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المحاضرة الثالثة

PISTON

PISTON

The piston is a disc which reciprocates within a cylinder. It is either moved by the fluid or it moves the fluid which enters the cylinder. The main function of the piston of an internal combustion engine is to receive the impulse from the expanding gas and to transmit the energy to the crankshaft through the connecting rod. The piston must also disperse a large amount of heat from the combustion chamber to the cylinder walls.

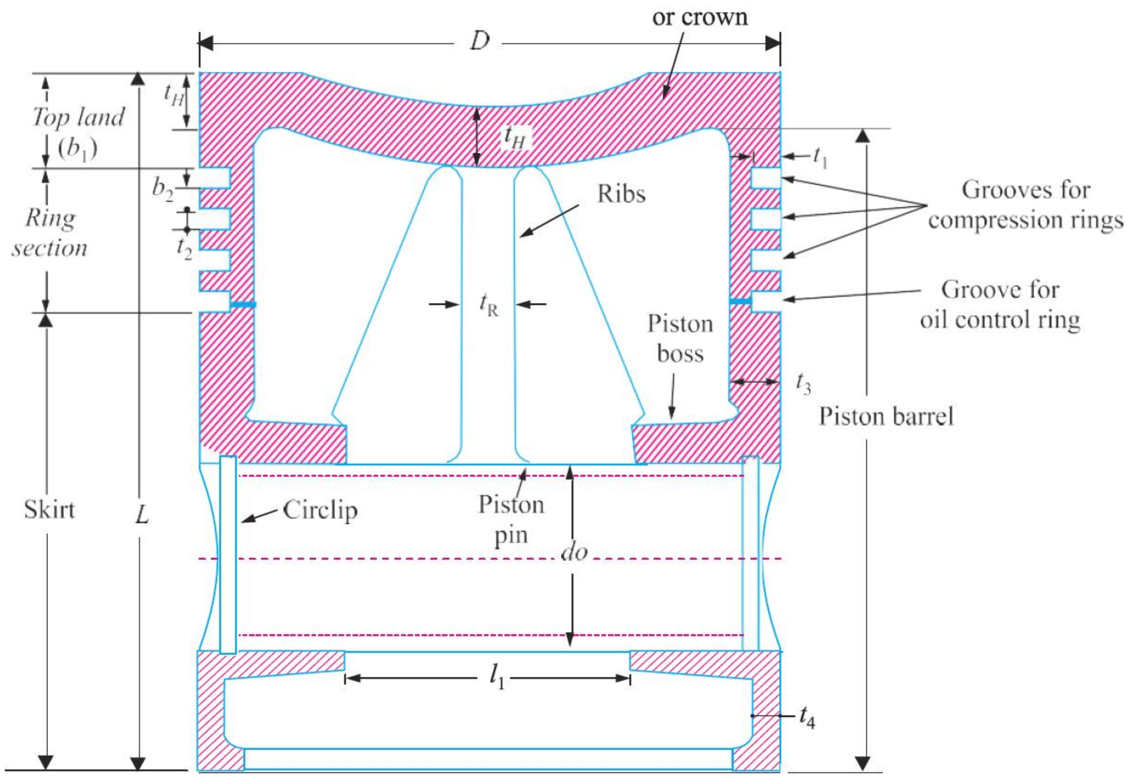

Fig.Piston for i.c Engine

The piston of internal combustion engines are usually of trunk type as shown in Fig.32.3. Such pistons are open at one end and consists of the following parts :

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.

DESIGN CONSIDERATIONS FOR A PISTON

In designing a piston for I.C. engine, the following points should be taken into consideration :

1. It should have enormous strength to withstand the high gas pressure and inertia forces.
2. It should have minimum mass to minimise the inertia forces.
3. It should form an effective gas and oil sealing of the cylinder.
4. It should provide sufficient bearing area to prevent undue wear.
5. It should disprese the heat of combustion quickly to the cylinder walls.
6. It should have high speed reciprocation without noise.
7. It should be of sufficient rigid construction to withstand thermal and mechanical distortion.
8. It should have sufficient support for the piston pin.

PISTON MATERIALS

Since the piston is subjected to highly rigorous conditions , it should have enormous strength and heat resisting properties to withstand high gas pressure. Its construction should be rigid enough to withstand thermal and mechanical distortion. Also the piston should be operated with least friction and noiseless. The material of the piston must possess good wear resisting operating temperature and it should be corrosive resistant.

The most commonly used materials for the pistons of I.C engines are cast-iron, cost-aluminium, forged aluminium, cast steel and forged steel. Cast iron pistons are used for moderate speed i.e below 6m/s and aluminium pistons are employed for higher piston speeds greater than 6 m/s.

DESIGN OF PISTON

When designing a piston, the following points must be considered such as

1. Adequate strength to withstand high pressure produced by the gas.
2. Capacity of piston to withstand high temperature.
3. Scaling of the working space against escape of gases.
4. Good dissipation of heat to the cylinder wall
5. Sufficient projected area (i.e surface area) and rigidity of the barrel.
6. Minimum loss of power due to friction.
7. Sufficient length to have better guidance and so on. The dimentions of various parts of the trunk-type piston are determined as follows.

PISTON HEAD

The piston head or crown is designed keeping in view the following two main considerations, i.e.

1. It should have adequate strength to withstand the straining action due to pressure of explosion inside the engine cylinder, and

2. It should dissipate the heat of combustion to the cylinder walls as quickly as possible.

On the basis of first consideration of straining action, the thickness of the piston head is determined by treating it as a flat circular plate of uniform thickness, fixed at the outer edges and subjected to a uniformly distributed load due to the gas pressure over the entire cross-section.

Based on strength consideration, the thickness of the piston head (t1), according to Grashoff's formula is given by

$$t_1 = \sqrt{\frac{3pmD2}{16\sigma tp}} \quad mm$$

where p_m = Maximum gas pressure N/mm^2

D = Allowable of piston or cylinder bore(mm)

σ_{tp} =Allowable tensile stress of the piston material

= 35 to 40 N/mm^2 for cast iron

= 60 to 100 N/mm^2 for steel

= 50 to 90 N/mm^2 for aliminium alloy

Based on heat dissipation, the head thickness is determined as,

$$t_1 = \frac{1000\,H}{12.56k(Tc - Te)} \; mm$$

where H= Heat following through the head (KW)

H = C x m x C_v x P_B

C =Constant (Usually 0.05). It is the piston of the hat supplied to the engine which is absorbed by the piston.

m = mass of the fuel used (i.e fuel consumption) (kg/kw/s)

C_v = Higher calorific value of the fuel(KJ/kg)

= 44 x10^3 KJ/kg for diesel fuel

= 11 x10^3 KJ/kg for petrol fuel.

P_B = Brake power of the engine per cycle (KW)

$$= \frac{\text{Pmb}L\text{An}}{60000000}\ kW$$

Pmb= Brake mean effective pressure (N/mm^2)

L= stroke length (mm)

A= Area of piston at its top side (mm^2)

n= Number of power strokes per minute

K= Heat conductivity factor (kw/m/^{0}C)

= 46.6 x 10^{-3} for cast iron

= 51 x 10^{-3} for steel

= 175 x 10^{-3} for aluminium alloys

T_c= Temperature at the centre of piston head (^{0}C)

T_e= Temperature at the edge of piston head (^{0}C)

=75 ^{0}C for aluminium alloys

RIBS:

To make the piston rigid and to present distortion due to gas load and connecting rod, thrust, four to six ribs are provided at the inner of the piston. The thickness of rib is assumed as t_2=(0.3 to 0.5)t_1

Where t_1 is thickness of the piston head.

PISTON RINGS:

To maintain the seal between the piston and the inner wall of the cylinder, some split-rings called as piston rings are employed. By making such sealing the escape of gas through piston side-wall to the connecting rod side can be prevented. The piston rings also serve to transfer the heat from the piston head to cylinder walls.

With respect to the location of piston rings, they are called as top rings, or bottom rings. Rings inserted at the top of the piston side wall are compression rings which may be 3 to

4 for automobiles and air craft engines and 5 to 7 for stationary compression ignition engines. Rings inserted at the bottom of the piston side wall are oil scraper rings, used to scraps the ol from the surface liner so as to minimize the flow of oil into the combustion chamber. The number of oil scrapper rings may be taken as 1 to 3. In the oil rings, the bottom edge is stepped to drain the oil.

The compression rings (i.e top side piston rings) are made of rectangular cross-section and their diameters are made slightly larger than the bore diameter. A part of the ring is cut off in order to permit the ring to enter into the cylinder liner.

Due to difference of diameters between the piston rings and liner, a pressure is exerted on the liner by the piston rings. Sufficient clearance should be given, between the cut ends (i.e free ends) of the piston-rings in order to prevent the ends contact at high temperature by thermal expansion.

Usually the piston rings are made of alloy cast iron with chromium plated to possess good wear resisting qualities and spring characteristics even at high temperatures. When designing on the liner wall should be limited between 0.025 N/mm^2 and 0.042 N/mm^2 .

Let t_3 = radial thickness of piston rings

t_4 = Axial thickness of piston rings

p_c= contact pressure (i.e wall pressure) in N/mm^2

Now radial thickness

$$t_3 = D\sqrt{\frac{3Pc}{\sigma br}} \quad mm$$

and the axial thickness t_4 =(0.7 to 1) t_3

or by empirical relation

$$t_4 = \frac{D}{10i}$$

where D = Bore diameter mm

σ_{br}= Allowable bending stress of ring material N/mm^2 = Alloy cast iron 84 to 112 N/mm^2

I = Number of rings.

Due to some advantages like, better scaling action, less wear of lands etc,. usually thinner rings are preferred. The first ring groove is cut at a distance of t1 to 1.2t1 from top.

The lands between the rings may be equal to or less than the axial thickness of ring t4. The gap between the free ends of the ring is taken as

C = (3.5 to 4) t3

Where t3 is the radial thickness of ring.

PISTON BARREL:

The cylindrical portion of the piston is termed as piston barrel. The barrel thickness may be varied (usually reduced) from top side to bottom side of the piston. The maxmum thickness of barrel nearer to piston head is given by, t5 = 0.03D+b+4.5 mm

Where b= radial depth of ring-groove b= t3+0.4mm

The thickness of barrel at the open end of the piston,t6=(0.25 to 0.35) t5 mm

PISTON SKIRT

The portion of the piston barrel below the ring selection upto the open end is called as portion-skirt. The piston skirt takes up the thrust of the connecting rod. The length of the piston skirt is selected in such a way that the side thrust pressure should not exceed 0.28 N/mm2 for slow speed engines and 0.5 N/mm2 for high speed engies.

The side thrust force is given by,

Fs = μFg

Where μ = coefficient of friction between lines and skirt=(0.03 to 0.1)

Fg= Gas force =$\frac{\pi}{4}\ D^2\ p_m$

The side thrust pressure, $p_s = \frac{side\ thrust\ fprce}{projected\ area} = \frac{Fs}{Ls \times D}$

Length of skirt (Ls) =$\frac{Fs}{Ps \times D}$ where D = Bore diameter.

LENGTH OF PISTON

The length of piston , Lp can be obtained as

Lp= Ls + Length of ring section + Top land

Empirically Lp= D to 1.5D

GUDGEON PIN or PISTON PIN

The piston pin should be made of case hardened alloy steel containing nickel, chromium, molybdenum etc with ultimate strength of 700 to 900 N/mm2 in order to with stand high gas pressure. The piston pin is designed based on the bearing pressure consideration.

Let l= length of piston pin, d= diameter of piston pin, pb= Allowable bearing pressure for piston pin=15 to 30 N/mm2.

Bearing strength of piston pin Fb=Bearing pressure x Projected area

Fb= pb.l.d

By equatning this bearing strength to gas force Gg, we get

Pb.l.d = Fg (there fore Fg = $\frac{\pi}{4}\ D^2\ p_m$)

Usually, l/d= 1.5 to 2 .

The piston pin is checked for bending as, the induced bending stress

$$\sigma_b = \frac{32\ M}{\pi d3} < \sigma_b$$

where M = Bending moment = $\frac{FgD}{8}$

D=Bore diameter

Fg= gas force

σ_b = Allowable bending stress= 84N/mm2 for case hardened steel and 140 N/mm2 for heat treated alloy steel

The gudgeon pin is fitted at a distance of (Ls/2) from open end where Ls is the skirt-length.

PISTON CLEARENCE

Proper clearance must be provided between the piston and liner to take care of thermal expansion and distortion under load. Usually the clearance may be between 0.04mm to 0.20 mm, depending upon the engine design and piston dia. small clearance may be adopted for the pistons cooled by oil(or) water.

Example 32.2. *Design a cast iron piston for a 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 per brake power per hour ; Higher calorific value of fuel = 42 × 10³ kJ/kg ; Speed = 2000 r.p.m.

Any other data required for the design may be assumed.

Solution. Given : D = 100 mm ; L = 125 mm = 0.125 m ; p = 5 N/mm² ; p_m = 0.75 N/mm²; η_m = 80% = 0.8 ; m = 0.15 kg / BP / h = 41.7×10^{-6} kg / BP / s; $HCV = 42 \times 10^3$ kJ / kg ; N = 2000 r.p.m.

The dimensions for various components of the piston are determined as follows :

1. *Piston head or crown*

The thickness of the piston head or crown is determined on the basis of strength as well as on the basis of heat dissipation and the larger of the two values is adopted.

We know that the thickness of piston head on the basis of strength,

$$t_H = \sqrt{\frac{3p.D^2}{16\,\sigma_t}} = \sqrt{\frac{3 \times 5(100)^2}{16 \times 38}} = 15.7 \text{ say } 16 \text{ mm}$$

...(Taking σ_t for cast iron = 38 MPa = 38 N/mm²)

Since the engine is a four stroke engine, therefore, the number of working strokes per minute,

$$n = N/2 = 2000/2 = 1000$$

and cross-sectional area of the cylinder,

$$A = \frac{\pi\,D^2}{4} = \frac{\pi\,(100)^2}{4} = 7855 \text{ mm}^2$$

We know that indicated power,

$$IP = \frac{p_m.L.A.n}{60} = \frac{0.75 \times 0.125 \times 7855 \times 1000}{60} = 12\,270 \text{ W}$$

$$= 12.27 \text{ kW}$$

∴ Brake power, $BP = IP \times \eta_m = 12.27 \times 0.8 = 9.8$ kW ...($\because \eta_m = BP/IP$)

We know that the heat flowing through the piston head,

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

$$= 0.05 \times 42 \times 10^3 \times 41.7 \times 10^{-6} \times 9.8 = 0.86 \text{ kW} = 860 \text{ W}$$

....(Taking C = 0.05)

∴ Thickness of the piston head on the basis of heat dissipation,

$$t_H = \frac{H}{12.56\,k\,(T_C - T_E)} = \frac{860}{12.56 \times 46.6 \times 220} = 0.0067 \text{ m} = 6.7 \text{ mm}$$

...($\because$ For cast iron , k = 46.6 W/m/°C, and $T_C - T_E$ = 220°C)

Taking the larger of the two values, we shall adopt

$$t_H = 16 \text{ mm}$$ **Ans.**

Since the ratio of L/D is 1.25, therefore a cup in the top of the piston head with a radius equal to 0.7 D (*i.e.* 70 mm) is provided.

2. *Radial ribs*

The radial ribs may be four in number. The thickness of the ribs varies from $t_H / 3$ to $t_H / 2$.

$\therefore$ Thickness of the ribs, $t_R = 16 / 3$ to $16 / 2 = 5.33$ to 8 mm

Let us adopt $t_R = 7$ mm **Ans.**

3. *Piston rings*

Let us assume that there are total four rings (*i.e.* $n_r = 4$) out of which three are compression rings and one is an oil ring.

We know that the radial thickness of the piston rings,

$$t_1 = D\sqrt{\frac{3 p_w}{\sigma_t}} = 100\sqrt{\frac{3 \times 0.035}{90}} = 3.4 \text{ mm}$$

...(Taking $p_w = 0.035$ N/mm^2, and $\sigma_t = 90$ MPa)

and axial thickness of the piston rings

$$t_2 = 0.7\, t_1 \text{ to } t_1 = 0.7 \times 3.4 \text{ to } 3.4 \text{ mm} = 2.38 \text{ to } 3.4 \text{ mm}$$

Let us adopt $t_2 = 3$ mm

We also know that the minimum axial thickness of the pistion ring,

$$t_2 = \frac{D}{10\, n_r} = \frac{100}{10 \times 4} = 2.5 \text{ mm}$$

Thus the axial thickness of the piston ring as already calculated (*i.e.* $t_2 = 3$ mm)is satisfactory. **Ans.**

The distance from the top of the piston to the first ring groove, *i.e.* the width of the top land,

$$b_1 = t_H \text{ to } 1.2\, t_H = 16 \text{ to } 1.2 \times 16 \text{ mm} = 16 \text{ to } 19.2 \text{ mm}$$

and width of other ring lands,

$$b_2 = 0.75\, t_2 \text{ to } t_2 = 0.75 \times 3 \text{ to } 3 \text{ mm} = 2.25 \text{ to } 3 \text{ mm}$$

Let us adopt $b_1 = 18$ mm ; and $b_2 = 2.5$ mm **Ans.**

We know that the gap between the free ends of the ring,

$$G_1 = 3.5\, t_1 \text{ to } 4\, t_1 = 3.5 \times 3.4 \text{ to } 4 \times 3.4 \text{ mm} = 11.9 \text{ to } 13.6 \text{ mm}$$

and the gap when the ring is in the cylinder,

$$G_2 = 0.002\, D \text{ to } 0.004\, D = 0.002 \times 100 \text{ to } 0.004 \times 100 \text{ mm}$$
$$= 0.2 \text{ to } 0.4 \text{ mm}$$

Let us adopt $G_1 = 12.8$ mm ; and $G_2 = 0.3$ mm **Ans.**

4. *Piston barrel*

Since the radial depth of the piston ring grooves (b) is about 0.4 mm more than the radial thickness of the piston rings (t_1), therefore,

$$b = t_1 + 0.4 = 3.4 + 0.4 = 3.8 \text{ mm}$$

We know that the maximum thickness of barrel,

$$t_3 = 0.03\, D + b + 4.5 \text{ mm} = 0.03 \times 100 + 3.8 + 4.5 = 11.3 \text{ mm}$$

and piston wall thickness towards the open end,

$$t_4 = 0.25\, t_3 \text{ to } 0.35\, t_3 = 0.25 \times 11.3 \text{ to } 0.35 \times 11.3 = 2.8 \text{ to } 3.9 \text{ mm}$$

Let us adopt $t_4 = 3.4$ mm

5. *Piston skirt*

Let l = Length of the skirt in mm.

We know that the maximum side thrust on the cylinder due to gas pressure (p),

$$R = \mu \times \frac{\pi D^2}{4} \times p = 0.1 \times \frac{\pi (100)^2}{4} \times 5 = 3928 \text{ N}$$

...(Taking $\mu = 0.1$)

We also know that the side thrust due to bearing pressure on the piston barrel (p_b),

$$R = p_b \times D \times l = 0.45 \times 100 \times l = 45\, l \text{ N}$$

...(Taking $p_b = 0.45$ N/mm^2)

From above, we find that

$$45\, l = 3928 \text{ or } l = 3928 / 45 = 87.3 \text{ say } 90 \text{ mm}$$ **Ans.**

∴ Total length of the piston ,

L = Length of the skirt + Length of the ring section + Top land

$= l + (4\, t_2 + 3b_2) + b_1$

$= 90 + (4 \times 3 + 3 \times 3) + 18 = 129$ say 130 mm **Ans.**

6. *Piston pin*

Let d_0 = Outside diameter of the pin in mm,

l_1 = Length of pin in the bush of the small end of the connecting rod in mm, and

p_{b1} = Bearing pressure at the small end of the connecting rod bushing in N/mm^2. It value for bronze bushing is taken as 25 N/mm^2.

We know that load on the pin due to bearing pressure

= Bearing pressure × Bearing area = $p_{b1} \times d_0 \times l_1$

$= 25 \times d_0 \times 0.45 \times 100 = 1125\, d_0$ N ...(Taking $l_1 = 0.45\, D$)

We also know that maximum load on the piston due to gas pressure or maximum gas load

$$= \frac{\pi D^2}{4} \times p = \frac{\pi (100)^2}{4} \times 5 = 39\,275 \text{ N}$$

From above, we find that

$1125\, d_0 = 39\,275$ or $d_0 = 39\,275 / 1125 = 34.9$ say 35 mm **Ans.**

The inside diameter of the pin (d_i) is usually taken as $0.6\, d_0$.

∴ $d_i = 0.6 \times 35 = 21$ mm **Ans.**

Let the piston pin be made of heat treated alloy steel for which the bending stress (σ_b)may be taken as 140 MPa. Now let us check the induced bending stress in the pin.

We know that maximum bending moment at the centre of the pin,

$$M = \frac{P.D}{8} = \frac{39\,275 \times 100}{8} = 491 \times 10^3 \text{ N-mm}$$

We also know that maximum bending moment (M),

$$491 \times 10^3 = \frac{\pi}{32}\left[\frac{(d_0)^4 - (d_i)^4}{d_0}\right]\sigma_b = \frac{\pi}{32}\left[\frac{(35)^4 - (21)^4}{35}\right]\sigma_b = 3664\,\sigma_b$$

∴ $\sigma_b = 491 \times 10^3 / 3664 = 134$ N/mm^2 or MPa

Since the induced bending stress in the pin is less than the permissible value of 140 MPa (*i.e.* 140 N/mm^2), therefore, the dimensions for the pin as calculated above (*i.e.* $d_0 = 35$ mm and $d_i = 21$ mm) are satisfactory.

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ENGINE-CYLINDER

ENGINE-CYLINDER:

At the time of compression and power strokes , more pressure is produced by the fuel-gas inside the cylinder. In order to with stand this high pressure, the cylinder ,cylinder head and piston should be fabricated with robust construction. The cylinder should also have the capacity to resist high temperature produced at the time of power stroke. It should be able to transfer the unused heat efficiency so as to escape from reaching the melting temperature of cylinder material.

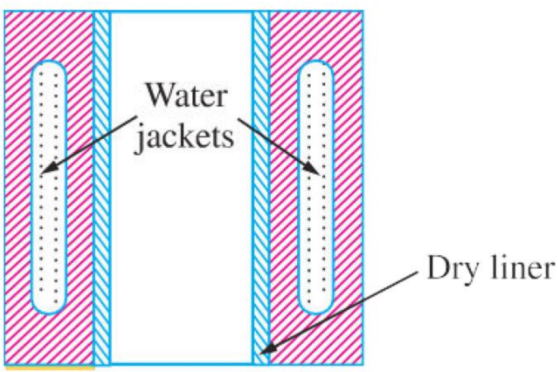
During operation of the engine, the piston slides inside the cylinder millions of times and thus the inside wall of the cylinder may be worn out. Since the cylinder is made as the integral part of the engine, the removal of the cylinder for repairing to rectify the wear by re-boring etc. will be very tendious and not economical and hence the cylinder is provided with another thin cylindrical piece called liner fitted concentric with the axis of the cylinder, by doing so worn out liner can easily replaced by new liner. Also by using strong liner, the good quality and strong material equal to that of liner material, need not be used for the entire cylinder and engine and thus the engine cost may be reduced. In the case of large sized engine, the cylinder with water jacket for cooling purpose.

MATERIALS : The cylinder and liner should be made of such a material which is strong enough to with stand high gas pressure and at the same time sufficiently hard enough to resist wear due to piston movement. It should also be capable of resisting thermal stresses due to heat flow through the liner-wall. In order to meet out the above requirements, the cylinder is usually made grey cast-iron and liners are made of nickel cast-iron, nickel chromium cast iron ,nickel chromium cast steel and so on.

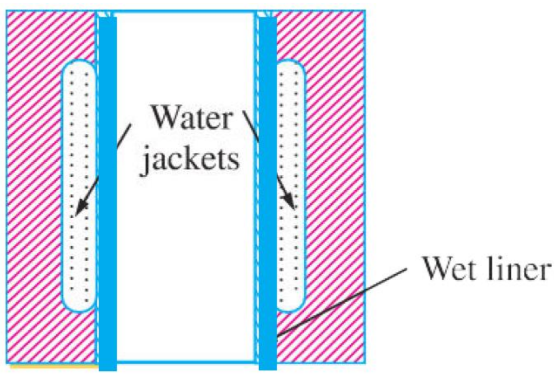
CYLINDER LINER

The cylinders are provided with cylinder liners so that in case of wear, they can be easily replaced. The cylinder liners are of the following two types :

1. Dry liner,   and   2. Wet liner.



a) **Dry liner**



b) wet liner

A cylinder liner which does not have any direct contact with the engine cooling water, is known as dry liner, as shown in Fig. (a). A cylinder liner which have its outer surface in direct contact with the engine cooling water, is known as wet liner, as shown in Fig. (b).The cylinder liners are made from good quality close grained cast iron (i.e. pearlitic cast iron), nickel cast iron, nickel chromium cast iron. In some cases, nickel chromium cast steel with molybdenum may be used. The inner surface of the liner should be properly heat-treated in order to obtain a hard surface to reduce wear.

DESIGN OF ENGINE CYLINDER

When designing a new engine, heat analysis must carried out to determine analytically the basic parameters of the engine under design with a sufficient degree of accuracy .This involves choice of data like engine type,power and speed, number and arrangement of cylinders, cylinder size, stroke bore ratio, piston speed and compression ratio etc.

Usually the piston speed and speed factor categorise the engine into high sped engine or low speed engine. The speed factor is defined as

$$C_S = \frac{0.3\,VN}{10^5}$$

Where V = piston speed in m/min

N = Crank shaft speed in r.p.m

The maximum piston speed for various applications is taken as follows :

Air craft engines 750 to 1000 m/min

Heave duty stationary engines 450 to 750 m/min

Large gas and diesel engines 300 to 450 m/min

The engines is classified as :

i) Low speed engine if Cs is less than 3

ii) Medium speed engine if Cs is between 3 to 9

iii) High speed engine if Cs is between 9 to 27

iv) Super speed engine if Cs is greater than 27.

The recommended piston speeds and the stroke-bore ratio for different types of engines are taking from jalal data book page number 15.12

Now considering the design of engine cylinder, when the gas expands inside the cylinder, two types of stresses will be induced in the walls of the cylinder liner which are

i) Tensile stress due to gas pressure and

ii) Thermal stress due to enormous heat.

