A Review Analysis on the Design of Internal Combustion Engine

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***Abstract*— The internal combustion engine ranks first among human technologies that have been the most successful. Theoretical searches for a vehicle have also been sparked by the current focus on fuel economy, pollution control, and other automotive sectors like low friction body profile. No other type has been identified by studies as offering appreciable benefits over traditional I C engines in terms of pollution control or fuel economy. However, these studies suggest that for the foreseeable future, the traditional diesel and spark-ignition engine types will continue to hold their dominant positions in land and marine transportation, as well as in industrial and portable power. Thus, this is a method for gathering design elements for every fundamental I C Engine part into a single document. Components include the piston, piston rings, cylinder, cylinder head, connecting rod, crank and crank shaft, cam and cam shaft, valve, and valve gear mechanism are all included in the design aspects. A paper can serve as the foundation for further in-depth I C Engine design work, stress analysis, and simulation.**

***Keywords*— Internal combustion engine (ICE), piston, piston ring, connecting rod, crank shaft, valve gear mechanism.**

I. INTRODUCTION

The majority of essential necessities in the world must be met via transportation, in one way or another. From this point on, internal combustion engines are discussed. For it to function, various fuels are required. The world has been working on improving the ICE and its systems for the past few years. Recently, pollution control and fuel economy have received more attention. Research has yielded no substitute variety that appears to offer noteworthy benefits in terms of fuel efficiency or pollution management, and most importantly, none that approaches the current engines' overall simplicity, safety, and versatility.

ICE needs different kinds of fuels to work. The world has been working on improving the ICE and its systems for the past few years. Recently, pollution control and fuel economy have received more attention. Research has yielded no substitute variety that appears to offer noteworthy benefits in terms of fuel efficiency or pollution management, and most importantly, none that approaches the current engines' overall simplicity, safety, and versatility (Alagumalai, 2014). For the foreseeable future, it seems that the traditional types of diesel and spark-ignition engines will continue to hold their current leading positions in land and marine transportation, as well as in industrial and portable power (Georgiev, 2011; Joseph E. Shigley, 1996; Taylor, 2020a). Hence, here is an approach to present basic design of ICE. This paper can provide foundation for next detail design stages, where stress analysis based on thermal fundamentals and simulations are possible.

II. COMPONENTS OF ICE

Major components of ICE are:

1. Cylinder and cylinder head
2. Piston and Piston ring assembly
3. Connecting rod
4. Crankshaft and flywheel
5. Valve mechanism

The ICE's cylinder drives the piston and holds the working fluid. The piston transfers the load to the crankshaft via the connecting rod after receiving the gas load. The connecting rod converts the piston's reciprocating action into the crankshaft's rotating motion, and the shaft provides the necessary power to the machine.

Major components of the reciprocating ICE are described and represented in Figure 1. Components of ICE are made up of variety of materials and perform certain functions. These functions are, cylinder block (g) integrated with crank case (m), and both are made of cast iron. The piston (e) reciprocates inside the cylinder, which include the combustion chamber. The purpose of the spark plug is to facilitate combustion within the cylinder block, above the piston. Spark plugs are avoided in compression ignition engines because internal combustion occurs in the engine owing to a high compression ratio. The piston is connected to the connecting rod (h) by piston pin (f). This end of the connecting rod is known as small end where as other end of the connecting rod is called the big end and 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]. A battery, an ignition coil, a distributor with cam and breaker points, and a spark plug for each cylinder make up the ignition system. An injection system replaces the ignition system in diesel engines (Khurmi & Gupta, 2005).

III. DESIGN CONSIDERATION OF ICE

Major components of ICE such as cylinder, piston, piston ring, connecting rod, crank shaft etc are included here for the design considerations. Different design considerations based on design fundamentals are represented here based on certain assumptions (DELPRETE ET AL., 2009).

A. Design Aspects of Cylinder: (Joseph E. Shigley, 1996; Khurmi & Gupta, 2005; Nguyen & Duy, 2018; Pulkrabek, 2013; Stone & Ball, 2004; Taylor, 2020a, 2020b; Viswabharathy & Vairamuthu, 2017; Willard, 1997).

