Rajasthan Institute of Engineering & Technology, Jaipur

B.Tech. II mid Term Examination

Session: 2017-18

IVSEMMECHANICAL ENGINEERING

KINEMATICS OF MACHINES 4ME1A

Time: -2 Hrs. **SET-(A)** **[Maximum Marks: -20]**

Instructions to Candidates: -

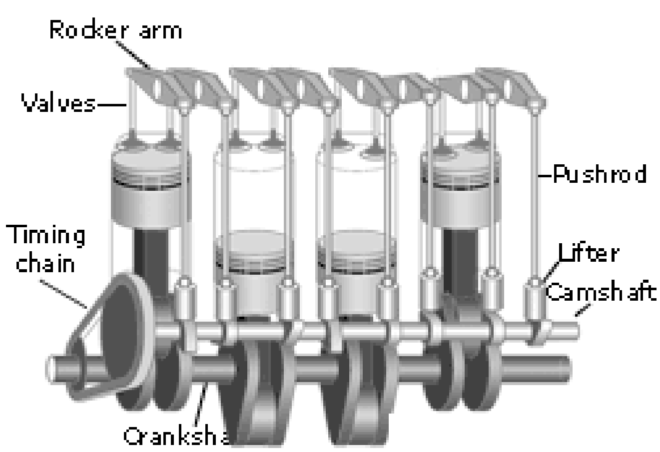
1. **No provision for Supplementary answer book**

Q.01 Explain the Overhead Valve Mechanism for Automobile Vehicle with neat sketch. 05

Ans.01 An overhead valve engine (OHV engine) is an engine in which the valves are placed in the cylinder head. This was an improvement over the older flathead engine, where the valves were placed in the cylinder block next to the piston. Overhead camshaft (OHC) engines, while still overhead valve by definition, are usually categorized apart from other OHV engines. The type of valve typically used are Poppet valves.

In a piston engine configuration where the valves are overhead but the camshaft is not, informally called pushrod engine or I-head engine, the camshaft is placed within the cylinder block (usually beside and slightly above the crankshaft in a straight engine or directly above the crankshaft in the V of a V engine), and uses pushrods or rods to actuate rocker arms above the cylinder head to actuate the valves. Lifters or *tappets* are located in the engine block between the camshaft and pushrods.[1] By contrast, overhead camshaft design avoids the use of pushrods by putting the camshaft directly above the valves in the cylinder head, thus simplifying the valvetrain.

OHV means OverHead Valve - an engine design where the camshaft is installed inside the engine block and valves are operated through lifters, pushrods and rocker arms. For this reason, an OHV engine is also known as a "Pushrod" engine. The OHV design has been successfully used for decades.

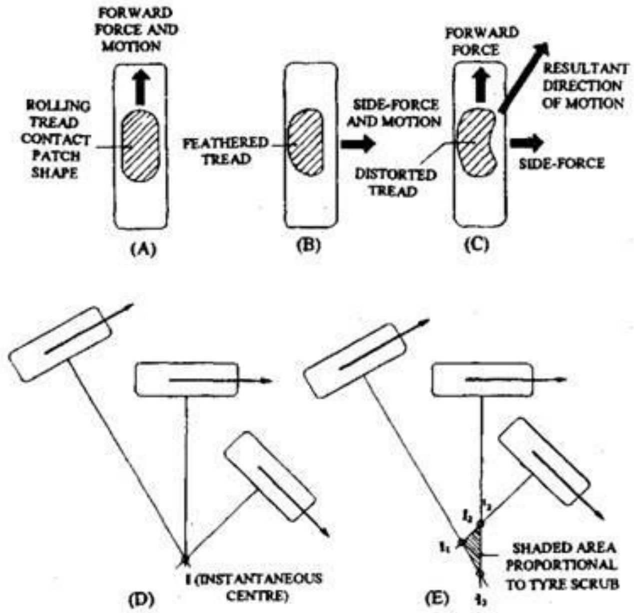


Or

Q.01. Explain with neat sketch the Correct or True steering of automobile, How can you identify the Ackerman & Davis Steering Gear Mechanism. 05

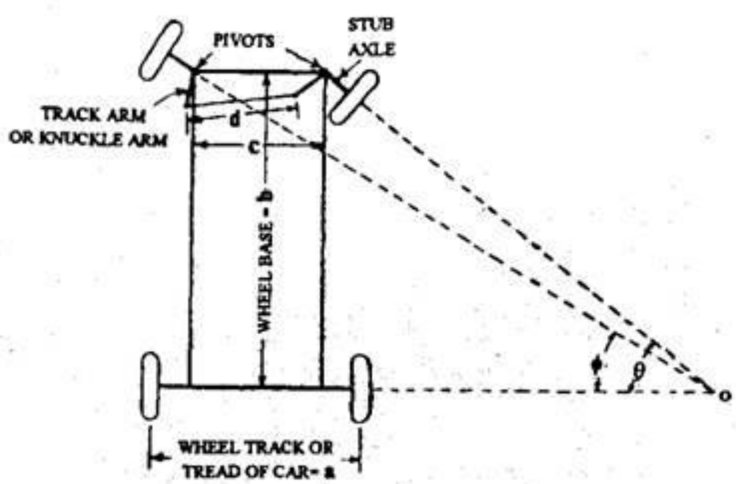
Ans.01 Condition for True Steering

True rolling occurs only when the direction of motion of the vehicle is perpendicular to the wheel axis i.e. the wheel is subjected to forward force. When wheel is subjected to side force that acts parallel to the wheel axis, a true scrub action is produced When the wheel is subjected to both forward and side forces, the movement is compounded of true rolling and lateral distortion This condition occurs when the wheels are being steered, i.e. the direction of motion is neither parallel nor perpendicular to the axis of rotation.On a circular path, true rolling condition occurs when the projected axes of several wheels all moving in different curved paths intersect at a single point called the instantaneous centre When these projected axes do not intersect at a single point, a degree of tyre scrub results



Road-wheel and tyre rolling conditions. A. True-rolling. B. True scrub. C. Tyre steer. D. Condition for true rolling. E. Condition for tyre scrub.

