Journal of Mechanical and Construction Engineering, 2023,
Vol. 03, Iss. 02, S. No. 039, pp. 1-12
ISSN (Online): 2583-0619

Failure Modes and Empirical Relations to Design Piston Pins for IC Engine

Mahesh Choudhary[1], Emarti Kumari*[2], Brijesh K. Gurjar[3], Mahesh Bishnoi[4], Deepak Sharma[5], Hemant Jagrat[6], Manish[7], Mukesh Choudhary[8], Deepika Choudhary[9], Saroj[10]

[1, 2, 3, 4, 5, 6, 7, 8, 9, 10]Department of Mechanical Engineering, MBM University, Jodhpur, Rajasthan, India
[1]choudharymahesh2001@gmail.com, [2]emarti.me@mbm.ac.in, [3]brijeshg8156@gmail.com, [4]mahesh29102001@gmail.com, [5]sharmadeeepak812@gmail.com, [6]hemantjagrat999@gmail.com, [7]manish.mns28@gmail.com, [8]choudhary8016@gmail.com, [9]deepikachoudhary8656@gmail.com, [10]saroj.kaswa2000@gmail.com

How to cite this paper: M. Choudhary, E. Kumari, B. K. Gurjar, M. Bishnoi, D. Sharma, H. Jagrat, Manish, M. Choudhary, D. Choudhary and Saroj, "Paper Title," *Journal of Mechanical and Construction Engineering (JMCE), Vol. 03, Iss. 02, S. No. 039*, pp. 1–12, 2023.

https://doi.org/10.54060/jmce.v3i2.39

Received: 27/06/2023
Accepted: 25/07/2023
Published: 25/11/2023

Abstract

In this article, authors discussed the various boundary conditions (fully floating, semi floating and stationary) and failure modes (transverse crack and longitudinal crack) of piston pin of IC engine. Moreover, authors given the empirical relations for shear stress, bending stress and ovalization stress to design piston pin for internal combustion engines. Furthermore, carried out the force analysis on piston pin and expressed the empirical relations for force analysis of piston pin that will be very useful for design of piston pin for petrol and diesel engines.

Keywords

piston pin, internal combustion engine, axial crack, circumferential crack

1. Introduction

The basic aim of a design of a system is to make that system work with maximum economy and efficiency. To follow this philosophy current design of IC engine components is moving towards higher compression ratio engines. With increases in demand of more economical and environment friendly engine, the need for more accurate and reliable parts of engine is be-coming prominent. While the engine developed quickly and engine performance improved greatly, the demand for engine parts is becoming higher and higher.

The piston pin or the wrist pin or the gudgeon pin is one such important part which is one of the highly stressed engine components, and therefore that must be designed carefully. The word gudgeon means pivot or journal. Piston pin connects the piston and small end of the connecting rod of IC engines providing bearing for the connecting rod to pivot upon as the piston assembly reciprocated. The central part is connected along the piston pin axis to small end of connecting rod and the lateral parts are supported in piston bosses. The piston pin is subjected to alternating loads during operation of the engine. It is a simple component with no moving parts but it must be engineered with precision as it has to satisfy some conflicting requirements such as it must have high strength but light in weight, it should be close fitting but also have freedom of movement, it should resist wear without scuffing, etc.

The piston pin forms a connection between the piston and the connecting rods in the internal combustion engine thus, it transfers gas pressure from piston to the connecting rod. The piston pin is placed in piston boss and when the gas pressure is transferred from piston to connecting rod, piston pin can experience bending. So, the piston and piston pin must have sufficient stiffness to endure the pressure and friction between contacting surfaces. Piston pins are generally made hol-low because of two reasons, one being they will have less weight than that of solid piston pin so it will reduce inertia force of the reciprocating system during operation of IC engine, secondly in hollow pins (act as short shaft) strain energy is more than solid pins. In addition, as an important part in engine, the working condition of piston is directly related to the reliability and durability of engine. So, it is important for the piston pin and the piston pin boss to carry out structural and optimal analysis which can provide reference for design of piston.

