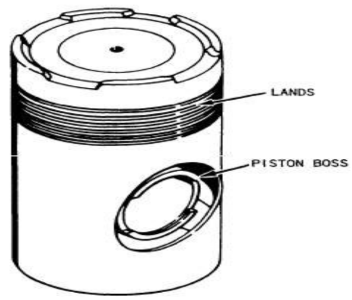
Introduction:

Function of Piston:



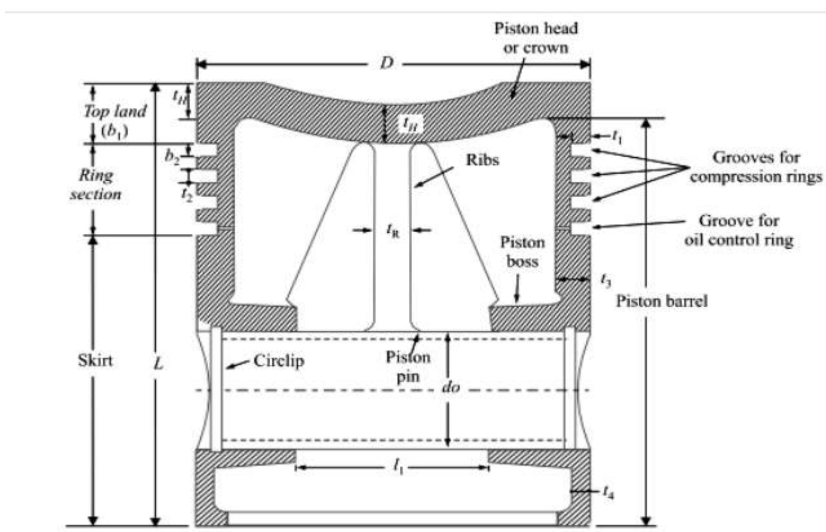
The purpose of the piston is to stand the expansion of gases and send it to the crankshaft. It transfers the force of the explosion to the crankshaft and, in turn, rotates it. The piston comes with rings that seal it and the cylinder wall.

Trunk Type Piston:

- Trunk pistons are long relative to their diameter.
- They act both as a piston and cylindrical crosshead.
- As the connecting rod is angled for much of its rotation, there is also a side force that reacts along the side of the piston against the cylinder wall. A longer piston helps to support this.
- Trunk pistons have been a common design of piston since the early days of the reciprocating internal combustion engine.
- A characteristic of most trunk pistons, particularly for diesel engines, is that they have a groove for an oil ring below the gudgeon pin.

Piston Nomenclature:

- **Head or crown**. The piston head or crown may be flat, convex or concave depending upon the design of combustion chamber. It withstands the pressure of gas in the cylinder.
- **Piston rings**. The piston rings are used to seal the cylinder in order to prevent leakage of the gas past the piston.
- **Skirt**. The skirt acts as a bearing for the side thrust of the connecting rod on the walls of cylinder.
- **Piston pin**. It is also called gudgeon pin or wrist pin. It is used to connect the piston to the connecting rod.



Schematic diagram of Piston

Design head thickness of piston or Piston Crown:

A) Thickness of Piston Head (According to Grashoff's Formula):

$$t_H = D\sqrt{\frac{3 \times p}{16 \times \sigma_t}}$$

Where, p= Maximum gas pressure or Explosion pressure (N/mm2)
D= Cylinder Bore Diameter (mm)
σt = Permissible Bending stress for the material of the Piston (Mpa or N/mm2)
= 35 to 40 (For Grey Cast Iron)
= 50 to 90 (For Nickel Cast Iron & Al Alloy)
= 60 to 100 (For Forged Steel)

OR

B) Thickness of Piston Head:

$$t_H = \frac{H}{12.56 \times K \times (T_c - T_E)}$$

Where, H = Heat flowing through the Piston Head (Watt or kJ/s)
K = Heat Conductivity (W/mm/ ^{0}C)
= 46.6(Grey Cast Iron), 51.25 (Steel) and 174.75 (Al Alloy)
Tc = Temperature at the center of the Piston Head (^{0}C)
TE = Temperature at the edges of the Piston Head (^{0}C)
(Tc – TE) = 220 ^{0}C (For Cast Iron) and 75 ^{0}C (For Al Alloy)

- **Heat flowing through Piston Head (H):**

$$H = C \times HCV \times m \times BHP$$

Where, C = Constant representing that portion of the heat supplied to that Engine which is absorbed by the Piston = 0.05
HCV = Higher Calorific value of the fuel (KJ/Kg)
= 45000 KJ/Kg (For Petrol)
= 47000 KJ/Kg (For Diesel)
m = Mass of the fuel used (Kg/BHP/s)
BHP = Break Horse Power per Cylinder

- **Break Horse Power (BHP):**

Now, we know that Mechanical Efficiency (**η m**) is,

$$\eta_m = \frac{\text{BHP (Output)}}{\text{IHP (Input)}}$$

Where, BHP = Break Horse Power
IHP = Indicated Horse Power

- **So, Break Horse Power (BHP) is,**

$$BHP = \eta m \times IHP$$

- **Indicated Horse Power (IHP):**

$$IHP = \frac{P_m \times L \times A \times n}{60{,}000}$$

Where, Pm = IMEP (Indicated Mean Effective Pressure) (N/mm2)

L = Stroke length (mm)

