LAB MANUAL FOR THE MACHINE DESIGN-II LAB

(Code No. : ETME 352)

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EXPERIMENT NO.1

AIM: To design a block brake with short shoe.

THEORY:

A brake is defined as a mechanical device that is used to absorb the energy possessed by a moving system or mechanism by means of friction. The primary purpose of brake is to slowdown or completely stops the motion of a moving system, such as rotating drum, machine or vehicle. It is also used to hold the parts of the system in position at rest.

TYPES OF BRAKES:

(1.) **Mechanical brakes:**

These are operated by mechanical means such as levers, spring &pedals. Type of block brake is Block Brake, Internal Brake or External Shoe Brake, disc brake, band brake.

(2.) **Hydraulic & pneumatic brakes:**

These are operated by fluid pressure such as oil or air pressure.

(3.) **Electrical brakes:**

These are operated by magnetic forces.

ENERGY EQUATIONS:

The braking torque depend upon the amount of energy absorb by the brake. For a translating body, the kinetic energy (K.E) absorbed by brake during

- Braking Period: $K.E=1/2\ m\ (v_1^2-v_2^2)$
- For a rotating body : $K.E=1/2\ I(w_1^2-w_2^2)$

- In hoist application:
  The potential energy(P.E) stored by the brake during braking period =mgh
  Where h=distance by which mass m falls during braking period.

$$E=M_T\theta$$

Where E= total energy absorbed by the brake.

M_T= braking torque

ϴ= angle through which brake drum rotate during the braking period.

DESIGN OF A BLOCK BRAKE WITH SHORT SHOE:

A block brake consists of a simple block, which is pressed against the rotating drum by mean of lever. The friction between the block & brake drum causes the retardation of drum..

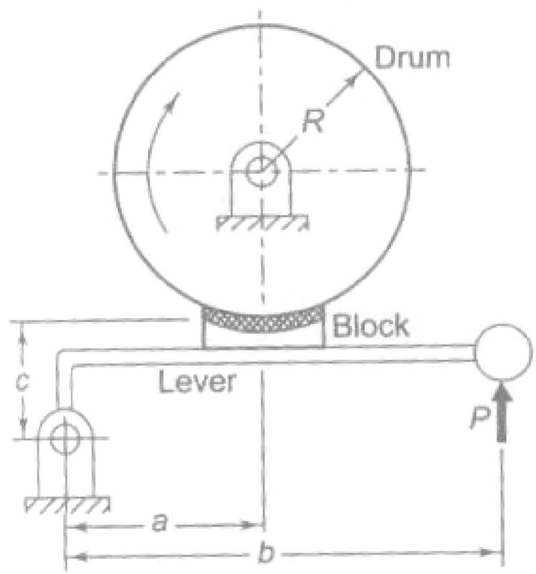

Fig. 1.1 *Block Brake*

The analysis is based on following assumption:

1. The block is rigidly attached to the lever.
2. The angle of contact between the block and brake drum is small resulting in a uniform pressure distribution. Considering the forces acting on the brake drum, $M_T= \mu NR$, where R=radius of brake drum.

The dimensions of block are determined by the following expression: $N=pl\omega$

Where p=permissible pressure between block & brake drum. l &ω= length and width of the block respectively .

Generally, drum dia./4< ω <drum dia./2

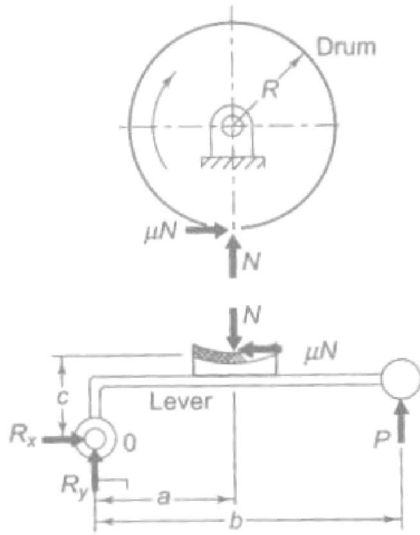

Fig. 1.2 *Free-body Diagram (Clockwise Rotation)*

Considering the equilibrium of forces in vertical and horizontal direction: Rx=μN

Taking moment of forces acting on the lever about hinge point 0.

P x b-N x a +μ N x C=0

P= (a-μC) x N/b

Case 1: a>μC: partially self energising brake.

Case 2: a=μC: self locking brake

Case 3: a<μC: uncontrolled braking and grabbing condition.