Various failure modes in piston pin of IC engine were discussed by Strozzi et al. [1]. Wen et al. [2] used an improved wireless instantaneous mean effective pressure measurement technique to conduct accurate measurement of piston assembly friction indirectly by measuring cylinder pressure, the connecting rod force, the crank position and the engine speed un-der various conditions. Along with friction force they also done work on variation of different forces in engine with crank angle. Empirical relation to design piston was given by Karwa [3]; Kolchin and Demidov [11]. Binaco et al. [4] investigated the influence of different piston bosses profiles were investigated on the elasto-hydrodynamic contact pressure distribution at the piston-gudgeon pin interface of a high performance supercharged engine. Design and shape optimization of connecting rod was carried out by Vijayvergiya et al. [5] and optimization of cylinder head was conducted by Kumari et al. [6]. Recently, analysis of thin-wall structures was carried out [7-10] in the presence of thermal and mechanical load considering various boundary conditions. Clark [12] and Fanghui [15] studied the free-floating piston pin behavior of heavy-duty engines'; further static responses of piston pin has been investigated by Debnath and Debnath [12]. The thermos-elastic analysis of piston was carried by Allmair and Sander [14]. Here, authors discussed the various failure modes (axial and transvers), boundary conditions (fixed with connecting rod, fixed with bosses and free floating), stresses and forces in piston pin for petrol and diesel engines.

1.1. Types of Piston Pin

On the basis of method of securing the piston pin in its position it is designed in following three configurations:

1.1.1. Fully floating piston pin

This arrangement of the piston pin is one of the most commonly used configurations in which piston pin is free to rotate in both small end of connecting rod and the piston bosses thus having double swivel action. The pin is secured in its position with help of various means which include circlips, caps, plugs, snap rings, etc as shown in Figure 1. In this type of arrangement since piston pin is free to rotate in both small end of connecting rod and in piston bosses it has improved lubrication compared to other types. It also helps in the reduction of thrust load on piston skirt due to its double swivel action.

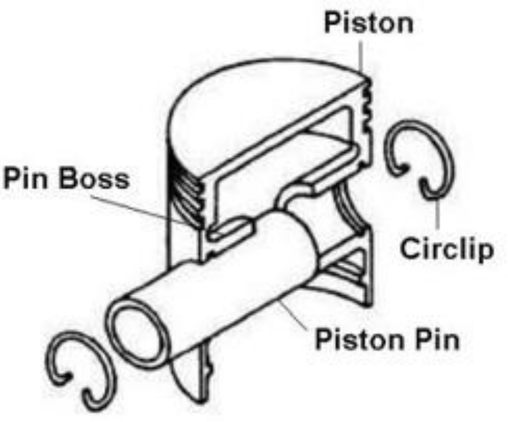

Figure 1. Full-floating piston pin (theengineerspost.com)

1.1.2. Semi floating piston pin

In this type of arrangement piston pin is free to rotate in piston bosses and its motion is restricted in small end of connecting rod. This can be done by various methods like providing a circumferential groove on piston pin and clamping small end of connecting rod by aligning it with that groove or heating the small end with help of oxyacetylene torch to expand it and then centrally aligning the piston pin till the small end cools down to shrink tight on pin. As it can be seen in Figure 2 where piston pin is clamped inside small end of connecting rod to restrict its motion and it is free to rotate in piston bosses.

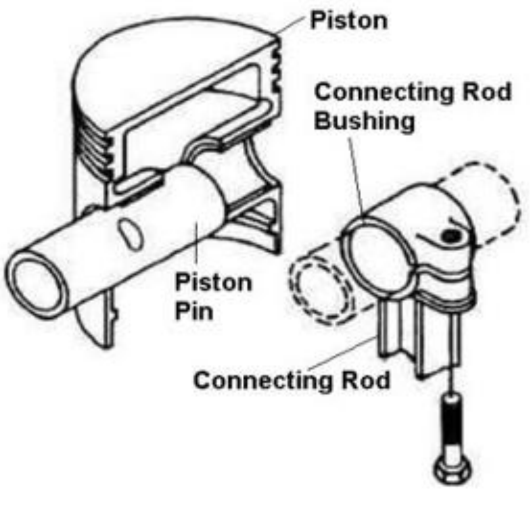

Figure 2. Semi-floating piston pin (theengineerspost.com)

1.1.3. Stationary piston pin

In this type of arrangement piston pin is secured in piston bosses and connecting rod is made free to rotate around the pin as shown in Figure 3. Since during heavy duty applications due to transferring of heavy thrust force via piston pin, connecting rod rotating about it can cause uneven wear, so this type of arrangement is generally not used.

