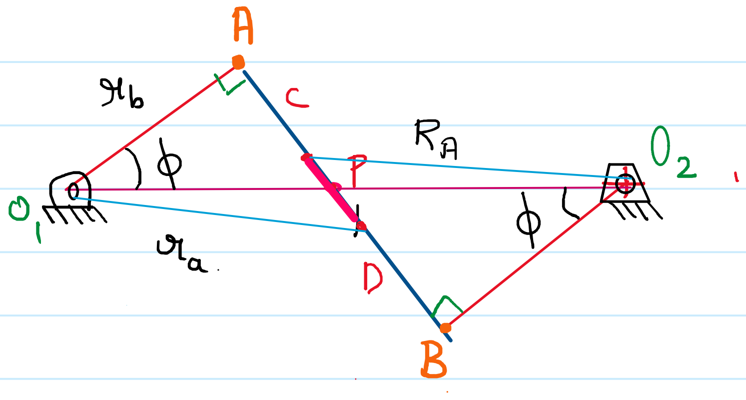
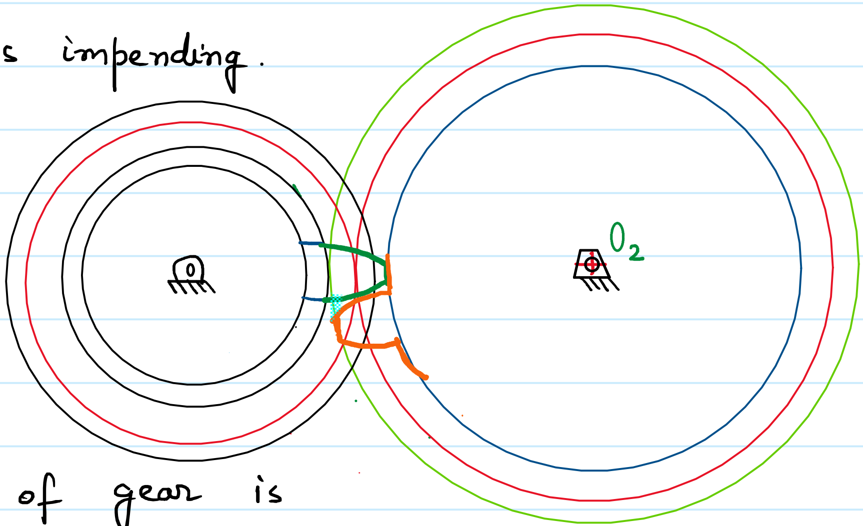


case (iii)

Interference is impending.

clearence $\neq 0$.

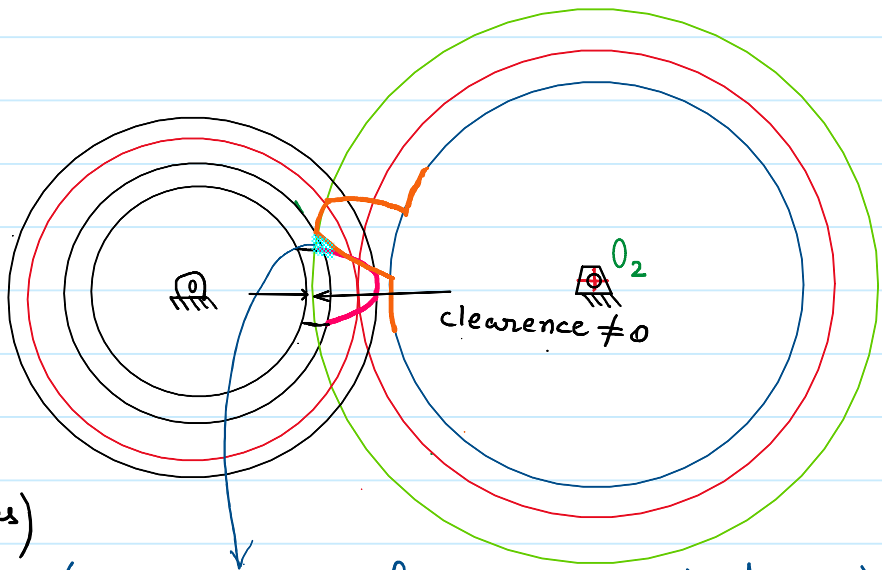


Involute face of gear is tending to get in action with non-involute flank of pinion.

Case (iv)

clearence $\neq 0$.

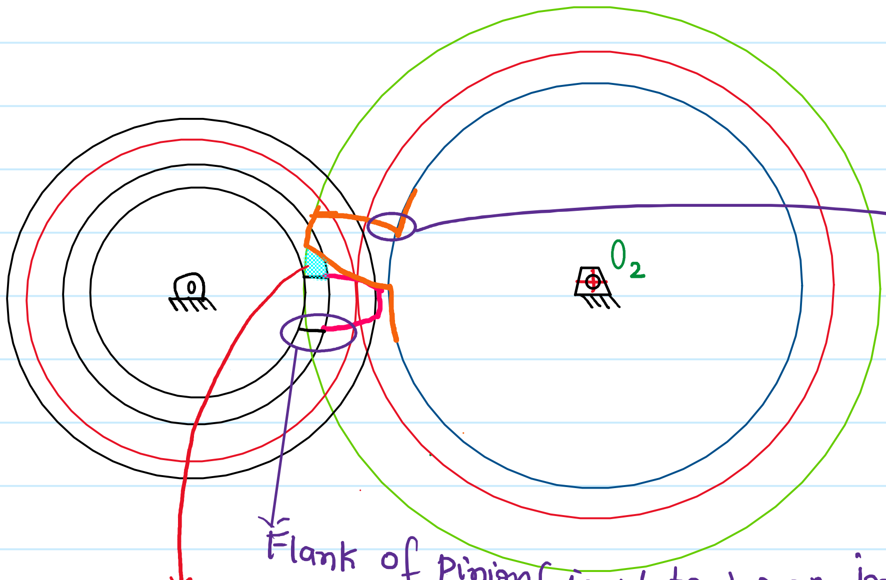
Total depth > Working depth.



(Interference occurs)

Non-conjugate action - (Involute face of gear is tending to get in action with non-involute flank of pinion) V.R = constant.

Case (V)



involute flank of gear.

Flank of Pinion (involute + non-involute)

Involute face of gear is about to get in action with non-involute flank of pinion. (non-conjugate action, V.R ≠ constant)

Clearence = 0

Total depth = Working depth.

Interference occurs.

INTERFERENCE:

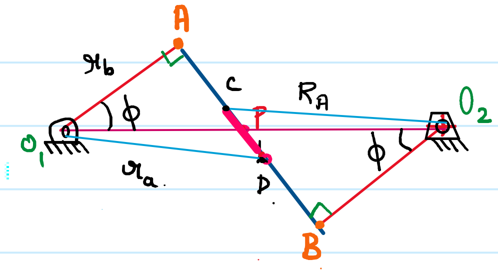
- Whenever non-conjugate meshing (meshing of involute profile with non-involute profile) will take place gears will join in each other and they will not be able to transfer the velocity ratio.
- Interference should be avoided.

Necessary condition for Interference:

- The presence of **non-involute profile is a necessary condition** for interference but, it is **not the sufficient one**.