Grain cast iron, which is less expensive and wear resistant, is used to make cylinders. Small engine construction eliminates the need for liners by using a single component for the cylinder, water jacket, and frame. However, they are employed in large, high-speed engines and can be changed out due to wear and tear. Good quality Gray CI, Ni, Chromium CI, or NI-Cr Cast steel is used to make liners. After machining, dry liners undergo heat treatment to achieve a hardness of RC 50 to 55 or higher. Liners are of wet or dry type in ICE.

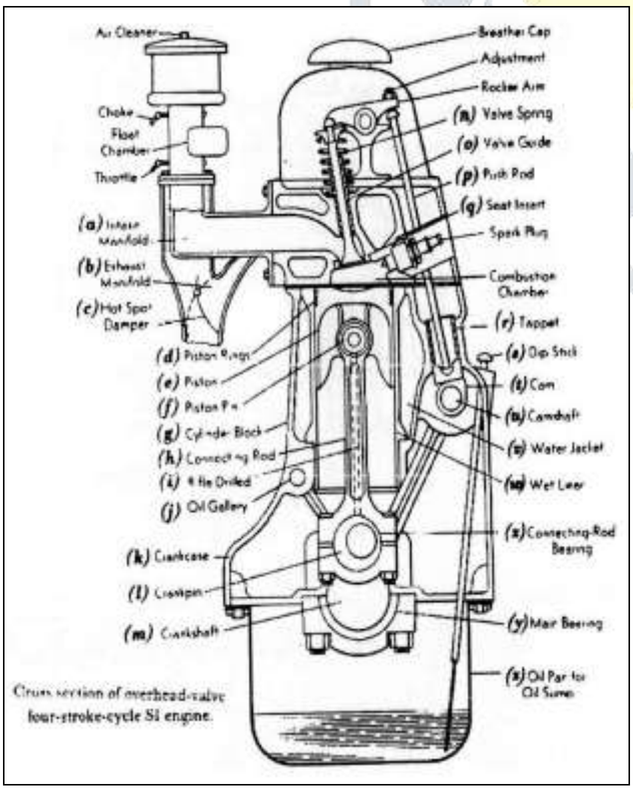

Fig. 1 Component details of Spark Ignition Engine (Pulkrabek, 2013; Willard, 1997).

Wet liners, used in engines with cylinder bore sizes 130mm or larger, reduce foundry issues and thermal stress, but can be challenging to replace due to water leakage. Dry liners, easily replaceable and preventing water leakage from jackets, improve cooling of upper liner, simplify cylinder casting, and reduce heat flow through composite wall (Khurmi & Gupta, 2005).

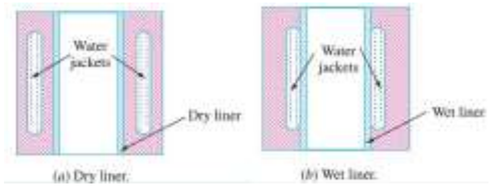

Fig. 2 Liners used in Spark Ignition Engine (Khurmi & Gupta, 2005).

Design equation used for thickness of cylinder wall: Assume bore for cylinder.

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

Where,
p = Maximum explosion pressure in N/mm^{2,}
D = Cylinder bore, mm,
σt = Allowable hoop stress in N/mm^{2,} = 35 to 100 N/mm^{2,}
K = Reboring factor.
(All notations used here are standard notations)

TABLE I
SELECTION OF CONSTANT 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

Engine cylinder bore and length derivation (Georgiev, 2011)**:** 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 = Mean effective pressure,
N = 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

Lower End Flange Thickness:

The upper half of the crank case is connected to the cylinder using studs and nuts.
Here thickness of flange at lower end = 1.25 t or 1.25 d to 1.5 d,
Where,

d= 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 = No. of studs or bolts,
*d*c =core diameter of bolt (bolts are made up of Ni steel),
σt = allowable stress for stud or bolt, = 60 to 100 MPa for nickel steel.

