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PART - A

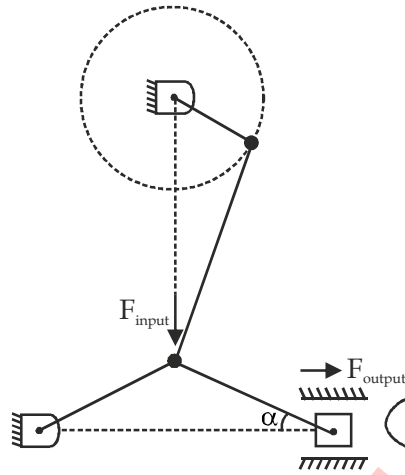
2 Marks Questions

1.
Sol.
 - A redundant degree of freedom (DOF) in a mechanism refers to having more independent movements (DOF) than are necessary to achieve a desired task.
 - In simpler terms, it's like having extra "degrees of freedom" that aren't needed to perform a specific action.
2.
Sol. **Transmission Angle:**
 - In a linkage mechanism, the transmission angle affects how effectively force is transmitted from one link to another.
 - For example, in a four-bar linkage, the transmission angle is the angle between the coupler link and the output link.
 - When the transmission angle is 90 degrees, the force is transferred most efficiently.
 - As the transmission angle deviates from 90 degrees, the efficiency of force transmission decreases.
 - Extreme values (close to 0 or 180 degrees) can lead to the mechanism stalling or jamming.
3.
Sol.
 - Equivalent mechanisms refer to different mechanical systems that produce the same output motion for a given input motion.
 - This means they achieve the same kinematic or dynamic function, even if their physical construction, number of links, or types of joints differ.
4.
Sol.
 - (a) **Rigid link:**
Deformation is negligible (microscopic) Ex:- Crank, C.R, Piston cylinder
 - (b) **Felexible links:**
Deformation are not negligible but are in perimissible limits.
Ex:- Belt drives, rope drives, chain drives.
 - (c) **Fluid Link:**
When the power is transmitted due to fluid pressure.
Ex:- Hydraulic brake, hydraulic accumulator, hydraulic ram, hydraulic crane, hydraulic jack.
5.
Sol. For the continious relative motion between the no of links in a mechanism the summation of shortest and longest link should not be greater than the summation of length of other two links.
For continous relative motion

$$s + l \leq p + q$$
 where,
 s = Shortest link
 l = Longest link
 p and q = other two link

6.

Sol. This is a special mechanism use where ∞ force or torque is required.



$$\tan \alpha = \frac{F_{i/p}}{F_{o/p}}$$

$$F_{o/p} = \frac{F_{i/p}}{\tan \alpha}$$

As $\alpha \rightarrow 0$ at the moment $\alpha \rightarrow 0$

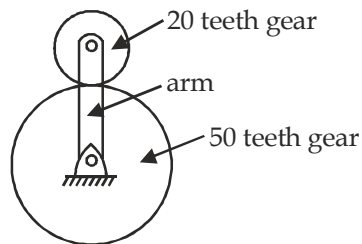
$$F_{o/p} \rightarrow \infty$$

$$M.A \rightarrow \infty$$

$$F_{o/p} \gggg F_{i/p}$$

7.

Sol. The number of degrees of freedom of the planetary gear train shown in the figure is



Number of links, $n = 4$

Number of revolute pair, $P_1 = 2 + 1 = 3$

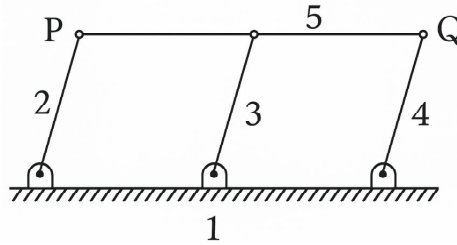
$\therefore$ 50 teeth gear and arm is double pair.

Number of lower pair, $P_2 = 1$

$$\begin{aligned} \therefore \text{DOF} &= 3(n - 1) - 2P_1 - P_2 \\ &= 3 \times 3 - 2 \times 3 - 1 \\ &= 9 - 6 - 1 = 2 \end{aligned}$$

8.

Sol.



As link 3 is treated as redundant link so it should not be considered while calculating degree of freedom.

Degree of freedom or mobility,

$$F = 3(l - 1) - 2j - h \quad \dots(1)$$

where,

$l \rightarrow$ Number of links = 4

$j \rightarrow$ Number of binary joints = 4

$h \rightarrow$ Number of higher pairs = 0

$$\begin{aligned} F &= 3(4 - 1) - 2 \times 4 - 0 \\ &= 9 - 8 \\ &= 1 \end{aligned}$$

9.

Sol. A body is said to be a resistant body, if after certain deformation it resist the forces.

10.

Sol. Given that

$$d = 25 \text{ mm}$$

$$\omega = \frac{2\pi N}{60} = 2\pi \times \frac{320}{60} = 33.51 \text{ rad / s}$$

$$\text{Maximum velocity of sliding} = \omega \cdot d = 33.51 \times 0.025 = 0.838 \text{ m/s.}$$

11.

Sol. Simple mechanism has four links. Whereas, compound mechanism has more than four links.

12.

Sol. It is the most common forms of the constraint Mechanism, that permits only relative motion of an oscillatory nature along a straight line. The mechanism used for this purpose are called straight line mechanism.

13.

- Sol.
1. Watt's Mechanism.
 2. Modified Scott - Russel Mechanism.
 3. Grasshopper Mechanism.
 4. Tchebicheff's Mechanism.
 5. Robert mechanism.

14.

Sol. Exact Straight Line Mechanisms

- Peaucellier Mechanism
- Hart Mechanism
- Scott - Russel Mechanism

15.

Sol. The steering gear mechanism is used for changing the direction of two or more of the wheel axles with reference to the chassis, so as to move the automobile desired path.



16.

- Sol.**
- The Ackerman steering gear mechanism is generally preferred over the Davis mechanism due to its simpler design and lower maintenance requirements, primarily because it utilizes turning pairs instead of sliding pairs.
 - While Davis achieves perfect steering geometry, its sliding pairs are prone to wear, leading to inaccuracies and increased maintenance over time.

17.

- Sol.** Hooke's Joint is used to connect two shafts that are not parallel shafts which are intersecting at a small angle.

18.

- Sol.** Velocity of driven shaft is not constant, it varies from maximum to minimum values. In order to have a constant velocity ratio of the driving and driven Shafts, an intermediate shaft with a hooke's joint. This type of joint is known as double Hooke's Joint.

19.

- Sol.**
1. **Geneva mechanism:-** It is used in indexing in shaper machine.
 2. **Ratchet mechanism:-** It is used in clocks.

20.

- Sol.** By velocity analysis we determine the velocity characteristic (linear, angular) of all the links of a mechanism at a given configuration, when the input velocity is specified.

21.

- Sol.**
1. Instantaneous centre method
 2. Relative velocity approach

22.

- Sol.** Instantaneous centre is the point from where the complete link will be rotation at that moment
or
For a rigid body in a plane motion there exist a point on a rigid body in a plane of motion whose instantaneous velocity is zero.

23.

Sol.

If n is the number of links in a mechanism then total number of relative I-centre is given by

$${}^nC_2 = \frac{n \times (n-1)}{2}$$

24.

Sol.

For the relative motion between the number of links in a mechanism, any three links their three I-centre must be in a single straight line.

25.

Sol.

It is used to define the velocity and acceleration of all the links of single slider crank mechanism in which cylinder is fixed, where input acceleration is zero and input angular velocity (crank) is given.

26.

Ans.

This acceleration will always be associated with slider when the slider is sliding on a rotating body. The magnitude is given by.

$$a_c = 2V\omega$$

where

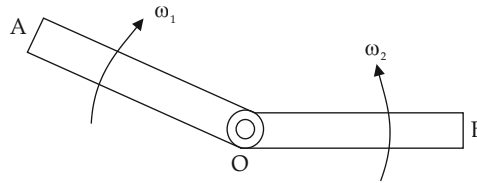
V is the velocity of slider.

ω is angular velocity of a body on which slider is sliding.

27.

Sol. Rubbing Velocity at a Pin Joint

The rubbing velocity is defined as the algebraic sum between the angular velocities of the two links which are connected by pin joints, multiplied by the radius of the pin.



Rubbing velocity at the pin joint O

= $(\omega_1 - \omega_2) r$, if the links move in the same direction

= $(\omega_1 + \omega_2) r$, if the links move in the opposite direction

28.

Sol. Given :

$$c = 1.2 \text{ m; } b = 2.7 \text{ m}$$

Let

α = Inclination of the track arm to the longitudinal axis.

We know that $\tan \alpha = \frac{c}{2b} = \frac{1.2}{2 \times 2.7} = 0.222$ or $\alpha = 12.5^\circ$

29.

- Sol.**
- Roller followers are generally preferred over knife-edged followers due to significantly lower friction and wear, especially at higher speeds.
 - While knife-edged followers have a point contact initially, they quickly develop a sliding action that leads to rapid wear and tear on both the follower and the cam.
 - Roller followers, with their rolling contact, minimize friction and extend the lifespan of the cam-follower system.

30.

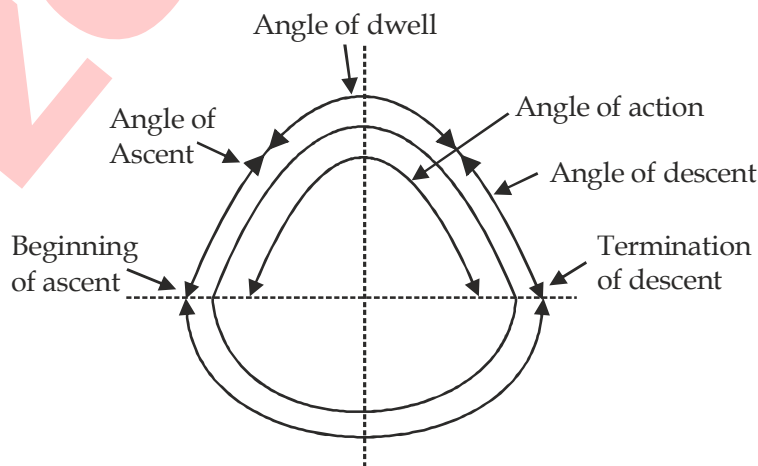
Sol.

$$t_p = 30 \text{ teeth; } m = 4 \text{ mm; } \phi = 20^\circ$$

To avoid interference, the maximum value of addendum

$$= r \sin^2 \phi = \frac{30 \times 4}{2} \times \sin^2 20^\circ = 7.0186 \text{ mm}$$

31.



Terms related to a cam's angular motion include the

1. Angle of ascent,
2. Angle of dwell,
3. Angle of descent
4. Angle of action

32.

Ans.

A cam is a mechanical member used to impart desired motion to a follower by direct contact.

33.

Ans.

If the line of movement of the follower passes through the center of rotation of cam is known as radial follower.

34.

Ans.

The pressure angle, representing the steepness of the cam profile, is the angle between the normal to the pitch curve at a point the direction of the follower motion. It varies in magnitude at all instant of the follower motion. A high value of the maximum pressure angle is not desired as it might jam the follower in the bearings.

35.

Ans.

The smallest circle drawn tangent to the pitch curve is known as the prime circle.

36.

Ans.

These type of governor a controlled by inertia forces.

37.

Ans.

Flywheel is used to control inter cycle fluctuation and governor is used to control intra cycle fluctuation.

38.

Ans.

A flywheel is used to control the variations in speed during each cycle of an engine. A flywheel of suitable dimensions attached to the crankshaft, makes the moment of inertia of the rotating parts quite large and thus, acts as a reservoir of energy.

39.

Ans.

The function of a governor is to maintain the speed of an engine within specified limits whenever there is a variation of load.

40.

Ans.

In this type of governor balls are controlled by spring. Initial the spring is fitted in compression so that a force is applied to the sleeve.

41.

Ans.

The mean force acting on the sleeve to change its equilibrium position for the fractional change in speed of governor is known as effort of governor.

42.

Ans.

The extent of sleeve movement for a change in speed of the governor is known as sensitivity of governor.

43.

Ans.

An excessively fast to and fro motion of sleeve between the stopers is known as hunting.

44.

Ans.

A governor said to be an isochronous excluding friction if its equilibrium speed is constant for all radius of rotation.

45.

Ans.

These are special type of bevel gear which is used to transmit same angular velocity to output at which input is running but at 90° .

46.

Ans.

It is a imaginary circle in the gear where pure rolling motion is absorbed when the mating gears are transmitting power. Being an imaginary circle. It cannot be the physical characteristic of the gear but being a most important circle it is one of the biggest specification of gears. The size of any gear is specified by the diameter of pitch circle.

47.

Ans.

It is the distance measured along the circumference of the pitch circle from a point on one tooth to the corresponding point on the adjacent tooth

$$p = \frac{\pi d}{T}$$

where p = circular pitch
 d = pitch diameter
 T = number of teeth

48.

Ans.

The force, which the driving tooth exerts on the driven tooth, is along a line from the pitch point to the point of contact of the two teeth. This line is also the common normal at the point of contact of the mating gears and is known as the line of action or the pressure line.

49.

Ans.

The angle between the pressure line and the common tangent to the pitch circles is known as the pressure angle or the angle of obliquity. For more power transmission and lesser pressure on the bearings, the pressure angle must be kept small. Standard pressure angles are 20° and 25° . Gears with 14.5° pressure angles have become almost obsolete.

50.

Ans.

The locus of a point on the pitch circle from the beginning to the end of engagement of two mating gears is known as the arc of contact.



51.

Ans.

An involute is defined as the locus of a point on a straight line which rolls without slipping on the circumference of a circle. Also, it is the path traced out by the end of a piece of taut cord being unwound from the circumference of a circle. The circle on which the straight line rolls or from which the cord is unwound is known as the base circle.

52.

Ans.

At any instant, the portions of the tooth profiles which are in contact must be involutes so that the line of action does not deviate. If any of the two surfaces is not an involute, the two surfaces would not touch each other tangentially and the transmission of power would not be proper. Mating of two non conjugate (non involute) teeth is known as interference.

53.

Ans.

1. Undercutting Gears
2. Increasing pressure angle (ϕ)
3. By stubbing the teeth
4. By increasing number of teeth

54.

Ans.

Due to change in centre distance, pitch circle changes, pressure angle change but base circle radius does not change and velocity ratio of gears depend upon base circle radius. Hence it will not change.

55.

Ans.

1. Ease of manufacturing
2. Constant velocity ratio maintain even when center distance is changing
3. Pressure angle remains constant throughout the engagement.

56.

Ans.

A gear train is a combination of gears used to transmit motion from one shaft to another. It becomes necessary when it is required to obtain large speed reduction within a small space.

57.

Ans.

A part from the rotation of the gears if any gear axis is also rotating with respect to some other axis then the train is known as epi-cyclic gear train. It may be simple epi-cyclic, compounded epicyclic, reverted epicyclic and so on.

To rotate the axis of the gear a link is used which is known as arm or carrier.

58.

Sol.

A Brake is a machine element which is used either to slow down or to stop a moving system, either by absorbing kinetic energy or by absorbing both kinetic energy and potential energy.

59.

Sol.

$$\frac{T_1}{T_2} = \left[\frac{1 + \mu \tan \theta}{1 - \mu \tan \theta} \right]^n$$

Where,

n = no. of shoe or blocks.



60.

Sol. There are two types of dynamometer

1. Absorption dynamometer.
2. transmission dynamometer.

61.

Sol. In absorption dynamometers, the entire power produced by the prime mover is absorbed by the Frictional resistance of the brake and is transformed into heat during the process of measurement.

62.

Sol. In transmission dynamometers, the energy is used for doing work. The power developed by the prime mover, is transmitted through the dynamometer to some other machine where the power is suitably measured.

63.

Sol. **Dry Friction**

Dry friction is said to occur when there is relative motion between two completely unlubricated surfaces. It is further divided into two types:

(a) **Solid Friction**

When the two surfaces have a sliding motion relative to each other, it is called a solid friction.

(b) **Rolling Friction**

Friction due to rolling of one surface over another (e.g., ball and roller bearings) is called rolling friction

64.

Sol. **Skin or Greasy Friction**

When the two surfaces in contact have a minute thin layer of lubricant between them, it is known as skin or greasy friction. Higher spots on the surface break through the lubricant and come in contact with the other surface.

Skin friction is also termed as boundary friction.

65.

Sol. **Film Friction**

When the two surfaces in contact are completely separated by a lubricant, friction will occur due to the shearing of different layers of the lubricant. This is known as film friction or viscous friction.

66.

Ans. **Laws of Friction**

Experiments have shown that the force of solid friction

- is directly proportional to normal reaction between the two surfaces
- opposes the motion between the surfaces
- depends upon the materials of the two surfaces
- is independent of the area of contact
- is independent of the velocity of sliding

The last of these laws is not true in the strict sense as it has been found that the friction force decreases slightly with the increase in velocity.

67.

Sol. Brakes are devices that slow down or stop motion, typically by converting kinetic energy into heat through friction. Dynamometers, on the other hand, measure the power output of a machine, often by applying a controllable load and measuring torque and speed.

68.

Sol. In an internal expanding shoe brake, the leading shoe is the one that rotates into the brake drum's surface when the brake is applied, while the trailing shoe moves away from the drum's surface. The leading shoe experiences a greater braking force due to the direction of rotation, which can create a self-energizing effect.

69.

Sol. • **Absorption:** Measures power by absorbing and dissipating it.
• **Transmission:** Measures power by allowing it to be transmitted, typically through a shaft.



70.

Sol. Balancing is the process of designing or modifying machinery so that the unbalance is reduced to an acceptable level and if possible is eliminated entirely.

71.

Sol. When several masses rotate in different planes, the centrifugal forces, in addition to being out of balance, also form couples. A system of rotating mass is in dynamic balance when there does not exist any resultant centrifugal force as well as resultant couple.

72.

Sol. In reciprocating mass system unbalance force is constant in direction but varying in magnitude due to which a single constant mass cannot be attached to the crank but there is no other position to connect a balance mass which is rotating at same speed of crank, that is why we need to balance it partially.

73.

Sol. Unbalanced primary forces along the lines of stroke are separated by a distance l apart and thus, constitute a couple. This is called swaying couple.

74.

Sol. Hammer blow is the maximum vertical unbalanced force caused by the mass provided to balance the reciprocating masses. Its value is $m r \omega^2$. Thus it varies as a square of speed. At high speeds, the force of the hammer blow could exceed the static load on the wheels and the wheels can be lifted off the rail when the direction of the hammer blow will be vertically upwards.

75.

Sol. If the axis of a spinning or rotating body is given an angular motion about an axis perpendicular to the axis of spin, an angular acceleration acts on the body about the third perpendicular axis. The torque required to produce this acceleration is known as the active gyroscopic torque.

76.

Sol. A gyroscope is specified as a device that can maintain orientation regardless of base movement and has a spinning disc installed on it that may freely rotate in several directions.

77.

Sol. Gyroscopic couple is a phenomenon resulting from the conservation of angular momentum. When a rotating disc is tilted, the gyroscopic couple acts perpendicular to the applied force and the axis of rotation, causing precession.

78.

Sol. Gyroscopic couple induces precession, causing the axis of the rotating disc to rotate around a vertical axis when subjected to an external force.

79.

Sol. Answer: Gyroscopic couple arises due to the conservation of angular momentum. The spinning disc resists changes in its orientation to maintain the original direction of angular momentum.

80.

Sol. Gyroscopic effects are utilized in gyroscopes for navigation systems, aircraft control, and stabilization in various engineering applications.

81.

Sol. The gyroscopic couple in rotating wheels contributes to stability by resisting changes in orientation, helping vehicles maintain balance.

82.

Sol. Yes, the gyroscopic effect can be intentionally used to control the orientation of a rotating disc. This is a fundamental principle in the design of gyroscopic devices.

83.

Sol. The gyroscopic couple resists the external torque, causing the rotating disc's axis to shift in a way that minimizes the applied torque. This behavior is part of the stabilizing effect.

84.

Sol. Yes, the gyroscopic effect becomes more pronounced at higher rotation speeds. The magnitude of the gyroscopic couple is proportional to the angular velocity of the rotating disc.



85.

- Sol.**
- Logarithmic decrement is a measure of how quickly the amplitude of a damped vibration decreases over time.
 - It's defined as the natural logarithm of the ratio of two successive amplitudes of oscillation, on the same side of the equilibrium position.
 - In essence, it quantifies the rate at which the oscillations decay.

86.