Whenever a vehicle takes a turn, the front wheels must turn in a definite manner both in relation to each other and to the axis of the rear wheels so that the lateral slip may be avoided



and true rolling for all the wheels is obtained. For this, as explained above, all the wheels must always rotate about the instantaneous centre. Since the rear wheels have a common and fixed axis, it is quite obvious that this common centre, 0, would lie somewhere on its extension .

From the figure, $\cot \phi = \frac{c+x}{b} = \frac{c}{b} + \frac{x}{b} = \frac{c}{b} + \cot \theta$

where, θ = Angle of inside lock.

ϕ = Angle of outside lock.

a = Wheel track, also known as tread of vehicle.

b = Wheel base of the vehicle.

c = Distance between the pivot centres.

d = Length of track rod.

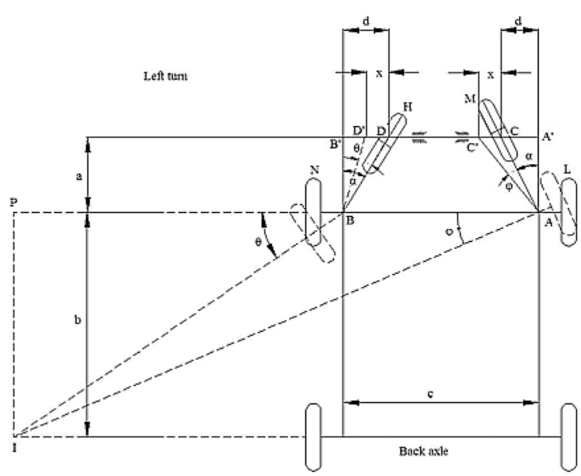
and x = Projected distances from instantaneous centre to the inner pivot centre.

Therefore, $\cot \phi - \cot \theta = \frac{c}{b}$

This equation gives the fundamental condition to be satisfied by all types of steering mechanism if true rolling for all the wheels is to be obtained avoiding any lateral slip. The steering linkage used in the vehicles must maintain the proper angles with the wheels when taking a turn. But practically it is not possible to maintain absolutely correct angles for the wheels for all turning angles.

Davis Steering Gear Mechanism

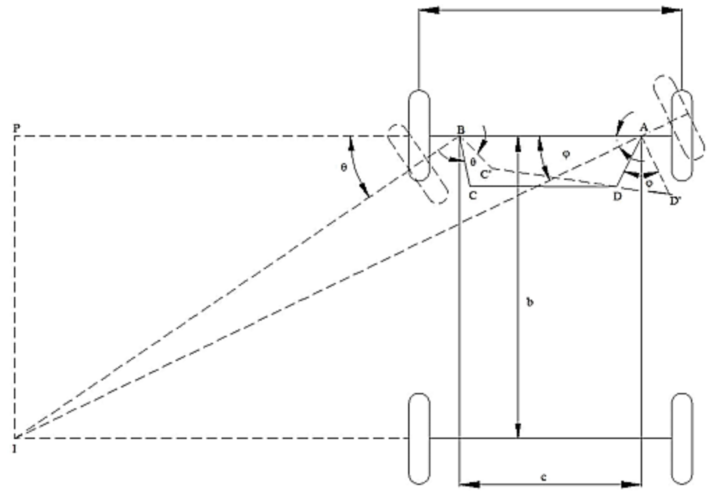
The slotted links AM and BH are attached to the front wheel axle, which turn on pivots A and B respectively. The rod CD is constrained to move in the direction of its length, by the sliding members at P and Q. These constraints are connected to the slotted link AM and BH by a sliding and a turning pair at each end. The steering is affected by moving CD to the right or left of its nominal position.



Ackerman steering gear mechanism

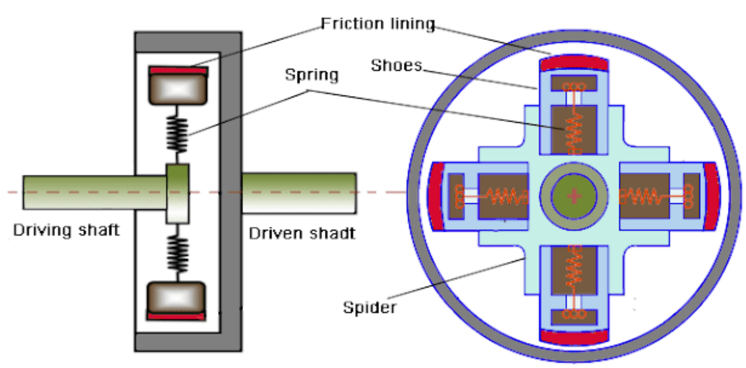
The Ackerman steering gear consists of turning pairs rather than sliding pairs. The whole of the mechanism is placed on the back of the front wheels. In Ackerman steering gear, the mechanism ABCD is a four bar crank chain. The shorter links BC and AD are equally inclined to the longitudinal axis of the vehicle. For the correct steering the following three positions are obtained.

1. When the vehicle moves along a straight path, the longer links AB and CD are parallel and the shorter links BC and AD are equally inclined to the longitudinal axis of the vehicle.
2. When the vehicle is moving to the left, the lines of the front wheel axle intersect on the back wheel axle at I for correct steering.



Q.02 Explain the working of Centrifugal Clutch with neat sketch. 05

Ans.02 The centrifugal clutch consists of a number of shoes or friction pads arranged radially symmetrical position inside the rim. It can slide along the guides integral with the boss on the driving shaft. The shoes are held against boss by using a spring that exerts a radially inward force. As the inner hub rotates, the weight of the shoe causes a radially outward force known as centrifugal force. This force depends on the weight of the shoe and the speed at which it rotates.



Centrifugal Clutch

At low speed, the centrifugal force also low, the shoes remain in the same position. As speed increases, the centrifugal force also increases, when centrifugal force becomes equal to spring force the shoes start floating. When the driver rotates fast enough the centrifugal force exceeds the spring force the shoes moves outward. At a certain speed, it gets contact with the inner surface of the drum and torque is transmitted. As the load increases, speed decreases; the shoes return to their original position and clutch Gets disengaged.