Minimum bolt size 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 Design:

Engine bore sizes more than 130 mm are fitted with wet liners, which are in direct contact with water. Although wet liners prevent thermal stress from water in the cylinder and lessen foundry problems when casting the cylinder, replacing the wet liner from the cylinder is challenging because there is a risk of water leaking into the crank case and combustion chamber.Considering it as flat plate, the thickness equation is as below:

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

Where,
C = constant = 0.1,
σt = allowable stress for CI head = 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:* (Joseph E. Shigley, 1996; Khurmi & Gupta, 2005; Pulkrabek, 2013; Stone & Ball, 2004; Taylor, 2020a, 2020b; Willard, 1997).

The piston's job in an internal combustion engine is to take up gas load and transfer it to the crank shaft via the connecting rod. Additionally, it transfers the heat of combustion from the combustion chamber to the walls of the cylinder.

The piston crown, skirt, gudgeon pin, and piston rings are the essential parts of a trunk piston. Key design requirements are that it should be robust and highly heat resistant, have a low weight and enough bearing area to avoid wear, efficiently stop gas leaks through the surrounding annular space, dissipate heat as soon as possible after combustion, and have a hardness of material that can withstand high, elevated temperatures while operating silently.

When it comes to materials, pistons are typically composed of forged aluminum, cast iron, cast steel, or cast aluminum. In addition to other characteristics, they differ in rubbing friction, density, strength, toughness, and thermal expansion and contraction (Heywood, 2018; Khurmi & Gupta, 2005; Stone & Ball, 2004).

Design of Thickness of Piston Head (th):

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

When loading, the piston head can be thought of as a circular plate that is positioned on the periphery and evenly distributes pressure. Thickness equation is given where,
P = maximum explosion pressure in N/mm^{2,}
D = cylinder bore in mm,
σt= allowable bending stress in N/mm^2

Checking of piston thickness with heat dissipation which is as below:
Heat absorbed by piston = 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 = piston radius in mm,
q= heat flow from gases in W/m^{2,}
dT/dr= temperature gradient,
k= thermal conductivity in W/m°C Hence,

Integrating the above equation between two temperature limits Tc and Te,
Tc= temperature at centre,
Te= temperature at edge

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

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

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

Where,
H= amount of heat dissipated through the piston head =W× CV of fuel× Brake power× C,
W= specific fuel consumption in kg/kW hr,
CV= calorific value of fuel = 36000 to 44000 kJ/kg for diesel and = 45000 kJ/kg for petrol,

C = fraction of the part of heat transmitted by piston =0.05 to 0.06 which is 5 to 6 % of heat,
K = thermal conductivity of piston material = 46 W/m^2 ^{o}K for CI and = 160 to 175 W/m^2 ^{o}K for aluminium alloy,
In case of unavailability of data, we can assume,
Tc-Te = 200^{o}K for CI and 55^{o}K for Al alloy.

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

$$BP = \frac{P_m LAN}{2\times 60} \quad for\ 4\ stroke\ single\ cylinder\ engine$$

Where,
Pm = power in N/m^2,
L=Length of piston in m,
A= 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]

Design of Piston Ring (Viswabharathy & Vairamuthu, 2017)**:**

Generally 3 to 4 compression rings and one oil ring are used which are made up of CI having permissible strength of 84 N/mm^2.
Equation for radial thickness of piston ring:

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

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

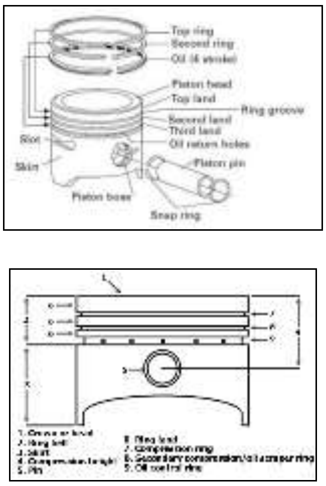

Fig. 3 Piston rings skeleton diagram.