Sol. There are three important types of vibrations from subject point of view:

1. Free or natural vibrations,
2. Damped vibrations,
3. Force vibrations.

87.

Sol. If the vibrations of a particle after giving it an initial displacement remain continued, then the vibrations are called free or natural vibrations. No external force acts on the particle. In other words, the vibrations of the particle with fundamental frequency under the influence of the restoring force are called free vibrations.

88.

Sol. The vibrations of a body whose amplitude goes on reducing over every cycle of vibrations are known as damped vibrations. This is due to the fact that a certain amount of energy possessed by the vibrating body is always dissipated in overcoming frictional resistance to the motion.

89.

Sol. When the body vibrates under the influence of external periodic force, then the vibrations are known as forced vibrations.
The body does not vibrate with its natural frequency, but it vibrates with the frequency of the driver.

90.

Sol. When the frequency of external force is equal to the natural frequency of the vibrations, resonance takes place, amplitude or deformation or displacement will reach to its maximum at resonance and the system will fail due to breakdown. This state of disturbing force on the vibrating body is known as the state of resonance.

91.

Sol. There are three planes in **Gyroscopic effect**-

1. plane of precession.
2. plane of Spinning
3. plane of active gyroscopic couple.

92.

Sol. **axis of spin:** It is axis about which the body revolves.

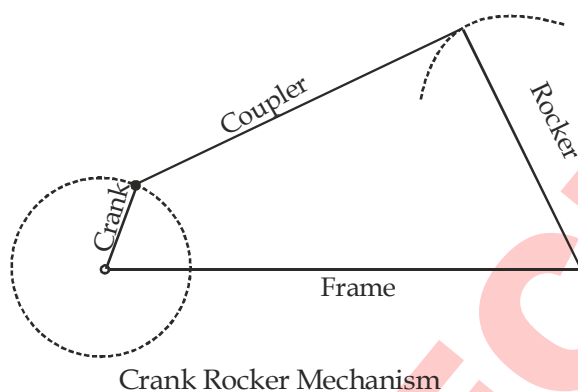
precession: it mean the rotation about third axis, which is perpendicular to both the Spin axis and couple axis, is called the axis of precession.

PART - B

5 Marks Questions

93.

Sol. Crank rocker mechanism:

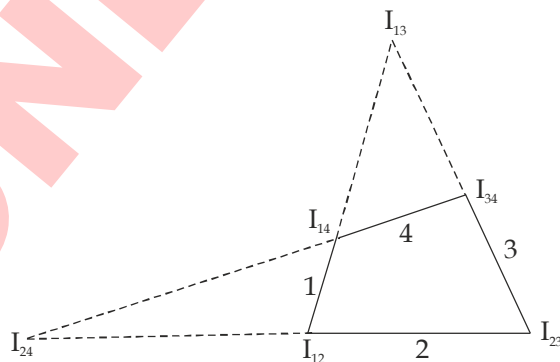
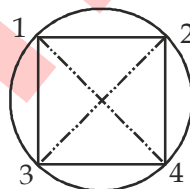


Crank Rocker Mechanism

Identified instantaneous centres.

Total number of link, $n = 4$

$$\text{Number of I centres} = \frac{n(n-1)}{2} = 6$$



Identification of Instantaneous Centres

94.

Sol. Mechanism :

- When one of the links of a kinematic chain is fixed, the chain is known as mechanism.
- It may be used for transmitting or transforming motion e.g. engine indicators, typewriter etc.
- A mechanism with four links is known as simple mechanism, and the mechanism with more than four links is known as compound mechanism.
- When a mechanism is required to transmit power or to do some particular type of work, it then becomes a machine.
- In such cases, the various links or elements have to be designed to withstand the forces (both static and kinetic) safely.



S.No.	Mechanism	Machine
1.	Mechanism transmits and modifies the motion.	Machine changes the mechanical work.
2.	A mechanism is an outline of the machine to produce define motion between various links of joints.	Machine will have many mechanisms for transmitting the mechanical work or power.
3.	Where kinematic chain is analyzed as a mechanism, no special consideration need to be given to the formed cross-section of the links.	In a machine, cross-section and proportions are to be considered to give stiffness, strength and clearance to the paths of the machine.
4.	Examples of mechanisms are clocks, typewriter, steering mechanism in a car, etc.	Examples or the machine are slotting, lathe, shaper in the workshop, (Shaper receives mechanical power which is suitably converted to do the work of cutting the metal), etc.

95.

Sol. (i) **Analysis vs. Synthesis of Mechanisms:****Analysis:**

Analysis of a mechanism involves studying the motion and forces within an existing mechanism. It determines how the mechanism will behave given its geometry and input motion, focusing on things like position, velocity, acceleration, and the forces acting on the links.

Synthesis:

Synthesis of a mechanism is the process of designing a mechanism to achieve a desired motion or task. It involves selecting the type of mechanism, determining the dimensions of its links, and ensuring it meets specific performance requirements.

Key Difference:

Analysis looks at the "what is" of a mechanism, while synthesis focuses on the "what should be".

(ii) **Kinematics vs. Dynamics:****Kinematics:**

Kinematics is the study of motion without considering the forces that cause the motion. It focuses on the geometric aspects of motion, like displacement, velocity, and acceleration.

Dynamics:

Dynamics is the study of motion that considers the forces and torques that cause the motion. It examines the relationship between forces, masses, and the resulting motion.

Key Difference:

Kinematics is a subset of dynamics. Kinematics provides the framework for describing motion, while dynamics adds the dimension of force and its influence on that motion.

96.

Sol. **Lower and Higher Pairs:****Lower Pair:**

In a lower pair, the two elements in contact have surface or area contact during motion. The relative motion is typically sliding or turning. Examples include a shaft rotating in a bearing, a screw and nut, or a slider-crank mechanism.



Higher Pair:

In a higher pair, the elements have line or point contact. This means the contact surfaces are dissimilar, and the motion is often a combination of rolling and sliding. Examples include a cam and follower, mating gears, or a wheel rolling on a surface.

Closed and Unclosed Pairs:**Closed Pair:**

A closed pair is one where the two elements are held together by mechanical means and their motion is constrained. This means they stay in contact without the need for external forces to maintain the connection. Examples include a pin joint, a slider in a guide, or a cam and follower mechanism.

Unclosed (or Force-Closed) Pair:

In an unclosed pair, external forces are required to maintain contact between the two elements. A cam and follower mechanism is a common example; it requires a spring or other force to keep the follower in contact with the cam.

Turning and Rolling Pairs:**Turning Pair:**

A turning pair, also known as a revolute pair, involves one link rotating or turning relative to another. An example is a shaft rotating within a bearing, or the links of a four-bar linkage.

Rolling Pair:

A rolling pair is formed when one link rolls on another without slipping. A wheel rolling on a surface is the most straightforward example. Rolling pairs can also be found in roller bearings and some gear systems.

97.

Sol. The various types of constrained motion include:

1. Completely Constrained Motion:

In this type, the motion is restricted in such a way that only one direction of motion is possible, and the motion is entirely predictable. An example is a piston moving inside a cylinder, where the motion is constrained to a linear path.

2. Incompletely Constrained Motion:

Here, the motion is not fully restricted, and more than one direction of motion is possible. The direction of motion depends on the external forces applied. An example is a shaft rotating in a bearing, which can rotate in either direction unless an external force dictates otherwise.

3. Successfully Constrained Motion:

This is a special case of incompletely constrained motion where the constraints are applied in such a way that only the desired motion is achieved, even though multiple motions are theoretically possible. This is often achieved through the design of the mechanism itself. For instance, in a footstep bearing, the weight of the shaft prevents upward movement, ensuring only rotational motion.

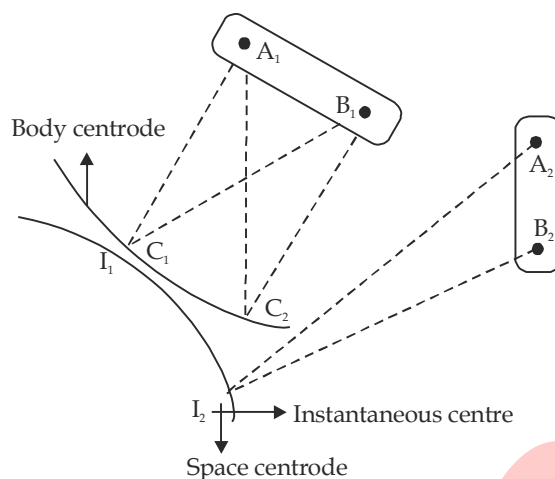
98.

Sol. **Body centrode and Space centrode:**

- The instantaneous centre is a point in the body which may be considered fixed at any particular instant of time, the locus of instantaneous centre in space during a definite motion of body is called space centrode. The locus of instantaneous centre relative to the body is called body centrode. The body centrode rolls without slipping over the space centrode.

$$A_1C_2 = A_2I_2$$

$$A_1C_2 = B_2I_2$$



In similar way $C_3, C_4, \dots$ can be found out and body centrode can be drawn.

99.

Sol. Three I-centers of three rigid bodies in relative motion shall lie on the same straight line. i.e. three I-centres corresponding to three rigid bodies in relative motion shall be collinear.

or

This theorem states that if three-plane bodies have relative motion among themselves their three Instantaneous Centres must lie on a straight line.

100.

Sol. In mechanisms, instantaneous centers represent points around which links appear to rotate at a specific instant.

There are three types: fixed, permanent, and neither fixed nor permanent (also called secondary).

(i) **Fixed Instantaneous Centers:**

These centers are located at fixed points in space relative to the mechanism. They don't move regardless of the mechanism's configuration. An example is the center of a fixed pivot or a point on a link that is rigidly attached to the ground.

(ii) **Permanent Instantaneous Centers:**

These centers are associated with joints that are part of the mechanism and remain in the same relative position throughout its motion. For instance, the center of a pin joint connecting two links is a permanent instantaneous center.

(iii) **Neither Fixed Nor Permanent Instantaneous Centers (Secondary Centers):**

These centers are neither fixed nor permanently associated with specific joints. Their location changes as the mechanism moves. They are determined through analysis using Kennedy's Theorem or other methods, and they are crucial for understanding the velocity of different points on the mechanism.

101.

Sol. Coriolis component of acceleration:-

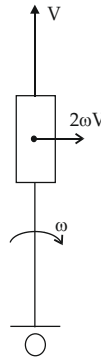
When a point on one link is sliding along another rotating link such as in quick return motion mechanism, then the Coriolis component of acceleration comes into account.

It is the tangential component of the acceleration of the slider with respect to the coincident point on the link.

Mathematically, Coriolis component of acceleration, $a_c = 2v\omega$

Where,

- a_c is the Coriolis acceleration.
- ω is the angular velocity vector of the rotating reference frame.
- V is the velocity vector of the moving object relative to the rotating frame.



The direction of the Coriolis component of acceleration is given by rotating the velocity of the slider by 90° in the direction of the angular velocity of the rotating link.

102.

Sol. It is uniform acceleration and retardation motion (outstroke angle)

$$\phi_a = 60^\circ = \frac{\pi}{3} \text{ rad}$$

$$N_{\text{cam}} = 300 \text{ rpm}$$

$$\omega_{\text{cam}} = \frac{2\pi \times 300}{60} = 5\pi \times 2 = 10\pi \text{ rad/s}$$

$$s = 20 \text{ mm} = 0.020 \text{ m}$$

In this motion:

$$(V_0)_{\text{max}} = 2(V_0)_{\text{mean}} = \frac{2s\omega}{\phi_a} = \frac{2 \times 0.020 \times 10\pi}{\pi/3} = 1.2 \text{ m/s}$$

In first half of stroke

$$(V_0)_{\text{max}} = 0 + a \times \frac{t_0}{2}$$

$$\frac{2\omega s}{\phi_a} = a \times \frac{\phi_a/2}{\omega}$$

$$a = \frac{4\omega^2 s}{\phi_a^2} = \frac{4 \times (10\pi)^2 \times 0.020}{\left(\frac{\pi}{3}\right)^2} = 8 \times 9 = 72 \text{ m/s}^2$$

103.

Sol. Given : $N = 500 \text{ r.p.m.}$ or $\omega = 2\pi \times 500 / 60 = 52.4 \text{ rad/s}$

Let α = Greatest permissible angle between the centre lines of the shafts.

Since the variation in speed of the driven shaft is $\pm 6\%$ of the mean speed (i.e. speed of the driving speed), therefore total fluctuation of speed of the driven shaft,

$q = 12\%$ of mean speed (ω) = 0.12ω

We know that maximum or total fluctuation of speed of the driven shaft (q),

$$0.12\omega = \omega \left(\frac{1 - \cos^2 \alpha}{\cos \alpha} \right) \text{ or } \cos^2 \alpha + 0.12 \cos \alpha - 1 = 0$$

and

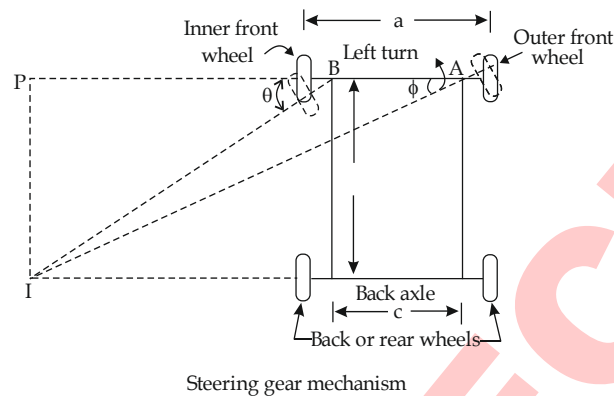
$$\cos \alpha = \frac{-0.12 \pm \sqrt{(0.12)^2 + 4}}{2} = \frac{-0.12 \pm 2.0036}{2} = 0.9418$$

...(Taking + sign)

$$\alpha = 19.64^\circ$$

104.

Sol.



- The condition for correct steering is that all the four wheels must turn about the same instantaneous centre. The axis of the inner wheel makes a larger turning angle θ than the angle ϕ subtended by the axis of outer wheel.

Let

 a = Wheel track, b = Wheel base, and c = Distance between the pivots A and B of the front axle.

Now from triangle IBP,

$$\cos \theta = \frac{BP}{IP}$$

and from triangle IAP,

$$\cot \phi = \frac{AP}{IP} = \frac{AB + BP}{IP} = \frac{AB}{IP} + \frac{BP}{IP} = \frac{c}{b} + \cot \theta$$

$$\therefore \cot \phi - \cot \theta = c / b$$

This is the fundamental equation for correct steering. If this condition is satisfied, there will be no skidding of the wheels, when the vehicle takes a turn.

- The Ackerman steering gear mechanism is much simpler than Davis gear. The difference between the Ackerman and Davis steering gears are:
 - The whole mechanism of the Ackerman steering gear is on back of the front wheels; whereas in Davis steering gear, it is in front of the wheels.
 - The Ackerman steering gear consists of turning pairs, whereas Davis steering gear consists of sliding members.

105.

Sol. For a constant acceleration cam, the cam accelerate in the first half of rise and decelerate with same value in the second half.

For follower displacement.

Displacement, $s = \frac{1}{2}ft^2$

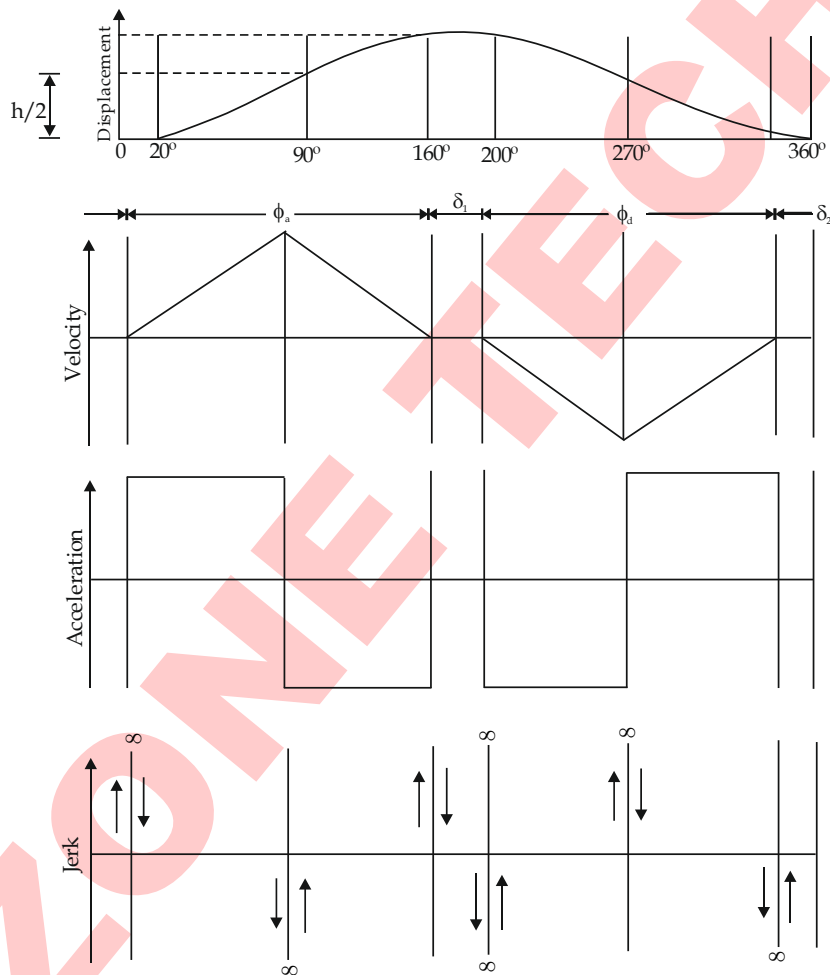
Acceleration, $f = \frac{2s}{t^2} = \text{const } a$

At halfway motion $s = \frac{h}{2}$ and $t = \frac{\phi/2}{\omega}$

$$f = \frac{2h/2}{(\phi/2\omega)^2} = \frac{4h\omega^2}{\phi}$$

Velocity is linear given by

$$\frac{ds}{dt} = ft = \frac{4h\omega^2}{\phi} \frac{\theta}{\omega} = \frac{4h\omega\theta}{\phi}$$



106.

Sol. It is uniform acceleration and retardation motion, ϕ_a (outstrike angle)

$$\phi_a = 60^\circ = \frac{\pi}{3} \text{ rad}$$

$$N_{\text{cam}} = 300 \text{ rpm}$$

$$\omega_{\text{cam}} = \frac{2\pi \times 300}{60} = 5\pi \times 2 = 10\pi \text{ rad/s}$$

$$s = 20 \text{ mm} = 0.020 \text{ m}$$

In first half of stroke

$$(V_0)_{\text{max}} = 0 + a \times \frac{t_0}{2}$$

$$\frac{2\omega s}{\phi_a} = a \times \frac{\phi_a / 2}{\omega}$$

$$a = \frac{4\omega^2 s}{\phi_a^2} = \frac{4 \times (10\pi)^2 \times 0.020}{\left(\frac{\pi}{3}\right)^2} = 8 \times 9 = 72 \text{ m/s}^2$$

107.

Sol. **Knife-edge Follower:**

- **Advantages:**

Simple design, compact size, suitable for low-speed applications.

- **Disadvantages:**

High wear on both the follower and cam surface due to line contact, not suitable for high-speed or heavy-load applications, can lead to cam surface damage over time.

Roller Follower:

- **Advantages:**

- Reduced wear due to rolling contact, suitable for high-speed applications, lower friction compared to knife-edge followers, can handle higher loads.

- **Disadvantages:**

More complex design than knife-edge, requires lubrication for the roller bearing.

Mushroom (Flat-faced) Follower:

- **Advantages:**

Large contact area reduces Hertzian stresses, suitable for high loads, less susceptible to wear than knife-edge followers.

- **Disadvantages:**

Requires a perfectly flat cam surface, susceptible to high side thrust, can experience high friction, especially with misalignment.

108.

Sol. **Sensitiveness:-** A governor is said to be sensitive when it readily responds to a small change of speed. The movement of the sleeve for a fractional change of speed is the measure of sensitivity.

$$\text{Sensitiveness} = \frac{N}{N_2 - N_1} = \frac{(N_2 + N_1)}{2(N_2 - N_1)}$$

Where $N_2 - N_1$ = speed range from no load to full load.

When N = mean speed

N_1 = minimum speed corresponding to full load conditions

N_2 = maximum speed corresponding to no-load conditions.

Stability: A governor is said to be stable if it brings the speed of the engine to the required value and there is not much hunting. The ball masses occupy a definite position for each speed for the engine within the working range. Obviously, the stability and the sensitivity are two opposite characteristics.

