Design and Analysis of Connecting ROD of 800cc Car

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Abstract:
Connecting rod basically connects the piston to the crankshaft which transmits power of the combustion from the combustion chamber crank shaft. Designing of connecting rod with standard dimensions based on Factor of Safety by theoretical analysis and design simulation has been done by using software CATIA and ANSYS & SOLID WORKS. Experimental work has been performed mechanical workshop and compares the results. The parameters such as stress, strain, deformation, fatigue analysis and working cycle has been analysed by taking the material Forged steel and Cast iron.

Keywords: Connecting Rod, Analysis of Connecting Rod, Four stroke engine Connecting Rod, Forge steel Connecting Rod, Design and Analysis of Connecting Rod.

I. INTRODUCTION:
The connecting rod is a major link inside a combustion engine. It connects piston to the crankshaft and is responsible for transferring power from the piston to the crankshaft & sending it to the transmission. There are different type of materials & production methods used in the design in connecting rod. The most common type of connecting rods are steel & aluminium. The most common type of manufacturing process are casting, forging, powder metallurgy. Connecting rods are widely used in variety of engines such as, in-line engine, V-engine, opposed cylinder engines, radial engines & opposed piston engines. A connecting rod consists of a pin end, a shank section, & a crank end. The small end attached to the piston pin or gudgeon pin which is currently most often press fit into the con rod but can swivel the piston, a “floating wrist pin design”. The big end of the rod is fabricated as a unit & cut in to two to establish precision fit around the big end bearing shell. The connecting rod is under tremendous stress from the reciprocating load represented by the piston, actually stretching & being compressed with every rotation & load increase with the third power with increasing engine speed. Connecting rod, automotive should be lighter & should consume less fuel & at the same time they should provide comfort & safety to passengers, that unfortunately leads to increase in weight of vehicle. This tendency in vehicle construction led the invention & implementation of quite new materials which are light & meet design requirements. Lighter connecting rods help to decrease load caused by inertia force in engine as it does not require big balancing weight on crankshaft. Geometrically it can be seen that longer connecting rod will reduce the amount of sideway force & therefore lead to longer engine life. Honda Company had already started the manufacturing of aluminium connecting rods reinforced with steel continuous fibres. Ford 4.6 litter engine & Chrysler 2.0 litter engine have connecting rod made by powder metallurgy which allows more precise control of size & weight with less machining & less excess mass to be machined off for balancing.

Nomenclature

t=Thickness of flange
B =Width of the flange section
H=Height of the flange section
H1 =Height of Smaller Head Section
H2=Height Of Bigger Head Section
A=Area of the section
Ixx=Moment of inertia about X axis
Iyy=Moment of inertia about Y axis
D =cylinder’s Bore diameter
r =crank radius
L = length of connecting rod
P=Mean effective pressure
F=Net Force
Fp=Force act on piston
Ft = Inertia force of connecting rod
Fc=Force act on connecting rod
Fn=Normal force act on small end
Ft =Tangential force
Θ=Angle between Connecting Rod & Piston
β = Angle between Connecting Rod & Piston
σ c =Compressive yield stress
Wcr= critical load of buckling
m p =mass of the reciprocating parts
m c =mass of the connecting rod
FOS = Factor of safety

II. THEORETICAL ANALYSIS:
The theoretical analysis of connecting rod for 800CC PETROL MARUTI SUZUKI OMNI has been done using the concept of buckling load in two dimensional sections as 1 section, considering the Inertia force due reciprocating parts as well as the weight of the connecting rod has been taken into the account. The factor of safety is very much effect has been also considered to get for safe design. The buckling of connecting rod in two different planes of motion. The buckling of connecting rod in the plane of motion, the ends of connecting rod is hinged in the crank pin & piston pin. Therefore, for buckling about the X-X axis, the end fixity co-efficient (n) is one. The buckling of connecting rod in a plane perpendicular to the plane of the motion is the ends of the connecting rod are fixed due to constraining effect of bearings at the crank pin & piston pin. Therefore, for buckling about the Y-Y axis the end fixity co-efficient (n) is four. A connecting rod is a machine member which is subjected to alternating direct compressive and tensile force. Since the compressive force are much higher than the tensile forces.
According to Rankine’s formula
\[ F = \left[ \frac{\sigma \times A}{1 + a \left( \frac{L}{K_x} \right)^2} \right] \]
Crippling load about X-axis
\[ F = \left[ \frac{\sigma \times A}{1 + a \left( \frac{L}{K_x} \right)^2} \right] \quad (L=L) \quad \text{(for both end hinged)} \]
Crippling load about Y-axis
\[ F = \left[ \frac{\sigma \times A}{1 + a \left( \frac{L}{2K_y} \right)^2} \right] \quad [L=L/2] \quad \text{(for both hinged)} \]

