teeth.

Advantages of cycloidal teeth –

1. ~~first~~ In cycloidal gear the contact takes place b/w a convex flank & concave surface this result in less wear in cycloidal gear as compared to involute gear.

2. In cycloidal gear teeth having wide flank. therefore the cycloidal gear are stronger than involute gear for the same pitch.

3. In cycloidal gears the interference does not occur at all.

Disadvantages of cycloidal teeth :-

1. Pressure angle is max. at begining of engagement, reduce to zero at pitch point, start increasing and again became max. at the end of engagement. there result is less smooth running of gear.

2. In cycloidal gear double curve are required for the face or flank (whereas in involute teeth only one curve required)

3. if the centre distance of a pair of cycloidal gear varry then change velocity ratio.

⇒ System of gear teeth :-

~~An involute of circle is defined as the locus of the point of the line which rolls without sleeping on the fixed circle & this fixed circle which is the generator of involute profile is known as base circle. The involute is~~

⇒ System of gear teeth :-

The following four system of gear teeth are commonly used in practice :-

1) $14\frac{1}{2}°$ composite system.

ii) $14\frac{1}{2}°$ full depth involute system.

iii) $20°$ full depth involute system

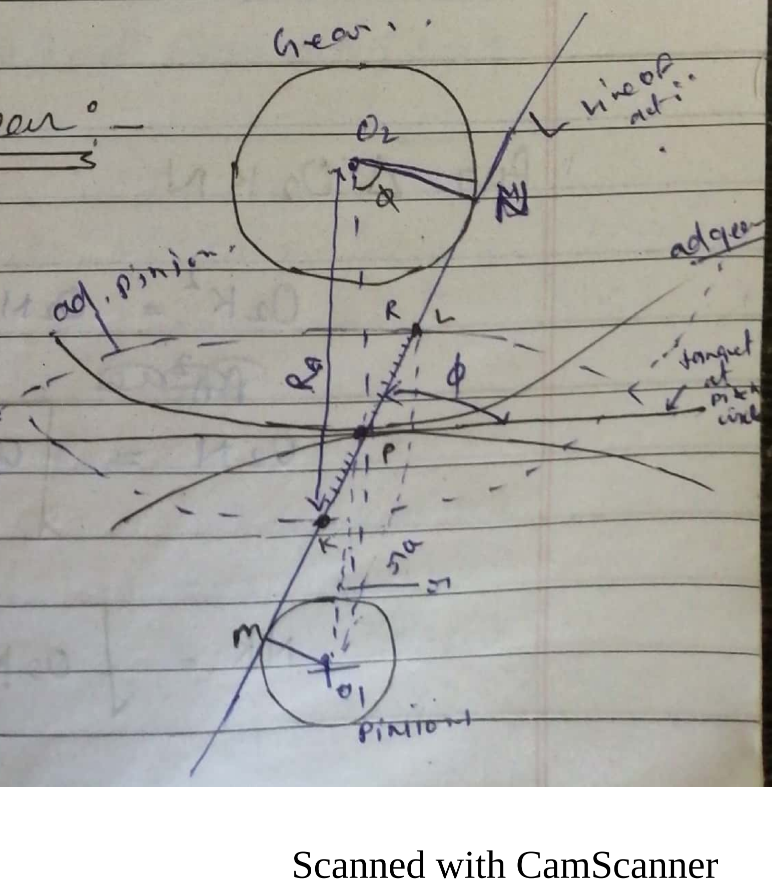
iv) $20°$ stub involute system.

Scanned with CamScanner

The $14\frac{1}{2}°$ composite system is used for general purpose of gear. It is stronger but has no chance of interchangebility. The tooth profile system have cycloidal curve at the top & bottom & involute curve at middle portion. The teeth are produced by formed milling cutter. The tooth profile of the $14\frac{1}{2}$ of full depth involute system was developed for the be with gear hub for spur & helical gear.

The tooth profile of the 20° full depth involute system may be cut by hubs. That increase the pressure angle from $14\frac{1}{2}°$ to 20° resulting strong teeth. Because the tooth aiting as a beam as ~~is~~ wider at the ~~dise~~ base. The 20° stub. involute system has a stronger tooth to nare to take heavy loads.