Isochronous : A governor with a range of speed zero is known as an isochronous governor. This means that for all positions of the sleeve or the balls, the governor has the same equilibrium speed. However, it is not practical due to the friction at the sleeve.

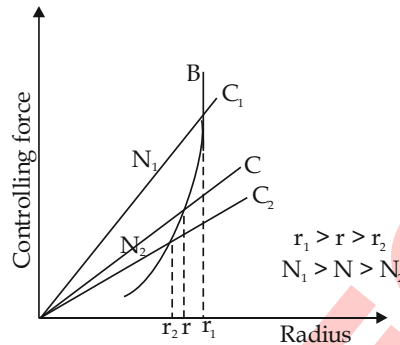
Hunting: A governor is said to be hunt if the speed of the engine fluctuates continuously above and below the mean speed. This is caused by too sensitive governor which changes the fuel supply by a large amount when a small change in speed of rotation takes place.

A governor is said to be isochronous when the equilibrium speed is constant (i.e., range of speed is zero) for Degree of hunting is more in unstable governors.

109.

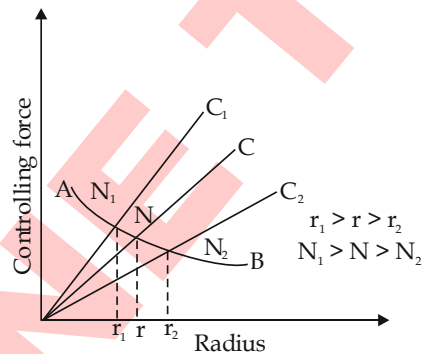
Sol.

(i)



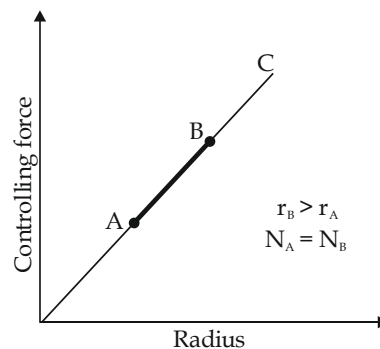
As speed N increase, radius increases. So this governor is stable.

(ii)



As speed N decreases, radius increases. So this governor is unstable.

(ii)



For all positions of governor, speed is constant. So this governor is isochronous.

110.

Sol. Difference Flywheel and Governor :

Flywheel	Governor
Flywheel reduces the fluctuation of speed during the thermodynamic cycles, but it does not maintain a constant speed.	Governor is a device to control the speed variation caused by the varying load.
The working of a flywheel does not depend upon the change in load or output required.	Governor operation depends upon the variation of load
The operation of flywheels is continuous from cycle to cycle.	The operation of a governor is intermittent.
Speed control in a single cycle.	Speed control over a period of time.
The function of a flywheel is to store energy when mechanical energy is more than required for the operation and release the same when the available energy is less than required its inertia helps to run machines at a dead center.	The function of a governor is to regulate the fuel supply according to the load requirement and run the machine at a constant speed irrespective of the output required.
Do not have any control over the supply of charge or fuel.	Control the supply of fuel to the engine.
It is relatively heavy and has large inertia force.	It's a light machine part.
It is used in engines and fabricating machines such as punching machines, rolling mill, etc.	Governors are provided on engines and turbines.
It is desired where the fluctuation in input torque, eg. four stroke engine.	Desired where the constant speed required, eg Generator (there is even electronic governor for diesel generator)

111.

Sol. Turning moment diagram for single cylinder four stroke cycle internal combustion engine :

- In case of a four-stroke internal combustion engine, the diagram repeats itself after every two revolutions (i.e. 720° or 4π radians).

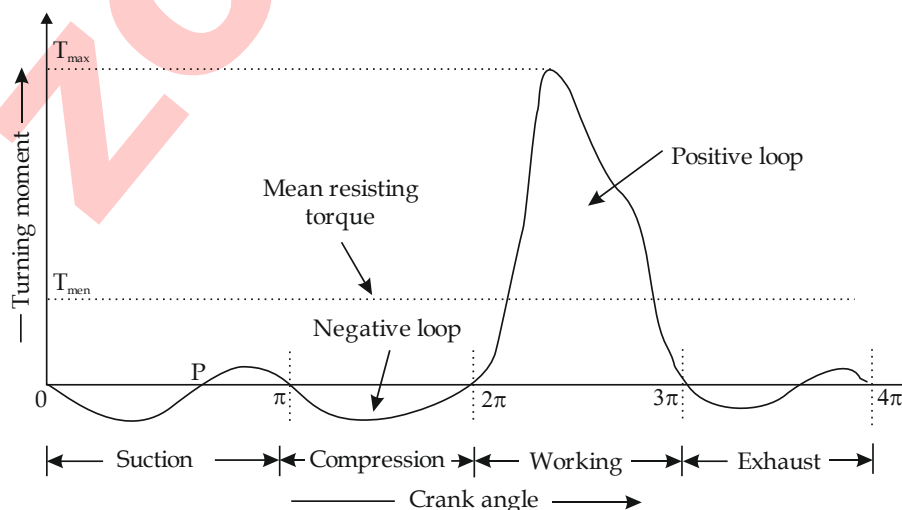
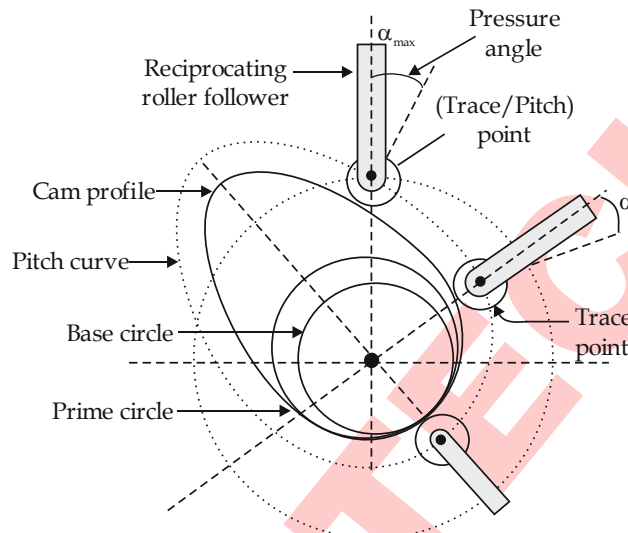


Fig.:- Turning moment diagram (single cylinder four stroke engine)

- It can be seen from the diagram that for the majority of the suction stroke, the turning moment is negative but becomes positive after the point P.
- During the compression stroke, it is totally negative.
- It is positive throughout the expansion stroke and again negative for most of the exhaust strokes.

112.

Sol. Terminology of radial cam:



1. **Base circle** : It is the smallest circle tangent to the cam profile (contour) drawn from the center of rotation of a radial cam.
2. **Pitch point** : It is point on the pitch curve of the cam having the maximum pressure angle.
3. **Pitch circle** : It is a circle drawn from the center of the cam through the pitch point. It is concentric with based circle.
4. **Prime circle**: It is the smallest circle that can be drawn from the center of the cam and tangent to the pitch curve. For a knife edge and a flat face follower, the prime circle and the base circle are identical. For a roller follower, the prime circle is larger than the base circle by the radius of the roller.
5. **Pressure angle (α)** : It represents the steepness of cam profile, is the angle between the normal to the pitch curve at a point and the direction of follower motion. A high value of maximum pressure angle is not desired. Its magnitude varies at all instant of follower motion.
6. **Pitch point** : It is point on the pitch curve of the cam having the maximum pressure angle.
7. **Trace point** : It is a reference point on the follower and is used to generate the pitch curve. In case of knife edge follower, the knife edge represents the trace point and the pitch curve corresponds to the cam profile. In a roller follower, the centre of the roller represents the trace point.
8. **Lift or stroke**: It is the maximum travel of the follower from its lowest position to the topmost position.
9. **Pitch curve**: It is the curve generated by the trace point as the follower moves relative to the cam. For a knife edge follower, the pitch curve and the cam profile are same whereas for a roller follower, they are separated by the radius of the roller.

113.

Sol. Pressure angle for gear train :

- The sides of each gear tooth incline at an angle called the pressure angle. For many years, the standard pressure angle was 14.5° . Today, the use of 14.5° gearing is generally limited to replacement work, while the 20° pressure angle is more commonly used. This switch has occurred because a 20° pressure angle results in a higher strength gear tooth that offers better wear-

resistance characteristics. The use of a 20° angle also permits pinions with fewer teeth to be used.

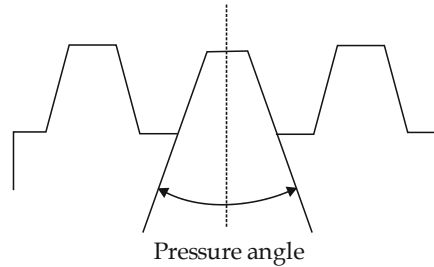


Fig.- Pressure angle (gearing)

- The pressure angle of a gear is also the angle between the line of action and the line tangent to the pitch circles of mating gears. Below figure, illustrates the relationship of the pressure angle to the line of action and the line tangent to the pitch circles.

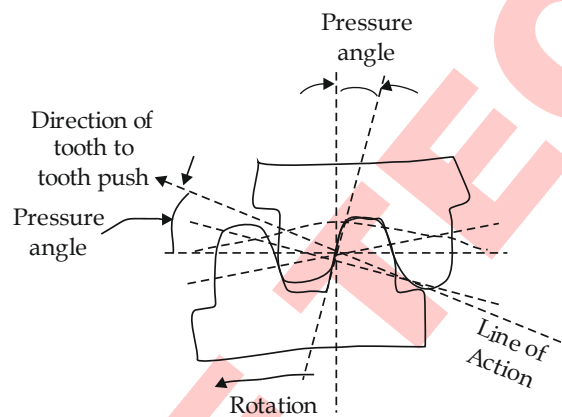


Fig.- Pressure angle (gearing)

Pressure angle for cam follower :

Pressure angle (α): It represents the steepness of cam profile, is the angle between the normal to the pitch curve at a point and the direction of follower motion. A high value of maximum pressure angle is not desired. Its magnitude varies at all instant of follower motion.

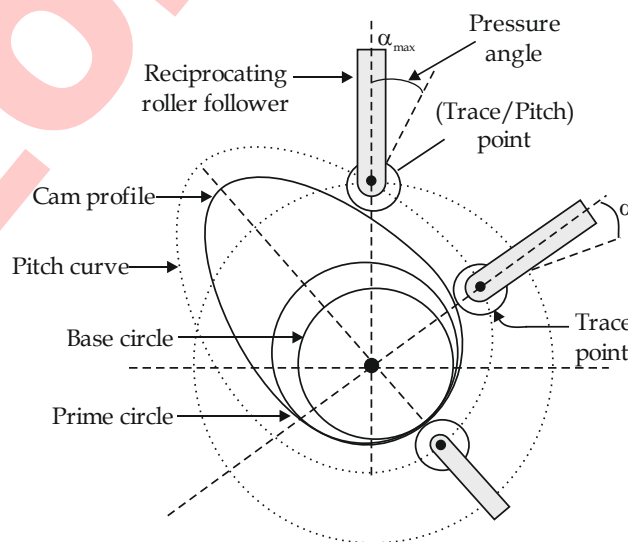


Fig.- Pressure angle (cam-follower)



114.

Sol.

(i) Given,

$$\phi = 14.5^\circ,$$

$$m = 5 \text{ mm},$$

$$t_p = 15,$$

$$T_g = 60,$$

$$A_G = m = 1 \text{ m},$$

$$G = \text{Gear ratio} = 4$$

$$\begin{aligned} \text{Minimum number of teeth on gear} &= \frac{2A_G}{m\sqrt{1 + \frac{1}{G}\left(2 + \frac{1}{G}\right)\sin^2\phi} - 1} = \frac{2}{\sqrt{1 + \frac{1}{4}\left(2 + \frac{1}{4}\right)\sin^2(14.5^\circ)} - 1} \\ &= \frac{2}{0.01747} = 114.42 \text{ teeth} \approx 115 \text{ so interference will occur} \end{aligned}$$

(ii) We keep teeth = 60

$$\begin{aligned} 60 &= \frac{2}{\sqrt{1 + \{0.25(2.25)\sin^2\phi\}} - 1} \\ \Rightarrow \sqrt{1 + 0.5625\sin^2\phi} - 1 &= 0.03333 \\ 1 + 0.5625\sin^2\phi &= 1.067 \\ \sin^2\phi &= 0.120 \\ \sin\phi &= 0.347 \\ \phi &= 20.31^\circ \end{aligned}$$

115.

Sol. Each have 30 teeth, minimum contact ratio = 2, circular pitch = 2.5 cm

$$\phi = 20^\circ$$

$$\pi m = 2.5 \text{ cm} = \text{circular pitch}$$

$$\Rightarrow m = 0.796 \text{ cm} = 7.96 \text{ mm} \approx 8 \text{ mm}$$

$$\text{Pitch radius of gear, } R = \frac{m \times z}{2} = \frac{8 \times 30}{2} = \frac{240}{2} \text{ mm} = 120 \text{ mm}$$

$$\text{Now, contact ratio} = \frac{\text{Area of contact}}{\text{Circular pitch}} = \frac{2\left[\sqrt{(R_A)^2 - (R \cos \phi)^2} - R \sin \phi\right]}{\cos \phi \times (2.5)} \quad (\text{as radius same})$$

$$\Rightarrow 0.085\sqrt{(R_A)^2 - (112.76)^2} - 41.0 \geq 2$$

$$\sqrt{R_A^2 - (112.76)^2} - 41 \geq 23.53$$

$$\sqrt{R_A^2 - (112.76)^2} = 64.53$$

$$R_A^2 = 4164.12 + 12714.8$$

$$R_A^2 = 16878.92$$

$$R_A = 129.91 \text{ mm}$$

$$A_G = R_A - R = 129.91 - 120 = 9.91 \text{ m}$$

$$A_G = 9.91 \text{ mm} = \text{addendum}$$

116.

Sol.

Pitch diameter of small gear = 120 mm

Module, $m = 8 \text{ mm}$ Gear ratio, $G = 4:3$

$$\text{Pitch diameter of gear, } D = 120 \times \frac{4}{3} = 160 \text{ mm}$$

$$(i) \quad \text{Number of teeth on gear, } T = \frac{D}{m} = \frac{160}{8} = 20 \text{ teeth}$$

$$(ii) \quad \text{Number of teeth on pinion, } t = \frac{3}{4} \times T = \frac{3}{4} \times 20 = 15 \text{ teeth}$$

$$(iii) \quad \text{Addendum for gears, } A = 1m = 1 \times 8 = 8 \text{ mm}$$

$$(iv) \quad \text{Dedendum} = 1.25m = 1.25 \times 8 = 10 \text{ mm}$$

$$(v) \quad \text{Whole depth} = \text{Addendum} + \text{Dedendum} = 8 + 10 = 18 \text{ mm}$$

$$(vi) \quad \text{Clearance, } c = 0.25m = 0.25 \times 8 = 2.00 \text{ mm}$$

117.

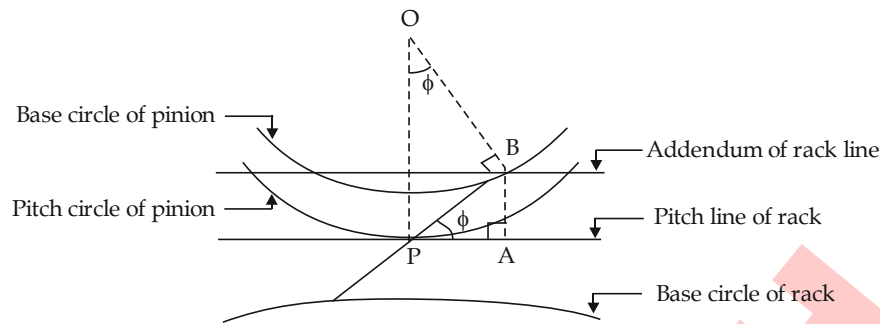
Sol.

Sl.	Cycloidal Teeth	Involute Teeth
1	Pressure angle varies from maximum at the beginning of engagement, reduce to zero at the pitch point and again increase to maximum at the end of engagement resulting in less smooth running of the gears	Pressure angle is constant throughout the engagement of teeth. This results in smooth running of the gears.
2	It involves double curve for the teeth, epicycloid and hypocycloid. This complicates the manufacturing.	It involves single curve for the teeth resulting in simplicity of manufacturing of tools.
3	Costlier as they are difficult to manufacture	Cheaper as they are easier to manufacture.
4	Exact center-distance is required to transmit a constant velocity ratio.	A little variation in the center distance does not affect the velocity ratio.
5	Phenomenon of interference does not occur at all.	Interference can occur if the condition of minimum number of teeth on a gear is not followed.
6	The teeth have spreading flanks and thus are stronger.	The teeth have radial flanks and thus are weaker as compared to the cycloidal form for the same pitch.
7	In this, a convex flank always has contact with a Concave face resulting in less wear.	Two convex surfaces are in contact and thus there is more wear.



118.

Sol. Consider an involute rack and pinion meshing having a module of 'm'.



The critical case for interference would be when addendum line of rack cuts the base circle of pinion. Let addendum of the rack be $A_R \times m$, where A is the fraction by which the standard addendum of one module for the rack is to be multiplied.

So, $A_B = A_R \times m$... (i)

From triangle OPB, $PB = OP \sin \phi$... (ii)

From triangle PBA, $AB = PB \sin \phi$... (iii)

Using equations (i), (ii) and (iii)

$$A_R \times m = OP \sin^2 \phi$$

Now,

$$OP = \frac{m \times t_{\min}}{2}$$

So,

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

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

119.

Sol. Laws of Friction :

(i) **Law of static friction**

- The frictional force always acts in a direction opposite to that in which the body tends to move.
- The frictional force is directly proportional to the normal reaction between the surfaces.
- The frictional force depends upon the nature of surfaces in contact.
- The frictional force independent of the area and shape of the contacting surfaces.

(ii) **Laws of dynamic or kinetic friction**

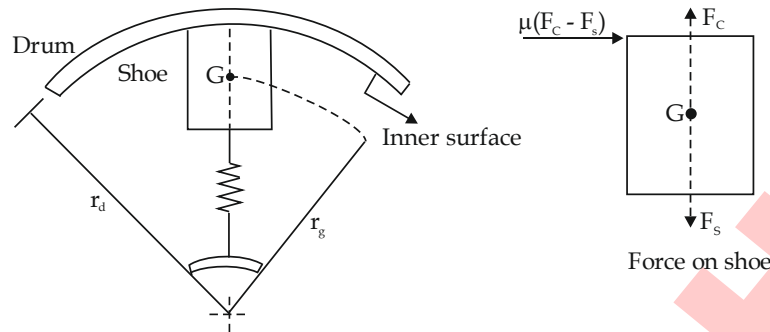
- The frictional force always acts in a direction opposite to that in which the body moves.
- The frictional force is directly proportional to the normal reaction between the two contacting surfaces.
- The magnitude of force of dynamic friction bears a constant ratio to the normal reaction between two surfaces but the ratio is slightly less than that in case of limiting friction.
- The frictional force remains constant for moderate speeds but it decreases slightly with the increase of speed.

It may be noted that :

- For extremely low pressure and for very high pressures sufficient to produce excessive deformation, the co-efficient of static friction, somewhat increases.
- For extremely low relative velocities, the co-efficient of kinetic friction increases and apparently becomes equal to the co-efficient of static friction.
- For very high velocities co-efficient of kinetic friction decreases appreciably.
- Ordinary changes in temperature do not materially affect co-efficient of friction.

120.

Sol.



The centrifugal clutch works on the principle of centrifugal force. As the angular speed of input shaft increases, the centrifugal force acting on the sliding shoes increases, causing the shoes to move in a radially outward direction. The shoes continue to move with increasing speed until they contact inner surface of drum. Power is transmitted due to frictional force between shoe lining and the inner surface of drum.

Application: Lawn movers, golf cart, cranes, cement mills, three wheelers.

121.

Sol.