Viva questions

1. What is the major drawback of single shoe brake?
2. How this drawback is overcome?

Practice problem

1. A single shoe brake with a torque capacity of 250Nm is there. The brake drum rotates at 100 rpm and coefficient of friction is 0.35. Calculate
   i) The actuating force and the hinge pin reaction for clockwise rotation of drum
   ii) The actuating force and the hinge pin reaction for anti clockwise rotation of drum
   iii) The rate of heat generated during the braking period

EXPERIMENT NO.2

Aim: To design a double or shoe brake.

THEORY:

When a single block brake is applied to a rolling wheel and additional load is thrown on the shaft bearing due to normal force (R_N).This produces bending the shaft. In order to overcome this drawback, a double block or shoe brake is used. It consists of two brake block applied at the opposite ends of a diameter of wheels which reduce the unbalanced force on the shaft.

Kinetic Energy (K.E) $=(Q\ (V_2^1-V^2{}_2))/(2g)$

Potential Energy (P.E) $=(Q\ (V_1+V_2)\ t)/2$

Where V_1 and V_2 are the speed of load before and after the brake is applied on m/sec and Q is the load.

Brake drum must absorb K.E of all rotating parts, so it would be

$E_r = WK^2\ (w_1^2-w_2^2)/2g$

Where w_1 and w_2 and angular velocity of rotating parts before and after the brake is applied in rod/sec and E_r is the rotational energy.

In case load is stopped completely w_2 and v_2 =0

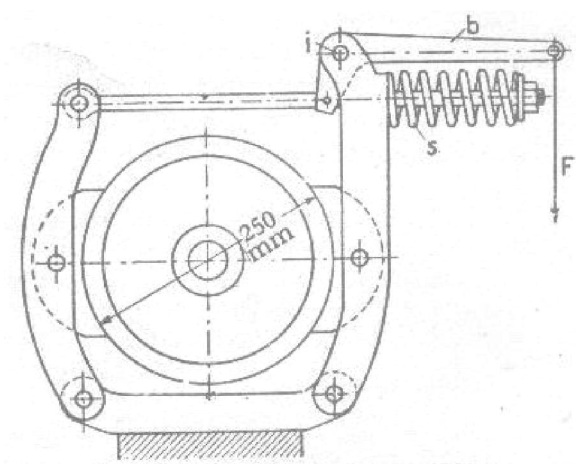

FIG. 2.1 . Double-block crane brake.

$E_t = E_r$+K.E+P.E

E_t =2(E_Er+K.E+P.E)/Πd (n_1+n_2) t

Where d is the diameter of brake drum and n_1 and n_2 is the speed of brake shear in rev.per sec.

W_K=F_tπd (n_1+n_2) t /2

A_{SF} = N/P where N is the normal reaction is the pressure and A_{SF} is the projected area normal to the direction N.

For moulded wooden or asbestos block, PV≤1 for continuous operation in lowering the load.

For Intermittent operation with comparatively longer period of the rest PV≤ 2 and

PV ≤ 3 for continuous operation

L=A_{SF}/b when F_t = μN and here b is the width of shoe

L=A_{SF}/2b in case of double shoe where b is the width of shoe.

Viva questions

1. Which brake is used for heavy load application?
2. What do you mean by self actuating and self energizing brakes?

Practice problem:

1. Determine a) the capacity and b) the main dimensions of a double block brake for the following conditions. The brake sheave is mounted on the drum shaft. The hoist with its load weighs 27kN and moves downwards with a velocity of 1.2 m/s. Pitch diameter of hoist drum is 1. M. The hoist must be stopped in a distance of 3 m, the kE of drum may be neglected. Assume brake dia =800mm

EXPERIMENT NO.3

AIM: To design a cone brake.

THEORY: A semi diagrammatic drawing of a cone brake as shown in fig. The outer cone O may from a part of the hoist drum can be attached to it while the inner cone is splined to shaft which can rotate is only one direction, being prevents from running in the opposite direction by a ratchet and pavel.

FORCE ANALYSIS:

The magnitude of the force F at the end of the operating level may be computed as follows:

The axial force F_a supplied at the cone surface can be revolved into a normal force N and a radial force R.

Normal force: $N=F_a/\sin\alpha$

Radial force is $R=F_a/\tan\alpha$

In a conical surface the radial force balance each other. The tangential force or braking force F_t is equal to the normal force multiplied by the friction coefficient.

$F_t = F_{N} = f. F_a/\sin\alpha$

The braking force torque is then,

$T= f. F_a D/2 \sin\alpha$ Where D is the mean diameter of cone.

