UNIT 2

ENGINE PARTS

Course objectives:

- To design the engine parts like piston, connecting rod and analyze design procedure different loading conditions

Course outcomes:

Student will be able to:

- Calculate the design parameter for energy storage element and engine components, connecting rod and piston

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 sometimes 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.

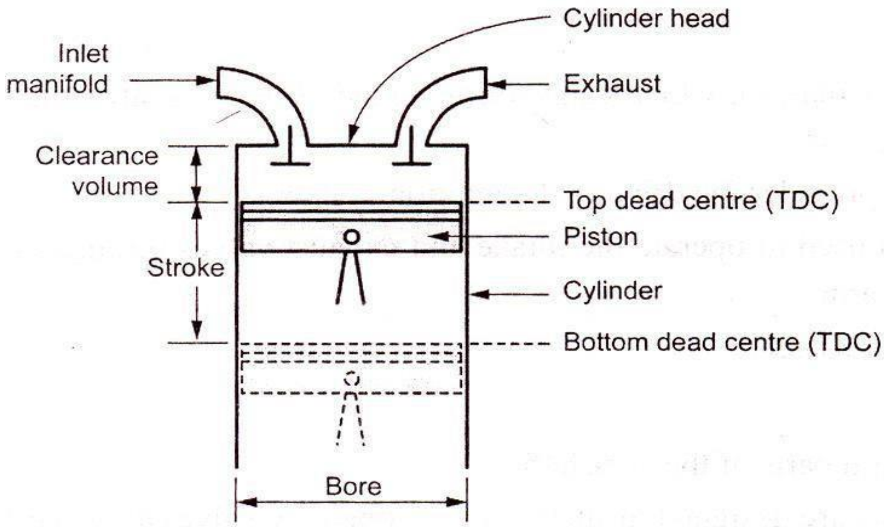
CLASSIFICATION OF I.C ENGINES

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), In case 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 (V_c): 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 (V_s): 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 (r_c): 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,

$$r_c = \frac{\text{maximum cylinder volume}}{\text{minimum cylinder volume}} = \frac{\text{Swept volume+clearance volume}}{\text{clearance volume}}$$

S.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
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

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.

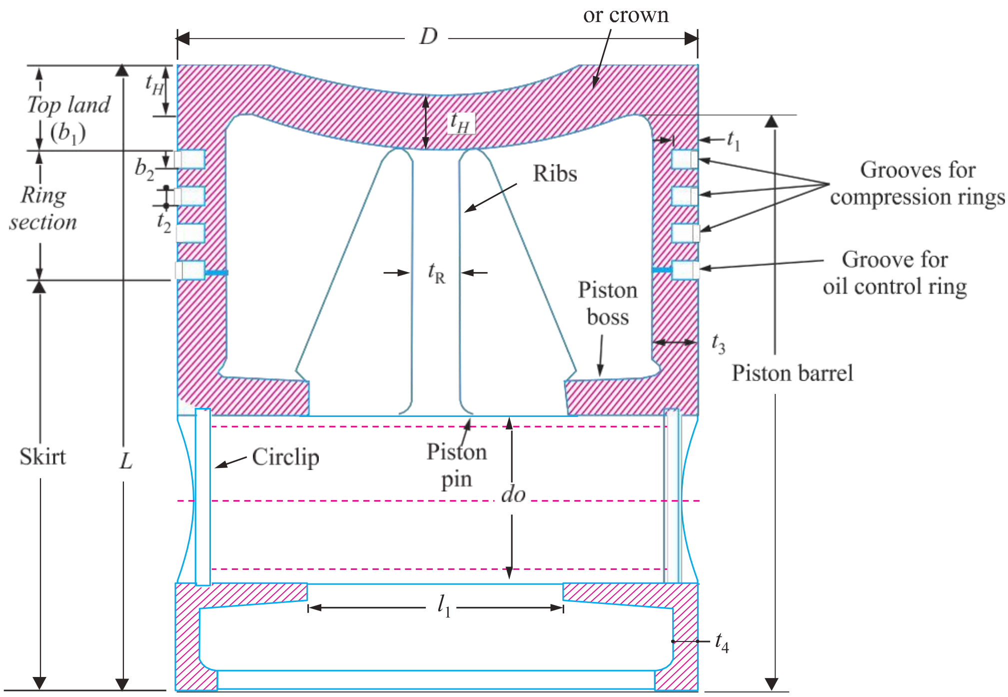


Fig: Piston for I.C Engine

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: e 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 minimize 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 disperse 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 dimensions 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 (t_1), according to Grashoff's formula is given by

$$t1 = \frac{\sqrt{3pnD2}}{16atp}\text{ mm}$$

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,

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

where H= Heat following through the head (KW)

$$H = C \times m \times C_v \times P_B$$

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)

$$= \frac{P\text{mbL}A\text{n}}{60000} \text{ kw}$$

P_{mb}= Brake mean effective pressure (N/mm^2)