Inner radius, $r_1 = 40 \text{ mm} = 0.04 \text{ m}$

Outer radius, $r_2 = 80 \text{ mm} = 0.08 \text{ m}$

$$W(\text{axial force}) = 2\pi P_{\max} r_1 (r_2 - r_1)$$

$$P_{\max} = \frac{W}{2\pi r_1 (r_2 - r_1)} = \frac{4 \times 10^3}{2 \times 3.14 \times 0.04 \times 0.04}$$

$$P_{\max} = 398 \text{ kN/m}^2$$

$$P_{\min} = \frac{W}{2\pi r_2 (r_2 - r_1)} = \frac{4 \times 10^3}{2 \times 3.14 \times 0.08 \times 0.04}$$

$$P_{\min} = 199 \text{ kN/m}^2$$

$$P_{\text{average}} = \frac{W}{\pi(r_2^2 - r_1^2)} = \frac{4 \times 10^3}{3.14(0.08^2 - 0.04^2)}$$

$$P_{\text{average}} = 265.4 \text{ kN/m}^2$$

122.

Sol.

Given: $D = 320 \text{ mm}$; $w = 0.03 \text{ m}$; $P_{\max} = P_n^l = 80 \text{ kPa}$, $\phi_1 = 15^\circ$, $\phi_2 = 145^\circ$, $a = 220 \text{ mm}$, $c = 125 \text{ mm}$, $\mu = 0.32$

For leading shoe,

$$F \times a - \int_{\phi_1}^{\phi_2} P_n^l c \sin \theta \int_{\phi_1}^{\phi_2} \mu P_n^l (r - c \cos \theta) = 0$$

$$F \times 0.22 - \frac{0.16 \times 0.125 \times 0.03}{4} \times (80 \times 10^3) \left[2 \times \frac{\pi}{180} \times 130 - \sin 290 + \sin 30 \right]$$

$$+ \frac{0.32 \times 0.16 \times 0.03 (80 \times 10^3)}{4} [4 \times 0.16 (\cos 15 - \cos 145) - 0.125 (\cos 30 - \cos 290)] = 0$$

$$F \times 0.22 - 12 \times 5.977 + 30.72 [1.0769] = 0$$

$$F = 175.673 \text{ N}$$



For the trailing shoe,

$$175.643 \times 0.22 - 8.9655 \times 10^{-4} P_n^t - 4.135 \times 10^{-4} \times P_n^t = 0$$

$$P_n^t = 29496.172 \text{ N / m}^2$$

∴ Braking torque

$$\begin{aligned} T_B &= r^2 \mu w (P_n^l + P_n^t) (\cos \phi_1 - \cos \phi_2) \\ &= 0.16^2 \times 0.32 \times 0.03 (80000 + 29496.172) (\cos 15 - \cos 145) \end{aligned}$$

$$T_N = 48.036 \text{ N}$$

123.

Sol.

• **Friction**

Friction is a force that opposes motion between two surfaces in contact. It arises from the irregularities and intermolecular forces between the surfaces.

• **Friction a Blessing or a Curse**

Friction can be both a blessing and a curse, depending on the context.

• **Blessing:**

Friction is essential for many everyday activities. For example, it allows us to walk without slipping, enables vehicles to brake and move, and helps us hold objects securely.

• **Curse:**

Friction can also be undesirable as it causes wear and tear on machinery, generates heat (leading to energy loss), and makes it harder to move objects, requiring more effort.

124.

Sol.

Torsion Dynamometer

A torsion dynamometer is used for measuring large powers particularly the power transmitted along the propeller shaft of a turbine or motor vessel. A little consideration will show that when the power is being transmitted, then the driving end of the shaft twists through a small angle relative to the driven end of the shaft. The amount of twist depends upon many factors such as torque acting on the shaft (T), length of the shaft (l), diameter of the shaft (D) and modulus of rigidity (C) of the material of the shaft. We know that the torsion equation is

$$\frac{T}{J} = \frac{C \cdot \theta}{l}$$

where

θ = Angle of twist in radians, and

J = Polar moment of inertia of the shaft.

For a solid shaft of diameter D, the polar moment of inertia

$$J = \frac{\pi}{32} \times D^4$$

and for a hollow shaft of external diameter D and internal diameter d, the polar moment of inertia,

$$J = \frac{\pi}{32} \times (D^4 - d^4)$$

From the above torsion equation,

$$T = \frac{C \cdot J}{l} \times \theta = k \cdot \theta$$

where $k = C \cdot J / l$ is a constant for a particular shaft. Thus, the torque acting on the shaft is proportional to the angle of twist. This means that if the angle of twist is measured by some means, then the torque and hence the power transmitted may be determined.

We know that the power transmitted

$$P = \frac{T \times 2\pi N}{60} \text{ watts, where N is the speed in r.p.m.}$$



125.

Sol. As per given data:

$$\text{Stroke length for lower piston} = 2r_{\text{lower}} = 80 \text{ mm}$$

$$(r_{\text{lower}})_{\text{crank}} = 40 \text{ mm}$$

Lets θ is the orientation of the centre crank

$$\text{Reciprocating piston mass, } (m)_{\text{lower}} = 10 \text{ kg}$$

$$\text{Reciprocating piston mass } (m)_{\text{upper}} = 20 \text{ kg}$$

and outer crank is oriented at $180 + \theta$.

By assuming primary forces is balanced then stroke of the upper piston.

As we know,

$$(F_{\text{primary}})_{\text{lower}} + (F_{\text{primary}})_{\text{upper}} = 0 \quad \dots(i)$$

$$(F_{\text{primary}})_{\text{lower}} = mr\omega^2 \cos \theta = 10 \times 40 \times \omega^2 \cos \theta \quad \dots(ii)$$

$$(F_{\text{primary}})_{\text{upper}} = mr\omega^2 \cos \theta (180 + \theta) = -20 \times r \times \omega^2 \cos \theta \quad \dots(iii)$$

From equations (i) and (ii), we get

$$\begin{aligned} r_{\text{upper}} &= 20 \text{ mm} \\ \text{Length of stroke} &= 40 \text{ mm} \end{aligned}$$

126.

Sol. As per given data

$$\text{speed of engine, } N = 300 \text{ rpm}$$

$$\Rightarrow \omega = \frac{2\pi \times 300}{60} = 10\pi \text{ rad / s}$$

$$\text{Stroke length, } 2r = 300 \text{ mm} = 0.3 \text{ m}$$

$$\text{Crank radius, } r = 0.15 \text{ m}$$

$$\text{mass of reciprocating part, } m = 50 \text{ kg}$$

$$\text{Mass of revolving part, } m_p = 37 \text{ kg at } 150 \text{ mm radius}$$

$$c = \frac{2}{3}rd \text{ of the reciprocating parts}$$

$$r_c = 300 \text{ mm } 0.3 \text{ m}$$

$$(i) \text{ mass to be balanced at the crank pin, } m = cm + m_p$$

$$= \frac{2}{3} \times 50 + 37 = 70.333 \text{ kg}$$

$$m_c r_c = mr$$

$$m_c \times 0.3 = 70.333 \times 0.15$$

$$m_c = 35.166 \text{ kg}$$

The balance mass required at a radius of 300 mm

$$m_c = 35.166 \text{ kg}$$

(ii) Residual unbalanced force (at $60^\circ = \theta$) from top dead centre,

$$= \sqrt{[(1-c)mr\omega^2 \cos \theta]^2 + (cmr\omega^2 \sin \theta)^2}$$

$$= \sqrt{\left[\left(1 - \frac{2}{3}\right)50 \times 0.15 \times (10\pi)^2 \cos 60^\circ\right]^2 + \left[\frac{2}{3} \times 50 \times 0.15 \times (10\pi)^2 \sin 60^\circ\right]^2}$$

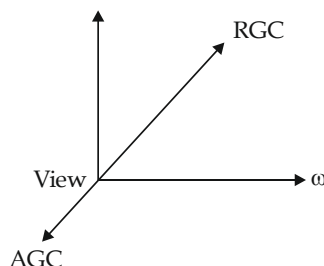
$$= \sqrt{1522017.047 + 18264204.57} = 4448.170 \text{ N}$$



127.

Given: $m = 2.5 \times 10^3 \text{ kg} = 2500 \text{ kg}$, $N = 1750 \text{ rpm (CW)}$, $\theta = \frac{\pi \times 1750}{30} = 183.2596 \text{ rad/s}$, $k = 0.32 \text{ m}$,

$$I = mk^2 = 256 \text{ kg.m}^2$$

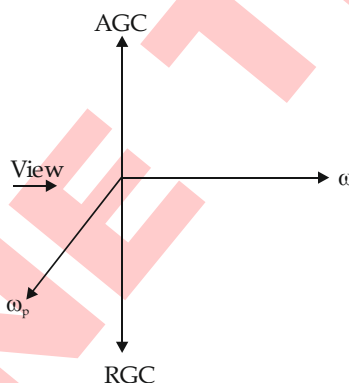


$$\omega_p = \frac{V}{R} = \frac{30 \times 5}{18 \times 250} = 0.0333 \text{ rad/s}$$

Gyroscopic couple,

$$C = I\omega\omega_p = 256 \times 183.2596 \times 0.0333 = 1562.25 \text{ N-m}$$

Effect: To lower the bow and raise the stern, when the ship turns right.
(ii)



$$\omega_p = 0.7 \text{ rad/s} \quad [\text{Given}]$$

$$\text{Gyroscopic couple, } C = I\omega\omega_p = 256 \times 183.2596 \times 0.7 = 32840.12 \text{ N-m}$$

Effect: The effect of the reaction couple when bow is rising, is to turn the ship towards right or towards starboard.

(iii)

Gyroscopic couple,

$$\omega_p = 0.2 \text{ rad/s}$$

$$C = I\omega\omega_p = 256 \times 183.2596 \times 0.2 = 9382.89 \text{ N-m}$$

Effect: As the axis of spin is always parallel to the axis of precession for all positions, there is no gyroscopic effect on the ship.

128.

Sol.

- **Gyroscopic Effect:** When a spinning object is subjected to a force that tries to change the direction of its spin axis, it doesn't immediately tilt in that direction. Instead, it moves in a direction perpendicular to both the applied force and its original spin axis. This phenomenon is called gyroscopic precession.



- **Gyroscopic Couple:** The torque that causes this precession is called the gyroscopic couple. It's the force that keeps the gyroscope stable and prevents it from simply tilting over when a force is applied to change its orientation.
- **Magnitude:** The magnitude of the gyroscopic couple (C) is given by the formula:

$$C = I\omega\omega_p$$

Where:

- I: is the moment of inertia of the rotating body (e.g., a gyroscope's rotor).
- ω : is the angular velocity of the spinning body (spin speed).
- ω_p : is the angular velocity of precession (the rate at which the spin axis changes direction).

PART - C

20 Marks Questions

129.

Sol. (i) **KINEMATIC PAIRS ACCORDING TO NATURE OF CONTACT :**

(a) **Lower Pair :**

- A pair of links having surface or area contact between the members is known as a lower pair.
- The contact surfaces of the two links are similar.
- **Examples :**
Nut turning on a screw, shaft rotating in a bearing, all pairs of a slider-crank mechanism, universal joint.

(b) **Higher Pair :**

- When a pair has a point or line contact between the links, it is known as a higher pair.
- The contact surfaces of the two links are dissimilar.
- **Examples :**
Wheel rolling on a surface cam and follower pair, tooth gears, ball and roller bearings, etc.

(ii) **KINEMATIC PAIRS ACCORDING TO NATURE OF MECHANICAL CONSTRAINT :**

(a) **Closed Pair :**

- When the elements of a pair are held together mechanically, it is known as a closed pair.
- The contact between the two can only be broken only by the destruction of at least one of the members.
- All the lower pairs and some of the higher pairs are closed pairs.

(b) **Unclosed Pair :**

- When two links of a pair are in contact either due to force of gravity or some spring action, they constitute an unclosed pair.
- In this the links are not held together mechanically.
- **Example :**
Cam and follower pair.

(iii) **KINEMATIC PAIRS ACCORDING TO NATURE OF RELATIVE MOTION :**

(a) **Sliding Pair :**

- If two links have a sliding motion relative to each other, they form a sliding pair.
- A rectangular rod in a rectangular hole in a prism is an example of a sliding pair.

(b) **Turning Pair :**

- When one link has a turning or revolving motion relative to the other, they constitute a turning pair or revolving pair.

(c) **Rolling Pair :**

- When the links of a pair have a rolling motion relative to each other, they form a rolling pair.
- A rolling wheel on a flat surface, ball and roller bearings, etc. is some of the examples for a Rolling pair.



(d) Screw Pair (Helical Pair) :

- If two mating links have a turning as well as sliding motion between them, they form a screw pair.
- This is achieved by cutting matching threads on the two links.
- The lead screw and the nut of a lathe is a screw Pair

(e) Spherical Pair :

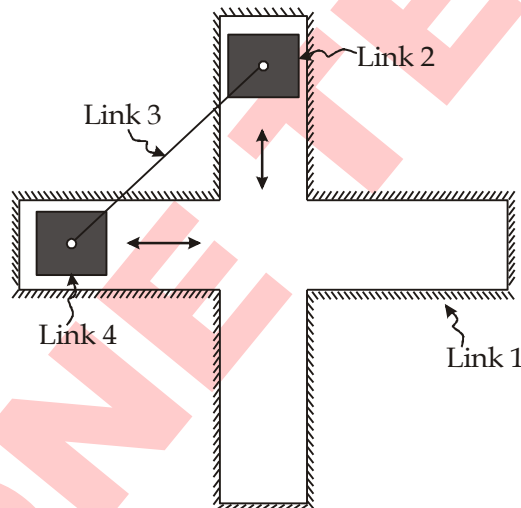
- When one link in the form of a sphere turns inside a fixed link, it is a spherical pair.
- The ball and socket joint is a spherical pair.

130.

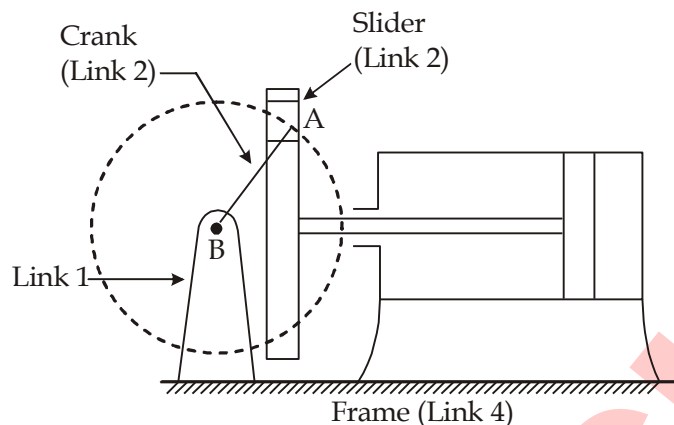
Sol. Inversion of Double Slider Crank Mechanism :

- The double slider crankshaft has two sliding pairs and two turning pairs.
- It is named double slider crankshaft because it has two sliding pairs.
- It has two sliders, a frame in which the slider moves, and a link that connects the two slides and fixes the distance between the two sliders.
- The frame in which sliders move is a combination of two straight grooves that form a single link and cut each other at right angles.

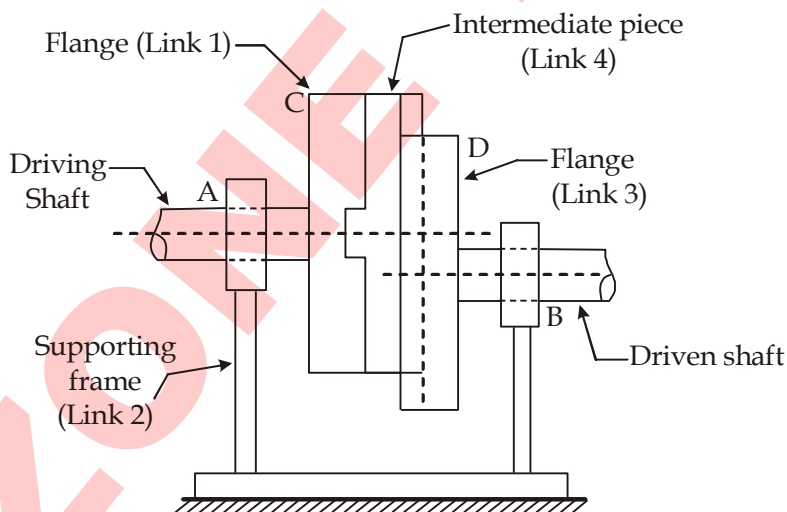
There are three important inversions of the double slider crank chain. These three are :

(i) Elliptical Trammels :

- This type of inverse is used to draw the ellipse. It is also known as the trend of Archimedes. It can be used to draw ellipses of different sizes.
- In this inverse link 1, i.e., the frame is fixed. Both sliders, link 2 and link 4, run in the grooves of this frame. One slider moves vertically, and the other slider moves horizontally.
- The connecting rod link 3, which connects the two sliding bars, link 2 and link 4, constrain the motion of these sliding bars. While the two sliders move, each point in the connecting rod is the location of an ellipse.
- The different points in the connecting rod will draw a different ellipse.

(ii) **Scotch Yoke Mechanism :**

- Scotch Yoke mechanisms are used for converting rotary motion into a reciprocating motion.
- The inversion is obtained by fixing either link 1 or link 3.
- In this mechanism, when link 2 (which corresponds to crank) rotates about B as center, link 4 (which corresponds to a frame) reciprocates. The fixed link 1 guides the frame.
- Now, the second slider will rotate around the fixed slider and will draw a circle if its path is traced. If the frame or slanted plate will move forward and if the horizontal moving slider is fixed and the frame will move up and down, then the vertical moving slider is fixed.

(iii) **Oldham's Coupling :**

- Oldham couplings are used to connect two shafts whose axles are separated from each other and rotate them at equal speed.
- An Oldham coupling mechanism is used to connect two parallel shafts whose axes are separated at short distances. The shaft is coupled in such a way that if one shaft rotates, the other shaft also rotates at the same speed. This inverse is achieved by fixing link 2. The shaft to be connected have two flanges (link 1 and link 3), fastened strictly to their ends by forging.
- It forms the coupling of the inverted Oldham. Now we will discuss how it is used to connect two shafts whose axis is slightly different from each other.
- This shaft is connected in such a way that if one shaft rotates, the other shaft also rotates at the same speed.



- The two flanges are placed at the end of the two shafts that are to be connected. These flanges are rigidly fastened at the end by forging. Both flanges have grooves.
- The groove of both flanges is perpendicular to each other. Between these two flanges, a central flange is inserted, which fits into the grooves of both flanges.
- In this coupling, which is fixed frame links 3, driving shaft and driving flange for link 2, intermediate flange link 1, and driveshaft and flange form link 4.
- When the first shaft, i.e., The driving shaft, rotates, the forged one also rotates, and then the central flange placed in the grooves of the driving flange starts rotating.
- Then the driven flange also starts rotating because the center flange is also fixed in the grooves of the driven flange, and as the driven flange rotates, the driven shaft also starts rotating at the same speed.

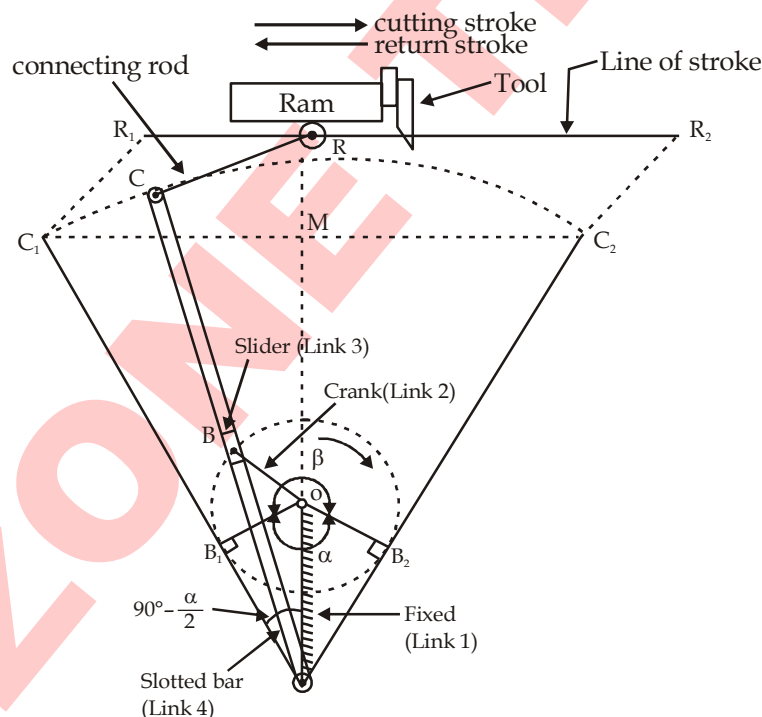
131.