Owing to the leverage

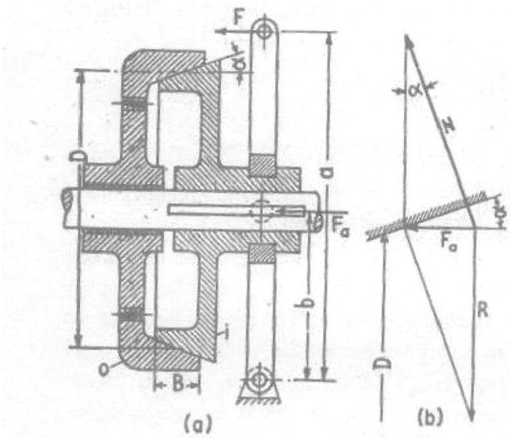

FIG. 3.1 . Cone brake

F_a= Fa/h

The relation between operating force F and braking force F_t

$F=F_t.b\ \sin\alpha/fa$

The area A of the contact surface can be determined by the relation

$A=(\Pi DB)/\cos\alpha$

Average pressure between contact surfaces is

$P=N/A = F_a/(\Pi DB\tan\alpha)$

The female cone is usually made of cast iron. The inner cone is also cast iron but it is often lined with wood or asbestos block in order to increase. The angle α is made from 10 to 18 degree. The axial width B is made from 0.12D to 0.22D

Viva Questions

1. Compare between cone brake and disk brake.
2. Why semicone angle be restricted to 12.5^0?

Practice Problem

A cone brake is mounted o a shaft which transmits 4.5kW at 225 rpm. The small diameter of the cone is 225mm, and the cone face is 50mm wide $\alpha=15^0$; the coefficient of friction is 0.33 and the lever dimensions are a=0.6 and b= 125mm. Find (a) The effort F necessary to stop the shaft and the specific normal pressure on cone surfaces.

EXPERIMENT NO.4

AIM: To design a differential band brake.

THEORY:

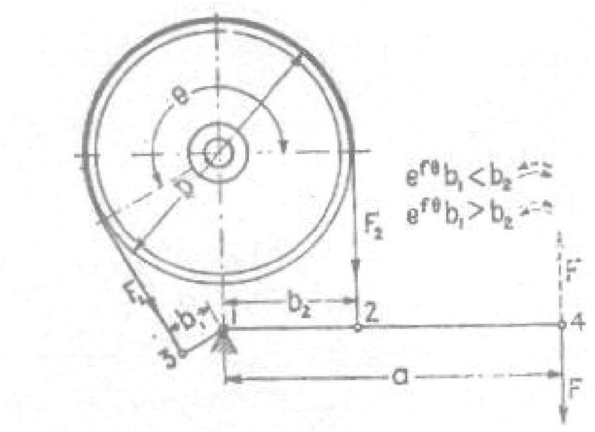

FIG. 4.1 Differential brake.

The band brake in which one of the band passes through the fulcrum is called simple band brake, while the band brakes in which neither of the band end passes through the fulcrum is called differential band brake.

$F1/F2=e^{f\Theta}$

$F1-F2=Ft$

Eliminating F2 from above equations

$F1= Ft\, e^{f\Theta}/ (e^{f\Theta}-1)$ (i)

$F2 = Ft/ (^{ef\Theta}-1)$ (ii)

Considering the operating lever as a free body and taking moments about fulcrum and assuming clockwise rotation

$Fa+F1b1=F2b2$

Substituting in equation (i) and (ii)

$F=Ft\ ((b2- e^{f\Theta}b1)/ ((e^{f\Theta}-1)a)$

The condition represented in figure requires that b2> efΘb1

$b2/b1>e^{f\Theta}$

if $b2/b1=e^{f\Theta}$

then F=0 and brake becomes self –locking and is undesirable and even dangerous.

$b2/b1<e^{f\Theta}$ the pull F becomes negative, the brake is applied automatically and a pull in opposite direction is in order to allow the sheave to turn and thus to lower the load.

If direction of load is reversed or is counterclockwise the greater tension F1 will act at the right end of the band and the smaller tension F2 will act at left end. A similar analysis gives

$$F=Ft\ (e^{f\Theta}b2-b1)/\ ((e^{f\Theta}-1)a)$$

The main factor determining the magnitude of F for a given F_t is the average ratio of lever arms or the ratio of (b1+b2)/2a

Pressure on band

$$p= (F1+F2)/(Dw)$$

here p =Average pressure

D=Diameter of brake drum

w= Band width

Viva- voce

1. What is the difference between the simple band brake and differential band brake?
2. What is the advantage of simple band brake over differential band brake?

Practice problem

1. Determine the capacity in kW at 125 rpm of brake sheave of a differential band brake. The principal dimensions are a=1.05 m b1 =50mm b2 =125 mm, OD=450 mm. the distance from fulcrum 1 to the sheave center is 300 mm. The band can stand a tensile load of 18kN. State the direction of force F upward or downward for a clockwise rotation of sheave. Find the magnitude of force F.