L= stroke length (mm)

A= Area of piston at its top side (mm2)

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 oil 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

$$t_3 = \frac{D\sqrt{b^2 - 4ac}}{\sigma br}$$

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 t_1 to $1.2t_1$ from top. The lands between the rings may be equal to or less than the axial thickness of ring t_4. The gap between the free ends of the ring is taken as

$$C = (3.5 \text{ to } 4)\, t_3$$

Where t_3 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 maximum thickness of barrel nearer to piston head is given by, $t_5 = 0.03D+b+4.5$ mm

Where b= radial depth of ring-groove $b= t_3+0.4$mm

The thickness of barrel at the open end of the piston, $t_6= (0.25 \text{ to } 0.35)\, t_5$ mm

PISTON SKIRT

The portion of the piston barrel below the ring selection up to 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/mm^2 for slow speed engines and 0.5 N/mm^2 for high speed engines.

The side thrust force is given by,

$Fs = \mu F_g$

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

Fg= Gas force = $\frac{\pi D2}{4} p_m$

The side thrust pressure, $p_s = \frac{\text{side thruct force}}{\text{projected area}} = \frac{Fc}{Lc*D}$

Length of skirt (Ls) = $\frac{Fc}{Pc*D}$

Where, D = Bore diameter.

LENGTH OF PISTON

The length of piston, L_p can be obtained as
$L_p = L_s$ + Length of ring section + Top land
Empirically L_p= 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 F_b=Bearing pressure x Projected area

$$F_b = p_b.l.d$$

By equating this bearing strength to gas force G_g, we get

$$P_b.l.d = Fg \text{ (therefore } Fg = \frac{\pi}{4} D^2 p_m$$

Usually, l/d= 1.5 to 2
The piston pin is checked for bending as, the induced bending stress

$$\sigma_b = \frac{32M}{\pi d^3} < \sigma_b$$

where M = Bending moment = $FgD/8$

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

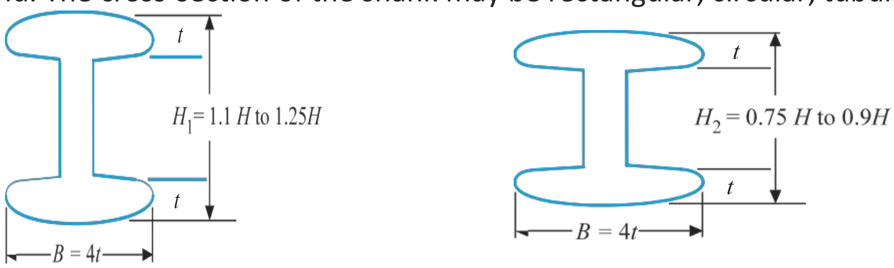
The gudgeon pin is fitted at a distance of (Ls/2) from open end where Ls is the skirt-length.