Sol. A quick return mechanism is a mechanism which converts circular motion (rotating motion following a circular path) into reciprocating motion (repetitive back and forth or to and fro linear motion) in presses and shaping machines.

There are three types of quick return mechanism

- Hydraulic shaper drive
- Crank and slotted link mechanism
- Whitworth mechanism

Following are the application of quick return mechanism



- Shaper
- Power driven saw
- Revolver mechanisms
- Screw press
- Mechanical actuator

In the extreme position AP_1 and AP_2 are tangential to the circle and the cutting tool is at the end of the stroke. The forward of cutting stroke occurs when the crank rotates from the position CB_1 to CB_2 (or through an angle β) in the clockwise direction. The return stroke occurs when the crank rotates from the position CB_2 and CB_1 (or through angle α) in the clockwise direction. Since the crank has uniform angular speed.

Therefore,

$$\frac{\text{Time of cutting stroke}}{\text{Time of return stroke}} = \frac{\beta}{\alpha}$$

$$= \frac{\beta}{360^\circ - \beta} \text{ or } \frac{360^\circ - \alpha}{\alpha}$$

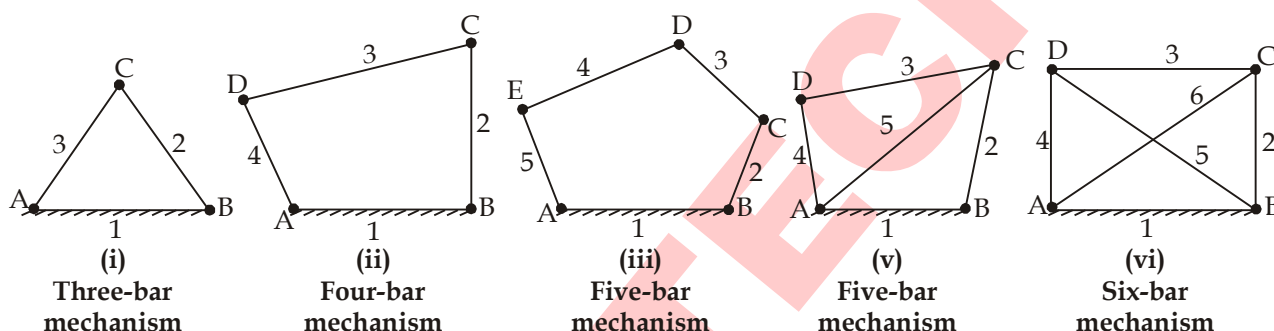
132.

Sol.

$$n = 3(L - 1) - 2j - h$$

Where, L = Number of links
 j = Number of binary joints

This equation is called Kutzbach's Criterion for determining the number of degrees of freedom of movability (n) of a plane mechanism.



The no. of degree of freedom or movability (n) for some simple mechanisms having no higher pair (i.e. $h = 0$) as shown in fig. are determined as follows;

(i) The mechanism as shown in figure (i) has three links and three binary joints.

i.e. $L = 3, j = 3$
 $n = 3(3 - 1) - 2 \times 3 = 0$

(ii) The mechanism as shown in figure (ii) has four links and four binary joints.

i.e. $L = 4, j = 4$
 $n = 3(4 - 1) - 2 \times 4 = 1$

(iii) The mechanism as shown in figure (iii) has five links and five binary joints.

i.e. $L = 5, j = 5$
 $n = 3(5 - 1) - 2 \times 5 = 2$

(iv) The mechanism as shown in figure (iv) has five links and six equivalent binary joints (because there are two binary joints at B and D, and two ternary joints at A and C)

i.e. $L = 5, j = 6$
 $n = 3(5 - 1) - 2 \times 6 = 0$

(v) The mechanism as shown in figure (v) has six links and eight equivalent binary joints (because there are four ternary joints at A, B, C, and D)

i.e. $L = 6, j = 8$
 $n = 3(6 - 1) - 2 \times 8 = -1$

Grubler's Criterion for plane mechanism :

The Grubler's criterion applies to the mechanism to the only single degree of freedom joints where the overall movability of the mechanism is unity.

Substituting $n = 1$ and $h = 0$ in kutzbach equation, we have,

$$1 = 3(L - 1) - 2j$$

or $3L - 2j - 4 = 0$

This equation is known as the Grubler's criterion for mechanisms with constrained motion.

For a simple mechanism, the degree of freedom (F) is given by the Grubler's criterion :

$$F = 3(L - 1) - 2j - h$$

Where, j = Number of revolute joints
 L = Number of links
 h = Number of higher pairs

For a constrained planer mechanism, DOF is 1 and $h = 0$

$$\Rightarrow 1 = 3(L - 1) - 2j - 0$$

$$\Rightarrow 3L - 2j = 4$$

$$\Rightarrow 2j = 3L - 4$$

The RHS of the equation should be positive and even number,

Put $L = 3$

$$2j = 3 \times 3 - 4 = 5 \text{ odd}$$

Put $L = 4$

$$2j = 3 \times 4 - 4 = 8 \text{ even}$$

So, the minimum value of n should be 4.

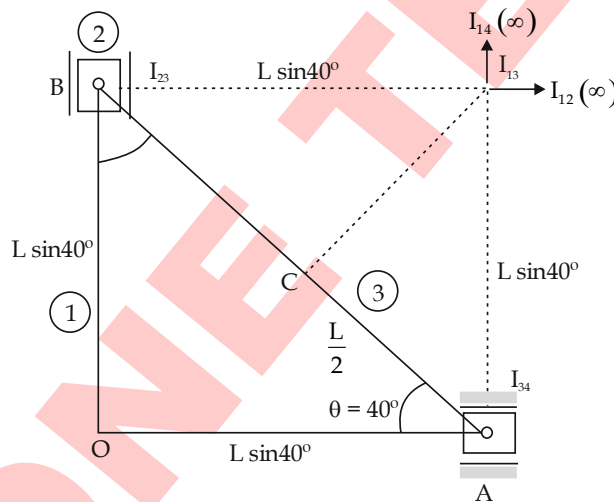
133.

Sol.

Given: Velocity of slider A, $V_A = 3 \text{ m/s}$

Angle made by slider with horizontal, $\theta = 40^\circ$

Let 'L' be the length of the link joining slider A and B.



For motion of link AB,

$$\frac{V_A}{I_{13}A} = \frac{V_B}{I_{13}B} = \frac{V_C}{I_{13}C}$$

$$\Rightarrow \frac{V_A}{I_{13}A} = \frac{V_C}{I_{13}C} \quad \dots(i)$$

From the above configuration,

$$\begin{aligned} (I_{13}C)^2 &= \left(\frac{L}{2}\right)^2 + (L \sin 40^\circ)^2 - 2\left(\frac{L}{2}\right)(L \sin 40^\circ) \cos 50^\circ \\ &= \frac{L^2}{4} + (L \sin 40^\circ)^2 - L^2 \sin 40^\circ \cos 50^\circ \\ &= L^2 (0.25 + \sin^2 40^\circ - \sin 40^\circ \cos 50^\circ) \end{aligned}$$

$$(I_{13}C)^2 = 0.25 \times L^2$$

$$(I_{13}C) = \sqrt{0.25 \times L^2}$$

$$I_{13}C = 0.5 \times L = \frac{L}{2} \quad \dots(ii)$$

Now, from equation (i) and (ii), we get

$$\Rightarrow \frac{V_c}{\left(\frac{L}{2}\right)} = \frac{3}{L \sin 40^\circ}$$

$$\Rightarrow V_c = \frac{3}{2 \sin 40^\circ} \Rightarrow V_c = 2.3336 \text{ m/s}$$

134.

Sol. As per given data: $l = 175.0 \text{ mm}$, $r = 50 \text{ mm}$, $n = \left(\frac{l}{r}\right) = \frac{175}{50} = 3.5$ and $N = 400 \text{ rpm}$

$$\Rightarrow \omega = 2 \times \pi \times 400 / 60 = 41.887 \text{ rad/s}$$

We know,
$$v = \omega r \left(\sin \theta + \frac{\sin 2\theta}{2n} \right)$$

Velocity of piston,
$$v = 2.09 \left(\sin \theta + \frac{\sin 2\theta}{7} \right)$$

at

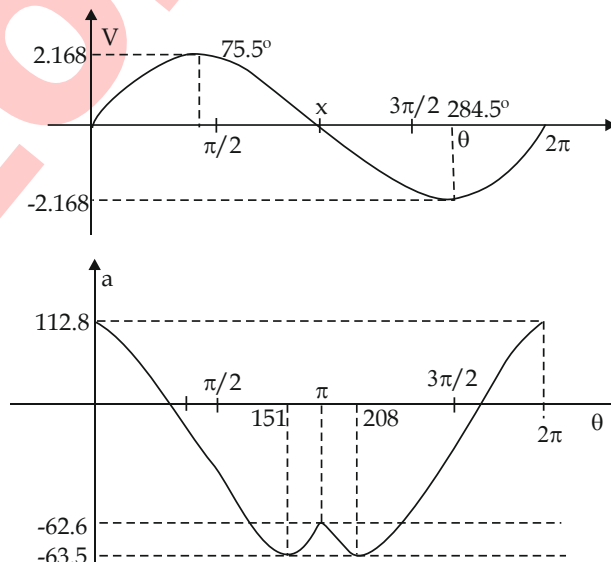
$$\theta = 0^\circ \quad V = 0$$

$$\theta = 90^\circ \quad V = \omega r = 2.094 \text{ m/s}$$

$$\theta = 180^\circ \quad V = -2.094 \text{ m/s}$$

$$\frac{dv}{d\theta} = \omega r \left(\cos \theta + \frac{1}{2n} \cdot (\cos 2\theta) 2 \right)$$

(for maximum or minimum)



$$\cos \theta + \frac{\cos \theta}{\pi} = 0 \quad (\text{for maximum or minimum})$$

Solving, $\cos \theta = \frac{1}{4}$

$$\Rightarrow \theta = 75.52^\circ, 284.478^\circ$$

$$\theta = 75.52^\circ$$

$$V = 2.09 \left(\sin 75.52^\circ + \frac{\sin 2 \times 75.52^\circ}{7} \right)$$

$$= 2.168 \text{ m/s}$$

Similarly for $\theta = 284.478^\circ$, $V = 2.168 \text{ m/s}$

Now,

$$a = 87.77 \left(\cos \theta + \frac{\cos 2\theta}{3.5} \right)$$

If $a = 0 \Rightarrow \cos \theta + \frac{\cos 2\theta}{3.5} = 0$

$$\Rightarrow 2 \cos^2 \theta - 1 = -3.5 \cos \theta$$

$$2 \cos^2 \theta + 3.5 \cos^2 \theta - 1 = 0$$

$$\cos \theta = 0.25$$

$$\theta = 75.52^\circ \text{ and } 284.5^\circ$$

at $\theta = 0^\circ \quad a = 112.80$

$$\theta = 90^\circ \quad a = -25.06$$

$$\theta = 180^\circ \quad a = 62.66$$

$$\theta = 270^\circ \quad a = -25.06$$

$$\theta = 360^\circ \quad a = 112.8$$

$$\frac{da}{d\theta} = \omega^2 y \left(-\sin \theta \frac{\sin 2\theta}{n} \cdot 2 \right) = 0$$

$$\Rightarrow \sin \theta = \frac{2 \sin \theta a u \theta \times 2}{n}$$

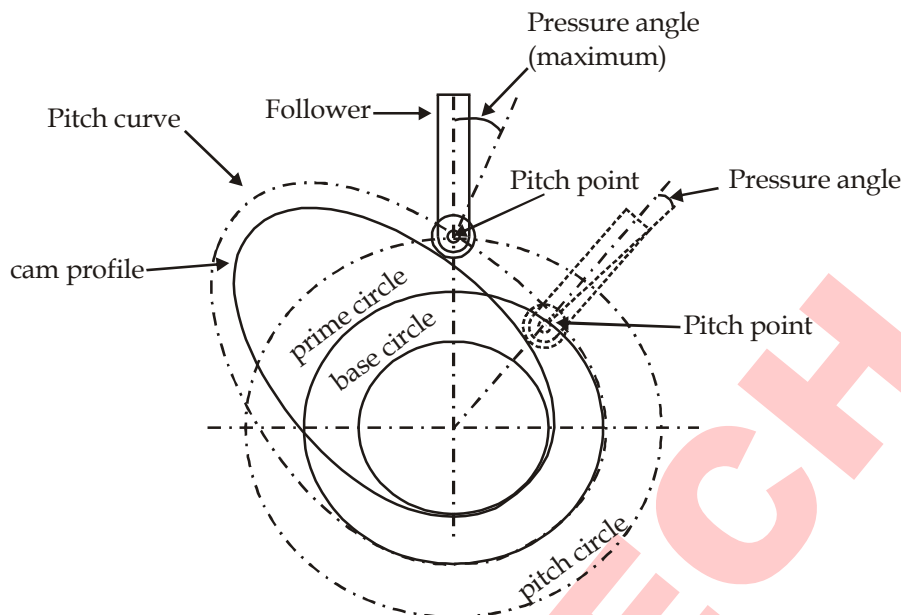
$$\cos \theta = -\frac{n}{4} = 0.875$$

$$\theta = 151.05^\circ, 208.95^\circ$$

135.

Sol. Terms Used in Radial Cams :

Figure shows a radial cam with reciprocating roller follower. The following terms are important in order to draw the cam profile.



(i) **Base Circle :**

It is the smallest circle that can be drawn to the cam profile.

(ii) **Trace Point :**

It is a reference point on the follower and is used to generate the *pitch curve*. In case of knife edge follower, the knife edge represents the trace point and the pitch curve corresponds to the cam profile. In a roller follower, the centre of the roller represents the trace point.

(iii) **Pressure Angle :**

It is the angle between the direction of the follower motion and a normal to the pitch curve, this angle is very important in designing a cam profile, if the pressure angle is too large, a reciprocating follower will jam in its bearings.

(iv) **Pitch Point :**

It is a point on the pitch curve having the maximum pressure angle.

(v) **Pitch Circle :**

It is a circle drawn from the centre of the cam through the pitch points.

(vi) **Pitch Curve :**

It is the curve generated by the trace point as the follower moves relative to the cam. For a knife edge follower, the pitch curve and the cam profile are same whereas for a roller follower, they are separated by the radius of the roller.

(vii) **Prime Circle :**

It is the smallest circle that can be drawn from the centre of the cam and tangent to the pitch curve. For a knife edge and a flat face follower, the prime circle and the base circle are identical. For a roller follower, the prime circle is larger than the base circle by the radius of the roller.

(viii) **Lift or Stroke**

It is the maximum travel of the follower from its lowest position to the topmost position.

136.

- Sol.** Given: Lift, $h = 1.2 \text{ cm} = 12 \text{ mm}$
 Least radius of cam, $r_c = 2 \text{ cm} = 20 \text{ mm}$
 Rotational speed, $N = 960 \text{ rpm}$
 Nose radius, $r_n = 0.3 \text{ cm} = 3 \text{ mm}$
 Mass of moving part, $m = 250 \text{ g} = 0.250 \text{ kg}$

Crank rotation duration of the exhaust valve = $180^\circ - 4^\circ + 50^\circ = 226^\circ$

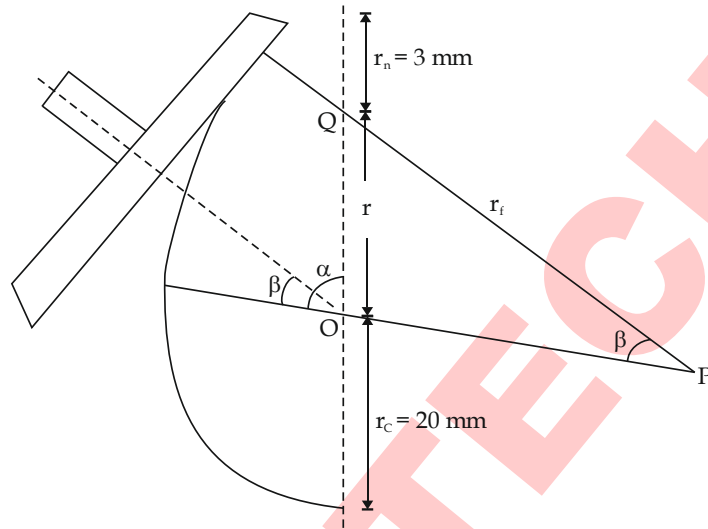
In four-stroke engines, the camshaft speed is half that of the crankshaft.

Angle of action of the camshaft,

$$2\alpha = \frac{226^\circ}{2} = 113^\circ$$

$\Rightarrow$

$$\alpha = 56.5^\circ$$



From the figure,

$$r + r_n = r_c + h$$

$\Rightarrow$

$$r = r_c + h - r_n = 20 + 12 - 3 = 29 \text{ mm}$$

In $\triangle OPQ$, we can write,

$$(PQ)^2 = (OP)^2 + (OQ)^2 - 2(OP)(OQ)\cos(\angle POQ)$$

$\Rightarrow$

$$(r_f - 3)^2 = (r_f - 20)^2 + (29)^2 - 2(r_f - 20)(29)\cos(180^\circ - 56.5^\circ)$$

$\Rightarrow$

$$r_f^2 + 9 - 6r_f = r_f^2 + 400 - 40r_f + 841 + (32.0123)(r_f - 20)$$

$\Rightarrow$

$$1.9877r_f = 1232 + 640.246$$

$\Rightarrow$

$$r_f = 297.707 \text{ mm}$$

Applying sine rule to $\triangle OPQ$,

$$\frac{OQ}{\sin \beta} = \frac{PQ}{\sin(180^\circ - \alpha)}$$

$\Rightarrow$

$$\frac{r}{\sin \beta} = \frac{r_f - r_n}{\sin(180^\circ - 56.5^\circ)}$$

$\Rightarrow$

$$\frac{29}{\sin \beta} = \frac{297.707 - 3}{\sin(123.5^\circ)}$$

$\Rightarrow$

$$\sin \beta = 0.08205$$

$\Rightarrow$

$$\beta = 4.7067^\circ$$

Velocity is maximum when the contact is on the point where the circular flank meets the circular nose.

$$\begin{aligned}
 V_{\max} &= \omega(r_f - r_c) \sin \beta \\
 &= \left(\frac{2\pi \times 960}{60} \right) \times (297.707 - 20) \times \sin(4.7067^\circ) \\
 &= 2290.832 \text{ mm/s} = 2.2908 \text{ m/s}
 \end{aligned}$$

Maximum acceleration is when $\theta = 0$,

$$\begin{aligned}
 f_{\max} &= \omega^2(r_f - r_c) = \left(\frac{2\pi \times 960}{60} \right)^2 \times (297.707 - 20) \\
 &= 2806638.827 \text{ mm/s}^2 = 2806.6388 \text{ m/s}^2
 \end{aligned}$$

Maximum retardation is when $\alpha - \theta = 0$

$$\begin{aligned}
 f_{\max} &= \omega r^2 = \left(\frac{2\pi \times 960}{60} \right)^2 \times 29 = 293087.77 \text{ mm/s}^2 \\
 &= 293.08 \text{ m/s}^2
 \end{aligned}$$

Spring force is needed to maintain contact during the retardation of the follower.

$$\text{Minimum force, } F = m \times f = 0.250 \times 293.088 = 73.272 \text{ N}$$

137.

Sol. Cams :

- A cam is a rotating machine element which gives reciprocating or oscillating motion to another element known as follower. The cam and the follower have a line contact and constitute a higher pair. The cams are usually rotated at uniform speed by a shaft, but the follower motion is pre-determined and will be according to the shape of the cam.
- The cam and follower mechanism is almost used in all modern machines, for example, for operating the valves in internal combustion engines, textile machine tools, paper cutting machines, feed mechanism of automatic lathe, switches etc.

Key Point:

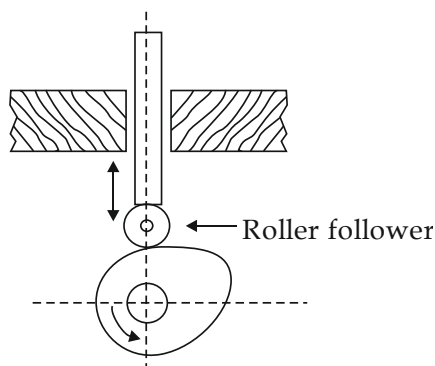
A cam and follower combination belongs to the category of a higher pairs

1. A driver member known as the cam
2. A driven member called follower
3. A frame which support, the cam and guides the follower

Classification of cams :

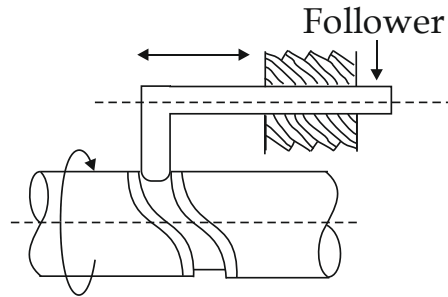
1. **According to shape:**

Radial or disc cams: In radial cams, the follower reciprocates or oscillates in a direction perpendicular to the cam axis.



