Important Instructions to examiners:

1) The answers should be examined by key words and not as word-to-word as given in the model answer scheme.
2) The model answer and the answer written by candidate may vary but the examiner may try to assess the understanding level of the candidate.
3) The language errors such as grammatical, spelling errors should not be given more Importance (Not applicable for subject English and Communication Skills).
4) While assessing figures, examiner may give credit for principal components indicated in the figure. The figures drawn by candidate and model answer may vary. The examiner may give credit for any equivalent figure drawn.
5) Credits may be given step wise for numerical problems. In some cases, the assumed constant values may vary and there may be some difference in the candidate’s answers and model answer.
6) In case of some questions credit may be given by judgement on part of examiner of relevant answer based on candidate’s understanding.
7) For programming language papers, credit may be given to any other program based on equivalent concept.

Que 1

a) Attempt any six of following. **Marks 12 (6 x 2)**

i) **kinematic pair**: It is a combination of two kinematic links such that there is relative motion between the two and the relative motion is completely or successfully constrained.

OR

The two links or elements of machine when contact with each other, are said to form a pair. The relative motion between them is completely or successfully constrained (ie in definite direction) the pair is known as kinematic pair.

Ex. Square bar reciprocates in a square hole**. (01 mark for def. , 01 mark for example**)

ii) **Module :** It is the ratio of the pitch circle diameter in millimeters to the number of teeth. This is usually denoted by m.

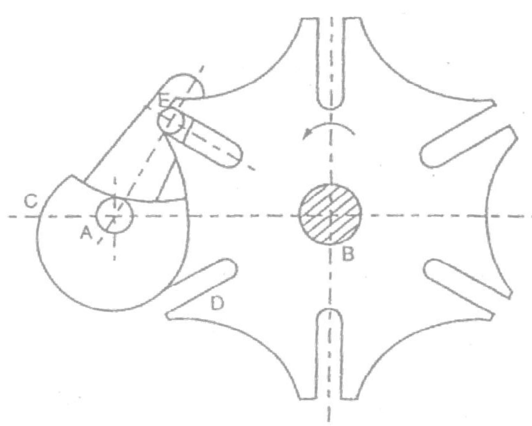
mathematically module m= D/T

where D = pitch circle diameter in mm

T = No of teeth

Unit of module is mm **(01 mark for def. , 01 mark for unit**)

iii) **Neat sketch of Geneva mechanism**

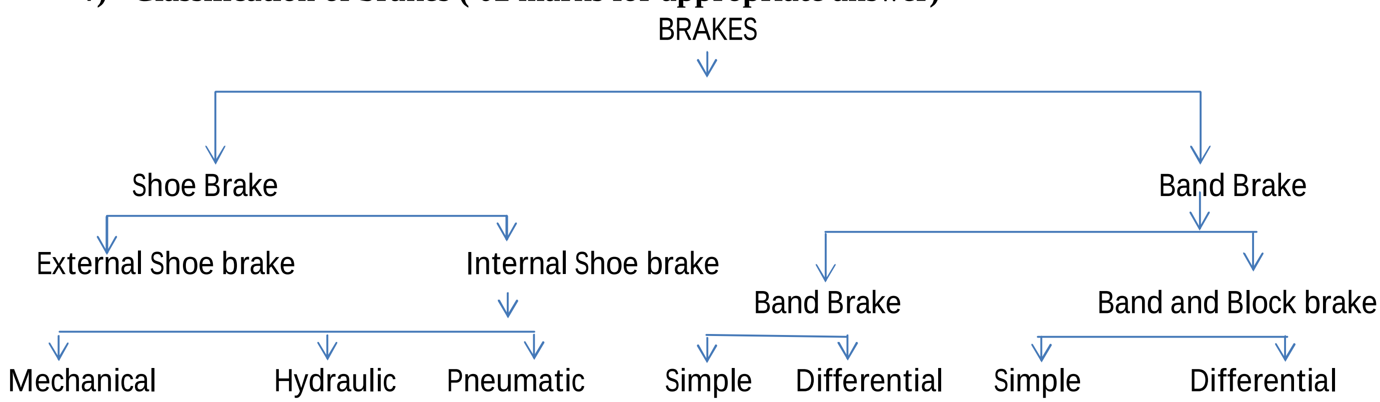


(02 marks for sketch)

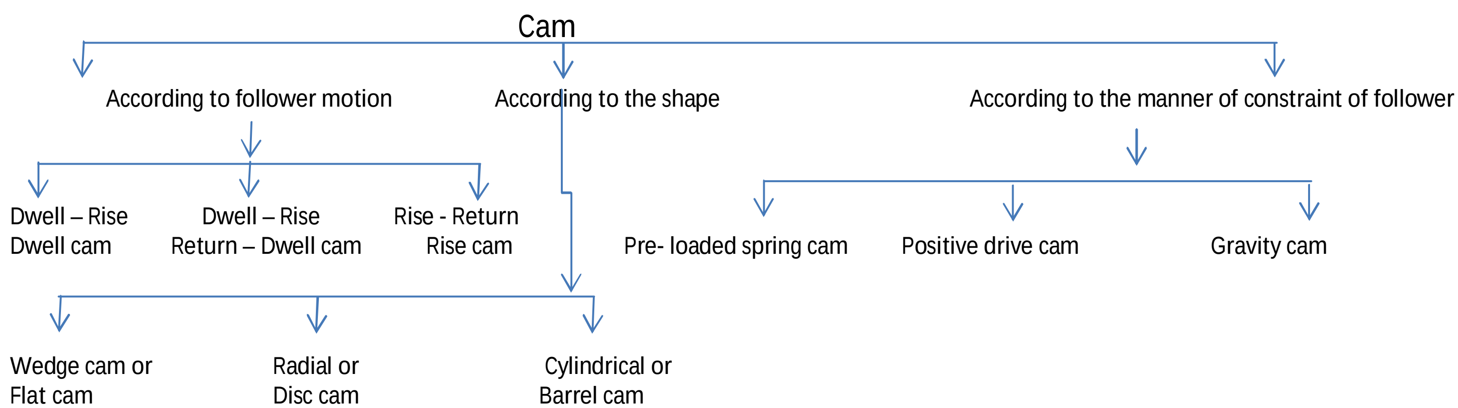
iv) **Fluctuation of Energy :** The variation of energy above or below mean torque line is known as fluctuation of energy

Fluctuation of speed: The variation of speed above or below mean speed of m/c during a cycle is called the fluctuation of speed. **(01 mark for each definition)**

v) **Classification of brakes (02 marks for appropriate answer)**



vi) Types of cams (02 marks for appropriate answer)



vii) Angle of Lap: The angle of lap is defined as the angle subtended by the portion of the belt which is in contact at the pulley surface of the pulley.

Slip : If the difference in tensions between tight and slack sides of the belt is too large to be resisted by friction between belt and the pulley, whole or the portion of the belt which is in contact with the pulley begin to slide. This results in relative motion between the belt the pulley and is termed as slip.

(01 mark for each definition)

viii) Natural Frequencies: The frequencies of vibration as a result of free vibrations of the system are known as natural frequencies. **(02 marks for appropriate answer)**

Q 1

b) Attempt any two of the following **Marks 8 (4 x 2)**

i) **Working of crank and slotted lever quick return mechanism**.

