 ISSN: 2321-8169
3843 – 3851

A Review of Internal Combustion Engine Design

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Abstract— The most successful inventions of human includes internal combustion engine (I C Engine) as top of the list. The recent emphasis on fuel economy, pollution control and other automobile fields like low friction body profile has also stimulated theoretical searches for an automobile. Studies have found no alternative type that promises to have significant advantages in fuel economy or pollution control than conventional I C Engines. But from these studies, it appears that the conventional types of spark-ignition and Diesel engines will remain in their present predominant position in land and sea transportation and for industrial and portable power for the foreseeable future. And so, here is an approach to have combined design aspects for all basic I C Engine components in one paper. Design aspects includes components like, piston, piston rings, cylinder, cylinder head, connecting rod, crank and crank shaft, cam and cam shaft along with valve and valve gear mechanism. A paper can be the base for future detailed designing work of I C Engine along with stress analysis and simulation.

Keywords*- Internal combustion engine, piston, piston ring, connecting rod, crank shaft, valve gear mechanism.*

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I. INTRODUCTION

World needs transportation to fulfill most of the basic need either in the form of one or the other. Internal combustion engine comes under the radar from this point. It needs different kinds of fuels to work. From past to recent future, world is working in the field of I C Engine, its systems and for its betterment. The recent emphasis is on fuel economy and pollution control. Studies have found no alternative type that promises to have significant advantages in fuel economy or pollution control, and, above all, none that has nearly the all-around simplicity, safety, and adaptability of present engines. It appears that the conventional types of spark-ignition and Diesel engines will remain in their present predominant position in land and sea transportation and for industrial and portable power for the foreseeable future [5]. Hence, here is an approach to present basic design of I C Engine.

This paper can provide foundation for next detail design stages, where stress analysis based on thermal fundamentals and simulations are possible.

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II. DIFFERENT PARTS OF INTERNAL COMBUSTION ENGINE

Principal parts of I C Engine are:

i. Cylinder and cylinder head
ii. Piston
iii. Connecting rod
iv. Valve mechanism
v. Crankshaft and flywheel

Cylinder of I C Engine contains the working fluid and guides the piston. Piston receives the gas load and transmits the load to the crankshaft through the connecting rod. The connecting rod transforms the reciprocating motion of piston into the rotary motion of crankshaft and the power is supplied to the necessary machine from the shaft.

The main components of the reciprocating internal combustion engine are shown in Figure 1. Engine parts are made of various materials and perform certain functions, some of which will be explained: cylinder block (g) it is integral with crank case (m), both are made of cast iron. The piston (e) reciprocates inside the cylinder, which include the combustion chamber. Spark plug is for combustion to take place inside the cylinder block above piston. In compression ignition engine, spark plug is avoided as due to high compression ratio, the combustion inside the engine takes place.

The piston is connected to the connecting rod (h) by piston pin (f). This end of the connecting rod is known as small end. The other end of the connecting rod called the big end is connected to the crank arm by crank pin (l).

Camshaft (u) makes the cam (t) to rotate and move up and down the valve rod through the tappet (r). Mainly each cylinder has two valves; one is admission or suction valve and the other is exhaust valve [6].

The ignition system consists of a battery, an ignition coil, a distributor with cam and breaker points, and spark plug for each cylinder. In diesel engines there is an injection system instead of ignition system.

Here we are not going in to much detail of two and four stroke engines as that is common fundamentals related to the I C Engines.

III. DESIGN ASPECTS OF INTERNAL COMBUSTION ENGINE

A. Design Aspects of Cylinder[1,3,5,6,7]

Cylinders are made up of grain cast iron as it is wear resistant and cheaper. For small size engines, the cylinder, water jacket and frame are made as single piece and liners are not used. But for large size high speed engines, they are used which can be replaced after wear and tear. Liners are made up of good grade of Gray CI, Ni, Chromium CI or NI-Cr Cast steel. Dry liners are heat treated after machining for hardness of RC 50 to 55 or above.

There are two types of liners:

1. Wet liners
2. Dry liners.

Wet liners have water in direct contact and are used in engine bore size 130 mm plus. Wet liners reduces the foundry

 ISSN: 2321-8169
3843 – 3851

problem of casting the cylinder and cylinder is not subjected to thermal stress due to water in it, but it is difficult to replace wet liner from the cylinder as chances of water leakage in the combustion space and crank case.

Dry liners are fitted in the cylinder wall and can be easily replaced as there is no chance of water leakage from jacket. The upper part of liner is cooled better with the use of dry liners, the casting of cylinder becomes complicated and heat flow through the composite wall of liner and cylinder is reduced [1].

