CAM ANALYSIS APPARATUS

Experiment -1

OBJECTIVE: -

1. To Study the motion of follower with the given profile of the cam.
2. To plot the n-q (Follower displacement Vs Angle of rotation) curves for different cam follower pairs

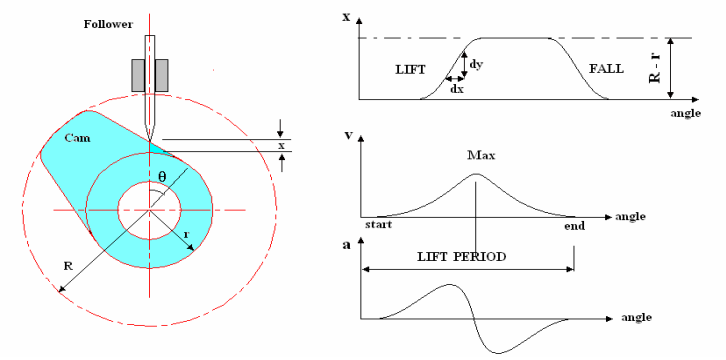
THEROY:-

Cams precede the development of modern Mechatronics but still have their uses in a variety of devices. They convert the rotation of a shaft into a specified linear motion of a follower. A typical example is the overhead cam shaft in an internal combustion engine. Many applications have been replaced by computer controlled machines using hydraulic, pneumatic and electric actuators. The motion of the follower depends on the shape of the cam. The most basic motion might be a simple quick stroke (e.g. to knock something off a conveyer belt. A cam as shown in the diagram would be suitable. A key part of your work is to understand how to produce the cam shape that produces the required motion. You should already know the relationship between displacement, velocity and acceleration.

For a displacement x, the velocity is v = dx/dt = gradient of the displacement - time graph. This is clearly zero at any time where the profile is circular. We must consider the inertia force produced by the sudden movement which produces acceleration and deceleration of the follower. The acceleration is a = dv/dt = the gradient of the velocity – time graph. It is normal to round the corners to avoid excessive wear.

DISPLACEMENT, VELOCITY AND ACCELERATION OF FOLLOWER

Consider a simple cam with a pointed follower. The centre lines of the follower and cam are the same. The follower is restrained to move along the centre line by the guides. The displacement of the follower is x and a typical displacement – angle graph is shown



Cam may be defined as a rotating and reciprocating element of a mechanism which imparts a reciprocating or oscillating motion to another element called follower. The cams are of the disc or cylindrical types and the follower are of knife edge, roller or flat faced.
The usual motions for the follower are **S.H.M**

Let S= lift of the follower

X= Displacement of the follower when crank has turned

Then $X=S/2\{1-\cos\theta)\}$

$V= \omega S / 2$

$V_{max} = \Pi \omega S / 2\theta$

$A=\omega^2 S \cos\theta/2$ during ascent

$A_{max}= \Pi^2\omega^2 S/2\theta^2$ during decent

Uniform acceleration or deceleration

Then displacement

$Y=1/2\ at^2$

$V_{average} =S/t$

$V_{max}= 2S/t$

$=2\omega S/\theta_0$ during ascent

$=V\omega^2 S/\theta^2$ during descent

The jump, phenomenon occurs in case of cam operating under the action of compression spring load. This is a transient condition that occurs only with high speed, highly flexible cam follower systems. With jump the cam and the follower separate owing to excessively unbalanced forced exceeding the spring force during the period of negative acceleration. This is indelible since the fundamental function of the cam-follower system; the constraint and control of follower motion are not maintained. Also related at the short life of the cam flank surface, high noise, and vibrations and poor action. A cam – follower retained against the cam with a compression retaining spring will under certain conditions, jump or bounce out of contact with the cam. This condition is most likely to occur with low values of damping and with high-speed cams or quite flexible follower trains.

Crossover shock occurs in a positive drive cam mechanism when contact moves from one side of the cam to the other. Clearance and backlash are taken up during the crossover, and impact occurs. Crossover takes place on the rise or return motion when the acceleration changes sign and when the velocity is at its peak. Preloading the system to remove backlash, by designing for a low peak velocity, and by using rigid follower train can reduce the effect. Roth art states that jump will not occur in high-speed system if at least two full cycles of vibration occur during the positive acceleration time-interval of the motion. If a smaller number of cycles exist during this period then, the states, the system should be investigated mathematically to determine if jump exists. This condition can be expressed by the equation:

Bi k > 2.

Where Bi is the angle through which the cam rotates during positive acceleration period.

This figure can probably be reduced slightly for appreciable amount of damping.

As shown in fig. 1. Spring K2 loses compression when jump beings and is carried in motion with the mass. The resulting motion now gets rather complicated because the mass too must be redistributed. Probably a good first approximation could be obtained by concentrating a portion of the mass at the bottom of spring and treating motion as a system of two degrees of freedom. It must be noted, and then the system will vibrate at a new frequency after jump being and analysis of the motion using the old frequency after jump being is not a true description of the motion.

The cam & follower assemblies are:

a) Circular arc cam with mushroom follower.

b) Tangent cam with roller follower.

c) An eccentric cam with knife-edge follower.

DESCRIPTION:

Apparatus is a motorized unit consisting of a camshaft driven by a variable speed motor. The shaft runs in a double bearing. The free end of the camshaft has a facility to mount the cam easily.

The follower is properly guided in gunmetal bushes and the type of follower can be changed according to the cam under test. Graduated circular protractor is fitted co-axial with the shaft. And a Dial Gauge fitted on the follower shaft is used to note the follower displacement for the angle of cam rotation. A spring is used to provide controlling force to the follower system. Weights on the follower shaft can be adjusted as per the requirement. An arrangement is provided to regulate the speed. The apparatus is very useful for testing the cam performance for jump phenomenon during operation. On this apparatus the effect of change of inertia forces on jump action of cam-follower during operation can be observed. It is useful for testing various cam & follower pairs. Three cams and three followers will be supplied with the apparatus. Theses are already hardened to reduce the wear.

UTILITIES REQUIRED:

Electricity supply 230 VAC, single Phase

Stroboscope (Optional)

PROCEDURE:

1. Fix the required cam and follower to be tested.
2. Bring the cam and follower to zero position
3. Rotate the cam slowly and note down the angle of rotation of the cam at regular interval and note the corresponding displacement of follower
4. Plot the graph between displacement of the follower and the angle of rotation of cam
5. Adjust the speed of motor to a particular mark with the help of variac.
6. Measure the RPM and stroke of the cam and follower

OBSERVATION TABLE:

S.No	Angle of rotation(θ)	Displacement of the follower(mm)
1.		
2.		
3.		
4.		
5.		