Design of Piston Barrel:

L = Length of barrel = D to 1.5D
The thickness of the piston barrel should match the thickness of the piston head tapering down to the bottom for the best heat flow.

tb = thickness of piston barrel = 0.03D+ b+ 5 mm
Thickness of barrel at open end = 0.25 tb to 0.35 tb

Design of Piston Skirt:

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
Checking of piston skirt is 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 = obliquity of connecting rod= 2^o to 5^o
From above equation:

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

Where,
p = explosion pressure.

Design of Ribs:

The number of ribs is given to prevent piston distortion. Additionally, this distributes the gas load. This is supplied around the skirt, up to the piston pin boss, and below the ring portion. In certain pistons, the ribs may stretch across the head. Typically, the head has four to six radial ribs.

Thickness of ribs = *th/3 to th/2*

Design of Piston Pin or Gudgeon Pin:

Piston and the tiny end of the connecting rod are typically connected using full or semi-floating pins. Because the connecting rod's full swinging action will only affect the tiny end bearing and result in uniform wear, fixed piston pins are never employed. The maximum gas load or the inertia force, whichever is higher, is the specification for piston pins.

Length of piston pin = 0.9 D
Length of piston pin in the connecting rod bush=lp= 0.45 D
Diameter of piston pin = dp

Allowable bearing pressure = pb = 15 to 30 N/mm^2
Pmax = lp×dp×pb

The piston pin is composed of a hollow part with a diameter ratio of 0.6 between the inner and outer to minimize weight. Assuming a uniform distribution of the gas load, the bending of this piston pin is checked.

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

The allowable bending stress for hardened steel is 84 N/mm^2 and 140 N/mm^2 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 of Piston Boss:

Inner diameter of piston boss = dp
Outer diameter of piston boss = 2dp
Length of boss= 0.2D

Design of Piston Clearance:

To account for thermal expansion, there is clearance between the piston and liner. In hot engines, the piston crown and skirt are reduced in size by a value δD in order to guarantee the required clearance.

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 = crank radius,
n= 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= 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 =dia. Of piston at gudgeon pin = Di+2th
Di = dia. Of inner side of piston = D-2[t +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.

$$\frac{\sigma_c \times A}{1 + a\left(\frac{l}{k_{xx}}\right)^2} = \frac{\sigma_c \times A}{1 + a\left(\frac{l}{2\,k_{yy}}\right)^2} \quad \text{or} \quad \left(\frac{l}{k_{xx}}\right)^2 = \left(\frac{l}{2\,k_{yy}}\right)^2$$

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

C. *Design of Connecting Rod Assembly : (Gaspari Cirne de Toledo et al., 2009; Joseph E. Shigley, 1996; Khurmi & Gupta, 2005; Pulkrabek, 2013; Stone & Ball, 2004; Taylor, 2020a, 2020b; Willard, 1997).*

A connecting rod in an internal combustion engine serves as a link between the piston and the crankshaft. They are often made of alloy or carbon steel that has been drop forged. Tensile strength of carbon steel can reach 550 to 950 MPa. Another material that is utilized is NI-Cr-steel. Rods can have circular, rectangular, I-, or H-shaped cross sections. For low speed engines, the circular section is utilized, and for high speed engines, the I-section.

A connecting rod's length can be four or five times the radius of the crank. Higher ratios result in enhanced angularity, which lessens side thrust-related wear. Reduced ratio led to more angularity and wear from side thrust. The majority of engines use standard two-piece connecting rods, in which the bearing cap is removed from the forged rod and fastened to the rod for final machining. Since the I section is typically employed in the connecting rod, the analysis for the same sample is done here.

A connecting rod is a part of a machine that experiences direct compressive and tensile forces that alternate. The connecting rod's cross-section is made to resemble a strut and uses Rankine's formula because the compression forces are significantly greater than the tensile forces. When a connecting rod experiences an axial load of W, it may buckle with the X or Y axes acting as neutral axes, respectively, in the connecting rod's plane of motion or in the plane perpendicular to the motion. When it comes to buckling about the X and Y axes, the connecting rod is thought of as having both hinged ends. A connecting rod ought to be able to buckle around both axes with equal strength.

