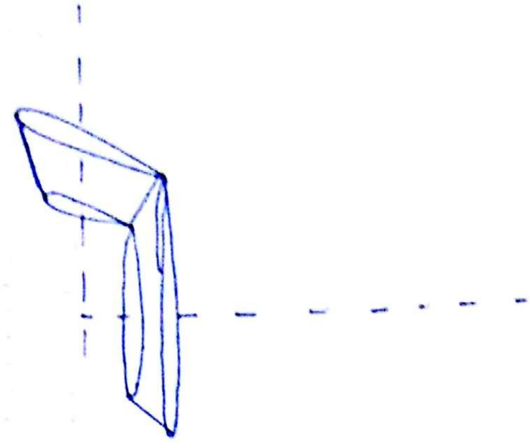
Axis

Axis

Impact stress (1% Use)

to transmited power b/w to mutually $\perp^{er}$ shafts.





$\frac{\omega_1}{\omega_2} = 1$

(Mitre Gear)

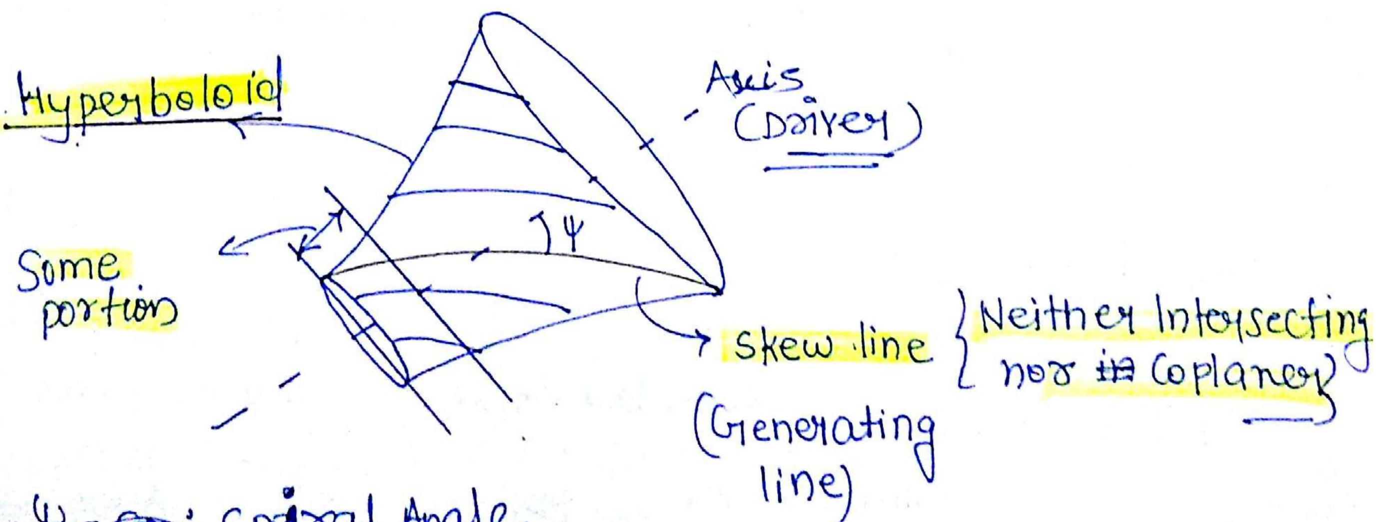
$\frac{\omega_1}{\omega_2}$ = Constant $\neq 1$

(iii) Axes Are Neither parellel nor Intersecting { Sliping -- it is a sliding in dirn of motion

- Pure rolling is impossible
- Rolling is possible

↳ (Rotation + Partial sliding)

↳ not in dirn of motion



ψ - ~~spi~~ Spiral Angle

(Skew - bevel Gears)

Spiral Gear :- Some times, when the space b/w the shaft (offset) is very less, then some portion of Hyperboloid is used to form spiral Gear called 'HYPOID Gears'

Worm and Worm wheel :-

Spiral

worm	worm wheel
• Very less dia.	• Very high dia.
• Very high spiral angle	• Very less spiral Angle

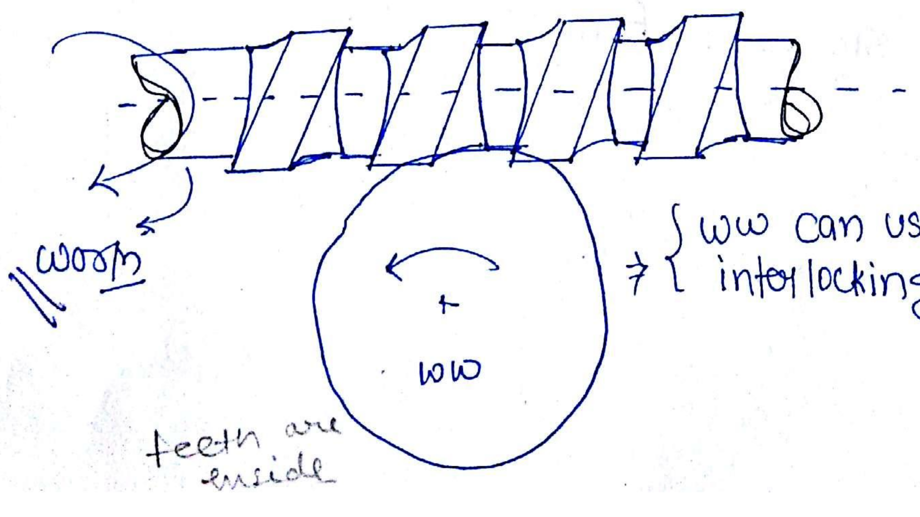
that's why it can not overcome static friction {Self locking}

due to less spiral angle w.w. can't turn worm due to static friction

⇒ Driver → worm (Always)

Famous :- For very high speed reduction

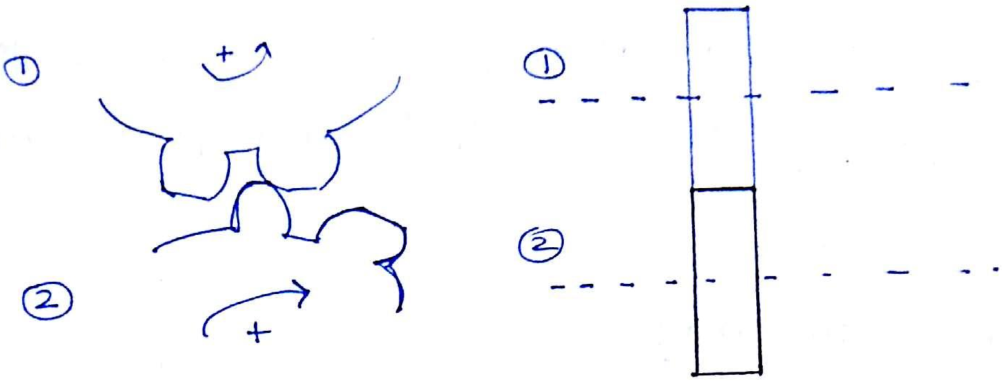
W : WW ⇒ 10:1, 30:1, 300:1, 1000:1, 1250:1