CALCULATION:

Plot a graph between displacement of the follower and the angle of rotation of the cam.

WHIRLING OF SHAFT APPARATUS
EXPERIMENT - 2

OBJECT: - To observe the modes of vibration
To determine the critical speed of the given shaft with the given end conditions**.**

THEORY:-

When any elastic body such as spring beam & shaft are displaced by the application of external forces & then released. They execute a vibratory motion. When the particles of the shaft or any disc move approximately perpendicular to the axis of shaft. Then the vibrations are known as transverse vibration. In this case, the shaft is straight & bends alternately & bending stresses are induced in the shaft. When a shaft is fixed at both ends, normally the centre of gravity will displace from the axis of rotation at a very lower amount. As a result of this displacement, the centre of gravity is subjected to a centripetal acceleration as soon as the shaft begins to rotate.

The inertia force acts radially outwards & bends the shaft. The bending of shaft is not only depends upon the value of eccentricity. But also depends upon the speed at which the shaft rotates. Consider a shaft AB of length L, carrying a load at any point of shaft. When some amount deflection will be given to it then released, it will make transverse vibration. The deflection of the shaft is proportional to load W & if the beam is deflected beyond the static equilibrium position, then the load will vibrate with some harmonic motion. If σ is the static deflection due to load W, then the natural frequency of free transverse vibration is.

$$F_n = \frac{1}{2\pi}\sqrt{\frac{g}{\sigma}} = \frac{0.4985}{\sqrt{\sigma}}$$

Then for simply supported beam with a uniformly distributed load of W per unit length, σ can be find out by using formula

$$\sigma = \frac{5}{384}\ X\ \frac{WL^4}{EI}$$

Where, W = weight / unit length (Nm)
Then for fixed beam with a uniformly distributed load of W per unit length
σ can be find out by using following formula

$$\sigma = \frac{WL^4}{384EI}$$

In actual practice, a rotating shaft carries different mountings & accessories in the from of gears, pulleys etc. when the gears or pulleys are put on the shaft, the centre of gravity of the pulley or gear does not coincide with the centre line of the bearing or with the axis of the shaft, when the shaft is stationary. This means that the centre of gravity of the pulley or gear is at a certain distance from the axis of rotation and due to this, the shaft is subjected to centrifugal force. This force will bent the shaft, which will further increase the distance of centre of gravity of the pulley or gear from the axis of rotation. This correspondingly increases the value of centrifugal force, which further increase the distance of centre of gravity from the axis of rotation. The bending of shaft is not only depends upon the value of eccentricity but also depends upon the speed at which the shaft rotates. The speed at which

the shaft runs. So that the additional deflection of the shaft from the axis of rotation becomes infinite, is known as critical or whirling speed.
If N_c is the critical or whirling speed in r.p.s.

Then

$$2\pi N_C = \sqrt{\frac{g}{\sigma}}$$

$$N_C = \frac{1}{2\pi}\sqrt{\frac{g}{\sigma}} \quad RPS$$

The critical or whirling speed is same as the natural frequency of transverse vibration, but its unit will be revolutions per second. Dunker ley's method is used to determine the critical speed of a shaft, which may be subjected to uniformly distributed load, A shaft supported with short bearings is assumed to be a simply supported shaft, while shaft supported with long bearings (or journal bearings) assumed to have both end fixed.

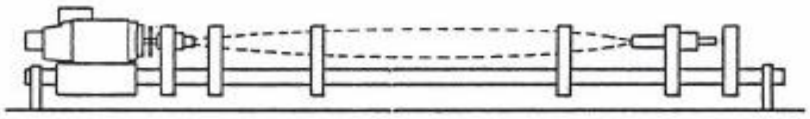


Figure 4. First mode whirl.

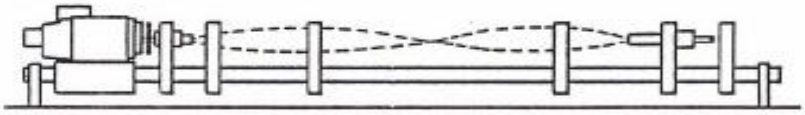


Figure 5. Second mode whirl

When a shaft rotates, it will go in to transverse vibration .If the shaft is out of balance, the resulting centrifugal force will induce the shaft to vibrate. When the shaft rotates at a speed equal to the natural frequency of transverse oscillations, This vibration becomes large and shows up as a whirling of the shaft. It also occurs at multiples of the resonant speed. This can be very damaging to heavy rotary machines such as turbine generator sets and the system must be carefully balanced to reduce this effect and designed to have a natural frequency different to the speed of rotation. When starting or stopping such machinery, the critical speeds must be avoided to prevent damage to the bearings and must be avoided to prevent damage to the bearings and turbine blades.
Suppose a shaft rotates, centrifugal force will cause it to bend out. Let the deflection of the shaft be r. The distance to the centre of gravity is then r + e.
The shaft rotates at ω rad/s. The transverse stiffness is k_t N/m
The deflection force is hence $F = K_t r$
The centrifugal force is $M \omega^2 (r + e)$

Equating force we have

$k_t = M\omega^2(r+e)$ from which $r = \frac{m\omega^2(r+e)}{k_t} = \frac{M\omega^2 r}{k_t} + \frac{M\omega^2 e}{k_t}$