By equating the Crippling load about X-axis and Y-axis,
\[ \frac{I_{xx}}{I_{yy}} = 4 \]

### III. SELECTION OF THE DIMENSION OF CONNECTING ROD:

Thickness of flange & web of the section=t

<table>
<thead>
<tr>
<th>B</th>
<th>H</th>
<th>A</th>
<th>Xc</th>
<th>Yc</th>
<th>I_{xx}</th>
<th>I_{yy}</th>
<th>I_{xx}/I_{yy}</th>
<th>Kx</th>
</tr>
</thead>
<tbody>
<tr>
<td>3t</td>
<td>5t</td>
<td>9t^2</td>
<td>1.5t</td>
<td>2.5t</td>
<td>26.75t^4</td>
<td>4.75t^4</td>
<td>5.64</td>
<td>1.72t</td>
</tr>
<tr>
<td>4t</td>
<td>5t</td>
<td>11t^2</td>
<td>2t</td>
<td>2.5t</td>
<td>34.91t^4</td>
<td>10.91t^4</td>
<td>3.2</td>
<td>1.78t</td>
</tr>
<tr>
<td>4t</td>
<td>6t</td>
<td>12t^2</td>
<td>2t</td>
<td>3t</td>
<td>56t^4</td>
<td>11t^4</td>
<td>5.09</td>
<td>2.16t</td>
</tr>
</tbody>
</table>

From above table it is observed that
Area of the section A=11t^2
Moment of inertia about x-axis I_{xx}
\[ I_{xx} = 4t \times (5t)^2 - 3t \times (3t)^2/12 = 34.91t^4 \]

Moment of inertia about y-axis I_{yy}
\[ I_{yy} = 2t \times \left[ \frac{1/12 \times t \times (4t)^2}{1} + \frac{1/12 \times (3t)^2}{3} \right] = 10.91t^4 \]
So, \[ \frac{I_{xx}}{I_{yy}} = 3.2 \] (since \[ \frac{I_{xx}}{I_{yy}} < 4 \) , design is safe

For the value of \[ \frac{I_{xx}}{I_{yy}}= 3.2 \] which is less than 4 so the design becomes safe. For this condition the width and the height of the section will be:

- Width of the section B = 4t
- Height of the section H = 5t

### SPECIFICATION OF 800CC PETROL MARUTI SUZUKI OMNI:

**Material Selected:** Forged Carbon Steel

1) Engine type: - Water cooled, 4-stroke.
2) Mass of the reciprocating part = 0.221 kg
3) Mass of connecting rod=0.361 kg, Displacement = 800 cc.
4) Maximum power = 35 bhp. @ 5000 r.p.m.
5) Maximum torque = 5.9 Kgm. @ 3000 r.p.m.
6) Compression ratio = 9.35/1 = 9.35
7) Density of petrol at 288 k = 737.22 \times 10^{-9} kg/mm^3.
8) Molecular weight (M) = 114.228 g/mole.
9) Ideal gas constant (R) = 8.314 kj/mole.k.

### CALCULATION OF NET FORCE:

The Mean effective pressure has been calculated and computed the net force and piston force acting on piston.

Mean Effective Pressure P=17.68 N/mm^2
Length of connecting Rod , L = 2 \times 74.4 mm = 147.8 mm.
Net force acting, (F = F_p - F_i)

- Piston Force F_p = (\pi d^2/4) \times p = 63798.6 N
- Reciprocating mass F_i = m_rw^2r (cos\theta + cos2\theta/n)

\[ m_r = \text{mass of the reciprocating parts} = 0.221 \text{kg}, \]
\[ r = \text{crank radius} \]
\[ w = \text{stroke of the piston} / 2 = 74.4 / 2 = 37.7 \text{mm} \]
\[ \theta = 0^\circ, \text{considering that crank is at the T.D.C. position}. \]

Buckling load (F) = F_p - F_i = 63798.6-2161.47 = 61637.12 N

Taking FOS =1

<table>
<thead>
<tr>
<th>( \theta =0 )</th>
<th>( \theta =50 )</th>
<th>( \beta=0 )</th>
<th>( \beta=30 )</th>
<th>( \theta =\beta=0 )</th>
<th>( \theta =50, \beta=30 )</th>
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</thead>
<tbody>
<tr>
<td>F_i</td>
<td>8986.04 N</td>
<td>4318N</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>F_c</td>
<td>54812N</td>
<td>68681N</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>F_t</td>
<td>0</td>
<td>67638N</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

\[ W_{cr} = (F_p-F_i) \times \text{FOS} \]