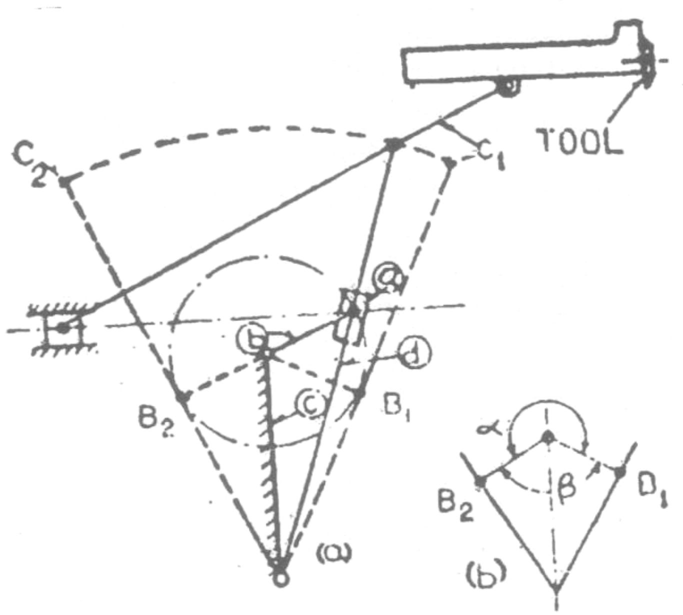
(02 marks for sketch,02 marks for description)

This type of mechanism is used in shaping and slotting machines. In this mechanism the slider ‘a’ reciprocates in oscillating slotted lever ‘d’ and ‘b’ rotates while ‘c’ is a stationary link ,as shown in fig. when link ‘b’ revolves the link ‘d’ oscillates as ‘b’ is smaller than ‘d’. Another link say ‘e’ connects the end d’ to the ram which carries the cutting tool which reciprocates perpendicular to the fixed link ‘c’.

The ram and hence the tool reverses its direction of motion when crank ‘b’ is perpendicular to lever ‘d’ thus the tool which performs cutting action during forward stroke does it during rotation of crank through the reflex angle ά . Similarly return stroke which is idle (non-cutting) stroke is executed during crank rotation of angle ß which is equal to $360^0 – ά$.

Thus, $\frac{\text{Time of cutting}}{\text{Time of return}} = \frac{ά}{ß} = \frac{ά}{360^0 – ά}$.

Necessary of quick return motion arises to increase effective working time of a shaper or a planner for which it is used.



Q 1 b

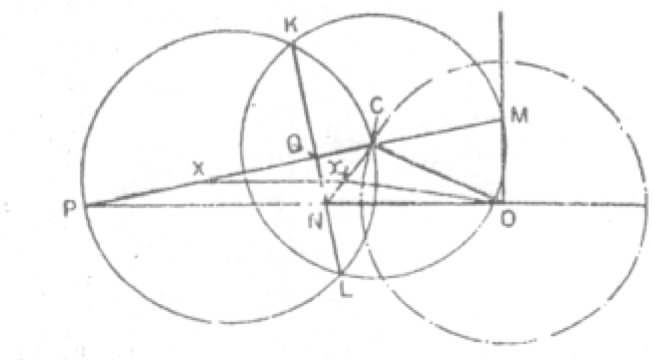
ii) Steps in Klein's construction (04 marks for appropriate answer)

Klein's construction is a simpler construction to get velocity and acceleration diagrams.

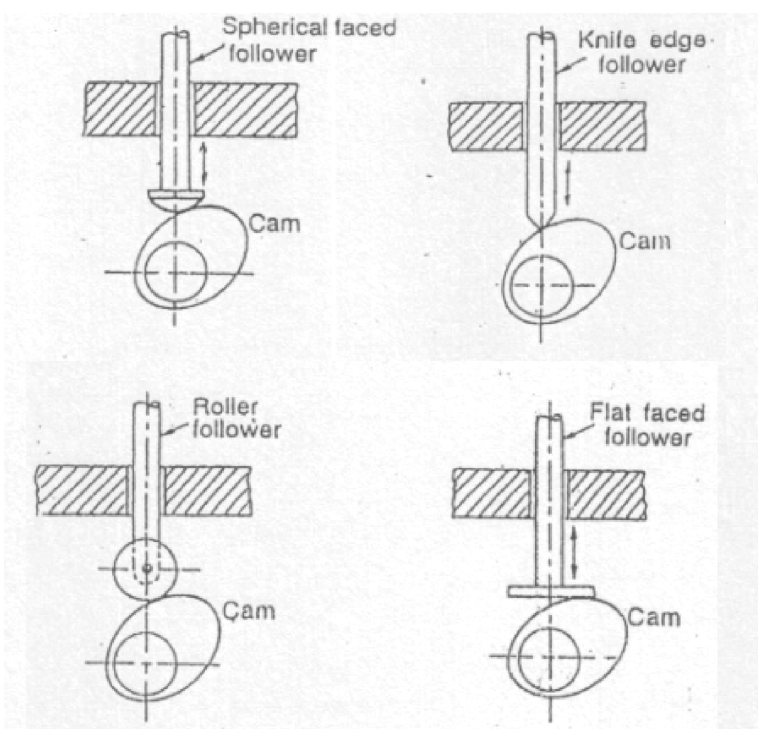
For example : for reciprocating engine mechanism OPC. draw a circle with PC as diameter as shown. and obtain velocity diagram OCM ie. produce PC to cut perpendicular to line of stroke in 'M' .

Draw another circle with 'C' as center and "CM" as radius cutting the first circle in points K and L.

Join "KL" which is the chord common to both the circles. Let it cuts PC and OP in "Q" and "N" respectively. Then "OCQN" is the required quadrilateral which is similar to acceleration diagram.



iii) Draw sketch of any four types of followers. (01 mark for each)



Q 2 Attempt any four of the following.

(a) **Inversion**: In a kinematic chain, the method of obtaining different mechanisms by fixing different links is known as inversion of the mechanism

It is a chain consisting of three turning pairs and one sliding pair & purpose of such a mechanism is to convert the reciprocating motion into rotary or vice versa.

Inversions of Single Slider Crank Chain are

Bull engine or pendulum pump

Oscillating cylinder engine

Rotary Engine

Crank & slotted lever QR mechanism

Whitworth QR mechanism

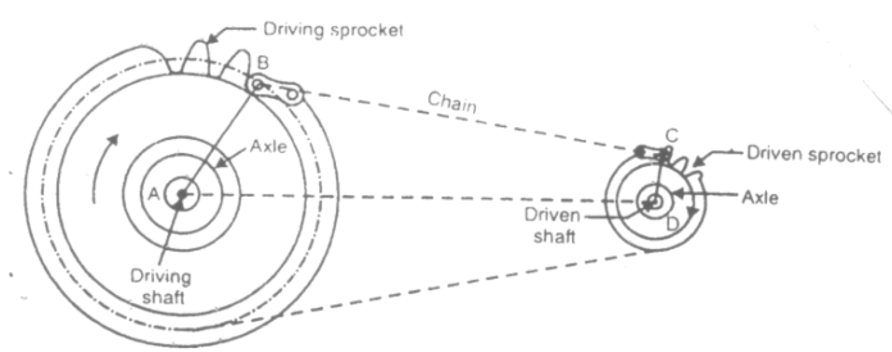
(02 marks for def. of inversion , 02 marks for inversions of single slider crank chain)

(b) Bicycle Free wheel Mechanism: (04 marks for appropriate answer)

It is a chain drive type mechanism. In this mechanism, the driving sprocket is mounted on the driving shaft 'A'. It is rotated by the force applied on the pedal. The driven sprocket is keyed to the output shaft 'D'.