Advantage

1. Simple and inexpensive and need little maintenance.
2. The centrifugal clutch is automatic any kind of control mechanism is not necessary.
3. They help to prevent the engine from stalling.
4. The engagement speed can precisely control by selecting spring.

Disadvantage

1. Loss of power due to friction and slipping.
2. This type of clutch not appropriate for the high amount of torque, the shoes will slip at the heavy loaded condition.
3. They engage at full or near-full power, shoes get heated very quickly may cause overheating.

Or

Q.02 Derive the equation of total frictional torque of single plate clutch (considering the uniform Pressure)

Ans.02 **Single plate clutch (considering the uniform Pressure)**

Now consider two friction surfaces, maintained in contact by an axial thrust W, as shown in

Let T = Torque transmitted by the clutch,

p = Intensity of axial pressure with which the contact surfaces are held together,

r_1 and r_2 = External and internal radii of friction faces, and

μ = Coefficient of friction.

Consider an elementary ring of radius r and thickness dr

We know that area of contact surface or friction surface,

$$= 2\pi\, r.dr$$

$\therefore$ Normal or axial force on the ring,

$$\delta W = \text{Pressure} \times \text{Area} = p \times 2\pi\, r.dr$$

and the frictional force on the ring acting tangentially at radius r,

$$F_r = \mu.\delta W = \mu.p \times 2\pi\, r.dr$$

$\therefore$ Frictional torque acting on the ring,

$$T_r = F_r \times r = \mu.p \times 2\pi\, r.dr \times r = 2\pi \times \mu\, .p.r^2\, dr$$

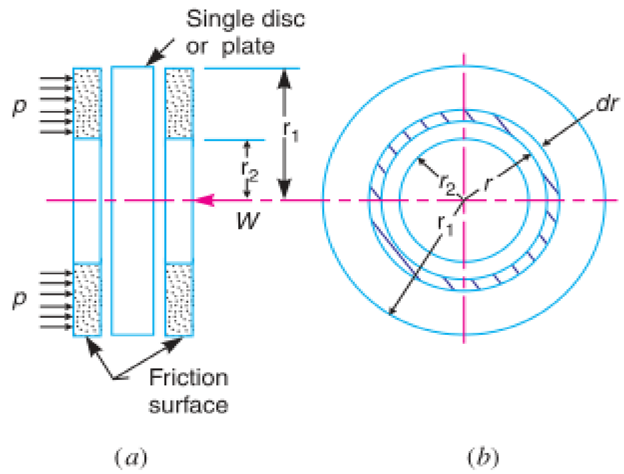


Fig. 10.22. Forces on a single disc or plate clutch.

We shall now consider the following two cases :

1. When there is a uniform pressure, and

2. When there is a uniform wear.

1. *Considering uniform pressure*

When the pressure is uniformly distributed over the entire area of the friction face, then the intensity of pressure,

$$p = \frac{W}{\pi[(r_1)^2 - (r_2)^2]} \quad \text{...}(i)$$

where W = Axial thrust with which the contact or friction surfaces are held together.

We have discussed above that the frictional torque on the elementary ring of radius r and thickness dr is

$$T_r = 2\,\pi\,\mu.p.r^2\,dr$$

Integrating this equation within the limits from r_2 to r_1 for the total frictional torque.

$\therefore$ Total frictional torque acting on the friction surface or on the clutch,

$$T = \int_{r_1}^{r_2} 2\pi\mu.p.r^2.dr = 2\,\pi\mu\,p\left[\frac{r_3}{3}\right]_{r_2}^{r_1} = 2\pi\mu\,p\left[\frac{(r_1)^3 - (r_2)^3}{3}\right]$$

Substituting the value of p from equation (i),

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

$$= \frac{2}{3} \times \mu.W\left[\frac{(r_1)^3 - (r_2)^3}{(r_1)^2 - (r_2)^2}\right] = \mu.W.R$$

where R = Mean radius of friction surface

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

Q.03 Derive the Equation of Braking of Four wheels Vehicle when Brakes are applied in front wheels only. 05

Ans.03 Braking of Four wheels Vehicle when Brakes are applied in front wheels

Now, consider a vehicle moving up an inclined plane,

Let α = Angle of inclination of the plane to the horizontal,

m = Mass of the vehicle in kg (such that its weight is $m.g$ newtons),

h = Height of the C.G. of the vehicle above the road surface in metres,

x = Perpendicular distance of C.G. from the rear axle in metres,

L = Distance between the centres of the rear and front wheels (also called wheel base) of the vehicle in metres,

R_A = Total normal reaction between the ground and the front wheels in newtons,

R_B = Total normal reaction between the ground and the rear wheels in newtons,

μ = Coefficient of friction between the tyres and road surface, and

a = Retardation of the vehicle in m/s^2.

2. *When the brakes are applied to front wheels only*

It is a very rare way of braking the vehicle, in which the braking force acts at the front wheels only.

Let F_A = Total braking force (in newtons) acting at the front wheels due to the application of brakes. Its maximum value is $\mu.R_A$.

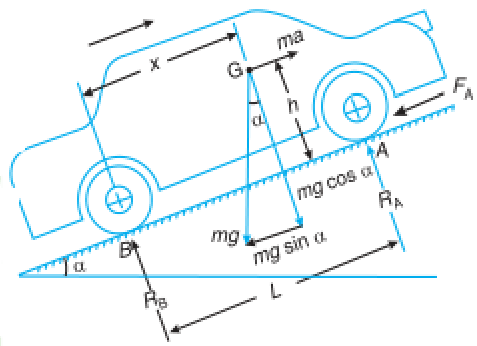
The various forces acting on the vehicle are shown in Fig. 19.28.

Resolving the forces parallel to the plane,

$$F_A + m.g \sin\alpha = m.a \qquad \ldots (i)$$

Resolving the forces perpendicular to the plane,

$$R_A + R_B = m.g \cos\alpha \qquad \ldots (ii)$$



. Motion of the vehicle up the inclined plane and brakes are applied to front wheels only.