Where
\[ F_i = m_rw^2r(\cos\theta + \cos2\theta/n) \]
\[ m_r = \text{mass of the reciprocating parts} = 0.221 \text{kg} \]
\[ r = \text{crank radius} \]
\[ w = \text{stroke of the piston} / 2 = 74.4 / 2 = 37.7 \text{mm} \]
\[ \theta = 0^\circ, \text{considering that crank is at the T.D.C. position}. \]

Buckling load (F) = F_p - F_i = 63798.6-2161.47 = 61637.12 N
Taking FOS =1

### IV. DESIGN CALCULATION OF CARBON STEEL

Using the Rankine Formula for critical load of buckling as
\[ W_{cr} = \frac{\sigma \times A}{1 + a \left( \frac{L}{K_x} \right)^2} \]

Only piston load taken i.e W_{ce} = F_p \times \text{FOS}
Taking FOS =1, \( \sigma = 415 \text{ Mpa}, \quad a = 0.0002 \),
The value of thickness of flange obtained as \( t =4 \text{mm} \)
Piston load and Reciprocating mass taken i.e

\[ W_{cr} = (F_p - F_i) \times \text{FOS} \]
The value of thickness of flange obtained as \( t = 3.61\text{mm} \)

Piston load, reciprocating mass along with include mass of connecting rod taken \( \text{W}_{\text{cr}}=(F_{\text{p}}-F_{\text{i}}) \times \text{FOS} \)

Where \( F_{\text{i}} = (m_{\text{r}}+m_{\text{c}}/3)w^{2}(\cos\theta + \cos2\theta/n) \)

\( m_{\text{r}}+m_{\text{c}}/3 \) = mass of the reciprocating parts = 0.341 kg

Taking FOS = 1

The value of thickness of flange obtained as \( t = 3.7\text{mm} \)

Table shows the value of components of connecting ROD VS FOS:

<table>
<thead>
<tr>
<th>FOS Range</th>
<th>( F_{\text{p}} = \text{FOS} \times F_{\text{ic}} )</th>
<th>( F_{\text{ic}} )</th>
<th>Width(mm)</th>
<th>Hc of section(mm)</th>
<th>Area ( A ) (mm²)</th>
<th>Small end Hc(mm)</th>
<th>Small end range(mm)</th>
<th>Big end Hc(mm)</th>
<th>Big end range(mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>( t \leq 1 )</td>
<td>( t &gt; 1 )</td>
<td>( t &gt; 1 )</td>
<td>( t' \leq 1 )</td>
<td>( t' &gt; 1 )</td>
<td>( t' &gt; 1 )</td>
<td>| ( t' &gt; 1 )</td>
<td>| ( t' &gt; 1 )</td>
<td>| ( t' &gt; 1 )</td>
<td>| ( t' &gt; 1 )</td>
</tr>
<tr>
<td>1</td>
<td>5</td>
<td>7</td>
<td>7</td>
<td>7</td>
<td>32</td>
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<td>35</td>
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<td>9</td>
<td>9</td>
<td>9</td>
<td>36</td>
<td>45</td>
<td>0.0009</td>
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<td>7</td>
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<td>10</td>
<td>10</td>
<td>40</td>
<td>50</td>
<td>0.0011</td>
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<td>45</td>
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<td>10</td>
<td>11</td>
<td>11</td>
<td>44</td>
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<td>0.0025</td>
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<td>0.0028</td>
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<td>13</td>
<td>13</td>
<td>17</td>
<td>68</td>
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<td>0.0032</td>
<td>63.75</td>
<td>76.5</td>
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<td>13</td>
<td>14</td>
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<td>72</td>
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<td>0.0036</td>
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<td>85.5</td>
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<td>0.0044</td>
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<td>90</td>
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<td>14</td>
<td>21</td>
<td>84</td>
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<td>0.0049</td>
<td>78.75</td>
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<tr>
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<td>11</td>
<td>15</td>
<td>15</td>
<td>22</td>
<td>88</td>
<td>110</td>
<td>0.0053</td>
<td>82.5</td>
<td>99</td>
</tr>
</tbody>
</table>

From the above table it is found that the value of FOS lies between 4 to 6 for safe connecting rod design. The thickness of the flange (\( t \)) = 7.7 mm = 8mm. which is very much significant

V. DIMENSION OF BIG END AND SMALL END BEARING:

Bearing pressure range taken as 7 N/mm² to 12.5 N/mm² and \( l_{c}/d_{c} \) ratio taken as 1.25 to 1.5.