Thickness of cylinder wall: Assume bore for cylinder.

$$t = \frac{pD}{2\sigma_t} + k$$

Where, p is maximum explosion pressure in N/mm^2, D is cylinder bore in mm, σt is allowable hoop stress in N/mm^2 and its approximate value is between 35 to 100 N/mm^2, K is Re-boring factor.

TABLE I
SELECTION OF FACTOR K

Bore (mm)	75	100	150	200	250	300	350	400 above
K (mm)	1.5	2.5	4	6	7.5	9.5	10.5	12.5

Cylinder bore and length:
Power developed by the engine,

$$P = \frac{p_m LAN}{60 \times 1000} kw - 2\, stroke cycle$$

$$P = \frac{p_m LAN/2}{60 \times 1000} kw - 4\, stroke cycle$$

Where, Pm is mean effective pressure, N is speed of engine in rpm, Assuming L/D, we can find the capacity of engine.
Length of cylinder = L+ clearance at ends
10 to 15% clearance at both ends is considered generally. Hence, Length of cylinder= L+ 0.15 L

Thickness of flange at lower end:
Studs and nuts are used to attach upper half of crank case with cylinder. Here thickness of flange at lower end = 1.25 t or 1.25 d to 1.5 d, where d is stud diameter. The distance between the stud centre and end of the flange is generally taken as 1.5d.

Size of studs or bolts for connecting the flange with the upper half of crank case:

$$Gas load = n \times \frac{\pi}{4} d_c^2 \times \sigma_t$$

Where, n is number of studs or bolts, dc is core diameter of bolt (bolts are made up of Ni steel), σt is allowable stress for stud or bolt and value is between 60 to 100 MPa for nickel steel.
Minimum size of bolt should not be less than 16 mm for gasket joint. The pitch of stud or bolt is taken 20 √(d) to 25 √(d) mm. Hence, outer diameter of cyl. Flange = pcd of bolt + 3d
The no. of bolts or studs can be taken as 8, 12, 16 in numbers with equations:

$$\left(\frac{D}{100} + 4\right) to \left(\frac{D}{50} + 4\right), where\, D\, in\, mm$$

Cylinder head:
Separate cylinder head is cased generally and connected with cylinder by the no. of bolts. The cylinder head accommodates valve mechanism, auto miser or spark plug depending on engine type. The cylinder head can be considered as circular plate fixed at the edges. Considering it as flat plate, the thickness equation is as below:

$$t = D\sqrt{C \times p_{\max} / \sigma_t}$$

Where, C is constant and its value is 0.1, σt is allowable stress for CI head and its value is between 35 to 60 N/mm^2. Maximum amount of heat is transferred through the cylinder head, so its thickness is taken sufficiently more.

B. Design Aspects of Piston and Piston Rings [1,3,5,6,7].

Function of piston in the internal combustion engine is to receive load of the gas and transmit it to the crank shaft through the connecting rod. It also disperses the heat of combustion from combustion chamber to the cylinder walls.

The main components of trunk piston are piston crown, skirt, gudgeon pin and piston rings. Important design criteria includes, it should be high heat resistant and strong, weight should be minimum with sufficient bearing area to prevent wear, should effectively prevent the leakage of gases through the annular space around, dissipate heat as quickly as possible after combustion, hardness of material should be maintained at high elevated temperatures with noise less operation.

In the name of materials, pistons are generally made up of cast iron, cast aluminium, cast steel or forged aluminium. They are different in rubbing friction, thermal expansion and contraction, density, strength and toughness along with other criteria [1].

Thickness of piston head (th):

$$t_h = \sqrt{\frac{3}{16} \frac{pD^2}{\sigma_t}}$$

The piston head can be considered circular plate fixed on the periphery with uniform pressure distribution while loading. Thickness equation is given where, P is maximum explosion pressure in N/mm^2, D is cylinder bore in mm, σt is allowable bending stress in N/mm^2. This thickness of piston is to be checked with heat dissipation. Hence, Heat absorbed by piston is equal to heat flow through the head from centre to the cylinder walls.

$$\pi r^2 q = \frac{dT}{dr}(2\pi r \times t_h)$$

Where, r is piston radius in mm, q is heat flow from gases in W/m^2, dT/dr is temperature gradient, k= thermal conductivity in W/m^{o}C Hence, integrating the above equation between two temperature limits Tc and Te, Tc= temperature at centre, Te= temperature at edge,

$$(T_c - T_e) = \frac{q}{2kt_h} \times \frac{r^2}{2} = \frac{D^2 q}{16kt_h}$$

$$q = \frac{H}{\frac{\pi}{4}D^2}$$

$$t_h = \frac{H}{12.56k(T_c - T_h)}$$

Where, H is amount of heat dissipated through the piston head =W× CV of fuel× Brake power× C, W is specific fuel consumption in kg/kW hr, CV is calorific value of fuel and is between 36000 to 44000 kJ/kg for diesel and between 45000 kJ/kg for petrol, C is fraction of the part of heat transmitted by piston and its value is between 0.05 to 0.06 which is 5 to 6 % of heat, K is thermal conductivity of piston material and its value is between 46 W/m^2 °K for CI and and its value is between 160 to 175 W/m^2 °K for aluminium alloy. In case of unavailability of data, we can assume, Tc-Te = 200°K for CI and 55°K for Aluminum Alloy.

