Comprehensive Design and FEM Analysis of Piston and Connecting Rod using ANSYS for Two Wheeler Engines

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Abstract- Connecting rod and piston are crucial components of IC engines. Connecting rod is the connecting member which is fitted to piston with small end and big end to the crankshaft. It is a vital member to transmit the activities of the piston to crank journal, thus the conversion of the reciprocating motion of the piston to rotary motion of the crankshaft. Piston is a reciprocating component of IC engine. Its sole function is to transfer combust gas force in the cylinder via connecting rod. Aluminium alloy is used as material for both piston and connecting rod. In this work, we took an Engine Specification of two wheelers as an input, we designed a Piston and Connecting rod by finding safe dimensions, and modelled using CREO Parametric Software and we did structural analysis, Thermal analysis and fatigue analysis of the connecting rod and Piston for 100CC engine of a Bike. In FEA analysis, it was found the Von misses stress; total displacement, Fatigue life, and Factor of safety are using ANSYS software are very much acceptable and reliable too.

Keywords: Connecting rod; Piston CREO Parametric; Analysis and ANSYS

1. INTRODUCTION

The connecting rod consists of small end to assemble the piston pin, a long shank with I sectional cross section, and a big end cap is split into two parts to fasten the crank journal or pin. The construction of connecting rod is illustrated in Fig 1. The basic function of connecting rod is to transfer the work force of piston to the crank journal. The connecting rod transmits the reciprocating motion of piston to the rotary motion of crankshaft. It also transfers lubricating oil from the crank pin to the piston pin through a small hole and provides a splashing of jet of oil to piston assembly. Pathade et al [1] have investigated the stress analysis of connecting rod by finite element method using pro-e wild fire 4.0 and ANSYS workbench 11.0 software and concluded that the induced stress in Small end of connecting rod was greater than that of the Stress induced at the big end of C.rod, therefore the chances of failure of the Connecting rod may be at the fillet section of both end. Kumar et al [2] has approached the problem with modeling and analysis of connecting rod. In this project Aluminium Boron Carbide material for connecting rod replaces the carbon steel with working factor of safety [FOS] is closer to theoretical factor of safety, which is to increase the stiffness by 48.55% and the stress was reduced by 7.84%.

2. LITERATURE REVIEW

The connecting rod of I.C. engine is normally produced by hot drop forging method and then finished by cutting, grinding, drilling and so on. Most internal combustion engines are forged as one piece connecting rod. The bearing cap is then cut off, faced and bolted in place for final machining of the big end. The connecting rod small end is generally made as a solid eye, then machined and fitted with phosphor bronze bush. The connecting rod is subjected repetitive forces due to engine strokes is shown in Fig. 1. It is one of the most heavily stressed parts of the I.C. engine [3].

![Fig. 1 View of A connecting rod with ‘I’ section](image)

2.1 Material Used - Aluminium alloy

Connecting rod for an automotive should be light in weight and at the same time much stronger and to withstand repetitive stresses. This tendency in vehicle construction compels the researchers to invent for quite new materials which are light and meet design requirements. Lighter connecting rods decrease load caused by inertia forces developed in engine and does not require huge balancing weights or counter weights on crankshaft in case of dynamic balancing. Many companies have already started the manufacturing of aluminium connecting rods reinforced with steel continuous fibres [4]. It results in less weight, increase of durability of the part. It also reduces the emission to large extent.

2.2 Cross Section of Connecting rod

Normally ‘o’, ‘I’ cross sections are there. But mostly I section is used in practice. The design of the
connecting rod length is an important factor. When the connecting rod is diminutive / small as compared to the crank radius, it has greater angular swinging, resulting in increase of side thrust on the piston. In high-speed engines, the ratio of the length of the connecting rod to the crank radius \((L/r)\) is generally 4 or >4. In low-speed engines, the \((L/r)\) ratio varies from 4 to 5. Most of the connecting rods in high-speed engines have an I-section because it reduces the weight and inertia forces and also easy to forge. Most rods are provided with rifle-drilled long hole throughout the length from the small end to the big end to carry the lubricating oil from crank journal to the piston pin bush bearing. In low-speed engines sometimes circular cross-section is used.

2.3 Buckling of Connecting rod

The connecting rod is an important engine component that has sizeable length in proportion to its width and breadth. It is subjected to axial compressive force equal to maximum combustion gas load on the piston. The compressive stress is of considerable in magnitude [5]. Therefore, the connecting rod is considered and designed as a column or a strut. The buckling of the connecting rod is seen in two different planes—plane of motion and a plane perpendicular to the plane of motion as illustrated in Fig 2.

The following observations are made with reference to this figure. The connecting rod buckling is seen in the plane of motion as shown in Fig. 2(a). In this plane, connecting rod ends are connected to crank pin and piston pin. Therefore, for buckling about the XX-axis, the end fixity coefficient \(n\) is one.

