Design and Analysis on Modified Design of Connecting Rod

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ABSTRACT

Connecting rod is a shaft member in an IC engine which is used to connect piston and crank and is used to create reciprocating motion to the engine that functions of transmitting the thrust from piston pin to crank pin. Together with the crank, it forms a simple mechanism that converts reciprocating motion of the piston into rotating motion. The connecting rods are generally made of steel alloys and cast iron. In this work the existing design of connecting rod is modified. And also the modeling of connecting rod and analysis of connecting rod is done through solidworks 2014 and ansys 15.0. FEA analysis was done out by software considering the material as cast iron. With the help of analyzing, the useful factors like displacement, Von mises stress and modal frequency and its deformation are calculated.

Keywords: buckling load, shank web thickness, piston, crankshaft, workbench, solidworks

I. INTRODUCTION

A connecting rod is a shaft member which is used to connect a piston to a crank or crankshaft in a reciprocating engine. With the crank, it forms a very simple mechanism that changes reciprocating motion into rotating motion. A connecting rod can also convert rotating motion into reciprocating motion. Former mechanisms, such as the chain, can only convey pulling motion. In a some two-stroke engines, the connecting rod is only essential to push. Now-a-days, the connecting rod is best recognized through its use in internal combustion piston engines, such as automobile engines. These are of a particularly different in design from earlier models of connecting rod used in steam engines and steam locomotives.

In current automotive IC engines, the connecting rods are generally made of steel for production engines. These can also be made of T6-2024 and T651-7075 aluminum alloys (for lightness and the ability to captivate high impact at the high expenditure of durability) or titanium (for a combination of lightness with strength and also at higher cost) for high-performance engines, or of cast iron for applications such as motor scooters. They are less rigidly fixed at one or other end, so that the angle between the piston and the connecting rod can change as the rod moves up and down and rotates around the crankshaft. The small end of connecting rod is attached to the piston pin. The big end is connected to the crankpin on the crank throw. Engines running on replaceable bearing shells which can be reachable through the connecting rod bolts that holds the bearing “cap” onto the big end. Mostly, small two-stroke engines and few single cylinder four-stroke engines avoids the need of a pumped lubrication system by using a rolling-element bearing. However, this needs the crankshaft to be constrained apart and then backed together in order to replace a connecting rod.

A main cause of engine’s wear is the sideway force employed on the piston via the connecting rod by the crankshaft, which naturally wears the cylinder into an oval cross-section rather than circular, making it not possible for the piston rings to correct the seal against cylinder walls. Yet, to a given engine block, the sum of the length of the connecting rod and the piston stroke will be a fixed number, which is determined by the fixed distance between the crankshaft axis and to the top of the cylinder block.

When connecting rod is under great stress, which is developed from the reciprocating load given by the piston, the connecting is actually being stretched and compressed with each rotation. And also the load will be increased as the square of the engine speed increased. Failure of a connecting rod, usually addressed as throwing a rod and is one of the most common reason for terrible engine failure in cars. Frequently, placing the broken rod through the sides of the crankcase and thereby interpreting the engine is irretrievable. It can result from fatigue next to a physical defect in the connecting rod, lubrication failure in bearing because of improper maintenance or
from the letdown of the rod bolts from a defect, improper tightening or over-revving of the engine. In an less maintained, unclean environment, a water or chemical emulsifies with the oil that lubricates the bearing can cause the bearing to fail. Re-use of rod bolts is a common practice as long as the bolts meet specifications defined by manufacturer. Regardless of their frequent occurrence on competitive automobile events, these failures are infrequent on production cars when normal daily driving. This is due to that the production of auto parts have a much larger factor of safety and frequent systematic quality control.