A = Area of Piston Head (mm2) = $\frac{\Pi D^2}{4}$

Where, D = Bore Diameter (mm)

= N/2 (If there is 4 Stroke engine)

= N (If there is 2 Stroke engine)

Where, N = RPM (Revolution per Minute

Assignment Questions:

1. Write material used for piston.

Ans:

2. Describe function for piston.

Ans:

Solve the Problems:

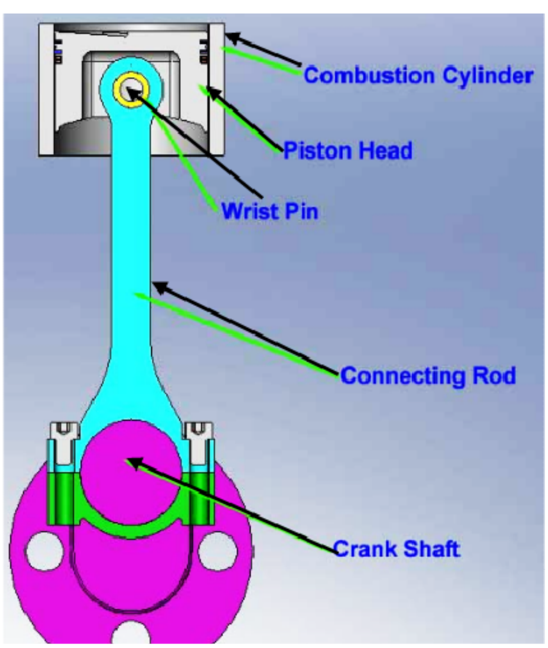
1. Design head thickness of piston for single acting four stroke engine for the following data:

Cylinder bore = 100 mm; Stroke = 125 mm; Maximum gas pressure = 5 N/mm2; Indicated mean effective pressure = 0.75 N/mm2; Mechanical efficiency = 80%; Fuel

consumption = 0.15 kg/BP/hr; HCV = 42 x 103 kJ/kg; Speed = 2000 rpm. Assume required data.

2. Find out piston head thickness for single acting four stroke engine from the following data:

Cylinder bore = 180 mm; Stroke = 270 mm; Maximum gas pressure = 6 N/m2; Indicated mean effective pressure = 0.7 N/m2; Mechanical efficiency = 80%; Fuel consumption = 0.2 kg/BP/hr; HCV = 42 x 103 kJ/kg; C = 0.05; K = 46.6 W/m/^{o}C; Tc-Te = 200^{o}C; Speed = 500 rpm. Take l/d = 1.5 Assume required data.



Introduction:

A Connecting rod is the link between the reciprocating piston and rotating crank shaft. Small end of the connecting rod is connected to the piston by means of gudgeon pin. The big end of the connecting rod is connected to the crankshaft.

They are not rigidly fixed at either end, so that the angle between the connecting rod and the piston can change as the rod moves up and down and rotates around the crankshaft.

The small end attaches to the piston pin, gudgeon pin or wrist pin, which is currently most often press fit into the connecting rod but can swivel in the piston, a "floating wrist pin" design. The big end connects to the bearing journal on the crank throw, in most engines running on replaceable bearing shells accessible via the connecting rod bolts which hold the bearing "cap" onto the big end.

Function:

- The function of the connecting rod is to convert the reciprocating motion of the piston into the rotary motion of the crankshaft.
- The function of the connecting rod also involves transmitting the thrust of the piston to the crank shaft.
- As a connecting rod is rigid, it may transmit either a push or a pull and so the rod may rotate the crank through both halves of a revolution, i.e. piston pushing and piston pulling.

Connecting rod shape:

- The classification of connecting rod is made by the cross sectional point of view i.e. I – section, H – section, Tabular section, Circular section. In low speed engines, the section of the rod is circular, with flattened sides. In high speed engines either an H – section or Tabular section is used because of their lightness. The rod usually tapers slightly from the big end to the small end. To provide maximum rigidity with minimum weight the main cross section of the connecting rod is made an I-section is made to blend smoothly into two rod ends called the small end (piston end) and big end (crank end).

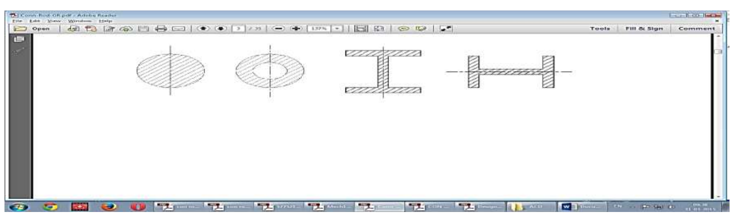


Fig. Shapes of connecting rod

Why connecting rod is made of I – section?

- The I-section of connecting rod is used due to its lightness and to keep the inertia forces as low as possible. Especially in case of high speed engine. It can also withstand high gas pressure. Since connecting rod is manufactured by I-section are rounded off for easy removal of the section from die.
- In high speed engines as the weight reduction are the major objectives. I-section is commonly used. I-section and H-section have higher section modulus per unit area as compared to circular, elliptical, or rectangular section. Hence they offer better resistance against bending as well as buckling.

Forces acting on the Connecting Rod:

The various forces acting on the connecting rod are:

- The combined effect of gas pressure on the piston and the inertia of the reciprocating parts.

- The friction of the piston rings, piston, piston rod and the cross head.
- Inertia of the connecting rod.
- The friction of the two end bearings i.e. of the piston pin bearing and the crank pin bearing.
- Thermal stresses.
- The longitudinal component of the inertia of the rod.
- The transverse component of the inertia of the rod.