PISTON CLEARENCE

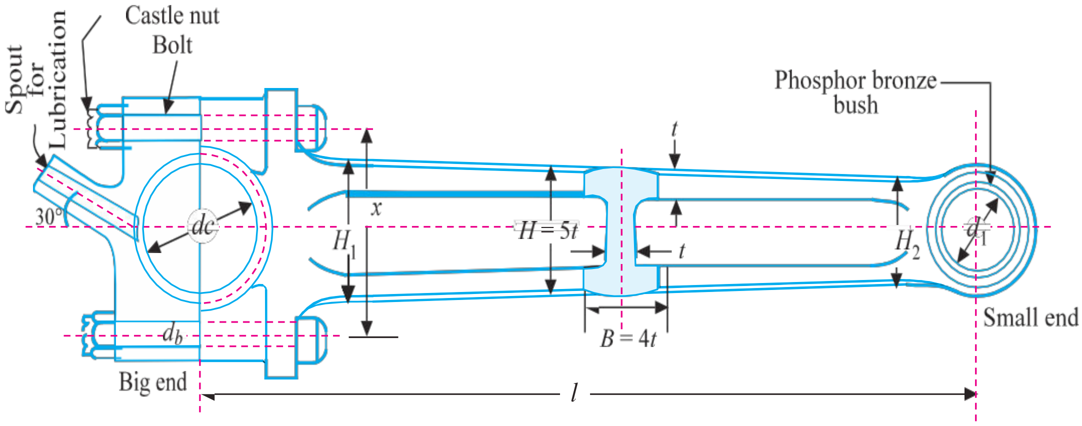
Proper clearance must be provided between the piston and liner to take care of thermal expansion and distortion under load. Usually the clearance may be between 0.04mm to 0.20 mm, depending upon the engine design and piston dia. small clearance may be adopted for the pistons cooled by oil(or) water.

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 aero engines 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, crank webs 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 cylinder liner by the oil scraper rings.

FORCES ACTING ON THE CONNECTING ROD

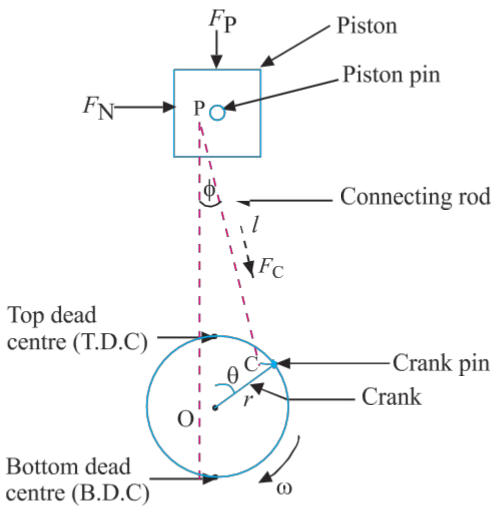
The various forces acting on the connecting rod are as follows

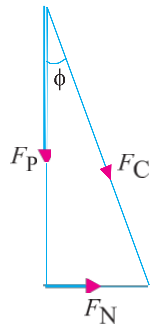
1. Force on the piston due to gas pressure and inertia of the reciprocating parts,
2. Force due to inertia of the connecting rod or inertia bending forces,
3. Force due to friction of the piston rings and of the piston, and
4. Force due to friction of the piston pin bearing and the crankpin bearing.

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 ofgas,

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,

Θ= 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 A_p

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,

F_p = Pressure × Area = $p \cdot A_p = p \times \pi D^2 /4$

And the inertia force of the reciprocating parts

Fi= mass x Acceleration

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

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

$$Fc = \frac{Fp - Fi}{coc\emptyset} = \frac{Fp}{coc\emptyset}$$

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

$$Fc = \frac{Fp}{coc\emptyset}$$

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 $\phi = \phi$max when ϴ=900.Hence the maximum force acting on the connecting rod, is given by

$$Fc_{max} = \frac{Fp}{coc\emptyset}$$

In general, n should be at least 3

Hence for n=l/r =3, $Fc=1.06F_p$

N=4, $F_c=1.03F_p$

N=5, F_c=1.02Fp

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

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 = $\frac{M_{max}}{Z}$

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 l_1, d_1=length and diameter of piston (i.e small end respectively)

l_2, d_2 = Length and diameter of crank pin (i.e big end respectively)

pb_1, pb_2 = Design bearing pressures for the small end and big end respectively

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

$$\mathbf{F_1 = pb_1.l_1.d_1}$$

And the bearing load applied on the crank pin (i.e big end) is given by $\mathbf{F_2 = pb_2.l_2.d_2}$

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

$$Pb_1 = 12.5 \text{ to } 15.4 \text{ N/mm}^2$$

$$Pb_2 = 10.8 \text{ to } 12.6 \text{ N/mm}^2$$

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

$$\text{Inertia force } F_i = mr\omega^2\left(\cos\Theta + \frac{\cos 2\theta}{n}\right)$$

$$n = \frac{S}{r} = \frac{\text{Length of connecting rod}}{\text{crank radius}}$$

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

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

For two bolts $F_{im} = 2 * \frac{\pi}{4} D^2 * S_t$

The nominal diameter may be selected from the manufacture's table (usually d_c=0.84 d_b , where d_b 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 = F_{im}

Hence $M = F_{im} l^l / 6$

l^l = Distance 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 = S_{be} = 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.