Break power can be found with the flowing equation in watts:

$$BP = \frac{P_m LAN}{2\times 60} \; for\ 4\,stroke\ \sin gle\ cylinder\ engine$$

Where, P_m is power in N/m^2, L is length of piston in m, A is area in m^2

For 4 stroke petrol engine = th = 0.12D to 0.14D -CI piston

For 4 stroke diesel engine = th = 0.12D to 0.16D -Al.Alloy

Piston Ring Design aspects:

Generally 3 to 4 compression rings and one oil ring are used which are made up of CI having permissible strength of 84 N/mm2. Equation for radial thickness of piston ring:

$$t_r = \sqrt{\frac{3P_w}{\sigma_t}}$$

Where, P_w is wall pressure = 0.02 to 0.04 N/mm^2, t_r = 0.04 D to 0.045D, b is 0.6 t_r to t_r, Land between rings= 0.75t_r

Piston Barrel Design aspects:

L is Length of barrel = D to 1.5D

For ideal heat flow, thickness of piston barrel should be equal to the thickness of piston head tapering down to bottom.

tb = thickness of piston barrel = 0.03D+ b+ 5 mm

Thickness of barrel at open end = 0.25 tb to 0.35 tb

Piston skirt design aspects:

Piston skirt it the portion of piston barrel below the ring section upto the open end. Its length should be such that the side thrust pressure does not exceed 0.2 to 0.3 N/mm^2.

Length of piston skirt = ls = 0.6 to 0.8D

Piston skirt is to be checked for side bearing pressure:

$$P_{\max} = \frac{R_{\max}}{l_s \times D} where, R_{\max}\ is\ side\ thrust\ of\ connecting\ rod\ on\ skirt = \frac{10}{100}\times gas\ load = P_{\max}\tan\theta$$

Where, θ is obliquity of connecting rod= 2° to 5°

From above equation:

$$P_{\max} = Gas\ load = \frac{\pi}{4}D^2 p$$

Where, p is explosion pressure.

Design aspect of ribs:

No. of ribs to avoid the distortion of piston is provided. This distributes the gas load as well. This is provided below the ring section and extending up to piston pin boss and extending around the skirt. The ribs may extend across the head in some pistons. Generally 4 to 6 ribs are provided radially in the head.

Thickness of ribs = th/3 to th/2

Design aspect of piston pin or gudgeon pin:

Usually full floating or semi floating pins are used to connect piston and small end of connecting rod. Fixed piston pins are never used as complete swinging motion of connecting rod will be confined to the small end bearing only and causing uniform wear. Piston pins is designed for maximum gas load or inertia force whichever is higher.

Length of piston pin = 0.9 D

Length of piston pin in the connecting rod bush is lp= 0.45 D

Diameter of piston pin = dp

Allowable bearing pressure = pb = 15 to 30 N/mm^2

Pmax=lp×dp×pb

To reduce the weight of piston pin, it is made of hollow section where diameter ratio of inner to outer is maintained as 0.6. This piston pin is checked for bending assuming the gas load to be uniformly distributed.

$$M = \frac{P_{\max} D}{8} = \frac{\pi}{32}\left[\frac{d_p^2 - d_i^2}{d_p}\right]\times \sigma_t$$

The allowable bending stress for hardened steel is 84 N/mm2 and 140 N/mm2 for heat treated alloy steel. Piston pin is subjected to double shear stress

at the boss, so it is to be checked for shear stress, and allowable limit for the same is 50 MPa.

Design aspect of piston boss:

Inner diameter of piston boss = dp

Outer diameter of piston boss = 2dp

Length of boss = 0.2D

Design aspect of piston clearance:

Clearance between the liner and piston is provided for the allowance for thermal expansion. To ensure the necessary clearance in hot engines, the piston crown and skirt are made smaller by a value δD.

δD = 0.005 to 0.007 D at crown in CI piston

δD = 0.001 to 0.0013 D at skirt in CI piston

δD = 0.006 to 0.01 D at crown in Al piston

δD = 0.0018 to 0.0025 D at skirt in Al piston

The clearance between the ring and groove wall is also important. And it is in compression ring is 0.7 to 0.95 mm and for oil scrapper ring it is 0.9 to 1 mm. This all clearance is to be checked under the hot state and hot condition which is not described here. Only important portion of inertia is described below.