Stresses in connecting rod:

- The stresses in connecting rod are set up by a combination of forces. They are induced in a connecting rod as combinations of axial stresses, bending stresses and thermal stresses which are subjected to during its operation. The axial stresses are produced due to cylinder gas pressure (compressive only) and the inertia force arising on account of reciprocating action (Both tensile as well as compressive), whereas bending stresses are caused due to the centrifugal effects.
- The connecting rod is under tremendous stress from the reciprocating load represented by the piston, actually stretching and being compressed with every rotation, and the load increases as the square of the engine speed increase.

Materials:

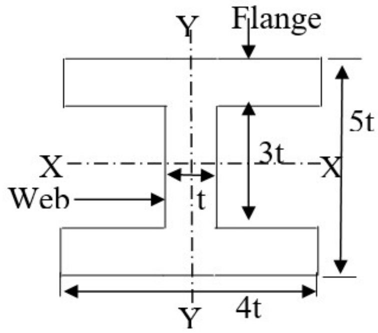
- Different types of materials and production methods are used in manufacturing of the connecting rods. The material for a connecting rod is selected based on the purpose of the connecting rod and depending upon the requirement of the I.C engines.
- The most common types of connecting rods are steel and aluminum. The most common type of manufacturing processes are casting, forging and powdered metallurgy.
- Some of the materials used in the manufacturing of connecting rod are
  Cast iron, Aluminum alloys, Carbon steel, Stainless steel, Magnesium, Titanium e.t.c

Design consideration for connecting rod:

- The connecting rod should have sufficient strength to withstand the momentary high gas pressure in the cylinder.
- It should be light weighted to keep the inertia forces as small as possible.
- Connecting rod is subjected to alternate tension and compression but these factors are considered as a strut while designing it.
- It should be robust enough to withstand bending stress.

Design I-section/circular section of connecting rod:

Design of I-section of connecting rod:



A connecting rod should be equally strong in buckling about both axis. The most suitable section for the connecting rod is I-section with proportional as shown in fig.

Let, thickness of flange and web of section = t.

Width of the section B = 4t.

Depth of the section H = 5t.

Area of the I-section.

$A = 2(4t \times t) + (3t \times t)$

$= 11t^2$

Moment of inertia of the I-section about neutral axis.

$I_{xx} =$

$= [4t\,(5t^3) - 3t\,(3t^3)]$

$= t^4$

Radius of gyration of section

$K_{xx} =$

$=$

$=$

$= 1.78\ t$

The gas pressure acting on connecting rod through piston along the axis of cylinder is given by

$F_P = \times D^2 \times P_{max}$

Where, F_P = maximum gas pressure acting on piston, in N/mm^2.

D = diameter of piston or cylinder bore, in mm.

P_{max} = maximum gas pressure, in N/mm^2.

Buckling load may be calculated by,

W_B = maximum gas force × factor of safety

$= F_P \times F_s$ -- (1)

Where, F_s = factor of safety

= 5 to 6

According to Rankines formula buckling load

$W_B =$ --- (2)

Where, σ_{cs} = ultimate compressive yield stress

A = cross section Area of connecting rod.

l = length of connecting rod.

K_{xx} = radius of gyration about axis of buckling

a = Rankin's constant
= 1/7500 for mild steel.
= 1/ 9000 for wrought iron.
= 1/1600 for cast iron.

From equation (1) and (2) we can obtain the cross sectional area of I-section of connecting rod and the cross section is taken at the centre of connecting rod.

Design of circular section of connecting rod:

Let, l = length of connecting rod.
d = diameter of connecting rod.
D = diameter of piston.
P_{max} = maximum gas pressure acting on connecting rod.
σ_t = allowable tensile stress in tension for connecting rod.
σ_c = compressive yield strength of the material.
E = young modulus of connecting rod material.
I = moment of inertia of the circular section of connecting rod.
= d^4, mm^4.

The gas pressure acting on connecting rod through piston along the axis of cylinder given by,
$F_P = \times D^2 \times P_{max}$

Buckling load may be calculated by,
W_B = maximum gas force × factor of safety
= $F_P \times F_s$ -- (1)

Where, F_s = factor of safety
=20

The buckling rod of connecting rod is taken in to consideration. The ends of connecting rod is fitting in such a fashion that the ends can be considered as fixed from Euler's formula
W_B =

From above equation we can obtain the diameter of the connecting rod for the circular section.

Assignment Questions:

1. Write material used for connecting rod.

Ans:

2. Describe function for connecting rod.

Ans:

3. Why connecting rod is made of I – section?

Ans:

4. List various forces acting on Connecting Rod.

Ans:

Solve the Problems:

1. Design connecting rod of I section for bellow given data:
Weight of reciprocating parts: 32 kg, Engine speed: 1700 rpm,
Length of connecting rod: 400 mm, Stroke: 200 mm, Cylinder bore: 100 mm,
Factor of safety: 6, Max. Gas pressure: 4.2 N/mm^2, a = 1/6400, Crushing stress: 345 N/mm^2,
Take 4t * t * 5t section.