The rear wheel is mounted on the output shaft. As the driving sprocket rotates, the chain will also rotates therefore, the driven sprocket will also rotate. Thus, the motion is transmitted to the rear wheel. It is a positive type drive. When the pedal is moved forward, then only it transmits the required motion to the rear wheel.

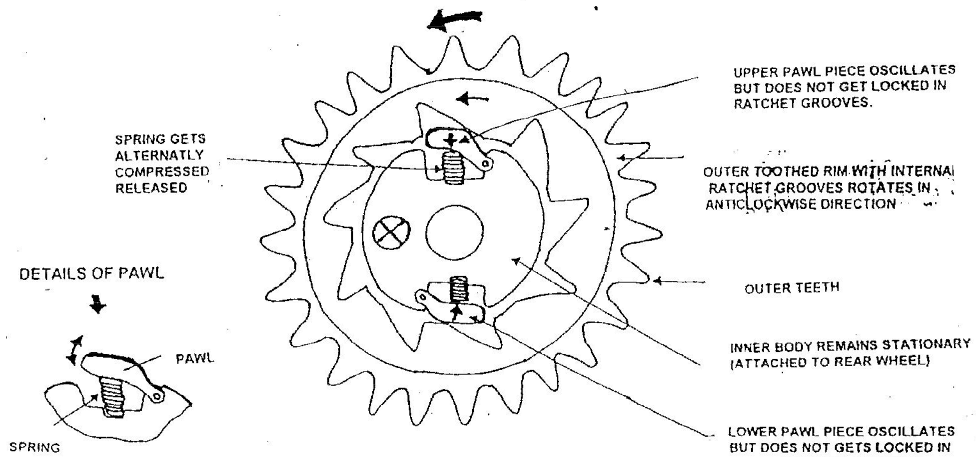
In case pedal is move reverse or held at a place, due to Paul & ratchet mechanism provided at the rear wheel, there is no motion transmission.



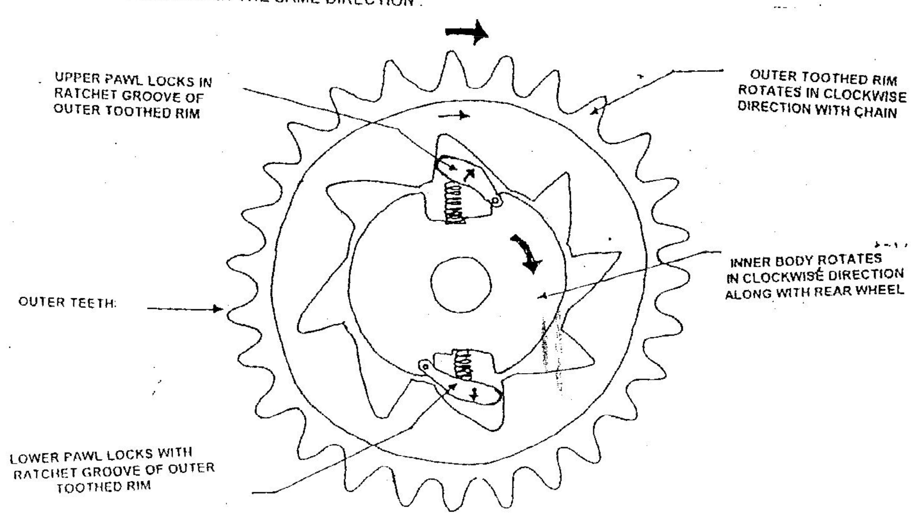
PRINCIPLE OF FREEWHEEL (PAWL & RATCHET MECHANISM)

BASIC MECHANISM USED IS "PAWL & RATCHET MECHANISM"

OUTER TOOTHED RIM ROTATING IN ANTI CLOCKWISE DIRECTION & INNER BODY STATIONARY (FREEWHEELING MODE)



OUTER TOOTHED RIM ROTATING IN "CLOCKWISE DIRECTION" & INNER BODY GETTING LOCKED TO OUTER RIM VIA PAWLS & ROTATING IN THE SAME DIRECTION .



(c) Difference (01 mark for each point)

Mechanism	**Structure**
1) A mechanism is a kinematic chain which transmits and transforms motion.	1) There is no relative motion between its links which transmits force only.
2) Mechanism is Skeleton diagram of a machine to transmit the definite motions between various links.	2) It serves to modify and transmit the forces only and no energy is transmitted for doing any mechanical work.
3) The motion & Force transmission is desirable	3) Only the Force transmission is desirable
4) Ex. Clockwork, Typewriter	4) Ex. Roof, Trusses, Buildings

d) Pantograph : **(02 marks for sketch, 02 marks for explanation)**

A Pantograph is an instrument used to reproduce to an enlarged or a reduced scale and as exactly as possible the path described by a given point. (See Fig.) The link ABQ is parallel to link OC and OA is parallel to link BC.

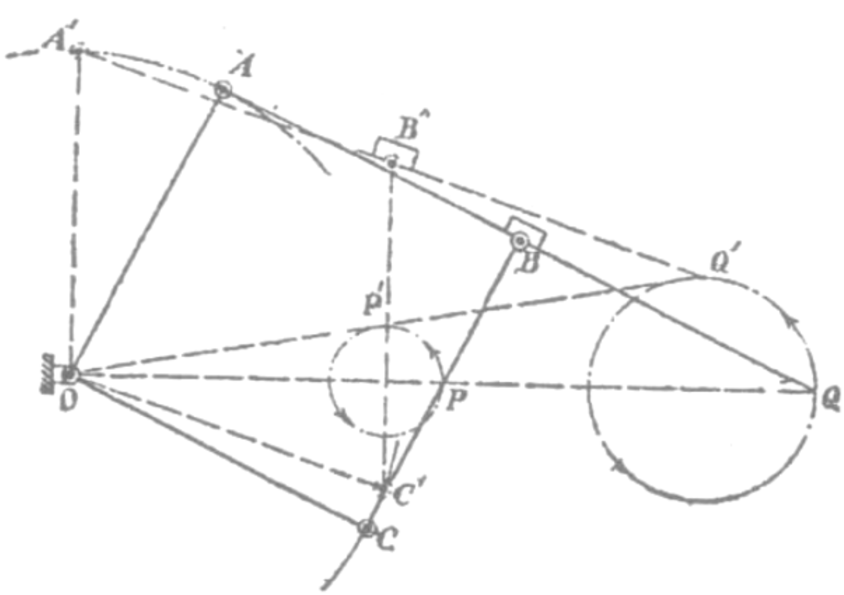
A line is drawn joining O to Q. It cuts BC at point P. It will be seen that points Q and P describe similar paths but to different scales. For the paths to be similar the essential requirement is that triangle OPC should be similar or triangle BPQ for all positions. For another position when point P has moved to P' in the process of tracing the path of a circle, it is seen that Q also traced the path of a circle from Q to Q'. The ratio

$\frac{\text{arc } PP'}{\text{arc } QQ'} = \frac{OP}{OQ}$ For any position the line will OQ will intersect line BC at point P such that.

$\frac{BQ}{OC} = \frac{PB}{PC} = \frac{PQ}{OP}$

and since OC and BQ are fixed lengths, the ratio $\frac{PC}{PB} = \frac{OP}{PQ}$

is always same for all positions.