Checking of piston wall thickness for inertia force:

$$Inertia\ force = F_i = mr\omega^2\left[1+\frac{1}{n}\right]$$

Where, m = mass of reciprocating parts, r is crank radius, n is length of connecting rod/ crank radius, ω= angular speed of rotation.

Always higher speed of rotation is considered. The weakest section of piston is the section across the oil holes.

Hence, A is c/s area of the weakest section

$$A=\frac{\pi}{4}\left[D_g^2-D_i^2\right]-No.of\ oil\ holes\times\left[\frac{D_g-D_i}{2}\right]d_0$$

Dg is diameter of piston at gudgeon pin= Di+2th

Di = dia. Of inner side of piston = D-2[th + tr + Δt]

Δt = radial clearance between the piston ring and groove.

do = diameter of oil hole and usually 1.2 mm is taken.

Usually 6 nos. of oil holes are provided in the oil scrapper ring.

Rupture stress= σt = Fi/A and permissible value is to N/mm^2

$$\frac{\sigma_c A}{1+a(\frac{l}{k_{xx}})^2}=\frac{\sigma_c A}{1+a(\frac{l}{2k_{yy}})^2} \quad where\ K_{xx}{}^2=4K_{yy}{}^2$$

$$I_{xx}=4I_{yy} \quad as\ I=AK^2$$

C. Design Aspects of Connecting Rod Assembly[1,3,5,6,7].

Connecting rod is a link between piston and crank shaft in the internal combustion engine. Generally they are drop forged from carbon steel or alloy steel. Carbon steel has ultimate tensile strength of 550 to 950 MPa. NI-Cr-steel is also used as one of the material. Cross section of rod may be circular, rectangular, I-section or H-section. Circular section is used for low speed and I-section is used for high speed engines. Length of connecting rod may be 4 to 5 times the crank radius. If higher ratios are used then angularity is increased which reduces wear due to side thrust. Lower ratio increased the angularity and hence the wear due to side thrust. Most engine have conventional two piece connecting rods, where bearing cap is cut off from forged rod and bolted in place for final machining. Generally I section is used in the connecting rod and hence the analysis is taken here for the same as sample.

A connecting rod is a machine member which is subjected to alternating direct compressive and tensile forces. Since the compressive forces are much higher than the tensile forces, therefore the cross-section of the connecting rod is designed as a strut and the Rankine's formula is used. A

connecting rod subjected to an axial load W may buckle with X-axis as neutral axis (i.e. in the plane of motion of the connecting rod) or Y-axis as neutral axis (i.e. in the plane perpendicular to the plane of motion). The connecting rod is considered like both ends hinged for buckling about X-axis and both ends fixed for buckling about Y-axis. A connecting rod should be equally strong in buckling about either axes.

Let, A is Cross-sectional area of the connecting rod, l is Length of the connecting rod, σc is Compressive yield stress, Wcr is Crippling or buckling load, Ixx and Iyy is Moment of inertia of the section about X-axis and Y-axis respectively, and kxx and kyy is Radius of gyration of the section about X-axis and Y- axis respectively.

According to Rankine's formula,

$$W_{cr\ aboutX-axis}=\frac{\sigma_c A}{1+a(\frac{L}{k_{xx}})^2}=\frac{\sigma_c A}{1+a(\frac{l}{k_{xx}})^2}\ asbothhingededge, l = L$$

$$W_{cr\ aboutY-axis}=\frac{\sigma_c A}{1+a(\frac{L}{k_{yy}})^2}=\frac{\sigma_c A}{1+a(\frac{l}{2k_{yy}})^2}\ asbothhingededge, L = \frac{l}{2}$$

In order to have a connecting rod equally strong in buckling about both the axes, the buckling loads must be equal, i.e.

$$\frac{\sigma_c A}{1+a(\frac{L}{k_{yy}})^2}=\frac{\sigma_c A}{1+a(\frac{l}{2k_{yy}})^2} \quad or \quad (\frac{l}{k_{xx}})^2=(\frac{l}{2k_{yy}})^2$$

$$k_{xx}{}^2=4k_{yy}{}^2 \quad or \quad I_{xx}=4I_{yy} \qquad (as, I=A\times k^2)$$

This shows that the connecting rod is four times strong in buckling about Y-axis than about X-axis. If Ixx > 4 Iyy, then buckling will occur about Y-axis and if Ixx < 4 Iyy, buckling will occur about X-axis. In actual practice, Ixx is kept slightly less than 4 Iyy. It is usually taken between 3 and 3.5 and the connecting rod is designed for buckling about X-axis. The design will always be satisfactory for buckling about Y-axis. The most suitable section for the connecting rod is I-section with the proportions as shown in Figure below.

