STUDY AND ANALYSIS OF CONNECTING ROD PARAMETERS USING ANSYS

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ABSTRACT

The present study conducted to analysis of the connecting rod parameters using ANSYS software. The main objective of this study has to investigate the stresses induced in connecting rod. This can be achieved by changing such design parameter in the existing design of single cylinder 4 stroke petrol engine by using FEA (Finite element analysis) for the study. During analysis of the parameters of connecting rod, it can be observed that several stresses are working during load condition of rod.

Key words: Analysis, Connecting Rod, ANSYS

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1. INTRODUCTION

Connecting rods are mostly used in variety of engines such as, in-line engines, V engines, opposed cylinder engines, radial engines and oppose-piston engines. A connecting rod consists of a pin-end, a shank, and a Pin-end and crank-end pin holes at the upper and lower both ends are machined to permit accurate fitting of bearings. These holes must be parallel. The upper end of the connecting rod is attached to the piston by the piston pin. If the piston pin is locked in the piston pin bosses in the piston and the connecting rod, the upper hole of the connecting rod will have a solid bearing of bronze or other same material. As the lower end of the connecting rod rotate with the crankshaft, the upper end is forced to turn back and forth on the piston pin. Although this crusade is rebuff, the bearing bushing is essential because of the high pressure and temperatures. The lower hole in the connecting rod is crack to permit it to be fixed around the crankshaft. The bottom part is made of the same material as the rod and is attached by two bolts. The surface that tolerate on the crankshaft is generally a bearing material in the form of a distinct crack shell. The two parts of the bearing are maintaining in the rod and cap by dowel pins, forecasts, or short brass screws. Split bearings may be of the accuracy or semi accuracy type.
The connecting rod in I.C. engines are subjected to high cyclic loads comprised of dynamic tensile and compressive load. Its primary function is to transmit the push and pull from the piston pin to the crank pin and thus convert the reciprocating motion of the piston into the rotary motion of the crank. It consists of a long shank small end and a big end. The cross section of the shank may be rectangular, circular, tubular, I-section or H-section. Commonly the circular section is used for low speed engine while I-section is preferred for high speed engine. Stress analysis of connection rod by finite element method using ANSYS 16.2 work bench software. And analyzed that the stress induced in the piston end of the connecting rod are greater than the stresses induced at the crank end. So that piston end more fractures compare to crank end.

![Design of Connecting Rod used in I.C Engine](image)

**Figure 1** Design of Connecting Rod used in I.C Engine

The automobile engine connecting rod is a high volume production, grave component. It connects reciprocating piston to rotating crankshaft, conveying the force of the piston to the crankshaft. Every vehicle that uses an I. C. engine requires as a minimum one connecting rod depending upon the number of cylinders in the engine. Connecting rods for automotive uses are normally manufactured by forging from either wrought steel or powdered metal. They could also be cast. However, castings could have blow-holes which are detrimental from durability and fatigue points of view.

**2. OBJECTIVE**
1. Study of connecting rod.
2. Geometry design through CAD Tool solid work.
3. Stress analysis through ANSYS.
3. ANALYTICAL DESIGN OF CONNECTING ROD

3.1. Dimension of I-Section of the Connecting Rod

Let us consider an I-section of the connecting rod, with the following proportions:

- Flange and web thickness of the section = \(t\)
- Width of the section, \(B = 4t\) and
- Depth or height of the section, \(H = 5t\)

The connecting rod should be equally strong in buckling about both the axes. We know that in order to have a connecting rod equally strong about both the axes,

\[
I_{xx} = 4I_{yy}
\]

Where,

- \(I_{xx}\) = Moment of inertia of the section about X-axis, and
- \(I_{yy}\) = Moment of inertia of the section about Y-axis.

In actual practice, \(I_{xx}\) is kept slightly less than 4 \(I_{yy}\). It is usually taken between 3 and 3.5 and the connecting rod is designed for buckling about X-axis. Now, for the section as shown in area of the section,

\[
A = 2(4t \times t) + 3t \times t = 11t^2
\]
\[
I_{xx} = \frac{1}{12}[4t(5t)^3 - 3t^3] = \frac{419}{12}t^4
\]
And \(I_{yy} = 2 \times \frac{1}{12} \times t(4t)^3 + \frac{1}{12} \times 3t \times t^3 = \frac{131}{12}t^4\)
\[
I_{xx}/I_{yy} = \frac{419/12}{12/131} = 3.2
\]
Since $I_{xx}/I_{yy} = 3.2$, therefore the section chosen is quite satisfactory.

