Design Optimization and Analysis of a Connecting Rod using ANSYS

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Abstract: ***The main Objective of this work is to explore weight reduction opportunities in the connecting rod of an I.C. engine by examining various materials such as Genetic Steel, Aluminum, Titanium and Cast Iron. This was entailed by performing a detailed load analysis. Therefore, this study has dealt with two subjects, first, static load and stress analysis of the connecting rod and second, Design Optimization for suitable material to minimize the deflection. In the first of the study the loads acting on the connecting rod as a function of time are obtained. The relations for obtaining the loads for the connecting rod at a given constant speed of crank shaft are also determined. It can be concluded from this study that the connecting rod can be designed and optimized under a comprising tensile load corresponding to 360° crank angle at the maximum engine speed as one extreme load, and the crank pressure as the other extreme load. Furthermore, the existing connecting rod can be replaced with a new connecting rod made of Genetic Steel.***

Keywords: Weight reduction, Genetic Steel, Detailed load analysis, Design Optimization, Comprising tensile load.

1. Importance of Connecting Rod

It interconnects the piston and the crank shaft and transmits the gas forces from the piston to the crankshaft. 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 rotary motion of the crank the usual form of the connecting rod in internal combustion engines. It consists of a long shank a small end and big end. The small end of connecting rod is usually made in the form of an eye and is provided with a bush. It is connected to the piston by means of piston pin. The big end of connecting rod is usually made into two halves so that it can be mounted easily on the crank pin bearing shells. The split is fastened to big end with two cap bolts. Big end bearing is allowed for by inserting thin metallic strip known as shims. The big end bearing is usually splash lubricated while the small end bearing is pressure lubricated.

2. Design Procedure for Connecting Rod

Being one of the most integral parts in an internal combustion engines design, the connecting rod must be able to withstand tremendous loads and transmit a great deal of power. It is no surprise that a failure in a connecting rod can be one of the most costly and damaging failures in an engine. But simply saying that isn't enough to fully understand the dynamics of the situation.

2.1. Forces acting on the Connecting Rod

The various forces acting on the connecting rod are as follows: Forces on the piston due to gas pressure and inertia of the reciprocating parts.

1. Forces on the piston due to gas pressure and inertia of the reciprocating parts.
2. Force due to inertia of the connecting or inertia bending forces
3. Force due to friction of the piston rings and of the piston, and
4. Forces due to friction of the piston pin bearing and crank pin bearing.

In this study, the first two forces have been considered.

2.1.1 Forces on the piston due to gas pressure and inertia of the reciprocating parts:

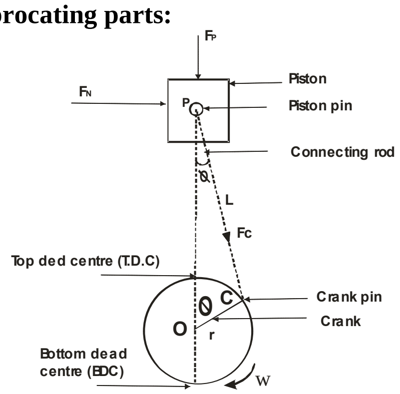

Fig: 1.Forces acting on connecting rod

It is known that the force on the piston due to pressure of gas,

F_L = Pressure X Area = p.A=p x $\pi D^2/4$ ---------- (1)

And inertia force of reciprocating parts,

F_I = Mass x acceleration = $m_R . w^2 . r\,(\cos\theta + \cos 2\theta/n)$---------- (2)

It may be noted that the inertia force of reciprocating parts opposes the force on the piston when it moves during its downward strokes (i.e., when the piston moves from the Top dead centre to Bottom dead centre). On the other hand, the inertia force of the reciprocating parts helps the force on the piston when it moves from the Bottom Dead Centre (BDC) to Top Dead Centre (TDC).

:: Net force acting on the piston or piston pin (or gudgeon pin or wrist pin)

F_p = Force due to gas pressure ± inertia force = $F_L \pm F_I$

The negative sign is used when piston moves from Top Dead Centre to Bottom Dead Centre and positive sign used when piston moves from Bottom Dead Centre to Top Dead Centre.