The buckling of the connecting rod is in a plane perpendicular to the motion of the plane as shown in Fig. 2(b). In this plane, connecting rod ends are fixed due to constraining effect of bearings at the crank pin and piston pin. Therefore, for buckling about the YY-axis, the end fixity coefficient \(n\) is four. Hence the connecting rod is four times stronger for buckling about the YY-axis as compared to buckling about the XX-axis. The design of connecting rod is done in such a way that it is equally resistant to buckling in either plane then [7].

\[
4 \ I_{yy} = I_{xx} \quad (1)
\]

where, \(I = \) Moment of inertia of cross-section (mm4)

Substituting \((I = AK^2)\), we have \(K_{yy} = 0.25 \ K_{xx}K\)

\(K = \) radius of gyration of cross-section (mm)

The above relationship proves that I-section is ideally suitable for the connecting rod. On the other hand, a circular cross-section is unnecessarily strong for buckling about the YY-axis. Fig. 3 shows the typical proportions for the ‘I’ cross-section of the connecting rod for IC engine. For this cross-section,

2.4 Piston

The piston is a reciprocating part of IC engine that performs a number of functions [6]. The main functions of the piston are as follows:

- It transmits the force due to combustion gas pressure of inside the combustion chamber to the crankshaft via connecting rod.
- Piston compresses the gas at compression stroke.
- It provides a seal for the cylinder from the crankcase by means of piston rings.
- It takes care of the side thrust resulting from rocking of the connecting rod.
- It helps in dissipating large amount of heat from the combustion chamber to the cylinder wall.

Trunk type piston, shown in Fig. 4 and Table 1, is used in IC engines. It consists of the following parts.
Table 1 Material Properties of Aluminium alloy

<table>
<thead>
<tr>
<th>Sl. No.</th>
<th>Properties</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Density in kg/m³</td>
<td>2770t</td>
</tr>
<tr>
<td>2</td>
<td>Coefficient of thermal expansion in C⁻¹</td>
<td>0.00035</td>
</tr>
<tr>
<td>3</td>
<td>Young Modulus in MPa</td>
<td>71000</td>
</tr>
<tr>
<td>4</td>
<td>Bulk Modulus in MPa</td>
<td>6960</td>
</tr>
<tr>
<td>5</td>
<td>Shear Modulus in MPa</td>
<td>2669.2</td>
</tr>
<tr>
<td>6</td>
<td>Tensile yield strength in MPa</td>
<td>280</td>
</tr>
<tr>
<td>7</td>
<td>Compressive yield strength in MPa</td>
<td>280</td>
</tr>
<tr>
<td>8</td>
<td>Poisson ratio</td>
<td>0.33</td>
</tr>
<tr>
<td>9</td>
<td>Melting Point in °C</td>
<td>670</td>
</tr>
</tbody>
</table>

3. DESIGN CALCULATION

3.1 Engine specification

Consider a 100 cc engine

- Engine type = air cooled, 4-stroke
- Bore x Stroke (mm) = 50 x 49.5
- Maximum Power. = 7.37 HP (5.4 kW) at 8000 rpm
- Maximum torque. = 7.95 Nm at 5000 rpm
- Compression Ratio = 9:1

3.2 Section of Connecting rod

From standards,

- Thickness of the flange and web of the section = t
- Width of the section B = 4t
- Height of the section H = 5t
- Area of the section A = 11 t²
- Moment of Inertia about X axis Ixx = 34.91 t⁴
- Moment of Inertia about Y axis Iyy = 10.91 t⁴

Therefore Ixx/Iyy = 3.2

Finite element analysis Length of the connecting rod (L) = 2 x stroke = 99 mm.

3.3 Maximum Force acting on connecting rod

\[ F_c = \frac{\pi}{4} \times d^2 \times P_{max} \]  
\[ F_c = \frac{\pi}{4} \times 50^2 \times 6; \quad F_c = 11.781 \text{ kN} \]

Critical buckling load:

\[ P_{cr} = F_c \times \text{FOS} = 11.781 \times 6 = 70.68 \text{ kN} \]

Rankine’s formula

\[ P_{cr} = \sigma A / \sqrt{1+a(L/K_{xx})^2} \]

where, \( A = 11t^2 \)
\( a = 1/5000 \)
\( K_{xx} = \sqrt{I / A} \)
\( 70680 = 280 \times 11t^2 / [1+0.002(99/1.78t)1] \)
\( t = 5.546 \text{ mm} \)