⇒ { ww can used for self interlocking

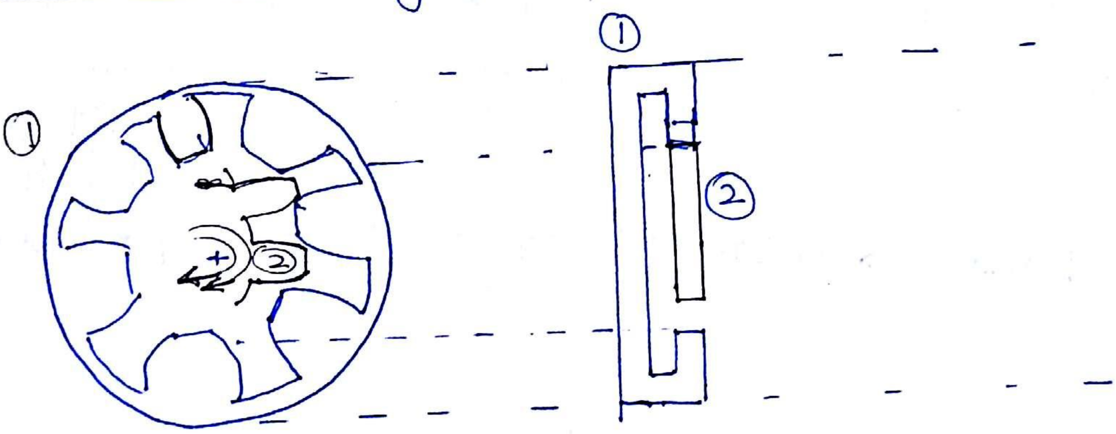
[B] According to the type of Gearing:-

- External Gearing:-



Bigger : Gear } opposite direction
Smaller : Pinion

- Internal Gearing:-



Bigger : Annular (Rins) } Same direction
Smaller : Pinion

Note:-

1. If ~~one~~ more than one gears are mounted one the same shaft

   ⇒ compound gear

   ⇒ Speed same

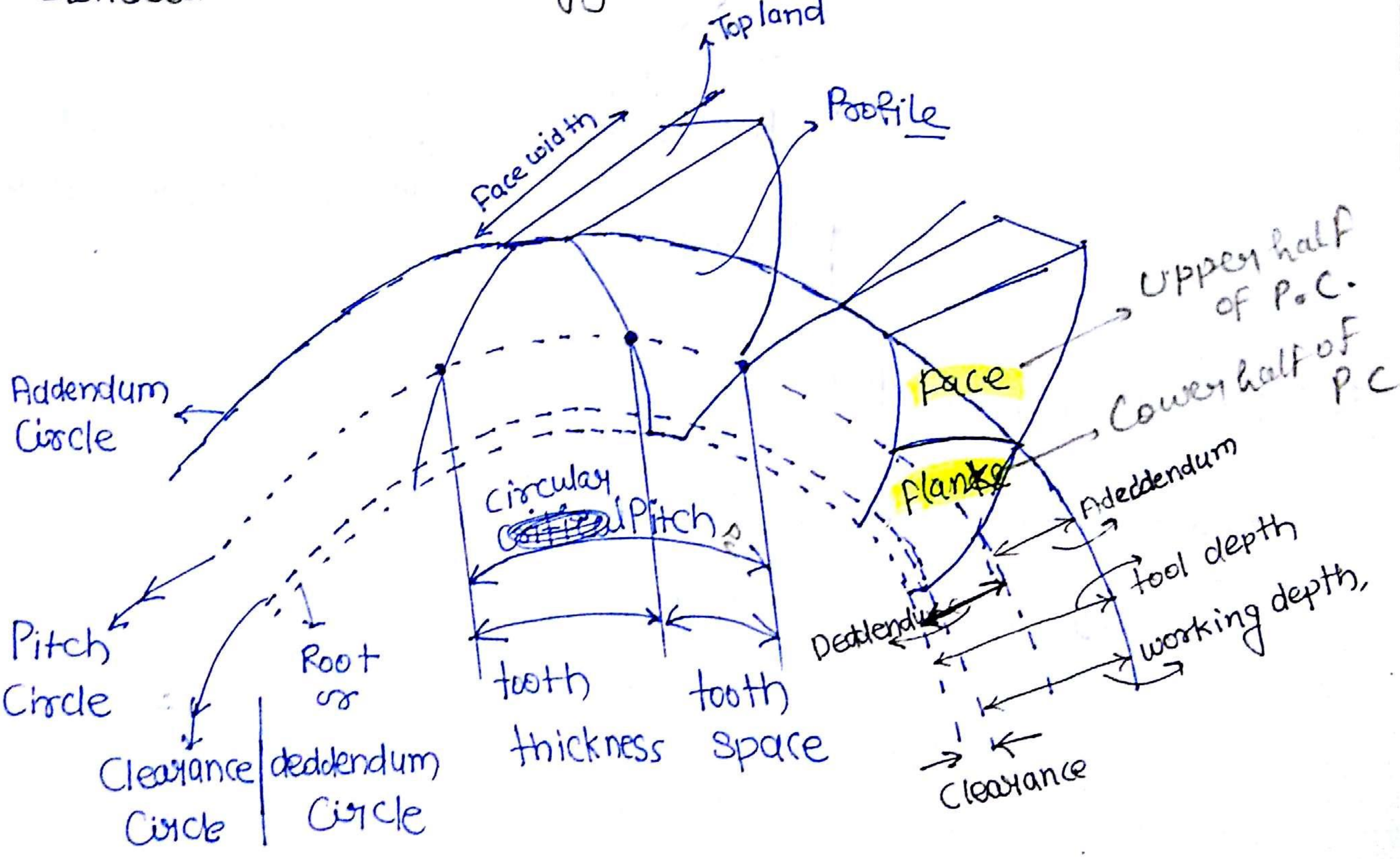
2. Generally in power transmission smaller bodies are made as Driver Pinion

$$\boxed{\text{Power} \quad P = T \times \omega}$$

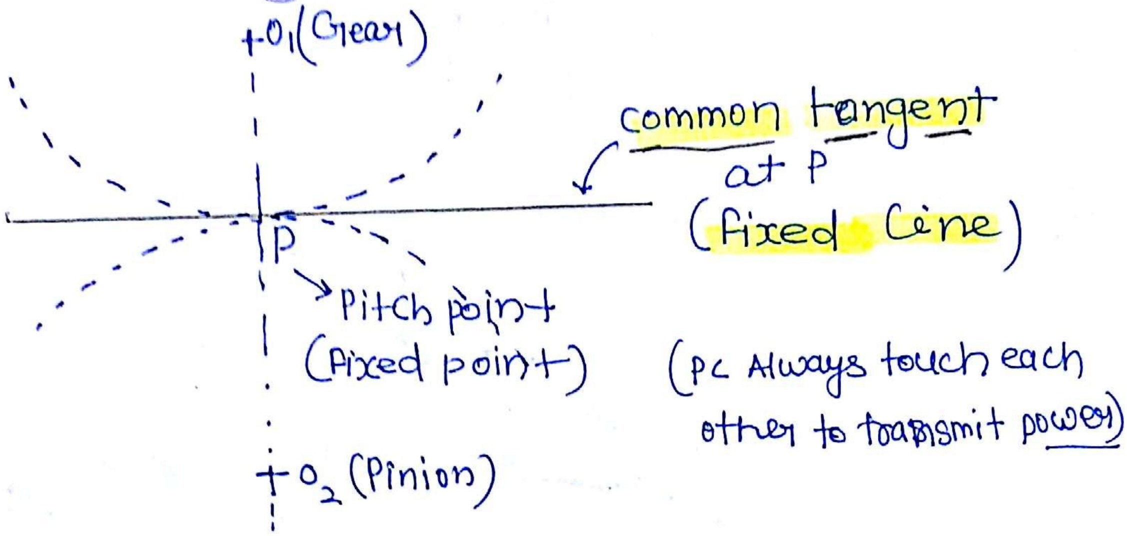
ω → High for smaller bodies

T → less Torque required

General ~~General~~ Gear terminology:-



1) Pitch Circle:- It is an imaginary circle in Gear where the pure rolling observe when the metting gear are transmitting power being an imaginary circle it can not be the physical characteristic of gear but being the most important circle, it is one of the biggest specification of the gear. The size of gear is specified by the diameter of pitch circle.



② circular Pitch (P_c):

if pitch circle dia (PCD) = D

No. of teeth = (T) = T

$$\boxed{P_c = \frac{\pi D}{T}}$$

For two mating Gears $P_{c_1} = P_{c_2}$

$$\frac{\pi D_1}{T_1} = \frac{\pi D_2}{T_2} \Rightarrow \boxed{\frac{D_1}{T_1} = \frac{D_2}{T_2}}$$

3. Module (m) :- always in mm

$$m = \frac{D\ (mm)}{T}$$

For two mating gear $m_1 = m_2$

4. Diametral Pitch :- (P_d)

$$\boxed{P_d = \frac{T}{D}}$$

$$P_c \cdot P_d = \frac{\pi D}{T} \times \frac{T}{D}$$

$$\boxed{P_c \cdot P_d = \pi}$$

5. Working Depth

By one body (Gear) = (Addendum + dedendum − clearance)

By mating Gear = sum of Addendum of both Gears.

* Tooth space − tooth thickness of mating Gear

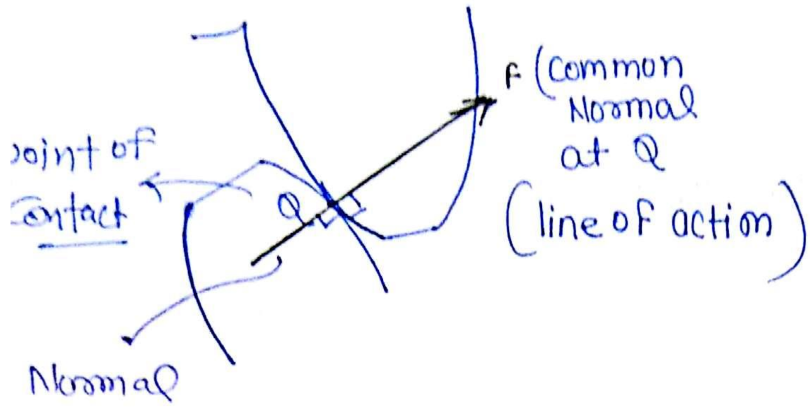
$\underbrace{\text{Tooth space − tooth thickness}}_{\text{Backlash}}$

Backlash

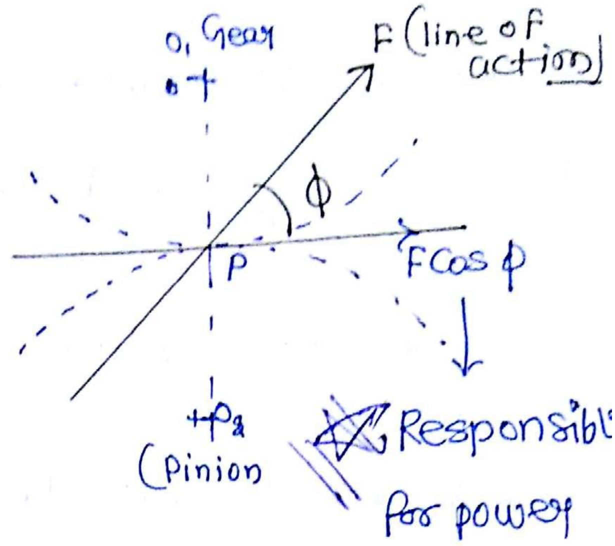
↳ To prevent the jamming of teeth due to thermal expansion.