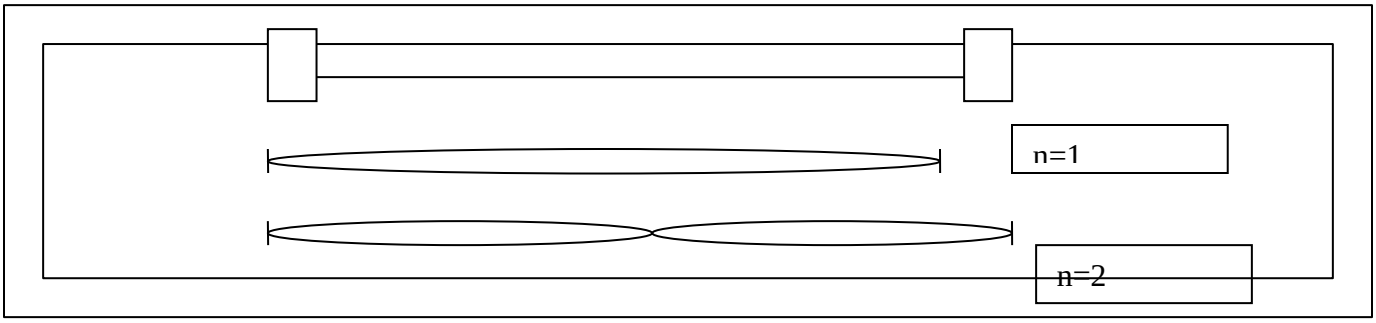
$$r = \frac{M\omega^2 e}{k_t(1 - \frac{M\omega^2}{k_t})}$$

It has already seen that $\frac{k_t}{M} = \omega^2{}_n$ $r = \frac{\omega^2 e}{\omega_n{}^2(1 - \frac{\omega^2}{\omega_n{}^2})} = \frac{e}{\left[\frac{\omega_n}{\omega}\right] - 1}$

From this we see that when $\omega_n = \omega$ r = e/o which is infinity. This means that no matter how small the imbalance distance e is, the shaft will whirl at the natural frequency. Balancing does help but can never be perfect so whirling is to be avoided on the best of machines.
The frequencies at which whirling occurs are calculated by the same methods as for transverse vibrations of beams

SIMPLY SUPPORTED – The ends are free to rotate normal to the axis (e.g. self aligning bearings)

$$f = \frac{\pi}{2} n^2 \sqrt{\frac{gEI}{WL^4}}$$ Where n is the mode (i.e. n=1, 2, 3, 4…….)

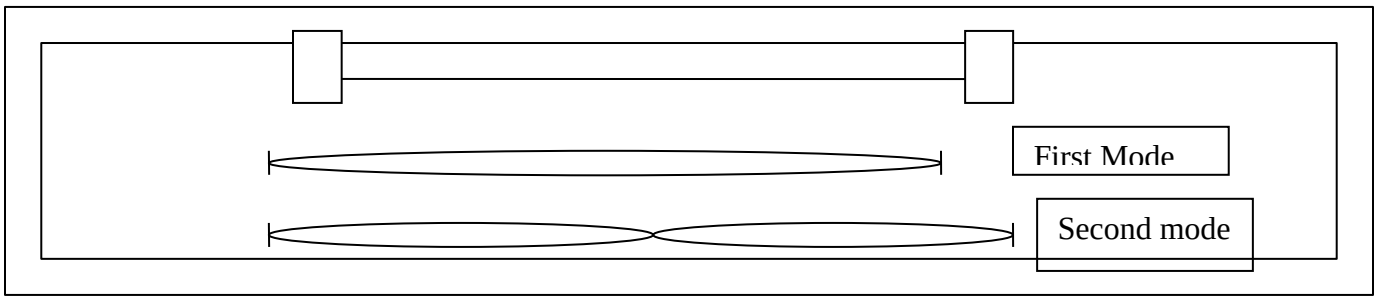


FIXED ENDS (i.e Fixed bearing or chucks)

The lowest critical speed is $f_n = 3.562\sqrt{\frac{gEI}{WL^4}}$

The highest Critical Speed is

$$f_n = \frac{\pi}{2}\left[n + \frac{1}{2}\right]^2 \sqrt{\frac{gEI}{WL^4}}$$ **Where n=1, 2, 3, 4……**

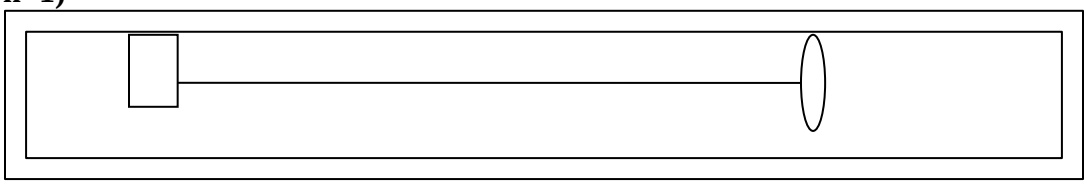


When Chuck at one end & free at the other

The lowest critical speed is $f_n = 0.565\sqrt{\frac{gEI}{WL^4}}$

The highest Critical Speed is

$f_n = \frac{\pi}{2}\left[n - \frac{1}{2}\right]^2 \sqrt{\frac{gEI}{WL^4}}$ **Where n=1, 2, 3, 4…… (Lowest speed corresponds to n=1)**



DESCRIPTION:

The setup consists of a sturdy m.s frame, variable speed motor bearing holders etc. Different bearings can be fitted in bearing block to have different end conditions. I.e.
(i) Both end fixed
(ii)Both end free (simply supported),
(iii)One end free & other end fixed
A variable speed motor is provided to drive the shaft along with the speed control unit. The setup is will designed to study the whirling of shaft.

UTILITIES REQUIRED:

Floor Area – 2 x 0.5 m^2
Electricity supply 230 V Ac
Tachometer

PROCEDURE:

1. Fix the bearings & shaft to be tested
2. Switch on the power supply
3. Start the motor & increase the speed of motor, with the help of varriac
4. Observe the modes of vibration
5. Take the rpm of shaft with the help of tachometer.
6. Switch of the power supply & fix the other shafts accordingly and takes the readings.

OBSERVATION:-

Data:
Elasticity of shaft = 189 x 10^9 N/m^2
g = 9.81 m/sec ^{2}
Mass of the shaft= -----Kg

When both ends are Simply Supported

OBSERVATION TABLE:

S.No	Shaft diameter(m)	Mode of vibration	Speed(RPM)

When One End Fixed & Other End Supported

OBSERVATION TABLE:

S.No	Shaft diameter(m)	Mode of vibration	Speed(RPM)

CALCULATION:

When Both end Supported

W= Weight / unit length

$= \frac{mg}{L}$, N/m

Moment of inertia (I)

$$I = \frac{\pi}{64} X \quad d^4 , \quad m^4$$

Frequency

Where n is the mode (i.e. n=1, 2, 3, 4…….)