Diameter of Small end and big end of connecting rod :

\[ F_c = D_p \times L_p \times P_{bc} \]  
\[ F_c = D_c \times L_c \times P_{bc} \]

where, \( P_{bc} = 10 \text{ MPa}, L_c/D_c = 1.25\)
\[ 11781 = 1.25 \times D_c \times 10000 \times 10^3 \]
\( D_c = 33.7 \text{ mm} \)

3.4 Results

- Thickness of flange and web of the section t = 5.546 mm
- Width of the section B (4t) = 22.18 mm
- Height at the middle of the section H (5t) = 27.73 mm
- Height at the small end of the section (0.9 H) = 24.95 mm
- Height at the big end of the section (1.25 H) = 34.66 mm
- Diameter of the big end of the connecting rod (Dcb) = 33.70 mm
- Diameter of the small end of the connecting rod (Dcs) = 28.71 mm
- Length of the big end of the connecting rod (Dcb) = 33.70 mm
- Length of the small end of the connecting rod (Dcs) = 28.71 mm.

3.5 Connecting rod - Thickness of the piston (tH)

The piston head or crown is designed by considering the following:

(a) It should be adequately strong to withstand the straining action due to in cylinder gas explosion pressure, and (b) Its ability to dissipate the combustion heat through the cylinder walls to cooling medium as quickly as possible.

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

\[ t_H = D \sqrt{\frac{3P_{max}}{16\sigma_b}} \]

\[ t_H = 4.74 \text{ mm} \]

where, Maximum pressure in N/mm², \( P = 6 \text{ N/mm²} \)

Cylinder bore outside diameter is in mm, \( D = 50 \text{ mm} \), Material is a AL alloy whose yield tensile strength is 285 MPa and F.O.S. is 2.25.

The material possess the Permissible tensile stress as in N/mm², \( \sigma_t = 125 \text{ in N/mm²} \). Next second consideration of heat transfer, the thickness of the piston head/crown thickness should be the heat absorbed by the piston during combustion of fuel is quickly transferred to the cylinder walls, Treating the piston head as a flat circular plate, its thickness is given by

\[ t_H = H / \{12.56(K_c - T_e)\} \]

\[ t_H = 4.74 \text{ mm} \]

where, Heat flowing through the piston head in kJ/s or kW,

\[ H = C \times HCV \times m \times B.P \]  
\[ = 407.53 \text{ W} \]
Heat conductivity in W/m°C, \(k = 175\) W/m°C for, the difference of temperature (TC – TE) =75°C for aluminium alloy; Constant of the heat supplied to the engine that is absorbed by the piston, \(C = 0.05\).

Higher Calorific value of the fuel in kJ/kg, HCV = \(47 \times 10^3\) kJ/ kg for petrol, Mass of the fuel used in kg per bp/ sec, \(m = 0.15\) kg per bp/ sec; Break Power in kW, B.P. = \(2\pi NT/60 = 4.162\) kW.

3.6 Radial thickness of ring (b)

The radial thickness (b) of the piston ring is obtained by considering the radial pressure between the cylinder wall and ring, from bending stress consideration in ring. The radial thickness is given by

\[ b = D \sqrt{3Pw/\sigma t} \]

\(Pw = 0.034\) MPa, \(\sigma t = 110\) MPa

\[ b = 50 \sqrt{3\times0.034/110} \]

\[ b = 1.523 \]

![Fig. 5 Nomenclature of Ring](image)

3.7 Axial thickness of ring (h)

The thickness of the rings may be taken is shown in Fig. 5. Nomenclature of Ring, \(t2 = 0.7b\) to b

Let \(b = 1.523\) mm

Minimum axial thickness, \(h = D/ (10\times nr) = 1.667\) mm; where, number of rings but the reference number, \(nr = 3\)

Width of the top land (h1):

The width of the top land varies from,

\(h1 = tH\) to 1.2\(tH\); \(h1 = 5.214\) mm

Width of ring land (h2):

Width of other ring lands varies from,

\(h2 = 0.75h\) to \(h\); \(h2 = 1.133\) mm

Maximum thickness of barrel (t3):

Maximum thickness of barrel can be calculated by

\[ t3 = (0.03xD) + b + 4.5\) mm = 7.923 mm; Length of the Piston skirt (Ls):

\[ Pc = 11.781\) KN

Side thrust = \(0.1Pc = Pb \times DLs\)

\(0.1 \times 11781 = 0.25 \times 50Ls\)

\(Ls = 94.24\) mm

Length of the Piston (L):

\[ L = Topland + length of ring section + Length of skirt 16, \]

\[ = 1.133+5.214+40 \]

\(L = 46.35\) mm

Result:

The piston crown Thickness, \(tH = 4.74\) mm

Maximum thickness of barrel, \(t3 = 7.923\) mm

Width of the top land, \(b1 = 5.214\) mm

Width of the other land, \(b2 = 1.133\) mm

Axial thickness of the ring, \(h = 1.295\) mm

Radial thickness of the ring, \(b = 1.523\) mm

4. RESULTS AND DISCUSSION

In this study, a three dimensional solid model of connecting rod is modeled in creo parametric software and introduced to the ANSYS software.