Experiment no. 5

Aim : To Design a single plate clutch by uniform wear theory and uniform wear theory.

Theory: The clutch is a mechanical device which is used to connect or disconnect the source of power from the remaining parts of power transmission system at the will of operator.

Operation: In the operation of clutch the conditions are as follows:

1. Initial condition: the driving member is rotating and driven member is at rest.
2. Final condition: both the members rotate at the same speed and have no relative motion.

Classification of clutches

i) Positive contact clutch

ii) Friction clutches

iii) Electromagnetic clutches

iv) Fluid clutches and couplings

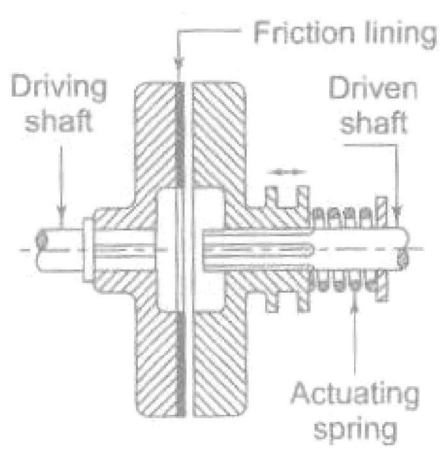

Fig. 5.1 *Single Plate Clutch*

Torque transmitting capacity

Two theories are used to obtain the torque capacity of the clutch. They are called uniform pressure theory and uniform wear theory.

Uniform pressure theory: uniform pressure theory is applicable in case of new clutches.

P= total operating force (N)

M_t= toque transmitted by clutch N-mm

p= intensity of pressure at radius r, N/mm^2

$P = \pi p (D^2-d^2)/ 4$

$M_t= (\mu P (D^3-d^3))/(3(D^2-d^2))$

Uniform Wear Theory: This theory is applicable only to worn out clutches or old clutches. According to this theory the wear is uniformly distributed over the entire surface area of the friction disk. The axial wear is proportional to friction work. The work done by friction force at radius is proportional to the friction al force μp and rubbing velocity 2πrn

Where n is speed in rev/ min.

Wear α (μp) (2πrn)

Assuming speed n and coefficient of friction μ to be constant,

wear α pr

pr= constant

$P = (\pi p_a d(D-d))/2$

$M_t= (\pi \mu p_a d(D^2-d^2))/8$

$M_t = (\mu P (D+d))/4$

Viva voce

1. What is major difference between the uniform pressure and uniform wear theory?
2. How the torque transmitting capacity of the clutches can be increased?

Practice problem

1. A plate clutch consists of one pair of contacting surfaces. The inner and outer diameters of the friction disk are 100 and 200 mm respectively. The coefficient of friction is 0.2 and permissible intensity of pressure is $1 N/mm^2$. Assuming uniform pressure theory and uniform wear theory calculate the power transmitting capacity of clutch at 750 rpm.

Experiment no. 6

Aim : To Design a connecting rod.

Function: The main function of the connecting rod is to transmit the push and pull from the piston pin to crank pin. In many cases its secondary function is to convey the lubricating oil from the bottom end to the top end i.e. from the crank pin to the piston pin and then for splash of jet cooling of piston crown.

Materials: The materials for connecting rods range from mild or medium carbon steels to alloy steel. For high speed engines the connecting rods may also be made of duralumin and aluminium alloys

Shape of connecting rod: I and H sections are most common sections used for connecting rod

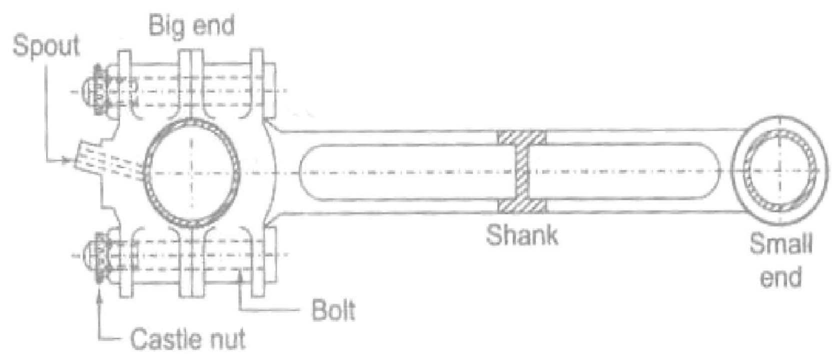

Fig. 6.1 *Connecting Rod*

Stresses in connecting rod

The various forces acting on the on connecting rod are

1. The combined effect of gas pressure on the piston and inertia of the reciprocating parts
2. Friction of piston rings and that of piston.
3. Inertia of connecting rod.
4. The friction of two end bearings i.e. piston pin bearing and crank pin bearing.