By selecting the high hat resisting material, the thermal stresses can be reduced at the maximum extent.

The gas pressure also produces two types of tensile stresses in the cylinder namely) Longitudinal stress and b) Circumferential stress which act at right angle to each other. We have already known that when the pressure vessel like boiler or engine cylinder is subjected to gas pressure the induced circumferential stress (hoop stress) will be more than the induced longitudinal stress and hence the cylinder is based circumferential(hop) stress.

The wall thickness of cylinder is usually calculated by applying thin cylinder formula.

Then the wall thickness of cylinder,

$$t = \frac{pD}{2\sigma t} + C$$

where p= maximum pressure of fuel-gas inside the cylinder

D= Inside diameter of cylinder(or) bore dia

σ_t = Allowable tensile stress of cylinder material N/mm^2

= (50 to 60 N/mm^2 for C.I Engine & 80 to 100 N/mm^2 for steel)

Where C = 6 to 12 mm to account for blow holes corrosion and reboring etc.

The thickness of the cylinder wall usually varies from 4.5 mm to 25 mm, or more depending upon the cylinder size.

The other parameters are empirically found out as follows

The thickness of liner t_l = 0.03D to 0.035 D

The thickness of jacket wall is given by,

t_j = 0.032D to 1.6 mm

The water space between the outer cylinder wall and the inner jacket wall is given by

t_w = 0.08D to 6.5 mm

The cylinder is usually attached to the upper half of the crank case with the help of flanges, studs and nuts.

The flange thickness is obtained as,

t_f = (1.2 to 1.4) t

where t= cylinder thickness

The stud or bolt diameter can be evaluated by comparing the tensile strength of all bolts at their root diameters to the gas load such as

$$n \times \frac{\pi}{4} \times d_c \times \sigma_{tb} = \frac{\pi}{4} \times D^2 \times p$$

Where

d_c = core(i.e.,root) diameter of bolt or stud

σ_{tb} = Allowable tensile strength of bolt material = (80 N/mm^2 to 100 N/mm^2)

n= Number of studs = (0.01D to 0.02D) + 4

The thickness of cylinder head may be calculated as

$$t = kD\sqrt{\frac{p}{2\sigma t}}$$

where k = constant = 0.5

σ_{th} = Allowable tensile stress of head material = (30 to 50 N/mm^2).

Ex .1.

A four stroke diesel engine has the following specifications: Brake power = 5 kW ; Speed = 1200 r.p.m. ; Indicated mean effective pressure = 0.35 N / mm 2 ;Mechanical efficiency = 80 %.Determine : 1. bore and length of the cylinder ; 2. thickness of the cylinder head ; and 3. size of studs for the cylinder head.

Solution. Given: $B.P. = 5\text{kW} = 5000$ W ; $N = 1200$ r.p.m. or $n = N/2 = 600$; $p_m = 0.35$ N/mm^2; $\eta_m = 80\% = 0.8$

1. Bore and length of cylinder

Let D = Bore of the cylinder in mm,

A = Cross-sectional area of the cylinder = $\frac{\pi}{4} \times D^2$ mm^2

l = Length of the stroke in m.

$= 1.5\,D$ mm $= 1.5\,D / 1000$ m ....(Assume)

We know that the indicated power,

$$I.P = B.P. / \eta_m = 5000 / 0.8 = 6250 \text{ W}$$

We also know that the indicated power ($I.P.$),

$$6250 = \frac{p_m . l . A . n}{60} = \frac{0.35 \times 1.5D \times \pi D^2 \times 600}{60 \times 1000 \times 4} = 4.12 \times 10^{-3}\, D^3$$

...($\because$ For four stroke engine, $n = N/2$)

$\therefore$ $D^3 = 6250 / 4.12 \times 10^{-3} = 1517 \times 10^3$ or $D = 115$ mm **Ans.**

and $l = 1.5\,D = 1.5 \times 115 = 172.5$ mm

Taking a clearance on both sides of the cylinder equal to 15% of the stroke, therefore length of the cylinder,

$$L = 1.15\,l = 1.15 \times 172.5 = 198 \text{ say } 200 \text{ mm} \quad \textbf{Ans.}$$

2. Thickness of the cylinder head

Since the maximum pressure (p) in the engine cylinder is taken as 9 to 10 times the mean effective pressure (p_m), therefore let us take

$$p = 9\,p_m = 9 \times 0.35 = 3.15 \text{ N/mm}^2$$

We know that thickness of the cyclinder head,

$$t_h = D\sqrt{\frac{C.p}{\sigma_t}} = 115\sqrt{\frac{0.1 \times 3.15}{42}} = 9.96 \text{ say } 10 \text{ mm} \quad \textbf{Ans.}$$

...(Taking $C = 0.1$ and $\sigma_t = 42$ MPa = 42 N/mm^2)

3. Size of studs for the cylinder head

Let d = Nominal diameter of the stud in mm,

d_c = Core diameter of the stud in mm. It is usually taken as 0.84 d.

σ_t = Tensile stress for the material of the stud which is usually nickel steel.

n_s = Number of studs.

We know that the force acting on the cylinder head (or on the studs)

$$= \frac{\pi}{4} \times D^2 \times p = \frac{\pi}{4}(115)^2 3.15 = 32\ 702 \text{ N} \quad ...(i)$$

The number of studs (n_s) are usually taken between 0.01 D + 4 (*i.e.* 0.01 × 115 + 4 = 5.15) and 0.02 D + 4 (*i.e.* 0.02 × 115 + 4 = 6.3). Let us take $n_s = 6$.

We know that resisting force offered by all the studs

$$= n_s \times \frac{\pi}{4}(d_c)^2\ \sigma_t = 6 \times \frac{\pi}{4}(0.84d)^2\ 65 = 216\ d^2 \text{N} \quad ...(ii)$$

...(Taking $\sigma_t = 65$ MPa = 65 N/mm^2)

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

$$d^2 = 32\ 702 / 216 = 151 \text{ or } d = 12.3 \text{ mm}$$

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المحاضرة الاولى

DESIGN OF I.C ENGINE PARTS

DESIGN OF I.C ENGINE PARTS

INTRODUCTION :

The internal combustion engine , shortly called as I.C Engine is one type of engines in which the thermal and chemical energies of combustion are released inside the engine cylinder. There is another type of heat engine called External combustion engine. For example steam engine , combustion takes place outside the engine cylinder and the thermal energy is first transmitted to water outside the cylinder and steam is produced and then this energized steam is injected inside the cylinder for further operation.

The I.C engines are commonly operated by petrol even fuels like petrol, diesel and some times by gas. Depending on the properties of these fuels, the construction of concerned engines may be slightly changed from one to another. But , whatever be the type of engines, they have the following basic components which are i) Cylinder ii) Piston iii) Connecting rod iv) Crank shaft and v) flywheel. Apart from these main elements they have some auxiliary parts like push rod, cams, valves, springs and so on.

The I.C Engines are employed in many places like in small capacity power plants, Industries and laboratory machines and their outstanding applications are in the field of transportation like automobiles, air-crafts, rail-engines, ships and so on.

CLASSIFICATION OF I.C ENGINES

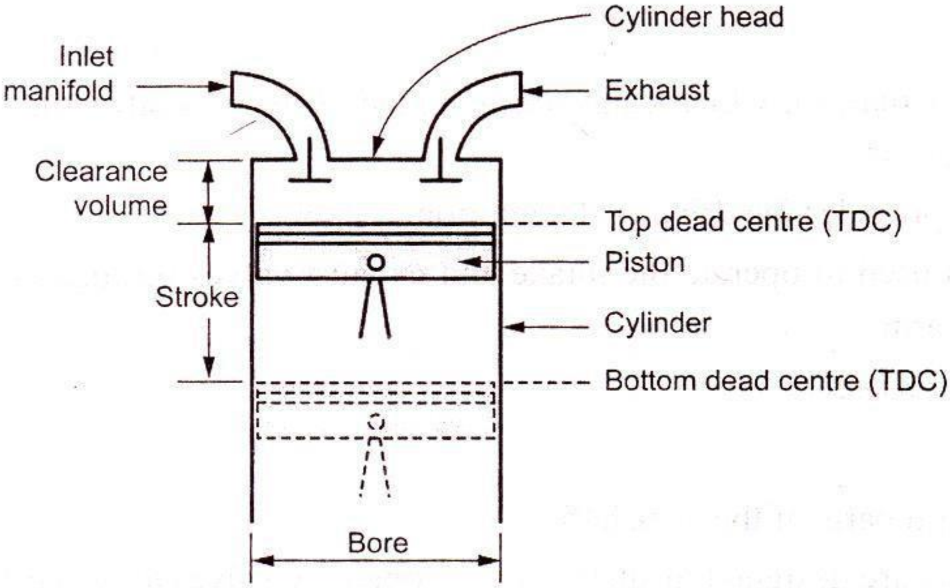
The I.C Engines are classified in many ways such as according to fuel used, method of ignition, work cycles, cylinder arrangement of applications etc:

a) According to fuel used
  i) Petrol Engine ii) Diesel Engine iii) Gas Engine
b) According to method of ignition
  i) Spark ignition engine ii) Compression ignition engine
c) According to working cycle
  i) Four stroke engine ii) Two stroke engine
d) According to cylinder arrangement
  i) Horizontal engine ii) Vertical engine iii) Inline engine
  iv) v-engine v) Radial engine

e) According to field of applications
  i) Automobile engine ii) Motor cycle engine iii) Aero engine
  iv) Locomotive engine v) Stationary engine

IC ENGINE TERMINOLOGY:

The following terms/Nomenclature associated with an engine are explained for the better understanding of the working principle of the IC engines



1. BORE: The nominal inside diameter of the engine cylinder is called bore.

2. TOP DEAD CENTRE (TDC): The extreme position of the piston at the top of the cylinder of the vertical engine is called top dead centre (TDC), Incase of horizontal engines. It is known as inner dead centre (IDC).

3. BOTTOM DEAD CENTRE (BDC): The extreme position of the piston at the bottom of the cylinder of the vertical engine called bottom dead centre (BDC). In case of horizontal engines, it is known as outer dead center (ODC).

4. STROKE: The distance travelled by the piston from TDC to BDC is called stroke. In other words, the maximum distance travelled by the piston in the cylinder in one direction is known as stroke. It is equal to twice the radius of the crank.

5. CLEARANCE VOLUME (Vc): The volume contained in the cylinder above the top of the piston, when the piston is at top dead centre is called the clearance volume.

6. SWEPT VOLUME (Vs): The volume swept by the piston during one stroke is called the swept volume or piston displacement. Swept volume is the volume covered by the piston while moving from TDC to BDC.

i.e Swept volume = Total volume – clearance volume

7. COMPRESSION RATIO (rc): Compression ratio is a ratio of the volume when the piston is at bottom dead centre to the volume when the piston is at top dead centre.

Mathematically,

$$Compression\ ratio = \frac{maximum\ cylinder\ volume}{minimum\ cylinder\ volume} = \frac{swept\ volume + clearance\ volume}{clearance\ volume}$$

The compression ratio varies from 5 : 1 to 10 : I for petrol engines and from 12:1 to 22 : I for diesel engines.

Sl.No	Classification Criteria	Types
1.	No of Strokes per cycle	1. Four Stroke Engine 2. Two Stroke Engine
2.	Types of Fuel Used	1. Petrol or Gasoline Engine 2. Diesel Engine 3. Gas Engine 4. Bi-Fuel Engine
3.	Nature of Thermodynamic Cycle	1. Otto Cycle Engine 2. Diesel Cycle Engine 3. Dual Combustion Cycle Engine
4.	Method of Ignition	1. Spark Ignition (SI) Engine 2. Compression Ignition (CI) Engine
5.	No of Cylinders	1. Single Cylinder Engine 2. Multi Cylinder Engine
6.	Arrangement of Cylinders	1. Horizontal Engine 2. Vertical Engine 3. V – Type Engine 4. Radial Engine 5. Inline Engine 6. Opposed Cylinder Engine 7. Opposed Piston Engine

Sl.No	Classification Criteria	Types
7.	Cooling System	1. Air Cooled Engine 2. Water Cooled Engine
8.	Lubrication System	1. Wet Sump Lubrication System 2. Dry Sump Lubrication System
9.	Speed of the Engine.	1. Slow Speed Engine 2. Medium Speed Engine 3. High Speed Engine
10.	Location of Valves	1. Over Head Valve Engine 2. Side Valve Engine

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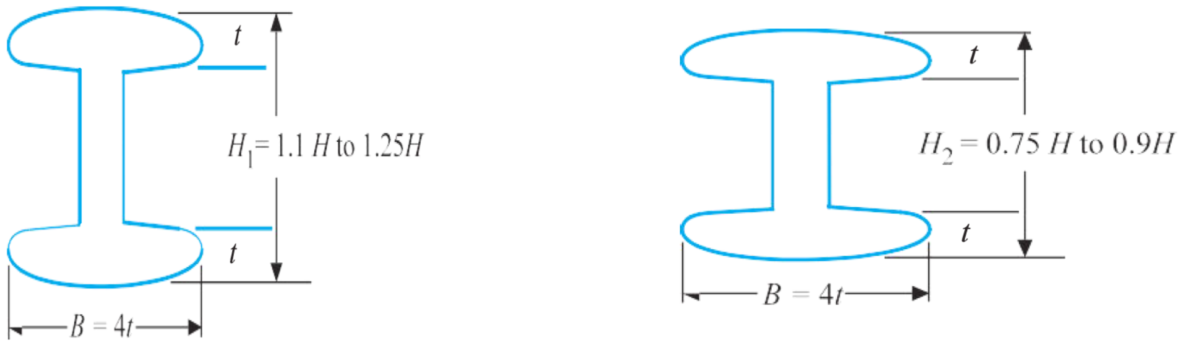
المحاضرة الرابعة

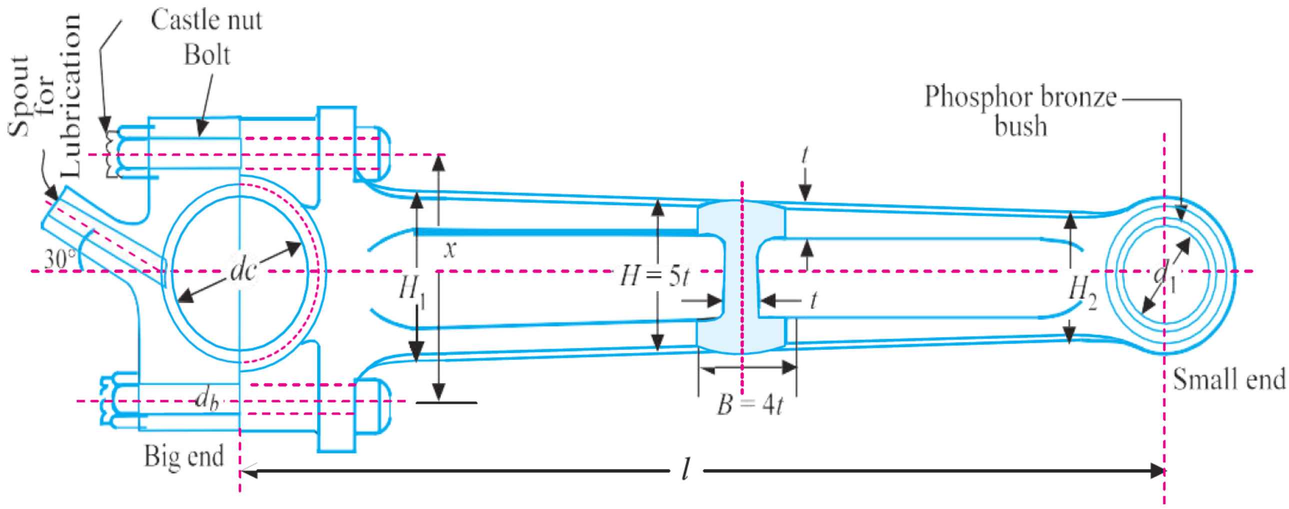
DESIGN OF A CONNECTING ROD

DESIGN OF A CONNECTING ROD

The connecting rod is the intermediate member between the piston and the crankshaft. Its primary function is to transmit the push and pull from the piston pin to the crankpin and thus convert the reciprocating motion of the piston into the rotary motion of the crank. The usual form of the connecting rod in internal combustion engines is shown in Fig. 32.9. It consists of a long shank, a small end and a big end. The cross-section of the shank may be rectangular, circular, tubular, I-section or H-



section. Generally circular section is used for low speed engines while I-section is preferred for high speed engines



The *length of the connecting rod (l) depends upon the ratio of l / r, where r is the radius of crank. It may be noted that the smaller length will decrease the ratio l / r. This increases the angularity of the connecting rod which increases the side thrust of the piston against the cylinder liner which in turn increases the wear of the liner. The larger length of the connecting rod will increase the ratio l / r. This decreases the angularity of the connecting rod and thus decreases the side thrust and the resulting wear of the cylinder. But the larger length of the connecting rod increases the overall height of the engine. Hence, a compromise is made and the ratio l / r is generally kept as 4 to 5.

The small end of the connecting rod is usually made in the form of an eye and is provided with a bush of phosphor bronze. It is connected to the piston by means of a piston pin.

The big end of the connecting rod is usually made split (in two **halves) so that it can be mounted easily on the crankpin bearing shells. The split cap is fastened to the big end with two cap bolts. The bearing shells of the big end are made of steel, brass or bronze with a thin lining (abou0.75 mm) of white metal or babbit metal. The wear of the big end bearing is allowed for by inserting thin metallic strips (known as shims) about 0.04 mm thick between the cap and the fixed half of the connecting rod. As the wear takes place, one or more strips are removed and the bearing is trued up.

The connecting rods are usually manufactured by drop forging process and it should have adequate strength, stiffness and minimum weight. The material mostly used for connecting rods varies from mild carbon steels (having 0.35 to 0.45 percent carbon) to alloy steels (chrome-nickel or chrome- molybdenum steels). The carbon steel having 0.35 percent carbon has an ultimate tensile strength of about 650 MPa when properly heat treated and a carbon steel with 0.45 percent carbon has a ultimate tensile strength of 750 MPa. These steels are used for connecting rods of industrial engines. The alloy steels have an ultimate tensile strength of about 1050 MPa and are used for connecting rods of aeroengines and automobile engines.

The bearings at the two ends of the connecting rod are either splash lubricated or pressure lubricated. The big end bearing is usually splash lubricated while the small end bearing is pressure lubricated. In the splash lubrication system, the cap at the big end is provided with a dipper or spout and set at an angle in such a way that when the connecting rod moves downward, the spout will dip into the lubricating oil contained in the sump. The oil is forced up the spout and then to the big end bearing. Now when the connecting rod moves upward, a splash of oil is produced by the spout. This splashed up lubricant find its way into the small end bearing through the widely chamfered holes provided on the upper surface of the small end.

In the pressure lubricating system, the lubricating oil is fed under pressure to the big end bearing through the holes drilled in crankshaft, crankwebs and crank pin. From the big end bearing, the oil is fed to small end bearing through a fine hole drilled in the shank of the connecting rod. In some cases, the small end bearing is lubricated by the oil scrapped from the walls of the cyinder liner by the oil scraper rings.

FORCES ACTING ON THE CONNECTING ROD

The various forces acting on the connecting rod are as follows :

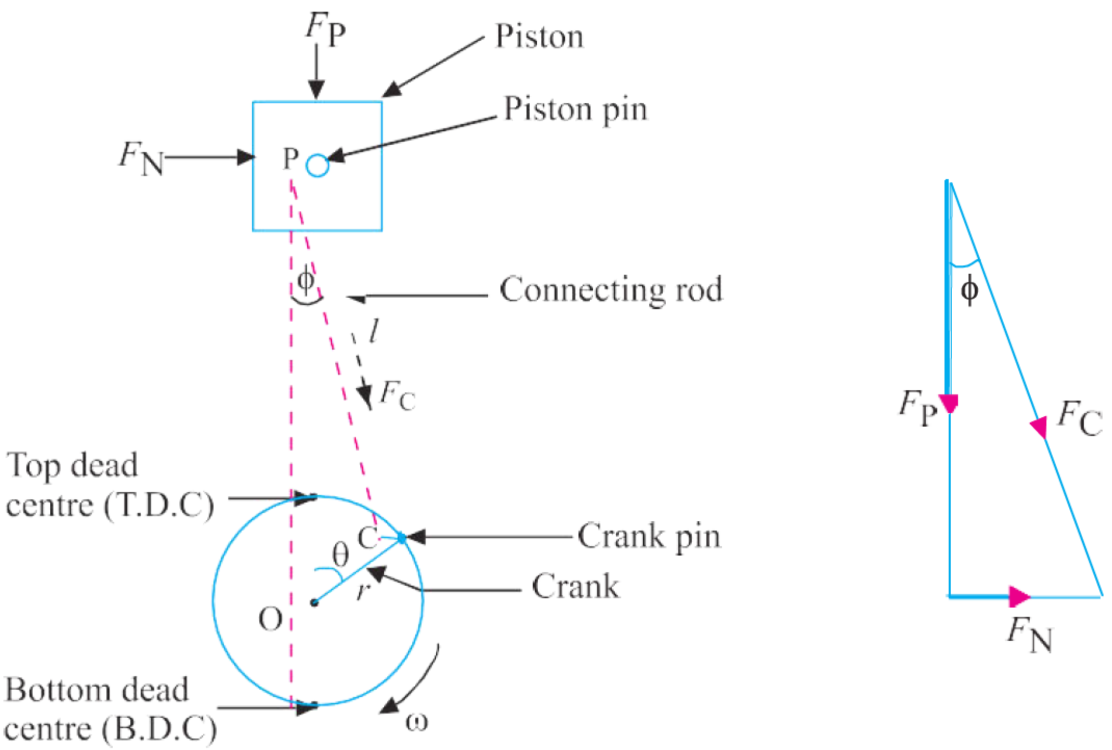
1. Force on the piston due to gas pressure and inertia of the reciprocating parts,
2. Force due to inertia of the connecting rod or inertia bending forces,
3. Force due to friction of the piston rings and of the piston, and

4. Force due to friction of the piston pin bearing and the crankpin bearing.

We shall now derive the expressions for the forces acting on a vertical engine, as discussed below.

1. Force on the piston due to gas pressure and inertia of reciprocating parts

Consider a connecting rod PC as shown in Fig. 32.10.



Let p = Maximum pressure of gas,

D = Diameter of piston,

A_P = Cross-section area of piston

m_R = Mass of reciprocating parts,

r= radius of crank shaft

ω = Angular speed of crank,

ϕ = Angle of inclination of the connecting rod with the line of stroke,

Ɵ= Angle of inclination of the crank from top dead centre,

r = Radius of crank,

l = Length of connecting rod, and

n = Ratio of length of connecting rod to radius of crank = l / r.

Fp= Force acting on the piston= p x Ap

Fc= Force acting on the connecting rod

Fi= Inertia force due to weight of the reciprocating parts

We know that the force on the piston due to pressure of gas,

Fp = Pressure × Area = p . Ap = p × π D^2 /4

And the inertia force of the reciprocating parts

Fi= mass x Acceleration

$$= \frac{Wr}{g} \times \omega^2 r \left(\cos\theta + (\cos 2\theta)/n\right)$$

The net load acting on the connecting rod, $F_C = F_P \pm F_i$

The –ve sign is used when the piston moves from TDC to BDC and +ve sign is used when the piston moves from BDC to TDC.

When weight of the reciprocating parts is to be considered ,then

$F_C = F_P \pm F_i \pm Wr$

The actual axial load acting on the connecting rod will be more than the next load due to the angularity of the rod.

Now ,the force acting on the connecting rod at any instant is given by

$$F_c = \frac{Fp - Fi}{\cos \emptyset} = \frac{Fp}{\cos \emptyset}$$

Normally inertia force due to the weight of reciprocating parts is very small, it can be neglected when designing connecting rod

$$F_c = \frac{Fp}{\cos \emptyset}$$

Since the piston is under reciprocating action, the connecting rod will be subjected to maximum force when the crank angle θ -900 and for other positions, the force values are reduced and for θ -00 and θ =1800, the forces are zeros. Also the inclination of the connecting rod max when θ -900. Hence the maximum force acting on the connecting trod, is given by

$$F_{c_{max}} = \frac{Fp}{\cos \emptyset}$$

In general, n should be at least 3

Hence for n=l/r=3, Fc=1.06Fp

N=4, Fc=1.03Fp

N=5, Fc=1.02Fp

Maximum bending moment due to inertia force is given by the relation $M_{max} = m.\omega^2.\mathrm{r}.\frac{l}{9\sqrt{3}}$

Where m= mass of connecting rod

ω = Angular speed in rad/s

L= length of connecting rod

R = radius of crank

The maximum bending stress = $\frac{\mathrm{Mmax}}{Z}$

Where Z = section modulus.

DIMENSIONS OF CONNECTING ROD ENDS

Now the other parts of connecting rod such as its small end, big end and bolts are designed as follows

The small end is made as solid eye without any split and is provided with brass bushes inside the eye and the big end is split and the top cap is joined with the remaining parts of connecting rod by means of bolts. By this set up the connecting rod can be dismantled without removing the crank shaft. In the big end also, the brass bushes of split type are employed.

The parameters of small end and big end are determined based on the bearing pressures

Let l1,d1=length and diameter of piston (i.e small end respectively)

L2,d2 = Length and diameter of crank pin (i.e bidend respectively)

Pb1,pb2 = Design bearing pressures for the small end and big end respectively

Bearingload applied on the piston pin(i.e small end) is given by

F1 = pb1.l1.d1

And the bearing load applied on the crank pin(i.e big end) is given by F2 = pb2.l1.d2

Usually the design bearing pressure for the small end and big end may be taken as,

Pb1 = 12.5 to 15.4 N/mm^2

Pb2 = 10.8 to 12.6 N/mm^2

Similarly, the ratio of length to diameter for small end and big end may be assumed as,

$L1/d1=1.5$ to 2, $l2/d2= 1.0$ to 1.25

Usually, low design stress value is selected for big end than that for small end.