When weight of the reciprocating parts ($W_R = m_R.g$) is to be taken in to consideration, then

$F_P = F_L \pm F_I \pm W_R$

The force F_P gives rise to a force F_C in the connecting rod and a thrust F_N on the sides of the cylinder walls from fig.1 we see that force in the connecting rod at any instant,

$F_C = Fp/Cos\Phi = FP\ /\sqrt{1-Sin^2\theta}\ /n^2$---------- (3)

The force in the connecting rod will be maximum when the crank and the connecting rod are perpendicular to each other (i.e. when $\theta = 90^o$). But at this position, the gas pressure would be decreased considerably. Thus, for all practical purposes, the force in the connecting rod F_C is taken equal to the maximum force on the piston due to pressure of gas F_I, neglecting piston inertia effects.

2.1.2 Forces due to inertia of the connecting rod or inertia bending forces

Consider a connecting rod PC and a crank OC rotating with uniform angular velocity 'w' rad/sec. In order to find the acceleration of various points on the connecting rod, draw the Kliens acceleration diagram CQNO as shown fig.2. CO represents the acceleration of C towards O and NO represents acceleration of P towards O. The acceleration of other points such as D, E, F and G etc., on the connecting rod PC may be found by drawing horizontal lines from these points two intersects CN at d, e, f and g respectively. Now do,eo,fo and go represents the acceleration of D, E, f ,and G all towards O. The inertia force acting on each point will be as follows:

Inertia force at C $=m*w^2*CO$
Inertia force at $D=m*w^2*DO$
Inertia force at $E=m*w^2*EO$, and so on.

The inertia forces will be opposite to the direction of acceleration or centrifugal forces. The inertia forces can be resolved in to two components, one parallel to the connecting rod and the other perpendicular to the rod. The parallel (or longitudinal) components add up algebraically to the force acting on the connecting rod (F_C) and produces thrust on the pins. The perpendicular (or transverse) components produces bending action (also called whipping action) and the stresses induced in the connecting rod is called whipping stress.

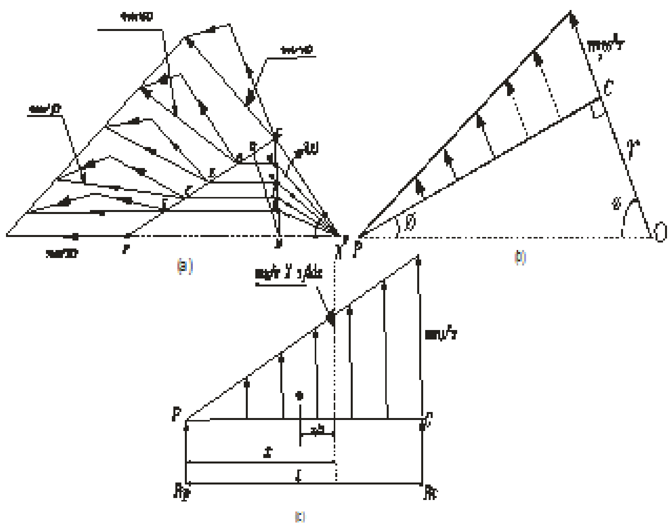

Figure 2: Inertia and bending forces

It may be noted that perpendicular components will be maximum, when the crank and the connecting rod are at right angles to each other. The variation of the inertia force on the connecting rod is linear and is like a simply supported beam of variable loading as shown in fig.2.Assuming that the connecting rod is of uniform cross-section and has mass m1 kg per unit length, therefore,

1. Inertia force per unit length at the crank pin$=m_1*w^2r$
2. Inertia force per unit length at piston pin=0
3. Inertia force due to small element of length dx at distance x from piston pin p, $dF_I=m_1*w^2*r*x/l*dx$
4. Resultant inertia force

$$F_I = \int_0^L m1 * w^2 r * x/l * dx = m1 * \frac{w^2 r}{l} * \left[\frac{x^2}{l}\right]_0^l$$
$$= m1 * \frac{1}{2} * w^2 r = \frac{m}{2} * w^2 r \text{ ---------- (4)}$$

This resultant inertia force acts at distance of 2l/3 from piston pin P.