For this analysis , Bearing pressure (\( P_{b,c} \)) = 12.5 N/mm² and \( l_{c}/d_{c} \) ratio taken as 1.5 to 1.5.

Load on Big end bearing = \( d_{c} \times l_{c} \times (P_{b,c})_c = d_{c} \times 1.5. \)

\( d_{c} = 58.3 \text{ mm} \) and \( l_{c} = 1.5 \times d_{c} = 87.5 \text{ mm} \)

Bearing pressure range taken as 12.5 N/mm² to 17 N/mm² and \( l_{p}/d_{p} \) ratio taken as 1.5 to 2.4.

For this analysis, Bearing pressure (\( P_{b,p} \)) = 17N/mm² and \( l_{p}/d_{p} \) = 2.4

Load on Small end bearing = \( d_{p} \times l_{p} \times (P_{b,p})_p = d_{p} \times 2.4d_{p} \times 17 = F_{p} \)

\( d_{p} = 39.5 \text{ mm} \) and \( l_{p} = 2.4d_{p} = 95\text{mm}. \)

Comparison of theoretical and actual dimension of components of connecting Rod:

<table>
<thead>
<tr>
<th>Components</th>
<th>Theoretical Dimension (in mm)</th>
<th>Actual Dimension(mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Thickness of connecting rod</td>
<td>7.7</td>
<td>8.1</td>
</tr>
<tr>
<td>length of connecting rod</td>
<td>147.8</td>
<td>149</td>
</tr>
<tr>
<td>Width Of Section(B)</td>
<td>32</td>
<td>32</td>
</tr>
<tr>
<td>Height of Section(H)</td>
<td>40</td>
<td>40</td>
</tr>
<tr>
<td>Height of Smaller Head Section(H1)</td>
<td>30</td>
<td>25</td>
</tr>
<tr>
<td>Height Of Bigger Head Section(H2)</td>
<td>44</td>
<td>38</td>
</tr>
<tr>
<td>Outer Dia Of Big Head</td>
<td>58.3</td>
<td>54.2</td>
</tr>
<tr>
<td>Outer Dia Of Small Head</td>
<td>39.5</td>
<td>27.6</td>
</tr>
</tbody>
</table>

DESIGN of Components & Assembly by CATIA:

DESIGN ANALYSIS OF CONNECTING ROD

Taking Cast Iron (C.I) material of connecting rod at static condition considering the force acting on small end during combustion results Buckling on I section
Result: Figure.1. Buckling on I section. Figure.2. Analytical Result Figure.3. Experimental Result

Taking Forged steel material of connecting rod at static condition considering the force acting on small end during expansion result: Deformation of Big End.

Result: Figure.1. Deformation of Big End. Figure.2. Analytical Result Figure.3. Experimental Result

Taking Forged steel material of connecting rod at Equilibrium static condition at TDC considering the distributed forces are acting on slider crank mechanism result: No Deformation

Force calculation by follow Slider crank mechanism:
\[
\theta = 0 \text{ degree}, \beta = 0 \text{ degree} \\
F_p = 63798 \text{N}, \ F_t = 8986.04 \text{N}, \ F_c = 54812 \text{N}, \ Fn = 0 \text{N}, \ Ft = 0 \text{N}
\]

Result: No deformation occurs.

Taking Forged steel material of connecting rod at Equilibrium static condition, considering the distributed forces are acting on slider crank mechanism result: Not very much deformation occur. High stress position is Small end due to Maximum piston force. Force calculation by follow Slider crank mechanism:
\[
\theta = 50 \text{ degree}, \beta = 30 \text{ degree} \\
F_p = 63798 \text{N}, \ F_t = 4318 \text{N}, \ F_c = 68681 \text{N}, \ Fn = 34340 \text{N}, \ Ft = 67638 \text{N}
\]

Result: Not very much deformation occur. High stress position is Small end due to Maximum piston force.

VI. CONCLUSION:

By checking and comparing the results theoretically and experimentally. It has been observed that the results are very much feasible. In design aspects nature of loading influence the buckling of the connecting rod which has been mainly mentioned in this work. Factor of safety also effects and plays big role for designing of connecting rod depending the physical and mechanical properties of material. In this work it has been seen that the range of factor of safety becomes 4 to 6.

Acknowledgement
The authors are very much grateful to Ashis Kumar Pradhan, Anik Pal, Pintu Kumar and Sourav Khatua for designing and analysis of the work.
VII. REFERENCE