2. Find the dimension of I-section of connecting rod from following data:
Weight of reciprocating parts: 32 kg, Engine speed: 1800 rpm,
Length of connecting rod: 140 mm, Stroke: 80 mm, Cylinder bore: 70 mm,
Factor of safety: 5, Max. Explosion pressure: 4.5 N/mm2, a = 1/6000,
Crushing stress: 100 N/mm^2, Take 4t*t*5t section, Take l/d =1.3, no. of bolts: 2.

3. Find the diameter of connecting rod,
If length of the connecting rod for slow speed diesel engine is 300 cm, cylinder diameter 110 cm and stroke 140 cm, Maximum gas pressure 60 kg/cm^2, Factor of safety = 20, E = 2.1 x 10^6 kg/cm^2

Introduction:

Function of Clutch:-

The use of a clutch is mostly found in automobiles. A little consideration will show that in order to change gears or to stop the vehicle, it is required that the driven shaft should stop, but the engine should continue to run. It is, therefore, necessary that the driven shaft should be disengaged from the driving shaft. The engagement and disengagement of the shafts is obtained by means of a clutch which is operated by a lever.

Types of Clutch:-

Following are the two main types of clutches commonly used in engineering practice:

1. Positive clutches
2. Friction clutches.

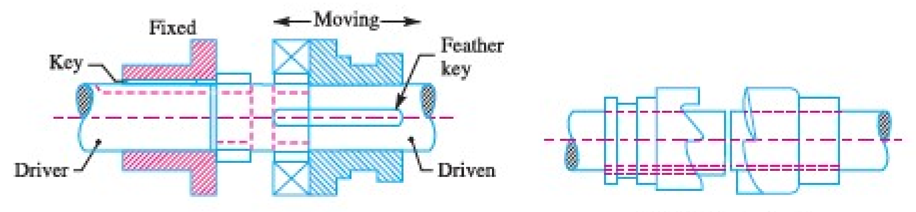
A. Disc or plate clutches (single disc or multiple disc clutch),

B. Cone clutches, and

C. Centrifugal clutches.

Positive Clutches

The positive clutches are used when a positive drive is required. The simplest type of a positive clutch is a jaw or claw clutch. The jaw clutch permits one shaft to drive another through a direct contact of interlocking jaws. It consists of two halves, one of which is permanently fastened to the driving shaft by a sunk key. The other half of the clutch is movable and it is free to slide axially on the driven shaft, but it is prevented from turning relatively to its shaft by means of feather key. The jaws of the clutch may be of square type as shown in Fig. (a) Or of spiral type as shown in Fig. (b).



(*a*) Square jaw clutch. (*b*) Spiral jaw clutch.

A square jaw type is used where engagement and disengagement in motion and under load is not necessary. This type of clutch will transmit power in either direction of rotation. The spiral jaws may be left-hand or right-hand, because power transmitted by them is in one direction only. This type of clutch is occasionally used where the clutch must be engaged and disengaged while in motion. The use of jaw clutches are frequently applied to sprocket wheels, gears and pulleys. In such a case, the non-sliding part is made integral with the hub.

Friction Clutches

A friction clutch has its principal application in the transmission of power of shafts and machines which must be started and stopped frequently. Its application is also found in cases in which power is to be delivered to machines partially or fully loaded. The force of friction is used to start the driven shaft from rest and gradually brings it up to the proper speed without excessive slipping of the friction surfaces. In automobiles, friction clutch is used to connect the engine to the drive shaft. In operating such a clutch, care should be taken so that the friction surfaces engage easily and gradually bring the driven shaft up to proper speed. The proper alignment of the bearing must be maintained and it should be located as close to the clutch as possible. It may be noted that:

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

Materials for Friction surfaces:-

The material used for lining of friction surfaces of a clutch should have the following characteristics:

1. It should have a high and uniform coefficient of friction.
2. It should not be affected by moisture and oil.
3. It should have the ability to withstand high temperatures caused by slippage.
4. It should have high heat conductivity.
5. It should have high resistance to wear and scoring.

The materials commonly used for lining of friction surfaces and their important properties are shown in the following table.

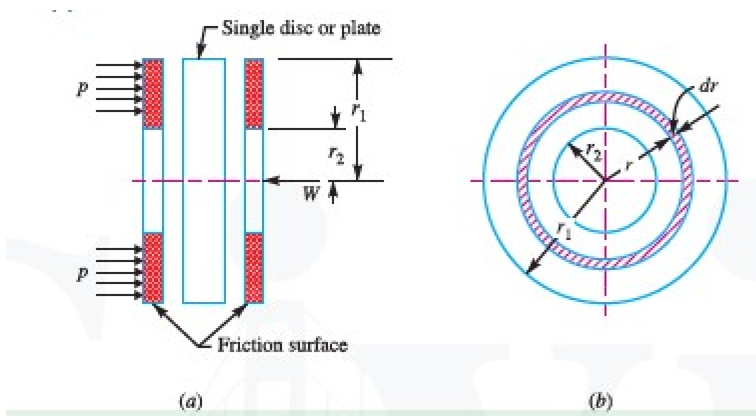
Design Consideration for Friction Clutches:-

The following considerations must be kept in mind while designing a friction clutch.

1. The suitable material forming the contact surfaces should be selected.
2. The moving parts of the clutch should have low weight in order to minimize the inertia load, especially in high speed service.
3. The clutch should not require any external force to maintain contact of the friction surfaces.
4. The provision for taking up wear of the contact surfaces must be provided.
5. The clutch should have provision for facilitating repairs.
6. The clutch should have provision for carrying away the heat generated at the contact surfaces.
7. The projecting parts of the clutch should be covered by guard.