The biggest load to be carried by these for bearings containing piston pin and crank pin is the maximum compressive load produced by the gas pressure neglecting the inertia force due to its small value

At the same time, the bolts are designed based on the inertia force of the reciprocating parts which is given by

Inertia force Fi= $m\, r\omega^2 \left(\cos\theta + \frac{\cos 2\theta}{n}\right)$

$$n = \frac{l}{r} = \frac{length\ of\ connecting\ Road}{crank\ radius}$$

The maximum inertia force will be obtained when the crank shaft is at dead centre position, i.e., at θ = 0.

By equating this maximum inertia force to the tensile strength of bolts and their core diameters, the size of bolts may be determined.

i.e for two bolts F_{im}= 2 x $\frac{\pi}{4} d_c^2 \times S_t$

The nominal diameter may be selected from the manufacture's table (uaually dc=0.84 db, where db is the nominal dia of bolt) .

The cap is usually treated as a beam freely supported at the bolts centre's and loaded in a manner intermediate between uniformly distributed load and centrally concentrated loaded.

Maximum bending moment at the centre of cap is given by $M = wl^{l} / 6$

Where w = maximum load equal to inertia force of reciprocating parts = Fim

Hence $M = Fiml^{l}/6$

l^{l} = Dstance between bolts centers

= Diameter of crank pin + (2 x wall thickness of bush) + dia of bolt + some extra marginal thickness.

Width of cap may be calculated as,

b = length of crank pin – 2 x flange thickness of bush

usually, the wall thickness and flange thickness of bush may be taken as about 5 mm.
Bending stress induced in the cap = Sbe = M / Z .

Where Z = Section modulus of the cap.

$Z = 1/6 .b.t_c^2$

Where t_c = Thickness of cap.

By comparing this induced bending stress with the design stress, the thickness of cap may be evaluated.

DESIGN PROCEDURE FOR CONNECTING ROD :

For the design of connecting rod, the following steps may be observed.

1. From the statement of problem, note the pressure of steam or gas, length of connecting rod, crank radius etc,. Then select suitable material usually mild steel for the connecting rod and find its design stresses. Assume the essential non given data suitably based on the working conditions.
2. Select I-section connecting rod if possible and determine its moment of inertia about x-axis and y-axis.
3. Equate the steam force with buckling strength of connecting rod using Rankine's formula and determine the dimensions of connecting rod.
4. Calculate the maximum bending stress and then compare it with design stress of the connecting rod for checking.

SLENDERNESS RATIO:

It is the ratio of the length of column (l) to its least radius of gyration (k)

Slenderness ratio =l/k

If l/k < 40 – then design of connecting rod be based on compressive load.

If l/k > 40 – then design of connecting rod may be based on Buckling load.

BUCKLING LOAD or CRIPPLING LOAD

The piston rod and connecting rod are designed mainly based on compressive failure load. Since the length of rods are more, they can buckle during compression, which is also considered as functional failure. That is , the compressive load which causes buckling of piston rod or connecting rod is called as buckling load or crippling load. For proper functioning without buckling the piston rod or connecting rod should be subjected to a compressive load with is less than crippling load.

When the connecting rod or piston rod are subjected to compressive load, they may fracture when the applied compressive load is more than their resisting compressive strength. At the same time, if the length of rods have been increased beyond certain limit with respect to their gross sectional dimensions (i.e $l/k > 40$) the rods may buckle for lower values of compressive load known as buckling load. This buckling load also considered as functional failure. Usually design of connecting & piston rod are designed based on buckling load.

Example 32.3. *Design a connecting rod for an I.C. engine running at 1800 r.p.m. and developing a maximum pressure of 3.15 N/mm². The diameter of the piston is 100 mm ; mass of the reciprocating parts per cylinder 2.25 kg; length of connecting rod 380 mm; stroke of piston 190 mm and compression ratio 6 : 1. Take a factor of safety of 6 for the design. Take length to diameter ratio for big end bearing as 1.3 and small end bearing as 2 and the corresponding bearing pressures as 10 N/mm² and 15 N/mm². The density of material of the rod may be taken as 8000 kg/m³ and the allowable stress in the bolts as 60 N/mm² and in cap as 80 N/mm². The rod is to be of I-section for which you can choose your own proportions.*

Draw a neat dimensioned sketch showing provision for lubrication. Use Rankine formula for which the numerator constant may be taken as 320 N/mm² and the denominator constant 1 / 7500.

Solution. Given : N = 1800 r.p.m. ; p = 3.15 N/mm² ; D = 100 mm ; m_R = 2.25 kg ; l = 380 mm = 0.38 m ; Stroke = 190 mm ; *Compression ratio = 6 : 1 ; $F.S.$ = 6.

The connecting rod is designed as discussed below :

1. *Dimension of I- section of the connecting rod*

Let us consider an *I*-section of the connecting rod, as shown in Fig. 32.14 (*a*), with the following proportions :

Flange and web thickness of the section = t

Width of the section, $B = 4t$

and depth or height of the section,

$$H = 5t$$

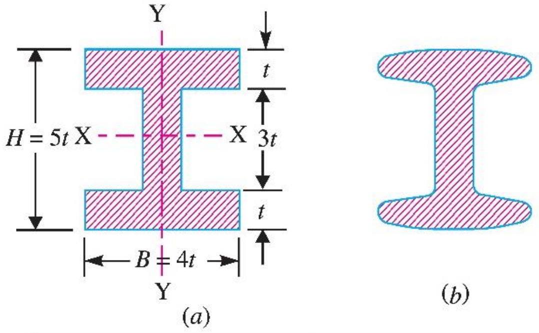


Fig. 32.14

First of all, let us find whether the section chosen is satisfactory or not.

We have already discussed that the connecting rod is considered like both ends hinged for buckling about *X*-axis and both ends fixed for buckling about *Y*-axis. The connecting rod should be equally strong in buckling about both the axes. We know that in order to have a connecting rod equally strong about both the axes,

$$I_{xx} = 4\, I_{yy}$$

where I_{xx} = Moment of inertia of the section about *X*-axis, and

I_{yy} = Moment of inertia of the section about *Y*-axis.

In actual practice, I_{xx} is kept slightly less than $4\, I_{yy}$. It is usually taken between 3 and 3.5 and the connecting rod is designed for buckling about *X*-axis.

Now, for the section as shown in Fig. 32.14 (*a*), area of the section,

$$A = 2\,(4\,t \times t\,) + 3t \times t = 11\, t^2$$

$$I_{xx} = \frac{1}{12}\left[4t(5t)^3 - 3t \times (3t)^3\right] = \frac{419}{12}t^4$$

and

$$I_{yy} = 2 \times \frac{1}{12} \times t(4t)^3 + \frac{1}{12} \times 3t \times t^3 = \frac{131}{12}t^4$$

$$\therefore \quad \frac{I_{xx}}{I_{yy}} = \frac{419}{12} \times \frac{12}{131} = 3.2$$

Since $\frac{I_{xx}}{I_{yy}} = 3.2,$ therefore the section chosen in quite satisfactory.

Now let us find the dimensions of this *I*-section. Since the connecting rod is designed by taking the force on the connecting rod (F_C) equal to the maximum force on the piston (F_L) due to gas pressure, therefore,

$$F_C = F_L = \frac{\pi D^2}{4} \times p = \frac{\pi(100)^2}{4} \times 3.15 = 24\ 740 \text{ N}$$

We know that the connecting rod is designed for buckling about *X*-axis (*i.e.* in the plane of motion of the connecting rod) assuming both ends hinged. Since a factor of safety is given as 6, therefore the buckling load,

$$W_B = F_C \times F.\,S. = 24\ 740 \times 6 = 148\ 440 \text{ N}$$

We know that radius of gyration of the section about X-axis,

$$k_{xx} = \sqrt{\frac{I_{xx}}{A}} = \sqrt{\frac{419t^4}{12} \times \frac{1}{11t^2}} = 1.78\ t$$

Length of crank,

$$r = \frac{\text{Stroke of piston}}{2} = \frac{190}{2} = 95 \text{ mm}$$

Length of the connecting rod,

$$l = 380 \text{ mm} \qquad \text{...(Given)}$$

∴ Equivalent length of the connecting rod for both ends hinged,

$$L = l = 380 \text{ mm}$$

Now according to Rankine's formula, we know that buckling load (W_B),

$$148\ 440 = \frac{\sigma_c . A}{1 + a\left(\frac{L}{k_{xx}}\right)^2} = \frac{320 \times 11\ t^2}{1 + \frac{1}{7500}\left(\frac{380}{1.78\ t}\right)^2}$$

... (It is given that σ_c = 320 MPa or N/mm^2 and $a = 1 / 7500$)

$$\frac{148\ 440}{320} = \frac{11\ t^2}{1 + \frac{6.1}{t^2}} = \frac{11\ t^4}{t^2 + 6.1}$$

$$464\ (t^2 + 6.1) = 11\ t^4$$

or $$t^4 - 42.2\ t^2 - 257.3 = 0$$

∴ $$t^2 = \frac{42.2 \pm \sqrt{(42.2)^2 + 4 \times 257.3}}{2} = \frac{42.2 \pm 53}{2} = 47.6$$

... (Taking +ve sign)

or $$t = 6.9 \text{ say } 7 \text{ mm}$$

Thus, the dimensions of I-section of the connecting rod are :

Thickness of flange and web of the section

$$= t = 7 \text{ mm} \quad \textbf{Ans.}$$

Width of the section, $B = 4\ t = 4 \times 7 = 28$ mm **Ans.**

and depth or height of the section,

$$H = 5\ t = 5 \times 7 = 35 \text{ mm} \quad \textbf{Ans.}$$

These dimensions are at the middle of the connecting rod. The width (B) is kept constant throughout the length of the rod, but the depth (H) varies. The depth near the big end or crank end is kept as $1.1H$ to $1.25H$ and the depth near the small end or piston end is kept as $0.75H$ to $0.9H$. Let us take

Depth near the big end,

$$H_1 = 1.2H = 1.2 \times 35 = 42 \text{ mm}$$

and depth near the small end,

$$H_2 = 0.85H = 0.85 \times 35 = 29.75 \text{ say } 30 \text{ mm}$$

∴ Dimensions of the section near the big end

$$= 42 \text{ mm} \times 28 \text{ mm} \quad \textbf{Ans.}$$

and dimensions of the section near the small end

$$= 30 \text{ mm} \times 28 \text{ mm} \quad \textbf{Ans.}$$

Since the connecting rod is manufactured by forging, therefore the sharp corners of *I*-section are rounded off, as shown in Fig. 32.14 (*b*), for easy removal of the section from the dies.

2. Dimensions of the crankpin or the big end bearing and piston pin or small end bearing

Let d_c = Diameter of the crankpin or big end bearing,

l_c = length of the crankpin or big end bearing $= 1.3\ d_c$ ...(Given)

p_{bc} = Bearing pressure = 10 N/mm² ...(Given)

We know that load on the crankpin or big end bearing

$$= \text{Projected area} \times \text{Bearing pressure}$$

$$= d_c \,.\, l_c \,.\, p_{bc} = d_c \times 1.3\ d_c \times 10 = 13\ (d_c)^2$$

Since the crankpin or the big end bearing is designed for the maximum gas force (F_L), therefore, equating the load on the crankpin or big end bearing to the maximum gas force, *i.e.*

$$13\ (d_c)^2 = F_L = 24\ 740 \text{ N}$$

$$\therefore \quad (d_c)^2 = 24\ 740 / 13 = 1903 \quad \text{or} \quad d_c = 43.6 \text{ say } 44 \text{ mm} \quad \textbf{Ans.}$$

and

$$l_c = 1.3\ d_c = 1.3 \times 44 = 57.2 \text{ say } 58 \text{ mm} \quad \textbf{Ans.}$$

The big end has removable precision bearing shells of brass or bronze or steel with a thin lining (1mm or less) of bearing metal such as babbit.

Again, let d_p = Diameter of the piston pin or small end bearing,

l_p = Length of the piston pin or small end bearing $= 2d_p$ ...(Given)

p_{bp} = Bearing pressure = 15 N/mm² ..(Given)

We know that the load on the piston pin or small end bearing

$$= \text{Project area} \times \text{Bearing pressure}$$

$$= d_p \,.\, l_p \,.\, p_{bp} = d_p \times 2\ d_p \times 15 = 30\ (d_p)^2$$

Since the piston pin or the small end bearing is designed for the maximum gas force (F_L), therefore, equating the load on the piston pin or the small end bearing to the maximum gas force, *i.e.*

$$30\ (d_p)^2 = 24\ 740 \text{ N}$$

$$\therefore \quad (d_p)^2 = 24\ 740 / 30 = 825 \quad \text{or} \quad d_p = 28.7 \text{ say } 29 \text{ mm} \quad \textbf{Ans.}$$

and

$$l_p = 2\ d_p = 2 \times 29 = 58 \text{ mm} \quad \textbf{Ans.}$$

The small end bearing is usually a phosphor bronze bush of about 3 mm thickness.

3. *Size of bolts for securing the big end cap*

Let d_{cb} = Core diameter of the bolts,

σ_t = Allowable tensile stress for the material of the bolts

$= 60 \text{ N/mm}^2$ ...(Given)

and n_b = Number of bolts. Generally two bolts are used.

We know that force on the bolts

$$= \frac{\pi}{4}(d_{cb})^2 \sigma_t \times n_b = \frac{\pi}{4}(d_{cb})^2 \; 60 \times 2 = 94.26\,(d_{cb})^2$$

The bolts and the big end cap are subjected to tensile force which corresponds to the inertia force of the reciprocating parts at the top dead centre on the exhaust stroke. We know that inertia force of the reciprocating parts,

$$F_I = m_R \cdot \omega^2 \cdot r \left(\cos\theta + \frac{\cos 2\theta}{l/r} \right)$$

We also know that at top dead centre on the exhaust stroke, $\theta = 0$.

$$\therefore \quad F_I = m_R \cdot \omega^2 \cdot r\left(1 + \frac{r}{l}\right) = 2.25\left(\frac{2\pi \times 1800}{60}\right)^2 0.095\left(1 + \frac{0.095}{0.38}\right) \text{N}$$

$$= 9490 \text{ N}$$

Equating the inertia force to the force on the bolts, we have

$$9490 = 94.26\,(d_{cb})^2 \quad \text{or} \quad (d_{cb})^2 = 9490 / 94.26 = 100.7$$

$$\therefore \quad d_{cb} = 10.03 \text{ mm}$$

and nominal diameter of the bolt,

$$d_b = \frac{d_{cb}}{0.84} = \frac{10.03}{0.84} = 11.94$$

say 12 mm **Ans.**

4. *Thickness of the big end cap*

Let t_c = Thickness of the big end cap,

b_c = Width of the big end cap. It is taken equal to the length of the crankpin or big end bearing (l_c)

= 58 mm (calculated above)

σ_b = Allowable bending stress for the material of the cap

$= 80 \text{ N/mm}^2$ ...(Given)

The big end cap is designed as a beam freely supported at the cap bolt centres and loaded by the inertia force at the top dead centre on the exhaust stroke (*i.e.* F_I when $\theta = 0$). Since the load is assumed to act in between the uniformly distributed load and the centrally concentrated load, therefore, maximum bending moment is taken as

$$M_C = \frac{F_I \times x}{6}$$

where x = Distance between the bolt centres

$= $ Dia. of crank pin or big end bearing + 2 × Thickness of bearing liner + Nominal dia. of bolt + Clearance

$= (d_c + 2 \times 3 + d_b + 3)$ mm $= 44 + 6 + 12 + 3 = 65$ mm

∴ Maximum bending moment acting on the cap,

$$M_C = \frac{F_I \times x}{6} = \frac{9490 \times 65}{6} = 102\ 810 \text{ N-mm}$$

Section modulus for the cap

$$Z_C = \frac{b_c (t_c)^2}{6} = \frac{58 (t_c)^2}{6} = 9.7\ (t_c)^2$$

We know that bending stress (σ_b),

$$80 = \frac{M_C}{Z_C} = \frac{102\ 810}{9.7\ (t_c)^2} = \frac{10\ 600}{(t_c)^2}$$

∴ $(t_c)^2 = 10\ 600 / 80 = 132.5$ or $t_c = 11.5$ mm **Ans.**

Let us now check the design for the induced bending stress due to inertia bending forces on the connecting rod (*i.e.* whipping stress).

We know that mass of the connecting rod per metre length,

$$m_1 = \text{Volume} \times \text{density} = \text{Area} \times \text{length} \times \text{density}$$

$$= A \times l \times \rho = 11t^2 \times l \times \rho \qquad ...(\because A = 11t^2)$$

$$= 11(0.007)^2\ (0.38)\ 8000 = 1.64 \text{ kg}$$

$$...[\because \rho = 8\ 000 \text{ kg/m}^3 \text{ (given)}]$$

∴ Maximum bending moment,

$$M_{max} = m \cdot \omega^2 \cdot r \times \frac{l}{9\sqrt{3}} = m_1 \cdot \omega^2 \cdot r \times \frac{l^2}{9\sqrt{3}} \qquad ...(\because m = m_1 \cdot l)$$

$$= 1.64 \left(\frac{2\pi \times 1800}{60} \right)^2 (0.095) \frac{(0.38)^2}{9\sqrt{3}} = 51.3 \text{ N-m}$$

$$= 51\ 300 \text{ N-mm}$$

and section modulus, $$Z_{xx} = \frac{I_{xx}}{5t/2} = \frac{419\ t^4}{12} \times \frac{2}{5\ t} = 13.97\ t^3 = 13.97 \times 7^3 = 4792 \text{ mm}^3$$

∴ Maximum bending stress (induced) due to inertia bending forces or whipping stress,

$$\sigma_{b(max)} = \frac{M_{max}}{Z_{xx}} = \frac{51\ 300}{4792} = 10.7 \text{ N/mm}^2$$

Since the maximum bending stress induced is less than the allowable bending stress of 80 N/mm^2, therefore the design is safe.

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المحاضرة السابعة

Gears

Gears

تستعمل التروس لنقل الحركة الدائرية بقدرة ودقة عالية، أي بدون انزلاق مقارنة بأجهزة نقل الحركة الأخرى (الأحزمة والسلاسل). كما تستخدم التروس لتخفيض أو زيادة السرعة المنقولة.

Classification of gears (Types of gears):

2.3 <u>Type of toothed gears</u>: — يمكن تصنيف التروس كما يلي:

1- According to the teeth shape: — 1- بالنسبة إلى نوع الأسنان:

a) Spur (straight) gears. — التروس العدلة

b) Helical (inclined) gears & spiral. — التروس الحلزوني

c) Bevel (curved) gears. — التروس المائلة

2- According the axes of shafts:

2- بالنسبة إلى وضع محاور الأعمدة التي تتركب عليها:

a) Parallel shaft (spur). — المحاور المتوازية

b) Intersecting shaft (bevel gear). — المحاور المتقاطعة

c) Non parallel and non intersecting shaft (skew shaft) (hedical).

3- According to the speed: — 3- بالنسبة إلى سرعة التروس:

a) Low speed (less than 3 m/sec). — سرعة واطئة

b) Medium speed (3-15) m/sec. — سرعة متوسطة

c) High speed (more than 15 m/sec). — سرعة عالية

4- According to the gearing : — 4- بالنسبة إلى نوع التعشيق:

a) External gears. — تعشيق خارجي

b) Internal gears. — تعشيق داخلي

c) Rack gears

2.4 Technical gear terms (Geometry of Gear)

1- Pitch circle: It is an imaginary circle drawn through the points where the teeth make contact. (d)
2- Circular pitch (pitch) "P": It is the distance from a point on one tooth to the corresponding point on an adjacent tooth measured on the pitch circle.

$$p = \frac{\pi d}{Z} \quad ,$$

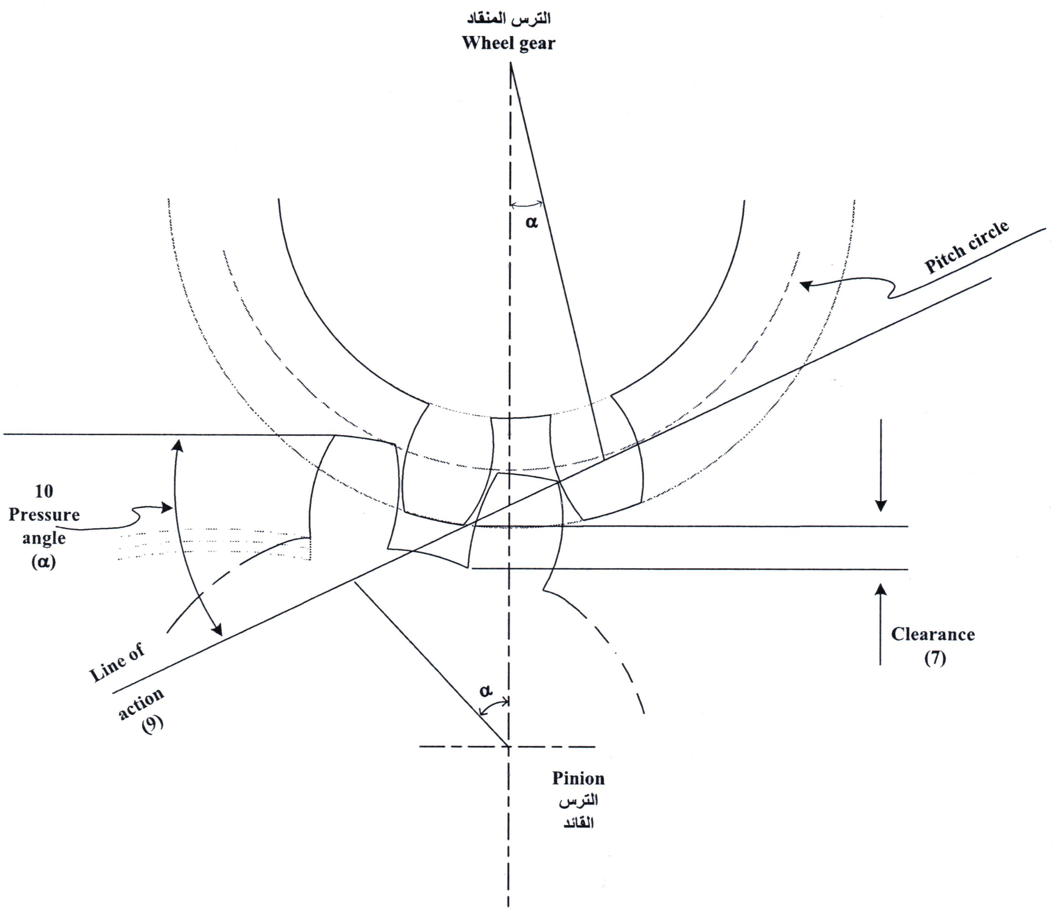
d = diameter of the pitch circle, Z = no. of teeth on the gear.
3- Addendum circle: It is the circle drawn through the top of the teeth. It's diameter = d + 2m, m: module , $\frac{d}{Z} = \frac{1}{Pd} = m$.
Where : m = module. The module is expressed as lenth of pitch diameter per tooth.
4- Dedendum circle: It is the circle drawn through the bottom of the teeth. It's dia = (1.157)m – (1)m = 0.157m.
5- Addendum: It is the radial distance from the pitch circle to the top of the tooth. It's value is (1)m. i.e. a =(1)m.
6- Dedendum: it is the radial distance from the pitch circle to the root of the tooth. It's value is : $\frac{(1+\pi)}{20} \times m = 1.157$ m.
7- Clearance: It is the difference between dedendum and addendum.
8- Depth of the tooth: It is the radial distance between the addendum circle and dedendum circle of a gear.
9- Line of action (Pressure line): It is a line normal to a pair of mating tooth profiles at their point of contact.
10- Pressure angle: It is the angle between the line of action and the common tangent to the pitch circle.
11- Diametral pitch (Pd): It is expressed as the number of teeth per unit length of the pitch circle diameter.
i.e. $Pd = \frac{Z}{d} = \frac{\pi}{P}$, P: circular pitch, Z = عدد الأسنان , d = diameter pitch of pitch circle.
12- The module (m): It is the pitch diameter of the gear in millimeters divided by the total number of gear – wheel teeth.
i.e. $m = \frac{d}{Z}$ (mm) d = pitch dia , Z = عدد الأسنان . For two meshing gears, their module pitch or their diametral pith must be same.

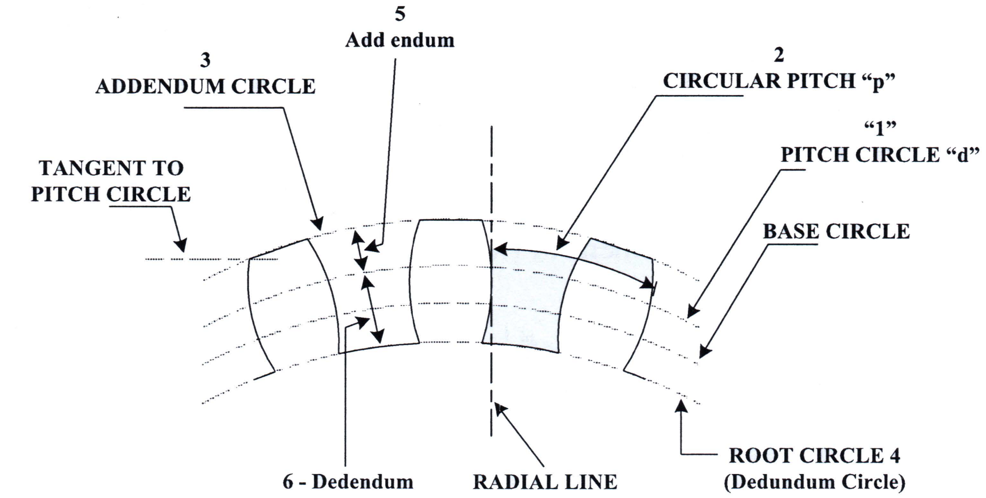
13- Thickness of teeth: It is the width of teeth measured along the pitch circle.