**II. DESIGN CALCULATIONS**

**Engine specifications**

<table>
<thead>
<tr>
<th>Type</th>
<th>1.3 litre 74 bhp 16v DDis diesel engine</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bore x stroke (mm)</td>
<td>69.6 x 82</td>
</tr>
<tr>
<td>Maximum power</td>
<td>74 bhp @ 4000rpm</td>
</tr>
<tr>
<td>Maximum torque</td>
<td>190 Nm @ 2000 rpm</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>17.6:1</td>
</tr>
</tbody>
</table>

Table 2.1 Specification of engine

Density of diesel= 8.206-7 kg/mm³
Mass = 8.20e-7 * 1248000
=1.02 kg
Molecular cut of diesel= 168.324 g/mol

\[
P = \frac{(1.02 \times 8.3143 \times 288.15)}{(0.168324 \times 0.001248)} = 11632789.2 \text{ N/m}^2
\]

Mean effective pressure;

\[
P_m = 2\pi(T_c/v_d)
\]

\[
P_m = \frac{(2\pi \times 190)}{(0.001248)} = 1163 \text{ N/mm}^2
\]

Design calculations for connecting rod:

1. Design of connecting rod shank

Let ‘t’ be the web thickness of connecting rod shank.

Since connecting rod is a cast iron,

\[S_c = 172 \text{ MPa}\]

Area of cross section,

\[A = 2(4t \times t) + 1(t \times 3t)\]

\[A = 11t^2\]

Moment of inertia about X-axis;

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

\[I_{xx} = (419t^4)/12\]

\[I_{xx} = 35t^4 \text{ mm}^4\]

Least radius of gyration about X-axis,

\[K_{xx} = \sqrt{\frac{I_{xx}}{A}}\]

\[K_{xx} = \sqrt{35t^4/11t^2}\]

\[K_{xx} = 1.78t\]

Since,
The maximum speed is 2000rpm.
Force acting on the piston due to gas pressure,

\[F_p = \frac{\pi}{4} \times D^2 \times P\]

\[F_p = \frac{\pi}{4} \times (69.6^2) \times 11.63\]

\[F_p = 44247.43 \text{ N}\]

Now according to rankine’s formula;

\[F_{cr} = \frac{S_c \times A}{(1+a \times (1/K_{xx})^2)}\]

Where a= rankine constant,

For cast iron, a= 1/1600.

For buckling about X-axis, the both ends of connecting rod are assumed as hinged.

The equivalent length of rod,

\[L = l = 2 \times \text{stroke length}\]

\[L = 2 \times 82\]

\[L = 164 \text{ mm}\]

Hence, by rankine’s formula,

\[44247.48 = \frac{(172 \times 11t^2)}{(1+ (1/1600) (164/1.78t)^2)}\]

\[t^4 = 22.91t^2 + 123.95\]

On solving by using quadratic equation, we get,

\[t= 5.23 \text{ mm}\]

then,

The thickness of web, \(t = 5 \text{ mm}\)

Height of connecting rod, 5t = 25 mm
Width of connecting rod, 4t = 20 mm
Length of connecting rod = 164 mm

Now the maximum bending moment acting on the rod,

\[ M_{\text{max}} = m \omega^2 r \cdot (l/9\sqrt{3}) \]

Where, \( m = \text{mass of connecting rod} \)
\[ = \text{Volume} \times \text{density} \]
\[ = \text{Area} \times \text{length} \times \text{density} \]
\[ = 11t^2 \times 164 \times (7874/10^9) \]
\[ = 0.35 \text{ kg} \]

Where, \( \rho = 7.874 \text{ g/cm}^3 \)
\[ = 7874 \text{ Kg/cm}^3 \]
\( \omega = \text{Angular speed} \)
\[ = (2\pi N)/60 \]
\[ = (2\pi \times 2000)/60 \]
\[ = 209 \text{ rad/sec} \]
\( r = \text{radius of crank} \)
\[ = \text{stroke} / 2 \]
\[ = 82 / 2 \]
\[ = 41 \text{ mm} \]

On substituting above values in \( M_{\text{max}} \), we get,
\[ M_{\text{max}} = 6594.55 \text{ N-mm} \]
\[ = 6.594 \times 10^3 \text{ N-mm} \]

Section modulus,
\[ z_{xx} = (I_{xx} / (5t / 2)) \]
\[ = (419t^4 / 12) / (2/5t) \]
\[ = 14t^3 \]
\[ = 1750 \]

Maximum bending stress,
\[ S_{\text{bmax}} = (M_{\text{max}} / Z_{xx}) \]
\[ = (6.5974 \times 10^3) / 1750 \]
\[ = 3.768 \text{ N/mm}^2 \]

Since, this is less than compressive stress 172 N/mm², our design is safe.

ii) Design of connecting rod end

The small end of connecting rod is connected with piston.
The big end is connected to Crank pin.