Design of a Disc or Plate Clutch

Consider two friction surfaces maintained in contact by an axial thrust (W) as shown in Fig.



T = Torque transmitted by the clutch,
p = Intensity of axial pressure with which the contact surfaces are held together,
r1 and r2 = External and internal radii of friction faces,
r = Mean radius of the friction face, and
⌠= Coefficient of friction.
Consider an elementary ring of radius r and thickness dr as shown in Fig. (b).
We know that area of the contact surface or friction surface = 2□ r.dr
So, 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,

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

Frictional torque acting on the ring,

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

We shall now consider the following two cases:
1. When there is a uniform pressure, and
2. When there is a uniform axial wear.
1. Considering uniform pressure. When the pressure is uniformly distributed over the entire area of the friction face as shown in Fig. (a), then the intensity of pressure,

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

Where W = Axial thrust with which the 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 r2 to r1 for the total friction torque.
So, Total frictional torque acting on the friction surface or on the clutch,

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

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

... (Substituting the value of p)

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

where $R = \frac{2}{3}\left[\frac{(r_1)^3 - (r_2)^3}{(r_1)^2 - (r_2)^2}\right]$ = Mean radius of the friction surface.

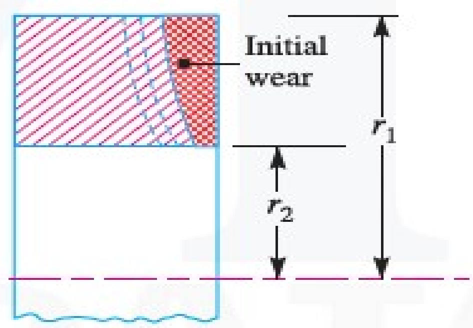
2. Considering uniform axial wear. The basic principle in designing machine parts that is Subjected to wear due to sliding friction is that the normal wear is proportional to the work of friction.

The work of friction is proportional to the product of normal pressure (p) and the sliding velocity

(V). Therefore,

$$\text{Normal wear} \propto \text{Work of friction} \propto p.V$$

or $$p.V = K \text{ (a constant) or } p = K/V \quad ...(i)$$



It may be noted that when the friction surface is new, there is a uniform pressure distribution over the entire contact surface.

This pressure will wear most rapidly where the sliding velocity is maximum and this will reduce the pressure between the friction surfaces. This wearing-in process continues until the product p.V is constant over the entire surface. After this, the wear will be uniform as shown in Fig.

Let 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

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

and the normal force on the ring,

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

Total force acing on the friction surface,

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

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

We know that the frictional torque acting on the ring,

$$T_r = 2\pi\,\mu.p.r^2.dr = 2\pi\,\mu \times \frac{C}{r} \times r^2.dr = 2\pi\mu.C.r.dr \qquad ...(\because p = C/r)$$

Total frictional torque acting on the friction surface (or on the clutch),

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

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

$$= \pi\mu \times \frac{W}{2\pi\,(r_1 - r_2)}[(r_1)^2 - (r_2)^2] = \frac{1}{2} \times \mu.W\,(r_1 + r_2) = \mu.W.R$$

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

NOTE:

1. In general, total frictional torque acting on the friction surfaces (or on the clutch) is given by

$$T = n.\mu.W.R$$

where n = Number of pairs of friction (or contact) surfaces, and

R = Mean radius of friction surface

$$= \frac{2}{3}\left[\frac{(r_1)^3 - (r_2)^3}{(r_1)^2 - (r_2)^2}\right] \qquad \text{... (For uniform pressure)}$$

$$= \frac{r_1 + r_2}{2} \qquad \text{... (For uniform wear)}$$

2. For a single disc or plate clutch, normally both sides of the disc are effective. Therefore a single disc clutch has two pairs of surfaces in contact (i.e. n = 2).

3. Since the intensity of pressure is maximum at the inner radius (r2) of the friction or contact surface, therefore equation **(ii)** may be written as

$$p_{max} \times r_2 = C \text{ or } p_{max} = C / r_2$$

4. Since the intensity of pressure is minimum at the outer radius (r1) of the friction or contact surface, therefore equation **(ii)** may be written as

$$p_{min} \times r_1 = C \text{ or } p_{min} = C / r_1$$

5. The average pressure (pav) on the friction or contact surface is given by

$$p_{av} = \frac{\text{Total force on friction surface}}{\text{Cross-sectional area of friction surface}} = \frac{W}{\pi\,[(r_1)^2 - (r_2)^2]}$$

Assignment Questions:

1. Write down the types of clutch.

Ans.

2. Write down the function of clutch.

Ans.

3. Explain the requirement of clutch.

Ans.

4. List the friction material and write the design consideration for clutch.

Ans.

Solve the Problems:

1. A single plate clutch both side effective is to transmit 145 kW at 2000rpm. The ratio of outer diameter to inner diameter is 1.25, Coefficient of friction as 0.4, Maximum intensity of pressure is 0.12 N/mm^2. Assuming UWC find diameters of surfaces and axial thrust.

2. A multiplate clutch having five plates has max.pressure limit 0.127 N/mm2. Find power transmitted by it at 500RPM if inner and outer radii are 75mm and 125mm respectively. Take co-efficient of friction=0.3

3.A single plate clutch both side effective is to transmit 25 kW at 3000 rpm. The ratio of outer diameter to inner diameter is 1.25, Coefficient of friction as 0.255, Maximum intensity of pressure is 0.1 N/mm2. Determine the outer and inner diameters of frictional surfaces and also determine axial thrust. Assume the theory of uniform wear.