Let,

A= Cross-sectional area of the connecting rod, l= Length of the connecting rod,

σc = Compressive yield stress,

Wcr= Crippling or buckling load,

Ixx and Iyy = Moment of inertia of the section about X-axis and Y-axis respectively, and

kxx and kyy= Radius of gyration of the section about X-axis and Y- axis respectively.

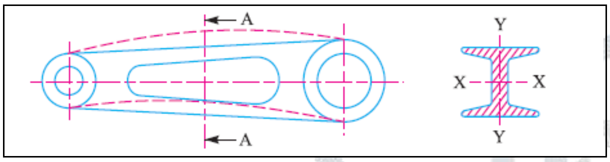

Fig. 4 Buckling of Connecting rod (I section) (Khurmi & Gupta, 2005)

According to Rankine's formula,

$$W_{cr} \text{ about } X\text{-axis} = \frac{\sigma_c \times A}{1 + a\left(\frac{L}{k_{xx}}\right)^2} = \frac{\sigma_c \times A}{1 + a\left(\frac{l}{k_{xx}}\right)^2} \quad \ldots (\because \text{ For both ends hinged, } L = l)$$

$$W_{cr} \text{ about } Y\text{-axis} = \frac{\sigma_c \times A}{1 + a\left(\frac{L}{k_{yy}}\right)^2} = \frac{\sigma_c \times A}{1 + a\left(\frac{l}{2k_{yy}}\right)^2} \quad \ldots \left(\because \text{ For both ends fixed, } L = \frac{l}{2}\right)$$

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

$$\frac{\sigma_c \times A}{1 + a\left(\frac{l}{k_{xx}}\right)^2} = \frac{\sigma_c \times A}{1 + a\left(\frac{l}{2\,k_{yy}}\right)^2} \quad \text{or} \quad \left(\frac{l}{k_{xx}}\right)^2 = \left(\frac{l}{2\,k_{yy}}\right)^2$$

$$\therefore \quad k^2_{xx} = 4\,k^2_{yy} \quad \text{or} \quad I_{xx} = 4\,I_{yy} \quad \ldots (\because I = A \times k^2)$$

This represents connecting rod is four times stronger 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 = $2(4t\times t)+3t\times t=11t^2$

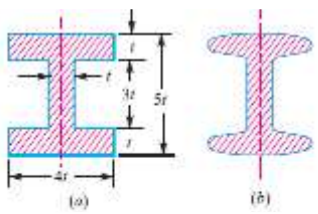

Fig. 5 I-section of connecting rod (Khurmi & Gupta, 2005).

∴ Moment of inertia about X-axis,

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

∴ Moment of inertia about Y-axis,

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

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

Since the value of Ixx and Iyy lies between 3 and 3.5, therefore I-section chosen is quite satisfactory.

Because the connecting rod's I-section is lightweight and minimizes inertia forces, it is utilized. High gas pressure is another thing it can handle. There are situations where a connecting rod has a rectangular form. Circular portions can be employed with slow-moving engines. The connecting rod's sharp edges are softened off since it is forged during production.

In a similar manner, a force analysis that takes into account two forces can be presented for a connecting rod. One is force produced by the inertia of reciprocating parts and the pressure of gas or steam. Inertia's bending forces come in second. The expressions for the forces acting on various engine types operating under various conditions can be derived. However, it is not considered a part of this paper here.

D. *Design of Crank Shaft Assembly: (Joseph E. Shigley, 1996; Khurmi & Gupta, 2005; Pulkrabek, 2013; Stone & Ball, 2004; Taylor, 2020a, 2020b; Willard, 1997).*

The crank pin, webs, and shaft work together to convert the reciprocating action of the piston into rotary motion. It could have a center crank or an overhung crank. either a single throw or a double throw type. The crank shaft needs to be strong enough to resist the forces because it will be subjected to both twisting and bending moments. Typically, 35 C or 40 C steel, with an ultimate strength of 525 to 600 MPa, is used to make it. Most of the time, they are fake. Since timing, pushrod, and cam designs, as well as parts of the valve and valve gear mechanism, which includes the design of the rocker arm, are essentially external engine components that are related to the engine moment, they are not included here.