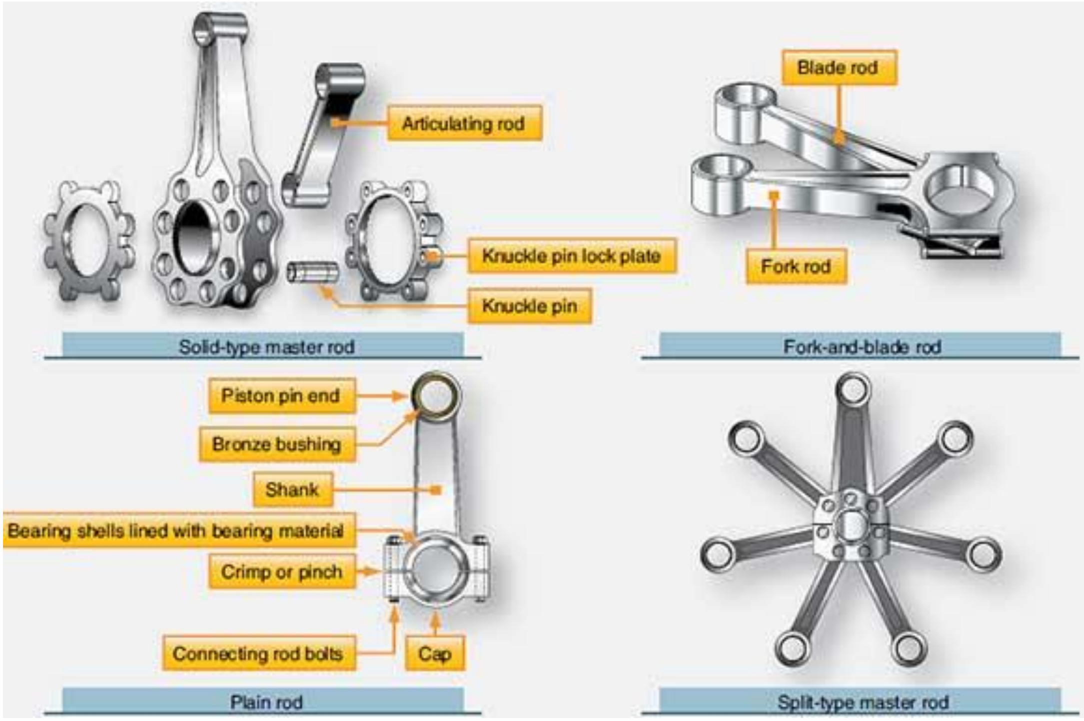
BUCKLING LOAD or CRIPPLING LOAD

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.

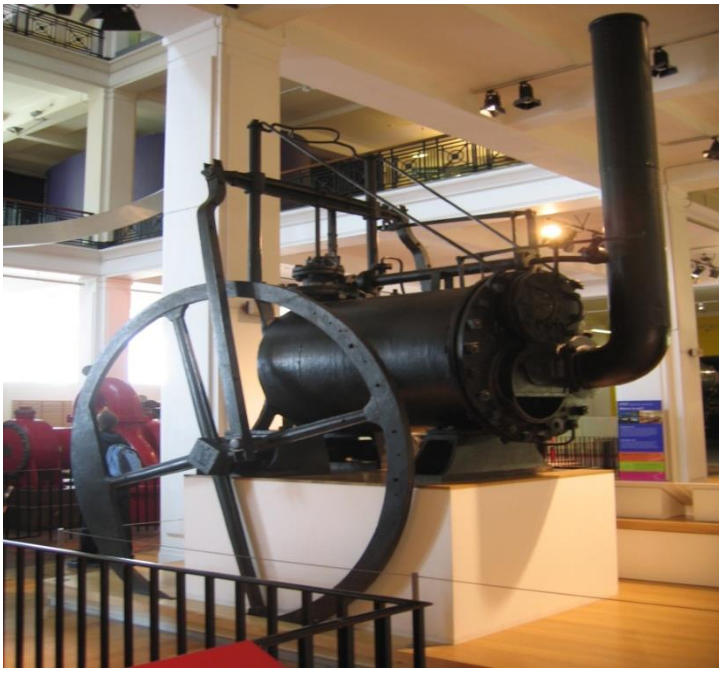
When the connecting rod or piston rod are subjected to compressive load, they may fracture when the applied compressive load is more than their resisting compressive strength. At the same time, if the length of rods have been increased beyond certain limit with respect to their gross sectional dimensions (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.