↳ Due to sliding thermal expansion so backlash

Pressure Angle :-



Normal on profile



ϕ = pressure angle

= Angle between the line of action & common tangent at P. (pitch circle)

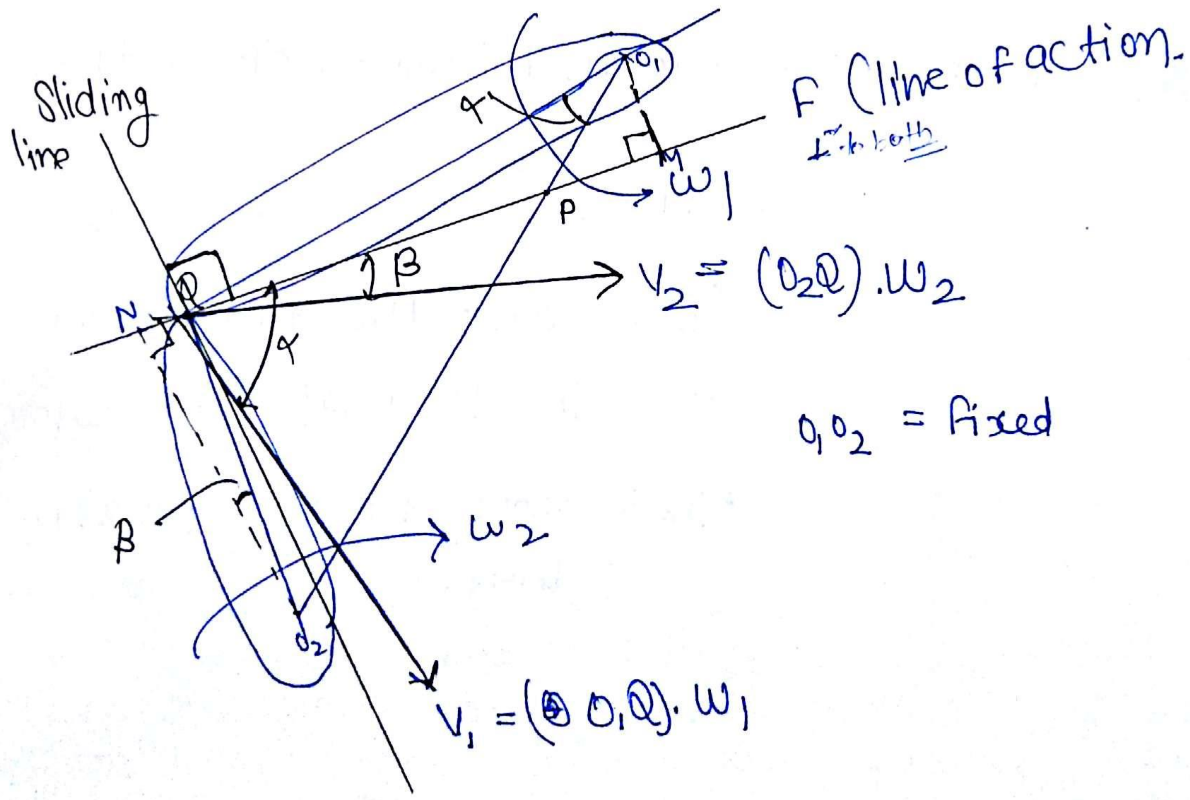
$\phi \rightarrow \max [20^\circ, 25^\circ]$

$\phi \uparrow,\ \cos\phi \downarrow,\ \Rightarrow F\cos\phi \downarrow$

* ~~[illegible]~~ point.

* Line of action always pass from P.

Law of Gearing :-



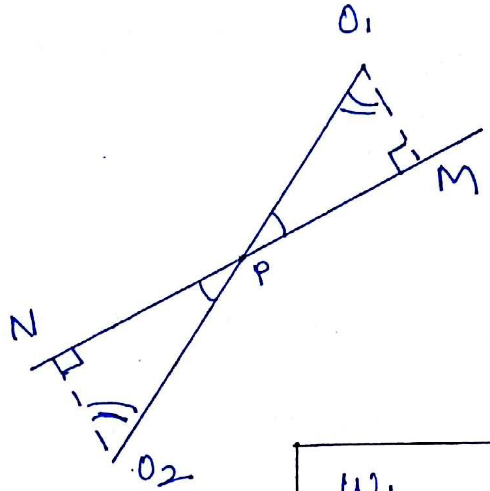
O_1O_2 = fixed

for proper Contact

$$V_1 \cos\alpha = V_2 \cos\beta$$

$$(O_1Q)\cdot\omega_1 \cdot \frac{O_1M}{O_1Q} = (O_2Q)\,\omega_2 \,\frac{O_2N}{O_2Q}$$

$$\frac{\omega_1}{\omega_2} = \frac{O_2N}{O_1M} \quad \text{--- (1)}$$



$\Delta O_1PM \sim \Delta O_2PN$

$$\boxed{\frac{\omega_1}{\omega_2} = \frac{O_2N}{O_1M} = \frac{O_2P}{O_1P} = \frac{PN}{PM}}$$

* If these two bodies are Gear.

$$\frac{\omega_1}{\omega_2} = \text{Constant}$$

$$\Rightarrow \frac{O_2P}{O_1P} = \text{Costant}$$

{ $O_1O_2 \Rightarrow$ Already fixed

so $P \Rightarrow$ fixed point (should be)

" Line of action must always pass through the fixed point on line joining the centre of rotating of Gear."

⇒ For the body to be Gear

↳ line of Action must always pass

through P

Common Normal at Q on both ~~bodies~~ of profile.

that's why almost 90% cost goes to profile making

" Mating profile must be designed in such a way such that always law of Gearing is satisfied"

↳ These are Called Conjugate profile

① Involute

② Cycloidal

③ Conjugate profile

Velocity of Sliding :-

$$V_{sliding} = |V_1 \sin\alpha - V_2 \sin\beta| = \left|O_1Q \cdot \omega \cdot \frac{QM}{O_1Q} - (O_2Q)\omega_2 \frac{QN}{O_2Q}\right|$$

$$= |\omega_1 (QP + PM) - \omega_2 (PN - QP)|$$

$$= |\omega_1 \cdot QP + \cancel{\omega_2} \cdot PM - \cancel{\omega_2} \cdot PN + \omega_2 QP|$$

$$\boxed{V_{sliding} = (\omega_1 + \omega_2) \cdot QP}$$

$$\boxed{V_{sliding} = (\omega_1 + \omega_2) \cdot QP}$$

Hence when $QP = 0$ ~~line of action~~ once Q pass through P. That is Condition of pure rolling (No sliding) → pitch circle

$QP = 0$ No sliding (Imaginary & pure rolling)

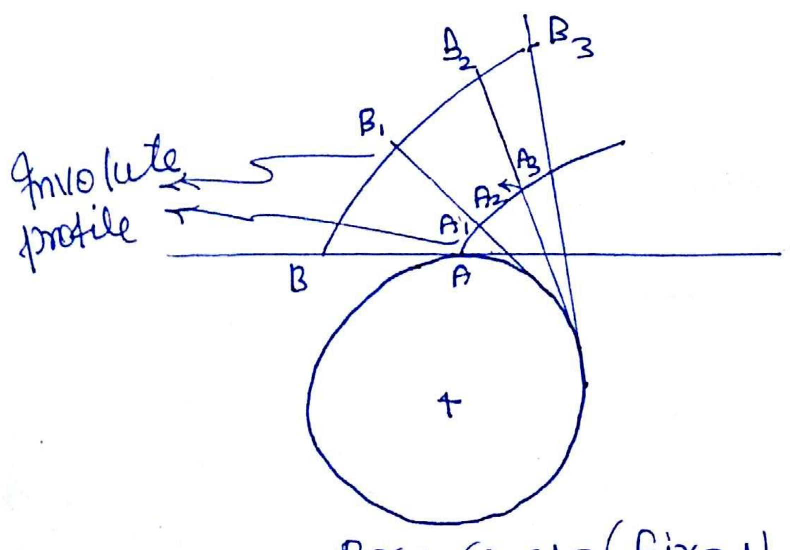
$QP \neq 0$ slinding → thermal expansion → Backlash require

Conjugate Profile

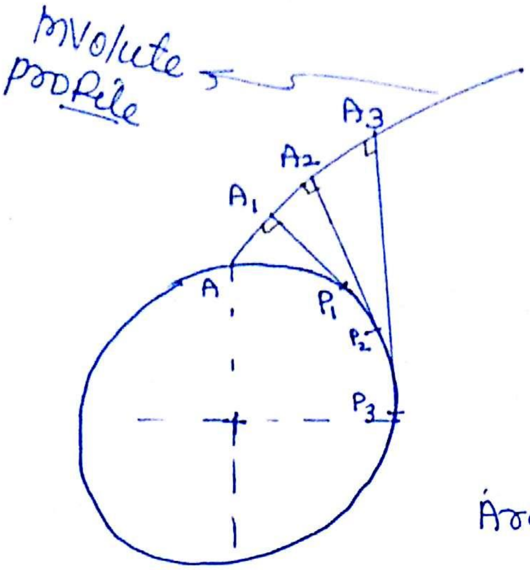
Involute Profile :- (By nature Conjugate)

"It is defined as the locus of the point on the line which rolls without sliping on the fixed circle."

This fixed circle which is basically the Generator of involute profile is known as base circle.