Since it has been assumed that 1/3rd mass of the connecting rod is concentrated at piston pin P and 2/3rd at the crank pin, therefore, the reaction at these two ends will be in the same proportion i.e.,

$$RP=F_I/3 \text{ and } RC=2F_I/3$$

Now the bending moment acting on the rod at section X-X at a distance x from P,

$$M_x = R_p\ X\ x\text{-}{*}m_1\ X\ w^2r\ X\ x/l\ X\ l/2x\ X\ x/3$$
$$=1/3F_I\ X\ x\text{-}m1.l/2xw2rXx3/3l2$$

(..Multiplying and dividing the letter expression by l)

$$\frac{F_I X x}{3} = F_I X \frac{x3}{3l^2} = \frac{F_I}{3}\left(x = \frac{x^2}{l^2}\right) \text{ ---------- (5)}$$

For maximum bending moment, differentiate Mx with respect to x and equate to zero, i.e.,

$$\frac{dMX}{dx} = 0 \text{ or } \frac{F_I}{3}\left[1-\frac{3x^2}{l^2}\right] = 0$$

$$1-\frac{3x^2}{l^2} = 0 \text{ or } 3x^2 = l^2 \text{ or } x = l/\sqrt{3}$$

Maximum bending moment,

$$M_{\max} = \frac{F_I}{3}\left[\frac{l}{\sqrt{3}} - \left\{\frac{l}{\sqrt{3}}\right\}^3 / l^2\right]$$

$$= \frac{FI}{3}\left[\frac{l}{\sqrt{3}} - \frac{l}{3\sqrt{3}}\right] = \frac{FI * l}{3\sqrt{3}} * \frac{2}{3} = \frac{2FI * l}{9\sqrt{3}}$$

$$= 2 * \frac{m}{2} * w2r * \frac{l}{9\sqrt{3}} = m * w2r * \frac{l}{9\sqrt{3}}$$

And the maximum bending stress, due to inertia of the connecting rod,

$\sigma_{max} = M_{max}/Z$, Where, Z=section modulus.

From above it can be seen that the maximum bending moment varies as the sware of speed therefore, the bending stress due to high speed will be dangerous. It may be noted that the maximum axial force and the maximum gas load occurs close to top dead centre whereas the maximum bending stress occurs when the crack angle $\theta = 65^o\text{-}70^o$ from top dead centre. The pressure of gas falls suddenly as the piston moves from dead centre. Thus the general practice is to design a connecting rod by assuming the force in the connecting rod (F_C) equal to the maximum force due to the pressure (F_L), neglecting piston inertia effects and then checked for bending stress due to inertia force (i.e., whipping stress).

3. Designing of Connecting Rod

In designing a connecting rod, the following dimensions are required to be determined.

- Dimensions of cross-section of the connecting rod
- Dimensions of the crankpin at the big end and the piston pin at the small end
- Size of bolts for securing the big end cap, and
- Thickness of the big end cap

The procedure adopted in determining the above mentioned dimensions are discussed as below:

3.1 Dimension and cross-section of the connecting rod

A connecting rod is a machine member which is subjected to alternating direct compressive and tensile forces. Since the compressive forces are much higher than the tensile forces, therefore the cross-section of the connecting rod is designed as a strut and the Rankine's formula is used.

A connecting rod, as shown fig.3 subjected to an axial load W may buckle with X-axis as neutral axis (i.e., in the plane of motion of the connecting rod) or Y-axis as neutral axis (i.e., in the plane perpendicular to the plane of motion). The connecting rod is considered like both ends hinged for buckling about X-axis and both ends fixed for buckling about Y-axis. A connecting rod should be equally strong in buckling about the axes.

I_{xx} and I_{yy} = Moment of inertia of the section about X-axis and Y-axis respectively, and
K_{xx} and K_{yy} = Radius of gyration of the section about X-axis and Y-axis respectively

According to Rankine'S formula

W_b about X-axis =

$$\frac{\sigma_c.A}{1+a\left(\frac{L}{k_{xx}}\right)^2}=\frac{\sigma_c.A}{1+a\left(\frac{l}{k_{xx}}\right)^2}$$

W_b about Y-axis =

$$\frac{\sigma_c.A}{1+a\left(\frac{L}{k_{yy}}\right)^2}=\frac{\sigma_c.A}{1+a\left(\frac{l}{2k_{yy}}\right)^2}$$

Where L = Equivalent length of the connecting rod, and
a = Constant
= 1/75000, for mild steel
= 1/9000 for wrought iron
= 1/1600, for cast iron