(e) Difference (any 04 points, 01 mark for each)

	Gear Drive		Belt Drive
1	Suitable for high power transmission when distance between the shafts is small	1	Suitable for moderate power transmission when the distance between the shafts is large.
2	Requires less space due to small distance between the center to center distance between the shafts	2	Requires large space due to large center to center distance between the shafts
3	Perfect velocity ratio may be obtained	3	Perfect velocity ratio may not be obtained
4	Operation noise less	4	It makes noise
5	The torque transmitted is more	5	The torque transmitted is less
6	Manufacturing is difficult	6	Manufacturing is bit easier
7	Cost is more	7	Cost is less

f)

V – Belt Drive _ IC Engine / Compressors / Lathe m/c / Drilling m/c

Flat belt drive _ Flour mill and similar

Gear Drive _ Gear box of automobile / wrist watches / Scooter motor cycles

Chain Drive _ Bicycles, motor cycles etc.

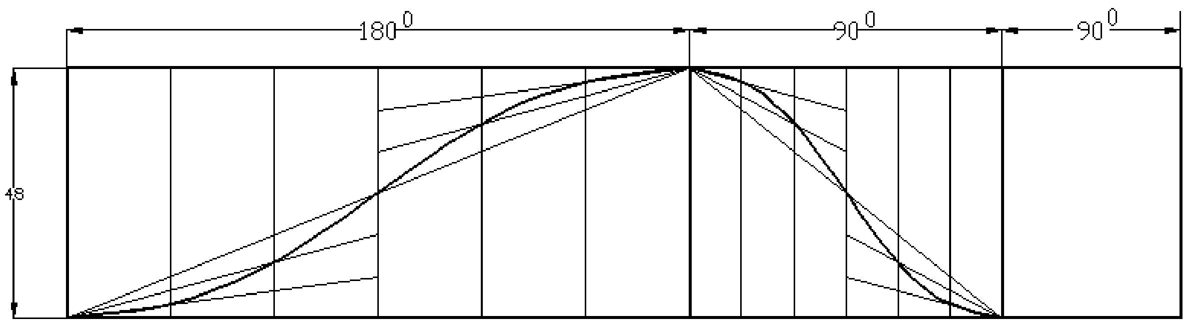
(01 mark for each)

Q3 a) Displacement diagram (04 marks for appropriate answer)

Scale on X-axis 1cm= 15^0 on Y-axis 1:1

Angle of rise $\Theta_1 = 180^0$ Dwell angle $\Theta_3 = 90^0$

Angle of return $\Theta_2 = 90^0$



Dwell period not to Scale

Q3 b) Crank OA ω = 20 rad/sec

$\propto = 25\, rad/sec^2$

$OA = 50\, mm$

Linear velocity = OA × ω

= 50 × 20 = 1000 mm/sec **(01 mark)**

Centripetal Acceleration :

$f_{OA}^{c} = \frac{(VOA)^2}{OA}$ = $(1000)^2/50$ = $20000\, mm/sec^2$ OR $20\, m/sec^2$ **(01 mark)**

Tangential Acceleration :

$f_{OA}^{T} = \propto_{OA} \times OA$

= 25 × 50 = $1250\, mm/sec^2$ = $1.25\, m/sec^2$ **(02 marks)**

Q 3 c) Flat Belt Problem

$$Tmax = T_1 + T_c$$

$$Centrigugal\ Tension\ T_c = \frac{Tmax}{3} = \frac{2500}{3}$$

$$T_c = 833.33\ N$$

$$\therefore\ T_1 = Tmax - T_c$$

$$= 2500 - 833.33$$

$\boldsymbol{T_1 = 1666.67\ N\ (Tight\ Side)}$ **02 Marks**

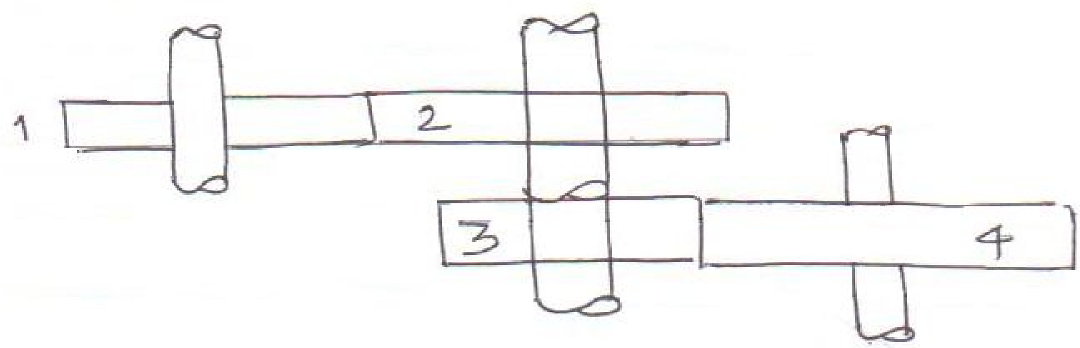
$$\frac{T_1}{T_2} = e^{\mu\vartheta} = e^{0.3\times170\times\pi/180} = 2.435$$

$$= \mathbf{2.435}$$

So, Tension on slack side

T_2= T_1 / 2.435 = 684.46 N **02 Marks**

Q. 3 d) Compound gear train :



Driving gear : $T_1 = 20$, $T_3 = 30$

Driven gear : $T_2 = 50$, $T_4 = 60$

Speed ratio :

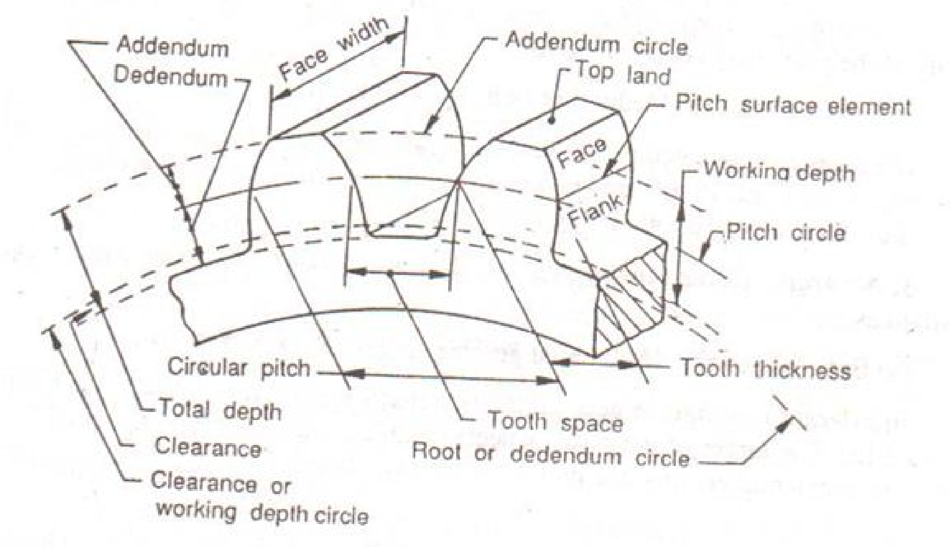
Speed Ratio : $\frac{N_1}{N_4} = \frac{T_2 \times T_4}{T_1 \times T_3}$ **(02 marks)**

$$\frac{N_1}{N_4} = \frac{50 \times 60}{20 \times 30} = 5$$

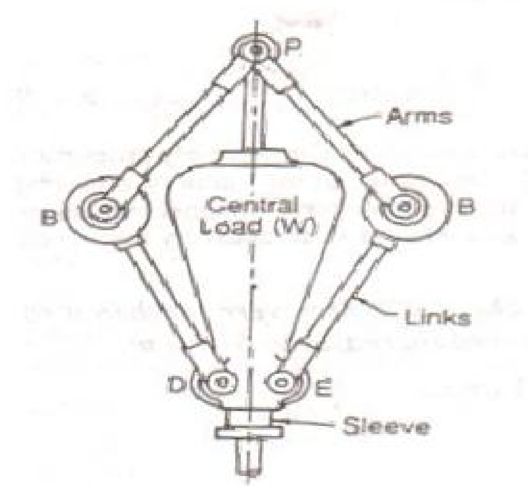
$N_1 = 400\, rpm$

$\therefore \boldsymbol{N_4 = \frac{400}{5} = 80 rpm \;\; clock\, wise}$ **(02 marks)**