Critical Speed

$N_C=f_n$ x 60 RPM

When One End Fixed & Other End Supported

W= Weight / unit length

$= \frac{mg}{L}$, N/m

Moment of inertia (I)

$$I = \frac{\pi}{64} X \quad d^4 , \quad m^4$$

The Critical Frequency is

$$f_n = \frac{\pi}{2}\left[n - \frac{1}{2}\right]^2 \sqrt{\frac{gEI}{WL^4}}$$ **Where n=1, 2, 3, 4……. (Lowest speed corresponds to n=1)]**

Critical Speed

$N_C=f_n$ x 60 RPM

Result:

APPARATUS USED: -. Arrangement of Gear train system.

THEORY: -

1. Definition of. Gear train
2. Classification of Gear train
3. Diagrams of different types of Gear train.
4. Working & Construction of different types of Gear train.
5. Advantages & Disadvantages of Gear train
6. Applications of Gear train .
7. Examples of Gear train

GEAR TRAIN:-

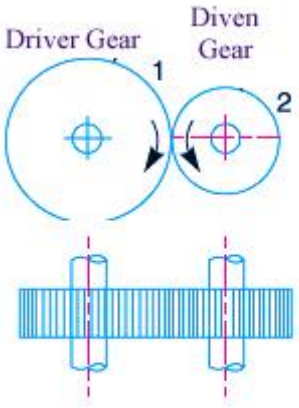
A gear train is a combination of gears used to transmit motion from one shaft to another. It becomes necessary when it is required to obtain large speed reduction With in a small space. The following are the main types of gear trains:

(i) Simple gear train
(ii) Compound gear train
(iii) Reverted gear train
(iv) Planetary gear train

SIMPLE GEAR TRAIN:-

A series of gears, capable of receiving and transmitting motion from one gear to another is called a simple gear train. In it, all the gear axes remain fixed relative to the frame and each gear is on a separate shaft.

Train Value = Number of teeth on driving gear / Number of teeth on driven gear

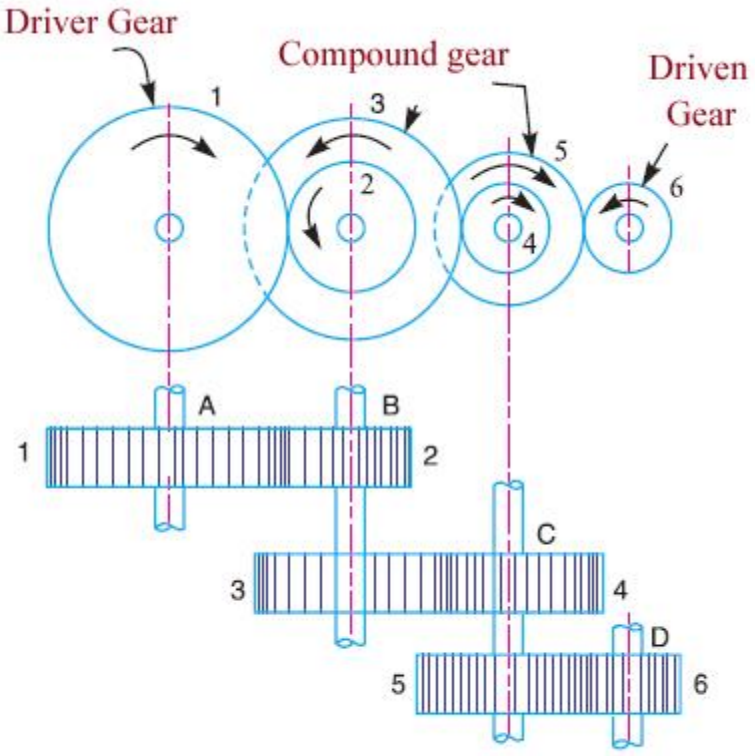


Simple Gear Train

COMPOUND GEAR TRAIN:-

When a series of gears are connected in such a way that two or more gears rotate about an axis with the same angular velocity, it is known as compound gear train. In this type, some of the intermediate shafts.

Train Value = Product of Number of teeth on driving gear / Product of Number of teeth on driven gear

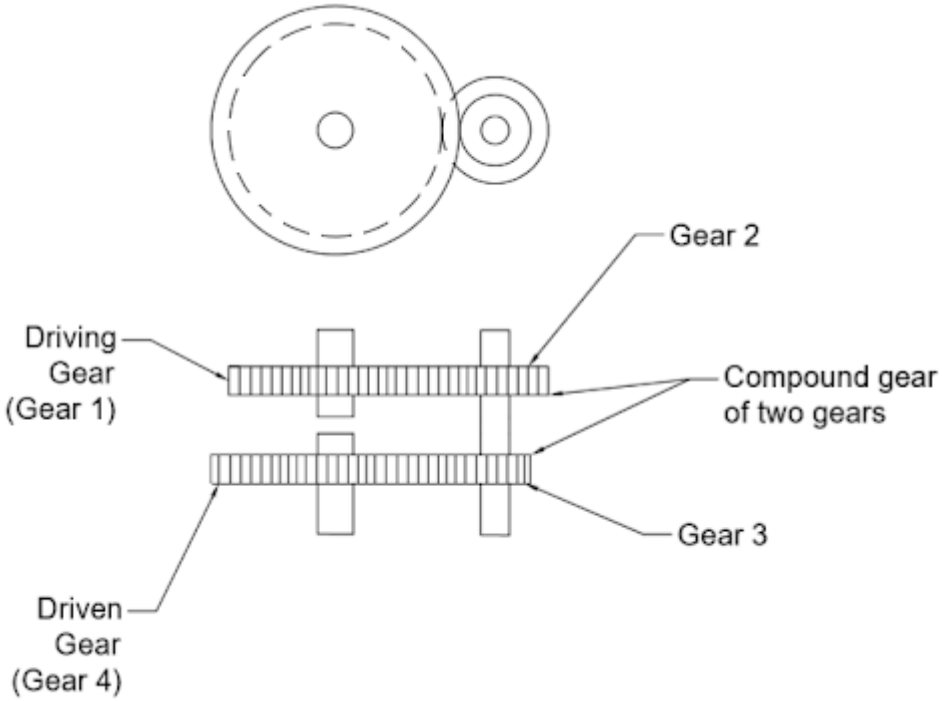


Compound Gear Train

REVERTED GEAR TRAIN :-

If the axes of the first and last wheels of a compound gear coincide; it is called a reverted gear train. Such an arrangement is used in clocks and in simple lathes where ‘back gear’ is used to give a slow speed to the chuck.