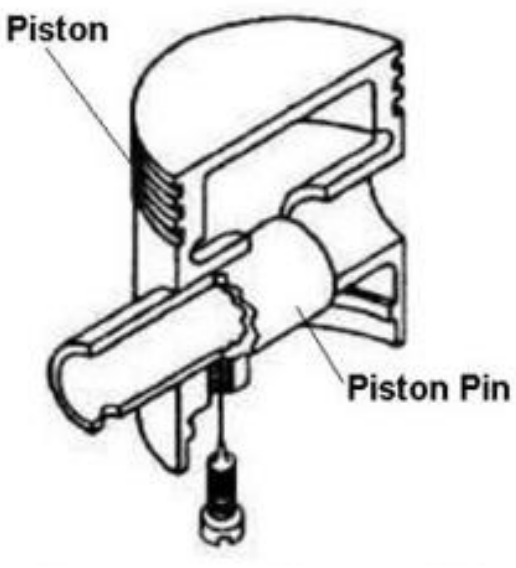

Figure 3. Stationary piston pin (theengineerspost.com)

1.2. Material selection and surface treatment of piston pin

The piston pin operates under high temperature in internal combustion engine and due to its location, it is one of the com-ponents that receives poor lubrication. Along with this it should have sufficient strength to bear all the bending, shear-ing and compressive loads it experiences during operation while remaining light weight so that it does not add much to re-ciprocating mass. To satisfy all these requirements piston pins are generally forged as a hollow cylinder made of alloy steel (Nickel / Chromium). In order to improve the wear resistance of the piston pin their surface treatment is done to harden the surface. Often choice of the surface treatment methods influences the selection of material.

There are mainly two types of surface hardening for the piston pin – carburising (also known as case hardening) and nitride hardening. These surface hardening methods impart some compressive residual stresses resulting in improved resistance to fatigue failure of piston pin increasing the durability of the component. Based on these methods piston pins are generally made out of either case hardened steel (mostly) or nitride hardened titanium. Though titanium has much lower density than steel resulting in light weight, it has much lower elastic modulus compared to steel which makes it unfavorable to use it as piston pin material as high stiffness material is desired for piston pin. Also, treatment of both inner and outer surface of the piston pin makes it more durable as it increases resistance to fatigue failure compared to treating either only outer or inner surface only.

Some low friction coatings are also applied on the piston pin surface along with hardening of surface to increase wear re-sistance and reduce any tendency of component to seize under heavy duty operation. The most common type of coating for piston pin is diamond like carbon (DLC) coating.

2. Failure modes in piston pin

Piston pin is one of the simplest and most important element internal combustion engine. The knowledge of pin failure point and various pin failure modes are always useful for a designer to design this element. The piston pin generally fails (fracture) due to fatigue loading during operation of the engine. Various points of probable failure- A, B, C, D, E, F at different locations is shown in **Figure 4**. It also fails due to wearing out of its surface which can be reduced by its surface treatment as discussed section 1.2.

In a piston pin the fracture crack initiates at a bore point. Among the three bore points shown in **Figure 4**, A is the most stressed point. On the basis of the propagation of crack, following failure modes are there in piston pin:

- Circumferential or transverse crack - crack originates due to shear stress in piston pin.
- Axial or longitudinal crack - this type of crack originates due to ovalizing stress.

Often both types of cracks coexist in piston pin.

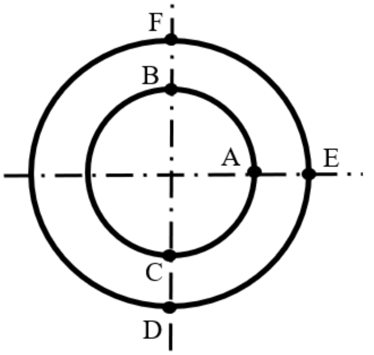

Figure 4. Schematic representation of various failure modes [1]

2.1. Circumferential or transverse crack

The general cross section of a piston pin is shown in **Figure 4** where A, B, C, D, E and F are the general points lying in a cross section of pin and are probable failure points. Among these points A is the most stressed point.

Figure 5. Circumferential crack in piston pin [1]

If we consider compressive loading on pin during combustion, since during combustion corresponding stresses are higher and at higher stresses failure of machine component take place. The highest stress during combustion occurs at point A of cross section i.e., at the pin bore side and at the support transition zone which is the gap where support to piston pin from piston bosses get transferred to support from small end of connecting rod. In this gap the piston pin attains its maximum ovalizing stress value and maximum shear stress value. So, at point A there is a compressive ovalizing stress, a compressive axial bending stress and shear stress, thus point A being the point of maximum stress the crack initiates from point A. Circumferential crack in a piston pin can be seen from the **Figure 5**.