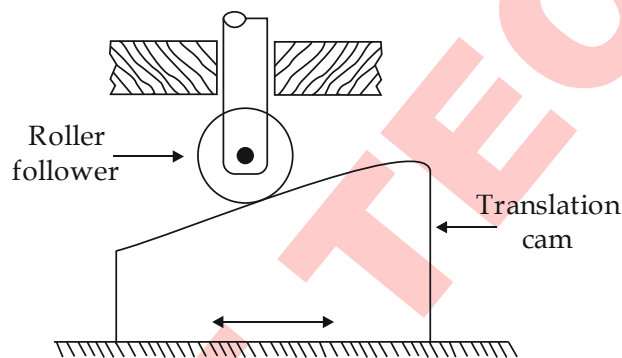
Cylindrical or barrel or drum cams:

A cylindrical cam is a 3D cam which drives its follower in a groove cut on periphery of a cylinder.



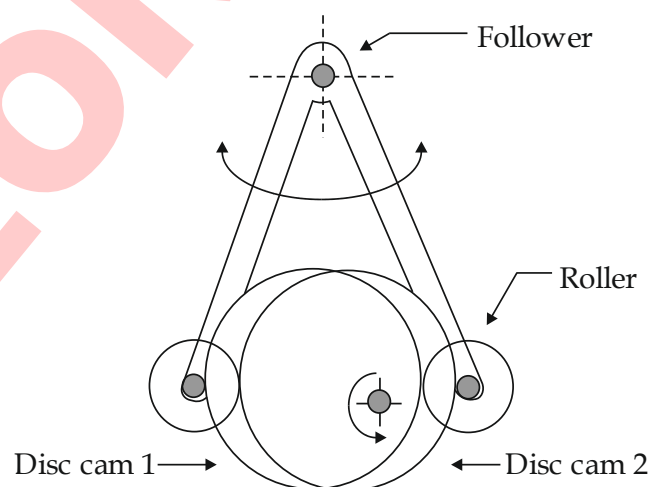
Translation or wedge or flat cams:

The flat plate cam profile moves to the left and the follower moves up and down, matching the shape of the profile. The flat plate cam then reverses in the opposite direction and the follower drops and rises again.



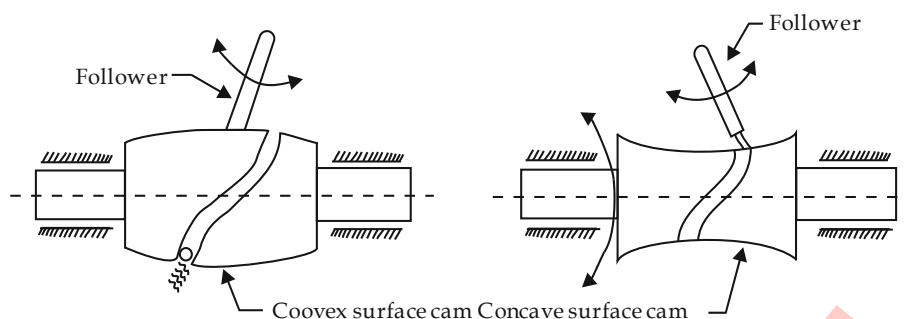
Conjugate Cams:

Conjugate, cams also called dual cams are a double disc cam that are bolted together. Because of the 2 rollers always touching the double disc cam the follower makes less noise and has better control at high speed.



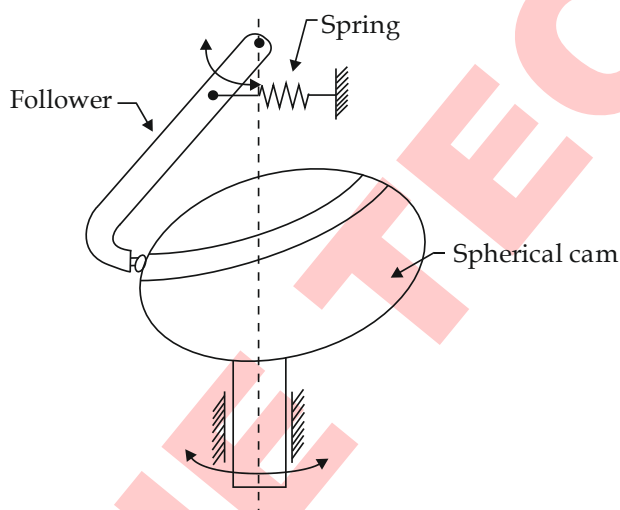
Globeulalaeos:

Input and output in glokpidal cam gears rotate, and the axes of rotation of cam and follower are perpendicular to each other. The special feature of glohoidal cams is that the roller axis is located in the plane of movement of the rotating follower.



Spherical or spiral Cams:

In a spherical cam, the follower oscillates about an axis perpendicular to the axis of rotation of the cam. A spherical cam is in the form of a spherical surface which transmits motion to the follower.

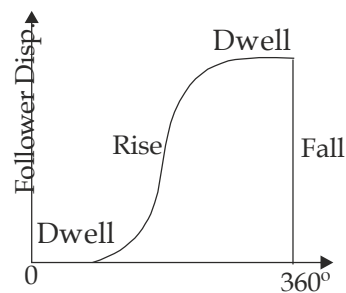
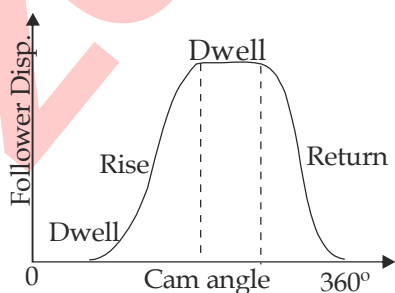


2. According to follower movement:

Motion of follower

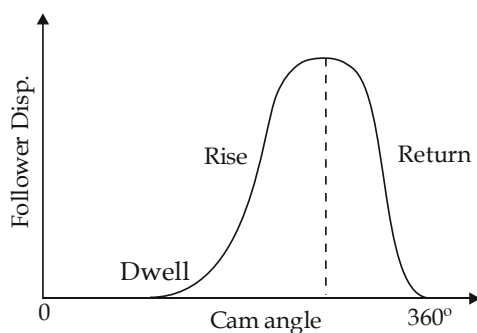
Dwell-Rise-Dwell-Return-Dwell (DRDRD):

- Also known as Dwell-Rise-Dwell (DRD)
- Most widely used type.



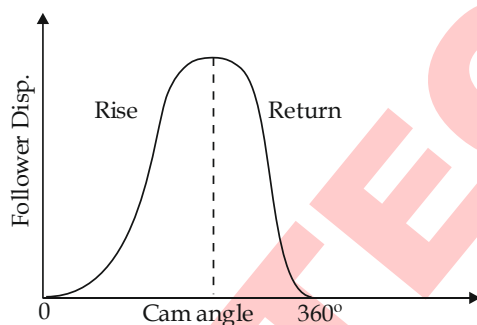
Dwell-Rise-Return-Dwell (DRRD):

- Used more frequently than RRR-type cam.



Rise-Return-Rise (RRR):

- Use is very limited in the industrie.



Followers: A follower is a translating or oscillating mechanical member that follows the motion of the cam. It can touch the surface profile of the cam or can be spring-loaded. It can have uniform velocity or can have uniform acceleration motion. Complicated output motion can be achieved with the help of the follower motion.

- A follower is a rotating or an oscillating element of a machine that follows the motion of cam by direct contact.
- If a cam moves in reciprocating motion the follower moves in vertically respect to the axis of the cam.
- Follower only follows cam and converts the rotary motion of cam into reciprocating or oscillating motion.

Classification of followers:

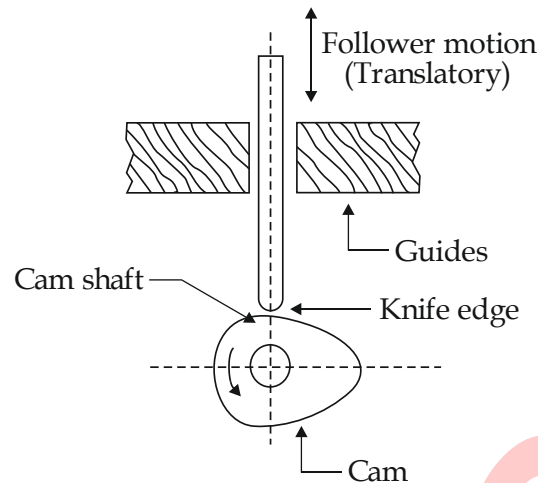
1. According to shape:

Types of followers:

Knife-edge followers :

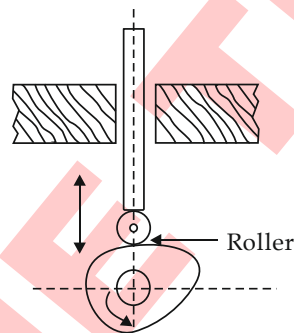
When the contacting end of the follower has a sharp knife edge, it is called a knife edge follower. The sliding motion takes place between the contacting surfaces (i.e. the knife edge and the cam surface).





Roller follower :

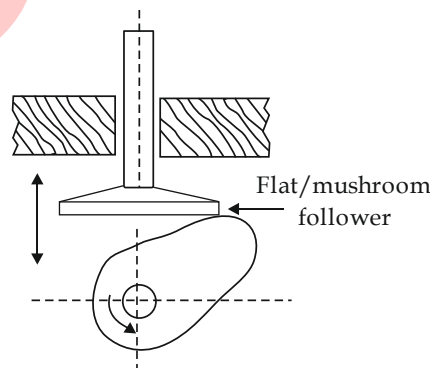
- When the contacting end of the follower is a roller, it is called a roller follower.
- At low speeds, follower has pure rolling, but at high speed, some sliding occurs.



- In case of steep rise, a roller follower jams the cam.

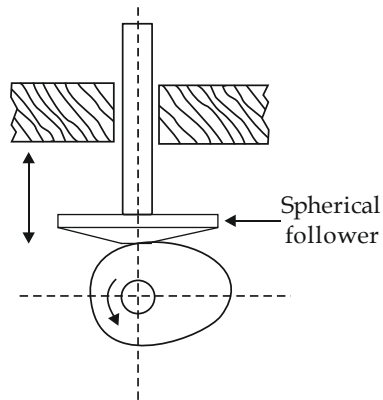
Mushroom follower:

- It does not pose the problem of jamming.
- High surface stresses and wear are quite high.



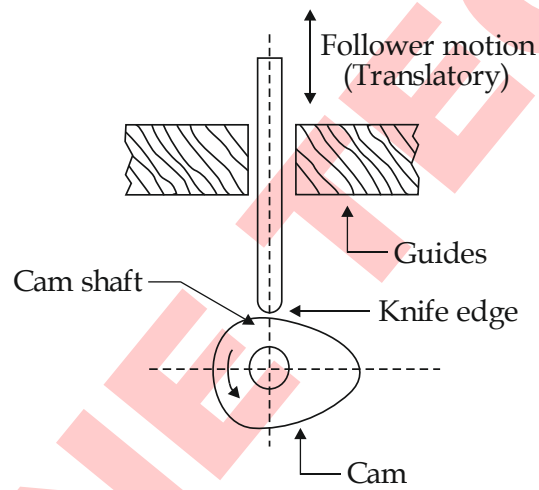
Spherical faced follower :

- When the contacting end of the follower is of spherical shape, it is called a spherical faced follower.
- When a flat-faced follower is used in automobile engines, high surface stresses are produced. In order to minimise these stresses, the flat end of the follower is machined to a spherical shape.

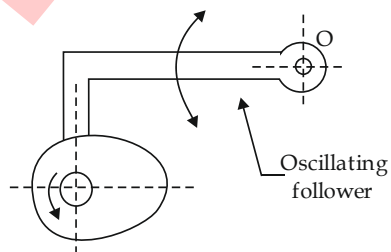


2. **According to movement:**

Translatory motion or reciprocating follower :



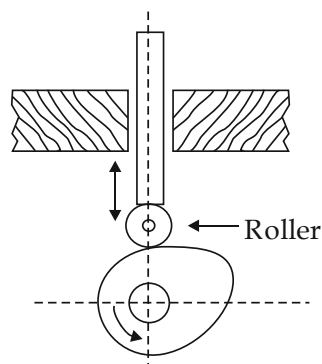
Oscillating follower :



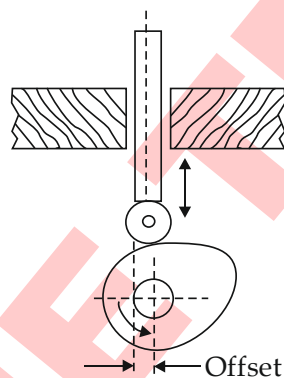
3. According to location of line of movement :

Line of motion

Radial follower :



Offset follower :

**Merits of using cam and follower :**

- By proper designing of cam profile, a variety of follower motions can be obtained
- Able to operate accurately
- Easy mechanism to convert the rotational motion to reciprocating or oscillating motion
- The number of components reduces with the use of a cam and follower mechanism

Demerits of using cam and follower :

- Subjected to wear, hence, can be used only when the transmission force is less
- Accurate manufacturing is required
- High manufacturing cost
- There is a limitation to the size of the mechanism. Large size is challenging to operate

Note:-

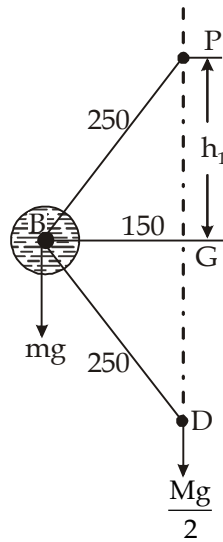
According to manner of constraint of follower :

- Pre-loaded spring cam, in this spring is used for keeping the contact.
- Positive-drive cam, in this follower cannot go out of his groove.
- Gravity cam, these cams are not preferred due to their uncertain behavior.

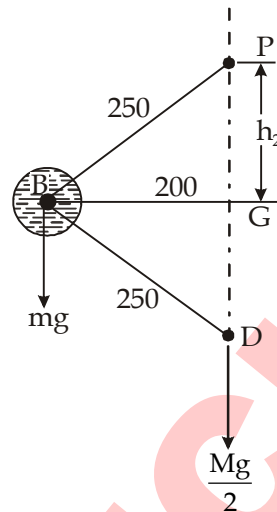
138.

- Sol** (a) Given: $BP = BD = 250 \text{ mm} = 0.25 \text{ m}$
 $m = 5 \text{ kg}$
 $M = 15 \text{ kg}$
 $r_1 = 150 \text{ mm} = 0.15 \text{ m}$
 $r_2 = 200 \text{ mm} = 0.2 \text{ m}$





(a) Minimum Position



(b) Maximum Position

The minimum and maximum positions of the governor are shown in figure (a) and (b) respectively.

Minimum speed when $r_1 = BG = 0.15$ m

Let N_1 = Minimum speed

From figure (a), we find the height of the governor,

$$h_1 = PG = \sqrt{(PB)^2 - (BG)^2} = \sqrt{(0.25)^2 - (0.15)^2} = 0.2 \text{ m}$$

We know that

$$(N_1)^2 = \frac{m+M}{m} \times \frac{895}{h_1} = \frac{5+15}{5} \times \frac{895}{0.2} = 17.900$$

$$\therefore N_1 = 133.8 \text{ r.p.m.}$$

Maximum speed when $r_2 = BG = 0.2$ m

Let N_2 = Maximum speed

From figure (b), we find that height of the governor.

$$h_2 = PG = \sqrt{(PB)^2 - (BG)^2} = \sqrt{(0.25)^2 - (0.2)^2} = 0.15 \text{ m}$$

We know that

$$(N_2)^2 = \frac{m+M}{m} \times \frac{895}{h_2} = \frac{5+15}{5} \times \frac{895}{0.15} = 23.867$$

$$\therefore N_2 = 154.5 \text{ r.p.m.}$$

Range of speed

$$\begin{aligned} \text{We know that range of speed} &= N_2 - N_1 \\ &= 154.4 - 133.8 = 20.7 \text{ r.p.m.} \end{aligned}$$

Sol. (b) I = Mass moment of inertia of the flywheel about its axis of rotation in $\text{kg-m}^2 = \text{m} \cdot \text{k}^2$

N_1 and N_2 = Maximum and minimum speeds during the cycle in r.p.m.

ω_1 and ω_2 = Maximum and minimum angular speeds during the cycle in rad/s.

$$N = \text{Mean speed during the cycle in r.p.m.} = \frac{N_1 + N_2}{2}$$

$$\omega = \text{Mean angular speed during the cycle in rad/s} = \frac{\omega_1 + \omega_2}{2}$$

$$C_s = \text{Coefficient of fluctuation of speed,} = \frac{N_1 + N_2}{N} \quad \text{or} \quad \frac{\omega_1 + \omega_2}{\omega}$$

We know that the mean kinetic energy of the flywheel.

$$E = \frac{1}{2} \times I \cdot \omega^2 = \frac{1}{2} \times m \cdot k^2 \cdot \omega^2 \quad (\text{in N-m or joules})$$

As the speed of the flywheel changes from ω_1 to ω_2 , the maximum fluctuation of energy.

$\Delta E = \text{Maximum K.E.} - \text{Minimum K.E.}$

$$\begin{aligned} &= \frac{1}{2} \times I(\omega_1)^2 - \frac{1}{2} \times I(\omega_2)^2 = \frac{1}{2} \times I[(\omega_1)^2 - (\omega_2)^2] \\ &= \frac{1}{2} \times I(\omega_1 + \omega_2)(\omega_1 - \omega_2) = I \cdot \omega(\omega_1 - \omega_2) \quad \dots(i) \end{aligned}$$

$$\dots \left(\because \omega = \frac{\omega_1 + \omega_2}{2} \right)$$

$$= I\omega^2 \left(\frac{\omega_1 - \omega_2}{\omega} \right) \quad \dots(\text{Multiplying and dividing by } \omega)$$

$$= I \cdot \omega^2 \cdot C_s = m \cdot k^2 \cdot \omega^2 \cdot C_s \quad \dots(\because I = k \cdot m^2) \quad \dots(ii)$$

$$= 2 \cdot E \cdot C_s \quad (\text{in N-m or joules}) \quad \dots \left(\because E = \frac{1}{2} \times I\omega^2 \right) \quad \dots(iii)$$

139.

Sol. Watt Governor

The simplest form of a centrifugal governor is a Watt governor, as shown in figure. It is basically a conical pendulum with links attached to a sleeve of negligible mass. The arms of the governor may be connected to the spindle in the following three ways :

- The pivot P, may be on the spindle axis as shown in figure (a).
- The pivot P, may be offset from the spindle axis and the arms when produced intersect at O, as shown in figure (b).
- The pivot P, may be offset, but the arms cross the axis at O, as shown in figure (c).