INDUSTRIAL APPLICATIONS

1. Engines in Automobile



2. In boilers



3. Marine steam boilers

1.

TUTORIAL QUESTIONS

UNIT 2

1. Design a cast iron piston for a single acting four stroke engine for the following specifications:
   Cylinder bore =100mm, Stroke=120mm, Maximum gas pressure =5 N/mm^2
   Brake mean effective pressure=0.65 N/mm^2, Fuel consumption= 0.227 kg/KW/hr
   Speed=2200 rev/min, Assume suitable data.

2. Determine the dimensions of small and big end bearings of the connecting rod for a diesel engine with the following data:

Cylinder bore = 100 mm
Maximum gas pressure = 2.45 MPa
(l/d) ratio for piston pin bearing = 1.5 (l/d)
ratio for crank pin bearing = 1.4
Allowable bearing pressure for piston pin bearing = 15 MPa
Allowable bearing pressure for crank pin bearing = 10 MPa

3. The following data is given for the piston of a four-stroke diesel engine: Cylinder head = 250 mm

Material of piston rings = Grey cast iron
Allowable tensile stress = 100 N/mm^2
Allowable radial pressure on cylinder wall = 0.03 MPa
Thickness of piston head = 42 mm
Number of piston rings = 4

Calculate all the dimensions related to piston and piston rings.

4. Design a connecting rod for four stroke petrol engine with the following data.
   Piston diameter = 0.10 m, stroke = 0.14 m, length of the connecting rod from centre to centre = 0.315 m, weight of reciprocating parts = 18.2 N, speed = 1500 rpm with possible over speed of 2500 compression ratio =4:1, probable maximum explosion pressure = 2.45 Mpa.

ASSIGNMENT QUESTIONS

1. Design a trunk type cast iron piston for a 4-stroke diesel engine with the following specifications: cylinder bore = 250 mm, stroke length = 375 mm, speed = 600 rpm, maximum gas pressure = 5Mpa, indicated mean effective pressure = 0.8 MPa, rate of fuel consumption = 0.3 Kg/BP/H, higher calorific value of fuel = 44 MJ/Kg, mechanical efficiency = 80 %. Sate clearly the design decision taken.
2. Design a trunk type cast iron piston for a 4-stroke diesel engine with the following specifications: cylinder bore = 250 mm , stroke length = 375 mm, speed = 600 rpm, maximum gas pressure = 5Mpa, indicated mean effective pressure = 0.8 MPa, rate of fuel consumption = 0.3 Kg/BP/H, higher calorific value of fuel = 44 MJ/Kg, mechanical efficiency = 80 %. Sate clearly the design decision taken.
3. A connecting rod is required to be designed for a high speed, four stroke I.C. engine. The following data are available. Diameter of piston = 88 mm; Mass of reciprocating parts = 1.6 kg; Length of connecting rod (centre to centre) = 300 mm; Stroke = 125 mm; R.P.M. = 2200 (when developing 50 kW); Possible over speed = 3000 r.p.m.; Compression ratio = 6.8: 1 (approximately); Probable maximum explosion pressure (assumed shortly after dead centre, say at about 3°) = 3.5 N/mm2.
4. Design a CI piston for a single acting four stroke petrol engine of the following specifications :
   Cylinder bore = 100mm
   Stroke Length =120mm
   Maximum gas pressure = 5MPa
   Break mean effective Pressure =0.65MPa
   Fuel Consumption = 0.17kg/bhp/min
   Speed =220rpm