If the piston is in floating type configuration, it may rotate inside its support as a result the direction of shear stress gets reversed, other stresses being in same direction as a result a new symmetric crack at any angle may propagate forming most common Y-shaped crack in piston pin as shown in **Figure 6.**

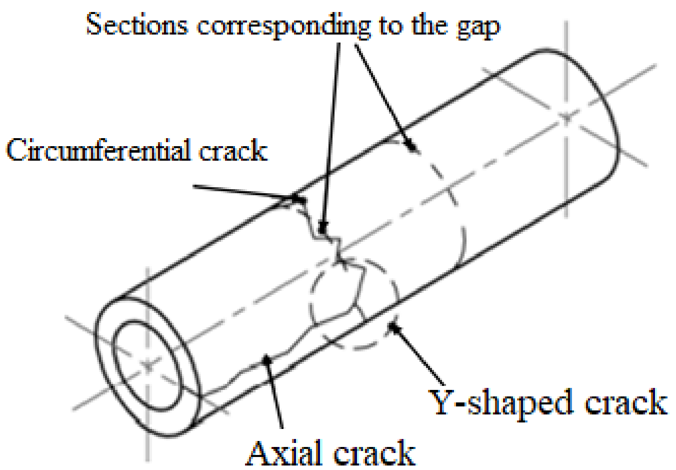

Figure 6. Y-shaped crack in piston pin *[1]*

2.2. Axial or longitudinal crack

Axial or longitudinal crack as shown in **Figure 7** is a type of crack initiated at point A and it propagate due to ovalizing stress towards the extreme ends of piston pin. On moving away from the transition zone, the ovalizing stresses dominate and shear stress diminishes resulting axial crack propagation.

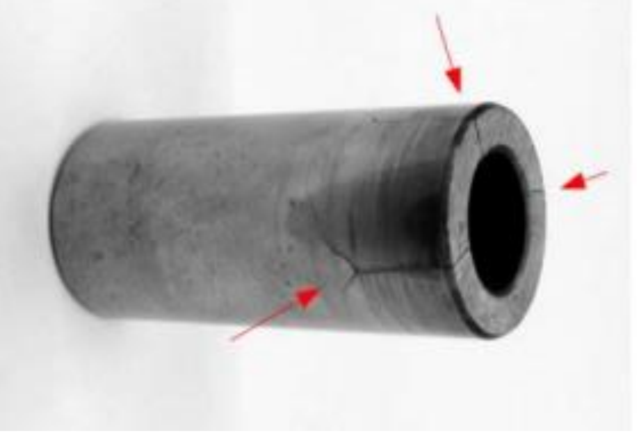

Figure 7. Axial crack propagation [1]

3. Stress Distribution in piston pin

Through literature it is noticed that most of the piston pins are hollow cylinders in which central part is contained in con-necting rod small end and extreme parts in the piston bosses. As it can be seen in **Figure 8 (a),** a little gap is provided between small end of connecting rod and piston bosses to allow thermal expansion of piston assembly and connecting rod. From **Figure 8 (b),** the maximum value of pressure distribution on piston pin is at the point where support to piston pin changes from piston bosses to connecting rod, thus we can see that this region experiences maximum stress.

During operation of internal combustion engine, gas force and inertia force are transmitted through piston pin from piston to connecting rod due to which piston pin is subjected to multiple distributed loads resulting in following types of stress in pis-ton pin:

3.1. Shear Stress

Due to inverse loading, piston bosses and small end of connecting rod as shown in **Figure 8** (a) the shear stress in the pin, shearing along two planes between the bosses and the small end of connecting rod is given by equation (1):

$$\tau = \frac{F_g}{2\left(\frac{\pi}{4}\left(1-\alpha^2\right)d_p^2\right)} \tag{1}$$

The allowable shear stress for piston pin is 60 MPa – 250 MPa where 250 MPa is the upper limit of alloy steel.