Q3 e) Spur gear Terminology (04 marks for appropriate answer)



Q3 f) Porter governor

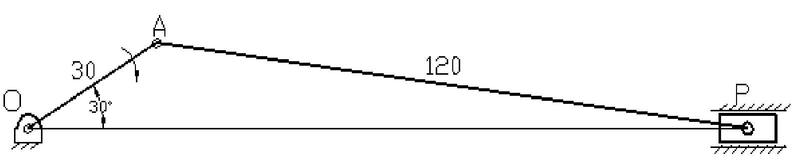


Porter governor is modification of watt's governor, with central load attached to the sleeve as shown in fig. the load moves up and down the central spindle. The additional downward force increases the speed of revolution required to enable the balls to rise to any predetermined level.

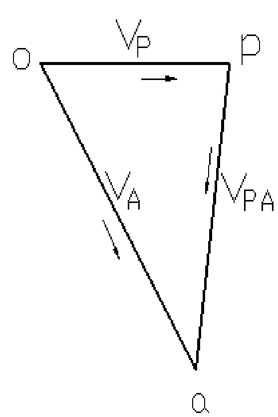
It regulates the speed when load is changing by changing the position of the sleeve. When balls rotates at higher speed sleeve moves in upward direction this happens when load reduces. Change in position of sleeves affects the supply of fuel and controls the speed of engine.

(02 marks for sketch, 02 marks for description)

Q. 4 a) Acceleration diagram :

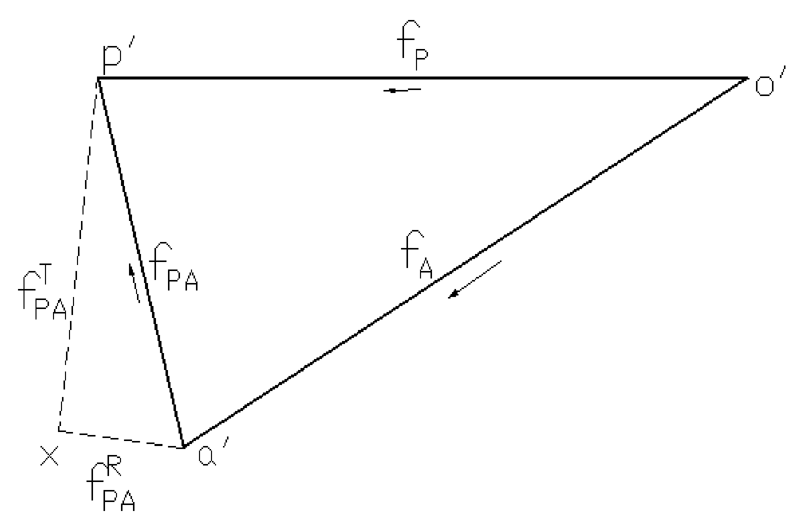


SPACE DIAGRAM SCALE 1:1



VELOCITY DIAGRAM

SCALE 1mm:10mm/sec



ACCELERATION DIAGRAM

SCALE 1mm :100 mm/sec^2

Q4 a) contd...

Angular velocity of link OA

$$\omega = \frac{2\pi N}{60} = \frac{2\pi \times 200}{60} = 20.94 \text{ rad/sec}$$

∴ Linear velocity of pt. A wrt O

$$v_{AO} = v_A = OA \times \omega$$
$$= 30 \times 20.94$$
$$= 628.20 \text{ mm/sec}$$

From velocity diagram

$$\overline{pa} = 54.81 \text{ mm}$$

∴ Linear velocity of pt P wrt A

$$v_{PA} = 54.81 \times 10 = 548.10 \text{ mm/sec}$$

also, $\overline{op} = 38.20$ mm

∴ velocity of slider P $v_P = 382.00$ mm/sec

Acceleration of pt A wrt O is

$$f^R_{AO} = f_A = \frac{[v_{AO}]^2}{OA} = 13140 \text{ mm/sec}^2$$

Also, radial acceleration of pt P wrt A is

$$f^R_{PA} \text{ or } f^c_{PA} = \frac{[v_{PA}]^2}{AP} = 2503 \text{ mm/sec}^2$$

From Acceleration diagram,

Acceleration of slider P, $f_P = \overline{o'p'} \times 100$

$$= \boxed{13099 \text{ mm/sec}^2}$$

Also, $f^T_{PA} = 6304$ mm/sec²

∴ Ang. Aceln. of link AP, $\alpha_{AP} = \frac{6304}{120} = \boxed{52.34 \text{ rad/s}^2}$

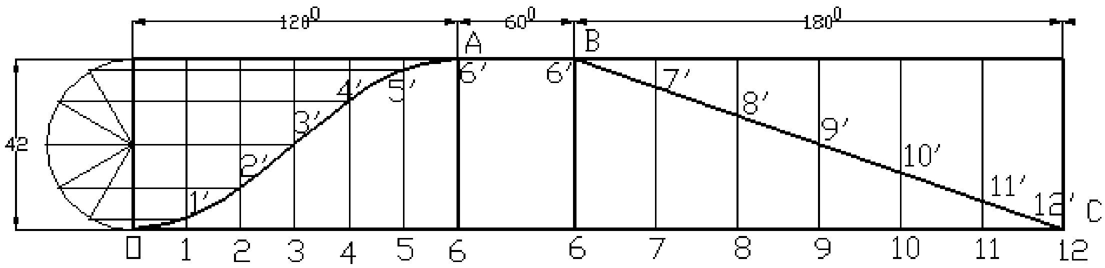
(02 marks for space diagram, 03 marks for velocity diagram & velocity values, 03 marks for acceleration diagram & its values)

Note : Due to graphical solution, small variation should be considered)

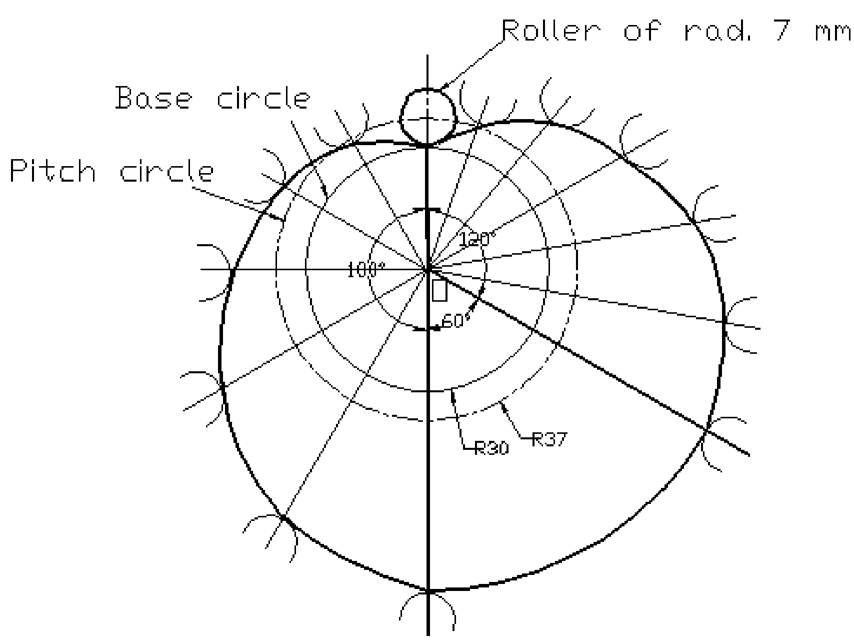
Q4 b) Cam Profile For Dis. Dig. 3 Marks , Cam Profile : 5 Marks