Design for Overhang Crank Shaft *(Abramchuk et al., 2018; Boysal & Rahnejat, 1997; Sowjanya et al., 2016):*

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

$$where, D\ is\ cylinder\ dia. \&$$

$$P_{max}\ is\ \max.\exp losion\ pressure$$

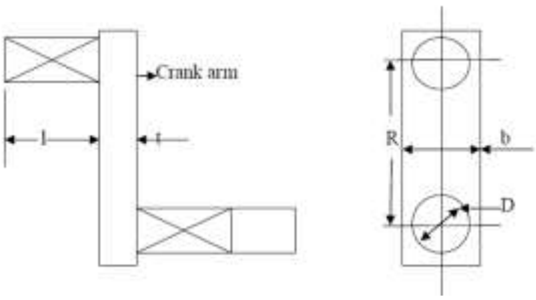

Fig. 6 Crank Arm Selection.

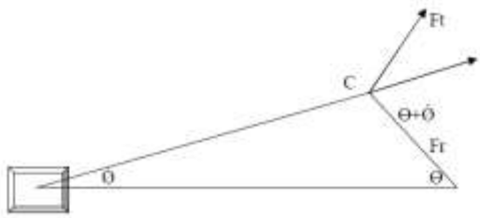

Fig. 7 Twisting Moment Diagram for Crank Pin.

Thrust in the connecting rod Q = P/cosǾ
Where,
Ǿ = Obliguity of connecting rod=$\sin^{-1}$[sinΘ/n],
Θ = position of crank for maximum twisting moment from IDC and n = l/r

Crank Pin Design: l/dp = sqrt(0.2σt/Pb)

Where,
σt = allowable stress for crank
Pb = allowable pressure for crank pin,
Then, P= l×dp×Pb
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.

Crank and Crank Arm Design:
Thickness of crank web= t = 0.6 to 0.75dp
Width of crank web = 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,
Ft = tangential component = Qsin(Θ+Ǿ)
Fr = radial component = Qcos(Θ+Ǿ)
Bending Moment due to Ft = Mt = Ft×R [R=crank radius]
= 1/6×tb^2×σbt

Stress value can be found from the equation.
Bending Moment due to Fr = Mr = Fr×e
= 1/6×bt^2×σbr
Direct compressive stress due to radial component Fr,

σcr = Fr/bt and σc = total compressive stress = σbt+σbr+σcr
Usually limit of this total compressive stress is 80MPa.
Twisting moment due to Ft = T = Ft×e = Zp×τ
Where,
Zp = polar modulus of section = 0.269 bt^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 (Joseph E. Shigley, 1996; Khurmi & Gupta, 2005; Pulkrabek, 2013; Stone & Ball, 2004; Taylor, 2020a, 2020b; Vasilyev & Bakhracheva, 2018; Willard, 1997).*

By controlling the suction and exhaust valves, the valve gear mechanism ensures that fuel is introduced into the engine at the appropriate time, allowing for the proper timing of the compression and expansion strokes. Before the following cycle begins, all of the burnt product is depleted. The inlet and exhaust valves, rocker arms or levers, tappets or push rods, cams and cam shaft, as well as the transmission system from crank shaft to cam shaft, are all parts of the valve gear mechanism for four-stroke engines.

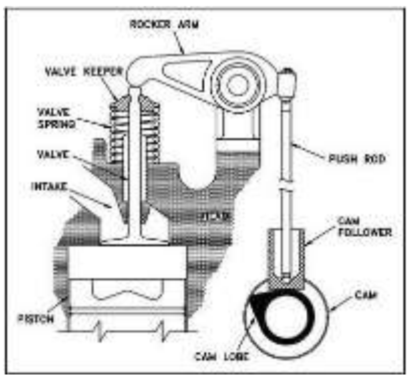

Fig. 8 Valve Gear Mechanism of I C Engine (Joseph E. Shigley, 1996).