Taking moments about G, the centre of gravity of the vehicle,

$$F_A \times h + R_B \times x = R_A (L - x)$$

Substituting the value of $F_A = \mu.R_A$ and $R_B = m.g \cos\alpha - R_A$ [from equation *(ii)*] in the above expression, we have

$$\mu.R_A \times h + (m.g \cos\alpha - R_A)\, x = R_A (L - x)$$

$$\mu.R_A \times h + m.g \cos\alpha \times x = R_A \times L$$

$$\therefore \qquad R_A = \frac{m.g \cos\alpha \times x}{L - \mu.h}$$

and

$$R_B = m.g \cos\alpha - R_A = m.g \cos\alpha - \frac{m.g \cos\alpha \times x}{L - \mu.h}$$

$$= m.g \cos\alpha \left(1 - \frac{x}{L - \mu.h}\right) = m.g \cos\alpha \left(\frac{L - \mu.h - x}{L - \mu.h}\right)$$

We know from equation *(i)*,

$$a = \frac{F_A + m.g \sin\alpha}{m} = \frac{\mu.R_A + m.g \sin\alpha}{m}$$

$$= \frac{\mu.m.g \cos\alpha \times x}{(L - \mu.h)m} + \frac{m.g \sin\alpha}{m} \qquad \text{. . . (Substituting the value of } R_A\text{)}$$

$$= \frac{\mu.g \cos\alpha \times x}{L - \mu.h} + g \sin\alpha$$

Or

Q.03 Derive the Equation of Braking torque on the Drum for Simple Band Brake. 05

Ans.03 **Simple Band Brake**

A band brake consists of a flexible band of leather, one or more ropes,or a steel lined with friction material, which embraces a part of the circumference of the drum.

simple band brake in which one end of the band is attached to a fixed pin or fulcrum of the lever while the other end is attached to the lever at a distance b from the fulcrum.

When a force P is applied to the lever at C, the lever turns about the fulcrum pin O and tightens the band on the drum and hence the brakes are applied. The friction between the band and the drum provides the braking force. The force P on the lever at C may be determined as discussed below :

Let T_1 = Tension in the tight side of the band,

T_2 = Tension in the slack side of the band,

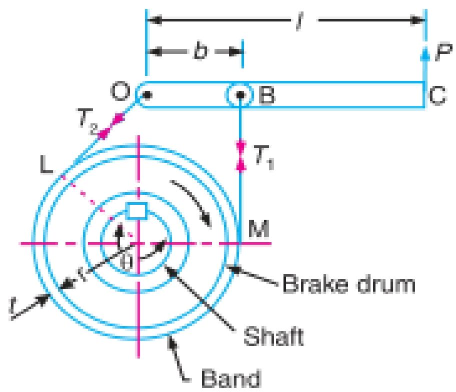
θ = Angle of lap (or embrace) of the band on the drum,

μ = Coefficient of friction between the band and the drum,

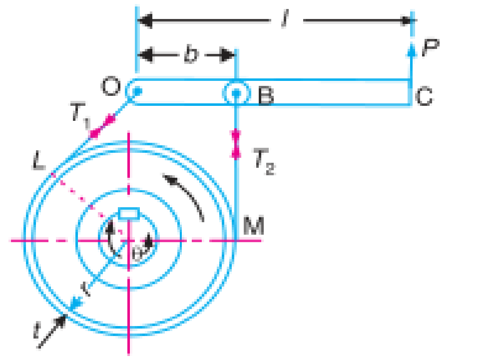
r = Radius of the drum,

t = Thickness of the band, and

r_e = Effective radius of the drum = $r + \dfrac{t}{2}$



(*a*) Clockwise rotation of drum.



(*b*) Anticlockwise rotation of drum.

Fig. 19.11. Simple band brake.

We know that limiting ratio of the tensions is given by the relation,

$$\frac{T_1}{T_2} = e^{\mu\theta} \qquad \text{or} \qquad 2.3\log\left(\frac{T_1}{T_2}\right) = \mu.\theta$$

and braking force on the drum = $T_1 - T_2$

$\therefore$ Braking torque on the drum,

$$T_B = (T_1 - T_2)\, r \qquad \text{. . . (Neglecting thickness of band)}$$

$$= (T_1 - T_2)\, r_e \qquad \text{. . . (Considering thickness of band)}$$

Now considering the equilibrium of the lever *OBC*. It may be noted that when the drum rotates in the clockwise direction, as shown in Fig. 19.11 (*a*), the end of the band attached to the fulcrum *O* will be slack with tension T_2 and end of the band attached to *B* will be tight with tension T_1. On the other hand, when the drum rotates in the anticlockwise direction, as shown in Fig. 19.11 (*b*), the tensions in the band will reverse, *i.e.* the end of the band attached to the fulcrum *O* will be tight with tension T_1 and the end of the band attached to *B* will be slack with tension T_2. Now taking moments about the fulcrum *O*, we have

$$P.l = T_1.b \qquad \text{. . . (For clockwise rotation of the drum)}$$

and

$$P.l = T_2.b \qquad \text{. . . (For anticlockwise rotation of the drum)}$$

Q.04 Explain the working of Throneycraft transmission Dynamometer(Froude) & write down the formula of BHP. 05

Ans.04 **Throneycraft transmission Dynamometer(Froude)**

When the belt is transmitting power from one pulley to another, the tangential effort on the driven pulley is equal to the difference between the tensions in the tight and slack sides of the belt. A belt dynamometer is introduced to measure directly the difference between the tensions of the belt, while it is running.

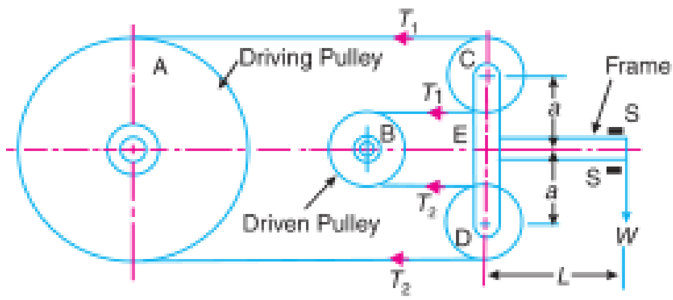


Fig. 19.34. Froude or Throneycroft transmission dynamometer.