⇒ **Analysis of Involute gear:-**



Note:-

1. Point of contact is changing but line of action not change. hence pressure angle ϕ constant.

2. Point of contact travelling along the line of action.

3. The time travelling which Q is travelling from start to end of engagement is known as one engagement period.

And the distance travel by the in this period is known as path of contact (KL)

$$KL = KP + PL$$

where

KP - path of approach

PL - path of Races

for ΔO_2KN

$$O_2K^2 = O_2N^2 + NK^2$$

$$O_2N = \sqrt{O_2K^2 - NK^2}$$

$$NK = \sqrt{O_2K^2 - O_2N^2}$$

$$KP + PN = \sqrt{O_2K^2 - O_2N^2}$$

$$KP = \sqrt{O_2K^2 - O_2N^2} - PN$$

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

KP - path of approach.
RA - attendum Radios of gear.
R - pitch circle Radius of gear.

ΔO_1ML **Similiarly** :-

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

$$\boxed{\text{Arc of contact} = \text{path of contact} / \cos\phi}$$

$$\boxed{\text{Arc of approach} = \text{Path of Approach} / \cos\phi}$$

$$\boxed{\text{Arc of Races} = \text{path of races} / \cos\phi}$$

$$\text{contact Ratio} = \frac{\text{Arc of contact}}{\text{Pitch circle } (P_c)}$$

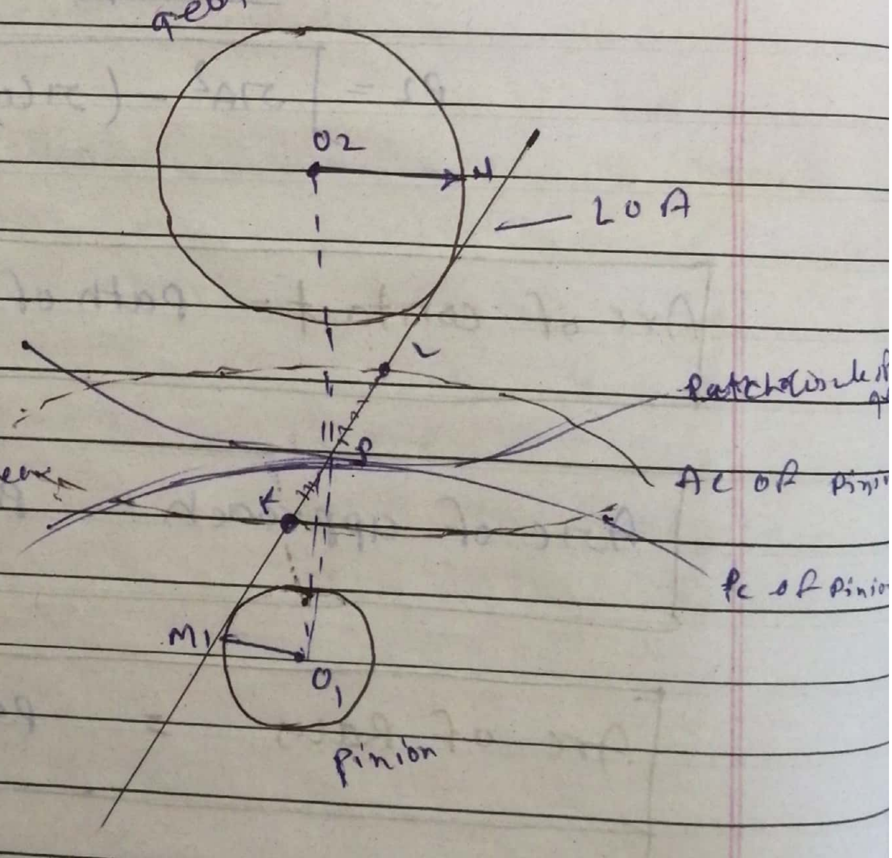
$$\boxed{P_c = \frac{\pi D}{T} = \pi m}$$

for e.g contact ratio is 1.28 —

One teeth is engage in full engagement period but 28% time of engagement period along with this pair. One more pair is engage therefore none of pair engage in one engagement period. its avg value comes out to be 1.28.

⇒ Interference in involute gears —

Interference in



if the radius of the addendum circle of pinion is increase to O_1N, the point of length contact (L) will move from L-N when this radius is further increase the point of contact (L) will be on the inside of base circle of gear (wheel) & not on the involute profile of tooth on wheel. Involute teeth of pinion will remove some material from not involute flank portion of the air, this Removal of material is a process called under cutting, is known as interference.

Similiarly if the Radius of the Addendum circle of the wheel increase beyond O_2M then the tip of tooth on wheel will cause interference with the tooth of pinion the point M & N called interference point.

from the even discussion be conclude the interference may only be avoid if the point of contact b/w the two teeth is always on involute profile of both the teeth.