Let,
\( l_1, d_1 \) be length and diameter of Piston pin.
\( l_2, d_2 \) be length and diameter of crank pin.
\( P_{b1}, P_{b2} \) be design bearing pressure for piston pin and crank pin.

\( F_1, F_2 \) be bearing load applied on piston pin and crank pin = Gas load (\( F_p \)).

Now,
\[ F_1 = P_{b1} \times l_1 \times d_1 \ & \]
\[ F_2 = P_{b2} \times l_2 \times d_2 \]

Assuming \( P_{b1} = 25 \text{ N/mm}^2 \)
\[ l_1 = 1.5d_1 \]

We get,
\[ F_1 = 25 \times 1.5d_1 \times d_1 \times F_p \]
\[ 25 \times 1.5d_1^2 = 44247.43 \]
\[ d_1^2 = 1179.93 \]
\[ d_1 = 34.35 \text{ mm} \]
\[ l_1 = 1.5d_1 \]
\[ l_1 = 1.5 \times 34.35 \]
\[ l_1 = 51 \text{ mm} \]

Similarly, by assuming,
\( P_{b2} = 23 \text{ N/mm}^2 \)
\[ l_2 = d_2 \]

We get,
\[ F_2 = P_{b2} \times l_2 \times d_2 = F_p \]
\[ 23 \times d_2 \times d_2 = 44247.43 \]
\[ d_2^2 = 1928.80 \]
\[ d_2 = 44 \text{ mm} \]
\[ l_2 = d_2 = 44 \text{ mm} \]
Now the inner diameter of small end,
\[ D_{in} = d_1 = 34 \text{ mm} \]
Outer diameter of small end, \( D_{os} \),
\[ D_{os} = D_{in} + (2 * \text{bush thickness}) + (2 * \text{marginal thickness}) \]
\[ D_{os} = 34 + (2*5) + (2*5) \]
\[ D_{os} = 54 \text{ mm} \]
Similarly for big end;
The inner diameter, \( D_{ib} \),
\[ D_{ib} = d_2 = 44 \text{ mm} \]
Outer diameter,
\[ = D_{ib} + (2*\text{bush thickness}) + (2*\text{bolt thickness}) + (2*\text{marginal thickness}) \]
Since bolt diameter is not known, \( D_{ob} \) cannot be found at present.

iii) Design of bolts

Bolts are subjected to inertia for by the reciprocating parts.
Inertia force,
\[ F_i = m\omega^2 r (\cos \theta + (\cos 2\theta/n)) \]
Maximum inertia force is obtained when \( \theta = 0^\circ \),
Which is given by,
\[ F_{im} = m\omega^2 r (1 + (1/n)) \]
Where, \( \omega = 209 \text{ rad/sec} \)
\[ r = 41 \text{ mm} \]
\[ n = l/r \]
\[ n = 164/41 \]
\[ n = 4 \]
Substituting, we get,
\[ F_{im} = 2247.872 \text{ N} \]
Let \( d_c \) be core diameter of bolt
\( S_t \) be Allowable tensile stress for bolt material
\( n' = \) number of bolts.
Now,
\[ S_t = (S_y/f.o.s) \]
Assume, \( S_y = 600 \text{ N/mm}^2 \)
\[ S_t = 600 / 6 \]
\[ S_t = 100 \text{ N/mm}^2 \]
\[ n' = 2 \]
Now equating \( F_{im} \) to bolt strength, we get,
\[ F_{im} = 2 * (\pi/4) * d_c^2 * S_t \]
\[ 2247.872 = 2 * (\pi/4) * d_c^2 * 100 \]
\[ d_c^2 = 14.32 \]
\[ d_c = 3.782 \text{ mm} \]
\[ d_c = 4 \text{ (standard)} \]
Hence M4 bolts can be selected.