Sufficient and necessary condition for Interference:

- If addendum of gear penetrate into the base circle of pinion or crosses the point of tangent to the base circle, it will result in interference.
- If the clearance is present interference may or may not occur, but if clearance is absent interference will take place certainly.
- If working depth is equal to full depth interference will take place and if working depth is less than full depth than interference may or may not occur.



- The **actual path of approach** will be decided by **gear dimensions** since gear will begin the engagement, whereas **pinion** will **end the engagement** so **actual path of recess** will be decided **by pinion.**

- The **maximum path of approach** will be decided by **pinion** and **maximum path of recess** will be decided by **gears**.

- The point of beginning of engagement 'C' has more possibility to cross point of tangency of the base circle of pinion (point A) therefore there will be more chances of interference at the **flank of pinion** than that of gear.

- Clearance is measured on the pinion side.

- If two wheels are in mesh having same size of addendum than the wheel which is driven will interfere first.

- In case of gear and pinion **gear will always interference first**.

- If two wheels are in mesh having different size of addendum than the **wheel having larger addendum will interfere** first.

- Interference occurs at the beginning of engagement.

Method to avoid interference

1. Increasing Centre distance
2. Undercutting
3. Stubbing
4. Using cycloidal teeth
5. By selecting the no. of teeth properly on pinion and Gear

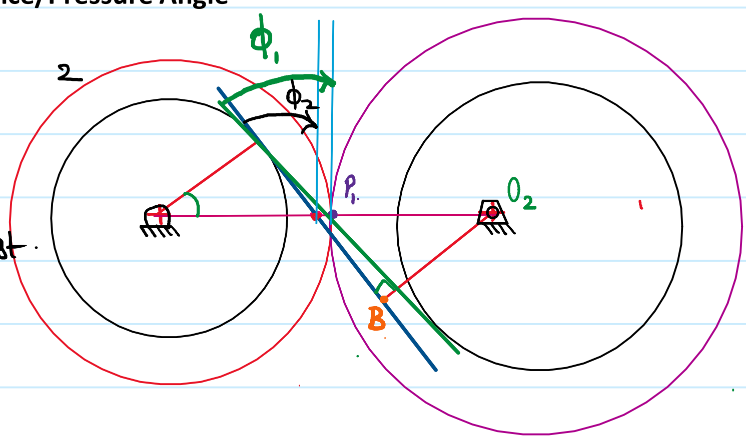
1. Increasing Centre distance/Pressure Angle

$r_b = \text{const.}$

$r_b = r\cos\phi$

$r\uparrow \quad \cos\phi\downarrow \quad r_b = \text{const.}$

$\cos\phi\downarrow \quad \phi\uparrow.$



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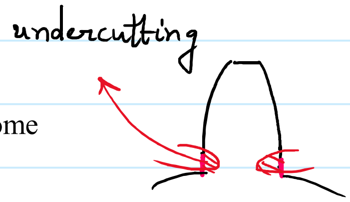
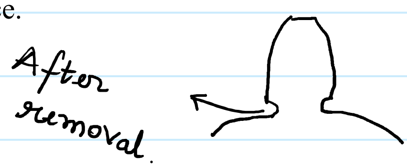
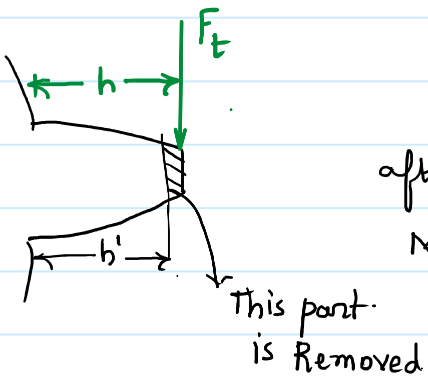
2. Under cutting:

- The process of removal of non-involute portion from the flank of pinion is known as under cutting.
- It may take place at the time of manufacturing.
- It may take place as a result of interference.
- It may done purposely to avoid the interference.
- Undercutting always result in stress concentration due to which the tooth become weaker and chances of failure will increase
- Therefore, undercutting is least preferred process to avoid interference.





3. Stubbing



$$M = F_t \times h.$$

after stubbing h decreases.

M decreases

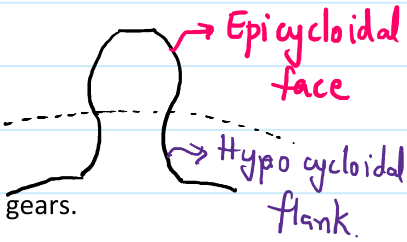
$\sigma \propto M$

$M \downarrow \ \sigma \downarrow$ strenght $\uparrow$.

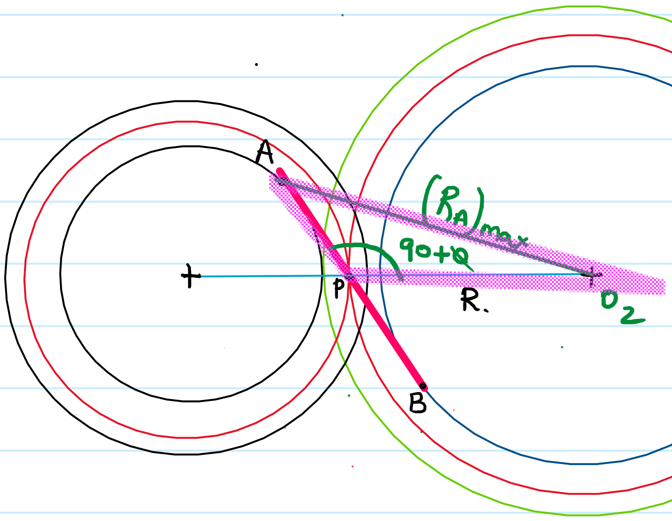
Modification of addendum.

4. Cycloidal teeth

1. Exact centre distance is required for cycloidal teeth
2. Tooth will be consisting of epi cycloidal face and Hypocycloidal flank.
3. Epi cycloidal face is in action with the hypo cycloidal flank of mating gears
4. This gives conjugate action.
5. Pressure angle is constant throughout the engagement in case of involute gears.
6. Pressure angle varies in case of cycloidal teeth
7. It is maximum at the beginning of engagement, zero at pitch point and maximum at the End of engagement.
8. Constant velocity ratio is not maintained during transmission of relative motion



5. **Selecting the number of teeth properly on Gear and Pinion**



In $\Delta^{le}\ O_2PA$.

$$O_2A^2 = O_2P^2 + AP^2 - 2.O_2P.\ AP.\ \cos\angle O_2PA.$$

$$(R_A)^2_{max} = R^2 + (r\sin\phi)^2 - 2\cdot R\cdot r\sin\phi.\ \cos(90+\phi)$$

$$(R_A)^2_{max} = R^2 + r^2\sin^2\phi - 2R.r\sin\phi.(-\sin\phi)$$

$$(R_A)^2_{max} = R^2 + r^2\sin^2\phi + 2R.r\sin^2\phi.$$

$$\text{Max. addendum.} \leq (R_A)_{Max} - R.$$

f_w - addendum coefficient of wheel.

$$(add)_{max.} \leq \sqrt{R^2 + r^2\sin^2\phi + 2Rr.\sin^2\phi} - R.$$

$$(add)_{max} \leq R\left[\sqrt{\left(1 + \left(\frac{r}{R}\right)^2 \cdot \sin^2\phi + 2\left(\frac{r}{R}\right)\sin^2\phi\right.} - 1\right]$$

$$\frac{r}{R} = d = 1/G = \frac{Z_p}{Z_G}$$

$$Z_p = d\cdot Z_G.$$

$$f_w \times m \leq \frac{m.Z_G}{2}\left[\sqrt{1 + d(d+2).\sin^2\phi} - 1\right]$$

$$Z_G \geq \frac{2.f_w.}{\sqrt{1 + d(d+2)\sin^2\phi} - 1.}$$

$$Z_p \geq. \frac{2.f_w.\times d.}{\sqrt{1 + d(d+2)\sin^2\phi} - 1.}$$

Minimum no. of teeth on Pinion.