Base Circle (Fixed)

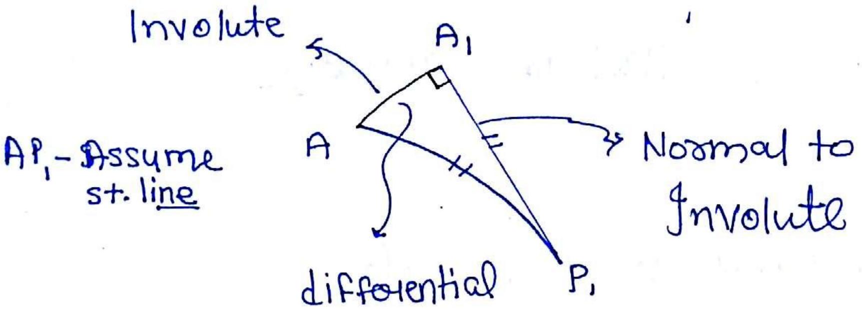


In reality

$$\left.\begin{array}{l} AP_1 \\ P_1P_2 \\ P_2P_3 \\ \vdots \end{array}\right\} \rightarrow 0$$

differential

$$\text{Arc}(AP_1) = P_1A_1$$
$$\text{Arc}(AP_2) = P_2A_2$$
$$\text{Arc}(AP_3) = P_3A_3$$
$$\vdots \qquad \vdots$$

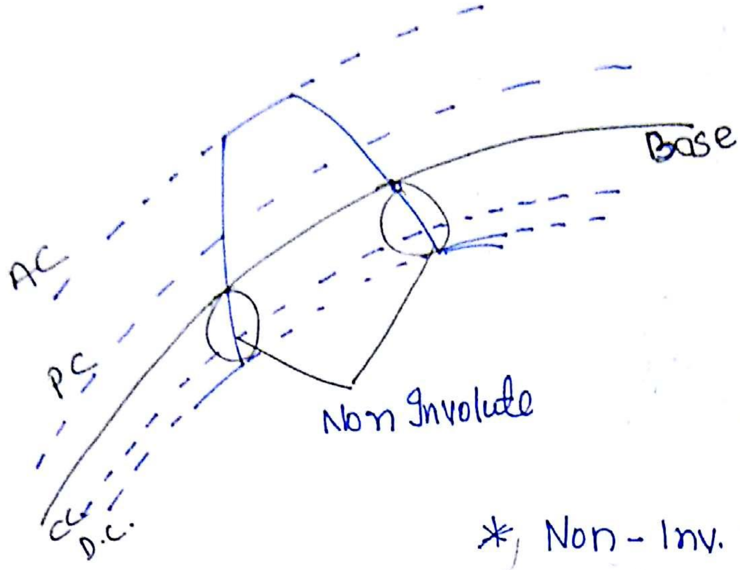


position of Circle
having center P_1 & Radius P_1A_1

" Normal drawn at any point on Involute curve will become tangent to its base circle automatically"

$$Inv(\theta) = (\tan\theta) - (\theta)\frac{\pi}{180} \qquad \theta \rightarrow deg.$$

Actual position of Base Circle in an involute Gear.



In external Gear
if $r_{base} \downarrow \Rightarrow \phi \uparrow$
↓
limitation
$[20°, 25°]$

* Non-Inv. portion in external Gear can never be eliminated
bcoz to eliminate this r_{base} कम करनी पड़ेगी तो ϕ बढ़ जायेगा $[20°, 25°]$

Velocity Ratio :-

$$\boxed{\frac{\omega_G}{\omega_P} = \frac{t}{T} < 1}$$

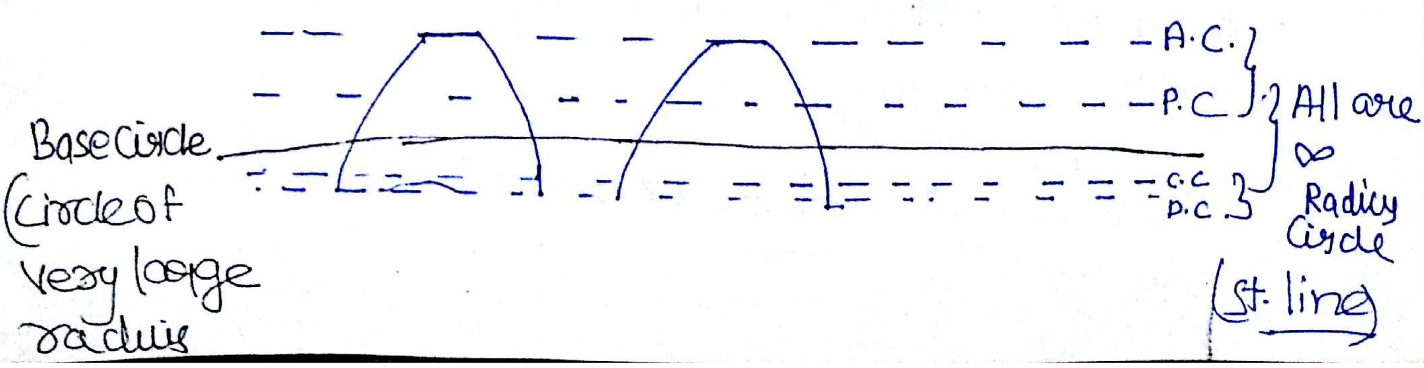
$$\boxed{\frac{\omega_P}{\omega_G} = \frac{T}{t} > 1}$$

Gear - Bigger
Pinion - small

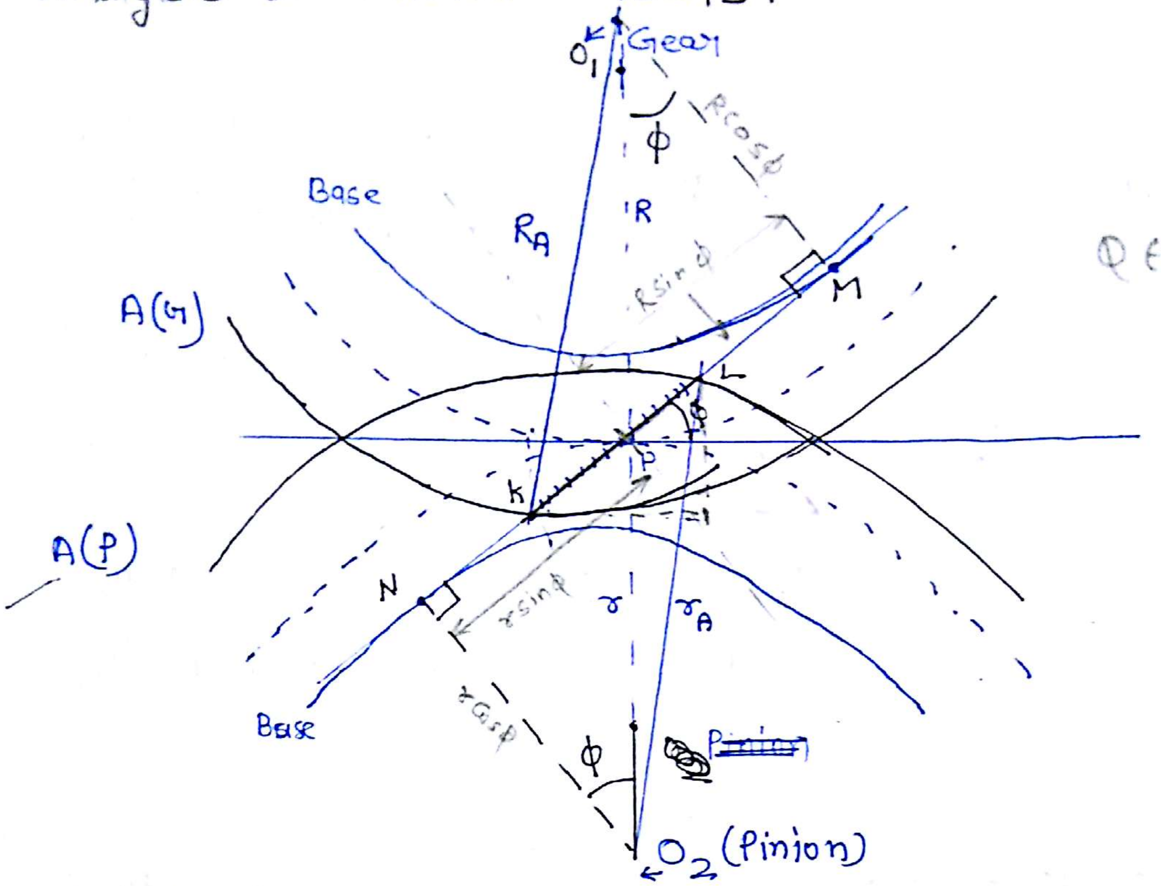
Gear Ratio :- (G)

$$\boxed{G = \frac{T}{t}} > 1 \text{ (Always)}$$

Rack :- Gear of ∞ P.C.D. ⇒ Biggest Gear.



Analysis of involute Gears :-



Start of engangment : K

End of engangment : L

line of action (K-L)

(i) Pass through P ← law of Gearing

(ii) Tangent to both of the Base Circle ← Involute property

- Point of contact is changing but line of action not changing

$\Rightarrow$ $\phi \Rightarrow$ Constant

(Q)

- Point of Contact is travelling along line of action :

  ⇒ locus of Q → straigh line

- The time interval in which Q is travelling from start to end of engangment is "one engangment period"

And distance travelled by the Q in this period is

$$\underbrace{KL}_{\boxed{\text{Path of Contact}}} = \underbrace{KP}_{\text{Path of approach}} + \underbrace{PL}_{\text{Path of Recess}}$$

* $O_1M = R\cos\phi$, $PM = R\sin\phi$

ΔO_1KM

$$R_A^2 = R^2\cos^2\phi + (KP + R\sin\phi)^2$$

$$KP = \sqrt{R_A^2 - R^2\cos^2\phi} - R\sin\phi$$

Similarly

$$PL = \sqrt{r_A^2 - r^2\cos^2\phi} - r\sin\phi$$

$KP + PL = KL$ (Path of Contact)

$$\boxed{\text{Path of Contact} = \left(\sqrt{R_A^2 - R^2\cos^2\phi} - R\sin\phi\right) + \left(\sqrt{r_A^2 - r^2\cos^2\phi} - r\sin\phi\right)}$$

Arc of Contact:- when the point of Contact is travelling from ~~start~~ start of engangment to end of engangment the distance travelled by the gear and pinion along their pitch circle in this period is Called Arc of Contact.

Arc of contact = travel of Pinion / Gear along their pitch circle in one engangment period

$$\text{Arc of App.} = \frac{\text{Path of App. (KP)}}{\cos\phi}$$

$$\text{Arc of Recess} = \frac{\text{Path of Recess (KL)}}{\cos\phi}$$

+

$$\boxed{\text{Arc of Contact} = \frac{\text{Path of Contact}}{\cos\phi}}$$

In one engangment period

$$\boxed{\left(\underset{\text{By Pinion}}{\text{Angle turned}}\right) = \left(\frac{\text{Arc of contact}}{r}\right) \text{rad}}$$

$$\boxed{\left(\underset{\text{By Gear}}{\text{Angle turned}}\right) = \left(\frac{\text{Arc of contact}}{R}\right) \text{rad}}$$

$$\boxed{\text{Contact Ratio} = \frac{\text{Arc of Contact}}{P_c}}$$

$P_c = \frac{\pi D}{T}$

P_c - Circular pitch.