A belt transmission dynamometer, as shown in Fig. 19.34, is called a Froude or Throneycroft transmission dynamometer. It consists of a pulley *A* (called driving pulley) which is rigidly fixed to the shaft of an engine whose power is required to be measured. There is another pulley *B* (called driven pulley) mounted on another shaft to which the power from pulley *A* is transmitted. The pulleys *A* and *B* are connected by means of a continuous belt passing round the two loose pulleys *C* and *D* which are mounted on a *T*-shaped frame. The frame is pivoted at *E* and its movement is controlled by two stops *S*,*S*. Since the tension in the tight side of the belt (T_1) is greater than the tension in the slack side of the belt (T_2), therefore the total force acting on the pulley *C* (*i.e.* $2T_1$) is greater than the total force acting on the pulley *D* (*i.e.* $2T_2$). It is thus obvious that the frame causes movement about *E* in the anticlockwise direction. In order to balance it, a weight *W* is applied at a distance *L* from *E* on the frame as shown in Fig. 19.34.

Now taking moments about the pivot *E*, neglecting friction,

$$2T_1 \times a = 2T_2 \times a + W.L \quad \text{or} \quad T_1 - T_2 = \frac{W.L}{2a}$$

Let D = diameter of the pulley *A* in metres, and

N = Speed of the engine shaft in r.p.m.

$\therefore$ Work done in one revolution $= (T_1 - T_2)\,\pi\, D$ N-m

and workdone per minute $= (T_1 - T_2)\,\pi\, DN$ N-m

$\therefore$ Brake power of the engine, $\text{B.P.} = \dfrac{(T_1 - T_2)\,\pi\, DN}{60}$ watts

Or

Q.04 In a Laboratory Experiment:-

Diameter of the flywheel (Drum) & Rope is 1.2 meter & 12.5 mm, Speed 200 rpm, Dead load on the brake 600 N, spring balance reading 150 N. Calculate the brake power of the engine. 05

Ans.04

Given : D = 1.2 m ; d = 12.5 mm = 0.0125 m ; N = 200 r.p.m ; W = 600 N ; S = 150 N

We know that brake power of the engine,

$$\text{B.P.} = \frac{(W - S)\,\pi\,(D + d)N}{60} = \frac{(600 - 150)\,\pi\,(1.2 + 0.0125)200}{60} = 5715 \text{ W}$$

$= 5.715$ kW **Ans.**

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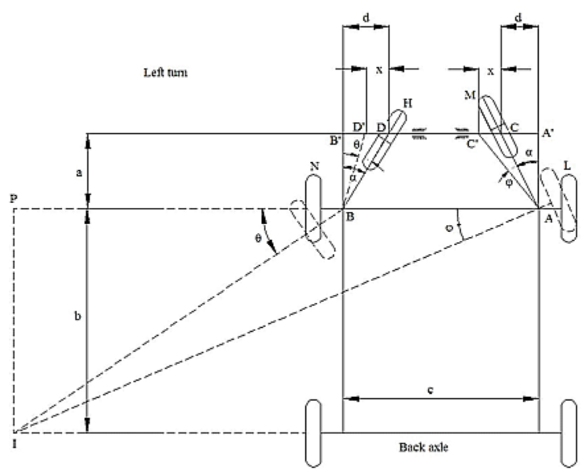
Q.01.Differentiate the Davis and Ackerman steering mechanism with their neat sketch 05

Ans.01

Davis Steering Gear Mechanism

The slotted links AM and BH are attached to the front wheel axle, which turn on pivots A and B respectively. The rod CD is constrained to move in the direction of its length, by the sliding members at P and Q. These constraints are connected to the slotted link AM and BH by a sliding and a turning pair at each end. The steering is affected by moving CD to the right or left of its nominal position.



Ackerman steering gear mechanism

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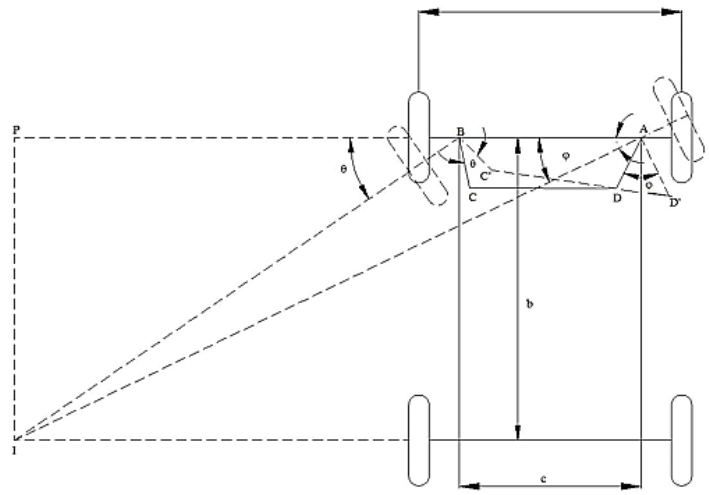


Or

Q.01 Write short Notes on:-

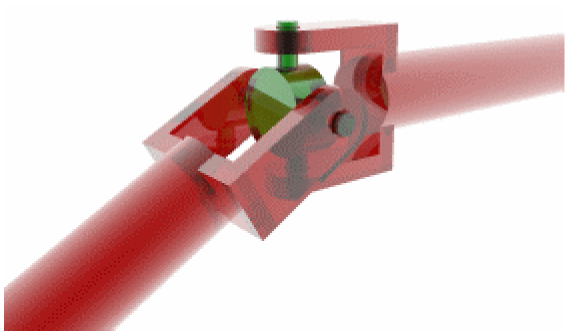
(a) Hooks Joint

(b) Trifler suspension 05

Ans.01

(a) A **universal joint** (**universal coupling**, **U-joint**, **Cardan joint**, **Spicer** or **Hardy Spicer joint**, or **Hooke's joint**)

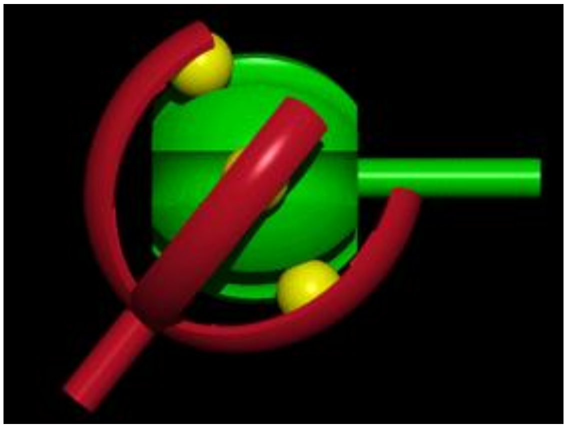
It is a joint or coupling connecting rigid rods whose axes are inclined to each other, and is commonly used in shafts that transmit rotary motion. It consists of a pair of hinges located close together, oriented at 90° to each other, connected by a cross shaft. The universal joint is not a constant-velocity joint.