⇒ Method to prevent interference:-

1. Under cutting:-

Under cut is done by the cutting tool at the time of mfg. result - strength of tooth is less at root. Limitation:- use of tooth low power transmission

⇒ Gear manufacturing Method :-

The various production method for producing the gear :-

1. Gear Milling :-

This is one of the initial & ~~waste~~ best, known as a ~~removal~~ material removal process for making gears. This method require the use of milling m/c. It is also to be noted direct this method & produce nearly all type of gears.

This method involve the use of a from cutter which is passed through the blank ~~the~~ create the tooth gap. this method is right known used only for the manufacturing of gear require very less dimensional accuracy.

2 Gear hubbing :-

Gear hubbing is a continuous generating process in which a tooth flank of the constantly move w/p are formed by equal space cutting edge of the hub.

The main advantage of this process it is used to producing a variety of gear including spur, helical, warm wheel etc. This main disadvantage of this method to the higher productivity rate of the gear.

3. **Gear shaping :-** Gear shaping is a generating process the cutter used is vertically a gear produced with the cutter is the tool is rotated at the require velocity ratio relatively to the gear to manufacture. This method can be used to produce cluster gear; Internal gear etc.

4. **Bevel gear :-** Bevel gear cutting is very specialised area in the field of gear cutting, this involving special type of m/c. for each variety of bevel gear to the manufacture. Some ~~part~~ of the bevel gear type along with the type of m/c require a hypoid zero etc.

=) **Design consideration of gear ~~type~~ drive :-** In the design of a gear drive the following data are :-

1) The power to be transmitted the speed of a driving gear, the speed of the driven gear, the centre distance.

following requirement must be meet in the design of gear drive:-

2) The gear teeth should be ~~streng~~ sufficient strength so that they be fill under static load or dynamic load during normal running condition

3) The used of space & material should be economical.

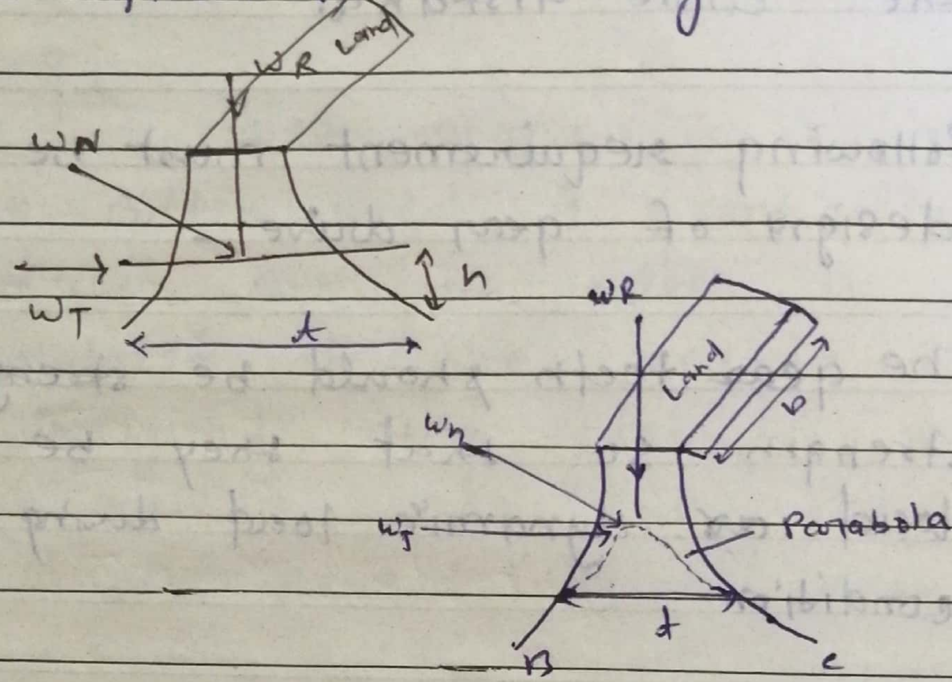
4) The alignment of the gear & deflection of the shaft must be consider bcoz they effect on the performance of the gear..

5) The lubrication of the gear must be satisfactory

=) **Beam strength of gear teeth :-**

It is defined as the maxi tensile load that the gear tooth can without failure, the beam strength of gear teeth is determined from an equation this eqn is known as Lewis eqn. & load carrying ability of tooth gear as determine by this eqn.

Lewis assume that as the load being transmitted from one gear to another. It is all given & taken by one tooth.