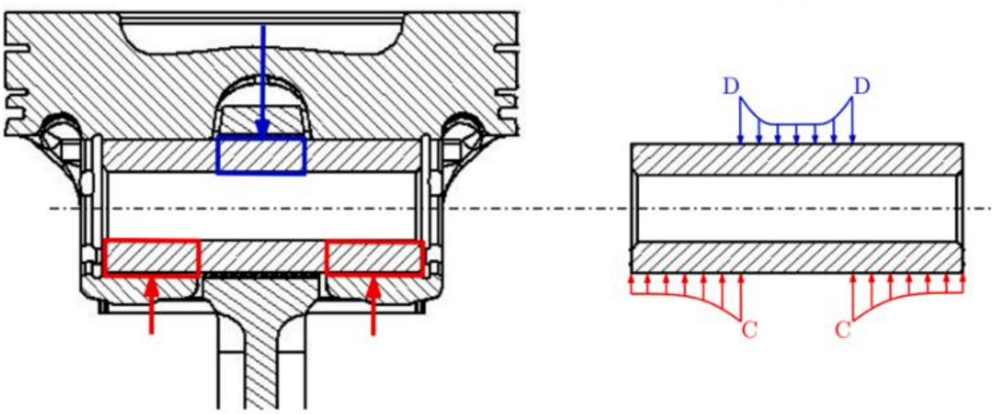

Figure 8. Schematic representation of (a) forces at top dead centre (b) pressure distribution on piston pin [2]

3.2. Bending Stress

The load distribution in the piston pin is such that it experiences a bending stress which is given by equation (2):

$$\sigma_b = \frac{M}{z} = \frac{\frac{F_g}{2}\left(\frac{2l_p+b}{12}\right)}{\frac{\pi}{32}\left(1-\alpha^4\right)d_p^3} = \frac{F_g(l_p+0.5b)}{1.2\left(1-\alpha^4\right)d_p^3} \tag{2}$$

3.3. Ovalization Stress

Due to vertical load, the piston pin ends are squeezed into somewhat elliptical shape called ovalization of piston pin cross section.

$$\sigma_0 = \frac{M_0(r_n - r_i)}{(r_0 - r_i)(r_m - r_n)r_i} \tag{3}$$

Where; Mo is ovalization moment of piston pin for section 1-2 and section 3-4 as shown in **Figure 9** is given by equation (4):

$$M_{1,2} = \frac{F_g(2r_m \ln\frac{r_0}{r_i} - 3(r_0 - r_i))}{L8\ln\frac{r_0}{r_i}} \tag{4}$$

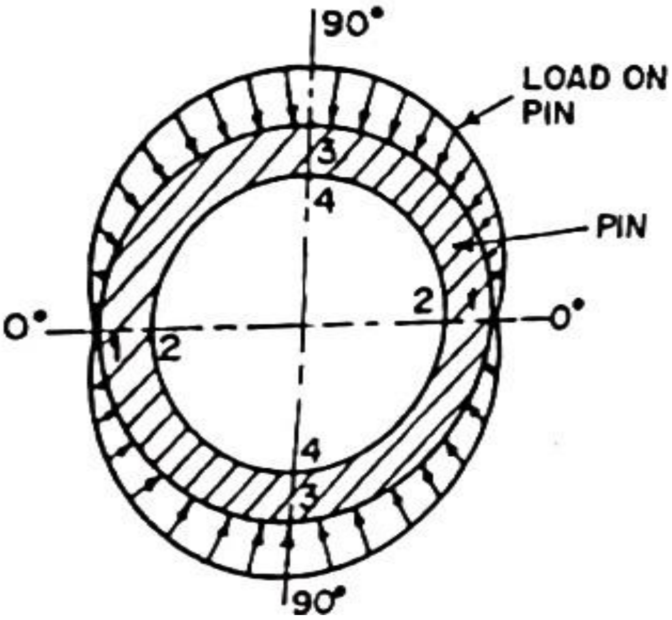

Figure 9. Ovalizing stress variation [3, 11]

$$M_{3,4} = \frac{F_g(4r_m \ln\frac{r_0}{r_i} - 3(r_0 - r_i))}{L8\ln\frac{r_0}{r_i}} \tag{5}$$

4. Pin Geometry

Piston pin is one of the highly stressed elements in IC engine. During operation it is mainly subjected to gas force and inertial force of reciprocating masses. In order to limit inertia force due to reciprocating mass the pin must be light weight. In order to achieve that piston pin are basically made cylindrical and hollow. The length of pin depends upon the application where the engine will be used. If it is not so demanding application, the pin length is made as long as possible to reduce contact pressure; the length piston pin is given in Table 1.