Constant-velocity joints (also known as **homokinetic** or **CV joints**)

It allow a drive shaft to transmit power through a variable angle, at constant rotational speed, without an appreciable increase in friction or play. They are mainly used in front wheel drivevehicles, and many modern rear wheel drive cars with independent rear suspension typically use CV joints at the ends of the rear axle halfshafts and increasingly use them on the drive shafts.

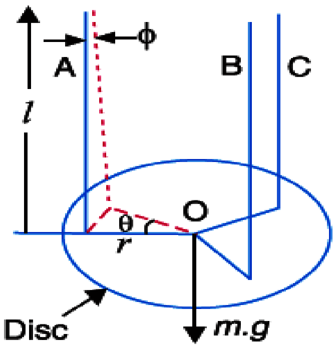
Constant-velocity joints are protected by a rubber boot, a CV gaiter, usually filled with molybdenum disulfide grease. Cracks and splits in the boot will allow contaminants in, which would cause the joint to wear quickly as grease leaks out. This way the friction parts don't get proper lubrication and get damaged due to minor particles that get in, while water causes metal components to rust and corrode. Wear of the boot often takes the form of small cracks, which appear closer to the wheel,because it is the wheel that produces vibration and up and down motions.[1] Cracks and tears in the areas closer to the axle are usually caused by external factors, such as packed snow, stones or uneven rocky off-road paths.



(b) Trifler suspension(**Torsional Pendulum)**

The trifilar suspension is used to determine the moments of inertia of a body about an axis passing through its mass center. The apparatus consists of a circular platform suspended by three equi-spaced wires of equal length. The body under consideration is placed with its mass centre exactly in the middle of the circular platform. The platform is given a small circular displacement about the vertical axis through its center, and is released. The periodic time of the subsequent motion is obtained by measuring the time taken to complete a definite number of oscillations. Then from the formula the moment of inertia of the body can be calculated.

In Addition the moment of inertia of a connecting rod was found. The apparatus was a knife edge to suspend the rod and record the oscillations



Trifilar suspension.

- The body (say a disc or flywheel) whose moment of inertia is to be determined is suspended by three long flexible wires A, B and C, as shown in Fig.
- When the body is twisted about its axis through a small angle θ and then released, it will oscillate with simple harmonic motion.
- Let,

  m = Mass of the body,

  W = Weight of the body in Newton = m.g,

  k_G = Radius of gyration about an axis through the centre of gravity,

  I = Mass moment of inertia of the disc about an axis through O and perpendicular to it = $m.k^2$,

  l = Length of each wire,

  r = Distance of each wire from the axis of the disc,

  θ = Small angular displacement of the disc,

  φ = Corresponding angular displacements of the wire, and

  α = Angular acceleration towards the equilibrium position.

Then, for small displacements,

$$r.\theta = l.\phi \quad \text{or} \quad \phi = r.\theta/l$$

Since the three wires are attached symmetrically with respect to the axis, therefore the tension in each wire will be one-third of the weight of the body.

$\therefore$ Tension in each wire $= m.g/3$

Component of the tension in each wire perpendicular to r

$$= \frac{m.g.\sin\phi}{3} = \frac{m.g.\phi}{3} = \frac{m.g.r.\theta}{3l} \quad \ldots (\because \phi \text{ is a small angle, and } \phi = r.\theta/l)$$

$\therefore$ Torque applied to each wire to restore the body to its initial equilibrium position *i.e.* restoring torque

$$= \frac{m.g.r.\theta}{3l} \times r = \frac{m.g.r^2.\theta}{3l}$$

Total restoring torque applied to three wires,

$$T = 3 \times \frac{m.g.r^2.\theta}{3l} = \frac{m.g.r^2.\theta}{l} \quad \ldots (i)$$

We know that disturbing torque

$$= I.\alpha = m.k_G^2.\alpha \quad \ldots (ii)$$

Equating equations (*i*) and (*ii*),

$$\frac{m.g.r^2.\theta}{l} = m.k_G^2.\alpha \quad \text{or} \quad \frac{\theta}{\alpha} = \frac{l.k_G^2}{g.r^2}$$

i.e. $$\frac{\text{Angular displacement}}{\text{Angular acceleration}} = \frac{l.k_G^2}{g.r^2}$$

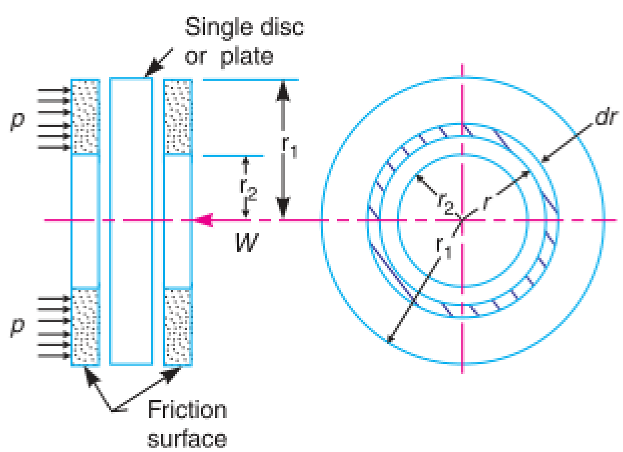
We know that periodic time,

$$t_p = 2\pi\sqrt{\frac{\text{Angular displacement}}{\text{Angular acceleration}}} = 2\pi\sqrt{\frac{l.k_G^2}{g.r^2}} = \frac{2\pi k_G}{r}\sqrt{\frac{l}{g}}$$

and frequency, $$n = \frac{1}{t_p} = \frac{r}{2\pi k_G}\sqrt{\frac{g}{l}}$$

Q.02 Derive the equation of total frictional torque of single plate clutch (considering the uniform wear). 05

Ans.02 **Single plate clutch (considering the uniform Wear)**



Now consider two friction surfaces, maintained in contact by an axial thrust W

Let T = Torque transmitted by the clutch,

p = Intensity of axial pressure with which the contact surfaces are held together,

r_1 and r_2 = External and internal radii of friction faces, and

μ = Coefficient of friction.