ENGINE PARTS

UNIT 2

MRCET CAMPUS

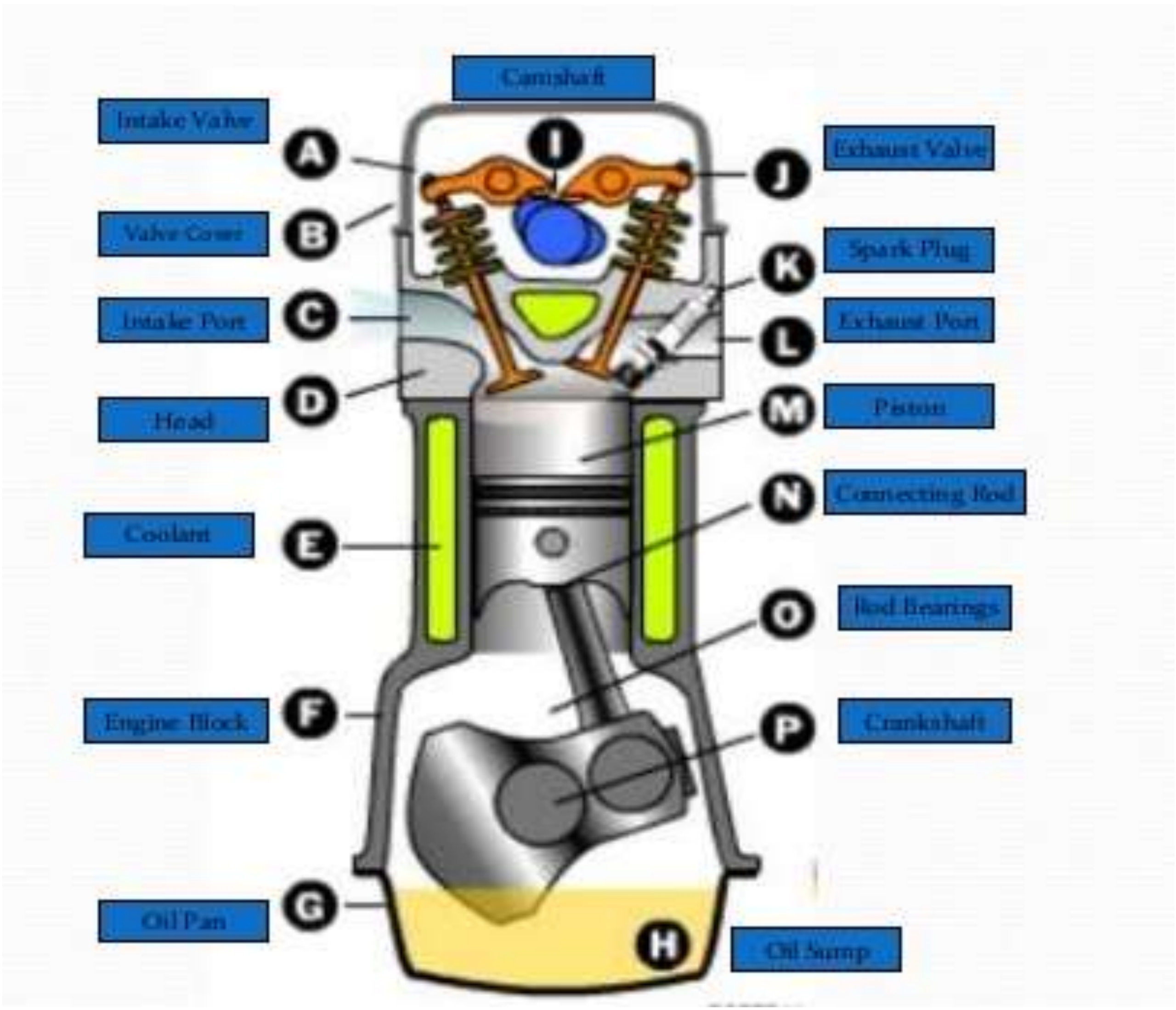
UGC Autonomous

DEPARTMENT OF MECHANICAL ENGINEERING

BASIC PARTS OF AN ENGINE

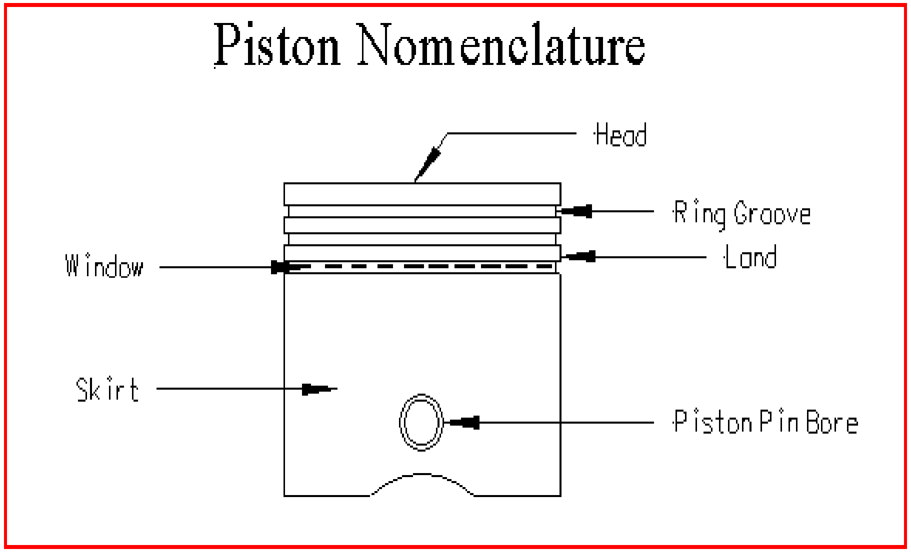
- Cylinder block
- Piston
- Piston rings
- Piston pin
- Connecting rod
- Crankshaft
- Cylinder head
- Intake valve
- Exhaust valve
- Camshaft
- Spark plug

BASIC COMPONENTS



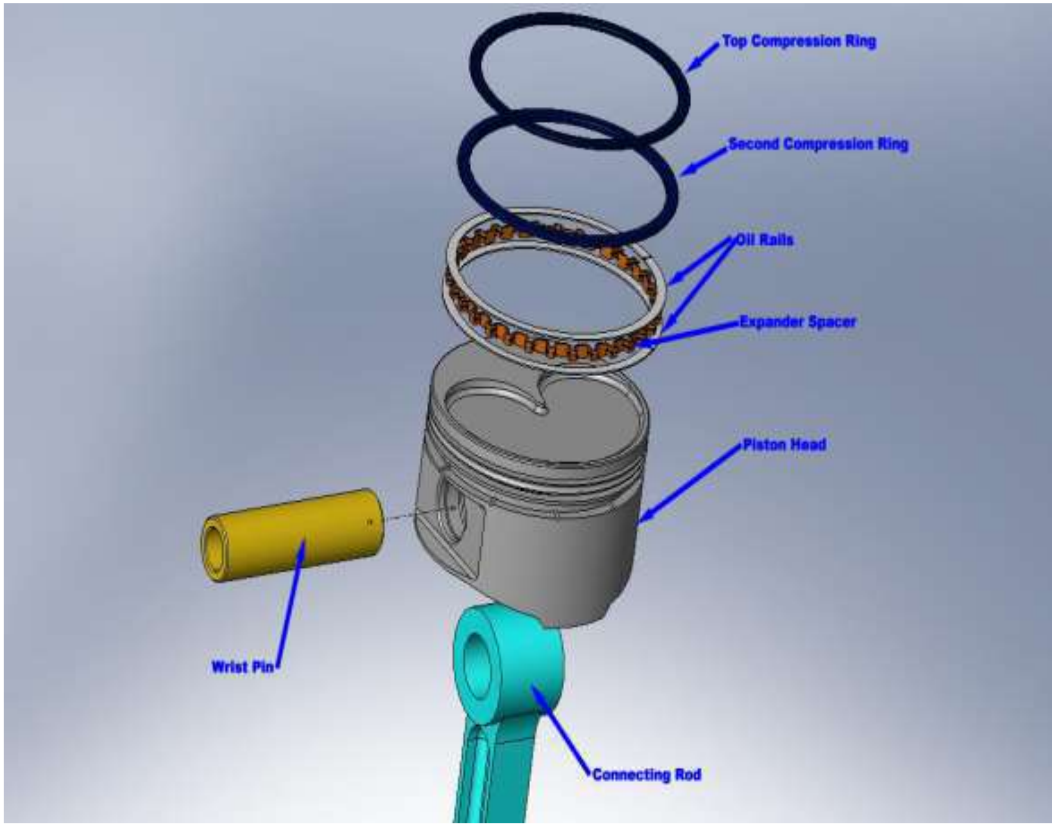
PISTON

The part of the engine that moves up and down in the cylinder converting the gasoline into motion



PISTON RING

- The rings seal the compression gases above the piston keep the oil below the piston rings.



PISTON PIN

- Also known as the wrist pin, it connects the piston to the small end of the connecting rod.
- It transfers the force and allows the rod to swing back and forth.

CONNECTING ROD

Links the piston to the crankshaft.