Now let us find the dimensions of this I-section. Since the connecting rod is designed by taking the force on the connecting rod ($F_c$) equal to the maximum force on the piston ($F_L$) due to gas Pressure, therefore,

$$F_c = F_L = \frac{\pi D^2}{4} \times P$$

$$= \frac{\pi 50^2}{4} \times 15.48$$

$$= 30394.9 \text{ N}$$

We know that the connecting rod is designed for buckling about X-axis (i.e. in the plane of Motion of the connecting rod) assuming both ends hinged. Since a factor of safety is given as 5, therefore the buckling load,

$$W_B = F_C \times F.S. = 30394.9 \times 5 = 151974.5 \text{ N}$$

We know that radius of gyration of the section about X-Axis

$$K = \sqrt{\frac{l_{xx}}{A}}$$

$$= 1.78 \text{ t}$$

Length of crank,

$$r = \text{Stroke of piston }/2 = \frac{56}{2} = 28 \text{ mm}$$

Length of the connecting rod,

$$L = 155 \text{ mm.}$$

Equivalent length of the connecting rod for both ends hinged,

$$L = l = 155 \text{ mm}$$

Now according to Rankine’s formula, we know that buckling load

$$151974.5 = \frac{415 \times 11 t^2}{1 + \frac{1}{1800 \times 1.78^2}}$$

$$366.2 t^2 + 2653.87 = 11 t^2$$

$$t^4 - 33.29 t^2 - 241.26 = 0$$

$$t = 6.27 \text{ or say } 7 \text{ mm}$$

Thus, the dimensions of I-section of the connecting rod are:

Thickness of flange and web of the section

$$t = 7 \text{ mm}$$

Width of the section, $B = 4 \times 4 = 28 \text{ mm and}$

Depth or height of the section,

$$H = 5 \times 7 = 35 \text{ mm}$$

These dimensions are at the middle of the connecting rod. The width ($B$) is kept constant throughout the length of the rod, but the depth ($H$) varies. The depth near the big end or crank end is kept as $1.1H$ to $1.25H$ and the depth near the small end or piston end is kept as $0.75H$ to $0.9H$. Let us take

Depth near the big end,

$$H_1 = 1.2H = 1.2 \times 35 = 42 \text{ mm}$$

And depth near the small end,

$$H_2 = 0.85H = 0.85 \times 35 = 29.75 \text{ say } 30 \text{ mm}$$

Dimensions of the section near the big end.
= 42 mm × 28 mm and
Dimensions of the section near the small end
= 30 mm × 28 mm

### 3.2. Dimensions of the Crankpin or the Big End Bearing and Piston Pin or Small End Bearing

Let,

\[ d_c \] = Diameter of the crankpin or big end bearing,

\[ l_c \] = Length of the crankpin or big end bearing = 1.3 \( d_c \)

\[ p_{bc} \] = Bearing pressure = 10 N/mm²

We know that load on the crankpin or big end bearing
= Projected area × bearing pressure
= \( d_c \times l_c \times p_{bc} = d_c \times 1.3 \times 10 = 13 \left( d_c \right)^2 \)

Since the crankpin or the big end bearing is designed for the maximum gas force (FL), therefore, equating the load on the crankpin or big end bearing to the maximum gas force, \( i.e. \)

\[ 13 \left( d_c \right)^2 = FL = 30394.9 \text{ N} \]

\[ (d_c)^2 = 30394.9 / 13 \]

\[ d_c = 48.35 \text{ say 49 mm and} \]

\[ l_c = 1.3 \ d_c = 1.3 \times 49 = 62.85 \text{ say 63 mm} \]

The big end has removable precision bearing shells of brass or bronze or steel with a thin lining (1mm or less) of bearing metal such as Babbit.

Again,

Let \( d_p \) = Diameter of the piston pin or small end bearing,

\[ l_p \] = Length of the piston pin or small end bearing = 2\( d_p \)

\[ p_{bp} \] = Bearing pressure = 15 N/mm²

We know that the load on the piston pin or small end bearing
= Project area × bearing pressure
= \( d_p \times l_p \times p_{bp} = d_p \times 2 \ d_p \times 15 = 30 \left( d_p \right)^2 \)

Since the piston pin or the small end bearing is designed for the maximum gas force (FL), therefore, equating the load on the piston pin or the small end bearing to the maximum gas force, \( i.e. \)

\[ 30 \left( d_p \right)^2 = 30394.9 \text{ N} \]

\[ (d_p)^2 = 30394.9 / 30 \]

\[ d_p = 31.83 \text{ say 32 mm and} \]

\[ l_p = 2 \ d_p = 2 \times 32 = 64 \text{ mm} \]

The small end bearing is usually a phosphor bronze bush of about 3 mm thickness

### 3.3. Size of Bolts for Securing the Big End Cap

Let,

\[ d_{cb} \] = Core diameter of the bolts,

\[ \sigma_t \] = Allowable tensile stress for the material of the bolts
= 60 N/mm² (assume) and

\[ n_b \] = Number of bolts. Generally two bolts are used.
We know that force on the bolts
\[ \frac{\pi}{4} (d_{cb})^2 \sigma_t \times n_b : (\sigma_t = 60, n_b = 2) \]
\[ = 94.26 (d_{cb})^2 \]
We know that inertia force of the reciprocating parts,
\[ F_I = M_R \times \omega^2 \times r \left( \cos \theta + \frac{\cos 2\theta}{l/r} \right) \]
We also know that at top dead center on the exhaust stroke, \((\theta = 0)\).
\[ F_I = M_R \times \omega^2 \times r \left( 1 + \frac{r}{l} \right) \]
\[ = 0.1617 \times \left( \frac{2\pi \times 7500}{60} \right)^2 \times 0.028 \left( 1 + \frac{0.028}{0.155} \right) N \]
\[ = 3297.36 N \]
Equating the inertia force to the force on the bolts, we have
\[ 3297.36 = 94.26 (d_{cb})^2 \]
\[ d_{cb} = 5.9 \text{ mm or say } 6 \text{ mm and} \]
Nominal diameter of the bolt,
\[ d_p = \frac{d_{cb}}{0.84} \]
\[ = 7.14 \text{ say } 8 \text{ mm} \]