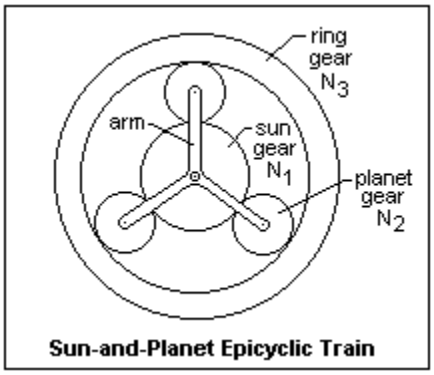
Train Value = Product of Number of teeth on driving gear / Product of Number of teeth on driven gear



PLANETARY OR EPICYCLIC GEAR TRAIN: -

When there exists a relative motion of
axis in gear train, it is called a planetary or an epicyclic gear train (or simply epicyclic gear or train). Thus in an 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 axis 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.

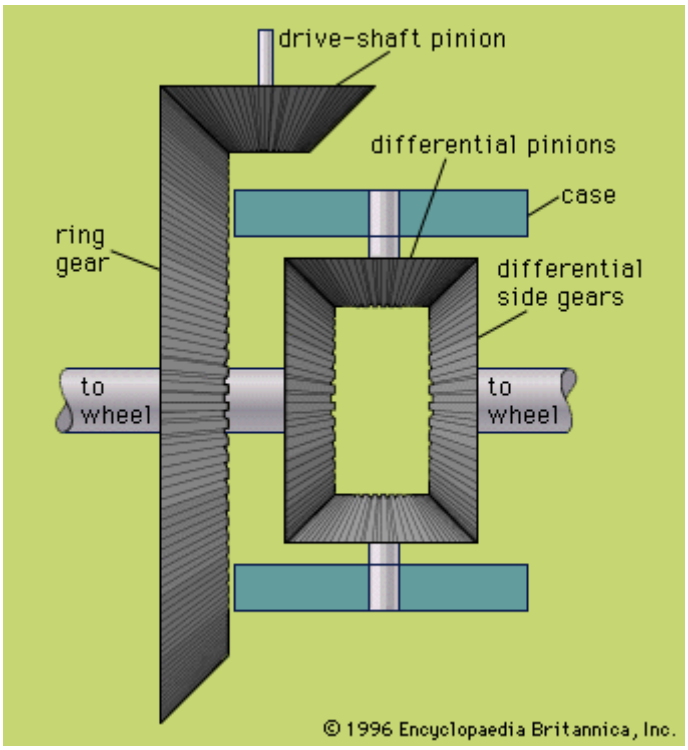


Sun-and-Planet Epicyclic Train

DIFFERENTIAL GEAR :-

When a vehicle takes a turn, the outer wheels must travel
farther than the inner wheels. In automobiles, the front wheels can rotate freely on their axis and thus can adapt themselves to the conditions. Both rear wheels are driven by the engine through gearing. Therefore, some sirt of automatic device is necessary so that the two rear

wheels are driven at slightly different speeds. This is accomplished by fitting a differential gear on the rear axle.



OBSERVATION & CONCLUSION: -

1. Comparison between simple, compound reverted, epicyclic and differential. Gear train.
2. To calculate the train value.
3. To calculate the speed of any gear.

GOVERNORS
Experiment -13

Aim: To study about different types of governors and its applications

Introduction:

Flywheel which minimizes fluctuations of speed within the cycle but it cannot minimize fluctuations due to load variation. This means flywheel does not exercise any control over mean speed of the engine. To minimize fluctuations in the mean speed which may occur due to load variation, governor is used. The governor has no influence over cyclic speed fluctuations but it controls the mean speed over a long period during which load on the engine may vary. The function of governor is to increase the supply of working fluid going to the prime mover when the load on the prime-mover increases and to decrease the supply when the load decreases so as to keep the speed of the prime-mover almost constant at different loads.

Example: when the load on an engine increases, its speed decreases, therefore it becomes necessary to increase the supply of working fluid. On the other hand, when the load on the engine decreases, its speed increases and hence less working fluid is required. When there is change in load, variation in speed also takes place then governor operates a regulatory control and adjusts the fuel supply to maintain the mean speed nearly constant. Therefore, the governor automatically regulates through linkages, the energy supply to the engine as demanded by variation of load so that the engine speed is maintained nearly constant.

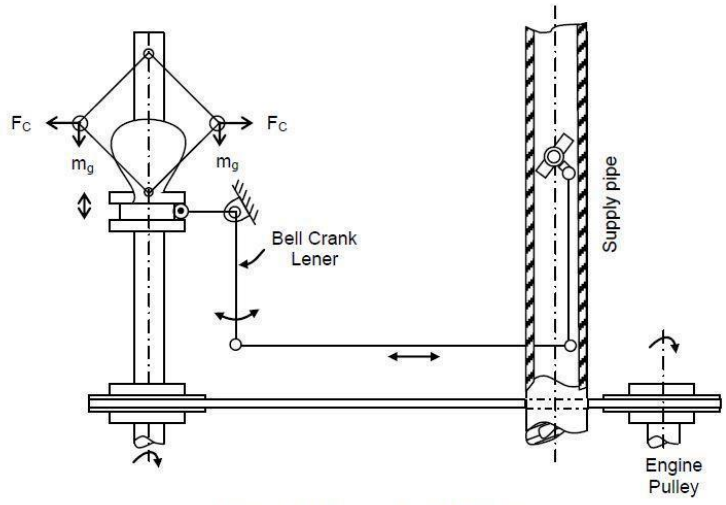


Figure 5.1 : Governor and Linkages

Types of Governors

The governors may, broadly, be classified as

1. Centrifugal governors
2. Inertia governors.
3. pickering governors.

Centrifugal Governors:

In these governors, the change in centrifugal forces of the rotating masses due to change in the speed of the engine is utilized for movement of the governor sleeve. One of this type of governors is shown in Figure. These governors are commonly used because of simplicity in operation.

Inertia and Flywheel Governors:

In these governors, the inertia forces caused by the angular acceleration of the engine shaft or flywheel by change in speed are utilized for the movement of the balls. The movement of the balls is due to the rate of change of speed instead of change in speed itself as in case of centrifugal governors. Thus, these governors are more sensitive than centrifugal governors.