Consider an elementary ring of radius r and thickness dr as shown

... p be the normal intensity of pressure at a distance r from the axis of the clutch. Since the intensity of pressure varies inversely with the distance, therefore

$$p.r. = C \text{ (a constant)} \quad \text{or} \quad p = C/r \qquad ...(i)$$

and the normal force on the ring,

$$\delta W = p.2\pi r.dr = \frac{C}{r} \times 2\pi C.dr = 2\pi C.dr$$

$\therefore$ Total force acting on the friction surface,

$$W = \int_{r_2}^{r_1} 2\pi C\,dr = 2\pi C\left[r\right]_{r_2}^{r_1} = 2\pi C(r_1 - r_2)$$

or

$$C = \frac{W}{2\pi(r_1 - r_2)}$$

We know that the frictional torque acting on the ring,

$$T_r = 2\pi\mu.p\,r^2.dr = 2\pi\mu \times \frac{C}{r} \times r^2.dr = 2\pi\mu.C.r.dr$$

$$...(\because \quad p = C/r)$$

$\therefore$ Total frictional torque on the friction surface,

$$T = \int_{r_2}^{r_1} 2\pi\mu.C.r.dr = 2\pi\mu.C\left[\frac{r^2}{2}\right]_{r_2}^{r_1} = 2\pi\mu.C\left[\frac{(r_1)^2 - (r_2)^2}{2}\right]$$

$$= \pi\mu.C[(r_1)^2 - (r_2)^2] = \pi\mu \times \frac{W}{2\pi(r_1 - r_2)}\left[(r_1)^2 - (r_2)^2\right]$$

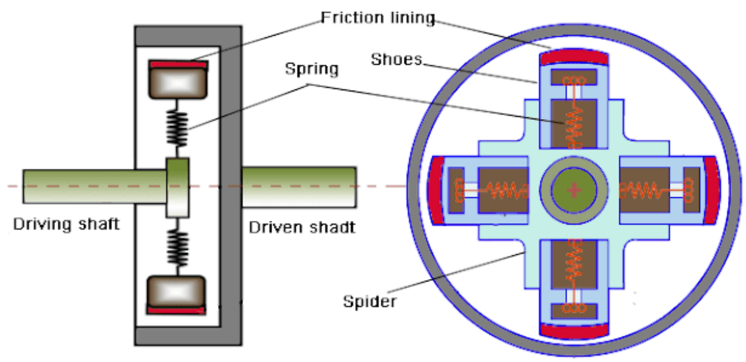
$$= \frac{1}{2} \times \mu.W(r_1 + r_2) = \mu.W.R$$

where R = Mean radius of the friction surface $= \dfrac{r_1 + r_2}{2}$

Or

Q.02 Explain the working of Centrifugal Clutch with neat sketch. 05

Ans.02 The centrifugal clutch consists of a number of shoes or friction pads arranged radially symmetrical position inside the rim. It can slide along the guides integral with the boss on the driving shaft. The shoes are held against boss by using a spring that exerts a radially inward force. As the inner hub rotates, the weight of the shoe causes a radially outward force known as centrifugal force. This force depends on the weight of the shoe and the speed at which it rotates.



Centrifugal Clutch

At low speed, the centrifugal force also low, the shoes remain in the same position. As speed increases, the centrifugal force also increases, when centrifugal force becomes equal to spring force the shoes start floating. When the driver rotates fast enough the centrifugal force exceeds the spring force the shoes moves outward. At a certain speed, it gets contact with the inner surface of the drum and torque is transmitted. As the load increases, speed decreases; the shoes return to their original position and clutch Gets disengaged.

Advantage

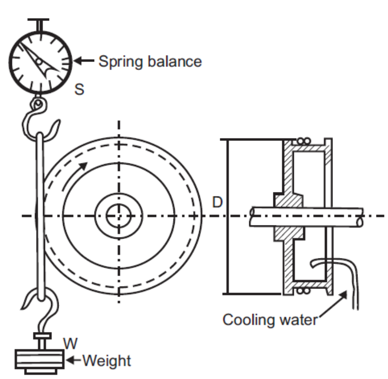
5. Simple and inexpensive and need little maintenance.
6. The centrifugal clutch is automatic any kind of control mechanism is not necessary.
7. They help to prevent the engine from stalling.
8. The engagement speed can precisely control by selecting spring.

Disadvantage

4. Loss of power due to friction and slipping.
5. This type of clutch not appropriate for the high amount of torque, the shoes will slip at the heavy loaded condition.
6. They engage at full or near-full power, shoes get heated very quickly may cause overheating.

Q.03 Explain the working of Rope Brake Dynamometer & write down the formula of BHP. 05

Ans.03



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. 19.32. 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.

$\therefore$ Brake power of the engine,

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

Or

Q.03 In a Laboratory Experiment:-

Diameter of the flywheel (Drum) & Rope is 1.2 meter & 12.5 mm, Speed 200 rpm, Dead load on the brake 600 N, spring balance reading 150 N. Calculate the brake power of the engine. 05

Ans.03

Given : D = 1.2 m ; d = 12.5 mm = 0.0125 m ; N = 200 r.p.m ; W = 600 N ; S = 150 N

We know that brake power of the engine,

$$\text{B.P.} = \frac{(W-S)\,\pi\,(D+d)N}{60} = \frac{(600-150)\,\pi\,(1.2+0.0125)200}{60} = 5715 \text{ W}$$

$$= 5.715 \text{ kW}$$ **Ans.**

Q.04 Derive the Equation of Braking of Four wheels Vehicle when Brakes are applied to all the four wheels. 05

Ans.04 **Braking of Four wheels Vehicle**

3. *When the brakes are applied to all the four wheels*

This is the most common way of braking the vehicle, in which the braking force acts on both the rear and front wheels.