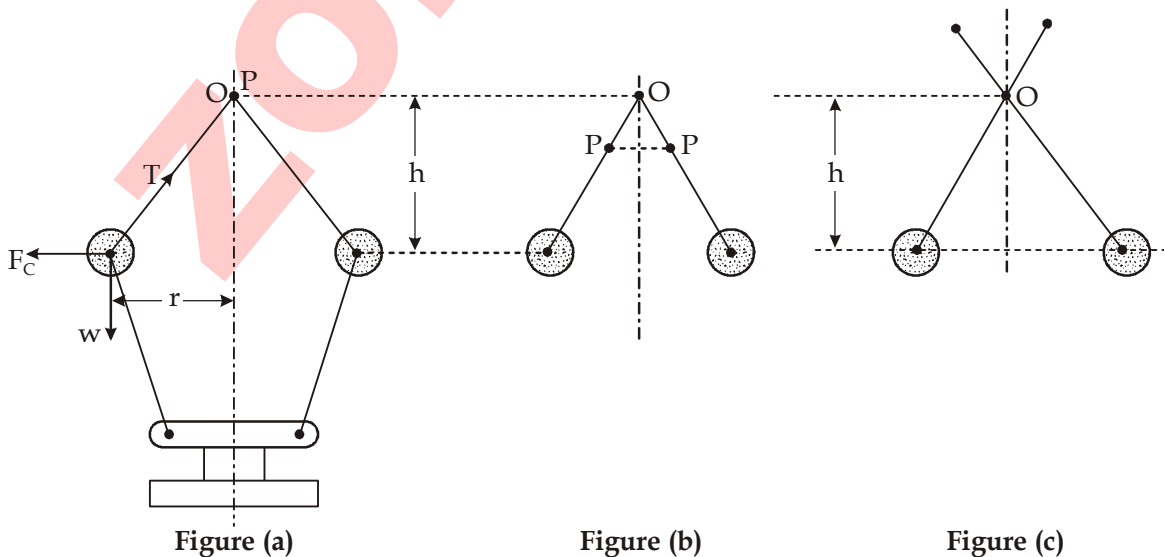


Figure : Watt Governor

Let

m = Mass of the ball in kg

w = Weight of the ball in newtons = $m \cdot g$

T = Tension in the arm in newtons

ω = Angular velocity of the arm and ball about the spindle axis in rad/s

r = Radius of the path of rotation of the ball i.e. horizontal distance from the centre of the ball to the spindle axis in metres

F_c = Centrifugal force acting on the ball in newtons = $m \cdot \omega^2 \cdot r$.

h = Height of the governor in metres

and

It is assumed that the weight of the arms, links and the sleeve are negligible as compared to the weight of the balls.

Now, the balls are in equilibrium under the action of

(i) The centrifugal force (F_c) acting on the ball

(ii) The tension (T) in the arm

(iii) The weight (w) of the ball

Taking moments about point O, we have

$$F_c \times h = w \times r = m \cdot g \cdot r$$

or

$$m \cdot \omega^2 \cdot r \cdot h = m \cdot g \cdot r$$

or

$$h = \frac{g}{\omega^2} \quad \dots(i)$$

When g is expressed in m/s^2 and ω in rads, then h is in metres. If N is the speed in r.p.m, then

$$\omega = \frac{2\pi N}{60}$$

$$\therefore h = \frac{9.81}{(2\pi N / 60)^2} = \frac{895}{N^2} \text{ metres} \quad (\because g = 9.81 \text{ m/s}^2) \quad \dots(iii)$$

Limitations :

- We see from the above expression that the height of a governor h , is inversely proportional to N^2 .
- Therefore at high speeds, the value of h is small.
- At such speeds, the change in the value of h corresponding to a small change in speed is insufficient to enable a governor of this type to operate the mechanism to give the necessary change in the fuel supply.
- This governor may only work satisfactorily at relatively low speeds i.e. from 60 to 80 r.p.m.

140.

Sol. (i) **Coefficient of Fluctuation of Speed (C_s) :**

The difference between the maximum and minimum speeds during a cycle is called the maximum fluctuation of speed. The ratio of the maximum fluctuation of speed to the mean speed is called the coefficient of fluctuation of speed.

Let N_1 and N_2 = Maximum and minimum speed in r.p.m. during the cycle

and N = Mean speed in r.p.m. = $\frac{N_1 + N_2}{2}$

$\therefore$ Coefficient of fluctuation of speed,

$$C_s = \frac{N_1 - N_2}{N} = \frac{2(N_1 - N_2)}{N_1 + N_2}$$

$$= \frac{\omega_1 - \omega_2}{\omega} = \frac{2(\omega_1 - \omega_2)}{\omega_1 + \omega_2} \quad \dots(\text{In terms of angular speeds})$$

$$= \frac{v_1 - v_2}{v} = \frac{2(v_1 - v_2)}{v_1 + v_2} \quad \dots (\text{In terms of linear speeds})$$

(ii) Coefficient of steadiness (m) :

The coefficient of fluctuation of speed is a limiting factor in the design of flywheel. It varies depending upon the nature of service to which the flywheel is employed.

Note: The reciprocal of the coefficient of fluctuation of speed is known as coefficient of steadiness and is denoted by m.

$$m = \frac{1}{C_s} = \frac{N_1 + N_2}{2(N_1 - N_2)}$$

(iii) Maximum fluctuation of energy (ΔE) :

The difference between the maximum and the minimum energies is known as maximum fluctuation of energy.

$$\Delta E = \text{Maximum K.E.} - \text{Minimum K.E.}$$

$$= \frac{1}{2} \times I(\omega_1)^2 - \frac{1}{2} \times I(\omega_2)^2 = \frac{1}{2} \times I[(\omega_1)^2 - (\omega_2)^2]$$

$$= \frac{1}{2} \times I(\omega_1 + \omega_2)(\omega_1 - \omega_2) = I \cdot \omega (\omega_1 - \omega_2) \quad \dots \left(\because \omega = \frac{\omega_1 + \omega_2}{2} \right) \quad \dots (i)$$

$$= I \cdot \omega \left(\frac{\omega_1 - \omega_2}{\omega} \right) \quad \dots (\text{Multiplying and dividing by } \omega)$$

$$= I \cdot \omega^2 \cdot C_s = m \cdot k^2 \cdot \omega^2 \cdot C_s \quad (\because I = m \cdot k^2) \quad \dots (ii)$$

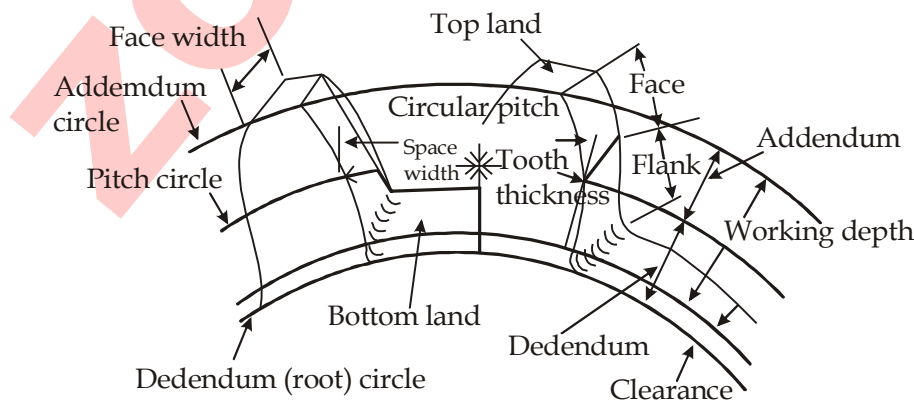
$$= 2 \cdot E \cdot C_s \quad (\text{in N-m or Joules}) \quad \dots \left(\because E = \frac{1}{2} \times I \cdot \omega^2 \right) \quad \dots (iii)$$

(iv) Coefficient of Fluctuation of energy

It may be defined as the ratio of the maximum fluctuation of energy to the work done per cycle. Mathematically, coefficient of fluctuation of energy,

$$C_E = \frac{\text{Maximum fluctuation of energy}}{\text{Work done per cycle}}$$

141.
Sol.



- 1. Pitch circle:** It is an imaginary circle which by pure rolling action, would give to same motion as the actual gear.
- 2. Pitch circle diameter:** It is diameter of the pitch circle. The size of the gear is usually specified by the pitch circle diameter. It is also known as pitch diameter.

3. **Pitch Point:** It is the common point of contact between two pitch surface.
4. **Pressure angle or angle of obliquity:** It is angle between the common normal to two gear teeth at the point of contact and common tangent at the pitch point.
5. **Addendum:** It is the radial distance of a tooth from the pitch circle to the top the tooth.
6. **Dedendum:** It is the radial distance of a tooth from the pitch circle to the bottom of the tooth.
7. **Addendum circle:** It is the circle drawn through the top of the teeth and is concentric with the pitch circle.
8. **Dedendum circle:** It is circle drawn through the bottom of the teeth. It is also called root circle.
9. **Circular Pitch:** It is distance measured on the circumference of the pitch circle from a point to one tooth to the corresponding point on the next tooth.

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

D = diameter of the pitch circle

T = Number of teeth on the wheel

10. **Diametral Pitch:** it is the ratio of Number of teeth of the pitch circle diameter in millimeters

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

11. **Module:** It is the ratio of pitch circle diameter in mm to the number of teeth, it is usually denoted by m.

$$\text{Module}(m) = D/T$$

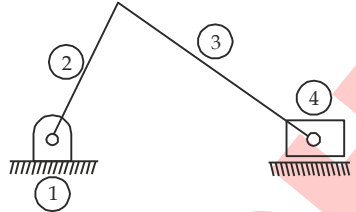
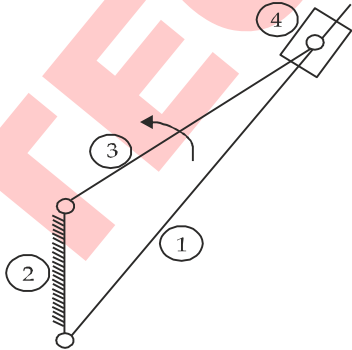
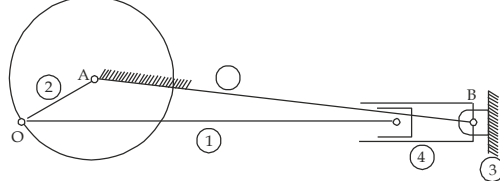
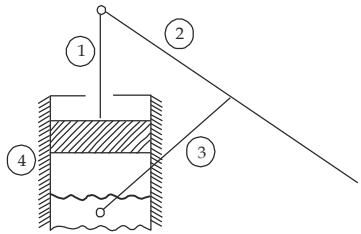
12. **Clearance:** It is the radial distance from the top of the tooth to the bottom of the tooth, in a meshing gear. A circle passing through the top the meshing gear is known as clearance circle.
 13. **Total depth:** It is the radial distance between the addendum and the dedendum circles of a gear. It is equal to the sum of the addendum and dedendum.
 14. **Face of tooth:** It is the surface of gear tooth above the pitch surface
 15. **Flank of tooth:** It is the surface of gear tooth below the pitch surface.
 16. **Working depth :** It is the radial distance from the addendum circle to the clearance circle. It is equal to the sum of the addendum of the two meshing gears.
 17. **Tooth thickness :** It is the width of the tooth measured along the pitch circle.
 18. **Tooth space :** It is the width of space between the two adjacent teeth measured along the pitch circle.
 19. **Backlash :** It is the difference between the tooth space and the tooth thickness, as measured along the pitch circle. Theoretically, the backlash should be zero, but in actual practice some backlash must be allowed to prevent jamming of the teeth due to tooth errors and thermal expansion.
 20. **Top land :** It is the surface of the top of the tooth.
 21. **Face width :** It is the width of the gear tooth measured parallel to its axis.
 22. **Profile :** It is the curve formed by the face and flank of the tooth.
 23. **Path of contact :** It is the path traced by the point of contact of two teeth from the beginning to the end engagement.
 24. **Arc of contact :** It is the path traced by a point on the pitch circle from the beginning to the end of engagement of a given pair of teeth. The arc of contact consists of two parts, i.e.,
Arc of approach
 It is the portion of the path of contact from the beginning of the engagement to the pitch point.
Arc of recess
 It is the portion of the path of contact from the pitch point to the end of the engagement of a pair of teeth.
 25. **Fillet radius :** It is the radius that connects the root circle to the profile of the tooth.
- Note :** The ratio of the length of arc of contact to the circular pitch is known as contact ratio i.e., number of pairs of teeth in contact.

142.

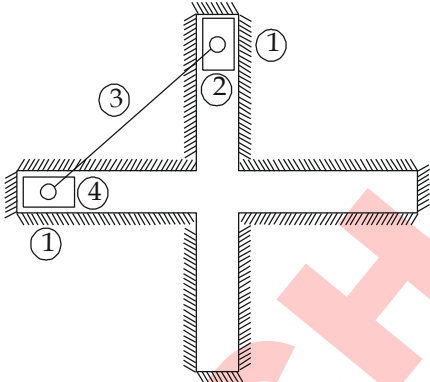
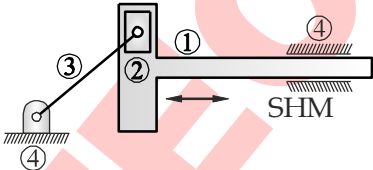
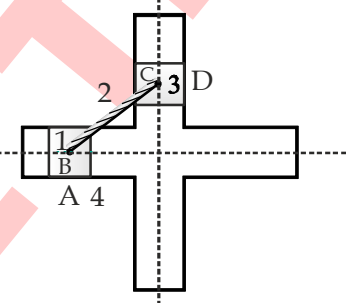
- Sol. Inversion of mechanism:** When one of links is fixed in a kinematic chain, it is called a mechanism. So we can obtain as many mechanisms as the number of links in a kinematic chain by fixing, in turn,

different links in a kinematic chain. This method of obtaining different mechanisms by fixing different links in a kinematic chain, is known as inversion of the mechanism. It may be noted that the relative motions between the various links is not changed in any manner through the process of inversion, but their absolute motions (those measured with respect to the fixed link) may be changed drastically.

(i) Properties of single slider-crank chain mechanism :

First inversion : • When link 1 is Fixed (cylinder Fixed). • Reciprocating engine. • Reciprocating compression.	 • Reciprocating engine. • Reciprocating compression.
Second inversion : • When link 2 is Fixed (crank Fixed). • Whitworth Quick-return mechanism. • Rotary engine.	 • Whitworth Quick-return mechanism. • Rotary engine.
Third inversion : • When link 3 is Fixed (connecting rod Fixed). • Oscillating cylinder engine. • Crank and slotted lever mechanism.	 • Oscillating cylinder engine. • Crank and slotted lever mechanism.
Fourth inversion : • When link 4 is Fixed (slider Fixed). • Hand pump	 • Hand pump

(ii) Properties of double slider-crank chain mechanism :

First inversion : • Elliptical trammel. • When link 1 is Fixed.	
Second inversion : • When any slider (link 2 or link 4) is Fixed. • Scotch-yoke mechanism.	
Third inversion : • Oldham's coupling. • It is obtain by Fixing the connecting link (link 2).	

Importance of inversion of mechanism :

- The inversion of a mechanism does not change the motions of its links relative to each other but does change their absolute motions.
- The inversion mechanism is used in shaper machine to reduce the backward stroke time.

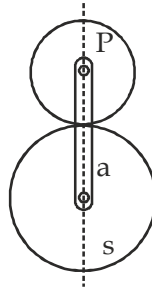
Note:-

The part of a mechanism which initially moves with respect to the frame or fixed link is called driver and that part of the mechanism to which motion is transmitted is called follower. Most of the mechanisms are reversible, so that same link can play the role of a driver and follower at different times. For example, in a reciprocating steam engine, the piston is the driver and flywheel is a follower while in a reciprocating air compressor, the flywheel is a driver.

143.

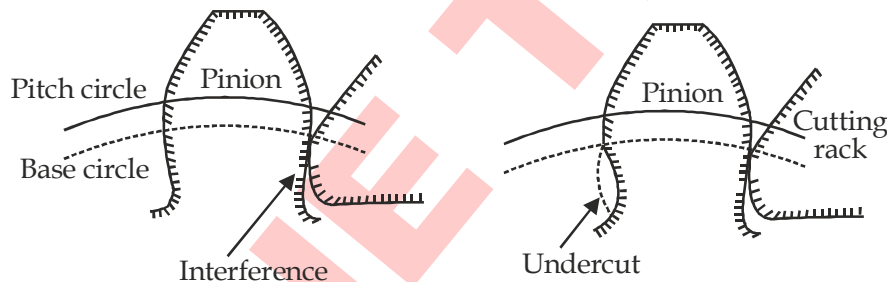
Sol. Epicyclic gear trains: A gear having a relative motion of axes is called a planetary or an epicyclic gear train (or simply epicyclic gear train). In a epicyclic train, the axis of at least one of the gears also moves relative to the frame.

Consider two gear wheels S and P, the axes of which are connected by an arm 'a'. If the arm a is fixed, the wheels S and P constitute a simple train. However, if the wheel S is fixed so that the arm can rotate about the axis of S, the wheel P would also move around S. Therefore, it is an epicyclic train.



Usually, the wheel P is known as the epicyclic wheel. The term epicyclic emerges from the fact that the wheel P rolls outside another wheel S and traces an epicyclic path. It is also possible that the fixed wheel is annular and another wheel rolls inside it. In that case, the path traced will be a hypocycloid. Large speed reductions are possible with epicyclic gears and if the fixed wheel is annular, a more compact unit could be obtained. Important applications of epicyclic gears are in transmission, coupling devices, and so on.

Interference in involute gears: Mating of two non-conjugate (non-involute) teeth is known as interference because the two teeth do not slide properly and thus, rough action and bending occurs. Owing to non-involute profile, the contacting teeth have different velocities which can lock the two gears. Figure shows a pinion. A portion of its dedendum falls inside the base circle. If the addendum of the mating gear is more than the limiting value, it interferes with the dedendum of the pinion and the two gears are locked.

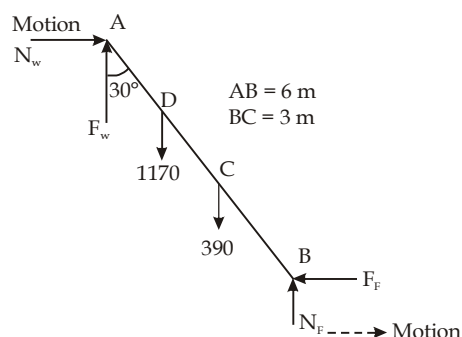


However, if a cutting rack having similar teeth is used to cut the teeth in the pinion, it will remove that portion of the pinion tooth which would have interfered with the gear as shown in figure. A gear having its material removed in this manner is said to be undercut and the process, undercutting. In pinion with small number of teeth, this can seriously weaken the tooth. However, when the actual gear meshes with the undercut pinion, no interference occurs.

Undercutting will not take place if the teeth are designed to avoid interference.

144.

Sol. When the ladder slips it moves, to the right at 'B'. Hence the direction of frictional force at 'B' acts from right to left. Similarly, due to slip end 'A' moves down. Hence the frictional force acts in upward direction at A.



Coefficient of friction between the wall and ladder is

$$\mu_w = \frac{F_w}{N_w} = 0.25$$

Where

F_w = Frictional force

N_w = Normal reaction

Coefficient of friction between the ladder and floor is

$$\mu_f = \frac{F_f}{N_f} = 0.38$$

Where

F_f = Frictional force

N_f = Normal reaction at floor

Equation of equilibrium of ladder.

$$\Sigma F_x = 0 :$$

$$N_w - F_f = 0; F_f = N_w$$

$$N_f = \frac{F_f}{0.38} = \frac{N_w}{0.38} = 2.6316N_w$$

$$\Sigma F_y = 0 :$$

$$F_w + N_f - 1170 - 390 = 0$$

$$F_w + N_f = 1560.$$

Since

$$F_w = 0.25N_w$$

$\therefore$

$$0.25N_w + N_f = 1560$$

$$0.25N_w + 2.6316N_w = 1560$$

$\therefore$

$$N_w = 541.366N; N_f = 1424.659N$$

Hence

$$F_w = 135.342N$$

$$F_f = 541.370N$$

$$\Sigma M_B = 0 : -N_w(6 \cos 30^\circ) - F_w(6 \sin 30^\circ) + 1170(x \sin 30^\circ) + 390(3 \sin 30^\circ) = 0$$

$$\therefore (1170 \sin 30^\circ)x = (541.366)6 \cos 30^\circ + (135.342)6 \sin 30^\circ - (390)(3 \sin 30^\circ)$$

$$x = \frac{1}{585} \{2813.020 + 406.026 - 585\}$$

$$x = 4.503m$$

$$\cos 30^\circ = \frac{h}{x}$$

$$h = x \cos 30^\circ$$

$$h = 3.89m$$

145.

Sol. Design Considerations of disc clutch :

1. The contact surface should develop a frictional force that may pick-up and hold the load with reasonable low pressure between the contact surface.
2. The heat of friction should be rapidly dissipated and tendency to grab should be minimum.
3. the surface should be backed by a material stiff enough to ensure a reasonably uniform distribution of pressure.