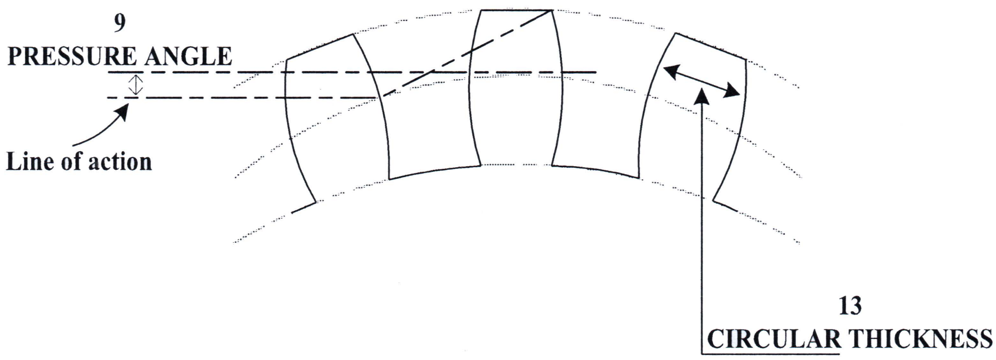
14- Centre distance: It is the distance between the centres of two gear in meshing. It's value is:

$$C = \frac{(d_1 + d_2)}{2}.$$

15- Arc of contact: It is the locus of a point on the pitch circle from the start to the end of engagement of two mating teeth.




5
Add endum
3
ADDENDUM CIRCLE
2
CIRCULAR PITCH "p"
"1"
PITCH CIRCLE "d"
TANGENT TO
PITCH CIRCLE
BASE CIRCLE
6 - Dedendum
RADIAL LINE
ROOT CIRCLE 4
(Dedundum Circle)


9
PRESSURE ANGLE
Line of action
13
CIRCULAR THICKNESS

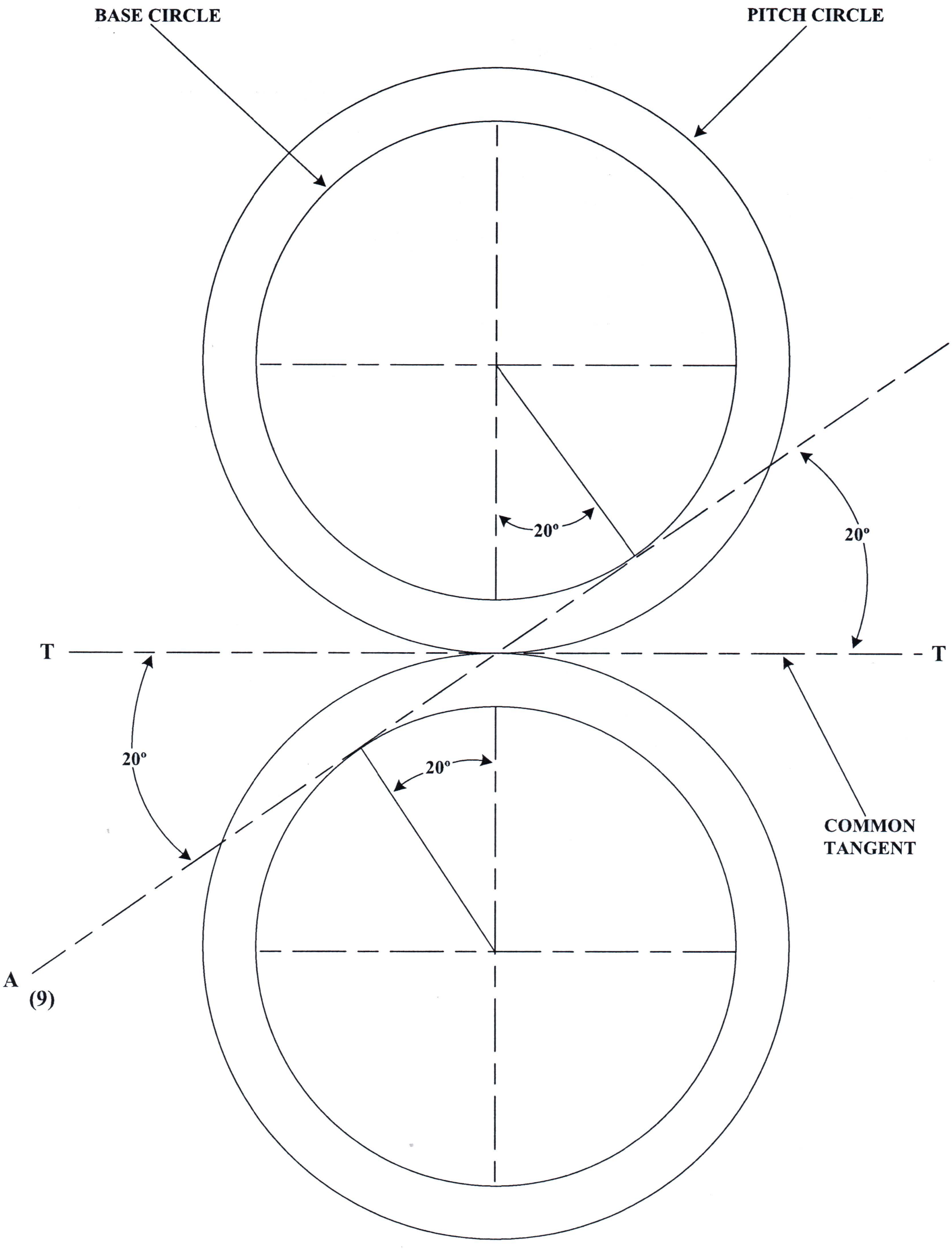


Fig. Drawing of base circle

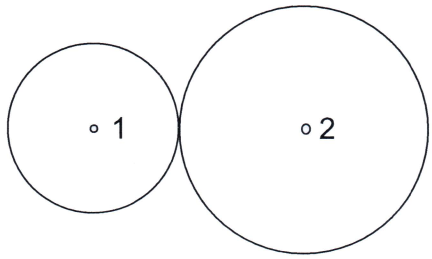
Velocity ratio of gears:

أن نسبة تغيير السرعة لزوج من التروس في حالة حركة تعتمد على عدد أسنان الترس القائد driver والترس المنقاد driven.

$$r = \frac{N_a}{N_b} = \frac{N_1}{N_2} = \frac{d_2}{d_1} = \frac{Z_2}{Z_1} = \frac{T_2}{T_1}$$

where:

r = velocity ratio.
N_1 = number of revolutions of driver gear.
N_2 = number of revolutions of driven gear.
d = diameter.
Z = number of teeth.
T = torque.



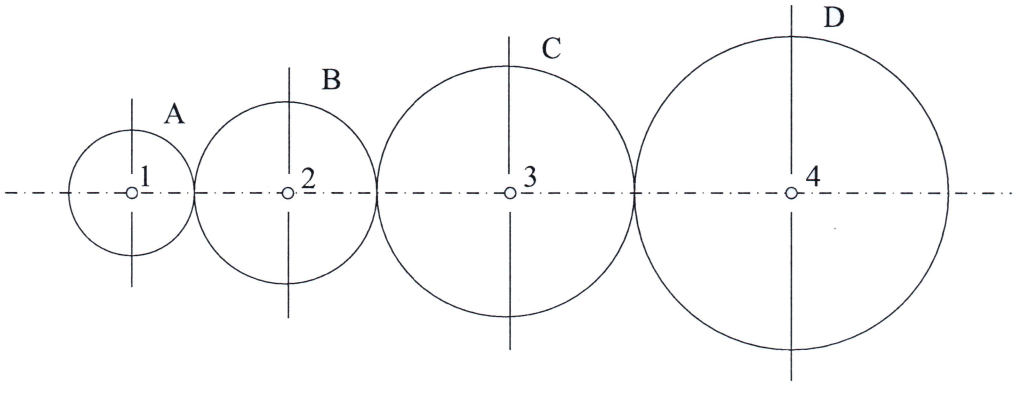
Gear trains: سلاسل التروس (نظام التروس):

سلاسل التروس هي عدد من التروس المعشقة مع بعضها لخفيض السرعة أو زيادة السرعة. وقد تكون سلسلة بسيطة أو سلسلة مركبة.

1. Simple gear train:

في هذه السلسلة تكون جميع التروس في مستوى واحد، وأبسطها سلسلة مكونة من ترسين فقط يسمى أحداهما القائد ويسمى الأخرى المنقاد (التابع).

وإلا أذا لاحظنا الشكل التالي نجد أن:-



$$r = \frac{N_a}{N_b} = \frac{Z_b}{Z_a}, \quad \frac{N_b}{N_c} = \frac{Z_c}{Z_b}, \quad \frac{N_c}{N_d} = \frac{Z_d}{Z_c}$$

$$r = \frac{N_a}{N_b} \cdot \frac{N_b}{N_c} \cdot \frac{N_c}{N_d} = \frac{Z_b}{Z_a} \cdot \frac{Z_c}{Z_b} \cdot \frac{Z_d}{Z_c}$$

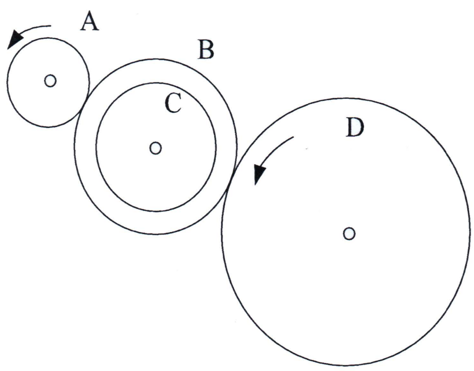
$$r = \frac{N_a}{N_d} = \frac{Z_d}{Z_a} = \frac{T_d}{T_a}$$

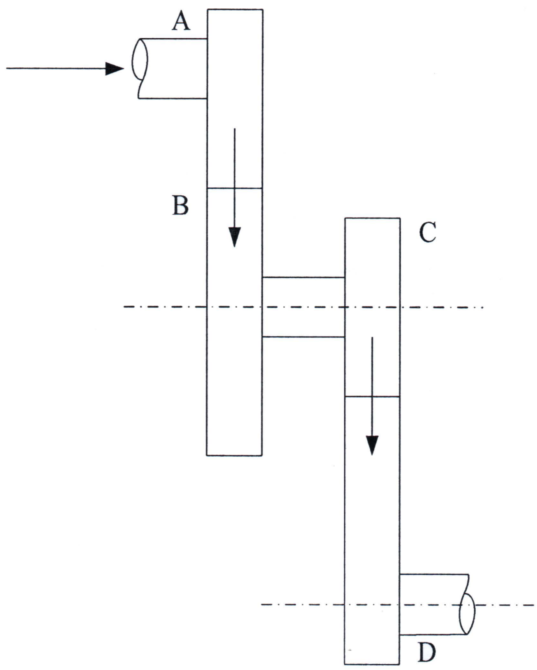
Ex: 1.

2. Compund gear train:

هذه السلسة تتكون من سلسلتين بسيطتين أو أكثر على التوالي. والترس المنقاد في سلسلة تكون مثبتة على نفس عمود الترس القائد للسلسة التالية:

وأذا لاحظنا الشكل التالي نجد أن:





$$\frac{N_a}{N_b}=\frac{Z_b}{Z_a} \quad \& \quad \frac{N_c}{N_d}=\frac{Z_d}{Z_c}$$

$$r=\frac{N_a}{N_b}.\frac{N_c}{N_d}=\frac{Z_b}{Z_a}.\frac{Z_d}{Z_c}$$

$$r=\frac{N_a}{N_d}=\frac{Z_b}{Z_a}.\frac{Z_c}{Z_b}.\frac{Z_d}{Z_c}=\frac{T_d}{T_a}$$

Ex: 2, 3.

1. Gear ratios:

صندوق التروس هو وسيلة لاختبار سرعات مختلفة بطريقة تتناسب وظروف تشغيل السيارة. ضمن خواص محركات الاحتراق فأن المحرك لا ينتج قدرة كافية عند سرعاته المنخفضة، فمثلاً عند بدأ الحركة أو عند صعود منحدر لابد من استعمال السرعة البطيئة لتوليد عزم دوران كبير على العجلات.

ويقوم صندوق التروس بثلاث مهمات:

1- اختبار سرعات مناسبة لحركة بحيث تبادل قوة المحرك بأكبر عزم لي.

2- اختبار وضع محايد يسمح بفصل القدرة الناتجة من المحرك عن بقية أجزاء نقل الحركة.

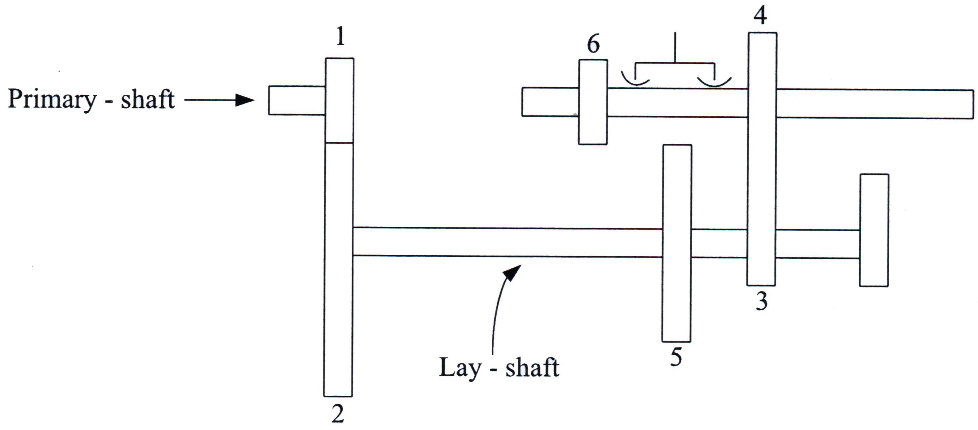
3- إعطاء حركة خلفية في اتجاه دوران معاكس لاتجاه المحرك الثابت دائماً.

ويتكون صندوق التروس من : (لاحظ الشكل التالي):

أ- العمود الابتدائي (عمود القابض) — Primary (input) shaft

ب- عمود التوزيع — Lay shaft

جـ- العمود الرئيسي — Main shaft



يتم الحصول على نسب تخفيض السرعة في صندوق التروس بواسطة سلسلة مركبة من التروس. لحساب التخفيض هذه نأخذ بنظر الاعتبار تخفيض سرعة العمود الرئيسي، ويمكن الحصول عليها من العلاقة الآتية:

$$r = \frac{\text{حاصل ضرب الأسنان التابعة}}{\text{حاصل ضرب الأسنان القائدة}} = \frac{\text{حاصل ضرب الأقطار التابعة}}{\text{حاصل ضرب الأقطار القائدة}}$$

بالنسبة لنسبة التخفيض في حالة السرعة المباشرة : $r = 1 : 1$

في هذه الحالة تكون سرعة الكرنك هي نفسها سرعة العمود الرئيسي الخارج من صندوق التروس.

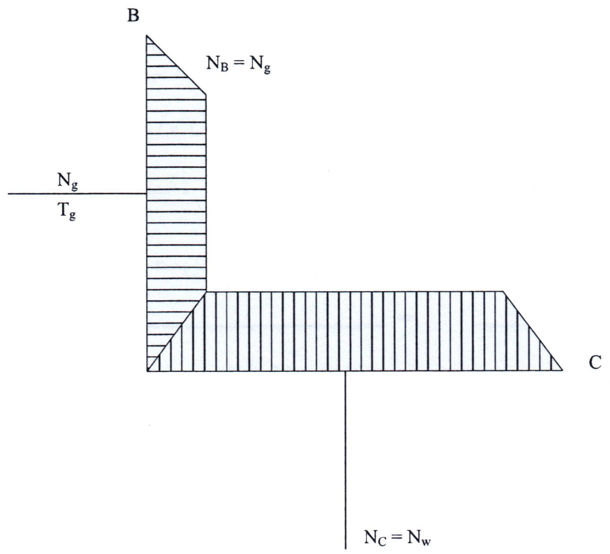
في بعض السيارات تكون سرعة العمود الرئيسي أعلى من السرعة الداخلة إلى صندوق التروس (عند السرعة المباشرة) وتسمى هذه نسبة فوق السرعة overdrive، وتكون عادة 0.72:1؛ أي أن الكرنك يدور 72 دورة فأن العمود الخارج من صندوق التروس يدور 100 دورة.

2. Rear – axle ratio: نسبة التخفيض في المجموعة الخلفية (الفرقية)

إن نسبة التخفيض في المجموعة الخلفية ثابتة دائماً وتعتمد على كل من أسنان ترس القائد (البنيون) وعدد أسنان ترس التاج. وتكون تروس المجموعة الفرقية عبارة عن تروس مخروطية. ويمكن إيجادها كما يلي:

$$r = \frac{N_b}{N_c} = \frac{Z_c}{Z_b} = \frac{T_c}{T_b}$$

$$r = \frac{N_g}{N_w} = \frac{T_w}{T_g}$$



3. Overall gear ratio:

هي عبارة عن حاصل ضرب نسبة التخفيض لصندوق التروس ونسبة التخفيض لمجموعة النقل النهائي (الفرقية). أي أن:

$$r_o = r_g \times r_r$$

Where:

r_o = overall gear ratio.
r_g = gear box ratio.
r_r = rear axle ratio.

4. Torque and speed in final drives:

يمكن زيادة عزم المحرك المنقول من خلال صندوق التروس لكل نقلة من نقلات الصندوق ما عدا السرعة العليا (السرعة المباشرة).

لذا فأن عزم دوران العجلات يكون أعلى عمود الكاردن وأعلى من عزم المحرك ولكن سرعتها أقل نتيجة تخفيض السرعة لذا فأن:

$$r_g = \frac{N}{N_g} = \frac{T_g}{T}$$

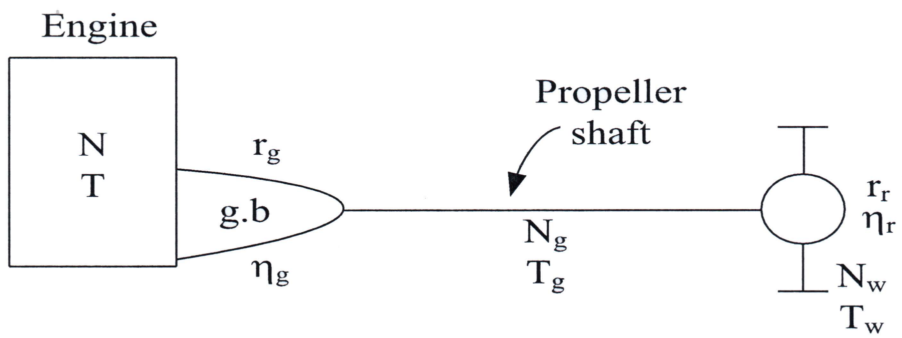
$$r_r = \frac{N_g}{N_w} = \frac{T_w}{T_g}$$

$$r_o = r_g \times r_r.$$

$$= \frac{N}{N_g} \times \frac{N_g}{N_w} = \frac{N}{N_w}$$

$$= \frac{T_g}{T} \times \frac{T_w}{T_g} = \frac{T_w}{T}$$

$$\therefore r_o = \frac{N}{N_w} = \frac{T_w}{T}$$



Ex: 8.

5. Tractive effort and engine Torque: (Te & T) جهد الجر وعزم المحرك

عندما تسير سيارة عند السرعة العليا فأن:

$$r_o = \frac{T}{T_w}$$

$$\therefore\ T_w = T \times r_o \times \eta t$$

Where:

T_w = driving wheels torque العزم للعجلات

T = egine torque عزم المحرك

→ r_r = rear axle ratio نسبة التخفيض للمحور الخلفي

ηt = transmission efficiency كفاءة النقل

ويمكن إيجاد عزم العجلات أيضاً من العلاقة التالية:

$$T_w = T_e \times R$$

Where:

T_e = tractive effort جهد الجر

R = effective radius of driving wheels القطر الفعال لعجلات القيادة

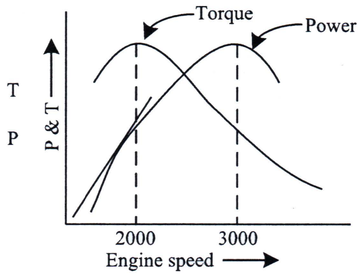
وبالتعويض

$$T_e \times R = T \times r_r \times \eta$$

$$\therefore\ T_e = T \times r_r \times \frac{\eta}{R}$$

Ex: 9, 10.

* نسبة سرعة المحرك:



لكل محرك سيارة سرعتان لهما أهمية خاصة وهما:

أ- السرعة التي يولد المحرك عندها أعظم قدره له.

ب- السرعة التي عندها يكون عزم المحرك قد وصل أعظم قيمة له.

ولكل محرك معلوم سرعة محدودة لا تتغير لأعظم قدرة وأخرى لأعظم عزم (لأحظ الشكل) والنسبة بين هاتين السرعتين تعرف بنسبة سرعة المحرك، وعليها يتوقف اختيار نسب السرعة في صندوق التروس والنقل النهائي.

* اختيار نسب التخفيض لصندوق التروس:

أن نسب التخفيض لصندوق التروس تتبع متوالية هندسية:

$r_g = K^{n-1}$ gear box ratio نسبة التخفيض لصندوق التروس

rg = نسبة التخفيض في صندوق التروس.

n = عدد النقلات في الصندوق.

K = مقدار ثابت تسمى الخطوة وهي أيضاً نسبة سرعة المحرك ويمكن إيجادها من العلاقة التالية:

$$K = \frac{N_p}{N_T}$$

Where:

N_p = egine speed for max. power. سرعة المحرك لأكبر قدرة

N_T = egine speed fpr max. torque. سرعة المحرك لأكبر عزم

Ex: 11, 12, 13.

العلاقة بين عدد دورات المحرك وسرعة المركبة على الطريق:

Relation between egine revolution (N) and Vehicle Linear speed (V):

Let	$N=$	engine speed	(rpm)	سرعة المحرك
	$N_w=$	road wheel speed	(rpm)	سرعة العجلات
	$V=$	vehicle's linear speed	(km/hr)	السرعة الخطية
	$R=$	Radius of the road wheel	(m)	نصف قطر العجلات
	$W=$	angular velocity of the road wheel	(rad/sec)	السرعة الزاوية للعجلات

إن صندوق التروس له نسب تخفيض مختلفة، لذا فله نسب مختلفة من $\frac{N}{V}$ ويمكن إيجادها كما يلي:

Since

$$V = w\,R$$

$$\frac{(V \times 1000)}{(60 \times 60)} = \left(\frac{2\pi\,Nw}{(60)}\right) \times R$$

$$\frac{N_W}{V} = 2.65 \times \frac{1}{R} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(1)$$

But

$$r_o = r_g \times r_r = \frac{N}{N_w} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (2)$$

Substituting:

$$N_w = \frac{N}{r_o} \times \frac{1}{V} = 2.65 \times \frac{1}{R}$$

$$\frac{N}{V} = 2.65 \times \frac{r_o}{R} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (3)$$

Ex: 14.

Crank engine of a vehicle rotates at (3800)rpm, if r_{g1} is (2.78:1) & r_r = (4:55:1).

a) Find out engine speed in km/hr when the wheel radius equal (0.375) m.
b) Calculate the torque at each of rear wheel when gear efficiencies η_g, η_r is (0.94) , (0.95) respectively and engine torque = (140) N.m.
c)

منحنيات الأداء Performance Curves

التغير بين جهد الجر ومقاومة الجر مع السرعة:

(Variation of tractive effort & tractive resistance with speed):

الشكل يبين منحني مجموع المقاومات وجهد الجر مع سرعة السيارة على طريق معين وذو انحدارات مختلفة. فالمنحنيات (1، 2، ....، 5) هي منحنيات مجموع المقاومات. حيث أن المنحني (1) قد يمثل مجموع المقاومات على طريق مستوي والمنحني (2) مجموع المقاومات على طريق مائل والمنحني (3) لطريق ذو ميل أكبر وهكذا.

أما المنحنيات A و B و C فيمثل منحنيات جهد جر السيارة على السرعة الأولى والثانية والثالثة على التوالي.

يتضح من المنحنيات أنه عندما يزيد جهد الجر عن مجموع المقاومات فأن السيارة تتعجل (تزداد سرعتها) إلى السرعة التي عندها يتساوى جهد الجر مع مجموع المقاومات، أما إذا زادت المقاومات عن جهد الجر فأن السيارة تتباطأ (تنقص سرعتها) إلى السرعة التي عندها يتساوى جهد الجر مع مجموع المقاومات.

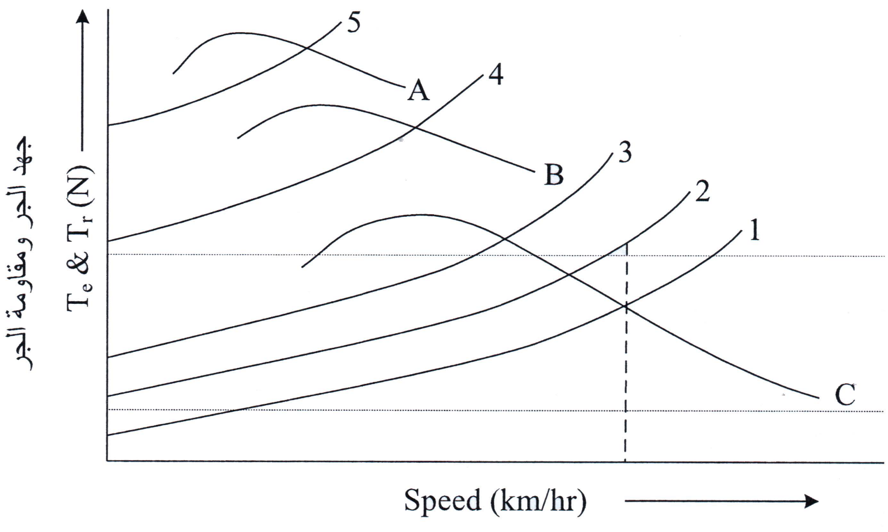
1، 2، 3، .... 5 منحنيات مجموع المقاومات.

1: يمثل مجموع المقاومات على طريق مستوي.

2: يمثل مجموع المقاومات على طريق مائل.

3: يمثل مجموع المقاومات على طريق ذو ميل أكبر.

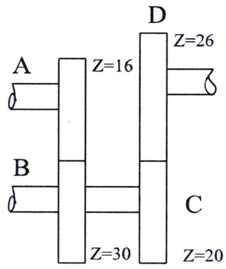
A، B، C: يمثل منحنيات جهد جر السيارة على السرعة الأولى والثانية والثالثة.