In order to have a connecting rod equally strong in buckling about the axis, the buckling load must be equal i.e.,

$$\frac{\sigma_c.A}{1+a\left(\frac{L}{k_{xx}}\right)^2}=\frac{\sigma_c.A}{1+a\left(\frac{l}{2k_{yy}}\right)^2}$$

$$K^2_{xx} = 4K^2_{yy} \text{ or } I_{xx} = 4I_{yy}$$

This shows that the connecting rod is four times strong in buckling about Y-axis than about X-axis. If $I_{xx} > 4I_{yy}$, then buckling will occur about y-axis and if $I_{xx} < 4I_{yy}$ buckling will occur about X-axis. In actual practice, I_{xx} is kept slightly less than 4Iyy. It is usually taken between 3 and 3.5 and the connecting rod designed for buckling about X-axis. The design will always be satisfactory for buckling about Y-axis.
The most suitable section for the connecting rod is I-section with proportional as shown fig. 3.

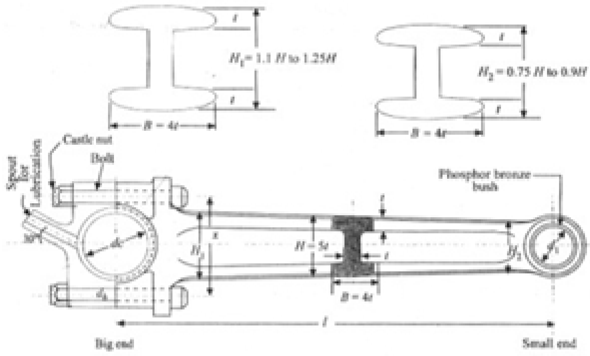

Figure 3: Parameters of connecting rod

Let,
1. Thickness of the flange and web of the section = t
2. Width of the section B=4t
3. Depth or height of section H = 5t from fig3,
4. Area of the crosssection $A = 2(4t * t) + 3t * t = 11t^2$
5. Moment of Inertia of the section about x-axis,

$$\text{Ixx} = 1/12[4t(5t)^3 - 3t(3t)^3] = 419\ t^4/12$$

6. Moment of inertia of the section about y-axis,

$$\text{Iyy} = \left[2 * \frac{t}{12} * (4t)3 + \left(\frac{3t^4}{12}\right)\right] = 131\ t^4/12$$

$$\frac{\text{Ixx}}{\text{Iyy}} = 3.2$$

Since the value of I_{xx}/I_{yy} lies between 3 and 3.5, therefore, I-section chosen is quite satisfactory. After deciding the proportions for I-sections of the connecting rod, its dimensions are determined by considering the buckling of the rod about X-axis (assuming both ends hinged) and applying Rankine's formula. It is known that buckling load,

$$W_B = \frac{\sigma_c.A}{1+a\left(\frac{L}{k_{xx}}\right)^2}$$

The buckling load (W_B) may be calculated by using the following relations, i.e., W_B = Max, gas force X factor of safety. The factor of safety may be taken as 5 to 6.

Note:

1. The I-section of the connecting rod is used due its lightness and to keep the inertia forces as low as possible especially in case of high speed engine. It can also withstand high gas pressure.
2. Sometimes a connecting rod may have rectangular section. For slow speed engines, circular cross-sections may be used.
3. Since connecting rod is manufactured by forging, therefore the sharp corners of I-section are rounded off as shown fig above. For easy removal of section from dies. [1,2,3,4]

4. Modeling of connecting rod using pro-engineering

The following steps are to be followed while designing a connecting rod for the study.

1. Create the solid model of connecting rod, by using pro-e.
2. Import the file from pro-e to ANSYS.
3. For ANSYS it is needed to do a Finite Element Analysis (FEA) what kind of element to be used for this connecting rod model. It is used a 10 node and 20 node tetra shape of element type solid92 and solid95 and are specified as follows.
4. The type of material should be specified for given connecting rod, in this work the materials considered are: Genetic steel, Aluminum, Cast iron, Titanium.