1. Load due to gas pressure and piston inertia

The load due to piston inertia = weight of reciprocating masses x acceleration

F_i= (Fw^2r (cos □ + (rcos2□)/l})/g

F= weight of reciprocating parts = weight of piston including that of rings+ weight of piston pin + one third portion of connecting rod(small end portion)

w= angular velocity of crank, rad/s

□= crank angle from TDC

r= crank radius, m

l= rod length, m

2. Force due to friction of piston rings and that of piston

$P_f = h\pi Dzp_r\mu$

h= axial width of the rings

D= cylinder bore

z= no. of rings

p_r= pressure of rings,

μ= coefficient of friction, about 0.1

In the design calculation the effect of friction of piston rings and of the piston can be calcultated.

3. Inertia of connecting rod: the inertia of connecting rod will have two components: along the rod i.e. longitudinal component and normal to rod i.e. the transverse component. The longitudinal component is taken into account by considering about one third portion of the connecting rod on the small end side as reciprocating and remaining two third as rotating with the crank.

Due to transverse component, a centrifugal force will act on every part of the rod the bending force will be zero at the piston pin and maximum at the crank pin. The variation can be assumed to be triangular.

If C is centripetal force acting on a unit length at the crank pin. The C is maximum when the crank and connecting rod are at right angles.

Max. Value of $C=\rho A w^2 r$

P= Density of material

A= cross section area of rod

w= angular speed

r= crank radius

Max. bending moment occurs at a distance of l/sqrt3

So maximum bending moment is given by

$M_{max}= 0.128F_n l$, where $F_n = Cl/2$

Maximum bending stress=Mmax/Z

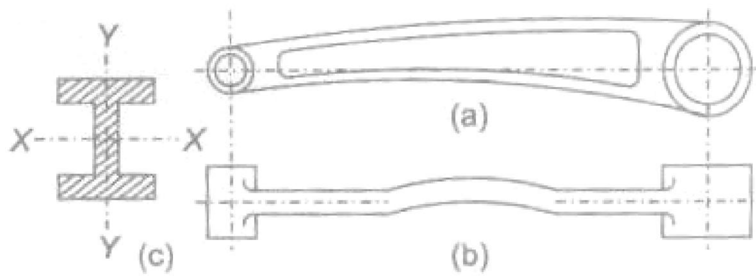

Fig. 6.2 Buckling of Connecting Rod

Buckling load = $f_{cu} A/(1+ a(l/k)^2)$, N

f_{cu}= ultimate crushing stress ,

A= section area

l= equivalent length

k = radius of gyration about axis of buckling, m

Buckling load = Max. gas load X FOS= $(\pi D^2 p_{max} X fos)/ 4$

Viva voce

1. Which section of connecting rod is generally used and why?
2. Why bigger end of connecting rod is made bigger.

Practice problem:

1. Design a connecting rod for four stroke petrol engine with following data

   Piston diameter=0.10 m, stroke =0.14 m, length of connecting rod center to center =0.315m

   Weight of reciprocating parts18.2N, Compression ratio =4:1, speed = 1500 rev/min with

   Possible over speed of 2500. , Probable maximum explosion pressure =2.45MPa

Experiment no. 7

Aim: To design center Crankshaft

Centre Crankshaft

The crankshaft is an important part of IC engine that converts the reciprocating motion of the piston into rotary motion through the connecting rod. The crankshaft consists of three portions-crank pin, crank web and shaft. The big end of the connecting rod is connecting rod is attached to the crank pin. The crank web connects the crank pin to the shaft portion. The shaft portion rotates in the main bearings and transmits power to the outside source through the belt drive, gear drive or chain drive.

There are two types of crankshafts-side crankshaft and centre crankshaft. The side crankshaft is called as the ‘overhang’ crankshaft. It has only one crankshaft and requires only two bearings for support. The centre crankshaft has two webs and three bearings for support. It is used in radial aircraft engines, stationary engines and marine engines. It is more popular in automotive engines.

Design of centre crankshaft

A crankshaft is subjected to bending and torsional moments due to the following three forces:

(i) Force exerted by the connecting rod on the crank pin.

(ii) Weight of flywheel (W) acting downward in the vertical direction.

(iii) Resultant belt tensions acting in the horizontal direction (P_1+P_2).

For the design two cases are considered:

Case1: The crank is at the top dead centre position and subjected to maximum bending moment and no torsional moment.