Area of the section=2(4t×t)+3t×t=11t^2

Moment of inertia about X-axis,

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

Moment of inertia about Y-axis,

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

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

Since the value of I_{xx} and I_{yy} lies between 3 and 3.5, therefore I-section chosen is quite satisfactory.

The I-section of the connecting rod is used due to its lightness and to keep the inertia forces as low as possible. It can also withstand high gas pressure. Sometimes a connecting rod may have rectangular section. For slow speed engines, circular sections may be used. Since connecting rod is manufactured by forging, therefore the sharp corners of I-section are rounded off. Same way the force analysis on connecting rod can be presented where two forces are considered. One is force due to gas or steam pressure and inertia of reciprocating parts. Second is inertia bending forces. We can derive the expressions for the forces acting on different types of engine working under

 ISSN: 2321-8169
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different conditions. But here it is not taken as a part of this paper.

D. Design Aspects of Crank Shaft Assembly [1,3,5,6,7].

Crank shaft transforms reciprocating motion of piston in the rotary motion which consists of crank pin, webs and shaft. It may be of overhung crank type or centre crank type. May be of single or double throw type. The crank shaft is subjected to both twisting moment and bending moment hence it must have sufficient strength to withstand the forces. It is generally made up of 35 C steel or 40 C steel having ultimate strength of 525 to 600 MPa. They are usually forged.

A part of valve, valve gear mechanism which includes rocker arm design as well as timing, pushrod and cam design is omitted here as they are basically outer components of engine and work in connection to engine moment.

Design procedure for overhang crank shaft:

$$Load\ on\ piston = \frac{\pi}{4} D^2 P_{\max}$$

$$where,\ D\ is\ cylinder\ dia.\ \&\ P_{\max}\ is\ \max.\exp losion\ pressure$$

Thrust in the connecting rod Q=P/cosǾ
Where, Ǿ in Obliguity of connecting rod=sin-1[sinϴ/n],
ϴ is position of crank for maximum twisting moment from IDC and n = l/r
Crank pin design: l/dp = sqrt(0.2σt/P_b)
Where, σt is allowable stress for crank
Pb is allowable pressure for crank pin, Then, P= l×d_p×P_b

Checking the crank pin for bending where assumption is uniformly distributed load for the crank pin,

$$M = \frac{pl}{2} = \frac{\pi}{32} d_p^2 \sigma_t$$

Here, stress value can be found and allowable limit for the material is 70 to 85MPa.

Design of crank and crank arm:
Thickness of crank web is t = 0.6 to 0.75d_p.
Width of crank web is b.
The crank arm is subjected to a compressive load and a bending load in dead centre position.

Direct compressive stress = p/bt
Bending stress = pe/z
Where, e = eccentricity of load = (l/2 + t/2) and Z= 1/6bt^2

Here allowable compressive stress is taken as 80MPa. Checking of crank arm cross section for maximum twisting moment:
In figure, F_t = tangential component = Qsin(ϴ+Ǿ)
F_r = radial component = Qcos(ϴ+Ǿ)
Bending Moment due to F_t = M_t = F_t×R [R=crank radius]
= 1/6×tb^2×σbt

Stress value can be found from the equation.
Bending Moment due to Fr = M_r = F_r×e = 1/6×bt^2×σbr
Direct compressive stress due to radial component F_r
$\sigma_{cr} = F_r/b_t$ and σ_c = total compressive stress = $\sigma_{bt}+\sigma_{br}+\sigma_{cr}$
Usually limit of this total compressive stress is 80MPa.

Twisting moment due to F_t = T = F_t×e = Z_p×τ
Where, Z_p = polar modulus of section = 0.269 b_t^2 (rectangular c/s) so, stress value can be found here.

$$Maximum\ principal\ stress = \sigma = \frac{\sigma_c}{2} + \sqrt{\left(\frac{\sigma_c}{2}\right)^2 + \tau^2}$$

E. Design Aspects of Valve and Valve Gear Mechanism [1,3,5,6,7].

Valve gear mechanism operates suction and exhausts valves and hence maintains air to fuel intake at desired time and hence compression and expansion strokes takes place at desired period of time. Also the remaining burnt product is exhausted before the next cycle is started. Component of valve gear mechanism for four stroke cycle engine are inlet and exhaust valves, rocker arms or levers, tappets or push rods, cams and cam shaft along with the transmission device from crank shaft to cam shaft.