Now the outer diameter of big end bearing;
\[ D_{ob} = D_{ib} + (2 * \text{bush thickness}) + (2 * \text{bolt thickness}) + (2 * \text{marginal thickness}) \]
\[ D_{ob} = 44 + (2*5) + (2*4) + (2*5) \]
\[ D_{ob} = 72 \text{ mm} \]

iv) Design of cap:

Bending stress induced in the cap is given by;
\[ S_{bc} = M/Z \]
\[ S_{bc} = (wl'/6) * (6/btc^2) \]
\[ S_{bc} = wl'/btc^2 \]
Now \( w = \) maximum load applied on the cap end
\[ = \text{Inertia force} [F_{im} = 2247.8 \text{ N}] \]
\[ l' = \text{Distance between bolt centers.} \]
\[ = \text{Diameter of crank pin} + (2 * \text{bush thickness}) + (\text{diameter of bolt}) + (2 * \text{marginal thickness}) \]
\[ = 44 + (2*5) + 4 + (7.5 * 2) \]
\[ l' = 81 \text{ mm} \]
\[ b = \text{width of cap} \]
\[ b = \text{length of crank pin} – (2 * \text{flange thickness of bush}) \]
\[ b = 44 – (2 * 5) \]
\[ b = 34 \text{ mm} \]
\[ S_{bc} = \frac{w l}{b t_c^2} = \frac{(2247.872 \times 81)}{(34 \times t_c^2)} \]

\[ S_{bc} = 100 \text{ N/mm}^2 \text{ (Assuming)} \]
\[ t_c^2 = \frac{(2247.872 \times 81)}{(34 \times 100)} \]
\[ t_c = 53.55 \]
\[ t_c = 7.31 \rightarrow t_c = 7 \text{ (say)} \]

Specifications:

<table>
<thead>
<tr>
<th>Thickness of web shank</th>
<th>5mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>Height of shank</td>
<td>25 mm</td>
</tr>
<tr>
<td>Width of shank</td>
<td>20 mm</td>
</tr>
<tr>
<td>Inner diameter of small end</td>
<td>34 mm</td>
</tr>
<tr>
<td>Outer diameter of small end</td>
<td>54 mm</td>
</tr>
<tr>
<td>Inner diameter of big end</td>
<td>44 mm</td>
</tr>
<tr>
<td>Outer diameter of big end</td>
<td>72 mm</td>
</tr>
<tr>
<td>Length of connecting rod</td>
<td>164 mm</td>
</tr>
<tr>
<td>Diameter of bolt</td>
<td>4 mm</td>
</tr>
<tr>
<td>Number of bolt</td>
<td>2</td>
</tr>
<tr>
<td>Width of cap</td>
<td>34 mm</td>
</tr>
<tr>
<td>Thickness of cap</td>
<td>7 mm</td>
</tr>
</tbody>
</table>

Table 2.2 Specifications of connecting rod

Material specifications

<table>
<thead>
<tr>
<th>Component</th>
<th>Material</th>
</tr>
</thead>
<tbody>
<tr>
<td>Connecting rod</td>
<td>Malleable Cast iron</td>
</tr>
<tr>
<td>Connecting rod cap</td>
<td>Malleable Cast iron</td>
</tr>
<tr>
<td>Hexagonal Bolt</td>
<td>Stainless steel</td>
</tr>
<tr>
<td>Hexagonal Nut</td>
<td>Stainless steel</td>
</tr>
</tbody>
</table>

Table 2.3 Specifications of Material used

<table>
<thead>
<tr>
<th>Material</th>
<th>Young’s modulus</th>
<th>Density</th>
</tr>
</thead>
<tbody>
<tr>
<td>Malleable CI</td>
<td>1.72E+11 (Pa)</td>
<td>7200 (Kg/m^3)</td>
</tr>
<tr>
<td>Stainless steel</td>
<td>1.93E+11 (Pa)</td>
<td>7750 (Kg/m^3)</td>
</tr>
</tbody>
</table>

Table 2.4 Properties of materials used

**III. MODELLING**

The detailing modelling and assembly of the connecting rod using calculated dimensions were done using Solidworks 2014.