Torque capacity : Consider there is no slipping takes place-

Area of contact of friction surface = $2\pi dr$

Normal or axial force on the ring $\delta w = P \times 2\pi r dr$

Friction force $F_r = \mu \delta w = \mu P 2\pi r dr$

Friction torque $T_r = F_r \times r = \mu P 2\pi r^2 dr$



We shall now consider two cases

1. Uniform pressure
2. Uniform wear
1. **Uniform pressure**

$$\text{Intensity of pressure } P = \frac{W}{\pi(r_1^2 - r_2^2)}$$

$$\begin{cases} P = \text{Pressure} \\ r = \text{Radius} \\ \mu = \text{Coefficient of friction} \\ W = \text{Load} \end{cases}$$

Frictional torque on elementary spring

$$T_r = 2\pi\mu Pr^2 dr$$

Total friction torque

$$T = \int_1^2 2\pi\mu Pr^2 dr = 2\pi\mu P \left[\frac{r_1^3 - r_2^3}{3} \right]$$

$$T = 2\pi\mu \frac{W}{(r_1^2 - r_2^2)\pi} \times \left(\frac{r_1^3 - r_2^3}{3} \right)$$

$$T = \frac{2}{3} \mu W \left[\frac{r_1^3 - r_2^3}{r_1^2 - r_2^2} \right] = \mu WR$$

R = Mean radius of friction surface

$$= \frac{2}{3} \left[\frac{r_1^3 - r_2^3}{r_1^2 - r_2^2} \right]$$

2. **Uniform wear** : If P is the intensity of pressure and r is the distance from the axis of clutch, the P varies inversely with the distance.

$$P \propto \frac{1}{r}$$

$$P = \frac{c}{r} \Rightarrow Pr = c \quad \dots(i)$$

Normal force on ring $\delta W = P \cdot 2\pi r dr = \frac{c}{r} 2\pi r dr$

$$= 2\pi c dr$$

Total force on friction surface

$$W = \int_1^2 2\pi c dr = 2\pi c(r_1 - r_2)$$

$$c = \frac{W}{2\pi(r_1 - r_2)} \quad \dots(ii)$$

Frictional torque acting on ring

$$T_r = 2\pi\mu Pr^2 dr = 2\pi\mu \frac{c}{r} r^2 dr$$

$$= 2\pi\mu c r dr$$

Total frictional torque

$$T = \int_1^2 2\pi\mu c r dr$$



$$= 2\pi\mu c \left[\frac{r_1^2 - r_2^2}{2} \right] = \pi\mu c (r_1^2 - r_2^2)$$

$$T = \pi\mu \frac{W}{2\pi(r_1 - r_2)} (r_1^2 - r_2^2)$$

$$T = \frac{1}{2} \mu W (r_1 + r_2) = \mu WR$$

R = Mean radius of friction surface

$$= \frac{r_1 + r_2}{2}$$

146.

Sol.

Where,

P_1 = Tension on the tight side of the band (N)

P_2 = Tension on the loose side of the band (N)

μ = Coefficient of friction between the friction lining and the brake drum

ϕ = Angle of wrap (rad)

The torque M_t absorbed by the brake is given by,

$$M_t = (P_1 - P_2)R$$

where,

M_t = Torque capacity of the brake (N-mm)

R = Radius of the brake drum (mm)

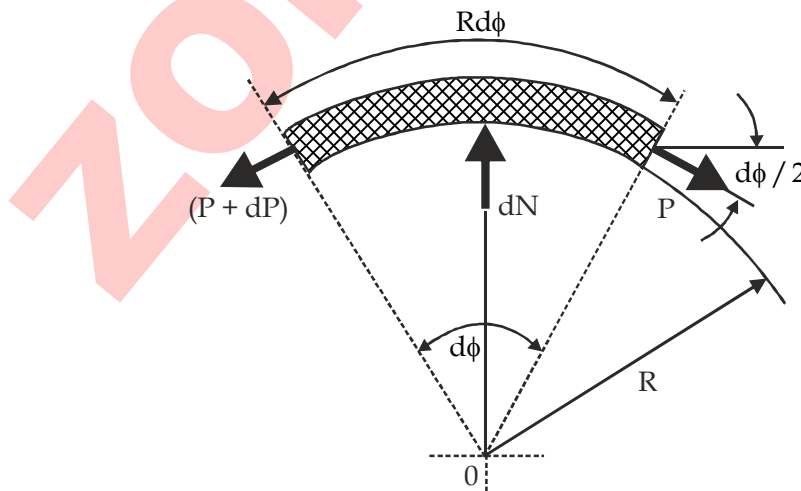
Considering the forces acting on the lever and taking moments about the pivot,

$$P_2 \times m = F_a \times a$$

$\Rightarrow$

$$F_a = \frac{P_2 \times m}{a}$$

An element of the band subtending an angle ($d\phi$) is shown in figure. The elemental area of the friction lining is ($Rd\phi b$), where b is the width of the lining parallel to the axis of the brake drum. In the figure (P) and ($P + dP$) are tensions in the band in the loose and tight sides respectively. If p is the intensity of pressure, the normal reaction (dN) is given by



Considering equilibrium of vertical forces on the element,

$$dN = P \sin\left(\frac{d\phi}{2}\right) + (P + dP) \sin\left(\frac{d\phi}{2}\right)$$

For small angles, $\sin\left(\frac{d\phi}{2}\right) = \left(\frac{d\phi}{2}\right)$

Neglecting higher order differentials,

$$dN = Pd\phi$$

From equation (i), $P = \frac{P}{Rb}$

The intensity of pressure is maximum, when the band tension P is equal to P_1 .

or, $P_{\max} = \frac{P_1}{Rb}$

$$\text{Torque}(T_f) = (P_1 - P_2)R = P_2 R \left(\frac{P_1}{P_2} - 1 \right) = P_2 R (e^{\mu\alpha} - 1)$$

$$\begin{aligned} \text{Torque}(T_f) &= (P_1 - P_2)R = P_1 R (1 - e^{-\mu\alpha}) = (P_{\max} Rb) R (1 - e^{-\mu\alpha}) \\ &= P_{\max} R^2 b (1 - e^{-\mu\alpha}) \end{aligned}$$

147.

Sol. **Angle of friction :** Consider a body of weight W resting on an inclined plane as shown in figure below. We know that the body is in equilibrium under the action of the following forces :

1. Weight (W) of the body, acting vertically downwards
2. Friction force (F) acting upwards along the plane
3. Normal reaction (R) acting at right angles to the plane

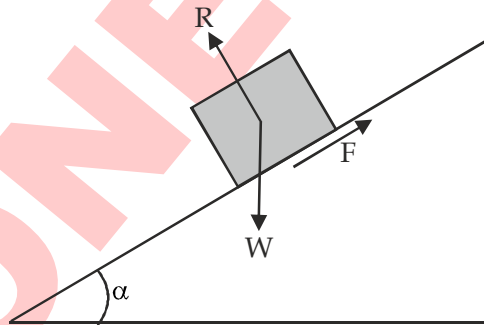


Fig.:- Angle of friction

Let the angle of inclination (α) be gradually increased, till the body just starts sliding down the plane. This angle of inclined plane, at which a body just begins to slide down the plane, is called the **angle of friction**. This is also equal to the angle, which the normal reaction makes with the vertical.

(ii) Co-efficient of friction : It is the ratio of limiting friction to the normal reaction, between the two bodies, and is generally denoted by μ .

Mathematically, coefficient of friction:

$$\mu = \frac{F}{R} = \tan \phi \quad \text{or} \quad F = \mu R$$

Where, ϕ = Angle of friction, F = Limiting friction, and R = Normal reaction between the two bodies.

(iii) Angle of repose/angle of sliding : The angle of repose or angle of sliding is defined as the minimum angle of inclination of a plane with the horizontal such that a body placed on the plane just begins to slide down. Its value depends on the material and nature of the surfaces in contact.



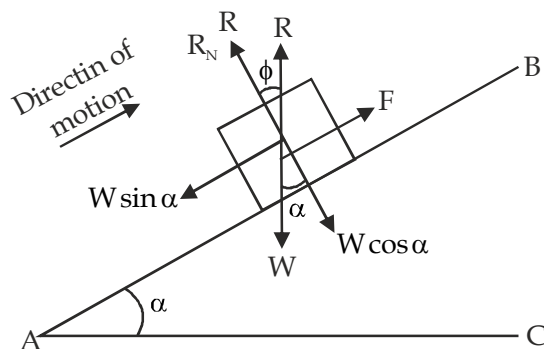


Fig.:- Angle of repose

From the above figure we can say that, AB is an inclined plane such that a body placed on it just begins to slide down. $\angle BAC = \alpha = \text{Angle of repose}$.

The various forces involved are :

1. Weight (mg), of the body acting vertically downwards
2. Normal reaction (R_N), acting perpendicular to AB
3. Force of friction (F), acting upon the plane AB

Now, mg can resolve into two rectangular components : $mg \cos \alpha$ opposite to R and $mg \sin \alpha$ opposite to F .

In equilibrium :

$$F = mg \sin \alpha \quad \dots (i)$$

$$N = mg \cos \alpha \quad \dots (ii)$$

Dividing equations (i) by (ii), we get :

$$\frac{f}{N} = \frac{mg \sin \alpha}{mg \cos \alpha} = \mu$$

i.e., $\mu = \tan \alpha$

Hence, the coefficient of limiting friction between any two surfaces in contact is equal to the tangent of the angle of repose between them.

(vi) Cone of friction :

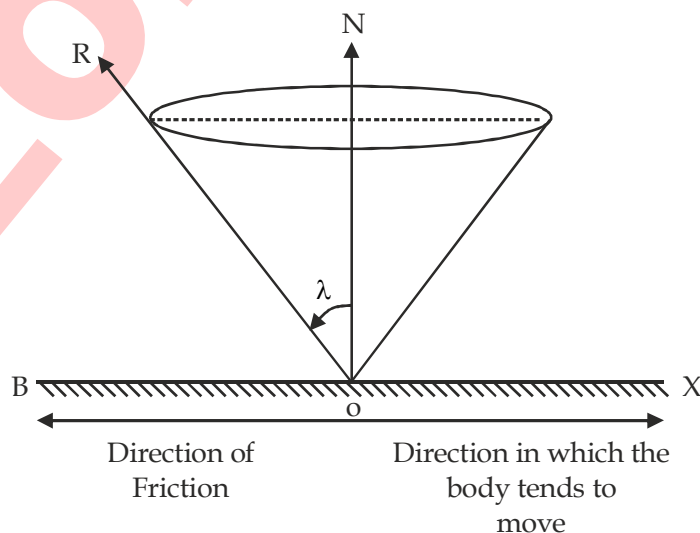


Fig.:- Cone of friction



Let ON represent the normal reaction offered by a surface on a body. If OX is the direction in which the body tends to move then the force of friction acts in the opposite direction i.e, along OE. If the body be in limiting equilibrium the resultant R makes an angle γ with the normal ON.

Suppose the body is at the point of sliding in other direction, it is easily seen that the resultant reaction will make the same angle λ with the normal. Hence, when limiting friction is offered the line of action of the resultant reaction should always lie on the surface of an inverted right circular cone whose semi-vertex is λ . This cone is called the **cone of friction**.

148.

Sol. We know that acceleration of the reciprocating mass of a slider crank mechanism is given by,

$$a = r\omega^2 \left(\cos \theta + \frac{\cos 2\theta}{n} \right)$$

Now, the accelerating force for mass m is,

$$F = mr\omega^2 \left(\cos \theta + \frac{\cos 2\theta}{n} \right) = mr\omega^2 \cos \theta + mr\omega^2 \frac{\cos 2\theta}{n}$$

$mr\omega^2 \cos \theta$ is known as primary unbalance force and $mr\omega^2 \frac{\cos 2\theta}{n}$ is known as secondary unbalance force.

Maximum value of primary unbalanced force = $mr\omega^2$

Maximum value of secondary unbalanced force = $\frac{mr\omega^2}{n}$

We know that,

$$n = \frac{l}{r}$$

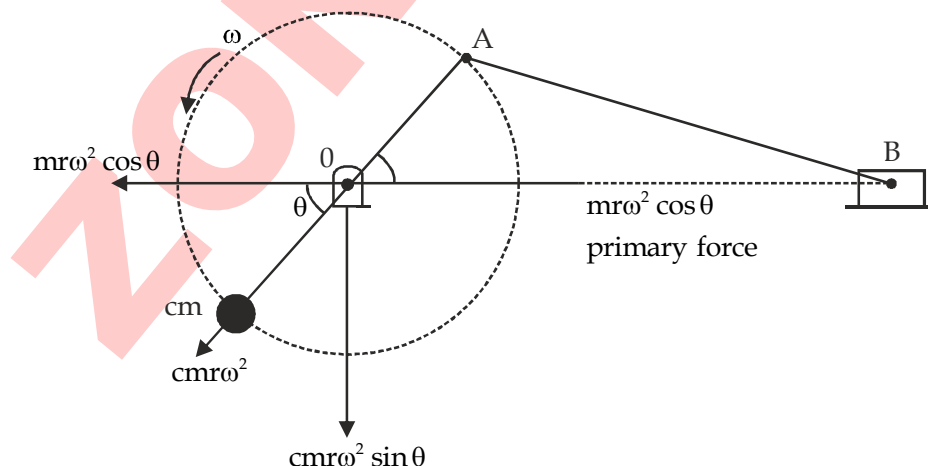
where,

r = Crank radius

l = Length of connecting rod

So, $n > 1$

We can neglect secondary unbalanced force for $n \gg 1$ and at very low speed.



It is not possible to obtain completely balance of unbalancing forces. It can be done only by partially balance of reciprocating mass, like 'cm' is the balanced reciprocating mass placed opposite to crank. where 'c' is the fraction of the reciprocating mass.

Primary force balanced by the mass = $cmr\omega^2 \cos \theta$

Unbalance for along the line of stroke = $(1 - c) mr\omega^2 \cos \theta$



Unbalance force perpendicular to the line of stroke $= cmr\omega^2 \sin \theta$

$$\text{Resultant unbalanced force} = \sqrt{[(1-c)mr\omega^2 \cos \theta]^2 + [cmr\omega^2 \sin \theta]^2}$$

The resultant unbalanced force is minimum when, $c = \frac{1}{2}$

149.

Sol. Given,

$$m = 250 \text{ kg};$$

$$r = 25 \text{ cm} = 0.25 \text{ m},$$

$$N = 12000 \text{ rpm},$$

$$V = 150 \text{ km/hr} = 416.67 \text{ m/s},$$

$$R = 2 \text{ km} = 2000 \text{ m}$$

$$I = mr^2 = 250 \times (0.25)^2 = 15.625 \text{ kg-m}^2$$

$\Rightarrow$

$$\omega = \frac{2\pi N}{60} = \frac{2\pi \times 12000}{60} = 400\pi \text{ rad/s}$$

Angle speed of precession,

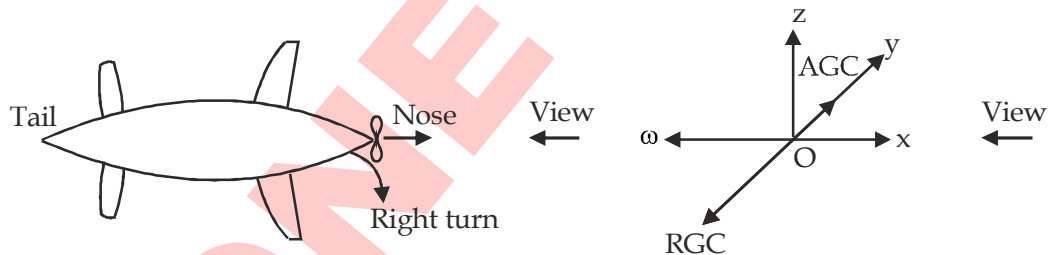
$$(\omega_p)_{\text{turning}} = \frac{V}{R} = \frac{416.67}{2000} = 0.208 \text{ rad/s}$$

Gyroscopic couple,

$$C = I\omega\omega_p = 15.625 \times 400\pi \times 0.208$$

$$C = 4090.07 \text{ Nm}$$

$$C = 4.0907 \text{ kNm}$$



AGC : Active gyroscopic couple

RGC : Reactive gyroscopic couple

Thus, the nose will go little bit up and tail will go little bit down.

150.

Sol.

Given: Mass of rotor,

diameter of shaft

distance between bearing

eccentricity

$$m = 10 \text{ kg}$$

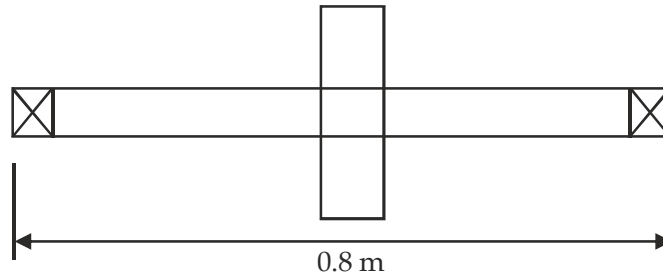
$$d = 20 \text{ mm} = 0.02 \text{ m}$$

$$l = 0.8 \text{ m}$$

$$e = 0.1 \text{ mm}$$

$$N = 3000 \text{ RPM}$$

$$E = 2 \times 10^6 \text{ bar}$$



Assuming bearing to be short, hence simply supported shaft

Deflection Now,

$$y = \frac{e}{\left(\frac{\omega_n}{\omega}\right)^2 - 1}$$

Now,

$$I = \frac{\pi}{64} d^4 = \frac{\pi}{64} (0.02)^4 = 7.854 \times 10^{-9} \text{ m}^4$$

$$\Delta = \frac{mg l^3}{48EI} = \frac{10 \times 9.81 \times 0.8^3}{48 \times 2 \times 10^6 \times 10^5 \times 7.854 \times 10^{-9}} = 6.66 \times 10^{-4} \text{ m}$$

$$\omega_n = \sqrt{\frac{g}{\Delta}} = \sqrt{\frac{9.81}{6.66 \times 10^{-4}}} = 121.35 \text{ rad / s}$$

$$\omega = \frac{2\pi N}{60} = \frac{2\pi \times 3000}{60} = 314.16 \text{ rad / s}$$

Hence,

$$y = \frac{0.1 \times 10^{-3}}{\left(\frac{121.35}{314.16}\right)^2 - 1} = -1.175 \times 10^{-4} \text{ m}$$

-ve sign shows that deflection is out of phase of centrifugal force.

Now, dynamic force on bearing = $sy = m\omega_n^2 y = 10 \times (121.35)^2 \times (-1.175 \times 10^{-4}) = 17.24 \text{ N}$

Total load = $mg + sy = 10 \times 9.81 + 17.24 = 115.34 \text{ N}$

151.

Sol. Vibration Isolation Theory

Vibration isolation theory involves strategies and techniques to minimise the transmission of mechanical vibrations from their source to surrounding structures or sensitive equipment. Vibration isolation theory plays a pivotal role in mitigating the adverse effects of mechanical vibrations in various applications, especially in scenarios where total vibration elimination is not always feasible, as is often the case in real-world engineering challenges.

Transmissibility Theory

Transmissibility theory explores the relationship between input and output vibrations, typically represented as the ratio of output to input vibrations, denoted as Transmissibility (T). The formula for Transmissibility is $T = \text{output/input}$.

When T is greater than 1, it indicates amplification, with maximum amplification occurring when the forcing frequency (f_f) aligns with the natural frequency (f_n) of the system. Transmissibility has no specific unit designation, although it may be referred to as the Q-factor. This concept plays a crucial role in calculating the efficiency of passive compensation methods.

152.

Sol. As per given data

For the vibrating system,

$$W = 19.62 \text{ kg} = 19.62 \times 9.81 = 192.4722 \text{ N}$$

$$K = \frac{8\text{kg}}{\text{cm}} = \frac{8 \times 9.81 \text{ N}}{10^{-2} \text{ m}} = 7848 \frac{\text{N}}{\text{m}}$$

$$c = 0.08 \text{ kg-s/cm} = \frac{0.08 \times 9.81}{10^{-2}} = 78.48 \text{ N-s/m}$$

$$(1) \quad 2\xi\omega_n = \frac{c}{m}$$

$$\xi = \frac{c}{2m\omega_n} = \frac{c}{2\sqrt{km}} = \frac{78.48}{2\sqrt{7848 \times 19.62}} = 0.1$$

$$(2) \text{ Natural frequency} \quad \omega_n = \sqrt{\frac{k}{m}} = \sqrt{\frac{7848}{19.62}} = 20 \text{ rad/s}$$

$$\text{Natural frequency of damped vibration} = \omega_d = \sqrt{1 - \xi^2} (\omega_n)$$

$$\omega_d = \sqrt{1 - 0.1^2} \times 20 = 19.899 \text{ rad/s}$$

$$(3) \text{ logarithmic decrement } (\delta) \quad \delta = \frac{2\pi\xi}{\sqrt{1 - \xi^2}} = \frac{2\pi \times 0.1}{\sqrt{1 - 0.1^2}} = 0.6314$$