Design procedure of valve port and valve:
Size of valve ports depends upon the cylinder bore and cylinder centre distance. Large valve is provided for large diameter cylinder. The size of valve port can be found as under:

$$A_{piston} \times V_{piston} = A_{port} \times V_{gas} \quad where,\ A\ indicates\ area\ and\ V\ indicates\ velocity\ in\ m/s$$

Piston velocity=2LN/60 m/s.
To maintain the flow, the pressure difference should not be high, so the gas velocity should not be high. Four automobile, it should be 50 to 75 m/s and for stationary engine, it should be 40 to 50 m/s. Here from the equation, area of port can be found and from that diameter of port can be found out in mm. For higher engine output, inlet port is made about 30 to 40 mm larger than exhaust port. Valve stem diameter is indicated by ds and valve seat is grinded with the angle of 45º normally. Generally poppet type valves are used in I c engine. Main part of valve is stem, head and face. For better performance of engine, opening and closing of valves should be done at perfect timing. Also valve should be tight on their seats to prevent any leakage while in close condition. Along with this, the weight of valve should be as light as possible to have flexibility in operation and reduce the working forces.
For flat headed valve, valve lift l = $d_{p0}/4$
For conical valves, valve lift l = $d_{p0}/4\cos\alpha$
Where usually α = 45 degree for exhaust and 30 degree for inlet valve.
Diameter of valve head d_v = d_{p0} to 1.16 d_{p0}

Thickness of valve head can be obtained by considering it as flat plate subjected to maximum uniform distributed gas load.

$$t = k\, d_{po} \sqrt{\frac{P_{\max}}{\sigma_t}}$$

Here, σ_t = 28MPa, k = 0.54 and if σ_t = 56MPa, k = 0.42
Generally value of k is taken as 0.5 if nothing is given. The diameter of valve stem is taken as, $d_s = d_{p0}/8$+6.5 to 11mm

Design of rocker arm:

 ISSN: 2321-8169
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Rocker arm transmits the motion from crankshaft to the valves. The force required to operate the inlet valve is comparatively less than that required to operate the exhaust valve. In practice both the rocker arms operating inlet valve and exhaust valve are made of same size for uniformity.
Maximum load on the valve during acceleration comprises of gas load, initial spring force to hold on its seat against suction stroke and valve inertia.

Gas load=Area of valve(which is $\pi/4d_v^2$)× exhaust pressure (P)
Exhaust pressure is generally 0.35 to 0.4 MPa.
Initial spring force = Valve area× Vacuum pressure
Vacuum pressure varies from 0.02 to 0.03 MPa for diesel eng.
Vacuum pressure varies from 0.05 to 0.07 MPa for petrol eng.
Inertia of valve = Mass of valve× Acceleration.
The design mass of valve is referred to the area of passage through the throat of the valve and reduces to its axis or it is the equivalent mass of valve gear referred to the valve axis. Empirically it can be found as under:
Mass of valve = m = 230× (port area in m^2)= 230×($\pi/4\ d_{p0}^2$) in kgs. Diameter here should be in meter.

The acceleration of valve is found as under:
Valve total opening angle =180+angle of EVO before BDC+ angle of EVC after TDC
= 180+ α+β
If the cam moves with uniform and equal acceleration and retardation, then,

$$Acceleration = a = \frac{4\omega^2 s}{\theta_0^2}$$

If it moves with simple harmonic motion, then,

$$a = \frac{\pi^2\omega^2}{\theta_0^2}\frac{S}{2}$$

Where, ω=2π(N/2)/60 rad/s and Cam shaft speed is half that of crank shaft speed and S= lift of valve.

Design aspects of rocker arm:
The rocker arm may be straight or angular with an included angle of 135 to 160 degree.
Arm length, X=5 times valve lift and Y=0.9 to 1X.
Allowable stress for caste steel is 50 to 60MPa and for forged steel is 60 to 70MPa.
Cross section of rocker arm is rectangular or I section generally.
For rectangular section: h = 3b
For I-section: Flanges=2.5t×t, Depth= 6t, Web= 4t×t
Now maximum bending moment on the arm, M=P×X=σbZ
Where, P is total force acting at the valve end and X is leverage of arm. Z=1/6bh2 for rectangular section and I/y for I-section.
Fulcrum pin diameter:
For angular rocker arm, included angle is Θ,

Reaction at fulcrum pin= $R = \sqrt{P^2 + Q^2 - 2PQ\cos\theta}$

Then, R = ld_pP_b
The allowable pressure is 10 to 18MPa and the ratio l/dp is usually taken as 1.25 to 1.5. So through this, pin diameter can be determined. Hub of rocker arm is fitted with brass bush.

Hence inner diameter of hub= $d_i = d_p+2$ × bush thickness.
Thickness of brass bush usually taken as 2 to 3 mm and outer diameter of hub is 2di. Checking of hub under bending with following equation and bending stress can be found out.

$$M = \sigma_b Z = \sigma_b \frac{1/12[BH^3 - Bh^3]}{H/2}$$

Here, H= outer diameter, h= inside diameter and B= width of the hub.