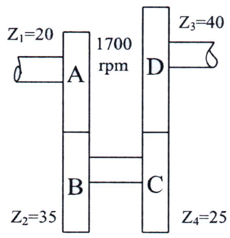
Gears

1- The number of teeth of the timing gear on a certain engine crankshaft is (14) teeth and the meshing gear on the camshaft has teeth. If the speed of the engine is (2400) rpm, Calculate: the speed of the camshaft.

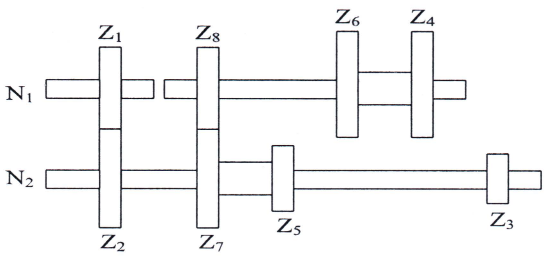
2- A compound gear train as shown in fig, find:
   a) Gear ratio.
   b) Speed of shaft (3) when shaft (1) rotates (2000) rpm.



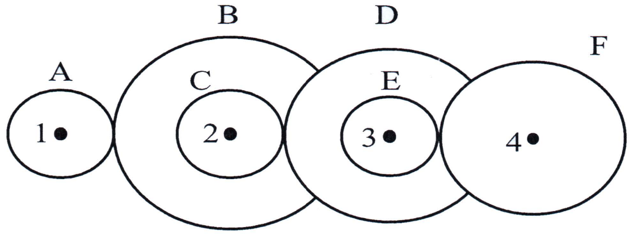
3- A compound gear train as shown in fig, Calculate:
   a) Overall gear ratio.
   b) Speed of shaft (3).
   c) Speed of shaft (2).



4- A gearbox as shown in fig, find gear ratios for fourwared speed, when:
   $Z_1 = 19.$
   $Z_2 = 40.$
   $Z_3 = 11.$
   $Z_4 = 34.$
   $Z_5 = 21.$
   $Z_6 = 32.$
   $Z_7 = 29.$
   $Z_8 = 24.$



5- A gearbox as shown in fig. If the number of teeth are as follows:
   $Z_a = 20.$
   $Z_b = 40.$
   $Z_c = 18.$
   $Z_d = 36.$
   $Z_e = 16.$
   $Z_f = 32.$
   Calculate:
   a) Gearbox ratio.
   b) Speed of shaft (4) when shaft (1) rotates (1800) rpm.



6- A car with four – speed gearbox traveling in 2nd gear, the engine speed (3200) rpm, engine torque (120) Nm, the constant – mesh pinions have (25) and (40) teeth respectively (Z_1 & Z_2). The 2nd gear layshaft pinions has (25) teeth (Z_5) and the meshing mainshaft gear has (40) teeth (Z6).
   Calculate:
   a) Second gear ratio.
   b) Speed of mainshaft gearbox.
   c) Torque at out put gearbox.

7- An engine develops a torque of (90) Nm at the flywheel at a speed of (1500) rpm and drives through a gearbox which has a low gear ratio of (3) to (1). If the efficiency of the drive is (9)%, what is the torque and speed of the propeller shaft.

8- In a four – speed gearbox the constant – mesh pinions have (20) and (35) teeth respectively. The second gear mainshaft pinion has (30) teeth and the meshing layshaft gear has (25) teeth. If the rear axle ratio is (5.5) to (1), Calculate: the overall gear ratio in second gear?

9- A motor car engine torque is (120) Nm, and the road wheel radius (0.3) m. if the rear axle ratio (4) to (1) and gearbox ratio (4) to (1), Calculate:

a) Maximum tractive effort.
b) Road wheel power, when the engine speed is (3000) rpm.

10- A vehicle traveling in first gear, has to exert a tractive effort of (2.4) KN to maintain a steady speed of (18) Km/hr. First gear ratio in gearbox is (4.2) to (1) and gearbox efficiency is (75)% and rear axle has final – driver ratio of (5) to (1) and efficiency of (88)%, whilst the rolling radius of driving wheels is (0.35) m. Calculate: the torque at the engine crankshaft?

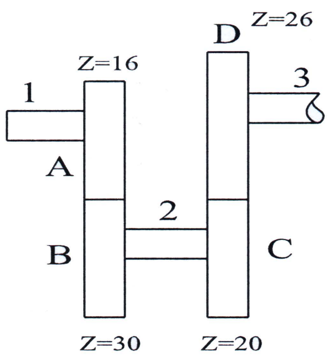
Gears Solution:

1.

$$r = \frac{N_1}{N_2} = \frac{Z_2}{Z_1}$$

$$= \frac{2400}{N_2} = \frac{48}{24}$$

N_2= 1200 rpm.



2.

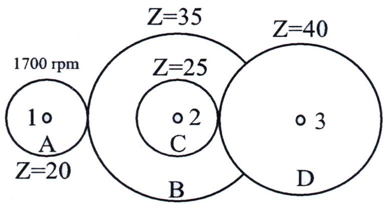
a) $r = \frac{Z_b}{Z_a} \times \frac{Z_d}{Z_c}$

$$r = \frac{30}{16} \times \frac{26}{20} = 2.44 : 1$$

b) $r = \frac{N_a}{N_d}$

$2.44 = 2000/N_d \rightarrow N_d = 820$ rpm.

3.



a) $r=\dfrac{Z_b}{Z_a}\times\dfrac{Z_d}{Z_c}=\dfrac{35}{20}\times\dfrac{40}{25}=2.5$

b) $r=\dfrac{N_a}{N_d}$

$$2.8=\frac{17000}{N_d}\rightarrow N_d=607\ \text{rpm}=N_3$$

c) $\dfrac{N_c}{N_d}=\dfrac{Z_d}{Z_c}$

$$\frac{N_c}{607}=\frac{40}{25}\rightarrow N_c=971\ \ \text{rpm}=N_2$$

4.

$$r_1=\frac{Z_2}{Z_1}\times\frac{Z_4}{Z_3}=\frac{40}{19}\times\frac{34}{11}=6.6$$

$$r_2=\frac{Z_2}{Z_1}\times\frac{Z_6}{Z_5}=\frac{40}{19}\times\frac{32}{21}=3.2$$

$$r_3=\frac{Z_2}{Z_1}\times\frac{Z_8}{Z_7}=\frac{40}{19}\times\frac{24}{29}=1.74$$

$$r_4=1:1$$

5.

$$r_1=\frac{Z_b}{Z_a}\quad,\quad r_2=\frac{Z_d}{Z_c}\quad,\quad r_3=\frac{Z_f}{Z_e}$$

a) $r_g=\dfrac{Z_b}{Z_a}\times\dfrac{Z_d}{Z_c}\times\dfrac{Z_f}{Z_e}$

$$=\frac{40}{20}\times\frac{36}{18}\times\frac{32}{16}=8:1$$

b) $r_g=\dfrac{N_1}{N_4}$

$$8=\frac{1800}{N_4}\rightarrow N_4=225\ \text{rpm}\ 8$$

6.

a) $r_2=\dfrac{Z_2}{Z_1}\times\dfrac{Z_6}{Z_5}=\dfrac{40}{25}\times\dfrac{40}{25}=2.56:1$

b) $r_2=\dfrac{N_1}{N_6}$

$$2.56 = \frac{3200}{N_6} \rightarrow N_6 = 1250 \text{ rpm} = N_g$$

c) $r_2 = \frac{T_6}{T_1}$

$$2.56 = \frac{T_6}{120} \rightarrow T_6 = 307.2 \text{ Nm} = T_g$$

7.

$$r_g = \frac{T_g}{T}$$

$$T_g = r_g \times T \times \eta_g$$
$$= 3 \times 90 \times 0.90$$
$$= 243 \text{ Nm}$$

also

$$r_g = \frac{N}{N_g}$$

$$3 = \frac{1500}{N_g}$$

N_g= 500 rpm

8.

$$r_g = \frac{Z_2}{Z_1} \times \frac{Z_6}{Z_5} = \frac{35}{20} \times \frac{30}{25} = 2.1:1$$

$$r_o = r_g \times r_r = 2.1 \times 5.5 = 11.55:1$$

9.

a) $r_o = r_g \times r_r = 4 \times 4 = 16$

$$r_o = \frac{T_w}{T}$$

$$T_w = r_o \times T \times \eta_T = 16 \times 120 = 1920 \text{ Nm}$$
$$T_w = T_e \times R_w$$
$$1920 = T_e \times 0.3 \rightarrow T_e = 6400 \text{ N}$$

b) $r_o = \frac{N}{N_w}$

$$16 = \frac{3000}{N_w} \rightarrow N_w = 187.5 \text{ rpm}$$

$$P_w = T_w \times \frac{2\pi N}{60} = 1920 \times \frac{2\pi \times 187.5}{60}$$
$$= 37680 \text{ watt}$$

11- A gearbox of 4 - speed ratio, the speed for maximum torque is (3000) rpm and for maximum power is (4800) rpm.

Calculate: the gear speed ration?

12-A car with four – speed gearbox traveling in 1st gear. If the overall gear ratio is (20) to (1) and the engine speed ratio is (1.6) find rear axle ratio?

13-A motor car engine torque is (120) Nm, and the road wheel radius (0.3)m. If the rear axle ratio is (4) to (1), the gearbox has 3 – speed, engine speed ratio is (2) and the car is traveling in 1st gear.

Calculate:

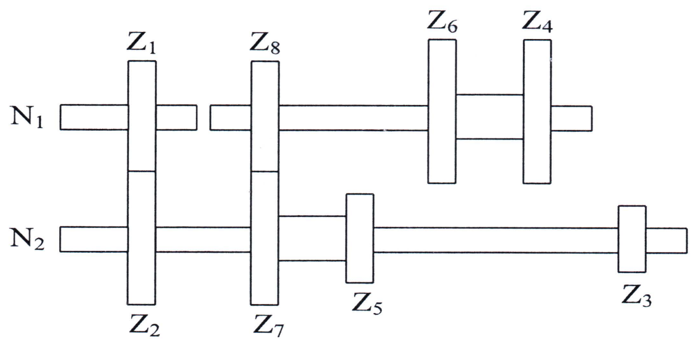
a) Maximum tractive effort.
b) Road wheel power, when the engine speed is (2000) rpm.

14- A motor vehicle engine produce a torque of (140) Nm at speed of (3800) rpm. First gear ratio in gearbox is (2.78:1) and gearbox efficiency is (94)% and rear axle ratio is (4.55:1) and efficiency of (95)%. If the diameter of the wheel is (75) cm, find:

a) Linear velocity of the wheel.
b) Torque of the wheel.

4) A gearbox as shown in fig, find gear ratios for forword speed, when

$Z_1 = 19.$
$Z_2 = 40.$
$Z_3 = 11.$
$Z_4 = 34.$
$Z_5 = 21.$
$Z_6 = 32.$
$Z_7 = 29.$
$Z_8 = 24.$



Solution:

$$r_1 = \frac{Z_2}{Z_1} \times \frac{Z_4}{Z_3} = \frac{40}{19} \times \frac{34}{11} = 6.6$$

$$r_2 = \frac{Z_2}{Z_1} \times \frac{Z_6}{Z_5} = \frac{40}{19} \times \frac{32}{21} = 3.2$$

$$r_3 = \frac{Z_2}{Z_1} \times \frac{Z_8}{Z_7} = \frac{40}{19} \times \frac{24}{29} = 1.74$$

$r_4 = 1:1$

10.

$r_o = r_g \times r_r = 4.2 \times 5 = 21{:}1$

$T_w = T_e \times R$

$T_w = r_o \times T \times \eta_t \ldots\ldots\ldots.(\eta = \eta_g \times \eta_r)$

$$T = \frac{(T_e \times R)}{(r_o \times \eta_g \times \eta_r)} = \frac{(2.4 \times 1000 \times 0.35)}{(21 \times 0.75 \times 0.8)} = 66.6 \text{ Nm}$$

11.

$N_T = 3000$ rpm , $N_p = 4800$ rpm , $r_g = ?$

$$K = \frac{N_p}{N_t} = \frac{4800}{3000} = 1.6$$

$R_g = k^{n-1}$

$R_{g1} = 1.6^{4-1} = 1.6^3 = 4.1{:}1$

$R_{g2} = 1.6^{3-1} = 1.6^2 = 2.56{:}1$

$R_{g3} = 1.6^{2-1} = 1.6{:}1$

$R_{g4} = 1{:}1$

12.

$r_o = 20$, $K = 1.6$, $r_r = ?$

$r_g = K^{n-1}$

$= 1.6^{4-1} = 1.6^3 = 4.1{:}1$

$r_o = r_g \times r_r$

$20 = 4.1 \times r_r$

$r_r = 4.96{:}1$

13.

a) $r_g = K^{n-1} = 2^{3-1} = 2^2 = 4{:}1$

$r_o = r_g \times r_r = 4 \times 4 = 16$

also,

$$r_o = \frac{T_w}{T}$$

$T_w = r_o \times T \times \eta = 16 \times 120 = 1920$ Nm

where $\eta = 1$.

also,

$T_w = T_e \times R$

$1920 = T_e \times 0.3$

$T_e = 6400$ Nm

14.

a) $r_o = r_g \times r_r = 2.78 \times 4.55 = 12.65$

$$\frac{N}{V} = 2.65\left(\frac{r_o}{R}\right)$$

N : rpm, V : km/hr, R : m

$$\frac{2800}{V} = 2.65\left(\frac{12.65}{0.375}\right)$$

$V = 42.5\ Km/hr$

OR:

$r_o = r_g \times r_r = 12.65$

$$r_o = \frac{N}{N_w}$$

$$12.65 = \frac{3800}{N_w}$$

$N_w = 300.4\ rpm$ (سرعة دوران العجلات)

$V = \pi dN$

$$V = \frac{(\pi \times 75 \times 300.4 \times 60)}{10^5}$$

$= 42.44\ Km/hr$

b)

$\eta_o = \eta_g \times \eta_r = 0.94 \times 0.95 = 0.593$

$$r_o = \frac{T_w}{T}$$

$T_w = r_o \times T \times \eta_o$

$= 12.65 \times 140 \times 0.593 = 1581.5\ Nm$

$$\text{Torque for one wheel} = \frac{T_w}{2} = \frac{1581.5}{2} = 790.75\ Nm$$

Q_B - Engine with gear box of (3) speeds, Calculate the over all ratio it the real axle ratio equal (4:1) and the max. speed ratio equal (2:1), Also Find out the torque exerted on the rear wheel if its diameter = (0.6) meter & engine torque is (120) N.m.

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المحاضرة التاسعة

Vehicle Suspension System

Vehicle Suspension System

1. Introduction:

أن هيكل وإطار السيارة تربط مع المحور الخلفي والأمامي بواسطة النوابض. حيث أن هذه النوابض تخمد الصدمات الناتجة من الطريق والتي تنتقل إلى هيكل السيارة عن طريق الإطارات، وذلك لحماية الأجزاء المركبة على الإطار المعدني (frame).

وعليه فأن جميع الأجزاء التي تساهم في حماية مكونات السيارة من الصدمات تسمى "نظام التعليق في السيارة".

2. Objective of suspension:

الغاية من نظام التعليق هي:

a- To prevent the transmission of the road shocks to the vehicle components.
b- To give the stability to the vehicle while in motion.
c- To provide the particular height to body structure and to bear the torque and braking reactions.

3. Springs:

النابض عبارة عن جسم مرن يتمدد أو يتقلص تحت تأثير قوة خارجية، ويستعيد شكله الأصلي بعد زوال القوة المؤثرة. ويعتبر التغير بطول النابض (الزيادة أو النقصان) مهماً جداً في حالة تصميم النوابض وذات تأثير مباشر عند اختيار النابض.

4. Types of Springs:

يوجد العديد من النوابض المستعملة في مجالات مختلفة، ويعتمد نوع النابض على الحالة التي يعمل بها أو وظيفتها وطريقة ربطه.

وتصنف النوابض على أساس الشكل كما يلي:

a- Leaf (Laminated) springs.
b- Helical (coil) springs.
c- Torsion springs.

يستعمل في كثير من السيارات نوابض لولبية في المقدمة ونوابض ورقية في المؤخرة أو قد يستعمل نوابض لولبية في المؤخرة أيضاً.

وقد يستخدم في بعض سيارات الخدمة الثقيلة نوابض ورقية في المقدمة أيضاً. وفي السنوات الأخيرة تم استخدام النوابض الالتوائية في بعض السيارات.

Leaf Spring

يستعمل النابض الورقي بصورة واسعة في نظام تعليق السيارة، ويقوم بامتصاص طاقة الصدمات المفاجئة وكافة الحركات الناتجة من تأثير سطح الطريق على السيارة. وهذه النوابض عبارة عن مجموعة أوراق حديدية متدرجة في الطول توضع فوق بعضها وتربط من وسطها بواسطة برغي مركزي centre bolt وبواسطة ماسكات على شكل حرف U.

والورقة الواحدة عبارة عن قطعة حديدية ذات سمك قليل نسبياً.

والنابض عندما يكون غي محمل فأنه يكون مقوس الشكل (بيضوي) ولهذا فأنه يسمى في هذه الحالة بالنابض الورقي شبه بيضوي Semi – elliptical leaf spring. تسمى الورقة الطويلة بالورقة الرئيسية (main) master ويكون نهايتيها على شكل دائرة من أجل ربطها مع هيكل السيارة:

Stiffness of the spring:

أن متانة النابض تعتمد على الأمور التالية:

1. Length of the spring.
2. Width of the spring.
3. Thickness of the spring.
4. Total number of the spring.

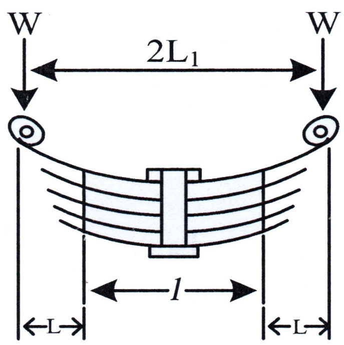
Leaf Spring Stresses Calculation:

أن الإجهاد المؤثرة على النابض الورقي هو إجهاد الانحناء bending ويمكن إيجادها من العلاقة التالية:

$$f_b = \frac{6WL}{nbt^2} \quad ....(1)$$

Where:

W = Load on the leaf spring end
L = effective length of the cantilevers spring
n = total number of the leaves
b = width of the leaf
t = thickness of the leaf



Deflection:

يمكن إيجاد مقدار النابض نتيجة الحمل W من العلاقة التالية:

$$\delta = \frac{2 f_b L^2}{3Et}$$

Where:

E = modules of elasticity

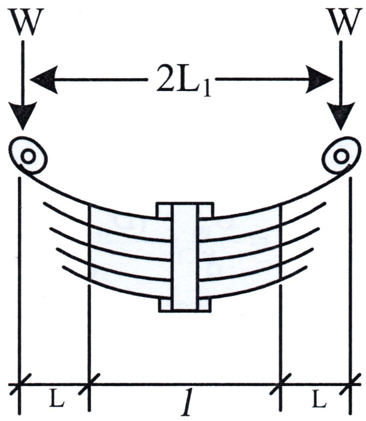
Calculation of spring leaves length:

من الشكل التالي نلاحظ أن طول النابض المؤثر هو (2L) حيث أن:

$$2L = 2L_1 - l$$

Where:

- $2L$ = effective length of spring
- $2L_1$ = Full length of the spring (over length)
- l = non effective length of the spring (width of band)



لحساب طول كل ورقة يكون بالشكل التالي، حيث يحسب طول الورقة الصغيرة رقم واحد (S_1) ثم التي تليها S_2 ، S_3 ، .... وحتى الورقة الرئيسية (S_m).

طول الورقة الأولى:

$$S_1 = \frac{2L_1 - l}{n-1} \times 1 + l$$

$$S_2 = \frac{2L_1 - l}{n-1} \times 2 + l$$

وهكذا تحسب أطوال الأوراق وحتى الورقة الرئيسية والتي يمكن إيجاد طولها من العلاقة التالية:

$$Sm = 2L_1 + \pi\,(d + t) \times 2$$

Where:

- d = inside diameter of eye
- t = thickness of the leaf

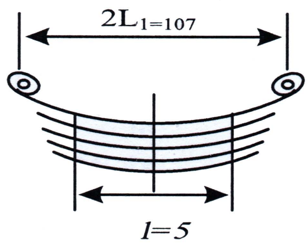
Ex: 1

A locomotive semi – elliptical laminated spring has an overall length of (107)cm and with of (5)cm, sustain a load of (3.56)KN at its center. The spring has one full length leaf which has a circular diameter of (1.5)cm, and seven graduated leaves with a central band of (5)cm long. If the bending stress of the spring is $(344\times10^{6})N/m^{2}$, and modulus of elasticity is $(206.9\times10^{9})N/m^{2}$.

Determine:

a) Thickness of the leaves.
b) Deflection of the spring.
c) Length of each leaf.

Solutions:



1) $L=\frac{(2L_1-l)}{2}=\frac{(107-5)}{2}=51\ cm.$

$$f_b=\frac{6WL}{(nbt^2)}$$

$$344\times10^6=\frac{6\times1.78\times10^3\times0.51}{(8\times0.05t^2)}$$

$$t^2=\frac{0.3958}{10^4}\quad\rightarrow\quad t=0.629\times10^{-2}\ m=6.29\ mm$$

2) $$\delta=\frac{2f_bL^2}{(3Et)}=\frac{2\times344\times10^6\times0.51^2}{(3\times206.9\times10^9\times6.29\times10^{-3})}$$

$$=0.0459m=45.9mm$$

3) $$S_1=\frac{2L_1-l}{n-1}\times1+l=\frac{107-5}{8-1}\times1+5=19.59\ cm$$

$$S_2=\qquad=\frac{107-5}{8-1}\times2+5=34.14\ cm.$$

$$S_3=\qquad=\frac{107-5}{8-1}\times3+5=48.7\ cm.$$

وهكذا نجد أطوال بقية الأوراق المتدرجة: S_4, S_5, S_6, S_7 أما طول الورقة الرئيسية:

$$S_m=2L_1+2\pi\,(d+t)$$
$$=107+2\pi\,(1.5+0.628)=120.36\ cm$$

Ex: 2

A leaf spring has an overall length of (100)cm and sustains a load of (70)KN at its center. The spring has (3) full length leaves and (15) graduated leaves with a central band of (10)cm long. All the leaves are to be stressed to (400)N/mm^2 when fully loaded. The ratio of the total spring depth to that of width is (2), E = (2000)N/mm^2.

Determine:

a) The thickness of the leaves.

b) Width of the leaves.

Solutions:

a) $L = \frac{(2L_1 - l)}{2} = \frac{(107 - 10)}{2} = 45 \text{ cm.} = 450 \text{ mm}$

$$\frac{18t}{b} = 2 \quad \rightarrow \quad \boxed{b = 9t}$$

$$f_b = \frac{6WL}{(nbt^2)}$$

$$400 = \frac{6 \times 35 \times 10^3 \times 450}{(18 \times 9t \times t^2)} \quad \rightarrow \quad t = 11.3 \text{ mm} \cong 12 \text{ mm.}$$

b) b = 9t = 9 × 12 = 108 mm = 10.8 cm.

Ex: 3

A semi – elliptical laminated vehicle spring to carry a load of (3000)N is to consist of seven leaves (6.5)cm wide, two of the leaves extending the full length of the spring. The spring is to be (110)cm in length and attached to the axle by two U-bolts (8)cm a part.

Assuming an allowable stress of (350)N/mm^2, modulus of elasticity (2.1×10^5)N/mm^2. and the master leaf has a circular diameter of (1.5)$_{cm}$.

a) Thickness of the leaves.
b) Deflection of the spring.
c) Length of the leaves.

Solution:

2W = 3000 N → ∴ W = 1500 N , n = 7 , b = 6.5 cm = 65 mm

$2L_1$ = 110 cm , l = 8cm = 80mm , f = 350 N/mm^2, E = 2.1 × 105 N/mm^2

$$L = \frac{2L1 - l}{2} = \frac{110-8}{2} = 51\,\text{cm} = 510\,\text{mm}$$

1) thickness : لإيجاد سمك الورقة:

$$f_b = \frac{6WL}{(nbt^2)}$$

$$350 = \frac{6\times1500\times510}{(7\times65t^2)}$$

$\therefore\ t^2 = 29.5936$

$t = 5.44\ \text{mm} \cong 5.5\ \text{mm}$

2) deflection الانحراف

$$\delta = \frac{2\,fL^2}{3EL} = \frac{2\times350\times510^2}{3\times2.1\times10^5\times5.5}$$
$$= 52.5\,\text{mm} = 5.25\,\text{cm}$$

3) Length of the leaves: لإيجاد طول الأوراق:

$$S_1 = \frac{2L_1 - l}{n-1}\times1 + l = \frac{110\text{ - }8}{7\text{ - }1}\times1 + 8 = 25\ \text{cm}$$

$$S_2 = \frac{2L_1 - l}{n-1}\times2 + l = \frac{110\text{ - }8}{7\text{ - }1}\times2 + 8 = 40\ \text{cm}$$

$$S_3 = \frac{2L_1 - l}{n-1}\times1 + l = \frac{110\text{ - }8}{7\text{ - }1}\times3 + 8 = 56\ \text{cm}$$

S_4 = - - - - - - = 72 cm, S_5 = - - - - - - = 88 cm, S_6 = - - - - - - = 104 cm

الورقة السادسة والسابعة بطول كامل، والورقة السابعة هي الورقة الرئيسية ولإيجاد طولها:

$S_m = 2L_1 + \pi\,(d + t) \times 2$

$= 110 + \pi\,(1.5 + 0.65) \times 2$

$= 135$ cm.

Helical (Coil) Spring

يصنع النابض اللولبي من سلك ذو قطر معين. ويتم اختيار القطر والمادة المصنوعة منها حسب ظروف عمل استخدام النابض. وبصورة عامة يصنع من مادة الفولاذ عالي الكربون.

Types of Coil springs:

a- Compression helical spring.
b- Tension helical spring.

ويكون الإجهاد المؤثر على هذه النوابض هو إجهاد القص «shearing stress» نتيجة لالتواء سلك النابض أثناء السحب أو الانضغاط.