Introduction:

Function of Flywheel:-

A flywheel used in machines serves as a reservoir which stores energy during the period when the supply of energy is more than the requirement and releases it during the period when the requirement of energy is more than supply. In case of steam engines, internal combustion engines, reciprocating compressors and pumps, the energy is developed during one stroke and the engine is to run for the whole cycle on the energy produced during this one stroke.

For example, in I.C. engines, the energy is developed only during power stroke which is much more than the engine load, and no energy is being developed during suction, compression and exhaust strokes in case of four stroke engines and during compression in case of two stroke engines. The excess energy developed during power stroke is absorbed by the flywheel and releases it to the crankshaft during other strokes in which no energy is developed, thus rotating the crankshaft at a uniform speed. A little consideration will show that when the flywheel absorbs energy, its speed increases and when it releases, the speed decreases. Hence a flywheel does not maintain a constant speed; it simply reduces the fluctuation of speed.

Fluctuation of Speed:-

The Co-efficient of Fluctuation of Speed:- The difference between the maximum and minimum speeds during a cycle is called the maximum fluctuation of speed. The ratio of the maximum fluctuation of speed to the mean speed is called coefficient of fluctuation of speed.

Let, *N*1 = Maximum speed in r.p.m. during the cycle,

*N*2 = Minimum speed in r.p.m. during the cycle, and

N = Mean speed in r.p.m. = (N1+N 2)/2

∴ Coefficient of fluctuation of speed,

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

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

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

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

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

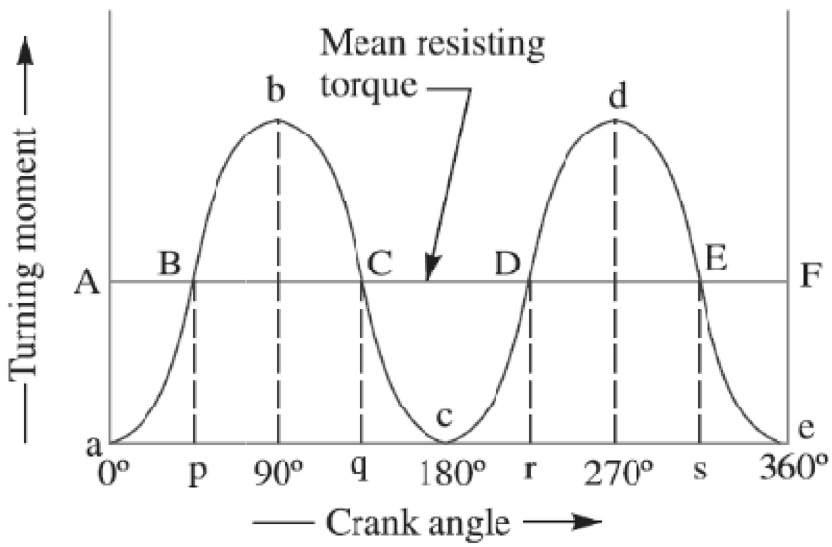
$$\therefore m = \frac{1}{Cs} = \frac{N}{N_1 - N_2} = \frac{\omega}{\omega_1 - \omega_2} = \frac{v}{v_1 - v_2}$$

Fluctuation of Energy:-

The fluctuation of energy may be determined by the turning moment diagram for one complete cycle of operation. Consider a turning moment diagram for a single cylinder double acting steam engine as shown in Fig. The vertical ordinate represents the turning moment and the horizontal ordinate (abscissa) represents the crank angle.

A little consideration will show that the turning moment is zero when the crank angle is zero. It rises to a maximum value when crank angle reaches 90º and it is again zero when crank angle is 180º. This is shown by the curve abc in Fig. 1 and it represents the turning moment diagram for outstroke. The curve cde is the turning moment diagram for in stroke and is somewhat similar to the curve abc. Since the work done is the product of the turning moment and the angle turned, therefore the area of the turning moment diagram represents

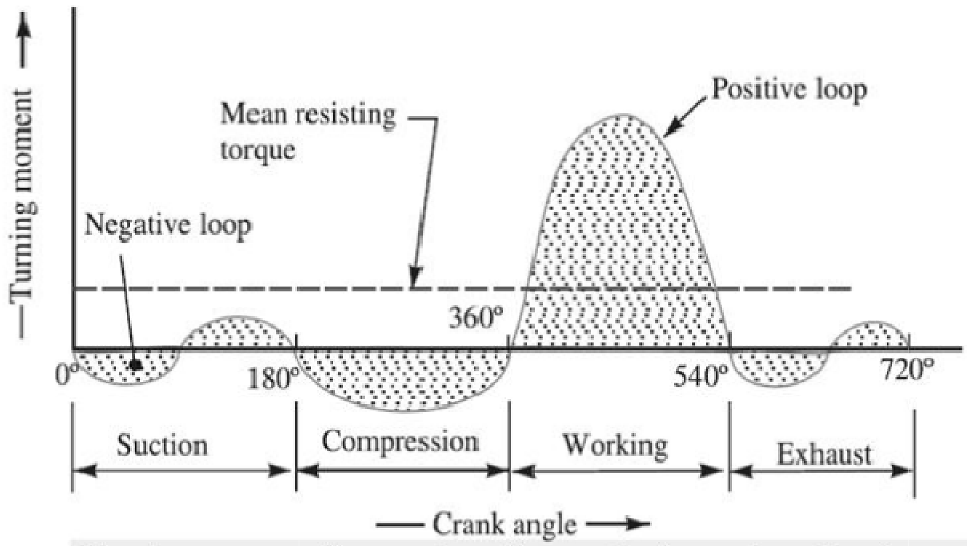
In actual practice, the engine is assumed to work against the mean resisting torque, as shown by a horizontal line AF. The height of the ordinate aA represents the mean height of the turning moment diagram. Since it is assumed that the work done by the turning moment per revolution is equal to the work done against the mean resisting torque, therefore the area of the rectangle aA Fe is proportional to the work done against the mean resisting torque.