Pickering Governors:

This type of governor is used for driving a gramophone. As compared to the centrifugal governors, the sleeve movement is very small. It controls the speed by dissipating the excess kinetic energy. It is very simple in construction and can be used for a small machine.

Types of Centrifugal Governors:

Depending on the construction these governors are of two types:

a) Gravity controlled centrifugal governors, and
b) Spring controlled centrifugal governors.

Gravity Controlled Centrifugal Governors:

In this type of governors there is gravity force due to weight on the sleeve or weight of sleeve itself which controls movement of the sleeve. These governors are comparatively larger in size.

Spring Controlled Centrifugal Governors:

In these governors, a helical spring or several springs are utilized to control the movement of sleeve or balls. These governors are comparatively smaller in size.

Gravity controlled centrifugal governors:

There are three commonly used gravity controlled centrifugal governors:

(a) Watt governor

(b) Porter governor

(c) Proell governor

Watt governor does not carry dead weight at the sleeve. Porter governor and proell governor have heavy dead weight at the sleeve.

In porter governor balls are placed at the junction of upper and lower arms.

In case of proell governor the balls are placed at the extension of lower arms. The sensitiveness of watt governor is poor at high speed and this limits its field of application.

Portergovernor is more sensitive than watt governor.

The proell governor is most sensitive out of these three.

Watt Governor:

This governor was used by James Watt in his steam engine. The spindle is driven by the output shaft of the prime mover. The balls are mounted at the junction of the two arms. The upper arms are connected to the spindle and lower arms are connected to the sleeve as shown in Figure.

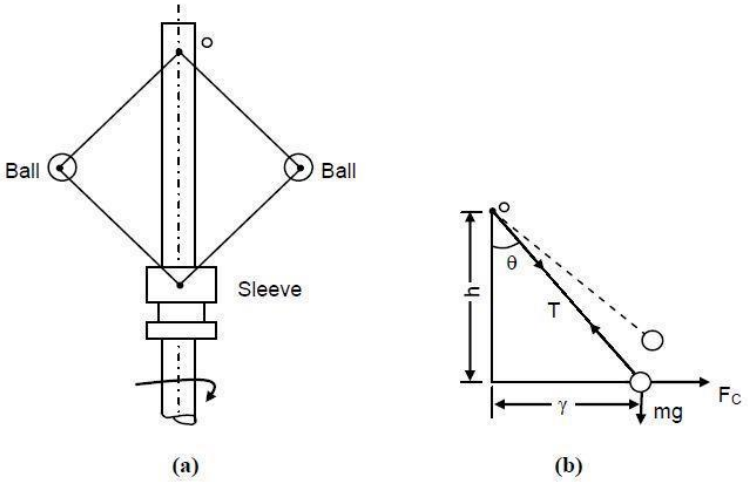


Figure 5.2 : Watt Governor

We ignore mass of the sleeve, upper and lower arms for simplicity of analysis. We can ignore the friction also. The ball is subjected to the three forces which are centrifugal force (*Fc*), weight (*mg*) and tension by upper arm (*T*). Taking moment about point *O* (intersection of arm and spindle axis), we get

$$F_C \, h - mg \, r = 0$$

Since, $F_C = mr\,\omega^2$

$\therefore$ $mr\,\omega^2 h - mg\,r = 0$

or $\omega^2 = \dfrac{g}{h}$

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

$\therefore$ $h = \dfrac{g \times 3600}{4\pi^2 N^2} = \dfrac{894.56}{N^2}$

where '*N*' is in rpm.

Figure shows a graph between height '*h*' and speed '*N*' in rpm. At high speed the change in height *h* is very small which indicates that the sensitiveness of the governor is very poor at high speeds because of flatness of the curve at higher speeds.

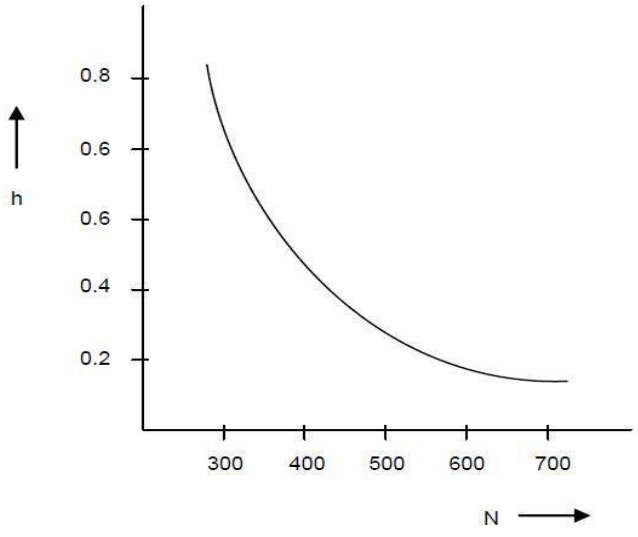


Figure 5.3 : Graph between Height and Speed

Porter Governor:

A schematic diagram of the porter governor is shown in Figure 5.4(a). There are two sets of arms. The top arms *OA* and *OB* connect balls to the hinge *O*. The hinge may be on the spindle or slightly away. The lower arms support dead weight and connect balls also. All of them rotate with the spindle. We can consider one-half of governor for equilibrium.

Let *w* be the weight of the ball,

*T*1 and *T*2 be tension in upper and lower arms, respectively,

Fc be the centrifugal force,

*r*be the radius of rotation of the ball from axis, and

I is the instantaneous centre of the lower arm.

Taking moment of all forces acting on the ball about I and neglecting friction at the sleeve, we get

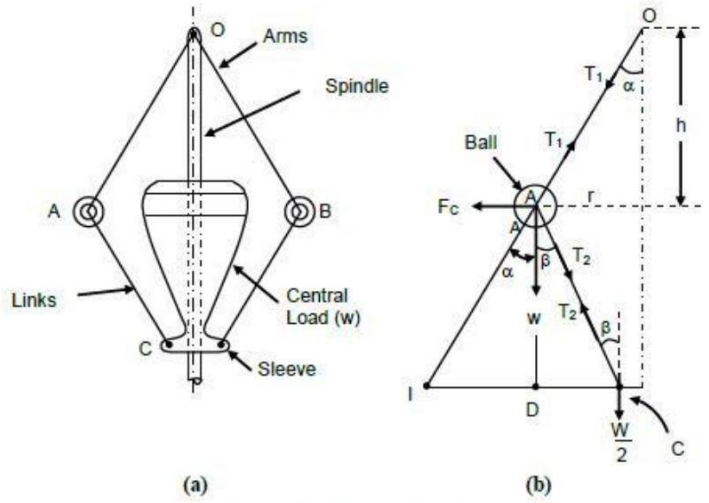


Figure 5.4 : Porter Governor

$$\omega^2 = \frac{g}{h}\left\{1 + \frac{(W \pm f)}{2w}(1 + K)\right\}$$

Spring controlled centrifugal governors:

In these governors springs are used to counteract the centrifugal force. They can be designed to operate at high speeds. They are comparatively smaller in size. Their speed range can be changed by changing the initial setting of the spring. They can work with inclined axis of rotation also. These governors may be very suitable for IC engines, etc.