SolidWorks is a solid modeling computer-aided design (CAD) and computer-aided engineering (CAE) computer program that runs on Microsoft Windows. SolidWorks is published by Dassault Systems.
IV. ASSEMBLY

The assembly of various parts into a component is also achieved using solidWorks 2014. Assembly is a component or an end item comprising of a number of parts or subassemblies put together to perform a specific function, and capable of disassembly without destruction.

V. MESHING

The meshing of existing design and new design of connecting rod, static structural, modal analysis were done through Ansys 15.0 workbench.

VI. ANALYSIS

Total deformation

Total Deformation is the change in the shape of an object when applying force. It is analyzed with the help of ANSYS 15.0.

ANSYS is a software used to determine how a product will function with different specifications, without building test products or conducting crash tests.
Fig 6.1 Total deformation – existing design

Fig 6.2 Total deformation – New design

Fig 6.3 Equivalent stress – existing design

Fig 6.4 Equivalent stress – New design

Modal analysis:

Existing design:

Fig 6.5 Deformation at 437.67 Hz

Fig 6.6 Deformation at 1195.9 Hz
Fig 6.7 Deformation @ 1354.3 Hz

Fig 6.8 Deformation at 2742.4 Hz

Fig 6.9 Deformation at 4799.2 Hz

Fig 6.10 Deformation at 6044.5 Hz

New design:

Fig 6.11 deformation at 436.08 Hz

Fig 6.12 Deformation at 1193.1 Hz
VII. RESULT

The difference between total deformation and equivalent stress is tabulated below.

<table>
<thead>
<tr>
<th></th>
<th>Existing design</th>
<th>New design</th>
</tr>
</thead>
<tbody>
<tr>
<td>Total deformation (mm)</td>
<td>0.12933</td>
<td>0.12846</td>
</tr>
<tr>
<td>Equivalent stress (N/mm²)</td>
<td>233.35 (max.)</td>
<td>246.72 (max.)</td>
</tr>
<tr>
<td></td>
<td>0.00523 (min.)</td>
<td>0.01815 (min.)</td>
</tr>
</tbody>
</table>

Table 7.1 difference between existing design and new design on total deformation and stress.

The modal analysis of existing and new design of connecting rod is tabulated below.

<table>
<thead>
<tr>
<th>Mode</th>
<th>Frequency(Hz)</th>
<th>Deformation(m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>436.88</td>
<td>2.5765</td>
</tr>
<tr>
<td>2</td>
<td>1193.9</td>
<td>2.5527</td>
</tr>
<tr>
<td>3</td>
<td>1323.4</td>
<td>3.2768</td>
</tr>
<tr>
<td>4</td>
<td>2727.6</td>
<td>2.6144</td>
</tr>
<tr>
<td>5</td>
<td>4761.4</td>
<td>2.1207</td>
</tr>
<tr>
<td>6</td>
<td>5916.3</td>
<td>3.3055</td>
</tr>
</tbody>
</table>

Table 7.2 Deformation with respect to frequency on existing design.

<table>
<thead>
<tr>
<th>Mode</th>
<th>Frequency(Hz)</th>
<th>Deformation(m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>436.08</td>
<td>2.5659</td>
</tr>
<tr>
<td>2</td>
<td>1193.1</td>
<td>2.5454</td>
</tr>
<tr>
<td>3</td>
<td>1344.3</td>
<td>3.2768</td>
</tr>
<tr>
<td>4</td>
<td>2681.2</td>
<td>2.5959</td>
</tr>
<tr>
<td>5</td>
<td>4724.5</td>
<td>2.1194</td>
</tr>
<tr>
<td>6</td>
<td>5941.9</td>
<td>3.2926</td>
</tr>
</tbody>
</table>

Table 7.3 Deformation with respect to frequency on new design.
VIII. CONCLUSION

On comparing the stress distribution and deformation formed between the existing and modified design, the stress developed is greater in modified design than in existing design & also the deformation produced in the modified design is lesser than the existing design of connecting rod.

References


Books

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