Design of Valve Port and Valve:

The cylinder bore and center distance determines the valve port size. For cylinders with a big diameter, a large valve is provided. The size of valve port can be found as under:

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

Piston velocity = 2LN/60 m/s.

A high pressure differential and hence low gas velocity are necessary to sustain the flow. It should be between 40 and 50 m/s for stationary engines and between 50 and 75 m/s for four-wheel drive vehicles. Here, the area of the port can be obtained using the equation, and the diameter of the port may be calculated in millimeters. The entrance port is constructed roughly 30 to 40 mm larger than the exhaust port to accommodate higher engine power. The valve stem diameter is denoted by ds, and the valve seat is typically ground at a 45º angle. In I.C. engines, poppet type valves are typically utilized.
Valve head, face, and stem are its main components. The timing of the opening and closing of the valves is crucial for optimal engine efficiency. In order to stop leaks while in a closed condition, the valves on their seats should also be tight. In addition, the valve's weight must to be as low as feasible to allow for maximum operating flexibility and minimize 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.

$$t = k\, d_{p0} \sqrt{\frac{P_{max}}{\sigma_t}}$$

Here, σt = 28MPa, k = 0.54 and if σt = 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, ds = dp0/8+6.5 to 11mm

The crankshaft moves the rocker arm, which then moves the valves. Comparatively speaking, less force is needed to operate the input valve than the exhaust valve. For uniformity, the rocker arms that operate the exhaust and inlet valves are typically the same size.
The gas load, the initial spring force to maintain the valve's seat against the suction stroke, and the valve's inertia constitute the maximum load on the valve during acceleration.

Gas load =Area of valve (which is $\pi/4 d_v^2$)× 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 engine.
Vacuum pressure varies from 0.05 to 0.07 MPa for petrol engine.
Inertia of valve = Mass of valve× Acceleration.

The area of passage through the valve's throat and reduction to its axis is referred to as the design mass of the valve, or alternatively, the valve axis is referred to as the equivalent mass of the valve gear.
Empirically it can be found as under:
Mass of valve = m = 230× (port area in m^2) = $230\times(\pi/4\, d_{p0}^2)$ 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+ $\alpha+\beta$
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,
$\omega = 2\pi(N/2)/60$ rad/s and cam shaft speed is half that of crank shaft speed and S= lift of valve.

Design 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, $R = l d_p P_b$

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 (Boysal & Rahnejat, 1997).
Hence inner diameter of hub= di = dp +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:

Push rod 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 σc 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.
The valve spring is designed in the same way as a compression spring. The purpose of a spring is to give spring force for the valve to close and to keep the cam in constant contact with the follower. When a spring surges excessively, as it does in a high-speed engine, the follower may separate from the cam contact.

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 = $\pi d_w^3 \tau / 16k$
When valve is closed, load on spring is minimum.
The allowable shear stress for carbon steel spring is 280 to 320MPa.

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

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 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 as below.
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.

IV. CONCLUSIONS

The internal combustion engine is the greatest invention ever made by humans. It is utilized everywhere including routine use. Due to shortage of fissile fuel and strict environment norms for emissions, it is necessary to evolve design of ICE. The work here reviews a fundamental designing various ICE components. When designing, no thermal or other factors are taken into account. In addition to valve and valve gear mechanism, we have reviewed design of many ICE components such as piston, piston rings, cylinder, cylinder head, connecting rod, crank and crank shaft, cam and cam shaft, and so on.

Aim of the work is to analyze different design parameters of ICE system in integration to design fundamentals with certain assumptions as mentioned in different sections. Expected outcome of this work is to have integration between design parameters and ICE systems which can be utilized for future design modifications as and when needed. 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 design data as per standard design data banks. This work creates a base for future design modifications in ICE cylinder and other relevant component design.

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