Spring Terms:

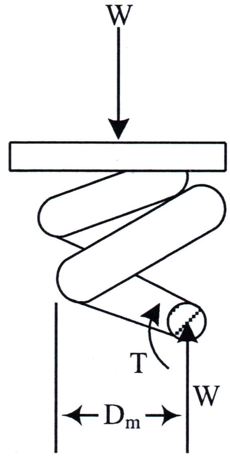
1. Solid length (L_s): It is the length of a coil spring when it is compressed until the coils come in contact with each other.
   $Ls = n' d$
   When , n' = total number of coils , d = dia of the wire.
2. Free length (L_f): It is the length of a free coil spring (uncompressed).
   $L_f = n' d + \delta_{max} + (n' - 1) \times 0.1$
3. Spring index (C): It is defined as the ratio of the mean diameter of the spring to the diameter of the wire.
   $$C = \frac{D_m}{d} = \frac{(D_o - d)}{d}$$
4. Spring rate (spring stiffness or spring constant) (R) : It is the load required per unit deflection of the spring. $R = \dfrac{W}{\delta}$
   Where , W = load, δ = deflection of the spring.
5. Pitch of the coil (P) : It is the axial distance between adjacent coils in uncompressed state.
   $$P = \frac{L_f}{n' - 1}$$

Coil spring stresses calculation:

الشكل يبين جزءاً من نابض لولبي انضغاطي تؤثر فيه قوة خارجية W.

أن النابض المبين في حالة اتزان للقوتين W وعزم اللي T.

أن عزم اللي المؤثر في النابض هو:



$$T = W \times \frac{D_m}{2} \ldots\ldots\ldots\ldots\ldots\ldots(1)$$

كما أن العزم:

$$T = f_s \times \frac{\pi d^3}{16} \ldots\ldots\ldots\ldots(2)$$

وبالتعويض:

$$W \times \frac{D_m}{2} = \frac{f_s \pi d^3}{16}$$

$$\therefore\ f_s = \frac{8WD_m}{\pi d^3} \ldots\ldots\ldots\ldots\ldots (3)$$

Where :

f_s = shear stress induced in the wire
W = axial load on the spring
D_m = mean diameter of the sptring
D = dia. of the spring

يمكن كتابة المعادلة السابقة (3) كما يل:

$$f_s = \frac{8KWD_m}{\pi d^3} \ldots\ldots\ldots\ldots\ldots\ldots(4)$$

Where :

K = shearing stress factor

(Wahl's factor)

حيث أن K هو معامل أجهاد القص والذي يمكن إيجاده من المعادلة الآتية، والتي تبين أن المعالم K يعتمد أساساً على دليل النابض:

$$K = \frac{4C-1}{4C-4} + \frac{0.615}{C}$$

Deflection of helical spring:

$$\delta = \frac{8nWD^3{}_m}{(Gd^4)}$$

Where :

N = number of active turns

G = modulus of rigidity for spring material

Ex: 4

A helical spring is made from a wire of (6)mm diameter and has an outside diameter of (7.5)cm. If the permissible shear stress is (350)MN/m^2 and modulus of rigidity (84)GN/m^2, find the axial load which the spring can carry and the deflection per active turn.

Solutions:

$$D_m = D_o - d = 0.075 - 0.006 = 0.069 \text{ m}$$

$$C = \frac{D_m}{d} = \frac{0.069}{0.006} = 11.5$$

$$K = \frac{4C-1}{4C-4} + \frac{0.615}{C} = \frac{4\times 11.5 - 1}{4\times 11.5 - 4} + \frac{0.615}{11.5} = 1.125$$

$$f_s = \frac{8KWD_m}{(\pi d^3)}$$

$$350\times 10^6 = \frac{8\times 1.125\times W\times 0.069}{\left(\pi\times (0.006)^3\right)}$$

W = 382.2 N

$$\delta = \frac{8W(D_m)^3 n}{(Gd^4)}$$

$$\frac{S}{n} = \frac{8W(D_m)^3}{Gd^4} = \frac{8\times 382.2\times (0.069)^3}{84\times 10^9\times (0.006)^4}$$ مقدار الانحراف لكل لفة فعالة:

$$= 9.22\times 10^{-3} \text{ m} = 9.22 \text{ mm.}$$

Ex: 5

A helical spring has a mean diameter of (25) mm and the diameter of the wire is (3) mm. if the shear stress 441 MN/m^2 and the total deflection of spring is (25) mm, find the load carried by the spring? and the number of active turns. Modulus of rigidity is (86.2) GN/m^2.

Solutions:

$D_m = 25$ mm , $d = 3$ mm.

$f_s = 441\ MN/m^2$

$S = 25$ mm, $W = ?$, $n = ?$,

$G = 86.2\ GN/m^2$,

$$C = \frac{D_m}{d} = \frac{0.025}{0.003} = 8.33$$

$$K = \frac{4 \times 8.33 - 1}{4 \times 8.33 - 4} + \frac{0.615}{8.33} = 1.176$$

$$f_s = \frac{8KWD_m}{(\pi d^3)}$$

$$441 \times 10^6 = \frac{8 \times 1.176 \times W \times 0.025}{(\pi \times (0.003)^3)}$$

$W = 159$ N

$$\delta = \frac{8W(D_m)^3 n}{(Gd^4)}$$

لحساب عدد لفات النابض

$$0.025 = \frac{8 \times 106.5 \times (0.025)^2 \times n}{86.2 \times 10^9 \times (0.003)^4}$$

$n = (13.107)$ active turns $= 14$

$n' = n + 2 = 14 + 2 = 16$ total turns.

Ex: 6

Design a helical compression spring for a maximum load of (1000) N for a deflection of (25) mm using the value of spring index as (5) stress, shear (420) N/mm^2, modulus of rigidity (84) KN/mm^2, Wahl's factor:

$$K = \frac{4C-1}{4C-4} + \frac{0.615}{C} \quad \text{, where } C \text{ is spring index.}$$

Solutions:

$W = 1000$ N, $S = 25$ mm, $C = 5$, $f_s = 420$ N/mm^2 , $G = 84$ KN/mm^2

$$K = \frac{4\times5-1}{4\times5-4} + \frac{0.615}{5} = 1.31$$

$$C = \frac{D_m}{d}$$

$$5 = \frac{D_m}{d} \quad \rightarrow \quad D_m = 5d$$

$$f_s = \frac{8KWD_m}{(\pi d^3)}$$

$$420 = \frac{8\times1.31\times1000\times5d}{(\pi\pi^3)}$$

$d^2 = 40 \quad \Rightarrow \quad d = 6.3$ mm

$D_m = 5d = 5 \times 6.3 = 31.5$ mm

$D_o = D_m + d = 31.5 + 6.3 = 37.8$ mm

$D_i = D_m - d = 31.5 - 6.3 = 25.2$ mm

$$\delta = \frac{8W(D_m)^3 n}{(Gd^4)}$$

$$25 = \frac{8\times1000\times(31.5)^3\times n}{(84\times10^3\times6.3^4)}$$

$n = 13.23 = 14$ active turn

$n' = n + 2 = 14 + 2 = 16$ total turn

$L_s = n'\,d = 16 \times 6.3 = 100.8$ mm

$L_f = n'\,d + \delta_{max} + (n' - 1) \times 0.1$

$= 100.8 + 25 + (16 - 1) \times 0.1 = 127.3$ mm

Pitch of the coil:

$$P = \frac{L_f}{(n'-1)}$$

$$= \frac{127.3}{(16-1)}$$

$= 8.84$ mm

Ex: 7

Design a close coiled helical compression spring for a service load ranging from (225) kg to (275) kg. The axial deflection of the spring for the load range is (6) mm. Assume a spring index of (5), shear stress (420) N/mm^2, Modulus of rigidity $G = 0.84 \times 10^5$ N/mm^2.

Solutions:

$W_1 = 225$ kg $= 2250$ N, $W_2 = 275$ kg $= 2750$ N

$\therefore\ W = 275 - 225 = 50$ kg $= 50 \times 10 = 500$ N

$\delta = 6$ mm, $C = 5$, $f_s = 420$ N/mm^2, $G = 0.84 \times 10^5$ N/mm^2

أولاً : سنقوم بإيجاد متوسط قطر النابض اللولبي لأقصى حمل مقداره (2750 N):

$$C = \frac{D_m}{d}$$

$$5 = \frac{D_m}{d} \quad \rightarrow \quad D_m = 5d$$

$$f_s = \frac{8W_2D_m}{\pi d^3}$$

$$420 = \frac{8 \times 2750 \times 5d}{\pi d^3}$$

$$420 = \frac{8 \times 2750 \times 5}{\pi d^2}$$

$d^2 = 78.1279$, $\therefore d = 8.839$ mm قطر سلك النابض

$Do = Dm + d = 44.195 + 8.839 = 53.034$ mm

$Di = Dm - d = 44.195 - 8.839 = 35.3$ mm

$$\delta = \frac{8W(Dm)^3 n}{Gd^4}$$

$$6 = \frac{8 \times 500 \times (44)^3 n}{0.84 \times 10^5 \times (8.8)^4}$$

$\therefore n = 8.9 \cong 9$ عدد اللفات الفعالة

$n' = n + 2 = 9 + 2 = 11$ عدد اللفات الكلية $\therefore$

الآن نقوم بإيجاد الانحراف الكلي الناتج من القوة (2750 N)

$$\delta_{max} = \frac{8 \times 2750 \times (44)^3 \times 9}{0.84 \times 10^5 \times (8.8)^4} = 33 \text{ mm}$$

$L_f = n' d + \delta_{max} + (n' - 1) \times 0.1$ الطول الحر:

$= 11 \times 8.8 + 33 + (11 - 1) \times 0.1 = 142$ mm

$P = \frac{L_f}{(n'-1)} = \frac{142}{(11-1)} = 14.2$ mm : خطوة النابض

Ex: 8

The following are the data for a helical spring used for an engine:

Length of the spring when value is open	= 4 cm,
Length of the spring when value is closed	= 5 cm,
Spring load when value is open	= 400 N,
Spring load when value is closed	= 200 N,
Inside diameter of spring	= 2.8 cm,
Max. shear stress	= 40 MN/m^2,
Modulus of rigidity	= 8 × 10 GN/m^2.

Design the spring. Take the Wahl's factor,

$$K = \frac{4C-1}{4C-4} + \frac{0.615}{C}$$, where C is spring index.

Solutions:

$$K = \frac{4C-1}{4C-4} + \frac{0.615}{C} = \frac{4\times 7.2-1}{4\times 7.2-4} + \frac{0.615}{7.2} = 1.2$$

$$C = \frac{D_m}{d}$$

$$7.2 = \frac{D_m}{d} \qquad \rightarrow \qquad D_m = 7.2d$$

أولاً : نقوم بإيجاد متوسط قطر النابض عند أقصى حمل = 400 N:

$$f_s = \frac{8W_2D_m}{\pi d^3}$$

$$40\times 10^6 = \frac{8\times 1.2\times 400\times 7.2d}{\pi d^3}$$

$$40\times 10^6 = \frac{8\times 1.2\times 400\times 7.2}{\pi d^2}$$

$d^2 = 0.00002304,\ d = 0.0048\ m = 4.8\ mm$

$D_m = 7.2 \times 4.8 = 35.06\ mm = 3.5\ cm$

$\delta = 5 - 4 = 1\ mm = 0.001\ m$ مقدار الانحراف في النابض:

يحدث هذا الانحراف (الانضغاط) نتيجة لقوة مقدارها:

$W = 400 - 200 = 200\ N$

$$\therefore\ \delta = \frac{8W(D_m)^3 n}{Gd^4}$$

$$0.001 = \frac{8\times 200\times (0.035)^3\times n}{(80\times 10^9\times (0.0048)^4)}$$

$\therefore\ n = 9.25 \cong 10$ turns عدد اللفات الفعالة:

$n' = n + 2 = 10 + 2 = 12$ عدد اللفات الكلية:

لإيجاد الطول الحر يجب إيجاد الانحراف الكلي الناتج من قوة (400 N):

$$S_{max} = \frac{8W(D_m)^3 n}{Gd^4}$$

$$= \frac{8 \times 400 \times (0.035)^3 \times 10}{(80 \times 10^9 \times (0.0048)^4} =$$

$$= 0.0323 \text{ m} = 3.23 \text{ cm}$$

$\therefore$ الطول الحر:

$$L_f = \bar{n}d + \delta_{max} + (\bar{n} - 1) \times 0.1$$

$$= 12 \times 0.48 + 3.23 + (12 - 1) \times 0.1 = 10.09 \text{ cm}$$

خطوة النابض:

$$P = \frac{L_f}{(n' - 1)} = \frac{10.09}{(12 - 1)} = 0.917 \text{ cm}$$

المادة : تصميم مكائن

مدرس المادة : أ.د حسين محمد علي

المحاضرة السادسة

PART 2

Belts

تستعمل السيور لنقل القدرة من عمود دائر إلى عمود آخر باستعمال بكرات مثبتة على هذه الأعمدة.

Types of Belts:

يوجد أنواع من السيور حسب شكل مقطعها وهي:

1. Flat belt:

ويكون مقطعه على شكل مستطيل. ويستعمل عادة في المعامل الكبيرة حيث تكون القدرة المنقولة كبيرة ولكن السرعة بطيئة. واستعماله في السيارات قليلة.

2. V – belt:

السير على شكل حرف V يستعمل عادة لنقل قدرة أكبر من السير المسطح ويستعمل في السيارات بصورة واسعة.

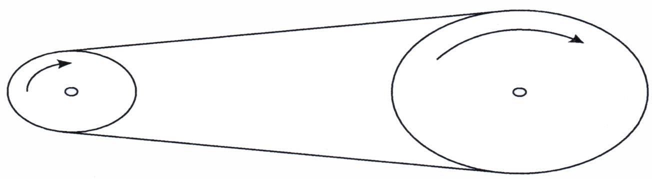
3. Circular belt (rope):

السير دائري المقطع يستعمل لنقل القدرات العالية والسرعات العالية. واستعمالها في السيارات قليلة. وقد يصنع من مواد حديدية كما في حالة الرافعات والمصاعد أو من مواد قطنية وذلك للاستعمالات الخفيفة.

Types of belts systems: **الأنواع الرئيسية لأنظمة السيور:**

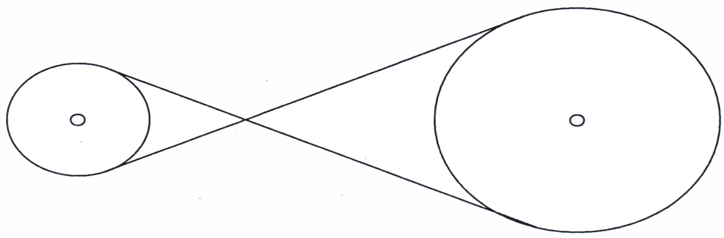
1. Open belt drive:

يستعمل هذا النظام لنقل الطاقة بين محورين متوازيين ويدوران بنفس الاتجاه.



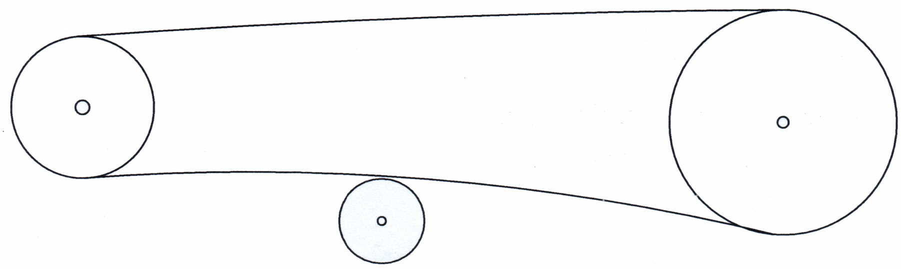
2. Cross belt drive:

يستعمل هذا النظام عند نقل الطاقة بين محورين متوازيين ولكن يدوران باتجاهين محتلفين.



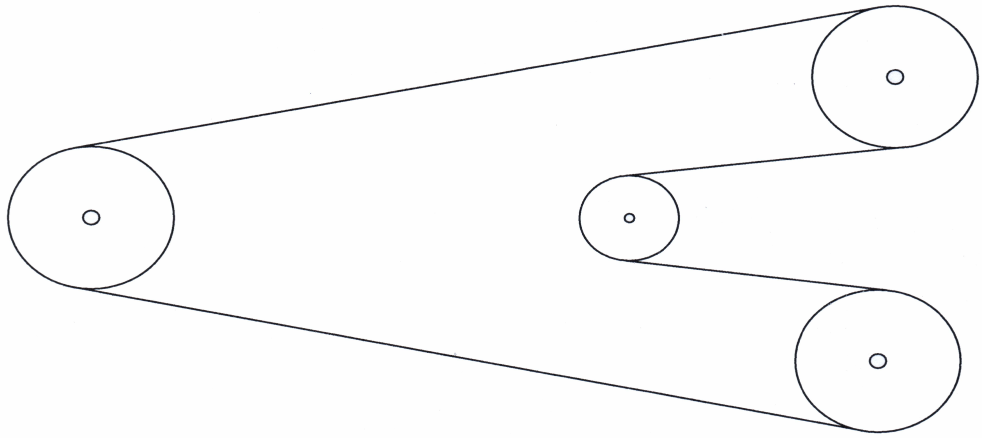
3. Belt drive with idler pulley: نظام السير الفتوح مع البركة الحرة

فائدة البكرة الحرة في هذا النظام هو زيادة قوس التماس بين السير والبكرات وكذلك زيادة الشد في السير.



4. Belt drive many pulleys نظام السير متعدد البكرات

يستعمل هذا النظام عند نقل الطاقة من محور إلى عدة محاور متوازية وباستعمال سبر واحد.



Speed ratio of belt drive:

يمكن حساب نسبة تغيير السرعة بين العمود الدائر والعمود المدار بدلالة أقطار البكرات ومن ثم تحديد سرعة العمود المدار بدلالة سرعة العمود الدائر أو سرعة المحرك.

والآن نفرض:

N_1 = speed of driver , N_2 = speed of driven

d_1 = dia. Of driver , d_2 = dia. of driven

$$\therefore \text{ speed ratio } = \frac{N_1}{N_2} = \frac{d_2}{d_1}$$

وفي حالة اعتبار سمك السير بالحسابات فأن نسبة السرعة تكون كالآتي:-

$$\frac{N_1}{N_2} = \frac{(d_2 + t)}{(d_1 + t)}$$

Power transmitted by belt:

Let

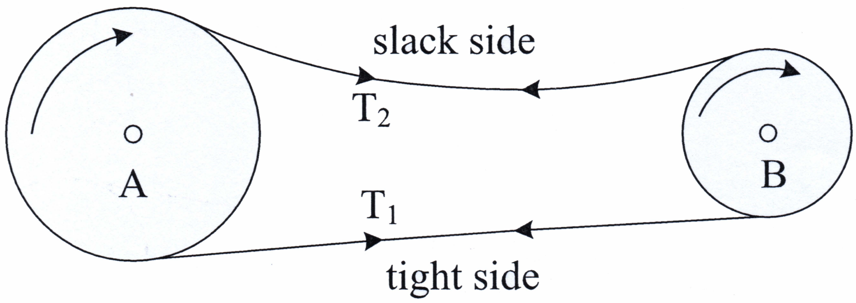
T_1 = tension in the tight side.

T_2 = tension in the slack side.

V = belt speed (m/sec).

$\therefore$ Power , $P = (T_1 - T_2)\, V$

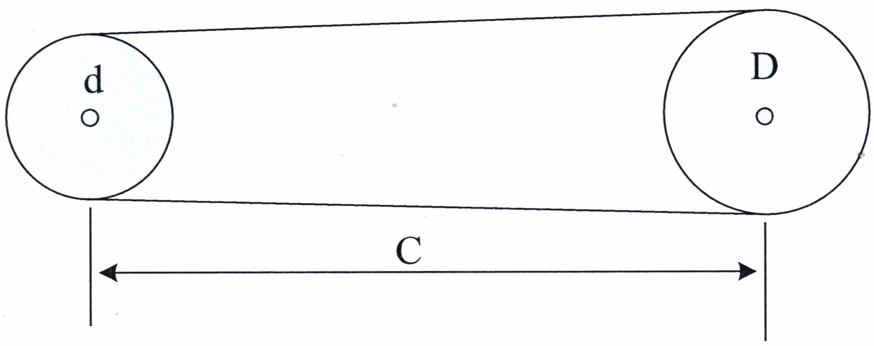
Ex: 1, 2, 3.



Total Length of belt:

a) Open belt:

$$L = 2C + \left(\frac{D+d}{2}\right)\pi + \frac{(D-d)^2}{4C}$$



b) Cross bet:

$$L = 2C + \left(\frac{D+d}{2}\right)\pi + \frac{(D+d)^2}{4C}$$

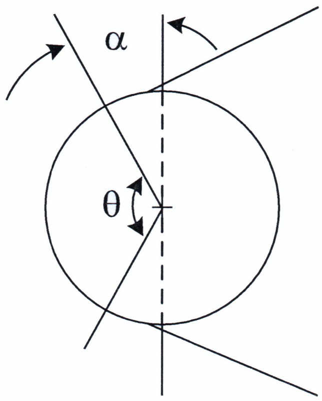
Angle of Contact:

a) Open belt:

ولإيجاد زاوية التماس للبكرة الصغيرة كما يلي:

$$\theta = 180 - 2\alpha$$

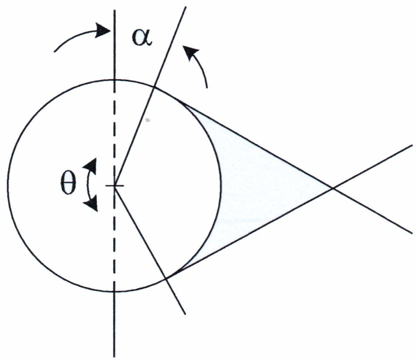
$$\alpha = \mathrm{Sin}^{-1}\frac{D-d}{2C}$$



b) Cross belt:

$$\theta = 180 + 2\alpha$$

$$\alpha = \mathrm{Sin}^{-1}\frac{D+d}{2C}$$



Ex: 4, 5.

Belts

1. The tension on the tight and slack sides of a belt drive are respectively (2.5) KN and (1) KN. The belt pulley is (600) mm diameter and rotates at (105) rev/min.
   Calculate the power transmitted by the belt in kilowatts.
2. A pulley – mounted on the crankshaft of an engine has an effective diameter of (150) mm and drives a dynamo pulley of (100) mm diameter by means of a belt. If the engine runs at (2800) rpm.
   Calculate:
   a) The speed of the dynamo pulley.
   b) The speed of the belt in m/sec.
3. A flat belt transmit (15) KW and drives a pulley (450) mm diameter at (420) rpm. If the maximum tension on the driving side is three times than on the slack side, Calculate the tension on both sides?
4. The diameters of two pulleys are (100) mm & (120) mm respectively, the distance between their centers is (300) mm.
   Find:
   a) The length of a belt required for open & cross belt.
   b) Angle of contact of belt with pulley for opened and cross belt.
5. A V – belt transmit power from an engine to a compressor rotates at a speed of (1250) rev/min.
   Determine:
   a) The diameter of the compressor's pulley.
   b) The belt length.
   c) The angle of contact of belt with pulley.

Belts

Solutions:

1.

$$\text{Belt speed, } V = \pi d_b N_b = \frac{22}{7} \times \frac{600}{1000} \times \frac{105}{60} = 3.3 \text{ m/sec.}$$

$$\text{Power, } P = (T_1 - T_2)V$$

$$= (2500 - 1000) \times 3.3 = 4950 \text{ watt} = 4.95 \text{ KW}$$

2.

a) $\frac{N_a}{N_b} = \frac{d_b}{d_a}$

$$\frac{2800}{N_b} = \frac{100}{150}$$

N_b=4200 rpm.

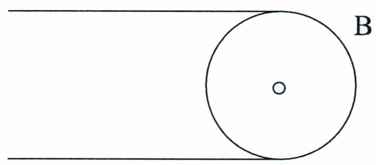
b) $\text{Belt speed, } V = \pi d_a N_a = \frac{22}{7} \times \frac{150}{1000} \times \frac{2800}{60} = 22 \text{ m/sec.}$

OR: $V = \pi d_b N_b =$

3.

d_b=450 mm, N_b=420 rpm, P=15 KW, $T_1=3T_2$, T_1=?, T_2=?

$V = \pi d_b N_b = \frac{22}{7} \times \frac{450}{1000} \times \frac{420}{60} = 9.9$ m/sec.



$P = (T_1 - T_2)V$

$15000 = (3T_2 - T_2) \times 9.9$

T_2=758 N

$T_1=3T_2=3\times758=2274$ N

4.

a) $L_o = 2C + \left(\frac{D+d}{2}\right)\pi + \frac{(D+d)^2}{4C}$

$= 2\times300 + \left(\frac{120+100}{2}\right)\pi + \frac{(120+100)^2}{4\times300} = 946$ mm

$L_c = 2C + \left(\frac{D+d}{2}\right)\pi + \frac{(D-d)^2}{4C}$

$= 2\times300 + \left(\frac{120+100}{2}\right)\pi + \frac{(120+100)^2}{4\times300} = 985.55$ mm

b) For Open Belt:

$\theta = 180 - 2\alpha$

$\alpha = \text{Sin}^{-1}\frac{D-d}{2C} = \text{Sin}^{-1}\frac{120-100}{2\times300} = \text{Sin}^{-1}0.033 = 1.9^\circ$

$\theta = 180 - 2\times1.9 = 176.2^\circ$

For Cross Belt:

$\theta = 180 + 2\alpha$

$\alpha = \text{Sin}^{-1}\frac{D+d}{2C}$

$\alpha = \text{Sin}^{-1}\frac{120+100}{2\times300} = \text{Sin}^{-1}0.366 = 21.5^\circ$

$\therefore\ \theta = 180 + 2\times21.5 = 223^\circ$

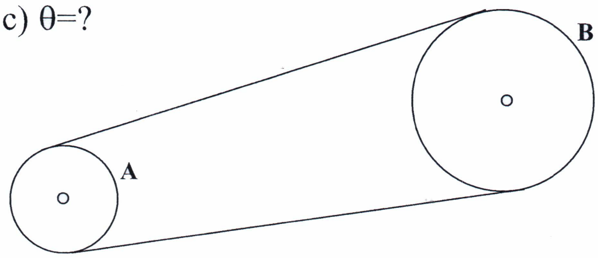
5.