Let F_A = Braking force provided by the front wheels = $\mu.R_A$, and

F_B = Braking force provided by the rear wheels = $\mu.R_B$.

A little consideration will show that when the brakes are applied to all the four wheels, the braking distance (*i.e.* the distance in which the vehicle is brought to rest after applying the brakes) will be the least. It is due to this reason that the brakes are applied to all the four wheels.

The various forces acting on the vehicle

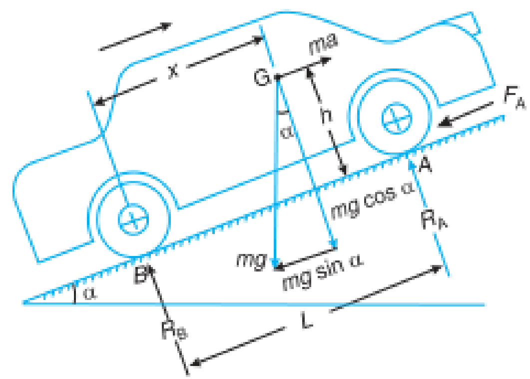


Fig. 19.29. Motion of the vehicle up the inclined plane and the brakes are applied to all the four wheels.

Resolving the forces parallel to the plane,

$$F_A + F_B + m.g\sin\alpha = m.a \qquad \ldots (i)$$

Resolving the forces perpendicular to the plane,

$$R_A + R_B = m.g\cos\alpha \qquad \ldots (ii)$$

Taking moments about G, the centre of gravity of the vehicle,

$$(F_A + F_B)h + R_B \times x = R_A(L - x) \qquad \ldots (iii)$$

Substituting the value of $F_A = \mu.R_A, F_B = \mu.R_B$ and $R_B = m.g\cos\alpha - R_A$ [From equation (*ii*)] in the above expression,

$$\mu(R_A + R_B)h + (m.g\cos\alpha - R_A)x = R_A(L - x)$$

$$\mu(R_A + m.g\cos\alpha - R_A)h + (m.g\cos\alpha - R_A)x = R_A(L - x)$$

$$\mu.m.g\cos\alpha \times h + m.g\cos\alpha \times x = R_A \times L$$

$$\therefore \qquad R_A = \frac{m.g\cos\alpha(\mu.h + x)}{L}$$

and

$$R_B = m.g\cos\alpha - R_A = m.g\cos\alpha - \frac{mg\cos\alpha(\mu.h + x)}{L}$$

$$= m.g\cos\alpha\left[1 - \frac{\mu.h + x}{L}\right] = m.g\cos\alpha\left(\frac{L - \mu.h - x}{L}\right)$$

Now from equation (*i*),

$$\mu.R_A + \mu R_B + m.g\sin\alpha = m.a$$

$$\mu(R_A + R_B) + m.g\sin\alpha = m.a$$

$$\mu.m.g.\cos\alpha + m.g\sin\alpha = m.a \qquad \ldots \text{[From equation (ii)]}$$

$$\therefore \qquad a = g(\mu.\cos\alpha + \sin\alpha)$$

or

Q.04 The wheel base of a car is 3 meter & its C G is 1.2 meter ahead the rear axle & 0.75 m above the ground level. The coefficient of friction between the wheels & the roads is 0.5 Determine the maximum deceleration of the car when it moves on level roads if the braking force on all the wheels is the same & no wheels slips occurs. 05

A.04 Solution

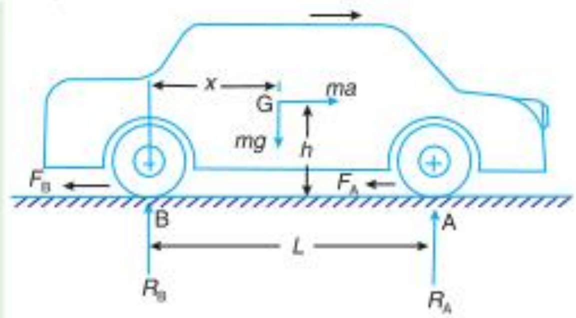
. Given : $L = 3$ m ; $x = 1.2$ m ; $h = 0.75$ m ; $\mu = 0.5$

Let a = Maximum deceleration of the car,

m = Mass of the car,

F_A and F_B = Braking forces at the front and rear wheels respectively, and

R_A and R_B = Normal reactions at the front and rear wheels respectively.



The various forces acting on the car are shown in Fig. 19.30.

We shall consider the following two cases:

(a) When the slipping is imminent at the rear wheels

We know that when the brakes are applied to all the four wheels and the vehicle moves on a level road, then

$$R_B = m.g\left(\frac{L - \mu.h - x}{L}\right) = m \times 9.81\left(\frac{3 - 0.5 \times 0.75 - 1.2}{3}\right) = 4.66\ m\ \text{N}$$

and $\quad F_A + F_B = m.a \quad$ or $\quad 2\mu.\ R_B = m.a \quad \ldots (\because\ F_B = F_A \text{ and } F_B = \mu.R_B)$

$\therefore \quad 2 \times 0.5 \times 4.66\ m = m.a \quad$ or $\quad a = 4.66\ \text{m/s}^2$

(b) When the slipping is imminent at the front wheels

We know that when the brakes are applied to all the four wheels and the vehicle moves on a level road, then

$$R_A = \frac{m.g(\mu.h + x)}{L} = \frac{m \times 9.81(0.5 \times 0.75 + 1.2)}{3} = 5.15\ m\ \text{N}$$

and $\quad F_A + F_B = m.a \quad$ or $\quad 2\mu\ .\ R_A = m.a \quad \ldots (\because\ F_A = F_B \text{ and } F_A = \mu\ .\ R_A)$

$\therefore \quad 2 \times 0.5 \times 5.15\ m = m.a \quad$ or $\quad a = 5.15\ \text{m/s}^2$

Hence the maximum possible deceleration is $4.66\ \text{m/s}^2$ and slipping would occur first at the rear wheels.