For Rack and Pinion Mechanism.

$\lambda = \frac{r}{R}$ for Rack. $R \to \infty$ $\lambda \to 0$

$$z_p \geq \frac{2 f_w \times \lambda}{\sqrt{1+\lambda(\lambda+2)\sin^2\phi} - 1}$$

if $\lambda \to 0$ $z_p \to \frac{0}{0}$

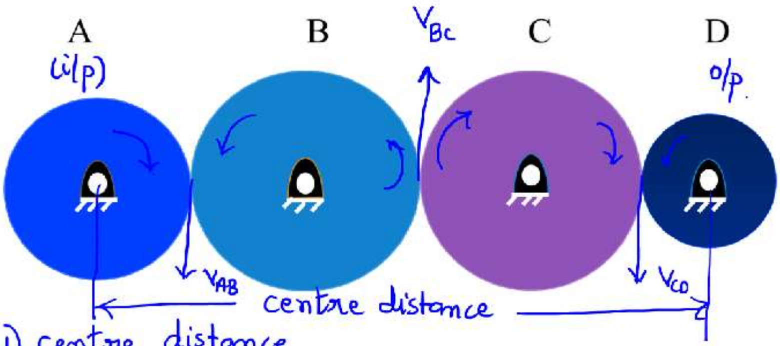
$$z_p = \frac{\frac{d}{d\lambda}(2 f_w \times \lambda)}{\frac{d}{d\lambda}\left(\sqrt{1+\lambda(\lambda+2)\sin^2\phi} - 1\right)}$$

$$z_p = \frac{2}{\sin^2\phi}$$

GEAR TRAIN:

Gear Train
- D.O.F = 1
  - When two gears are mounted on same shaft
    - Reverted gear train (R.G.T)
    - Compound gear train (C.G.T)
  - All gears are mounted on same plane and on different shaft. (S.G.T) → Simple Gear Train.
- D.O.F = 2 → Epicyclic Gear Train.

01. Simple Gear Train (S.G.T):



module

$m_A = m_B = m_C = m_D$

Pressure Angle

$\phi_A = \phi_B = \phi_C = \phi_D$

1) centre distance.

2) Train value.

for Pure Rolling

$V_{AB} = V_{BC} = V_{CD}$

Centre distance:

$$= r_A + 2r_B + 2r_C + r_D$$

$$= \frac{m_A \cdot Z_A}{2} + m_B \cdot Z_B + m_C \cdot Z_C + \frac{m_D \cdot Z_D}{2}.$$

$V = r\omega$

$V = \frac{m \cdot Z}{2} \cdot \omega$

$\boxed{Z \propto 1/\omega}$

Train value.

T.V.

$$= \frac{\omega_{o/p}}{\omega_{i/p}} = \frac{\omega_D}{\omega_A} = \left(-\frac{\omega_B}{\omega_A}\right)\left(-\frac{\omega_C}{\omega_B}\right)\cdot\left(-\frac{\omega_D}{\omega_C}\right)$$

$$= \left(-\frac{Z_A}{Z_B}\right)\left(-\frac{Z_B}{Z_C}\right)\cdot\left(-\frac{Z_C}{Z_D}\right)$$

$$\frac{\omega_D}{\omega_A} = \frac{-Z_A}{Z_D}$$

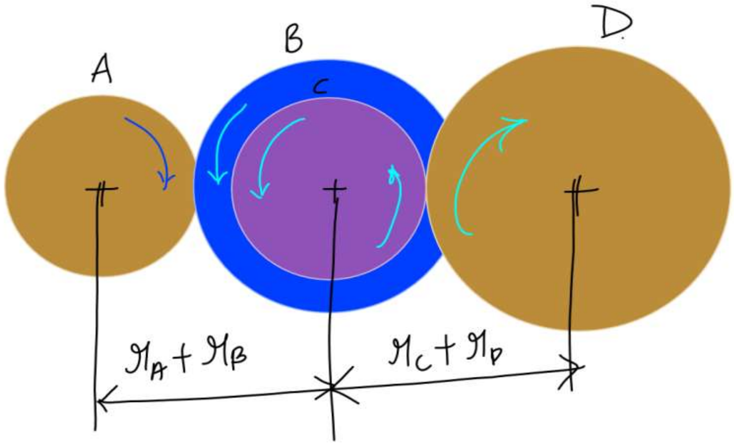
- Idler gears: The gear whose teeth does not appears in the final expression of train value are known as idler gear.
- They are use to fill the space between driven and driven element.
- Idler cannot effect the magnitude of train value but they certainly effect the direction of output element with the respect to input.

Gear B and C are called idler gears.

1. If odd no. of idlers are used direction of i/p and o/p gear is same.
2. In case of even no. of idler gear the direction of i/p and o/p is different.

02. Compound Gear Train (C.G.T):

- In this alternate gears are called driver – driven – driver – driven.



$\phi_A = \phi_B$

$m_A = m_B$

$\phi_C = \phi_D$

$m_C = m_D$

$m_B = m_C$ ✓

$m_B \neq m_C$ ✓

Centre distance:

$$= r_A + r_B + r_C + r_D$$

$$= \frac{m_A \cdot Z_A}{2} + \frac{m_B \cdot Z_B}{2} + \frac{m_C \cdot Z_C}{2} + \frac{m_D \cdot Z_D}{2}$$

$m_A = m_B$, $m_B = m_C$ / $m_B \neq m_C$

$m_C = m_D$,

Train value:

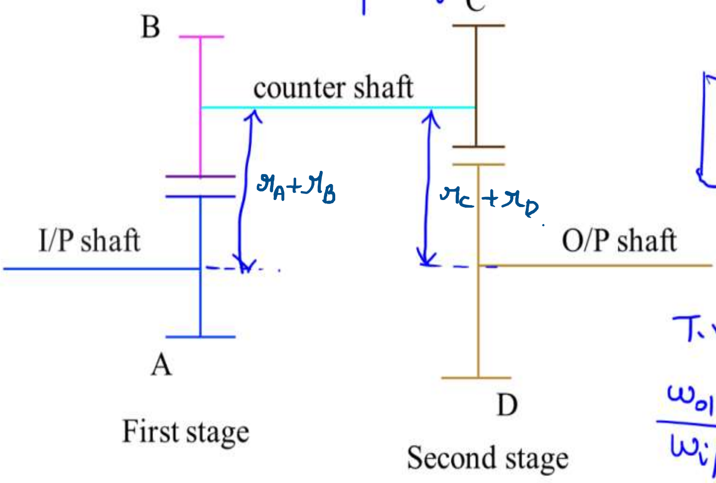
$$T.V. = \frac{\omega_{o/p}}{\omega_{i/p}} = \frac{\omega_D}{\omega_A} = \left(\frac{-\omega_B}{\omega_A}\right)\underbrace{\left(\frac{\omega_C}{\omega_B}\right)}_{1}\left(\frac{-\omega_D}{\omega_C}\right)$$

$$\boxed{\frac{\omega_D}{\omega_A} = \frac{Z_A \cdot Z_C}{Z_B \cdot Z_D}}$$

03. Reverted Gear Train (R.G.T): If the input and output shafts are co-axial then we use reverted gear train.

- EX. Gear box and Watches etc.