Table 1. The length of piston pin for SI and CI engines.

Pin Length	SI Engine	CI Engine
Stationary Pin	(0.88 – 0.93) D	(0.88 – 0.93) D
Floating Pin	(0.78 – 0.88) D	(0.8 – 0.9) D

For the case of high-performance engines, in order to limit the weight, the piston pin is made shorter by 0.4 times of piston diameter D. By making the piston pin hollow the weight does get reduced but ovalizing stress come into picture which is absent in solid pins. In order to limit the effect of ovalizing stress it should not be reduced beyond certain limit. The empirical relations for piston pin diameter and connecting rod bushing length are given in **Table 2**.

Table 2. Empirical relations for piston pin diameter and connecting rod bush length.

Piston pin diameter	SI Engine	CI Engine
outer diameter (d_p)	(0.22 – 0.28) D	(0.3 – 0.38) D
inner diameter (d_i)	(0.65 – 0.75) d_p	(0.5 – 0.7) d_p
Connecting rod bushing length (l_{cr})		
Retained pin	(0.28 – 0.32) D	(0.28 – 0.32) D
Floating pin	(0.33 – 0.45) D	(0.33 – 0.45) D

In order to achieve the same mean pressure along the lateral and central supports, the axial length of lateral supports should be 1.3 times the central support length. Weight of the pins is also reduced by providing taper to the pin at extremities, making complex taper in pin, providing reinforcement at high stress zone and reducing pin thickening, etc.

4.1. Clearance in piston pin

There should be an initial clearance between pin outer surface, connecting rod small end and piston bosses. The clearance should be adequate in all operating conditions. It must not be too small so that seizure of engine might occur at high operating temperature, and it should not be too high so that it affects the film or lubrication system over pin and make unwanted noise in engine. So, it is suggested that the diametral clearance falls in range of 0.0008 – 0.003 times the pin outer radius.

4.2. Piston Pin offset

Piston pin is not often placed at the centre of piston, it is being bit offset on one side in the piston. The reason for the piston pin offset is to reduce the stress on connecting rod when it reaches top dead centre or bottom dead centre. When piston pin is placed at centre of piston, whenever the connecting rod reaches TDC or BDC it experiences load as it becomes straight up or down instead if a little offset is provided the rod travels more in an arc which makes it easy for rod to rotate at TDC and BDC giving more output power.

5. Forces analysis for design of piston pin

The piston pin connects piston to connecting rod thus all the forces acting on piston are transmitted to connecting rod through piston pin. In this mechanism gas force (F_g) acts on piston due to combustion of air fuel mixture, connecting rod force (F_c) acts in connecting rod whose direction changes according to rotation of crank and there is friction force (F_f) between the piston and cylinder, thrust force (F_n) perpendicular to the movement of piston and inertia force (F_{in}) due to reciprocating motion of the piston assembly as shown in **Figure 10**.