4.1 Meshing of Connecting rod

A characteristic of quadrilateral/hexahedral elements that might make them more economical in some situations is that they permit a much larger aspect ratio than triangular/tetrahedral cells. A large aspect ratio in a triangular/tetrahedral cell will affect invariably the skewness of the cell, which is undesirable as it may hamper the accuracy and convergence. For relatively complex geometry triangular/tetrahedral mesh is preferred (Fig. 6).

Boundary Condition: Big end is fixed (Fig. 8). A pressure of six MPa is applied at the Small end (Fig. 7). The big end of the connecting rod is fixed and small end is applied with a pressure of 6 MPa. The Fig. 9 shows the total deformation due to tensile load is Maximum deformation: 0.012814 mm.

The Fig. 10 shows the total deformation due to compressive load near the small is visible. The value of minimum deformation could be read as 0.006685 and closer to 0.0042713 and the deformation is not affecting the C.rod very closer to small end is observed.

![Fig. 6 Meshing of connecting rod](image)
4.2 Von Misses stress due to tensile load

Maximum Von Misses stress: 103.76 MPa, Minimum Von Misses stress: 0.00066257 MPa. The Fig. 11 and 12 shows the maximum Von misses stress developed due to tensile and compressive load application. For the chosen dimension for analysis the stresses developed is very well within the acceptable range and fatigue life due to tensile load is not affecting much. The Fig. 13 & 14 indicate the fatigue life and safety factor for tensile load and Fig. 15 & 16 represent the fatigue life and safety factor for the application of compressive load.

The dimensions considered for the small end is acceptable and it is not alarming to add more factor of safety either for tensile load or for compressive load.

4.3 Von Misses stress due to compressive load

Maximum compressive stress: 103.76 Mpa; Minimum compressive stress: 0.00066257 Mpa. Von misses stress analysis helps to understand where stress concentration is more and what design consideration to be made is clearly seen in the diagram. But this analysis is self explanatory and no appreciable stresses are developed in the connecting rod. The following data shows the results from the analysis.

4.4 Fatigue analysis of connecting rod

Fatigue Life due to tensile load:
Minimum Life: 3.4388e5;
Maximum Life: 1e6
Factor of safety due to tensile load:
Maximum FOS: 0
Minimum FOS: 15

Fatigue Life due to compressive load:
Minimum Life: 3.4388e5
Maximum Life: 1e6

Factor of safety due to compressive load:
Maximum FOS: 0
Minimum FOS: 15

4.5 Model of Piston in ANSYS

A gas pressure of 6mpa was considered to apply on the piston head. The Fig. 17 to 20 indicates the creation of the model and consideration of boundary conditions.

Structural analysis by Von misses stress by applying gas load on the piston shows the pressure effecting area of the piston top and stress is not much and deformation due to stress is at all noteworthy from Fig. 21 and 22. Gas pressure of 6 MPa is at the top end of the Piston.

Von Misses stress due to gas pressure
Maximum tensile stress: 116.06 MPa
Minimum tensile stress: 1.9398e-9
Total deformation due to gas pressure:
Maximum deformation: 0.024017 mm
Minimum deformation: 0 mm

4.6 Thermal Analysis

Boundary condition:
Maximum temperature at Piston head of 400°C
Temperature distribution:
Min temperature of 22°c
Total heat flux:
Maximum heat flux: 6.0528 W/mm²
Minimum heat Flux: 1.7484e-10 W/mm².

The Fig. 23-25 show the thermal analysis of a piston with gas load applied on top of the piston and heat flux and thermal fatigue analysis from Fig. 26 and 27 are showing very much less values and very safe to operate with the present dimensions considered for piston. The piston will with stand the heat and further it will not deform or get damaged or melt during working.

4.7 Fatigue Analysis

Fatigue Life of Piston:
Minimum Life: 1.8742e5
Maximum Life: 1e6

5. CONCLUSION

From the following data it is evident from the analysis conclude that

- Tensile strength of Al alloy is 280 MPa
- Compressive strength of Al alloy is 280 MPa
- Maximum Tensile stress of connecting rod is 103.76 MPa
- Maximum Tensile stress of Piston is 116.06 MPa
- Maximum temperature in the engine is 400°C and Melting point of alloy is 671°C.
- Working stress is less than the allowable stress
- Maximum working temperature is less than the melting temperature.
- Therefore, the two main components of I.C. engine namely Piston and Connecting rod is safe as well as economical and suits for the engine of two wheelers.

REFERENCES