Push rod design aspects:
It is considered to be strut or column. It is usually hollow, with inner diameter 0.6 times outer diameter. Euler's equation or rankine equation is used for l/k<80 for CI and l/k<100 for steel.

$$P_g = \frac{\sigma_c A}{1 + a[l/k]^2} \qquad A = \frac{\pi}{4}[d_0^2 - d_i^2]$$

$$k^2 = I/A = \frac{\pi}{64}[d_0^2 - d_i^2] / \frac{\pi}{4}[d_0^2 - d_i^2]$$

Where, a=I/7500 for steel, l is length of push rod and σ_c is allowable stress for push rod material.
Push rod is made up of bright drawn mild steel with 0.4% carbon content. Allowable stress is 65 to 75MPa.
Design of valve spring is done in usual manner as compression spring. Function of spring is to provide spring force for the closure of the valve and to maintain contact of cam with the follower all the times. In the excessive surging of spring, as in high speed engine, the follower may leave the contact of cam.

Whal factor = k = (4c-1/4c-4)+(0.615/c), Where, C= spring index = D/d_w
Maximum load on the valve is the load when valve is opened.
Pmax = D/2 = πdw3τ/16k
When valve is closed, load on spring is minimum. The allowable shear stress for carbon steel spring is 280 to 320MPa.

$$No.of\ active\ coil = n = \frac{\delta G d_w}{8pc^3}$$

Where, δ is valve lift, G is modulus of rigidity and P is net load which is subtraction of maximum to minimum load. Total no. of coils is n+2. Free length of spring can be finding in the usual solid length added to deflection and clearance.

Design aspects of cam:
Cam is integral part of the cam shaft. In this case following are the calculation,
Diameter of cam shaft = 0.16D+12.5mm [D=bore diameter]
If the cam is keyed on the cam shaft, then
Diameter of cam shaft = 1.175D+20mm
Base circle diameter = cam shaft dia.+3mm for integral cam
= 1.5×shaft dia.+20 mm for keyed cam
Usually roller follower is used with cam.
Diameter of roller = 0.5 to 0.75 × cam shaft diameter
Width of follower = 0.3×roller diameter
Width of cam = 0.09D+6 mm to 0.11D+12 mm
However the width of cam depends upon the load acting on it.

If the cam is well lubricated, then the cam width is determined a sunder.

If cam is face hardened then, it can sustain load of 200N to 250N per mm width.
For plain carbon steel cam, it can sustain a load of 100N to 150N per mm width.
From the given valve lift and given angles of action, the ca can be designed.

CONCLUSIONS

The work is base approach to have design of different I C Engine and its components. No thermal or other considerations are taken in to account while designing. We have designed different components like piston, piston rings, cylinder, cylinder head, connecting rod, crank and crank shaft, cam and cam shaft along with valve and valve gear mechanism. Purpose of the paper is to better understanding of different design parameters and systems in integration to design with certain assumptions as mentioned in different sections. Expected outcome of this work is to have integration between design parameters and systems. Same way any existing or new single cylinder engine can be designed, thermal considerations can be approached and a simulation of the engine can be workout. For safer limits, certain stress limits have been assumed which can be change as per the material selection and data can be drawn from design data book.

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ISSN: 2321-8169
3843 – 3851

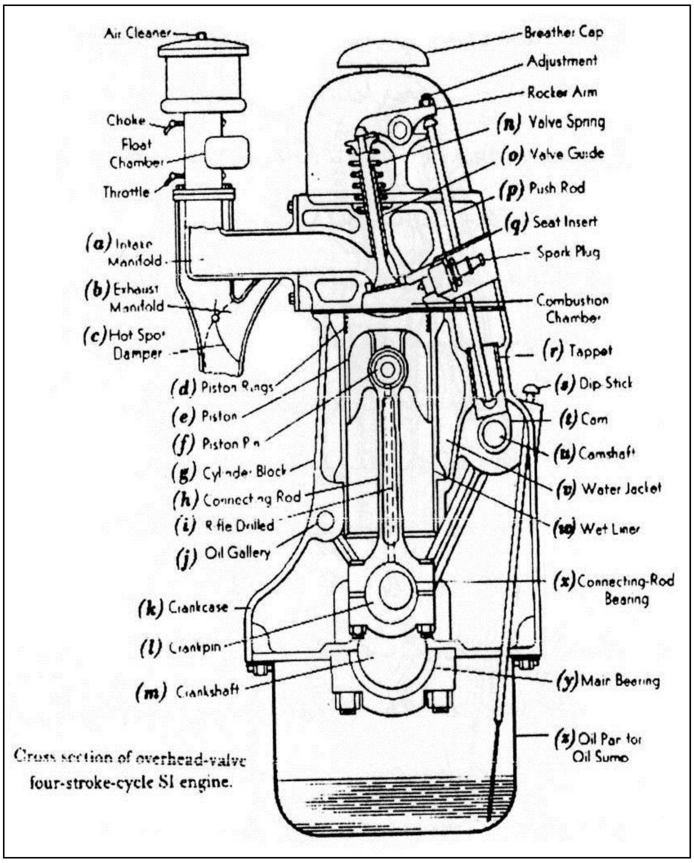
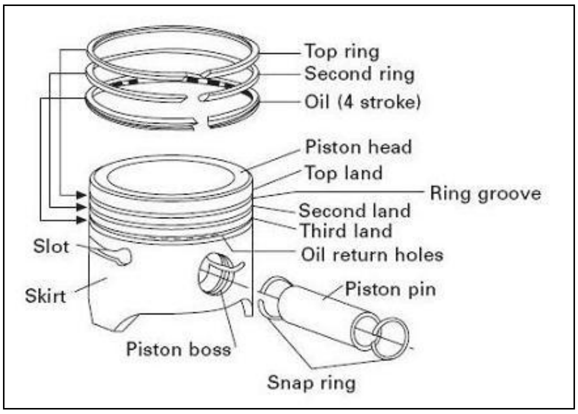