B, C are compound gear.



Centre distance.

$$r_A + r_B = r_C + r_D$$

$$\boxed{\frac{m_A \cdot Z_A}{2} + \frac{m_B \cdot Z_B}{2} = \frac{m_C Z_C}{2} + \frac{m_D \cdot Z_D}{2}}$$

$m_A = m_B$

$\phi_A = \phi_B$

$m_C = m_D$

$\phi_C = \phi_D$ ✓

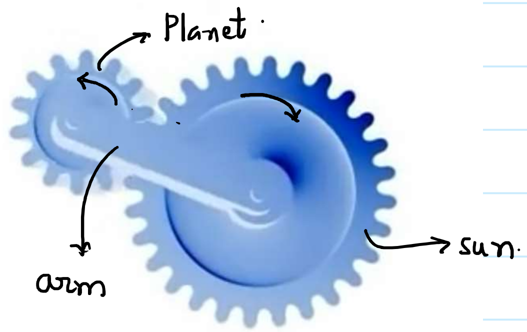
$m_B = m_C$ ✓

$m_B \neq m_C$

T.V.

$$\frac{\omega_{o/p}}{\omega_{i/p}} = \frac{\omega_D}{\omega_A} = \left(\frac{-\omega_B}{\omega_A}\right)\underbrace{\left(\frac{\omega_C}{\omega_B}\right)}_{1}\left(\frac{\omega_D}{\omega_C}\right)$$

$$\boxed{\frac{\omega_D}{\omega_A} = \frac{-Z_A}{Z_B} \cdot \frac{-Z_C}{Z_D}}$$



$$\omega_s \cdot Z_s = \omega_p \cdot Z_p$$

$$\omega_p = -\frac{Z_s}{Z_p} \cdot (+1)$$

Advantages:

- They are compact in size.
- Both static and Dynamic forces are balanced if multiple planets are used.
- High torque ratio or velocity ratio can be achieved.
- Bi-directional output can be obtained from a single unidirectional input. DOF = 2
- It is used in automobile transmission system

SNo.	Condition of motion	Arm.	Sun Gear	Planet Gear.
1.	Arm is fixed and sun rotated with (+1rev)	0	+1.	$-\frac{Z_s}{Z_p}$.
2.	Arm. is fixed and sun. rotated $+x$·rev	0	$+x$.	$-x \cdot \frac{Z_s}{Z_p}$
3.	Arm rotated by $(+y)$	$+y$	$+y$	$+y$
4.	Total	y	$x+y$	$y - x \cdot \frac{Z_s}{Z_p}$

Speed of Sun $N_s = x + y$. $\Rightarrow N_s = x + N_{arm} \longrightarrow$ (A)

Speed of Planet $N_p = y - x \cdot \frac{Z_s}{Z_p}$ $\quad N_p = N_{arm} - x \cdot \frac{Z_s}{Z_p} \longrightarrow$ (B)

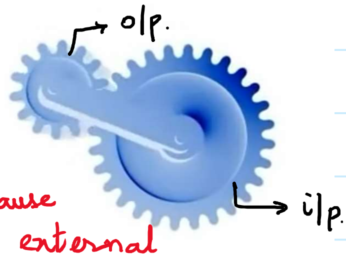
$\left(\frac{A}{B}\right)$

$$\boxed{\frac{N_s - N_{arm}}{N_p - N_{arm}} = -\frac{Z_p}{Z_s}}$$

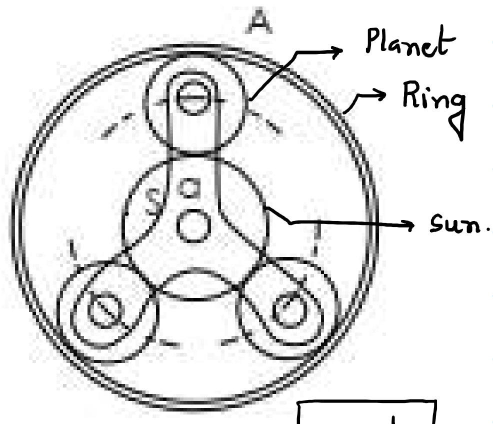
T.V.

$$\frac{\omega_{o/p}}{\omega_{i/p}} = \frac{N_P - N_{arm}}{N_S - N_{arm}} = -\frac{Z_S}{Z_P}$$

because of external gearing.



→ sun – i/p. Ring – o/p.
Arm – i/p Planet – (coupler link)



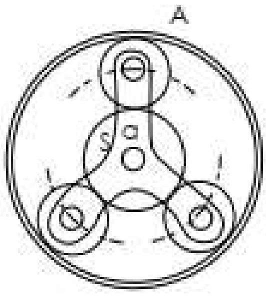
$$\boxed{\omega \propto \frac{1}{Z}}$$

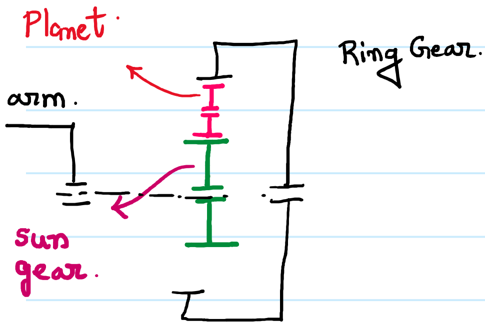
T.V.

$$\frac{\omega_{o/p}}{\omega_{i/p}} = \frac{\omega_R - \omega_{arm}}{\omega_S - \omega_{arm}} = \frac{\omega_P - \omega_{arm}}{\omega_S - \omega_{arm}} \cdot \frac{\omega_R - \omega_{arm}}{\omega_P - \omega_{arm}}$$

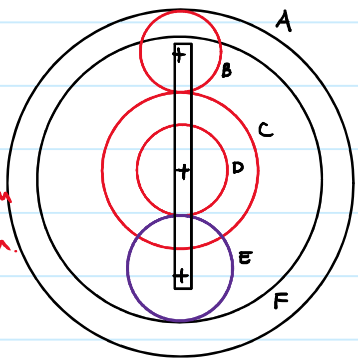
$$\frac{\omega_R - \omega_{arm}}{\omega_S - \omega_{arm}} = \left(-\frac{Z_S}{Z_P}\right)\left(+\frac{Z_P}{Z_R}\right)$$

$$\frac{\omega_R - \omega_{arm}}{\omega_S - \omega_{arm}} = -\frac{Z_S}{Z_R}$$





→ A – i/p, F – o/p. C–D – compound gears.
Arm – i/p. $\omega_C = \omega_D$.



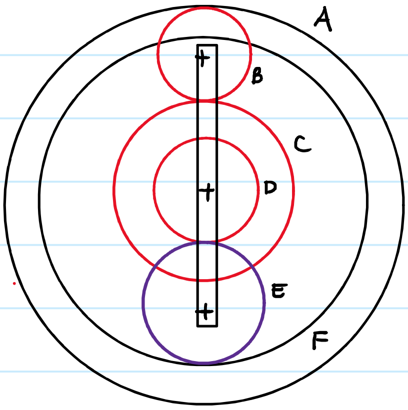
T.V.