C=75 cm, d_A=10 cm, N_A=2750 rpm

N_B=1250 rpm, a) d_B=? , b) L=? , c) θ=?

a) $\frac{N_A}{N_B} = \frac{d_B}{d_A}$

$\frac{2750}{1250} = \frac{d_b}{10}$



d_B=22 cm قطرة بكرة الكوميرسر

b) $L = 2C + \left(\frac{D+d}{2}\right)\pi + \frac{(D-d)^2}{4C}$

$= 2\times 75 + \left(\frac{22+10}{2}\right)\pi + \frac{(22-10)^2}{4\times 75} = 200.76\ \text{cm}$ طول السير

c) $\theta = 180 - 2\alpha$

$\alpha = \text{Sin}^{-1}\frac{D+d}{2C} = \text{Sin}^{-1}\frac{22+10}{2\times 75} = \text{Sin}^{-1}0.21333 = 12.3^\circ$

$\therefore\ \theta = 180 - 2\times 12.3 = 180 - 24.6 = 155.4^\circ$

CHAINS

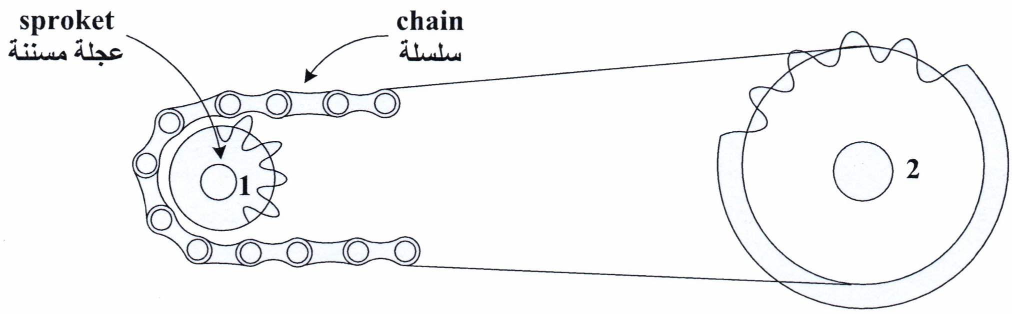
<u>السلسلة</u>:- هي إحدى وسائل نقل الحركة. وتتكون في أبسط أشكالها من:

سلسلة وعجلتين sprockets إحداهما قائدة (1) والأخرى منقادة (2).

وعادة تغلف (تحاط) بصندوق خاص وتزود بنظام تزييت.

وقد شاع استخدام السلاسل في وسائد نقل الحركة التي تصل قدرتها إلى (100KW) وعدة آلاف من الكيلومترات، وسرعة يمكن أن تصل إلى (15 m/s) (54 Km/hr)، وكذلك بنسب تخفيض عالية.

وتستخدم في الدرجات الهوائية والنارية وفي ادارة الاليات المساعدة في عمليات الدرفلة، ومعدات استخراج البترول.



أجزاء السلاسل:

تعتبر السلسلة هي العنصر الأساس في وسيلة نقل الحركة والذي يحدد مدى كفاءتها وتحملها، وهي تتكون من حلقات متصلة ببعضها مفصلياً وتصنع في مصانع خاصة حسب مواصفات قياسية.

توصل نهاية السلسلة بحلقة إضافية قابلة للفك وتكون حلقة أعتيادية.

أهم مميزات السلاسل:

1- لها قابلية على الاستخدام في حالة المسافة الكبرى بين الأعمدة.

2- لها كفاءة عالية تصل إلى حوالي 98%.

3- لها أقل حمل مؤثر في الأعمدة مقارنة بالأحزمة.

4- لها إمكانية نقل الحركة إلى عدة أعمدة باستخدام سلسلة واحدة.

أهم عيوب (مساوئ) السلاسل:

1- تكاليف عالية نسبياً.

2- تحدث ضوضاء أثناء العمل.

3- تحتاج إلى صيانة مستمرة.

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المحاضرة الثامنة

POWER TRANSMITTED BY A SHAFT

Power transmitted by a shaft

<u>العمـود</u>:- هـو جـزء دوار يسـتخدم لنقل القـدرة. وتحديـد أسـلوب نقل القـدرة مهـم بالنسبة للمصـمم وذلك لتجنب تركيز الحمل في نقطـة معينـة من المـادة والتي تؤدي إلى تشقق أو زيادة تشوه العمود.

لدراسة هذه الحالة من الضروري معرفة المصطلحات التالية:

1- الإجهاد المباشر (Direct stress): عندما يتعرض مادة إلى حمل فأنه يكون في حالة أجهاد. $f = \frac{W}{A}$ (). والإجهاد قد يكون: إجهاد شد، إجهاد انضغاط، إجهاد انحناء، أو إجهاد قص.

2- الانفعال (Strain): هو مقياس تشوه المادة عند تعرضها إلى حمل. $\varepsilon \frac{\delta L}{L}$.

3- معامـل اللدونـة (Modulus of elasticity): هـي النسـبة بـين الإجهـاد المباشـر والانفعال. $E = \frac{f}{\varepsilon}$.

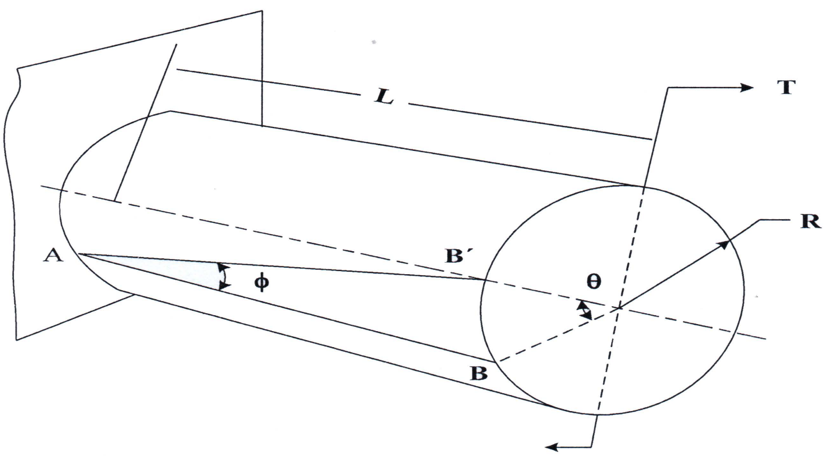
4- اللي البسيط {Pure (Simple) torsion}: هو تعرض مادة إلى عزم لي أو عزم دوران حول محورها الطولي.

5- العلاقة بين إجهاد القص وزاوية اللي: (لاحظ الشكل):

$$\frac{f_s}{R} = \frac{G\theta}{L} \quad \ldots\ldots\ldots\ldots\ldots (1)$$

Where

f_s = shearing stress.
R = radius of the shaft.
G = modulus of rigidity.
θ = angle of twist (in radian) in length L.



6- عزم لي المقاومة: من أجل التوازن فأن العزم المقاوم الكلي يجب أن يعادل العزم المسلط على العمود.

ويعطي بالصيغة التالية:

$$T = \frac{\pi R^4}{2} \times \frac{f_s}{R} = J \times \frac{f_s}{R} \quad \ldots\ldots\ldots\ldots\ldots \ (2)$$

حيث أن : $J = \frac{\pi R^4}{2}$ وتسمى العزم القطبي الثاني لمساحة المقطع العرضي للعمود.

$$\therefore \quad \frac{T}{J} = \frac{f_s}{R} = \frac{G\theta}{L} \quad \ldots\ldots\ldots\ldots\ldots \ (3)$$

والآن يمكن إيجاد العزم في حالة عمود صلد وعمود مجوف كما يلي:

a) Solid circular shaft of dia. (d) :

$$T = \frac{\pi d^3}{16} f_s \quad \ldots\ldots\ldots\ldots\ldots \ (4)$$

b) Hollow circular shaft of outside dia (D) and inside dia. (d):

$$T = \frac{\pi}{16} f_s \left(\frac{D^4 - d^4}{D} \right) \quad \ldots\ldots\ldots\ldots\ldots \ (5)$$

7- القدرة المنقولة بواسطة العمود:

Power transmitted by a shaft,

$$P = T \times \frac{2\pi N}{60} \quad \ldots\ldots\ldots\ldots\ldots \ (6)$$

Where

T = torque in Nm

N = the shaft speed in rev/min.

Ex: 1

A specimen of steel, (20) mm diameter, showed that the elastic limit shear stress was reached when the torque applied was (220) Nm. At this condition, the angle of twist on a length of (200) mm was 2.3°. Determine the modulus of rigidity, and the elastic limit shear for this material.

Solution:

$$\text{Polar second moment, } J = \frac{\pi R^4}{2} = \frac{\pi d^4}{32}$$

$$= \frac{\pi \times 10^4}{2} = 5000\pi \text{ mm}^4$$

$$\text{Angle of twist, } \theta = 2.3^\circ = \frac{2.3}{57.3} = 0.04 \text{ red} \qquad (1 \text{ rad} = 57.3^\circ)$$

$$\frac{T}{J} = \frac{G\theta}{L}$$

$$\frac{220 \times 1000}{5000\,\pi} = \frac{G \times 0.04}{200}$$

$$\therefore \quad G = 70 \times 10^3 \text{ N/mm}^2$$

$$= 70 \text{ KN/mm}^2 = 70 \text{ GN/mm}^2 = 70 \text{ KN/m}^2$$

$$\frac{T}{J} = \frac{f_s}{R}$$

$$\therefore \quad \text{Shear stress, } f_s = \frac{TR}{J} = \frac{220 \times 10^3}{5000\,\pi} = 140 \text{ N/mm}^2 = 140 \text{ MN/m}^2$$

Ex: 2

A tubular steel propeller shaft, (1.25) m long, has an outside diameter of (50) mm and an inside diameter of (43.5) mm.
Determine:

a) The torque which can be transmitted by the shaft if the shear stress is (15) N/mm^2.
b) The power transmitted if the shaft makes (3000) rpm.
c) The angle through which the shaft twists.
(Take $G=85\times10^3\ N/mm^2$).

Solution:

$$T=\frac{\pi}{16}f_s\left(\frac{D^4-d^4}{D}\right)=\frac{\pi}{16}\times\frac{15}{50}\left((50)^4-(43.5)^4\right)$$

$T=157.3\times10^3$ Nmm $=157.3$ Nm

$$\text{Power, } P=T\times\frac{2\pi N}{60}$$

$$=157.3\times\frac{2\pi\times3000}{60}=49.4\times10^3\ \text{watt}=49.4\ \text{KW}$$

$$\frac{f_s}{R}=\frac{G\theta}{L}$$

$$\therefore \text{ Angle of twist, } \theta=\frac{2f_sL}{GD} \qquad (D=2R)$$

$$\theta=\frac{2\times15\times1.25\times10^3}{(85\times10^3\times50)}$$

$=0.0088$ rad

$=0.0088\times57.3=0.5^\circ$

Ex: 3

Determine the diameter of a solid steel shaft which can transmit (33) KW at (1050) rpm, if the shears stress in the material is not to exceed (60) N/mm^2.

Solution:

$$\text{Power, } P = T \times \frac{2\pi N}{60}$$

$$33000 = T \times \frac{2\pi \times 1050}{60}$$

العزم $T = 300\ Nm = 300000\ Nmm$

الإجهاد $f_s = 60\ MN/m^2 = 60 \times \frac{10^6}{10^6} = 60\ N/mm^2$

والآن، فأن العزم لعمود صلد قطره "d" هو:

$$T = \frac{\pi d^3}{16} f_s$$

$$300000 = \frac{\pi d^3}{16} \times 60$$

$d^3 = 25460\ mm^3$

$\therefore d = 29.42\ mm$ قطر العمود

Ex: 4

A specimen of brass, (12) mm diameter, showed that the limit of proportionality was reached when the torque applied was (8.25) Nm. Under this condition, the angle of twist on a length of (400) mm was $(2.35)^\circ$. Determine the modulus of rigidity and the stress for the material.

Solution:

$$d = 12\ mm,\ T = 8.25\ Nm,\ \theta = 2.35^\circ = \frac{2.35}{57.3} = 0.04\ rad$$

$$L = 400\ mm,\ G = ?,\ f_s = ?$$

$$\frac{T}{J} = \frac{G\theta}{L}$$

$$J = \frac{\pi d^4}{32} = \frac{\pi \times (12)^4}{32} = 2036.57\ \ mm^4$$

$$\therefore \frac{8.25 \times 10^3}{2036.57} = \frac{G \times 0.04}{400}$$

$$G = \frac{405}{0.6}\ N/mm^2 = 40.5\ GN/m^2$$

$$\frac{T}{J} = \frac{f_s}{R}$$

$$f_s = \frac{TR}{J} = \frac{8.25 \times 10^3 \times 6}{2036.57}$$

$$= 24.3\ N/mm^2$$

$$= 24.3\ MN/m^2$$

Ex: 5

The propeller shaft of a car is the form of a hollow tube, (50) mm outside diameter and (3) mm thick. Determine the maximum shear stress in the tube when the shaft transmits (52) KW at a speed of (4000) rev/min.

Solution:

$$\frac{D-d}{2}=3\,\text{mm} \quad \rightarrow \quad d=44\,\text{mm}$$

$$P=T\times\frac{2\pi N}{60}$$

$$52000=T\times\frac{2\pi\times 4000}{60}$$

T=124 Nm

$$T=\frac{\pi}{16}f_s\left(\frac{D^4-d^4}{D}\right)$$

$$124\times 10^3=\frac{\pi}{16}f_s\left(\frac{(50)^4-(44)^4}{50}\right)$$

f_s =12.64 N/mm2

=12.64 MN/m2

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المحاضرة السادسة

POWER TRANSMISSION SYSTEMS AND PULLEYS

POWER TRANSMISSION SYSTEMS AND PULLEYS

When the power is to be transmitted between two co-axial shafts, connecting elements like couplings or clutches can be employed on the other hand, if the power is to be transmitted between two non co-axial shafts but may be kept parallel or non parallel and at some distances, we need some intermediate driving elements like belts, chains, or gears. When the co-axial shafts are connected by couplings, their speeds will not differ whereas we can reduce or increase the speed of driven shaft through these intermediate driving links .Shortly saying that the drives are the intermediate mechanism between the driving and driven shafts in order to transmit the power or energy produced in one machine to another or between two members of a machine along with the variation of shaft speeds.

The belts or ropes are used to transmit power from one shaft to another by means of pulleys which rotate at the same speed or at different speeds. The amount of power trans- mitted depends upon the following factors :

1. The velocity of the belt.
2. The tension under which the belt is placed on the pulleys.
3. The arc of contact between the belt and the smaller pulley.
4. The conditions under which the belt is used.

It may be noted that

a. The shafts should be properly in line to insure uniform tension across the belt section.
b. The pulleys should not be too close together, in order that the arc of contact on the smaller pulley may be as large as possible.
c. The pulleys should not be so far apart as to cause the belt to weigh heavily on the shafts, thus in- creasing the friction load on the bearings.
d. A long belt tends to swing from side to side, causing the belt to run out of the pulleys, which in turn develops crooked spots in the belt.
e. The tight side of the belt should be at the bottom, so that whatever sag is present on the loose side will increase the arc of contact at the pulleys.
f. In order to obtain good results with flat belts, the maximum distance between the shafts should not exceed 10 metres and the minimum should not be less than 3.5 times the diameter of the larger pulley

CLASSIFICATION OF POWER TRANSMITTING DRIVES

Modern machines utilize mechanical, hydraulic, pneumatic and electrical drives. The design principles of some commonly adopted mechanical drives are discussed, i.e,. the power transmitting elements may be mechanical items.

Mechanical drives may be classified based on the following conditions.

a) According to the physical conditions of transmission they may be classified into.

i) Friction drives such as belt and rope drives, and

ii) Toothed drives such as gears and chain drives

b) According to the method of linking the driving and driven members, they may be grouped into,

i) Drives with direct contact between the driving and driven members such as gears,

ii) Drives with an intermediate link between the driving and driven members aauch as belts, ropes and chain drives.

c) According to positions of shaft axes as ,

i) Flexible drives: Here the slight variation of shaft axes from parallelism may be permitted because this variation will not affect much the proper function of drive and also the slight variation of centre distance may not be minded much.

Ex: Belt drives, rope drives, chain Drives.

ii) Rigid drives: Here the variations of shaft axes from parallelism and centre distance will not be permitted because of the rigid construction and direct contact of the driving and driven members.

iii) Ex: Gear drives.

ELEMENTS OF A POWER DRIVE

Each transmission mechanism comprises two essential shafts namely the driving(input) shaft and the driven(output) shaft.

The members of power suppliers, lke shafts and pulleys of a motor are called as driving members and the members of power receivers like shafts and pulleys of a machine(say lathe or Rice mill) may be called as driven members.

Each drive, whether it may be a belt, chain, or gear drive, has its specific features and fields of application. The choice of drive depends on the amount of power to be transmitted, peripheral distance between the axes of the mating members.

BELT DRIVE:

It is a mechanical drive in which the driving shaft and driven shaft are connected by a flexible link(i.e belt) through pulleys mounted on the shafts.

Generally, the belts and chin drives are called as flexible drives because they allow the designer considerable flexibility in location of driving and driven machineries and tolerances are not critical as in the case of gear drives. Another advantage of flexible drives, especially of belt drives, is that they reduce vibration and shock transmission.

SELECTION OF A BELT DRIVE

Following are the various important factors upon which the selection of a belt drive depends:

1. Speed of the driving and driven shafts,

2. Speed reduction ratio,

3. Power to be transmitted,

4. Centre distance between the shafts,

5. Positive drive requirements,

6. Shafts layout,

7. Space available, and

8. Service conditions.

TYPES OF BELT DRIVES

The belt drives are usually classified into the following three groups :

1. LIGHT DRIVES. These are used to transmit small powers at belt speeds upto about 10 m/s, as in agricultural machines and small machine tools.

2. MEDIUM DRIVES. These are used to transmit medium power at belt speeds over 10 m/s but up to 22 m/s, as in machine tools.

3. HEAVY DRIVES. These are used to transmit large powers at belt speeds above 22 m/s, as in compressors and generators.

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المحاضرة الخامسة

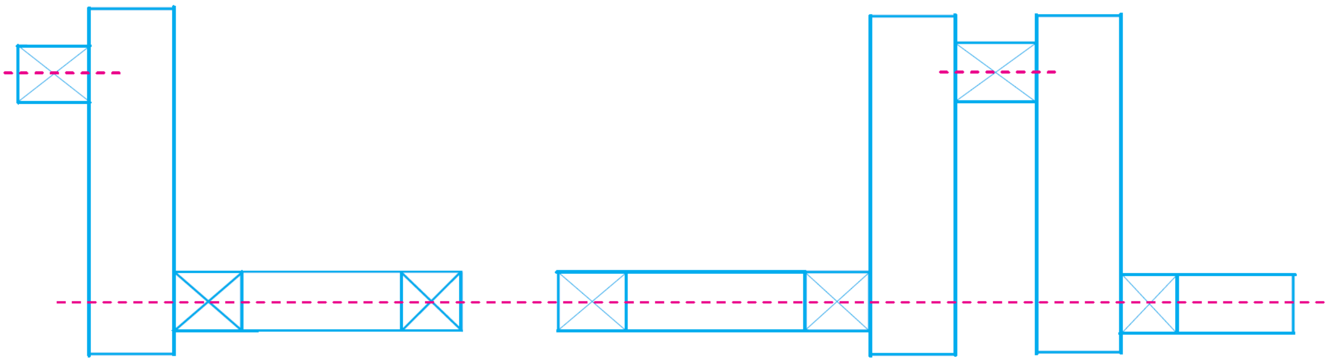
CRANK SHAFT

CRANK SHAFT

A crank shaft (i.e a shaft with a crank) is used to convert reciprocating motion of the piston into rotary motion or vie versa. The crank shaft consists of the shaft parts which revolve in the main bearings, the crank pins to which the big ends of the connecting are connected , the crank arms or webs (also called cheeks) which connect the crank pins and the shaft parts. The crankshaft, depending upon the position of crank, may be divided into the following two types.

1. side crank shaft

2. centre crank shaft.



The crankshaft, depending upon the number of cranks in the shaft, may also be classfied as single throw or multi-throw crankshafts. A crankhaft with only one side crank or centre crank is called a single throw crankshaft whereas the crankshaft with two side cranks, one on each end or with two or more centre cranks is known as multi-throw crankshaft.

The side crankshafts are used for medium and large size horizontal engines.

MATERIAL AND MANUFA CTURE OF CRANKSHAFTS

The crankshafts are subjected to shock and fatigue loads. Thus material of the crankshaft should be tough and fatigue resistant. The crankshafts are generally made of carbon steel, special steel or special cast iron.

In industrial engines, the crankshafts are commonly made from carbon steel such as 40 C 8, 55 C 8 and 60 C 4. In transport engines, manganese steel such as 20 Mn 2, 27 Mn 2 and 37 Mn 2 are generally used for the making of crankshaft. In aero engines, nickel chromium steel such as 35 Ni 1 Cr 60 and 40 Ni 2 Cr 1 Mo 28 are extensively used for the crankshaft.

The crankshafts are made by drop forging or casting process but the former method is more common. The surface of the crankpin is hardened by case carburizing, nitriding or induction hardening.

DESIGN OF OVERHUNG CRANKSHAFT

Overhung crank shaft or side crankshaft of one crank pin, one shaft part (i.e Jounal) and one web which connects the crank pin with the journal. When designing the crankshaft, it is required to discuss about the nature of stresses induced in various parts of the crankshaft.

Let

F = Force transmitted from connecting rod to the crankshaft

A = Area of cross section of crank pin

L = Length of crank pin

d= Diameter of crank pin

w – width of crank web

t = Thickness of crank web

r = Distance between axes of crankpin and journal (i.e crank radius)

x =Distance between the centres of crank pin and journal

ϴ = Angle of inclination of crank from inner dead centre

ϕ = Angle of inclination of the connecting rod with the line of stroke

β = Angle between crank and connecting rod

Fr = radial component of force

Ft= Tangential component of force

Sb= Allowable bending stress

Ss= Allowable shear stress

Sc= Allowable crushing (or) bearing stress.

STRESS INDUCED IN THE CRANKPIN

When the force is transmitted from the connecting rod to the crankshaft, the crankpin is subjected to three types of stresses namely,

i) Plain shear stress due to direct shear force

ii) Bending stress at the fixed end due to the bending moment

iii) Crushing (or) bearing stress acting over the projected area

At any crank angle Ɵ, the force F can be resolved into radial component of force Fr, and tangential component of force Ft. Their magnitudes are

$Fr = F\cos(\theta+\emptyset)$ and $Ft = F\sin(\theta+\emptyset)$

In the case of crank pin, these componetnts of force will not produce any effect on the pin and hence, for the design of crankpin, the actual force F may be considered for all positions of the crank.

Now, the various stresses induced in the crankpin are evaluated as follows.

Plain shear stress $Ss= F/ A = 4 F/ \pi d^2$

Bending moment at the fixed end $M = F \times (l/2)$

(Assuming the force is acting at the centre of pin) Hence bending stress $Sb = 32M/\pi d^3$

$$=16Fl/\pi d^3$$

Bending stress $Sc=$ Force/projected Area $= F/l.d$

It is found that the bearing pressure is a limiting factor in design as it insures proper lubrication.

STRESSES INDUCED IN THE CRANK WEB

Since the force acting on the crank web is having different values for different positions of the crank with respect to the line of stroke, the web is designed based on maximum loading conditions. Usually two positions of crank may be considered for the web design, that is, at zero crank angle and when the included angle between connecting rod and the crank web is 900. When $\theta = 0$, the radial component $Fr = F$ and tangential component $Ft = 0$. Similarly when $\beta = 900$, $Ft = F$, and $Fr = 0$. For other positions of crank, the force is resolved into radial and tangential components and the corresponding induced stresses are evaluated properly.

The various induced stresses in the web at any crank angle are AS FOLLOWS.

I) Direct (or) axial stress by radial force

II) Bending stress due to radial force

III) Bending stress due to tangential force.

Direct stress $So = Fr / w.t$

Bending stress induced in the web due to eccentric application of radial force,

$S_{br} = M / Z$

i.e $S_{br} = (Fr(l/2 + t/2)) / ((1/6) \times wt^2$

$= 3Fr(l + t) / wt^2$

Bending stress induced in the web nearer to the main journal,
$Sbt = M / Z = Ft.r/((1/6) \times w^2t$

$= 6Ft.r / t.w^2$

Resultant maximum stress acting on the web,

$S = S_o + S_{br} + S_{bt}$.

STRESSES INDUCED IN THE CRANK-SHAFT NAIN JOURNAL

The main journal of the crank shaft is also designed similar to web based on induced bending and torsional stresses corresponding to maximum loading positions. The induced stresses on the main journal are

I) Bending stress due to radial force

II) Bending stress due to tangential force

III) Torsional shear stress due to tangential force

Bending stress due to radial force,

$S_{br} = 32M/D^3\pi = 32F_r.x / \pi D^3$

Bending stress due to tangential force,

$S_{bt} = 32M / \pi D^3 = 32 Ft.x / \pi D^3$

These two bending stresses are acting at right angles and hence the resultant bending stress is given by

$Sb= \surd S_{br}^2 + S_{bt}^2 = 32 / \pi D^3(\surd F_r^2 + F_t^2.x) = 32 F x / \pi D^3$

Torsional shear stress due to tangential force ,

$Ss = 16T/ \pi D^3 = 16Ft.r / \pi D^3$

Since the main journal is subjected to bending stress and shear stress, the induced equivalent bending stress and shear stress must be found out.

Equivalent bending stress, $Sbe = \frac{1}{2} [Sb + \surd S_b^2 + 4S_s^2]$

Equivalent shear stress $Sse = \frac{1}{2} [\surd S_b^2 + 4S_s^2]$

Also the main journal must be checked for bearing pressure. For the optimum design of crankshaft, the dimensions of crank shaft parts are selected in such a way that the induced stresses should be less than their allowable values.

DESIGN OF CENTRE CRANK SHAFT

In this type of crankshaft, one crank pin is supported by two webs and the webs are fitted with main journals at both ends. Since the crankshaft resembles a simply supported beam with central loading, the force received from the connecting rod is shared equally by the two journals and the maximum bending moment is developed at the centre of crank pin.