Scale on X-axis 1cm = 15^0 on Y-axis 1:1

Angle of rise $\theta_1 = 120^0$ Dwell angle $\theta_2 = 60^0$ Angle of return $\theta_3 = 180^0$



DISPLAMENT DIAGRAM



CAM PROFILE

Note: Dwell period not to scale

Q. 4 c) A leather Belt : Note: Read perm. Tension as Perm. Stress

(02 marks each for Tmax, m, Tc & power should be given)

density $\rho = 1\ gm/cm^3$ stress $\sigma = 21 \times 10^5\ N/m^2$

width $b = 25\ cm$, $t = 1.1\ cm$ $\frac{T_1}{T_2} = 2$

Max. Tension in belt is

$T_{max} = \sigma \times b \times t = 21 \times 10^5 \times 0.25 \times 0.011$

$= 5775\ N$

Mass per meter length of belt

$m = \rho \times A \times l$ $(\because l = 1\ m)$

$= \boxed{2.75}\ kg/meter$

critical velocity for Max. Power Transmission

$v = \sqrt{\frac{T_{max}}{3m}} = 26.45\ m/sec$

$\therefore$ Centrifugal Tension

$T_c = mv^2 = \boxed{1923.90\ N}$

$\therefore$ Tension on Tight side $T_1 = T_{max} - T_c$

$= \boxed{3851.09\ N}$

Since $\frac{T_1}{T_2} = 2$

$\therefore T_2 = 1925.54\ N$

$\therefore P_{max} = (T_1 - T_2) \times v$

$= (3851.09 - 1925.54) \times 26.45$

$= 50930.62\ W$

$= \boxed{50.93\ kW}$

Note:- Read permissible Tension as Permissible Stress

5 a Difference between Flywheel & Governer (Full marks any 4 differences)

	Flywheel	Governor
i)	Flywheel is a device which stores or supply energy when produced in excess & releases when required by m/c	i) Governor is a device controls the fuel to engine & controls mean speed
ii)	It regulates fluctuation of speed when there there is a variation in cyclic torque of m/c	ii) It regulates speed of engine when there is a external load variation.
iii)	It acts by virtue of its inertia	iii) It acts as a mechanism to control fuel supply
iv)	If torque variation is high, flywheel required is larger size	iv) If external load variation is higher, more control necessary on fuel supply
v)	Used in Engines, forging m/c, Sheet metal press, shearing m/c	v) Used in Engines

5 b) Sensitiveness & Hunting of Governor(02 marks each)

Sensitiveness:

Consider two governors A and B running at the same speed. When this speed increases or decreases by a certain amount, the lift of the sleeve of governor A is greater than the lift of the sleeve of governor B. It is then said that the governor A is more sensitive than the governor B.

In general, the greater the lift of the sleeve corresponding to a given fractional change in speed, the greater is the sensitiveness of the governor. It may also be stated in another way that for a given lift of the sleeve, the sensitiveness of the governor increases as the speed range decreases. This definition of sensitiveness may be quite satisfactory when the governor is considered as an independent mechanism. But when the governor is fitted to an engine, the practical requirement is simply that the change of equilibrium speed from the full load to the no load position of the sleeve should be as small a fraction as possible of the mean equilibrium speed. The actual displacement of the sleeve is immaterial, provided that it is sufficient to change the energy supplied to the engine by the required amount. For this reason, the sensitiveness is defined as the ***ratio of the difference between the maximum and minimum equilibrium speeds to the mean equilibrium speed.***

Let N_1 = Minimum equilibrium speed,

N_2 = Maximum equilibrium speed, and

N = Mean equilibrium speed $= \dfrac{N_1 + N_2}{2}$.

∴ Sensitiveness of the governor

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

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

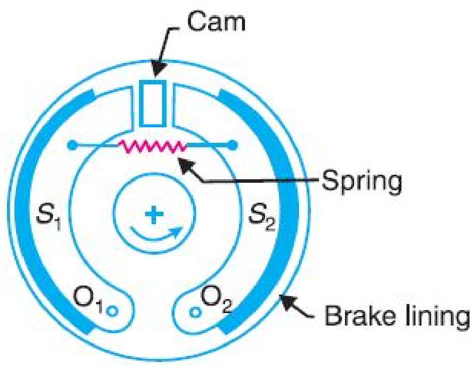
Hunting:

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 a too sensitive governor which changes the fuel supply by a large amount when a small change in the speed of rotation takes place.

For example, when the load on the engine increases, the engine speed decreases and, if the governor is very sensitive, the governor sleeve immediately falls to its lowest position. This will result in the opening of the control valve wide which will supply the fuel to the engine in excess of its requirement so that the engine speed rapidly increases again and the governor sleeve rises to its highest position. Due to this movement of the sleeve, the control valve will cut off the fuel supply to the engine and thus the engine speed begins to fall once again. This cycle is repeated indefinitely.

Such a governor may admit either the maximum or the minimum amount of fuel. The effect of this will be to cause wide fluctuations in the engine speed or in other words, the engine will hunt.

Q5 c) Internal Expanding Brake: (02 marks for sketch, 02 marks for explanation)



Internal expanding brake.

An internal expanding brake consists of two shoes S_1 and S_2 as shown in Fig. The outer surface of the shoes are lined with some friction material (usually with Ferodo) to increase the coefficient of friction and to prevent wearing away of the metal. Each shoe is pivoted at one end about a fixed fulcrum O_1 and O_2 and made to contact a cam at the other end. When the cam rotates, the shoes are pushed outwards against the rim of the drum. The friction between the shoes and the drum produces the braking torque and hence reduces the speed of the drum. The shoes are normally held in off position by a spring as shown in Fig. The drum encloses the entire mechanism to keep out dust and moisture. This type of brake is commonly used in motor cars and light trucks.

Q5 d) Rope Brake Dynamometer: (02 marks for sketch, 02 marks for explanation)

It is another form of absorption type dynamometer which is most commonly used for measuring the brake power of the engine. It consists of one, two or more ropes wound around the flywheel or rim of a pulley fixed rigidly to the shaft of an engine. The upper end of the ropes is attached to a spring balance while the lower end of the ropes is kept in position by applying a dead weight as shown in Fig.

In order to prevent the slipping of the rope over the flywheel, wooden blocks are placed at intervals around the circumference of the flywheel.

In the operation of the brake, the engine is made to run at a constant speed. The frictional torque, due to the rope, must be equal to the torque being transmitted by the engine.

Let W = Dead load in newtons,
S = Spring balance reading in newtons,
D = Diameter of the wheel in metres,
d = diameter of rope in metres, and
N = Speed of the engine shaft in r.p.m.