Turning moment diagram for a single cylinder double acting steam engine.

We see in Fig. 1, that the mean resisting torque line AF cuts the turning moment diagram at points B, C, D and E. When the crank moves from ‘a’ to ‘p’ the work done by the engine is equal to the area aBp, whereas the energy required is represented by the area aABp. In other words, the engine has done less work (equal to the area aAB) than the requirement. This amount of energy is taken from the flywheel and hence the speed of the flywheel decreases. Now the crank moves from p to q, the work done by the engine is equal to the area pBbCq, whereas the requirement of energy is represented by the area pBCq. Therefore the engine has done more work than the requirement. This excess work (equal to the area BbC) is stored in the flywheel and hence the speed of the flywheel increases while the crank moves from p to q.

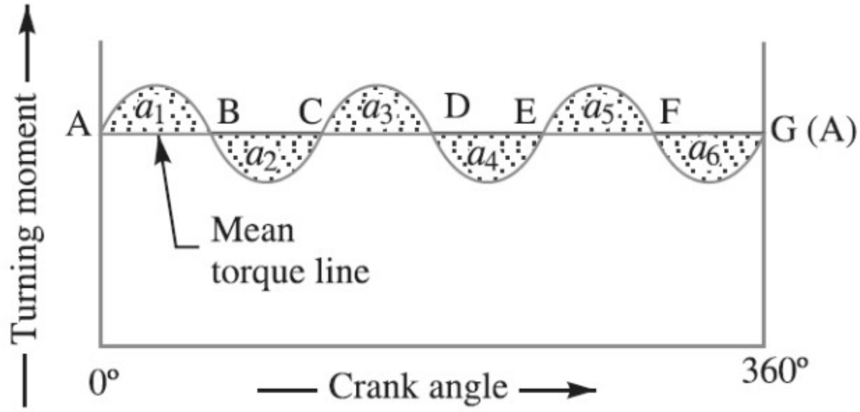
Similarly when the crank moves from q to r, more work is taken from the engine than is developed. This loss of work is represented by the area CcD. To supply this loss, the flywheel gives up some of its energy and thus the speed decreases while the crank moves from q to r. As the crank moves from r to s, excess energy is again developed given by the area DdE and the speed again increases. As the piston moves from s to e, again there is a loss of work and the speed decreases. The variations of energy above and below the mean resisting torque line are called fluctuation of energy. The areas BbC, CcD, DdE etc. represent fluctuations of energy.



Tunring moment diagram for a four stroke internal combustion engine.

Maximum Fluctuation of Energy

A turning moment diagram for a multi-cylinder engine is shown by a wavy curve in Figure. The horizontal line *AG* represents the mean torque line. Let *a*1, *a*3, *a*5 be the areas above the mean torque line and *a*2, *a*4 and *a*6 be the areas below the mean torque line. These areas represent some quantity of energy which is either added or subtracted from the energy of the moving parts of the engine.



Turning moment diagram for a multi-cylinder engine.

Let the energy in the flywheel at $A = E$, then from Fig., we have

Energy at B $= E + a1$

Energy at C $= E + a1 - a2$

Energy at D $= E + a1 - a2 + a3$

Energy at E $= E + a1 - a2 + a3 - a4$

Energy at F $= E + a1 - a2 + a3 - a4 + a5$

Energy at G $= E + a1 - a2 + a3 - a4 + a5 - a6 =$ *Energy at A*

Let us now suppose that the maximum of these energies is at *B* and minimum at *E*.

∴ Maximum energy in the flywheel

$= E + a1$

And minimum energy in the flywheel

$= E + a1 - a2 + a3 - a4$

∴ Maximum fluctuation of energy,

ΔE = Maximum energy – Minimum energy

$= (E + a1) - (E + a1 - a2 + a3 - a4) = a2 - a3 + a4$

COEFFICIENT OF FLUCTUATION OF ENERGY

It is defined as the ratio of the maximum fluctuation of energy to the work done per cycle. It is usually denoted by *C*E. Mathematically, coefficient of fluctuation of energy,

$$CE = \frac{Maximum\ \ fluctuation\ \ of\ Energy}{Work\ done\ per\ cycle}$$

The work done per cycle may be obtained by using the following relations:

1. Work done / cycle = Tmean × θ

Where, Tmean = Mean torque, and

θ = Angle turned in radians per revolution

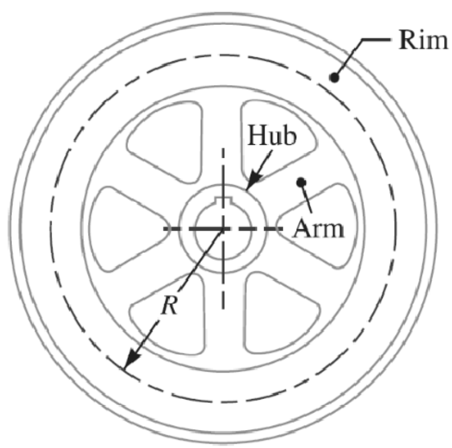
= 2 π, in case of steam engines and Two Stroke

Internal Combustion Engines.