The most commonly used spring controlled centrifugal governors are :

a) Hartnell governor
b) Wilson-Hartnell governor
c) Hartung governor

Hartnell Governor:

The Hartnell governor is shown in Figure 5.5. The two bell crank levers have been provided which can have rotating motion about fulcrums *O* and *O* one end of each bell crank lever carries a ball and a roller at the end of other arm. The rollers make contact with the sleeve. The frame is connected to the spindle. A helical spring is mounted around the spindle between frame and sleeve. With the rotation of the spindle, all these parts rotate.

With the increase of speed, the radius of rotation of the balls increases and the rollers lift the sleeve against the spring force. With the decrease in speed, the sleeve moves downwards. The movement of the sleeve are transferred to the throttle of the engine through linkages.

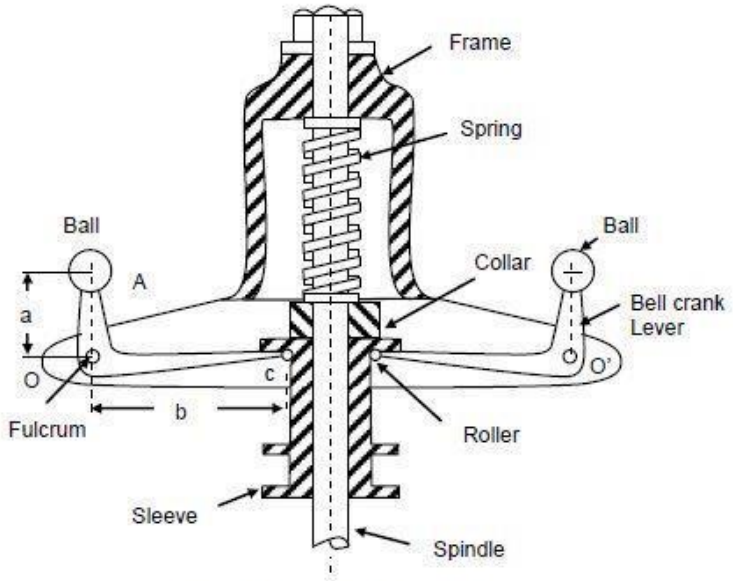


Figure 5.5 : Hartnell Governor

CHARACTERISTICS OF GOVERNORS:

Different governors can be compared on the basis of following characteristics:

Stability:

A governor is said to be stable when there is one radius of rotation of the balls for each speed which is within the speed range of the governor.

Governors Sensitiveness:

The sensitiveness can be defined under the two situations:

1. When the governor is considered as a single entity.
2. When the governor is fitted in the prime mover and it is treated as part of prime mover.

(a) A governor is said to be sensitive when there is larger displacement of the sleeve due to a fractional change in speed. Smaller the change in speed of the governor for a given displacement of the sleeve, the governor will be more sensitive.

(b) The smaller the change in speed from no load to the full load, the more sensitive the governor will be. According to this definition, the sensitiveness of the governor shall be determined by the ratio of speed range to the mean speed. The smaller the ratio more sensitive the governor will be

Where $N2 - N1$ = Speed range from no load to full load.

Hunting:

Whenever there is change in speed due to the change in load on the engine, the sleeve moves towards the new position but because of inertia if overshoots the desired position. Sleeve then moves back but again overshoots the desired position due to inertia. This results in setting up of oscillations in engine speed. If the frequency of fluctuations in engine speed coincides with the natural frequency of oscillations of the governor, this results in increase of amplitude of oscillations due to resonance. The governor, then, tends to intensity the speed variation instead of controlling it. This phenomenon is known as hunting of the governor. Higher the sensitiveness of the governor, the problem of hunting becomes more acute.

Conclusion:

STUDY OF DIFFERENT TYPES OF CAM AND FOLLOWER ARRANGEMENTS

Experiment -14

AIM: - To study various types of cam and follower arrangements.
APPARATUS USED: - Cam and follower arrangements.

THEORY: - 1. Definition of Cam & Follower.
2. Classification of Cam & Follower.
3. Diagrams of Cam & Follower.
4. Working & Construction of Cam & Follower.
5. Advantages & Disadvantages of Cam & Follower.
6. Applications of Cam & Follower. .
7. Examples of Cam & Follower.

CAM & FOLLOWER: - A cam is a mechanical member used to impart desired motion to a follower by direct contact. The cam may be rotating or reciprocating whereas the follower may be rotating, reciprocating or oscillating. A cam and the follower combination belongs to the category of higher pairs.
• A driver member known as the cam.
• A driven member called the follower

Types of Cams:
A. According to shape

(i) Wedge and flat Cams :- A wedge cam has a wedge W which, in general, has a translational motion. The follower F can either translate or oscilate.

(ii) Radial or Disc Cams :- A cam in which the follower moves radially from the centre of rotation of the cam is known as a radial or a disc cam.

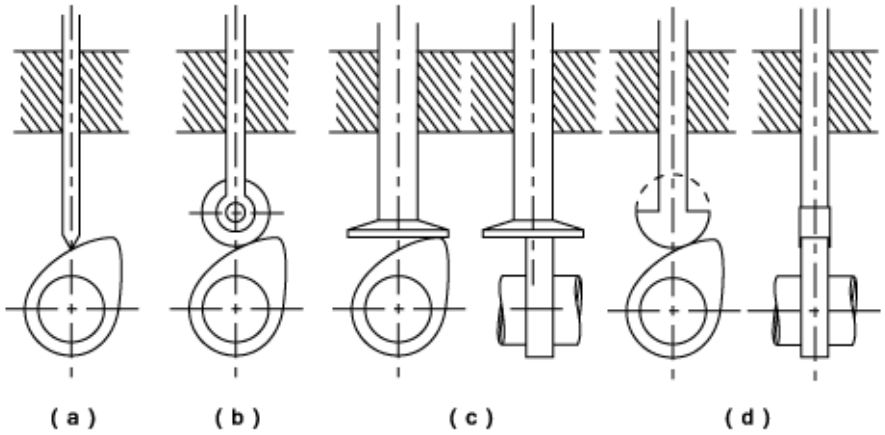
(iii) Spiral cams :- A spiral cam is a face cam in which a groove is cut in the form of a spiral as shown in fig. the spiral spiral groove consists of teeth which mesh with a pin gear follower.