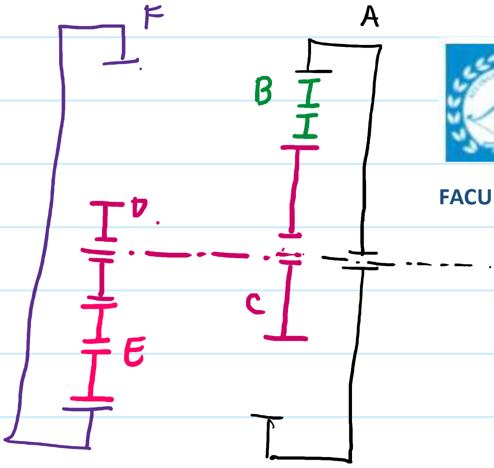
$$\frac{\omega_{o/p}}{\omega_{i/p}} = \frac{\omega_F - \omega_{arm}}{\omega_A - \omega_{arm}} = \frac{\omega_B - \omega_{arm}}{\omega_A - \omega_{arm}} \cdot \frac{\omega_C - \omega_{arm}}{\omega_B - \omega_{arm}} \cdot \frac{\omega_D - \omega_{arm}}{\omega_C - \omega_{arm}} \cdot \frac{\omega_E - \omega_{arm}}{\omega_D - \omega_{arm}} \cdot \frac{\omega_F - \omega_{arm}}{\omega_E - \omega_{arm}}$$

1.

$$\frac{\omega_F - \omega_{arm}}{\omega_A - \omega_{arm}} = \left(+\frac{Z_A}{Z_B}\right)\left(-\frac{Z_B}{Z_C}\right)\left(-\frac{Z_D}{Z_E}\right)\left(+\frac{Z_E}{Z_F}\right)$$

Gear B and E are idler gears.





Assumption.

1. Inertia of wheels and arms is ignored.

Torque Analysis.

$$\Sigma \text{Torque} = 0$$

$$T_{i/p} + T_{arm} + T_{o/p} = 0$$

$$T_s + T_{arm} + T_p = 0.$$

$$\Sigma \text{Power} = 0$$

$$P_{i/p} + P_{arm} + P_{o/p} = 0$$

$$T_s.\omega_s + T_{arm}\cdot\omega_{arm} + T_p.\omega_p = 0.$$

01. In gears, interference takes place when

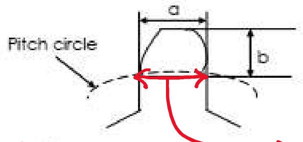
(a) $\boxed{\text{the tip of a tooth of a mating gear digs into the portion between base and root circles}}$ → Flank portion of pinion.

(b) gears do not move smoothly in the absence of lubrication → Result of interference

(c) pitch of the gear is not same

(d) gear teeth are undercut → Result of interference.

02. One tooth of a gear having 4 module and 32 teeth is shown in the figure.
Assume that the gear tooth and the corresponding tooth space make equal intercepts on the pitch circumference. The dimensions 'a' and 'b', respectively, are closest to **(GATE– 08)**



(a) ~~6.08 mm, 4 mm~~
(b) 6.48 mm, 4.2 mm
(c) 6.28 mm, 4.3 mm
(d) 6.28 mm, 4.1 mm ✓

$m = 4\,mm \qquad z = 32$

circular pitch = Tooth space + Tooth Thickness

tooth space = tooth thickness.

$\text{Tooth thickness} = \frac{\pi m}{2}$

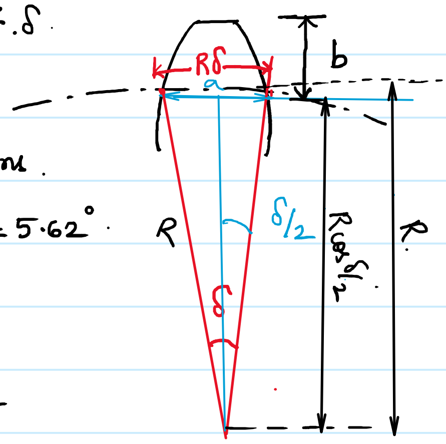
$\text{tooth thickness} = R\delta.$

$\frac{\pi m}{2} = \frac{m \cdot Z}{2} \cdot \delta.$

$\delta = \frac{\pi}{z} = \frac{3.1414}{32}$

$\delta = 0.0981$ radians.

$= 0.0981 \times \frac{180}{\pi} = 5.62^\circ.$



$a = \left(R \cdot \sin \delta/2\right) \times 2$

$= \frac{m \cdot Z}{2} \sin \delta/2 \cdot \times 2$

$= \frac{32 \times 4}{2} \times \sin\left(\frac{5.62}{2}\right) \times 2$

$= 6.28$

$b = \text{add} + \left(R - R \cdot \cos \delta/2\right)$

$= 4 + 64\left(1 - \cos\left(\frac{5.62}{2}\right)\right)$

$= 4.077 \cong 4.1\,mm$

03. An involute pinion and gear are in mesh. If both have the same size of addendum, then there will be an interference between the

(a) $\boxed{\text{tip of the gear tooth and flank of pinion}}$
(b) tip of the pinion and flank of gear
(c) flanks of both gear and pinion
(d) tips of both gear and pinion

04. For a pinion of 15 teeth, undercutting decreases (increases/decreases) with increase (increases/decreases) of pressure angle. / centre distance.

05. For spur with gear ratio greater than one, the interference is most likely to occur near the
(a) pitch point
(b) point of beginning of contact
(c) point of end of contact
(d) root of the tooth

$G > 1.$

06. Consider the following specifications of gears A, B, C and D:

Gears	A	B	C	D
Number Of teeth	20	60	20	60
Pressure Angle	14.5°	14.5°	20°	14.5°
Module	1	3	3	1
Material	Steel	Brass	Brass	Steel

Which of these gears form the pair of spur gears to achieve a gear ratio of 3?
(a) A and B
(b) A and D
(c) B and C
(d) C and D

G.R. = 3.

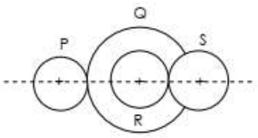
A, B
↓ ↓
20 60
$\phi_A = \phi_B$
$m_A \neq m_B$

cannot mesh.