* Contact Ratio tells No. of pairs engang in one engangment period.

Min ≥ 1

Generally [1.2, 1.8] ≈

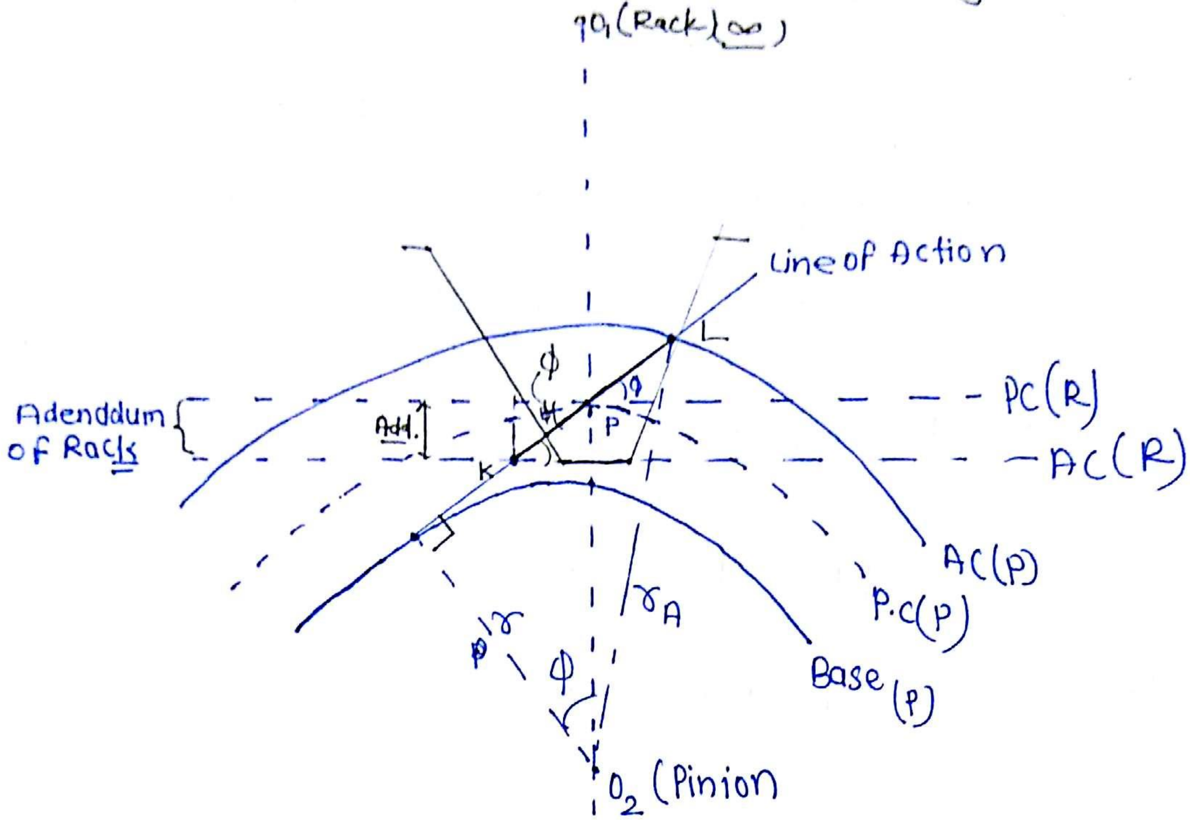
In spur Gears $\in [2, 3]$

For eg: Contact Ratio = 1.21

One pair is enganged in full engangment period But 21% time of engangment period along with this pair one more ~~m~~ pair i.e. total two pairs. enganged.

Therefore No. of pair engaged in one engangment period it avg. Value is comes out to be 1.21

Path of Contact in Rack and Pinion Arrangment:-



Path of contact

$$KL = KP + PL$$

$$\boxed{KL = \frac{\text{Add(Rack)}}{\sin\phi} + \left[\sqrt{r_A^2 - r^2\cos^2\phi} - r\sin\phi\right]}$$

IES 20M
Q.51
Page 25

$t = 24$ $\quad m = 8\,mm$ $\quad$ Add. (each) $= 7.5\,mm$.

$T = 36$ $\quad \phi = 20°$

$N_p = 450$ rpm

$\frac{N_s}{N_p} = \frac{t}{T} = \frac{24}{36} \Rightarrow N_s = 450 \times \frac{24}{36} = 300$ rpm

Gear:- $m = \frac{D}{T}$ & $R = \frac{mT}{2} = \frac{8 \times 36}{2} = 144\,mm$ $\quad R_A = 144 + 7.5 = 151.5\,mm$

pinion $m = \frac{d}{t}$, $r = \frac{mt}{2} = \frac{8 \times 24}{2} = 96\,mm$ & $r_A = 96 + 7.5 = 103.5\,mm$

Point of Contact

$$KL = KP + PL$$

$$= \left[\sqrt{R_A^2 - R^2\cos^2\phi} - R\sin\phi\right]$$

$$+ \left[\sqrt{r_A^2 - r^2\cos^2\phi} - r\sin\phi\right]$$

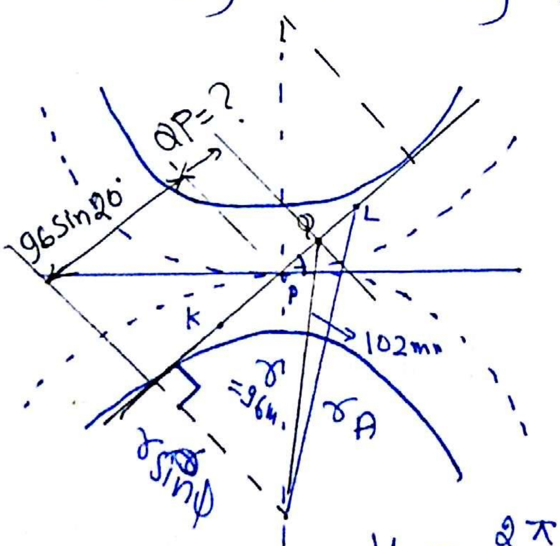
$$= \sqrt{(151.5)^2 - (144\cos 20^\circ)^2} - (144\sin 20^\circ) + \sqrt{(103.5)^2 - (96\cos 20^\circ)^2} - (96\sin 20^\circ)$$

Path of Contact $= (KL) = 36.78\,\text{mm}$?

Arc of contact $= \dfrac{KL}{\cos\phi} = \dfrac{36.78}{\cos 20} = 39.14$ mm ?

(i) (Angle turned By pinion) $= \dfrac{\text{Arc of Contact}}{r} = 0.407$ rad. ?

(ii) Velocity of sliding $V_s = (\omega_1 + \omega_2) \cdot QP = \dfrac{2\pi}{60}(300+450) \cdot QP$ & ?



$$(102)^2 = (96\cos 20^\circ)^2 + (96\sin 20^\circ + QP)^2$$

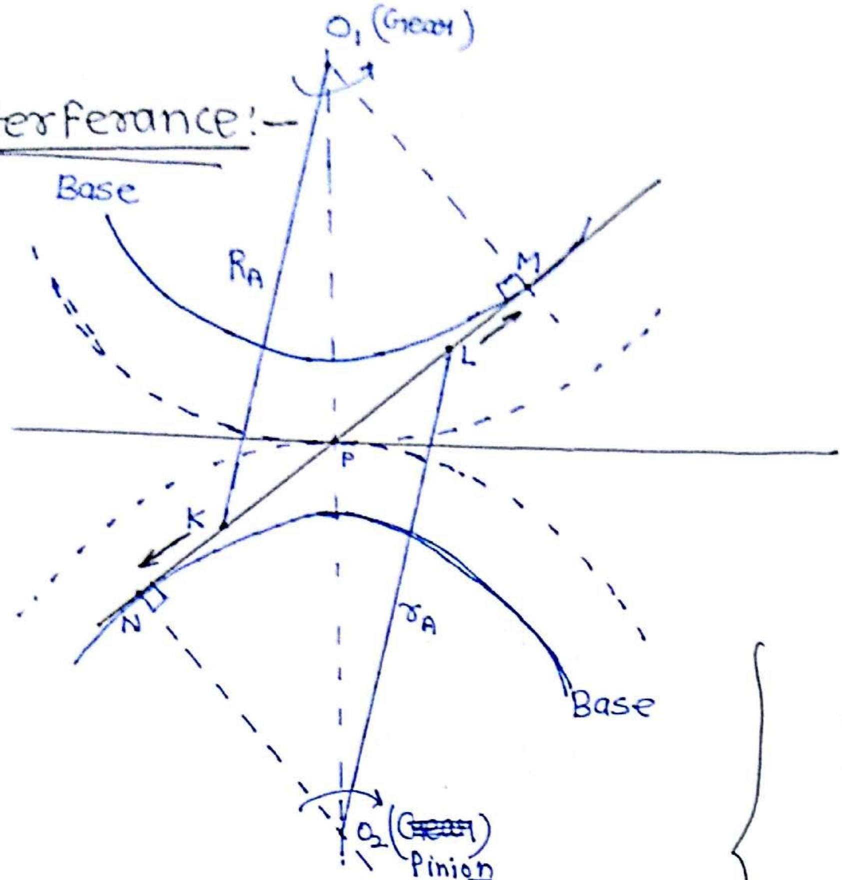
$$QP = \sqrt{(102)^2 - (96\cos 20)^2} - 96\sin 20^\circ$$

$$QP = 14.76\text{ mm}$$

$$V_s = \frac{2\pi}{60}(300+450) \times \frac{14.76}{1000}$$

$$V_s = 1.15\text{ m/s}$$

Interferance:-



Interferance

last suffer points of K and L ~~or~~ are

critical points → N M

(Interferance point)

If $r_A > O_2M$

⇒ Involute tip of pinion will touch non-Involute flank portion of Gear.