(iv) Cylindrical cams :- In a cylindrical cam, a cylinder which has a circumferential contour cut in the surface, rotate about its axis.

(v) Conjugate cams :- A conjugate cam is a double – disc cam, the two discs being keyed together and are in constant touch with the two rollers of a follower. It is used for low noise, high speed and dynamic loads.

(vi) Globoidal cams :- A globoidal cam can have two types of surfaces, convex or concave. A circumferential contour is cut on the surface of rotation of the cam to impart motion to the follower which has an oscillatory motion.

(vii) Spherical cams :- In a spherical cam, the follower oscillates about an axis perpendicular to the axis of rotation of the cam.



B. According to Follower Movement

(i) Rise-Return-Rise (R-R-R) :- In this, there is alternate rise and return of the follower with no periods of dwells. Its use is very limited in the industry. The follower has a linear or an angular displacement.

(ii) Dwell-Rise-Return-Dwell (D-R-R-D) :- In such a type of cam, there is rise and return of the follower after a dwell. This type is used more frequently than the RR- R type of cam

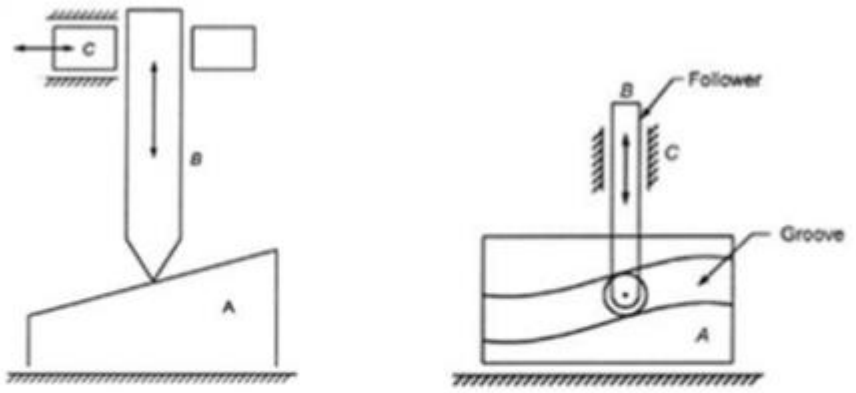
(iii) Dwell-Rise-Dwell-Return (D-R-D-R) :- It is most widely used type of cam. The dwelling of the cam is followed by rise and dwell and subsequently by return and dwell as shown in fig.

C. According to Manner of Constraint of the Follower

(i) Pre-loaded Spring Cam
(ii) Positive-Drive Cam
(iii) Gravity Cam

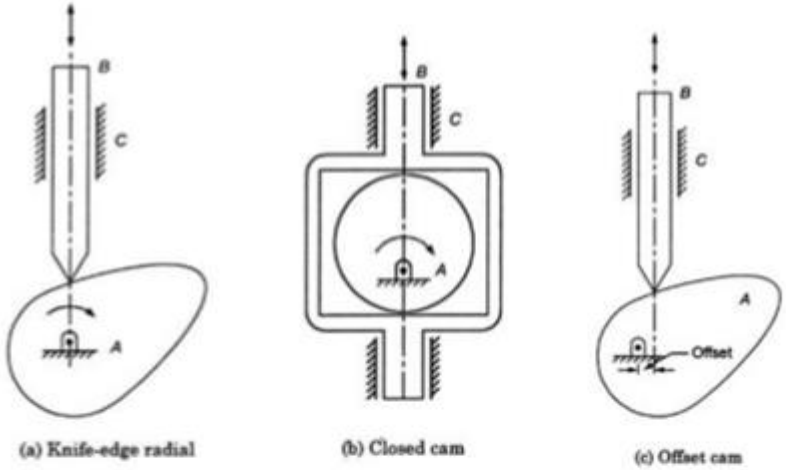
Types of Followers :-
A. According to shape
(i) Knife-edge Follower
(ii) Roller Follower
(iii) Mushroom Follower
B. According to Movement
(i) Reciprocating Follower
(ii) Oscillating Follower

C. According to Locating of Line of Movement
(i) Radial Follower
(ii) Offset Foolower



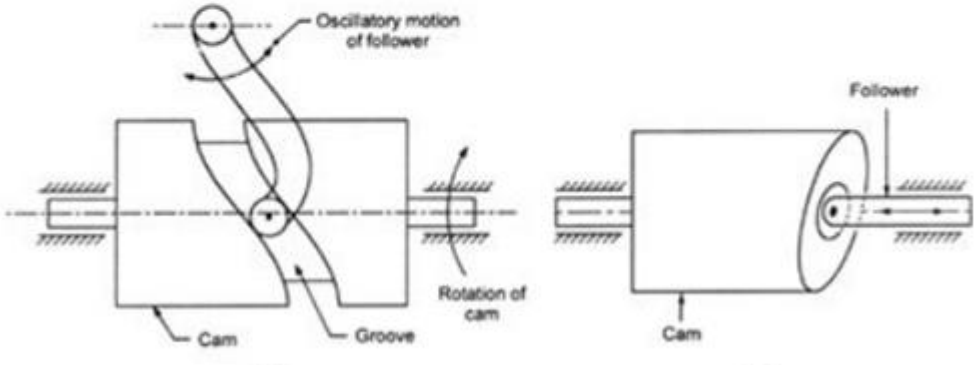
(c) (d)

Fig.8.1 Wedge and flat cams



(a) Knife-edge radial (b) Closed cam (c) Offset cam

Fig.8.2 Radial and offset cams



(a) (b)

Fig.8.3 Cylindrical cams

OBSERVATION & CONCLUSION: -

1. Comparison between Cam & Follower.
2. Type of Motion to be named.