$\therefore$ Net load on the brake

$$= (W - S)\text{ N}$$

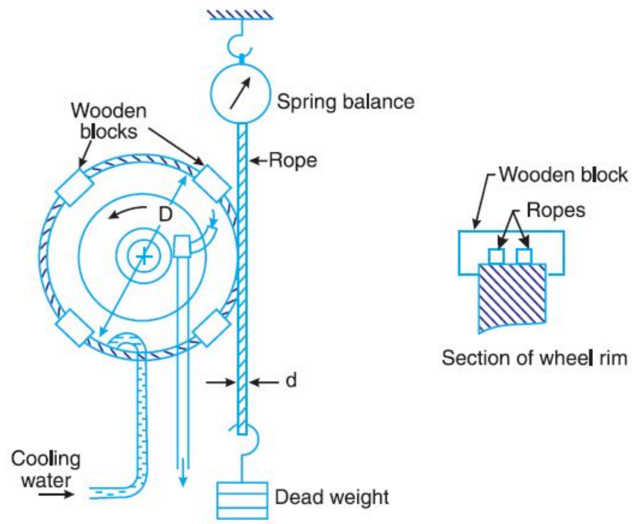
We know that distance moved in one revolution

$$= \pi(D + d)\text{ m}$$

$$= (W - S)\,\pi(D + d)\text{ N-m}$$

and work done per minute

$$= (W - S)\,\pi(D + d)\,N\text{ N-m}$$



Rope brake dynamometer.

$\therefore$ Brake power of the engine,

$$\text{B.P} = \frac{\text{Work done per min}}{60} = \frac{(W - S)\,\pi\,(D + d)N}{60}\text{ watts}$$

If the diameter of the rope (d) is neglected, then brake power of the engine,

$$\text{B.P.} = \frac{(W - S)\,\pi\,D\,N}{60}\text{ watts}$$

5 e) Difference between Disc brake & Internally Expanding Brakes

(any 4 differences, 01 mark each)

Disc brake	Internally Expanding Brakes
i) Rubbing / Friction surfaces are flat	i) Surfaces are cylindrical
ii) Heat dissipation easy & better since assembly is exposed to atmosphere	ii) Assembly closed & so, heat dissipation lower The drum housing always retains heat.
iii) Cooling of surfaces is quicker	iii) Cooling time gets prolonged
iv) It has heat dissipation holes	iv) No such provision
v) Braking is quicker (Vehicle stops in quick time)	v) Braking is slow as excessive braking causes drum heating

Q5 f) Problem on Balancing of masses: (02 marks for Resultant, 02 marks for m)

Given : $m_1 =$ 10 kg ; $m_2 =$ 20 kg ; $m_3 =$ 15 kg ;

$r_1 = 0.2$ m ; $r_2 = 0.25$ m ; $r_3 = 0.15$ m; $r = 0.30$m

$\theta_1 = 0°$; $\theta_2 = 60°$; $\theta_3 = 150°$

Let m = Balancing mass, and

θ = The angle which the balancing mass makes

Since the magnitude of centrifugal forces are proportional to the product of each mass and its radius,

therefore

$m_1 \cdot r_1 = 10 \times 0.2 = 2$ kg-m

$m_2 \cdot r_2 = 20 \times 0.25 = 5$ kg-m

$m_3 \cdot r_3 = 15 \times 0.15 = 2.25$ kg-m

Resolving $m_1.r_1$, $m_2.r_2$, $m_3.r_3$ and $m_4.r_4$ horizontally,

$$\Sigma H = m_1 \cdot r_1 \cos\theta_1 + m_2 \cdot r_2 \cos\theta_2 + m_3 \cdot r_3 \cos\theta_3$$

$$= 2 \cos 0° + 5 \cos 60° + 2.25 \cos 150°$$

$$= \boxed{2.55 \text{ kg-m}}$$

Now resolving vertically,

$$\Sigma V = m_1 \cdot r_1 \sin\theta_1 + m_2 \cdot r_2 \sin\theta_2 + m_3 \cdot r_3 \sin\theta_3$$

$$= 2 \sin 0° + 5 \sin 60° + 2.25 \sin 150°$$

$$= \boxed{5.455 \text{ kg-m}}$$

$\therefore$ Resultant, $R = \sqrt{(\Sigma H)^2 + (\Sigma V)^2} = \boxed{6.02 \text{ kg-m}}$

We know that

$m \cdot r = R = 6.02$ m = 6.02 / 0.30 = 20.067 kg

and $\tan\theta' = \Sigma V / \Sigma H$ = $\boxed{\theta' = 64.94°}$

So, Balancing Mass = 20.067 kg to be located at ϴ = 244.94^0

6a) Causes of Vibrations **(Full marks to any 4 points)**

There are various factors those act as disturbing means to the machines from their equilibrium conditions. These are the disturbing forces that cause vibrations in machines. From safe running & life point of view, the vibrations are dangerous. Vibrations can result from a number of conditions, acting alone or in combination. Vibration problems might be caused by auxiliary equipment, not just the primary equipment.

Following are some causes of vibrations

i) I**mbalance**—An unbalanced mass in a rotating component will cause vibration when the unbalanced weight rotates around the machine's axis, creating a centrifugal force. Imbalance could be caused by manufacturing defects (machining errors, casting flaws) or maintenance issues (deformed or dirty fan blades, missing balance weights). As machine speed increases the effects of imbalance become greater. Imbalance can severely reduce bearing life as well as cause undue machine vibration.

ii) **Misalignment/shaft run-out**—Vibration can result when machine shafts are out of line. Angular misalignment occurs when the axes of (for example) a motor and pump are not parallel. When the axes are parallel but not exactly aligned, the condition is known as parallel misalignment. Misalignment can be caused during assembly or develop over time, due to thermal expansion, components shifting or improper reassembly after maintenance. The resulting vibration can be radial or axial (in line with the axis of the machine) or both.

iii) **Wea**r—As components such as ball or roller bearings, drive belts or gears become worn, they might cause vibration. When a roller bearing race becomes pitted, for instance, the bearing rollers will cause a vibration each time they travel over the damaged area. A gear tooth that is heavily chipped or worn, or a drive belt that is breaking down, can also produce vibration.

iv) **Looseness / Clearance** —Vibration that might otherwise go unnoticed can become obvious and destructive if the component that is vibrating has loose bearings or is loosely attached to its mounts. Such looseness might or might not be caused by the underlying vibration. Whatever its cause, looseness can allow any vibration present to cause damage, such as further bearing wear, wear and fatigue in equipment mounts and other components.

v) **Excessive load-** This is also one reason for cause of vibrations. If load on m/c is beyond its capacity, vibrations are caused.