### 3.4. Thickness of the Big End Cap

Let,
- \( t_c \) = Thickness of the big end cap,
- \( b_c \) = Width of the big end cap. It is taken equal to the length of the crankpin or big end bearing \((l_c) = 63 \text{ mm (calculated above)}\)
- \( \sigma_b \) = Allowable bending stress for the material of the cap
  = 80 N/mm² ... (assume)
  Maximum bending moment is taken as
  \[ M_C = F_I \times x / 6 \]
  Where
  \( x \) = Distance between the bolt centers
  = Dia. of crankpin or big end bearing + 2 × Thickness of bearing liner + Nominal dia. of bolt + Clearance
  = \((d_c + 2 \times 3 + d_b + 3) \text{ mm} = 49 + 6 + 8 + 3 = 66 \text{ mm} \)
  Maximum bending moment acting on the cap,
  \[ M_C = 36270.96 \text{ N-mm} \]
  Section modulus for the cap
  \[ Z_C = \frac{b_c \times t_c^2}{6} \]
  \[ = \frac{63 \times t_c^2}{6} \]
  \[ = 10.5 \times t_c^2 \]
  We know that bending stress \(( \sigma_b )\),
  \[ 80 = \frac{M_C}{Z_C} \]
  \[ = 36270.96 / 10.5 \times t_c^2 \]
Let us now check the design for the induced bending stress due to inertia bending forces on the Connecting rod (i.e. whipping stress).

We know that mass of the connecting rod per metre length,

\[ M_1 = \text{Volume} \times \rho = \text{Area} \times \text{length} \times \rho \]

\[ = A \times l \times \rho = 11t^2 \times l \times \rho \quad (Q.A = 11t^2) \]

\[ = 11(0.007)2 \quad (0.155) \quad 8000 = 0.66 \text{ kg} \quad (\rho = 8000 \text{ kg/m}^3 \text{ (given)}) \]

Maximum bending moment

\[ M_{\text{max}} = m \omega^2 r \times \frac{1}{9\sqrt{3}} \]

\[ = 0.668 \left( \frac{2\pi \times 7500}{60} \right)^2 \times 0.028 \left( \frac{0.155)^2}{9\sqrt{3}} \right) \]

\[ = 17.78 \text{ N-m} \]

\[ = 17781 \text{ N-mm} \]

And section modulus,

\[ Z_{xx} = \frac{I_{xx}}{t} \]

\[ = \frac{419 t^4}{12} \times \frac{2}{5t} \]

\[ = 13.97 t^3 = 4792 \text{ mm}^3 \]

Maximum bending stress (induced) due to inertia bending forces or whipping stress,

\[ \sigma_{b(max)} = \frac{M_{\text{max}}}{Z_{xx}} \]

\[ = 17781/4792 \]

\[ = 3.71 \text{ N/mm}^2 \]

4. FINITE ELEMENT ANALYSIS OF CONNECTING ROD

**Figure:** Solid Model of Connecting Rod

**Figure:** Meshes Model of Connecting Rod
Study and Analysis of Connecting Rod Parameters using Ansys

5. LOAD DISTRIBUTION OF CONNECTING ROD

Table Loading Data

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Crank end loading</th>
<th>Pin end loading</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>max</td>
<td>min</td>
</tr>
<tr>
<td>Load magnitude</td>
<td>30394.9 N</td>
<td>30394.9 N</td>
</tr>
<tr>
<td>Maximum shear stress</td>
<td>143.36</td>
<td>0.17154</td>
</tr>
<tr>
<td>Equivalent shear stress</td>
<td>271.84</td>
<td>0.3253</td>
</tr>
</tbody>
</table>

Figure: Maximum Shear Stress

Figure: Equivalent Stress
6. CONCLUSION

- Above study gives the idea about designing of the connecting rod. It explains about the various stresses to be considered while designing the connecting rod and different materials used and comparing the result of all material.
- The Finite element Analysis of the connecting rod is done in ANSYS Workbench 16.2 considering all loading condition.
- The maximum pressure stress was obtained between pin end and rod of connecting rod.
- The maximum shear stress was obtained in pin end. So the chance of failure of the connecting rod may be fitted section of both end but at piston end more chance of failure compare to at crank end.

7. FUTURE WORK AND SCOPE

Further analysis can be done by choosing different parameters for the connecting rod. Maximum stress concentration at the fillet of big and small end can be changed by changing the material. Dynamic analysis of connecting rod can be done in future through ANSYS. Other parameters for the failure can be considered.

REFERENCES

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