→ Involute to Non Involute connection

↳ law of Gearing is not be satisfied

↳ Involute tip of pinion will remove some material from ~~Involute~~ Non-Involute flank portion of Gear.

(This removal of the material is a process called under-cutting.

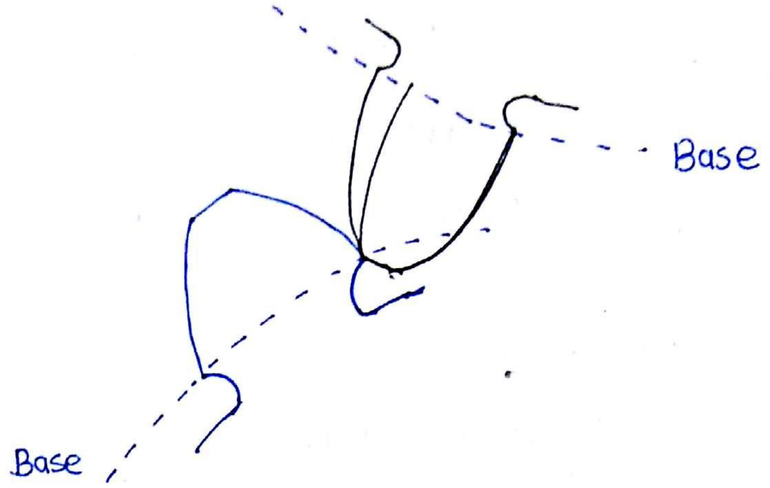
→ This Complete phinomena called "Interferance"

Similar will be there

If $R_A > O_1N$

Methods to prevent Interferance

1. Under Cut Method :-



* Under cutting is provided by the cutting tool at the time of manufacturing on non Involute portion.

↳ Interferance eliminated

But Tooth strength is less at the Root (Due to less material)

limitation :- Used in low power transmission.

In medium and high power transmission

2. $\phi \uparrow$, Pressure angle increase.

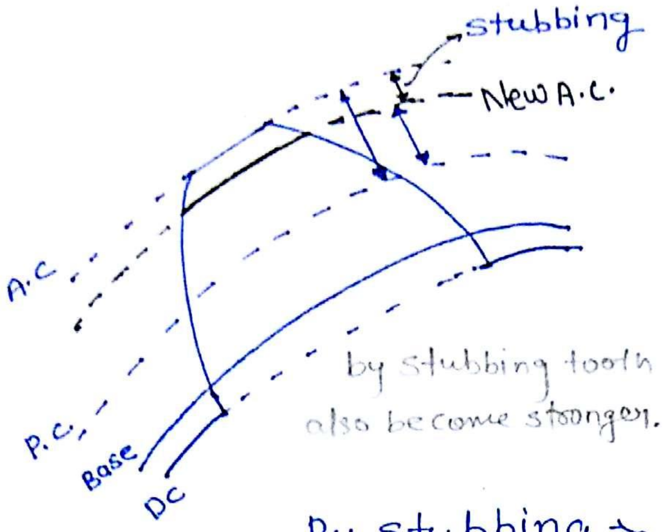
If pressure angle increase by decreasing radius of base ($r_b \downarrow$), Interferance will decrease

limitation $\phi = [20^\circ, 25^\circ]$

$\phi \uparrow \rightarrow r_b \downarrow$

↳ $(20^\circ, 25^\circ)$

3. By stubbing the teeth:-



By stubbing

$\phi \rightarrow$ No. change

But Addendum decrease

$\Rightarrow$ Addendum circle radius $\downarrow$

$\Rightarrow$ Interferance $\downarrow$

By stubbing $\Rightarrow$ Path of Contact decrease because $R_A \downarrow$

So Arc of contact will also decrease (but $\boxed{P_c = \frac{\pi D}{T}} \rightarrow$ No change)

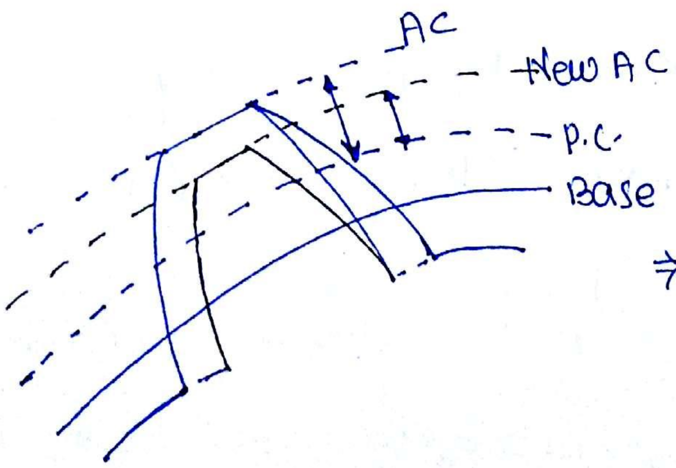
therefore $\underbrace{\text{contact ratio}}$ will decrease

$\downarrow$

Min ≥ 1

$\hookrightarrow$ limitation

4. Increase the Number of teeth:-



If teeth (T) increase

$\hookrightarrow \phi \rightarrow$ Constant

$\Rightarrow$ If $T \uparrow \Rightarrow$ Add. Cir Rad. $\downarrow$

$\Rightarrow$ Path of Contact $\downarrow$

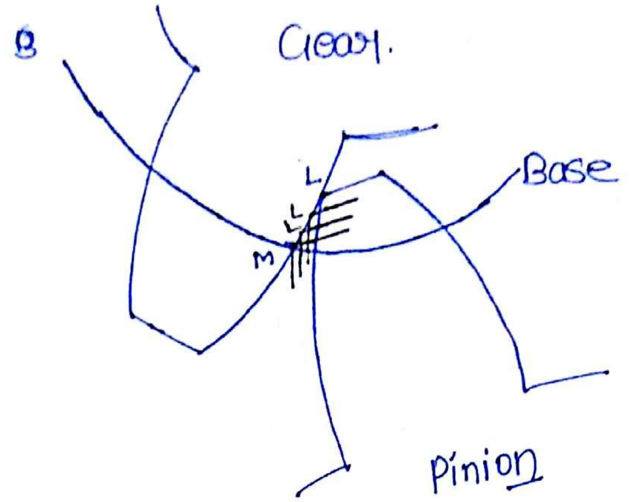
$\Rightarrow$ Arc of Contact $\downarrow$

But here $\boxed{P_c = \frac{\pi D}{\uparrow T}}$ will decrease $\downarrow$

* No limitation

So Contact Ratio will increase

For example



Note:-

A → Fractional Addendum for 1 mm module in order to, avoid Interferance

Gear A_G.
Pinion A_P
Rack. A_R

There-fore Addendum required in order to avoid interferance = mA

m - module

$\left.\begin{matrix} mA_G \\ mA_P \\ mA_R \end{matrix}\right\}$

For eg.

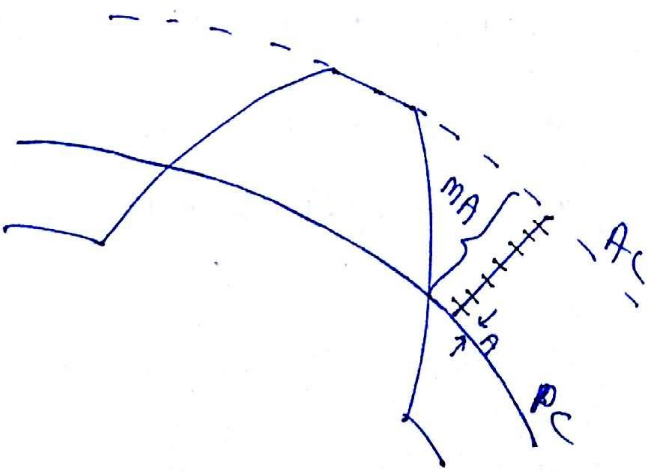
$m = 8\ mm$

Add. = 7.5 mm

$mA = 7.5$

$A = \frac{7.5}{8}$

$A = 0.9375$



Involute Gear System

Full depth Involute ($\phi = 14\frac{1}{2}°, 20°$)

Add. = standard add.

= One module value

$mA = 1m$

$\boxed{A = 1} \longrightarrow$ $A_G \to 1$, $A_P \to 1$, $A_R \to 1$

Stub Involute [20°, 25°]

Add. < standard add.