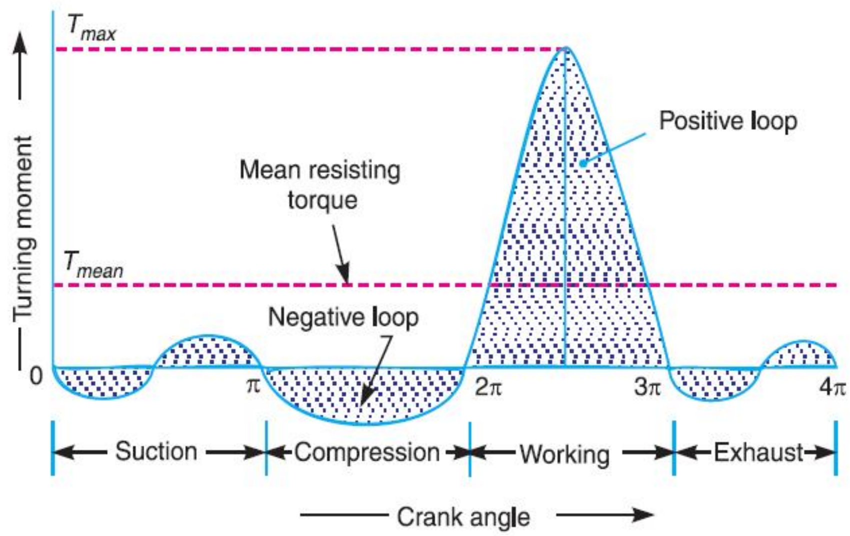
vi) **Extreme speeds**- If there is an extreme variation in operational speeds of the m/c , then also vibrations are created.

vii) **Foundation of m/c-** This is basic cause if it is not proper to absorb vibrations**.**

6 b) T-Ө Diagram of Single Cylinder 4 stroke engine (For Fig: 2 M, Ans: 2M)

A turning moment diagram for a four stroke cycle internal combustion engine is shown.

We know that in a four stroke cycle internal combustion engine, there is one working stroke after the crank has turned through two revolutions, *i.e.* 720° (or 4 π radians).



Turning moment diagram for a four stroke cycle internal combustion engine.

Since the pressure inside the engine cylinder is less than the atmospheric pressure during the suction stroke, therefore a negative loop is formed as shown in Fig. During the compression stroke, the work is done on the gases, therefore a higher negative loop is obtained. During the expansion or working stroke, the fuel burns and the gases expand, therefore a large positive loop is obtained. In this stroke, the work is done by the gases. During exhaust stroke, the work is done on the gases, therefore a negative loop is formed.

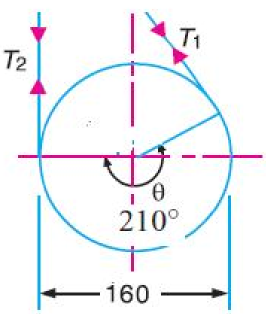
Q6c) Problem on Simple Band Brake (02 marks each for torque & power)

Given :

$N = 460$ r.p.m. or $\omega = 2\pi \times 460/60 = 48.15$ rad/s ; $\mu = 0.35$

$d = 160$ mm i.e. $r = 0.08$ m

First of all, let us find the tensions in the tight and slack sides of the band *i.e.* T_1 and T_2 respectively.



angle of lap of the band on the drum,

$$\theta = 210° = 210 \times \frac{\pi}{180} = \boxed{3.666 \text{ rad}}$$

We know that

$$\frac{T_1}{T_2} = e^{\mu.\theta} = e^{0.35 \times 3.666}$$

i.e. $\frac{T_1}{T_2} = 3.60$ But $T_1 = 3200$ N

$$\therefore \quad T_2 = 3200 / 3.60 = \boxed{888.88 \text{ N}}$$

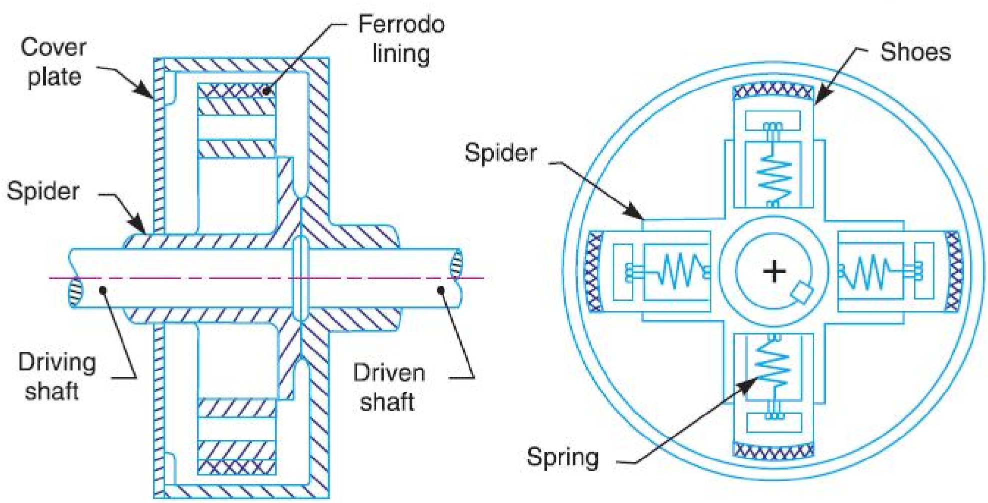
We know that torque applied,

$$T_B = (T_1 - T_2)\, r = (3200 - 888.88) \times 0.08 = 184.88 \text{ N-m}$$

Also, Power absorbed in braking P = 2 π N T / 60 = 8901.81 W = 8.9 kW

Q6d) Centrifugal Clutch (02 marks for sketch, 02 marks for answer)

The centrifugal clutches are usually incorporated into the motor pulleys. It consists of a number of shoes on the inside of a rim of the pulley, as shown in Fig. The outer surface of the shoes are covered with a friction material. These shoes, which can move radially in guides, are held



Centrifugal clutch.

against the boss (or spider) on the driving shaft by means of springs. The springs exert a radially inward force which is assumed constant. The mass of the shoe, when revolving, causes it to exert a radially outward force (*i.e.* centrifugal force). The magnitude of this centrifugal force depends upon the speed at which the shoe is revolving.

When the centrifugal force is less than the spring force, the shoe remains in the same position as when the driving shaft was stationary, but when the centrifugal force is equal to the spring force, the shoe is just floating.

When the centrifugal force exceeds the spring force, the shoe moves outward and comes into contact with the driven member and presses against it. The force with which the shoe presses against the driven member is the difference of the centrifugal force and the spring force.

Q6e) Problem on Foot step bearing (02 marks each for torque & power)

Given : $D =$ 0.2 m ; $N =$ 150 rpm

load W = 20 kN = 20000 N Coeff. of frction = μ = 0.05

We know that for uniform pressure distribution, the total frictional torque,

$$T = \frac{2}{3} \times \mu . W . R = \frac{2}{3} \times 0.05 \times 20 \times 10^3 \times 0.1 = \boxed{66.67 \text{ N-m}}$$

$\therefore$ Power lost in friction,

$$P = 2\pi N T / 60 = 2 \times \pi \times 150 \times 66.67 / 60$$

$$= \boxed{1046.67 \text{ W}} = \boxed{1.046 \text{ kW}}$$

Q6f) Problem on Single plate clutch (02 marks each for torque & power)

Note: Give due consideration for following assumptions ii & iii (Not given in Que.)

Given : $d_1 = 25$ cm or $r_1 = 0.125$ m $d_2 = 20$ cm or $r_2 = 0.1$ m

$N = 700$ r.p.m. Axial Load W = 1500 N

Assumptions : i) Uniform wear ii) Coeff. of friction = $\mu = 0.3$

iii) Both surfaces of clutch are effective i.e. n =2

We know that torque transmitted,

$$T = 1/2\ n.\mu.W.\ (r_1 + r_2)$$

$$= 1/2 \times 2 \times 0.3 \times 1500 \times (0.125 + 0.1) \quad (\because n = 2, \text{for both sides of plate effective})$$

$$= \boxed{101.25 \text{ N-m}}$$

$\therefore$ Power transmitted by a clutch,

$$P = 2\pi NT / 60 = 2 \times \pi \times 700 \times 101.25 / 60$$

$$= \boxed{7418.25 \text{ W}} = \boxed{7.418 \text{ kW}}$$