Consider each tooth as a cantilever beam loaded by a normal load (wn). It is resolve into component i.e tangential component ( ) & radial component ( ) acting $\perp^r$ to the centre line of the tooth respectively. The tangential component include of bending stress which tend to break the tooth. The Radial component WR induced as compresive stress of relatively small magnitude therefore its effect on the tooth neglected hence the bending stress is used as the basis for design calculation. The critical section on the section of max. bending stress may be obtained by drawing a parabola through A & tangential to the teeth curve B&C. If the teeth of shape like a parabola it will have same stress at all section. But the tooth is larger than parabola at every section BC, we therefore conclude at the section BC is the section of max. stress or the critical section. The max. value of bending stress BC is given by

$$\boxed{\frac{M}{I} = \frac{\sigma}{y}}$$

$$\boxed{M_{max} = (\sigma_{max}) - \frac{I_{NA}}{Y}}$$

where:-

$M \rightarrow$ max. bending moment at critical section BC.

$M_{max} \rightarrow W_T \times h$

W_T - tangential load acting on the tooth.

$h \rightarrow$ length of the tooth.

$Y \rightarrow$ half the thickness of teeth at the critical section BC.

$Y \rightarrow t/2$

$I \rightarrow$ MOI about the centre line of the tooth.

$I = \frac{Bd^3}{12}$

$b \rightarrow$ with of gear face.

$\sigma_w \rightarrow$ bending stress at section BC.

$$\boxed{Y = .124 - \frac{.684}{T}}$$

$$(\sigma_w)_{max} = \frac{M_{max} \cdot Y_{max}}{I} \Rightarrow \frac{W_T \times h \times t/2}{bt^3/12}$$

$$(\sigma_w)_{max} = W_T \times \frac{6h}{bt^2}$$

Safe Condition :- $\sigma_w \leq (\sigma_w)_{max}$

$$(\sigma_w)_{permissible} = W_T \frac{6h}{bt^2}$$

$$W_T = \frac{bt^2}{6h} (\sigma_w)_{per}$$

$$W_T = \frac{\pi m bt^2}{\pi m b n} (\sigma_w)_{per}$$

$$\boxed{W_T = \pi m b y (\sigma_w)_{per}} \quad \star \quad \boxed{Y = \frac{t^2}{6\pi m n}}$$

Beam strength

Y – Lewis form factor or tooth form factor.

$$\boxed{Y = 0.124 - \frac{0.684}{T}}$$

for $14\frac{1}{2}°$ composite & full depth involute system 20° full involute system.

$$\boxed{Y = 0.154 - \frac{0.912}{T}}$$

for 20° full depth involute system.

$$\boxed{Y = 0.175 - \frac{0.841}{T}}$$

for 20° stup system.

Hence lewis form factor depend on no. of teeth pressure angle, geometry of tooth profile (basically module).

=) Dynamic tooth load :→

The dynamic tooth are due to following reason :-

1. Deflection of tooth under load.
2. In accuracy of tooth profile.
3. Presence of stress concentration factor
4. Presence of jurk & impact.
5. Inertia of the rotating part.

A closer approximation to the actual condn may be made by the use of eqn on extansive series of test as follows :-

$$\boxed{W_D = W_T + W_I}$$

$$\boxed{W_I = \frac{W_T + 21V\,(bc + W_T)}{21V + \sqrt{b \cdot c + W_T}}}$$

where :-

$W_D \rightarrow$ Total Dynamic load (N)

W_T - Steady transmitted load

$V \rightarrow$ pitch line velocity (m/s)

$C \rightarrow$ Determination factor or dynamic factor.

$b \rightarrow$ face width.

e - tooth error action.

E_P - modulus of elasticity of material for pinion.

$$\boxed{C = \frac{e}{\left[\frac{1}{E_P} + \frac{1}{E_G}\right] K_1}}$$

E_G - modulus of elasticity of material for gear.

$K_1 \rightarrow$ A factor depending upon the form of teeth

⇒ Static tooth load :-

The static tooth load is obtained by lewi's formula by substituting floxure indurance limit or elastic limit stress (σ_e). In place of permissible bending.

$$\omega_T = \pi m b y (\sigma_\omega) per$$

$$\boxed{\omega_B = \pi m b y (\sigma_e)}$$

σ_e – 1·75 × B·H·N → for steel.
B·H·N – Brinell hardness no.
σ_e – flaxure indurance limit.

Wear tooth load :-

It is defined as the max. value of load, direct of gear tooth can be use or without any wear.
In gear wear comes out :-

1. Abrasive wear.
2) Corrosion wear.
3. metal pitting.

$$\boxed{\omega_\omega = Q D_P b K}$$

$$\boxed{Q = \frac{2G}{G \pm 1}}$$

$$\left\{ G = \frac{T}{t} = \frac{N_P}{N_g} \right\}$$

D_P – pitch diameter for pinion.