$mA < 1m$

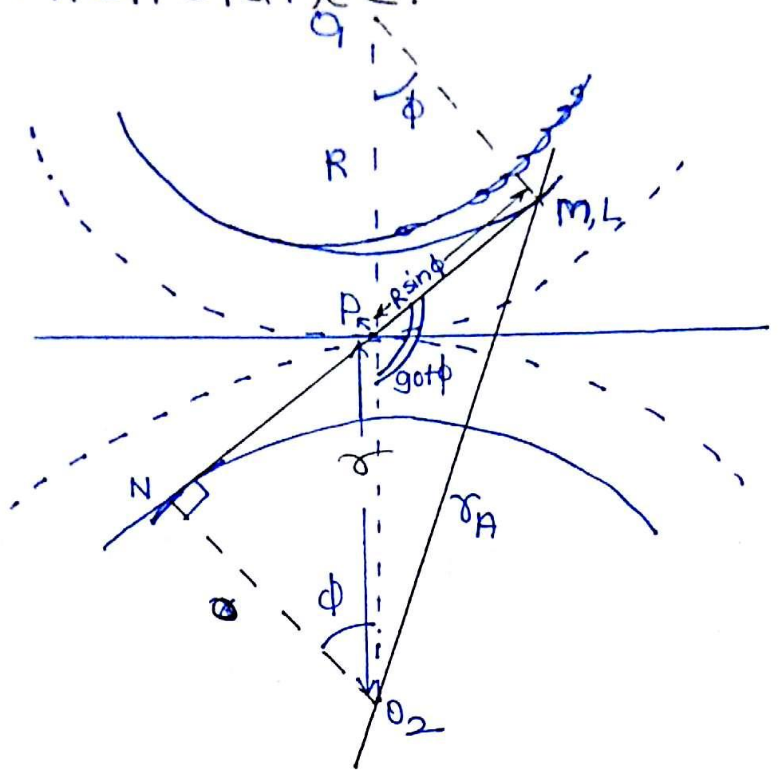
$A < 1$

$\left.\begin{array}{l} A_G \\ A_P \\ A_R \end{array}\right\} < 1$

20° Stub Involute :– (Best Gear)

1. lesser Interferance
2. min. no. of theeth req. is less
3. Cost is less
4. Stooger teeth.

Min. No. of teeth on Pinion/Gear in order to avoid interferance.



$\Delta O_2 PL$:- $\quad r_A^2 = r^2 + R^2 \sin^2\phi - 2(r)(R\sin\phi)\cos(90+\phi)$

$$= r^2 + (R^2 + 2rR)\sin^2\phi$$

$$r_A^2 = r^2\left[1 + \frac{R}{r}\left(\frac{R}{r}+2\right)\sin^2\phi\right]$$

$$r_A = r\sqrt{\left[1 + G(G+2)\sin^2\phi\right]}$$

$(\text{Addendum})_{\text{pinion}} = r_A - r$

$$m A_p = r\sqrt{\left[1+G(G+2)\sin^2\phi\right]} - r$$

$$m = \frac{2r}{t} \quad \left[\because m = \frac{D}{T}\right]$$

$$\frac{2r}{t} A_p = r\sqrt{1+G(G+2)\sin^2\phi} - r$$

$$t_{min} = \frac{2A_p}{\left[\sqrt{1+G(G+2)\sin^2\phi} - 1\right]}$$

$$\boxed{t_{min} = \frac{2A_p}{\left[\sqrt{1+G(G+2)\sin^2\phi} - 1\right]}}$$

minimum No. of teeth require to Avoid interferance for pinion.

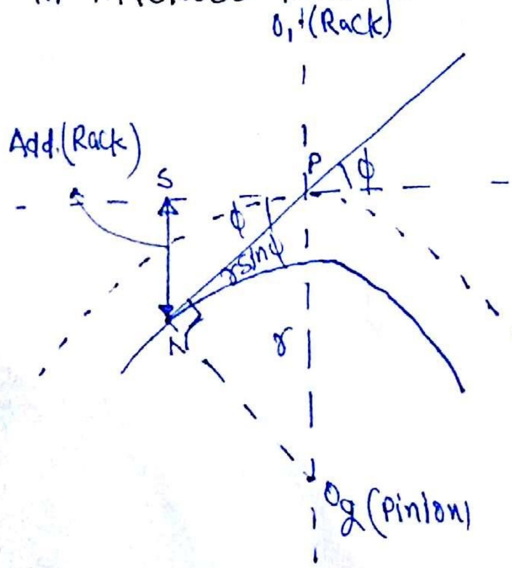
Similarly

$$T_{min} = \frac{2A_G}{\left[\sqrt{1+\frac{1}{G}\left(\frac{1}{G}+2\right)\sin^2\phi} - 1\right]}$$

For Gear.

24/10/2016

Minimum Number of teeth on the pinion to avoid Interferance in Involute Rack and pinion arrangment.



Δ SPN

$$\sin\phi = \frac{\text{Add (Rack)}}{r\sin\phi}$$

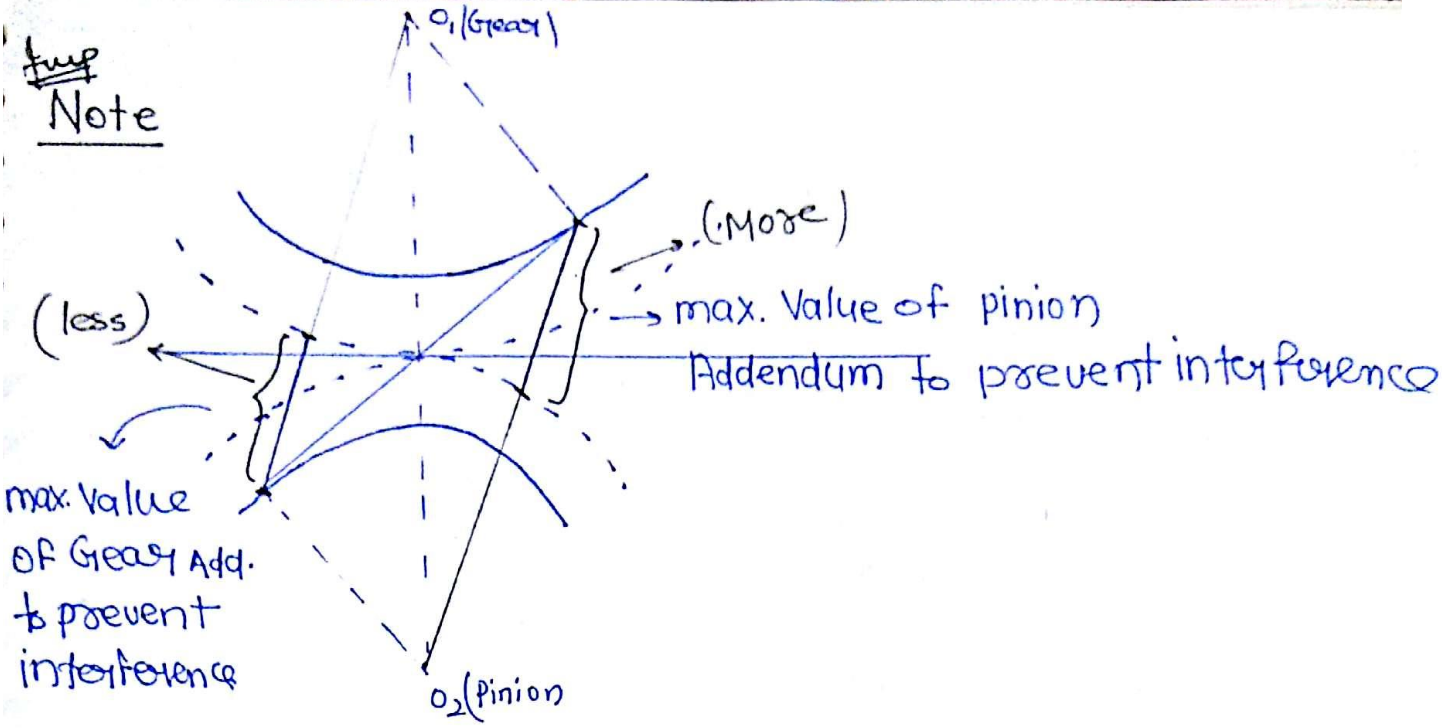
$$\text{Add (Rack)} = r\sin^2\phi = mA_R$$

$$\frac{m\,t_{min}}{2}\sin^2\phi = mA_R$$

$$t_{min} = \frac{2A_R}{\sin^2\phi}$$

Imp

Note



Gear Design

(i) If Gear / Pinion having same value of Addendum

- First Gear must be safe

$$T_{min} = \frac{2A_G}{\sqrt{1+\frac{1}{G}\left(\frac{1}{G}+2\right)\sin^2\phi}-1}$$

- Pinion will automatically safe

- $t_{min} = \dfrac{T_{min}}{G}$ — Check this

eg $G = 3$

$T_{min} = 28$

$t_{min} = \frac{28}{3} = 9.03$
$= 10$

$G = \frac{28}{10} = 2.8$ ✗

$T_{min} = 30$, $t_{min} = 10$ $\Big\} G = 3$ ✓

(ii) If Gear / Pinion Having different Addendum (3 Condⁿ Must be satisfied)

- $T_{min} = \dfrac{2A_G}{\sqrt{1+\frac{1}{G}\left(\frac{1}{G}+2\right)\sin^2\phi}-1}$

- $t_{min} = \dfrac{2A_p}{\sqrt{1+G(G+2)\sin^2\phi}-1}$

- $G = \dfrac{T_{min}}{t_{min}}$

For eg if $G = 3$ (Give)

$T_{min} = 39.4 \rightarrow 40$

$t_{min} = 12.6 \rightarrow 13$

$G = \frac{40}{13} = 3.07$ ✗

So $T_{min} = 42$, $t_{min} = 14$ $\Big\}$ Minimum

$G = \frac{42}{14} = 3$ (satisfied)

Q.52

$G = 3$

$A_p = A_G = 1$, $\phi = 20°$

$t_{min} = ?$

A_p & A_G same so we will safe Gear first

$$T_{min} = \frac{2A_G}{\sqrt{1+\frac{1}{G}\left(\frac{1}{G}+2\right)\sin^2\phi} - 1} = \frac{2\times 1}{\sqrt{1+\frac{1}{3}\left(\frac{1}{3}+2\right)(\sin 20)^2} - 1}$$

$$= 44.94$$

$$\boxed{T_{min} = 45}$$

$$t_{min} = \frac{45}{3} = \underline{15} \text{ teeth.}$$

Now $t_{min} = 15 - 3 = 12$

$G = 3$ (same)

$T_{min} = 12 \times 3 = \underline{36}$

Stubbing

$$T_{min} = \frac{2A_G}{\sqrt{1+\frac{1}{G}\left(\frac{1}{G}+2\right)\sin^2\phi} - 1} \Rightarrow 36 = \frac{2A_G}{\sqrt{1+\frac{1}{3}\left(\frac{1}{3}+2\right)(\sin 20)^2} - 1}$$

$A_G = 0.801$

So $\underline{\underline{20\%}}$ stubbing Require

If $A_G = \text{same} = \underline{1}$

$T_{min} = 36$

$\phi = ?$

$$= 36 = \frac{2A_G}{\sqrt{1+\frac{1}{G}\left(\frac{1}{G}+2\right)\sin^2\phi} - 1} \Rightarrow 1+\frac{1}{3}\left(\frac{1}{3}+2\right)\sin^2\phi = \left(\frac{19}{18}\right)^2$$

$$\phi = 22.53°$$

~~$\phi = 10.84$~~ ?