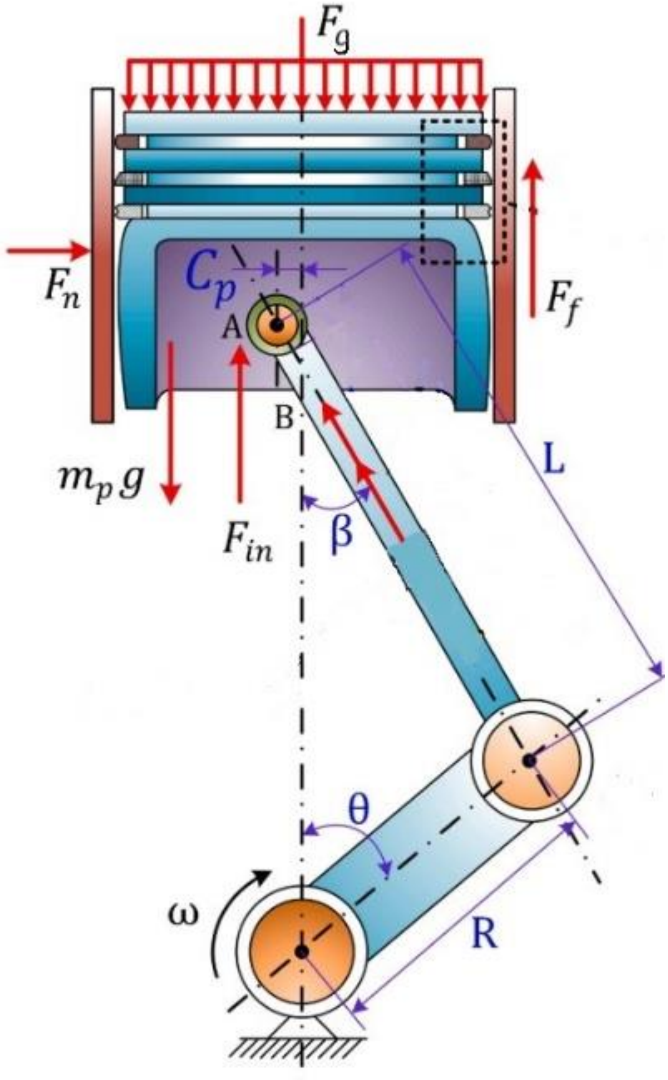

Figure 10. Forces in piston assembly [2]

Calculation of gas force on piston is done by calculating the cylinder pressure during operation of the engine. If pressure inside combustion chamber is P_c then gas force F_g is given by:

$$F_g = \frac{\pi D^2}{4} P_c \tag{6}$$

In order to calculate inertia force we do analysis of kinematic relations as schematic forces are shown in **Figure 10**:

$$S = R\cos\theta + L\sqrt{1-\lambda^2} \tag{7}$$

Here, *S* is the distance from the piston pin centre to the main bearing centre in axial direction of cylinder liner, *R* is the crank radius, θ is crank angle. The axial acceleration of piston can be calculated as given in equation (8) below:

$$a_p = \frac{d^2 s}{dt^2} = -\alpha R\left(\sin\theta + \frac{\lambda\cos\theta}{\sqrt{1-\lambda^2}}\right) - \omega^2 R\left[\cos\theta - \frac{\lambda\sin\theta}{\sqrt{1-\lambda^2}} + \frac{R\cos^2\theta}{L(1-\lambda^2)^{3/2}}\right] \quad (8)$$

Here, α is the angular acceleration, ω is the rotation speed of crankshaft, *L* is the length of connecting rod and $\lambda = \frac{C_p + R\sin\theta}{L}$. Thus, inertial force can be given as $F_{in} = -m_p a_p$ where m_p is the mass of piston pin assembly including piston.

Now if we calculate forces on piston pin (F_p) due to gas force and inertia force of reciprocating mass system, it is given by:

$$F_p = F_c + F_{in} \quad (9)$$

$$F_p = \frac{\pi D^2 P_c}{4} + kF'' \quad (10)$$

Here, P_c = maximum gas pressure; kF'' is inertia force of mass of piston assembly excluding the mass of piston pin; k is a factor to take account of mass of piston pin where 0.76 to 0.86 for SI engine and 0.68 to 0.81 for CI engine;

F'' is inertia force of piston mass at speed '*n*' which is equal to $F'' = -m_p R\omega^2(1+\varphi)$; where φ is ratio of crank radius R to connecting rod length *L.*

Along with inertia force and gas force piston pin also experiences bearing pressure due to both small end of connecting rod and piston pin bosses. For modern engines the limit of bearing pressure on small end of connecting rod is 20 MPa – 60 MPa and for piston pin bosses it is 15 MPa – 50 MPa. The bearing pressure exerted by piston pin on the small end of the connecting rod is expressed by equation (11).

$$(pb_1) = \frac{F_g}{d_0 L_{cr}} \quad (11)$$

Where, F_g is gas pressure, d_o is the outer diameter of piston pin and L_{cr} is the length of the pin bearing surface at the small end of connecting rod.

Similarly, the bearing pressure exerted by floating pin, if employed on piston bosses where (L_p – *b*) is the length of pin bearing surface in the bosses is written as:

$$(pb_2) = \frac{F_g}{d_0(L_p - b)} \quad (12)$$

6. Conclusion

In this communication authors discussed the failure modes such as axial and circumferential crack which are occurs in piston pin of IC engine. Thereafter, discussed the various boundary used to mount piston pin in IC engine. Furthermore, empirical relations are given to calculate bending, shear and ovalization stresses induced in piston pin. Also discussed the pin geometry and empirical relations in the terms of piston diameter. Moreover, force analysis of piston is carried out to design a piston pin for IC engine.

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