Centre crank shaft is divided into single crank type (or single throw) and multi crank type (or multi throw) depending upon the number of crank pins , which may be employed in single cylinder engine or multi-cylinder engine. For the single throw and multi throw crankshafts, the number of crankpins, webs and the main journals required are as follows.

If we consider as

np = Number of crank pins

nw= Number of webs

nj = Number of main journals.

Then for single throw crank shaft.

np = 1, nw = 2, nj = 2.

For multi throw crankshaft, the number of main journals is usually one more than the number of crankpins. However, the number of main journals and web can be reduced, excluding some between the crankpins, if the rigidity of the crankshaft is increased sufficiently.

i.e for multi throw crank shaft, (say, for four crank model)

$n_p = 4$, $n_w=2$, $n_p = 8$, $n_j = np + 1 = 5$

(or) $n_p = 4$, $n_w = 6$, $n_j = 3$ (in special case)

Similarly for six crank model

$n_p = 6$, $n_w=2n_p=12$, $n_j = n_p + 1 = 7$ or $n_p =6$, $n_w = 10$, $n_j = 5$.

DSIGN OF SINGLE THROW CRANK SHAFT

The single throw crank shaft consists of one crank pin, two webs and two main journals which are rotating inside the main bearings.

The stresses induced in various parts are discussed as follows.

Let

F = Force applied by the connecting rod to the crank shaft.

A = Area of cross-section of crank pin

L = Length of crank pin

D = Diameter of crankl pin

L = Length of main journal

D= Diameter of main journal

W = width of crank web

T = Thickness of crnk web

R = Radius of crank

X = Distance between centres of main journals

Fr = Radial component of force

Ft = Tangential component of force

The centre crank shaft may be considered as a simply supported beam, loaded at the centre (i.e at the crank pin) and supportedat the bearings.

Since the force F is shared by the two journals equally the reaction on each journal is F/2 and the maximum bending moment is developed at the centre of crank pin and is equal to $(\frac{Fx}{4})$

STRESSES INDUCED IN THE CRANK PIN

In this case also, the crank pin is subjected to three types of stresses, similar to overhung crank shaft. They are

I) plain shear stress (or transverse shear stress) due to direct shear force at the area of cross-section. Ss = F/A =4F/ πd^2

II) Bending stress due to bending moment at the centre of the pin

Sb = 32M/ πd^3 = 4Fx/ πd^3

Iii 0 Bearing stress over the projected area,

Sc = F/l.d

STRESSES INDUCED IN THE CRANK WEB

Since this crank shaft is containing two webs, the force supplied by the connnecting rod is shared by these two webs equally and hence the force applied on one web is only half of force. The induced stresses are

I) Direct axial stress by the radial force, S_o = Fr/wt

II) Bending stress due to radial force,

$$Sbr = \frac{M}{Z} = \frac{Force \times distance\ of\ action}{section\ modulus} = \frac{Fr\ \left[\left(\frac{x}{2}\right) - \left(\frac{l}{2} + \frac{t}{2}\right)\right]}{\frac{1}{6}t2w}$$

$$= \frac{3Fr\ [(x) - (l + t)]}{wt2}$$

III) Bending stress induced by the tangential force,

$$Sbr = \frac{M}{Z} = \frac{Fr\ .\ r}{\frac{1}{6}tw2} = \frac{6\ Fr\ .\ r}{tw2}$$

Total resultant stress induced on the web,

$S = S_o + S_{br} + S_{bt}$

Here radial force Fr = $\frac{F}{2}$ cos(θ+Ø)

And tangential force, Ft = $\frac{F}{2}$ sin(θ+Ø)

STRESSES INDUCED IN THE CRANK SHAFT MAIN JOURNAL

Since the centre crank shaft is similar to simply supported beam, the bending moment at the journals is zero. Hence the possible induced stress is due to twisting moment produced by the tangential force.

The torsional shear stress, Ss = $\frac{16T}{\pi D^3} = \frac{16Ft.r}{\pi D^3}$

Where F_t = $\frac{F}{2}$ sin(θ+Ø)

Sometimes fly wheels may be connected at the end of journal. For such cases, the bending moment produced by the weight of the fly wheel on the journal may be taken into account for the design consideration.

DESIGN OF MAIN BEARINGS

The main bearings, into which the crankshaft journals are rotating, are designed based o the bearing pressure developed over the projected area.

If D = Diameter of bearing

L = Length of bearing

Then bearing pressure, pb = $\frac{Load}{projected\ area} = \frac{W}{L.D}$

Where W = F for overhung crank shaft

and W = F/2 for centre crank shaft.

DESIGN OF MULTI-THROW CRANK SHAFT

Since the multi throw crankshaft is simply the multiple structure of single throw crank

shaft, the design of multi-throw crank shaft is very similar to the design of single throw crankshaft.

In this, case , since all the cylinders of the engine posses equal capacity, the force supplied by one cylinder is used for designing one portion of the crank shaft, (i.e one set of crank pin web and journal etc) and for the remaining portions, the same design values are adopted.

DESIGN STRESS VALUES

All parts of crankshaft (i.e crank pin, web & journal) are made of same material and hence they must have common design stress values. Usual design stress values for the crank shaft

material (I.e for mild steel) are

i) In bending : 60 to 100 Mpa

ii) In torsion & compression : 80 to 120 Mpa

iii) In shear : 40 to 60 Mpa

iv) In bearing : 10 to 20 Mpa

The design bearing pressure for the bearings are

i) In crank pin : 4 to 12 Mpa

ii) In main shaft : 1.5 to 2 Mpa

STEPS INVOLVED IN THE DESIGN OF CRANKSHAFT

1. From the given problem, identify the type of crankshaft to be designed, material, steam, or gas pressure and other given parameters.
2. Determine the maximum load acting on the crank pin, maximum torque and bending moments.
3. Find out the parameters of crankpin such as its length, and diameter etc. based on the bearing pressure and check the induced bending and shear stresses with their allowable values.
4. Design the main journal (i.e shaft) based on maximum torque and bending moment conditions and check the bearing pressure.
5. Select the web parameters proportionately and check their induced stresses.
6. In any case, if the induced stress is more than the allowable value, then alter the corresponding dimensions suitably.
7. Usually the following proportions are adopted for the crankshaft parts:

Let d = Diameter of crankpin

D= Diameter of main journal.

Then for overhung crankshaft.

a) Diameter of main journal D = 1.25 to 1.5d
b) Length of main journal l = 1.25D
c) Length of journal inside the crank L1 = 1.0 to 1.25D
d) Length of crank pin l = 1.0 to 1.25d
e) Length of pin inside the crank l1 = 1.0 to 1.25d
f) Thickness of web t=0.7 to 1.0d
g) Width of web nearer to crank pin a= 1.5d
h) Width of web nearer to journal b = 1.5d

For centre crankshaft

a) Diameter of journal D = d
b) Thickness of web t = 0.7d
c) Width of web w = 1.5d

The remaining parameters may be calculated based on design stress values.

Example 32.4. *Design a plain carbon steel centre crankshaft for a single acting four stroke single cylinder engine for the following data:*

Bore = 400 mm ; Stroke = 600 mm ; Engine speed = 200 r.p.m. ; Mean effective pressure = 0.5 N/mm²; Maximum combustion pressure = 2.5 N/mm²; Weight of flywheel used as a pulley = 50 kN; Total belt pull = 6.5 kN.

When the crank has turned through 35° from the top dead centre, the pressure on the piston is 1N/mm² and the torque on the crank is maximum. The ratio of the connecting rod length to the crank radius is 5. Assume any other data required for the design.

Solution. Given : $D = 400$ mm ; $L = 600$ mm or $r = 300$ mm ; $p_m = 0.5$ N/mm² ; $p = 2.5$ N/mm² ; $W = 50$ kN ; $T_1 + T_2 = 6.5$ kN ; $\theta = 35°$; $p' = 1$N/mm² ; $l / r = 5$

We shall design the crankshaft for the two positions of the crank, *i.e.* firstly when the crank is at the dead centre ; and secondly when the crank is at an angle of maximum twisting moment.

1. *Design of the crankshaft when the crank is at the dead centre* (See Fig. 32.18)

We know that the piston gas load,

$$F_P = \frac{\pi}{4} \times D^2 \times p = \frac{\pi}{4}(400)^2\, 2.5 = 314\,200 \text{ N} = 314.2 \text{ kN}$$

Assume that the distance (b) between the bearings 1 and 2 is equal to twice the piston diameter (D).

$$\therefore \quad b = 2D = 2 \times 400 = 800 \text{ mm}$$

and
$$b_1 = b_2 = \frac{b}{2} = \frac{800}{2} = 400 \text{ mm}$$

We know that due to the piston gas load, there will be two horizontal reactions H_1 and H_2 at bearings 1 and 2 respectively, such that

$$H_1 = \frac{F_P \times b_1}{b} = \frac{314.2 \times 400}{800} = 157.1 \text{ kN}$$

and
$$H_2 = \frac{F_P \times b_2}{b} = \frac{314.2 \times 400}{800} = 157.1 \text{ kN}$$

Assume that the length of the main bearings to be equal, *i.e.*, $c_1 = c_2 = c / 2$. We know that due to the weight of the flywheel acting downwards, there will be two vertical reactions V_2 and V_3 at bearings 2 and 3 respectively, such that

$$V_2 = \frac{W \times c_1}{c} = \frac{W \times c/2}{c} = \frac{W}{2} = \frac{50}{2} = 25 \text{ kN}$$

and
$$V_3 = \frac{W \times c_2}{c} = \frac{W \times c/2}{c} = \frac{W}{2} = \frac{50}{2} = 25 \text{ kN}$$

Due to the resultant belt tension ($T_1 + T_2$) acting horizontally, there will be two horizontal reactions H_2' and H_3' respectively, such that

$$H_2' = \frac{(T_1 + T_2)\, c_1}{c} = \frac{(T_1 + T_2)\, c/2}{c} = \frac{T_1 + T_2}{2} = \frac{6.5}{2} = 3.25 \text{ kN}$$

and
$$H_3' = \frac{(T_1 + T_2)\, c_2}{c} = \frac{(T_1 + T_2)\, c/2}{c} = \frac{T_1 + T_2}{2} = \frac{6.5}{2} = 3.25 \text{ kN}$$

Now the various parts of the crankshaft are designed as discussed below:

(a) Design of crankpin

Let d_c = Diameter of the crankpin in mm ;

l_c = Length of the crankpin in mm ; and

σ_b = Allowable bending stress for the crankpin. It may be assumed as 75 MPa or N/mm^2.

We know that the bending moment at the centre of the crankpin,

$$M_C = H_1 \cdot b_2 = 157.1 \times 400 = 62\ 840 \text{ kN-mm} \quad ...(i)$$

We also know that

$$M_C = \frac{\pi}{32}(d_c)^3\sigma_b = \frac{\pi}{32}(d_c)^3 75 = 7.364(d_c)^3 \text{ N-mm}$$

$$= 7.364 \times 10^{-3}\ (d_c)^3 \text{ kN-mm} \quad ...(ii)$$

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

$$(d_c)^3 = 62\ 840 / 7.364 \times 10^{-3} = 8.53 \times 10^6$$

or d_c = 204.35 say 205 mm **Ans.**

We know that length of the crankpin,

$$l_c = \frac{F_P}{d_c \cdot p_b} = \frac{314.2 \times 10^3}{205 \times 10} = 153.3 \text{ say } 155 \text{ mm } \textbf{Ans.}$$

...(Taking p_b = 10 N/mm^2)

(b) Design of left hand crank web

We know that thickness of the crank web,

$$t = 0.65\ d_c + 6.35 \text{ mm}$$

$$= 0.65 \times 205 + 6.35 = 139.6 \text{ say } 140 \text{ mm } \textbf{Ans.}$$

and width of the crank web, $w = 1.125\ d_c + 12.7$ mm

$$= 1.125 \times 205 + 12.7 = 243.3 \text{ say } 245 \text{ mm } \textbf{Ans.}$$

We know that maximum bending moment on the crank web,

$$M = H_1\left(b_2 - \frac{l_c}{2} - \frac{t}{2}\right)$$

$$= 157.1\left(400 - \frac{155}{2} - \frac{140}{2}\right) = 39\ 668 \text{ kN-mm}$$

Section modulus, $Z = \frac{1}{6} \times w.t^2 = \frac{1}{6} \times 245\ (140)^2 = 800 \times 10^3 \text{ mm}^3$

$\therefore$ Bending stress, $\sigma_b = \frac{M}{Z} = \frac{39\ 668}{800 \times 10^3} = 49.6 \times 10^{-3} \text{ kN/mm}^2 = 49.6 \text{ N/mm}^2$

We know that direct compressive stress on the crank web,

$$\sigma_c = \frac{H_1}{w.t} = \frac{157.1}{245 \times 140} = 4.58 \times 10^{-3} \text{ kN/mm}^2 = 4.58 \text{ N/mm}^2$$

$\therefore$ Total stress on the crank web

$$= \sigma_b + \sigma_c = 49.6 + 4.58 = 54.18 \text{ N/mm}^2 \text{ or MPa}$$

Since the total stress on the crank web is less than the allowable bending stress of 75 MPa, therefore, the design of the left hand crank web is safe.

(c) Design of right hand crank web

From the balancing point of view, the dimensions of the right hand crank web (*i.e.* thickness and width) are made equal to the dimensions of the left hand crank web.

(d) Design of shaft under the flywheel

Let d_s = Diameter of the shaft in mm.

Since the lengths of the main bearings are equal, therefore

$$l_1 = l_2 = l_3 = 2\left(\frac{b}{2} - \frac{l_c}{2} - t\right) = 2\left(400 - \frac{155}{2} - 140\right) = 365 \text{ mm}$$

Assuming width of the flywheel as 300 mm, we have

$$c = 365 + 300 = 665 \text{ mm}$$

Allowing space for gearing and clearance, let us take $c = 800$ mm.

$$\therefore \qquad c_1 = c_2 = \frac{c}{2} = \frac{800}{2} = 400 \text{ mm}$$

We know that bending moment due to the weight of flywheel,

$$M_W = V_3 \cdot c_1 = 25 \times 400 = 10\ 000 \text{ kN-mm} = 10 \times 10^6 \text{ N-mm}$$

and bending moment due to the belt pull,

$$M_T = H_3' \cdot c_1 = 3.25 \times 400 = 1300 \text{ kN-mm} = 1.3 \times 10^6 \text{ N-mm}$$

$\therefore$ Resultant bending moment on the shaft,

$$M_S = \sqrt{(M_W)^2 + (M_T)^2} = \sqrt{(10 \times 10^6)^2 + (1.3 \times 10^6)^2}$$
$$= 10.08 \times 10^6 \text{ N-mm}$$

We also know that bending moment on the shaft (M_S),

$$10.08 \times 10^6 = \frac{\pi}{32}(d_s)^3 \sigma_b = \frac{\pi}{32}(d_s)^3 42 = 4.12\ (d_s)^3$$

$$\therefore \qquad (d_s)^3 = 10.08 \times 10^6 / 4.12 = 2.45 \times 10^6 \text{ or } d_s = 134.7 \text{ say } 135 \text{ mm}$$ **Ans.**

2. *Design of the crankshaft when the crank is at an angle of maximum twisting moment*

We know that piston gas load,

$$F_P = \frac{\pi}{4} \times D^2 \times p' = \frac{\pi}{4}(400)^2 1 = 125\ 680 \text{ N} = 125.68 \text{ kN}$$

In order to find the thrust in the connecting rod (F_Q), we should first find out the angle of inclination of the connecting rod with the line of stroke (*i.e.* angle ϕ). We know that

$$\sin\phi = \frac{\sin\theta}{l/r} = \frac{\sin 35°}{5} = 0.1147$$

$$\therefore \qquad \phi = \sin^{-1}(0.1147) = 6.58°$$

We know that thrust in the connecting rod,

$$F_Q = \frac{F_P}{\cos\phi} = \frac{125.68}{\cos 6.58°} = \frac{125.68}{0.9934} = 126.5 \text{ kN}$$

Tangential force acting on the crankshaft,

$$F_T = F_Q \sin(\theta + \phi) = 126.5 \sin(35° + 6.58°) = 84 \text{ kN}$$

and radial force, $$F_R = F_Q \cos(\theta + \phi) = 126.5 \cos(35° + 6.58°) = 94.6 \text{ kN}$$

Due to the tangential force (F_T), there will be two reactions at bearings 1 and 2, such that

$$H_{T1} = \frac{F_T \times b_1}{b} = \frac{84 \times 400}{800} = 42 \text{ kN}$$

and $$H_{T2} = \frac{F_T \times b_2}{b} = \frac{84 \times 400}{800} = 42 \text{ kN}$$

Due to the radial force (F_R), there will be two reactions at bearings 1 and 2, such that

$$H_{R1} = \frac{F_R \times b_1}{b} = \frac{94.6 \times 400}{800} = 47.3 \text{ kN}$$

$$H_{R2} = \frac{F_R \times b_2}{b} = \frac{94.6 \times 400}{800} = 47.3 \text{ kN}$$

Now the various parts of the crankshaft are designed as discussed below:

(a) Design of crankpin

Let d_c = Diameter of crankpin in mm.

We know that the bending moment at the centre of the crankpin,

$$M_C = H_{R1} \times b_2 = 47.3 \times 400 = 18\ 920 \text{ kN-mm}$$

and twisting moment on the crankpin,

$$T_C = H_{T1} \times r = 42 \times 300 = 12\ 600 \text{ kN-mm}$$

$\therefore$ Equivalent twisting moment on the crankpin,

$$T_e = \sqrt{(M_C)^2 + (T_C)^2} = \sqrt{(18\ 920)^2 + (12\ 600)^2}$$

$$= 22\ 740 \text{ kN-mm} = 22.74 \times 10^6 \text{ N-mm}$$

We know that equivalent twisting moment (T_e),

$$22.74 \times 10^6 = \frac{\pi}{16}(d_c)^3 \tau = \frac{\pi}{16}(d_c)^3 35 = 6.873\ (d_c)^3$$

...(Taking τ = 35 MPa or N/mm^2)

$\therefore$ $(d_c)^3 = 22.74 \times 10^6 / 6.873 = 3.3 \times 10^6$ or $d_c = 149$ mm

Since this value of crankpin diameter (*i.e.* d_c = 149 mm) is less than the already calculated value of d_c = 205 mm, therefore, we shall take d_c = 205 mm. **Ans.**

(b) Design of shaft under the flywheel

Let d_s = Diameter of the shaft in mm.

The resulting bending moment on the shaft will be same as calculated eariler, *i.e.*

$$M_S = 10.08 \times 10^6 \text{ N-mm}$$

and twisting moment on the shaft,

$$T_S = F_T \times r = 84 \times 300 = 25\ 200 \text{ kN-mm} = 25.2 \times 10^6 \text{ N-mm}$$

$\therefore$ Equivalent twisting moment on shaft,

$$T_e = \sqrt{(M_S)^2 + (T_S)^2}$$

$$= \sqrt{(10.08 \times 10^6)^2 + (25.2 \times 10^6)^2} = 27.14 \times 10^6 \text{ N-mm}$$

We know that equivalent twisting moment (T_e),

$$27.14 \times 10^6 = \frac{\pi}{16}(d_s)^3 \tau = \frac{\pi}{16}(135)^3 \tau = 483\ 156\ \tau$$

$\therefore$ $\tau = 27.14 \times 10^6 / 483\ 156 = 56.17$ N/mm^2

From above, we see that by taking the already calculated value of ds = 135 mm, the induced shear stress is more than the allowable shear stress of 31 to 42 MPa. Hence, the value of d_s is calculated by taking τ = 35 MPa or N/mm^2 in the above equation, *i.e.*

$$27.14 \times 10^6 = \frac{\pi}{16}(d_s)^3 35 = 6.873\ (d_s)^3$$

$\therefore$ $(d_s)^3 = 27.14 \times 10^6 / 6.873 = 3.95 \times 10^6$ or d_s = 158 say 160 mm **Ans.**

(c) Design of shaft at the juncture of right hand crank arm

Let d_{s1} = Diameter of the shaft at the juncture of the right hand crank arm.

We know that the resultant force at the bearing 1,

$$R_1 = \sqrt{(H_{T1})^2 + (H_{R1})^2} = \sqrt{(42)^2 + (47.3)^2} = 63.3 \text{ kN}$$

$\therefore$ Bending moment at the juncture of the right hand crank arm,

$$M_{S1} = R_1\left(b_2 + \frac{l_c}{2} + \frac{t}{2}\right) - F_Q\left(\frac{l_c}{2} + \frac{t}{2}\right)$$

$$= 63.3 \left(400 + \frac{155}{2} + \frac{140}{2}\right) - 126.5 \left(\frac{155}{2} + \frac{140}{2}\right)$$

$$= 34.7 \times 10^3 - 18.7 \times 10^3 = 16 \times 10^3 \text{ kN-mm} = 16 \times 10^6 \text{ N-mm}$$

and twisting moment at the juncture of the right hand crank arm,

$$T_{S1} = F_T \times r = 84 \times 300 = 25\,200 \text{ kN-mm} = 25.2 \times 10^6 \text{ N-mm}$$

∴ Equivalent twisting moment at the juncture of the right hand crank arm,

$$T_e = \sqrt{(M_{S1})^2 + (T_{S1})^2}$$

$$= \sqrt{(16 \times 10^6)^2 + (25.2 \times 10^6)^2} = 29.85 \times 10^6 \text{ N-mm}$$

We know that equivalent twisting moment (T_e),

$$29.85 \times 10^6 = \frac{\pi}{16}(d_{s1})^3 \tau = \frac{\pi}{16}(d_{s1})^3 42 = 8.25\,(d_{s1})^3$$

...(Taking τ = 42 MPa or N/mm²)

∴ $(d_{s1})^3 = 29.85 \times 10^6 / 8.25 = 3.62 \times 10^6$ or $d_{s1} = 153.5$ say 155 mm **Ans.**

(d) Design of right hand crank web

Let σ_{bR} = Bending stress in the radial direction ; and

σ_{bT} = Bending stress in the tangential direction.

We also know that bending moment due to the radial component of F_Q,

$$M_R = H_{R2}\left(b_1 - \frac{l_c}{2} - \frac{t}{2}\right) = 47.3 \left(400 - \frac{155}{2} - \frac{140}{2}\right) \text{kN-mm}$$

$$= 11.94 \times 10^3 \text{ kN-mm} = 11.94 \times 10^6 \text{ N-mm} \quad ...(i)$$

We also know that bending moment,

$$M_R = \sigma_{bR} \times Z = \sigma_{bR} \times \frac{1}{6} \times w.t^2 \quad ...(\because Z = \frac{1}{6} \times w.t^2)$$

$$11.94 \times 10^6 = \sigma_{bR} \times \frac{1}{6} \times 245\,(140)^2 = 800 \times 10^3 \sigma_{bR}$$

∴ $\sigma_{bR} = 11.94 \times 10^6 / 800 \times 10^3 = 14.9$ N/mm² or MPa

We know that bending moment due to the tangential component of F_Q,

$$M_T = F_T\left(r - \frac{d_{s1}}{2}\right) = 84\left(300 - \frac{155}{2}\right) = 18\,690 \text{ kN-mm}$$

$$= 18.69 \times 10^6 \text{ N-mm}$$

We also know that bending moment,

$$M_T = \sigma_{bT} \times Z = \sigma_{bT} \times \frac{1}{6} \times t.w^2 \quad ...(\because Z = \frac{1}{6} \times t.w^2)$$

$$18.69 \times 10^6 = \sigma_{bT} \times \frac{1}{6} \times 140(245)^2 = 1.4 \times 10^6 \sigma_{bT}$$

∴ $\sigma_{bT} = 18.69 \times 10^6 / 1.4 \times 10^6 = 13.35$ N/mm² or MPa

Direct compressive stress,

$$\sigma_b = \frac{F_R}{2w \cdot t} = \frac{94.6}{2 \times 245 \times 140} = 1.38 \times 10^{-3} \text{ kN/mm}^2 = 1.38 \text{ N/mm}^2$$

and total compressive stress,

$$\sigma_c = \sigma_{bR} + \sigma_{bT} + \sigma_d$$
$$= 14.9 + 13.35 + 1.38 = 29.63 \text{ N/mm}^2 \text{ or MPa}$$

We know that twisting moment on the arm,

$$T = H_{T2}\left(b_1 - \frac{l_c}{2}\right) = 42\left(400 - \frac{155}{2}\right) = 13\ 545 \text{ kN-mm}$$
$$= 13.545 \times 10^6 \text{ N-mm}$$

and shear stress on the arm,

$$\tau = \frac{T}{Z_P} = \frac{4.5T}{w.t^2} = \frac{4.5 \times 13.545 \times 10^6}{245\ (140)^2} = 12.7 \text{ N/mm}^2 \text{ or MPa}$$

We know that total or maximum combined stress,

$$(\sigma_c)_{max} = \frac{\sigma_c}{2} + \frac{1}{2}\sqrt{(\sigma_c)^2 + 4\tau^2}$$
$$= \frac{29.63}{2} + \frac{1}{2}\sqrt{(29.63)^2 + 4\ (12.7)^2} = 14.815 + 19.5 = 34.315 \text{ MPa}$$

Since the maximum combined stress is within the safe limits, therefore, the dimension w = 245 mm is accepted.

(*e*) *Design of left hand crank web*

The dimensions for the left hand crank web may be made same as for right hand crank web.

(*f*) *Design of crankshaft bearings*

Since the bearing 2 is the most heavily loaded, therefore, only this bearing should be checked for bearing pressure.

We know that the total reaction at bearing 2,

$$R_2 = \frac{F_P}{2} + \frac{W}{2} + \frac{T_1 + T_2}{2} = \frac{314.2}{2} + \frac{50}{2} + \frac{6.5}{2} = 185.35 \text{ kN} = 185\ 350 \text{ N}$$

$\therefore$ Total bearing pressure

$$= \frac{R_2}{l_2 \cdot d_{s1}} = \frac{185\ 350}{365 \times 155} = 3.276 \text{ N/mm}^2$$

Since this bearing pressure is less than the safe limit of 5 to 8 N/mm^2, therefore, the design is safe.