Q. 50

$\phi = 20^\circ \quad m = 10\,mm, \quad A_p = A_G = 1$

$\left\{ \begin{array}{l} T = 50 \\ t = 13 \end{array} \right. \Rightarrow G = \frac{50}{13}$

(i) $T_{min} = \dfrac{2A_G}{\sqrt{1+\frac{1}{G}\left(\frac{1}{G}+2\right)\sin^2\phi} - 1} = \dfrac{2 \times 1}{\sqrt{1+\frac{13}{50}\left(\frac{13}{50}+2\right)(\sin 20)^2} - 1}$

$= 59.17$

$T_{min} = 60$ (Yes Interferance will occurs.

(ii) $T_{min} = 50 = \dfrac{2A_s}{\sqrt{1+\frac{1}{G}\left(\frac{1}{G}+2\right)\sin^2\phi} - 1} = \dfrac{2 \times 1}{\sqrt{1+\frac{13}{50}\left(\frac{13}{50}+2\right)(\sin\phi)^2} - 1}$

?

$1+\frac{13}{50}\left(\frac{13}{50}+2\right)\sin^2\phi = \left(\frac{26}{25}\right)^2 \Rightarrow \phi = 21.87$

Pb. if $G = 4, \quad A_p = A_G = 1, \quad \phi = 20^\circ$

$T_{min} = ?$

$T_{min} = \dfrac{2A_G}{\sqrt{1+\frac{1}{G}\left(\frac{1}{G}+2\right)\sin^2\phi} - 1} = \dfrac{2 \times 1}{\sqrt{1+\frac{1}{4}\left(\frac{1}{4}+2\right)\sin^2 20} - 1}$

$= 61.77$

$T_{min} = 62$ ✗

$t_{min} = \frac{62}{4} = 15.5$
$= 16$

~~G = 16~~ $T_{min} = 16 \times 4 = 64$ Au

* Effect of Variation of Centre distance due to vibration on the performance of Involute Gear.

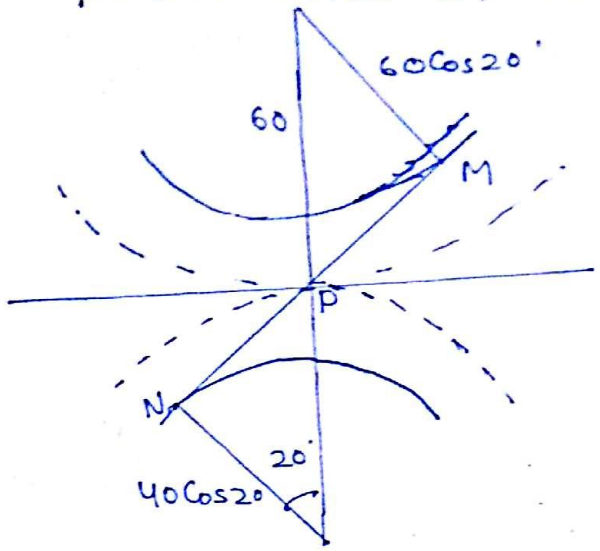
For eg.

Centre distance = 100 mm

$R = 60$ mm

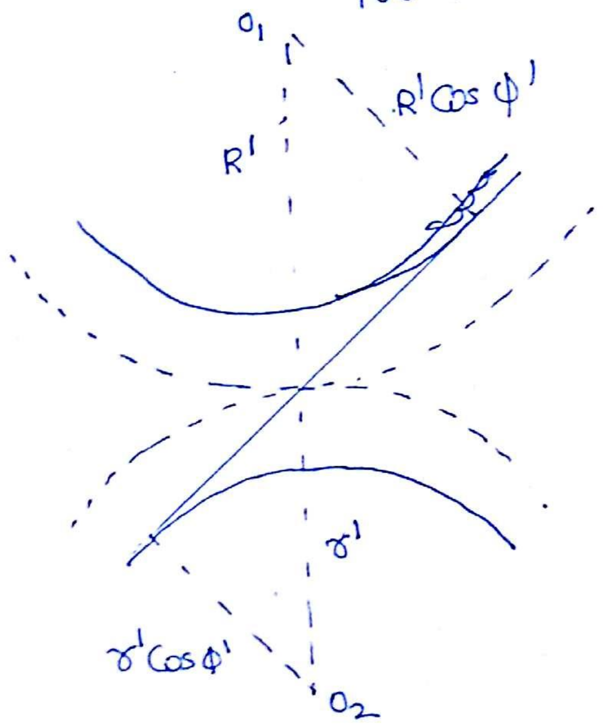
$r = 40$ mm

$\phi = 20^\circ$



At a moment let centre distance is ↑ by 2%.

100 mm → 102 mm {due to Vibration}



$$R'\cos\phi' = 60\cos 20^\circ$$

$$r'\cos\phi' = 40\cos 20^\circ$$

$$(R'+r')\cos\phi' = (60+40)\cos 20^\circ$$

$$R' + r' = 102$$

$$\cos\phi' = \frac{100\cos 20^\circ}{102}$$

$$\phi' = 22.88$$

$R' =$ $\qquad$ $r' =$

Due to Vibration

- Centre distance changing
- P.C. changing Pitch circle
- P changing
- ϕ changing

$\Big\}$ Close to zero (But $\neq 0$ practically)

Imp $\boxed{r_{Base} \rightarrow \text{No Change}}$

Radius of Base Circle will never changes in the case of involute teeth.

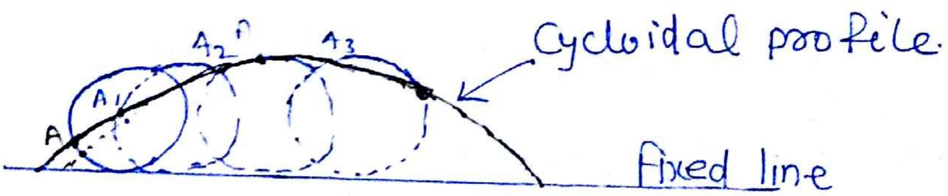
$$\frac{\omega_1}{\omega_2} = \cancel{\frac{O_2M}{O_1M}}\ \frac{O_2N}{O_1M}$$ Base radii

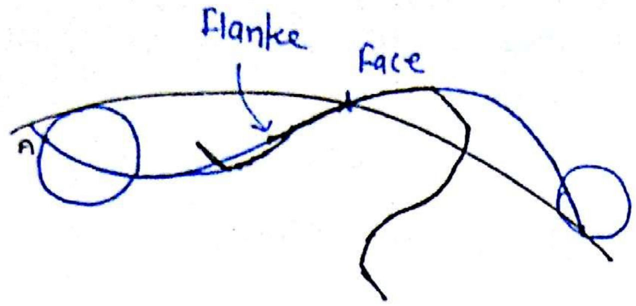
Velocity Ratio No change

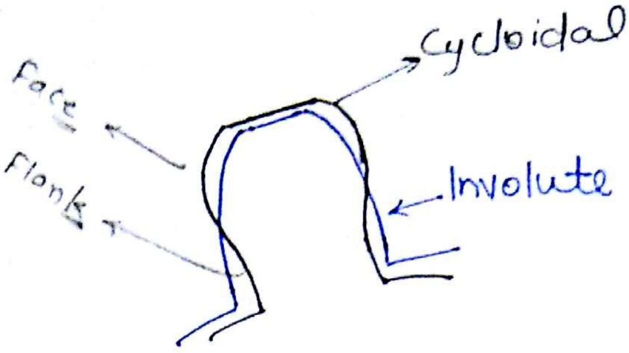
Important thing $\boxed{\frac{\omega_1}{\omega_2} = \frac{O_2N}{O_1M}}$

Cycloidal Profile (By nature Conjugate)

"It is defined as the locus of the point on the circumferance of the circle which rolls without sliping on fixed ~~line~~ straight line"







(1) Per tooth cost is more but overall cost of Gear is less.

Interferance - Absent

(2) Flank wide (stronger teeth)

(3) concave - convex connection → Very high life due to less wear.

(4) ϕ (pressure angle changing)

→ ϕ_{max} → At start of engangment

→ ϕ → 0 At pitch point

→ ϕ_{max} → At end of engangment

↳ (But in reverse dir^n)

Drawback :- $\frac{\omega_1}{\omega_2}$ changing so we don't use these Gears due to vibration

Note:-

Normal thrust between the teeth $= F$
Along line of Action.

tangential force Component $= F \cos\phi$
(power Component)
Along line Normal to profile

Radial force component $= F \sin\phi$

Torque $= T = (F \cos\phi \times \text{Pitch Circle Radius})$

Power $= P = T \times \omega = T \times \dfrac{2\pi N}{60}$

$$P = \frac{2\pi N T}{60} \quad \text{watt}$$

Efficiency $\eta = \dfrac{P_o}{P_i} = \dfrac{T_o \times \omega_o}{T_i \times \omega_i}$

Note:- the power component is 95% of normal thrust b/w teeth

$$F \cos\phi = 0.95\, F$$

$$\cos\phi = 0.95$$