A & D
↓ ↓
20 60
$\phi_A = \phi_D$
$m_A = m_D$

Compound Gear Trains

07. A compound gear train with gears P,Q,R and S has number of teeth 20, 40,15 and 20, respectively. Gears Q and R are mounted on the same shaft as shown in the figure below. The diameter of the gear Q is twice that of the gear R. If the module of the gear R is 2 mm, the center distance in mm between gears P and S is **(GATE–13)**



(a) 40
(b) 80
(c) 120
(d) 160

$Z_P = 20$
$Z_Q = 40$
$Z_R = 15$
$Z_S = 20.$

Q − R → compound gear

$d_Q = 2d_R \Rightarrow m_Q \cdot Z_Q = 2 \cdot m_R \cdot Z_R.$

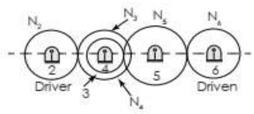
$m_R = 2\,mm.$
$m_S = 2\,mm.$
$m_Q = 1.5$
$m_P = 1.5\,mm.$

$m_Q = \frac{2\times2\times15}{40} = 1.5\,mm.$

centre distance $= r_P + r_Q + r_R + r_S.$

$= \frac{m_P \cdot Z_P}{2} + \frac{m_Q \cdot Z_Q}{2} + \frac{m_R \cdot Z_R}{2} + \frac{m_S \cdot Z_S}{2}$

08. A gear train is made up of five spur gears as shown in the figure. Gear 2 is driver and gear 6 is driven member. N_2, N_3, N_4, N_5 and N_6 represent number of teeth on gears 2,3,4,5, and 6 respectively, The gear(s) which act(s) as idler(s) is/ are



(GATE –15)

(a) Only 3
(b) Only 4
(c) Only 5
(d) both 3 and 5

T.V.

$$\frac{\omega_6}{\omega_2} = \left(-\frac{\omega_3}{\omega_2}\right)\left(+\frac{\omega_4}{\omega_3}\right)\left(\frac{\omega_5}{\omega_4}\right)\left(-\frac{\omega_6}{\omega_5}\right)$$

$$\frac{\omega_6}{\omega_2} = \left(-\frac{Z_2}{Z_3}\right)\cdot\left(-\frac{Z_4}{\cancel{Z_5}}\right)\cdot\left(-\frac{\cancel{Z_5}}{Z_6}\right)$$

Gear 5 is idler wheel.

Reverted Gear Trains

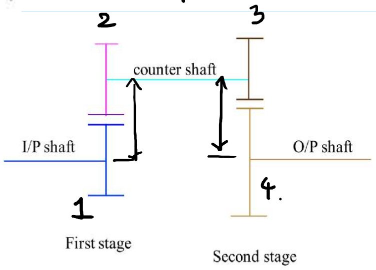
Data for Q. 09 & 10 are given below.
The overall gear ratio in a 2 stage speed reduction gear box (with all spur gears) is 12. The input and output shafts of the gear box are collinear. The countershafts which is parallel to the input and output shafts has a gear (Z_2 teeth) and pinion (Z_3 = 15 teeth) to mesh with pinion (Z_1 = 16 teeth) on the input shaft and gear (Z_4 teeth) on the output shaft respectively. It was decided to use a gear ratio of 4 with 3 module in the first stage and 4 module in the second stage. **(GATE-03)**

Reverted.

09. Z_2 and Z_4 are
(a) 64 and 45
(b) 45 and 64
(c) 48 and 60
(d) 60 and 48

10. The centre distance in the second stage is
(a) 90 mm
(b) 120 mm
(c) 160 mm
(d) 240 mm

$G = 12 \Rightarrow G = G_1 G_2$
$G_1 = 4.$
$G_2 = G/G_1 = 12/4 = 3.$



Gear 1 – pinion. $z_1 = 16$
Gear 2 – wheel. $z_2 = ?$
Gear 3 – Pinion. $z_3 = 15$
Gear 4 – wheel. $z_4 = ?$

$m_1 = m_2 = 3\,mm.$
$m_3 = m_4 = 4\,mm.$

$G_1 = \frac{\omega_1}{\omega_2} = \frac{z_2}{z_1} = 4.$

$z_2 = 4 \cdot z_1 = 4\times16 = 64.$

$G_2 = \frac{\omega_3}{\omega_4} = \frac{z_4}{z_3} \Rightarrow z_4 = 3z_3 = 3\times15 = 45$

centre distance $= r_1 + r_2 = \frac{m_1 z_1}{2} + \frac{m_2 z_2}{2} =$

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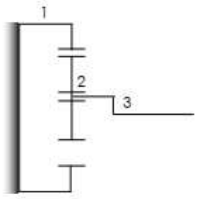
Cycloidal Gear Trains

11. A planetary gear train is shown in Fig. Internal gear (1) has 104 teeth and is held fixed and planet gear (2) has 96 teeth. How much does the planet gear rotate for sixty revolutions of the planet carrier (3) in clockwise direction ? (GATE-99)



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Gear1- internal. $z_1 = 104$, $\omega_1 = 0$

Gear-2 $z_2 = 96$

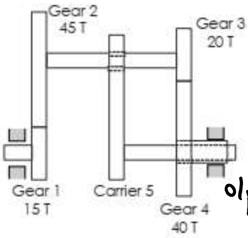
$$\frac{\omega_1 - \omega_{arm}}{\omega_2 - \omega_{arm}} = +\frac{z_2}{z_1} \Rightarrow \frac{0-60}{\omega_2 - 60} = +\frac{96}{104}$$

$\omega_2 = ?$ $\omega_{arm} = 60$ rev. C.W.

$\omega_2 =$ ______ rev.

Data for Q. 12 & 13 are given below.

A planetary gear train has four gears and one carrier, angular velocities of the gears are ω_1, ω_2, ω_3, and ω_4, respectively. The carrier (arm) rotates with angular velocity ω_5. (GATE – 06)



2-3 compound-gear.

i/p o/p.

T.V.

$$\frac{\omega_{o/p}}{\omega_{i/p}} ; \frac{\omega_4 - \omega_{arm}}{\omega_1 - \omega_{arm}} = \frac{\omega_2 - \omega_{arm}}{\omega_1 - \omega_{arm}} \cdot \frac{\omega_3 - \omega_{arm}}{\omega_2 - \omega_{arm}} \cdot \frac{\omega_4 - \omega_{arm}}{\omega_3 - \omega_{arm}}$$

$$\frac{\omega_4 - \omega_{arm}}{\omega_1 - \omega_{arm}} = \left(-\frac{z_1}{z_2}\right)\left(-\frac{z_3}{z_4}\right)$$

$$\frac{\omega_4 - \omega_{arm}}{\omega_1 - \omega_{arm}} = \left(-\frac{15}{45}\right)\cdot\left(\frac{-20}{40}\right) = 1/6.$$

12. What is the relation between the angular velocities of Gear 1 and Gear 4 ?

(a) $\boxed{\frac{\omega_1 - \omega_5}{\omega_4 - \omega_5} = 6}$ ✓ (b) $\frac{\omega_4 - \omega_5}{\omega_1 - \omega_5} = 6$

(c) $\frac{\omega_1 - \omega_2}{\omega_4 - \omega_3} = \left[\frac{2}{3}\right]$ (d) $\frac{\omega_2 - \omega_5}{\omega_4 - \omega_5} = -\frac{8}{9}$

13. For ω_1 = 60 rpm clockwise (cw) when looked from the left, what is the angular velocity of the carrier and its direction so that Gear 4 rotates in counter clockwise (ccw) direction at twice the angular velocity of Gear 1 when looked from the left?
(a) 130 rpm, cw
(b) 223 rpm, ccw
(c) 256 rpm, cw
(d) 156 rpm, ccw ✓

$\omega_1 = 60$ rpm.

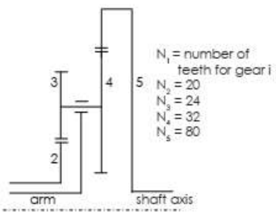
$\omega_{arm} = ?$

$\omega_4 = -120$ rpm.

$$\frac{-120 - \omega_{arm}}{60 - \omega_{arm}} = 1/6.$$

$\omega_{arm} =$ ___.