Case2: The crank is at an angle with the line of dead centre position and subjected to maximum torsional moment.

Centre crankshaft at top dead centre position

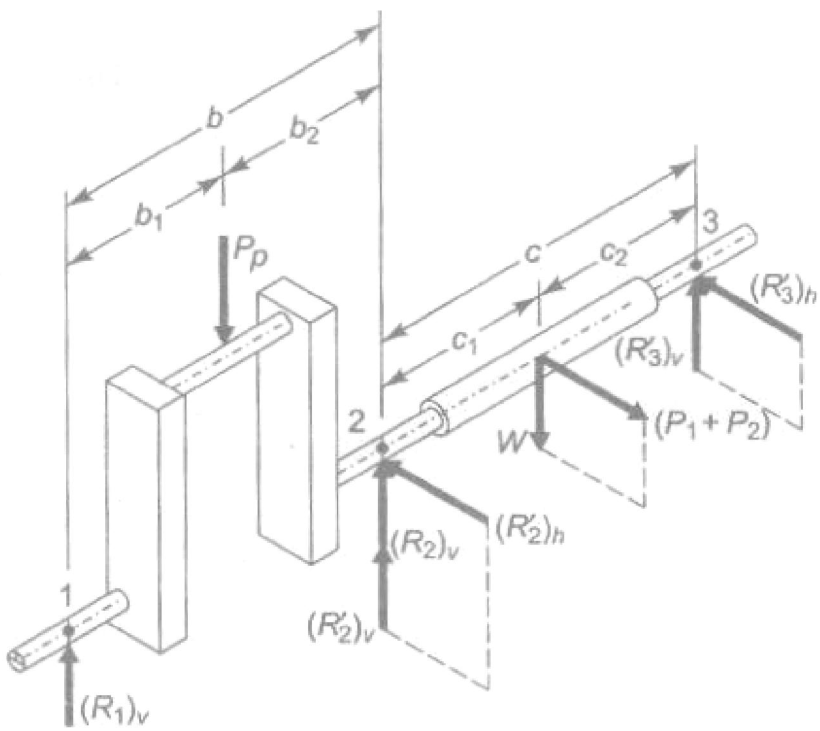

Fig. 7.1 *Centre Crankshaft at Dead Centre*

The crankshaft is supported on three bearings 1, 2 and 3.

Assumptions

(i) The engine is vertical and the crank is at top dead centre position.

(ii) The belt drive is horizontal.

(iii) The crankshaft is simply supported on bearings.

(i)Bearing reactions

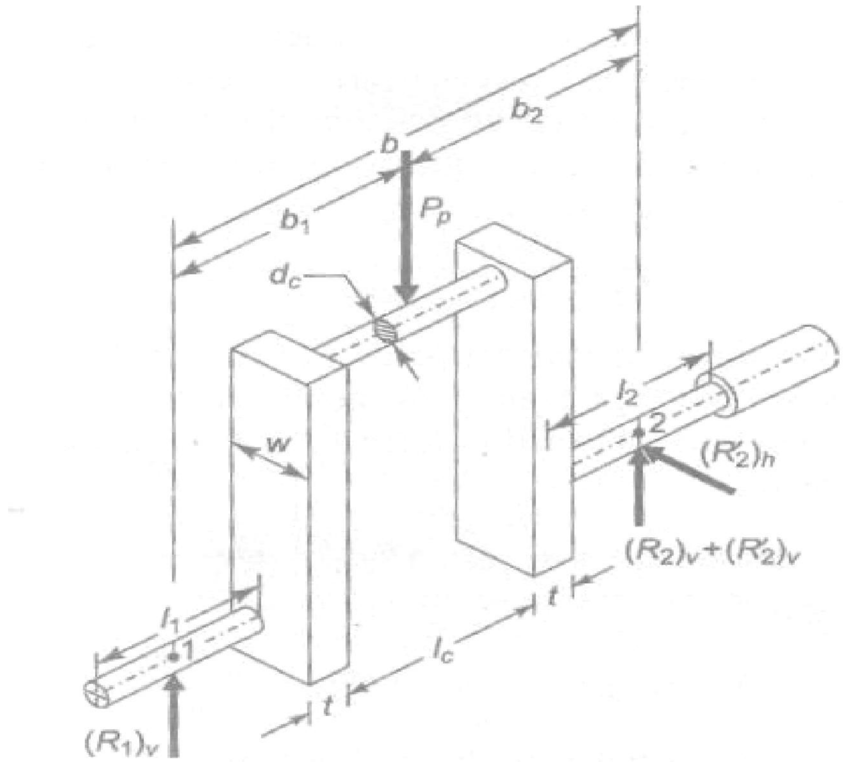

Fig. 7.2 . *Crank Pin and Web*

(a) The reactions at the bearings 1 and 2 due to force on the crank pin(P_p) are denoted by R_1 and R_2 followed by suffix letter v and h.