= 4 π, in case of four stroke internal combustion engines.

The mean torque (Tmean) in N-m may be obtained by using the following relation *i.e.*
Tmean = $(P \times 60)/2\pi N$
Where P = Power transmitted in watts,
N = Speed in r.p.m., and
ω = Angular speed in rad/s = $2\pi N / 60$
2. The work done per cycle may also be obtained by using the following relation:
Work done / cycle = $(P \times 6)/n$
Where n = Number of working strokes per minute.
= N, in case of steam engines and two stroke internal combustion engines.
= $N / 2$, in case of four stroke internal combustion engines.

Energy Stored in the Flywheel:-



Flywheel.

A flywheel is shown in Fig.

A flywheel is shown in Fig.
m = Mass of the flywheel in kg,
k = Radius of gyration of the flywheel in meters,
I = Mass moment of inertia of the flywheel about the axis of rotation in kg · m2
= m · k2
N1 and N2 = Maximum and minimum speeds during the cycle in r.p.m.,
ω1 and ω2 = Maximum and mi nimum angular speeds during the cycle in rad / s,
N = Mean speed during the cycle in r.p.m = $(N_1+N_2)/2$

ω = Mean angular speed during the cycle in rad / s = $(\omega_1+\omega_2)/2$
Cs = Coefficient of fluctuation of speed = $(N_1 - N_2)/N = (\omega_1 - \omega_2)/\omega$
We know that mean kinetic energy of the flywheel,

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

As the speed of the flywheel changes from ω1 to ω2, the maximum fluctuation of energy,
ΔE = Maximum K.E. — Minimum K.E.

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

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

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

$$= I\,\omega(\omega_1 - \omega_2) \qquad [\because \omega = \frac{(\omega_1+\omega_2)}{2}] \qquad \text{.... (1)}$$

$$= I\,\omega^2\,\frac{\omega_1 - \omega_2}{\omega} \qquad [\because I = m \cdot k^2] \qquad \text{.... (2)}$$

$$= I\,\omega^2\,C_s \qquad [\because E = \frac{1}{2} \times I \times \omega^2] \qquad \text{.... (3)}$$

$$= m \cdot k^2 \times \omega^2 \times C_s$$

$$= 2\,E\,C_s$$

The radius of gyration (k) may be taken equal to the mean radius of the rim (R), because the thickness of rim is very small as compared to the diameter of rim. Therefore substituting $k = R$ in equation (2), we have
$\Delta E = m \cdot R^2 \cdot \omega^2 \cdot Cs = m \cdot \mathcal{V}^2 \cdot Cs \; (\because \mathcal{V} = \omega \cdot R)$
From this expression, the mass of the flywheel rim may be determined.
Notes:
1. In the above expression, only the mass moment of inertia of the rim is considered and the mass moment of inertia of the hub and arms is neglected. This is due to the fact that the major portion of weight of the flywheel is in the rim and a small portion is in the hub and arms. Also the hub and arms are nearer to the axis of rotation, therefore the moment of inertia of the hub and arms is very small.
2. The density of cast iron may be taken as 7260 kg / m^3 and for cast steel; it may taken as 7800 kg / m3.
3. The mass of the flywheel rim is given by m = Volume × Density = $2\pi R \times A \times \rho$
From this expression, we may find the value of the cross-sectional area of the rim. Assuming the cross-section of the rim to be rectangular, then
A = b × t
Where b = Width of the rim, and
t = Thickness of the rim.
Knowing the ratio of b / t which is usually taken as 2, we may find the width and thickness of rim.
4. When the flywheel is to be used as a pulley, then the width of rim should be taken 20 to 40 mm greater than the width of belt.

Assignment Questions:

1. Write the function of fly wheel.

Ans.

2. What is turning moment diagram?

Ans.

3. Explain Co-efficient of fluctuation of speed in flywheel.

Ans.

4. Derive formula to find out mass and cross sectional area of flywheel rim for given energy fluctuation.

Ans.

Solve the Problems:

1. The scales for the turning moment diagram for an engine are as Follow. Turning moment: 1mm= 70 N-m and Crank angle: 1mm=4.5^0.The turning moment diagram is repeated at every ½ of engine revolution and the areas are -35, +410, -285, +325, -335, +260, -365, +285, -260 mm^2. The engine speed is 900 r.p.m. and the fluctuation of speed is 2% of the mean speed. Find the mass and cross section of the flywheel rim having 650 mm mean diameter. The rim is rectangular with the width 2 times the thickness. Take density as 7250 kg/m^3.

2. A single cylinder double acting engine develops 240 kW at a mean speed of 160 rpm. The coefficient of fluctuation of energy is 0.1 and the fluctuation of speed is + 4% of mean speed. If the mean diameter of flywheel rim is 4 m and the hub and spokes provides 5% of the rotational inertia of the wheel, find the mass of flywheel and cross sectional area of the rim. Take density as 7450 kg/m^3.