14. For the epicylic gear arrangement shown in the figure, ω_2 = 100 rad/s clockwise (CW) and ω_{arm} = 80 rad/s counter clockwise (CCW). The angular velocity ω_5 (in rad / s) is (GATE–10)

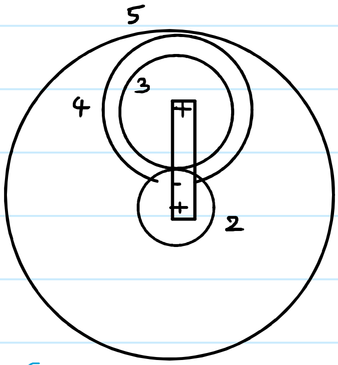


N_i = number of teeth for gear i
$N_2 = 20$
$N_3 = 24$
$N_4 = 32$
$N_5 = 80$

(a) 0 (b) 70 CW
(c) 140 CCW (d) 140 CW

$\omega_2 = 100$ rad/s, $\omega_{arm} = -80$ rad/s

$\omega_5 = ?$



$z_2 = 20$
$z_3 = 24$
$z_4 = 32$
$z_5 = 80$

$$\frac{\omega_5 - \omega_{arm}}{\omega_2 - \omega_{arm}} = \frac{\omega_3 - \omega_{arm}}{\omega_2 - \omega_{arm}} \cdot \frac{\omega_4 - \omega_{arm}}{\omega_3 - \omega_{arm}} \cdot \frac{\omega_5 - \omega_{arm}}{\omega_4 - \omega_{arm}}$$

1

$$= \left(\frac{-z_2}{z_3}\right)\cdot\left(\frac{+z_4}{z_5}\right)$$

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15. In an automobile service station, an automobile is in a lifted up position by means of a hydraulic jack. A person working in the service station gave a tap to one rear wheel and made it rotate by one revolution. The rotation of another rear wheel is

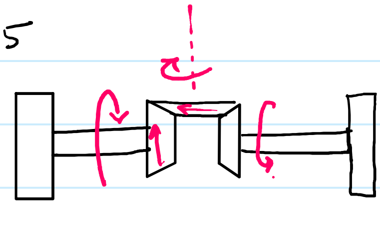
(a) zero

(b) also one revolution in the same direction

(c) also one revolution but in the opposite direction

(d) unpredictable

15



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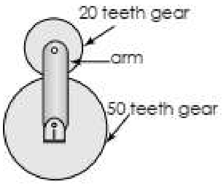
16. The number of degrees of freedom of the planetary gear train shown in the figure is **(GATE –15)**



(a) 0 (b) 1

(c) 2 (d) 3

16)

$\boxed{DoF = 2.}$

17. An Involute pinion has16 teeth and a module of 5 mm. What is the minimum value of the pressure angle so that the dedendum (with a standard value of one module) consists of purely an involute profile to eliminate the interference completely.

(a) 29° (b) 25°

(c) 20° (d) 32.5°

A) Possible only wheels not pinion complete (for involute profile)

$R_{Dedendum} \geq R_{Base}$.

$R - \text{dedendum} \geq R.\cos\phi$.

$40 - 5 \geq 40\cos\phi$.

$\phi \leq \cos^{-1}\left(\frac{35}{40}\right)$ $\phi \leq 28.95°$.

ϕ increases $\Rightarrow$ $\cos\phi \downarrow$ $R\uparrow$;

so $R_b = \text{constant}$.

$m = 5$. $z = 16$

$R = \frac{5\times 16}{2} = 40$.

dedendum $= 1m$.

$\phi = 14.5°$

$\phi = 20°$.

18. In a sun and planet gear train, the fixed sun gear has 40 teeth and planet gear has 20 teeth. The carrier rotates at 10 rad/s and transmits a torque of 5 N-m. The holding torque on sun gear is _____ N-m.

19. Train value of a gear train is

(a) equal to speed ratio

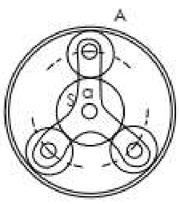
(b) equal to reciprocal of speed ratio

(c) always greater than one

(d) always less than one

$T.V. = \frac{1}{\text{Speed Ratio}}$

20. The three armed spider of given figure is driven at 180 rpm. The annulus A in the gear rotates at 300 rpm about the axis of the fixed wheel S which has 80 teeth. What will be the number of teeth required on wheel P?



(a) 15 (b) 25

(c) 30 (d) 20

$\omega_{arm} = 180$ rpm.

$\omega_{Ring} = 300$ rpm.

$\omega_s = 0$.

$Z_s = 80$.

$Z_p = 20$.

$R_R = R_s + 2.R_p$.

$\frac{m_R Z_R}{2} = \frac{m_s.Z_s}{2} + m_p.Z_p$

$Z_R = Z_s + 2Z_p$.

$80 + 2Z_p$.

T.V.

$$\frac{\omega_R - \omega_{arm}}{\omega_s - \omega_{arm}} = \frac{\omega_p - \omega_{arm}}{\omega_s - \omega_{arm}} \cdot \frac{\omega_R - \omega_{arm}}{\omega_p - \omega_{arm}}.$$

$$\frac{\omega_R - \omega_{arm}}{\omega_s - \omega_{arm}} = \left(\frac{-Z_s}{\cancel{Z_p}}\right)\left(\frac{+\cancel{Z_p}}{Z_R}\right)$$

$$\frac{300-180}{0-180} = \frac{-80}{80+2Z_p} \Rightarrow Z_p = \_\_\_.$$

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21. Which of the following statement(s) is/are correct ?
   (a) Bevel gear is used for connecting two non-parallel or, intersecting but coplanar shafts. ✓
   (b) Spur gear is used for connecting two parallel and coplanar shafts with teeth parallel to the axis of the gear wheel. ✓
   (c) Mitre gear is used for connecting two shafts whose axes are mutually perpendicular to each other. ✓
   (d) Helical gear is used for connecting two parallel and coplanar shafts with teeth inclined to the axis of the gear wheel. ✓

} Mitre gear - identical gears are used.

$VR = 1$

18. In a sun and planet gear train, the fixed sun gear has 40 teeth and planet gear has 20 teeth. The carrier rotates at 10 rad/s and transmits a torque of 5 N-m. The holding torque on sun gear is _____ N-m.

$\omega_s = 0$

$Z_s = 40$

$Z_p = 20.$

$\omega_{arm} = 10 \text{ rad/s}.$

$T_{arm} = 5 \text{ N-m}.$

$T_{sun/fixed} = ?$

T.V. $\dfrac{\omega_p - \omega_{arm}}{\omega_s - \omega_{arm}} = -\dfrac{Z_s}{Z_p}$

$\dfrac{\omega_p - 10}{0 - 10} = -\dfrac{40}{20} = -2$

$\omega_p = 30 \text{ rad/s}.$

$\Sigma Power = 0$

$P_s + P_p + P_{arm} = 0$

$T_s \cdot \omega_s + T_p \cdot \omega_p + T_{arm} \cdot \omega_{arm} = 0$

$T_p \cdot (30) + 5(10) = 0 \Rightarrow T_p = -50/30 = -1.67 \text{ N-m}.$

$\Sigma Torque.$

$T_s + T_p + T_{arm} = 0$

$T_s + (-1.67) + 5 = 0 \Rightarrow T_s = \underline{-3.33 \text{ N-m}}$

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What is BALANCING ?

- The process of either removing or reducing the unbalanced force or couple from a system is known as balancing.
- Balancing can be done either by adding the counter masses or by removing the extra masses present in the system.