(b) The reactions at the bearings 2 and 3 due to weight of the flywheel (W) and sum of the belt tensions (P_1+P_2) are denoted by R_2' and R_3' followed by suffix letters v and h.

Suppose,

P_p= force exerted on crank pin (N)

D= diameter of piston (mm)

pmax.= maximum gas pressure inside the cylinder (Mpa or N/mm^2)

W= weight of flywheel(N)

P1=tension in tight side of belt(N)

P2= tension in slack side of belt(N)

b= distance between main bearings 1 and 2

c= distance between bearings 2 and 3

At the top dead centre position, the thrust in the connecting rod will be equal to the force acting on the piston.

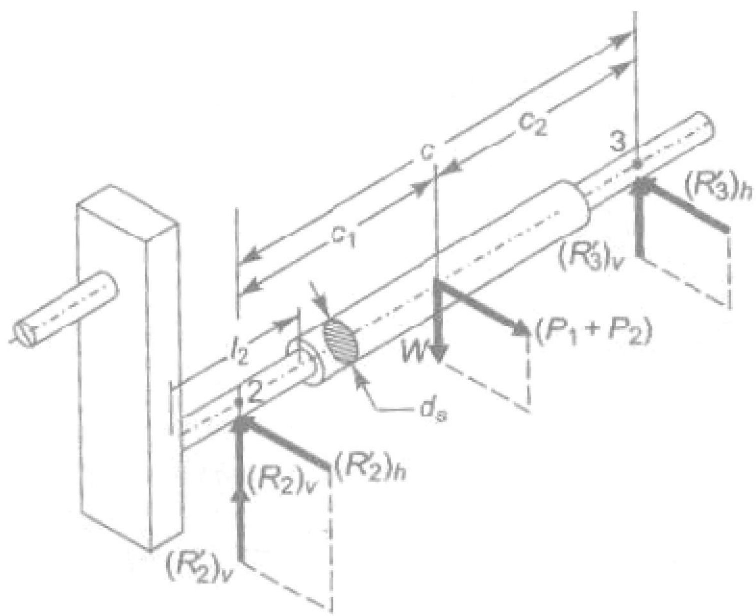

Fig. 7.3 *Shaft Under Flywheel*

$P_p=\{\pi D^2/4\}p_{max}$.

Taking moment of forces,

$P_p*b_1=(R_2)_v*b$ or $\mathbf{(R_2)_v=P_p*b_1/b}$

Similarly,

$P_p*b_2=(R_1)_v*b$ or $\mathbf{(R_1)_v=P_p*b_2/b}$

It is also assumed that the portion of the crankshaft between the bearings 2 and 3 is simply supported on bearings and subjected to a vertical force W and horizontal force (P1+P2).

Taking moment of forces,

$W*c_1=(R_3')_v*c$ or $\mathbf{(R_3')_v=W*c_1/c}$

$W*c_2=(R_2')_v*c$ or $\mathbf{(R_2')v=W*c_2/c}$

$(P_1+P_2)*c_1=(R_3')_h*c$ or $\mathbf{(R_3')_h=(P_1+P_2)*c_1/c}$

$(P_1+P_2)*c_2=(R_2')_h*c$ or $\mathbf{(R_2')_h=(P_2+P_2)*c_2/c}$

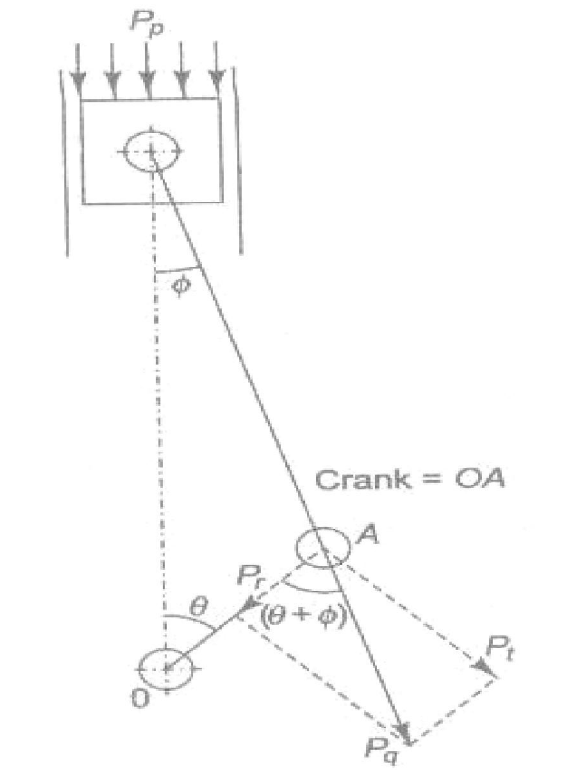

Fig. 7.4 *Force Acting on Crank*

The resultant reactions at the bearings are as follows:

$R_1=(R_1)_v$

$R_2=sqrt\{[(R_2)_v+(R_2')_v]^2 + [(R_2')_h]^2\}$

$R_3=sqrt\{[(R_3')_v]^2 + [(R_3')_h]^2\}$

Note: When the distance b between the bearings 1 and 2 is not specified, it is assumed by the following empirical relationship:

b=2*piston diameter or b=2*D

(ii)Design of crank pin

The central plane of the crank pin is subjected to maximum bending moment. Suppose,

d_c=diameter of crank pin(mm)

l_c=length of crank pin(mm)

σ_b=allowable bending stress for crank pin(N/mm^2)

The bending moment at the central plane is given by,

$\mathbf{(M_b)_c=(R_1)_v b_1}$

$I=\pi dc^4/64$

$y=d_c/2$

$\sigma_b=(M_b)_c y/I$

Substituting,

$(M_b)_c=(\pi d c^3/32)\ \sigma_b$

The diameter of the crank pin can be determined using the above equations.

The length of the crank pin is determined by bearing consideration. Suppose,

p_b=allowable bearing pressure at the crank pin bush(N/mm^2)

$p_b=P_p/d_c*I_c$ or $\mathbf{I_c= P_p/d_c*p_b}$

(iii)Design of left-hand crank web

Suppose,

w=width of crank web(mm)

t=thickness of crank web(mm)

Here,

$\mathbf{t=0.7d_c}$

$\mathbf{w=1.14d_c}$ where d_c=diameter of the crank pin(mm)

The left-hand crank web is subjected to eccentric load$(R_1)_v$. There are two types of stresses in the central plane of the crank web, viz., direct compressive stress and bending stress due to eccentricity of reaction$(R_1)_v$.

The direct compressive stress is given by,

$\boldsymbol{\sigma_c=(R_1)_v/wt}$

The bending moment is given by,

$\boldsymbol{M_b=(R_1)_v*[b_1-(l_c/2)-(t/2)]}$

$I=wt^3/12$

$y=t/2$

$\sigma_b=M_b*y/I$

Substituting,

$$\sigma_b=\frac{(R1)v*[b1-(lc/2)-(t/2)](t/2)}{wt^3/12}$$

$$=\frac{6(R1)v*[b1-(lc/2)-(t/2)]}{wt^2}$$

The total compressive stress is given by,

$\boldsymbol{(\sigma_c)_t=\sigma_c+\sigma_b}$

It should be less than the total allowable bending stress.

(iv)Design of right-hand crank web

The thickness and width of the right-hand crank web are made identical to that of the left-hand crank web(since they are identical from balancing considerations).

(v)Design of shaft under flywheel

The central plane of theshaft is subjected to maximum bending moment. Suppose,

d_s=diameter of shaft under flywheel(mm)

The bending moment in the vertical plane due to resultant belt tension is given by,

$(M_b)_v=(R_3')_v c_2$

The bending moment in the horizontal plane due to resultant belt tension is given by,

$(M_b)_h=(R_3')_h c_2$

The resultant bending moment is given by,

$\mathbf{M_b=\sqrt{[(M_b)_v]^2 + [(M_b)_h]^2}}$

$\mathbf{=\sqrt{[(R_3')_v * c_2]^2 + [(R_3')_h * c_2]^2}}$

Also, $\mathbf{M_b= (\pi(d_s)^3/32)\, \sigma_b}$

Therefore, the diameter of the shaft under flywheel (d_s) can be calculated.

Viva Questions

1. What is crank shaft and why is it used?
2. What are the major stresses induced in the crankshaft?
3. What is difference between center crankshaft and side crankshaft?
4. What is single throw and multi throw crankshaft?

Practice problem

1. Design a plain carbon steel crank shaft for a 0.40 m by 0.60m single acting four stroke single cylinder engine to operate at 200 rev/min. The mean effective pressure is 0.49 MPa, and the maximum combustion pressure is 2.625 MPa. At a Maximum torsional moment when the crank angle is 36 degree, the gas pressure is 0.975 MPa. l/r=4.8. the flywheel is used as pulley weighing 54.50 kN and total belt pull is 6.75kN.