Figure 1: Spark ignition engine parts and details [6].



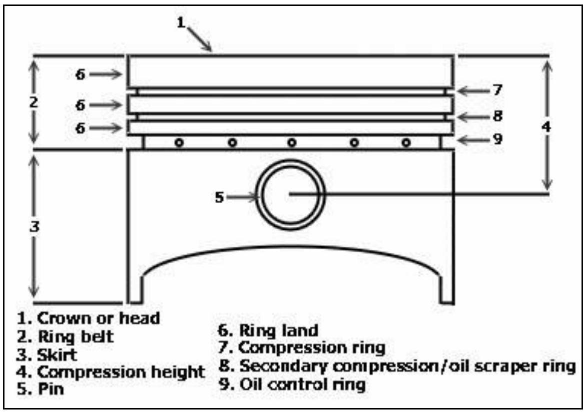


Figure 2: Piston rings skeleton diagram.

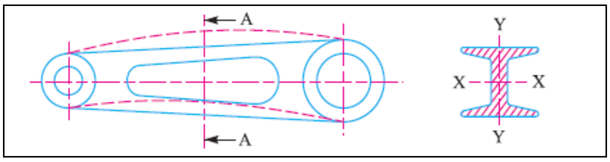


Figure 3: Buckling of Connecting rod (I section)[1]

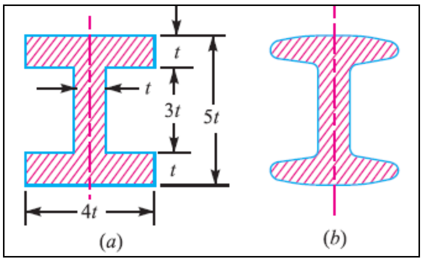


Figure 4: I-section of connecting rod [1].

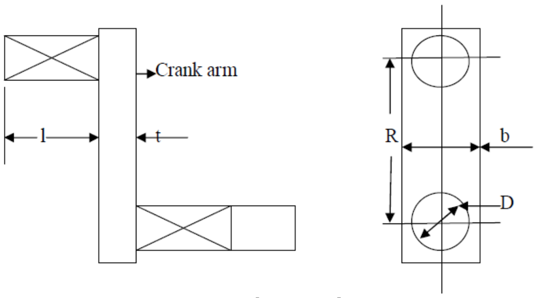


Figure 5: Crank Arm Selection.

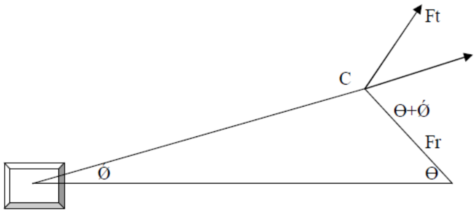


Figure 6: Twisting Moment Diagram for Crank Pin.

ISSN: 2321-8169
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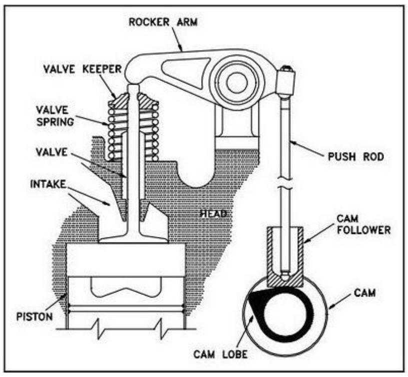

Figure 7: Valve Gear Mechanism of I C Engine [5]