Why Balancing is required ?

- Inertia force and inertia moment of a body are equal and opposite to the resultants of all external forces and moments.
- Accelerating parts of a machine generate inertia forces and moments, which are transmitted to the machine's frame or foundation.
- Variations in acceleration over time result in dynamic forces and moments on the foundation, leading to harmful vibrations and noise during operation.
- Balancing these inertia forces and moments is essential for increasing the life expectancy and ensuring smoother operation of the machine.

1. Types of balancing
   1. Static balancing.
   2. Dynamic balancing.

Aspects of Balancing problem

- The evaluation and analysis of inertia forces and moments
- The determination of convenient methods of balancing these quantities.

BALANCING OF ROTATING MASSES:
(A) STATIC BALANCING:

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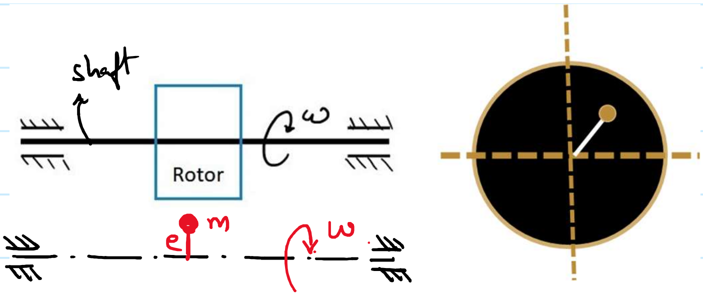
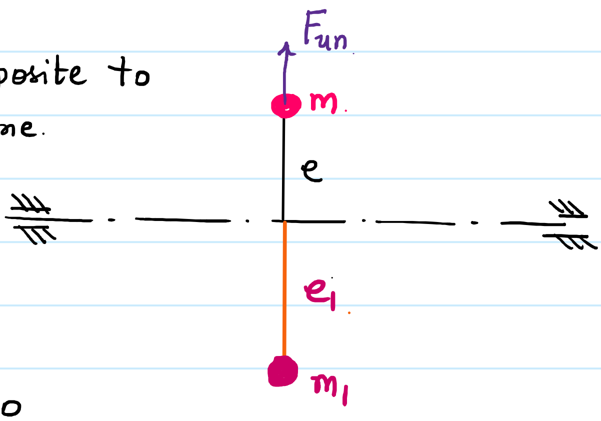
- A system is said to be statically balanced if there is no unbalanced force in the system.
- The centre of mass will lie on the axis of rotation.
- The force polygon will be completely closed.



METHOD(1): Adding the balancing masses in the same plane:

a) BALANCING BY SINGLE COUNTER MASS:

Mass is added diametrically opposite to unbalanced mass in same plane.
m = mass of rotor



- Force polygon will be closed
- C.O.M will lie on the axis of rotation

$\Sigma F_X = 0$
$\Sigma F_Y = 0$

$$\Sigma F_y = 0$$

$$m e \omega^2 - m_1 e_1 \omega^2 = 0$$

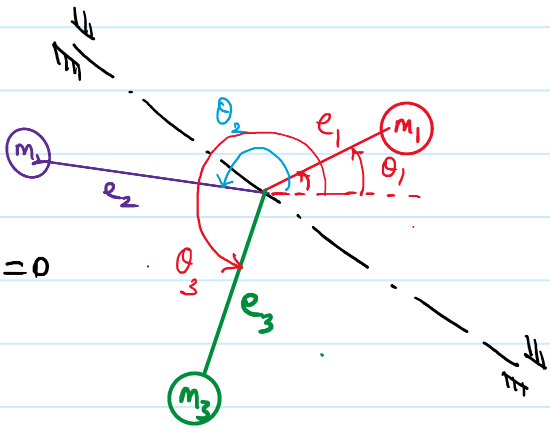
$$m e = m_1 e_1$$

METHOD(2): Adding the balancing masses in the same plane:

b) BALANCING BY MORE THAN ONE COUNTER MASS:

- If the system is statically balanced then:

$\Sigma F_X = 0$
$\Sigma F_Y = 0$



$$\Sigma F_x = 0$$

$$m_1 e_1 \omega^2 \cos\theta_1 + m_2 e_2 \omega^2 \cos\theta_2 + m_3 e_3 \omega^2 \cos\theta_3 = 0$$

$$\sum_{i=1}^{n} m_i e_i \cos\theta_i = 0$$

$$\Sigma F_y = 0$$

$$m_1 e_1 \omega^2 \sin\theta_1 + m_2 e_2 \omega^2 \sin\theta_2 + m_3 e_3 \omega^2 \sin\theta_3 = 0$$

$$\sum_{i=1}^{n} m_i e_i \sin\theta_i = 0$$

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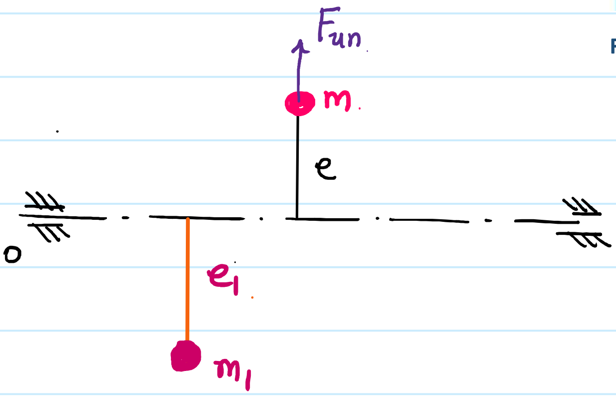
METHOD(2): Adding the balancing masses in the parallel plane:

a) BY ADDING A SINGLE BALANCING MASS:

$$\Sigma F_y = 0$$

$$m e \omega^2 - m_1 e_1 \omega^2 = 0$$

$$m e = m_1 e_1$$



b) STATIC BALANCING OF SYSTEM BY ADDING MORE THAN ONE MASS IN PARALLEL PLANE:

$$\Sigma F_x = 0$$

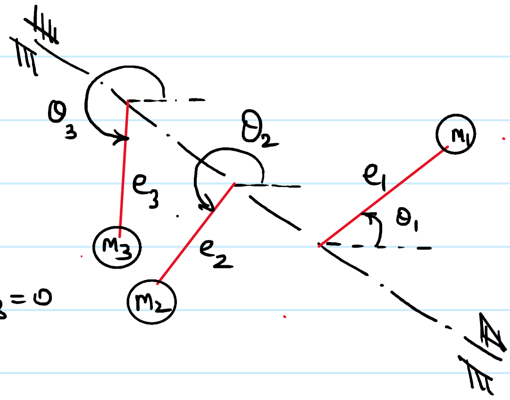
$$m_1 e_1 \omega^2 \cdot \cos\theta_1 + m_2 e_2 \cdot \omega^2 \cdot \cos\theta_2 + m_3 e_3 \cdot \omega^2 \cdot \cos\theta_3 = 0$$

$$\Sigma m_i e_i \cos\theta_i = 0$$

$$\Sigma F_y = 0$$

$$m_1 e_1 \omega^2 \cdot \sin\theta_1 + m_2 e_2 \cdot \omega^2 \cdot \sin\theta_2 + m_3 e_3 \cdot \omega^2 \cdot \sin\theta_3 = 0$$

$$\Sigma m_i e_i \sin\theta_i = 0$$



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