4. Areas are divided at the big end of the connecting rod. The parts are divided in to 8 parts.

5. *Meshing of the model:* The procedure to generate a mesh is called meshing. Meshing is done on the connecting rod for easy solving and accurate results in ANSYS. In 3D modeling we have only two types of meshes are there, namely, i. Hexa ii.Tetra. In tetra meshing only we have free meshing and in hexa either we go for mapped meshing or free meshing are available, but in hexa meshing always the mapped mesh had been used. In 3D meshing there are three types of planers namely, (i) Solid-45, (ii) Solid-95, (iii) Solid-92.
   In solid-92 the no. of nodes are 10 and it has two degrees of freedom per node. In solid-95 the no., of nodes are 20 and it has two degrees of freedom.
6. *Applying Boundary conditions to the model:* First arrest the degrees of freedom of smaller end of the connecting rod. After that apply the loads on the big end of the connecting rod at different areas and the result obtained are shown in the figure4.

Figure 4: Applying pressures to the connecting rod at the big end and small end.

7. Transferring of geometric model in to the FEA (finite element analysis):
8. *Solution:* After solving this model it gives the deflections and von misses stress produced in the connecting rod.

5. Design Optimization of Connecting Rod

5.1 Introduction to Design Optimization

The purpose of design optimization is to modify the stress in the model by increasing or decreasing the area. The stress value can be decreased. From the formula, $stress = Load/Area$, it is known that, stress is inversely proportional to area, so by increasing the area the stress value could be decreased. This is the concept of design optimization. The design optimization of a proto type of a connecting rod is described as shown below:

5.2 Initiating the values to scalar parameters of connecting rod

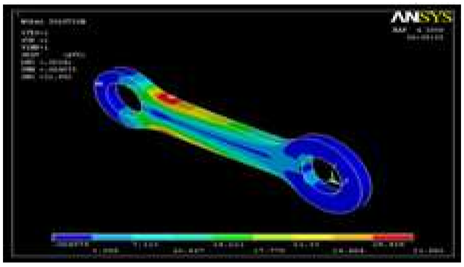

Figure 5: Stress and Volume values for prototype connecting rod

The radius value of big end and small end of connecting rod and the length of the connecting rod and the fillet radius are given to the scalar parameters. With the help of these parameters the prototype of connecting rod is drawn and the stress value in the model is analyzed.

When we have done solution on the prototype with a uniform pressure (1MPa) at the big end of the connecting rod the stress value is as shown in the figure5 below. The stress value at the beginning stage is 31.992 N/mm^2. The stress value for the prototype of the connecting rod S_V is 31.6273407 and the volume value for prototype of the connecting rod V_V is 1.64271953. It is observed from the fig 5 that the maximum stress is obtained at small end of the connecting rod. [5, 6, 7]

5.3 Changing the fillet radius value and analyzing the model

The maximum stress value which would occur at small end of the connecting rod near to the fillet radius. By changing the fillet radius the stress value would be changed as shown in the figure below. The stress value for the prototype of the connecting rod is Stress value, S_V is 28.369251 and the fillet radius value is 1.335. As discussed above the stress value for the first case that is 31.6273407 and by changing the fillet radius the stress value obtained is 28.369251.The radius value for the first case is 1.5and the fillet radius value for the second case is 1.335.

The values of deflections, maximum and minimum stresses for the different cases (applying forces at different areas) for different materials for element type solid92 and solid 95 are shown in the table 1 & 2 respectively.

Table 1: Deflections, maximum and minimum stresses using Solid-92

Solid92	*Genetic steel*			*Titanium*			*Cast Iron*			*Aluminum*		
	Dmx	Smx	Smn	Dmx	Smx	Smn	Dmx	Smx	Smn	Dmx	Smx	Smn
Case1	0.00173	21.839	0.00255	0.00329	21.264	0.00268	0.00344	22.933	0.00237	0.00512	21.092	0.00272
Case2	0.5107	400.295	0.03189	0.97258	397.826	0.06265	1.021	404.917	0.02384	1.513	397.202	0.63888
Case3	0.00389	37.902	0.00255	0.00741	36.315	0.00268	0.00774	38.394	0.00237	0.01153	36.068	0.00272
Case4	0.51058	399.437	0.05983	0.97258	397.826	0.03429	1.021	404.917	0.0464	1.513	397.202	0.35587
Case5	0.36318	285.951	0.09378	0.68329	285.799	0.01932	0.72603	290.728	0.08169	1.063	285.34	0.19396

Table 2: Deflections, maximum and minimum stress using Solid-95

S95	**Genetic steel**			**Titanium**			**Cast Iron**			**Aluminum**		
case	Dmx	Smx	Smn	Dmx	Smx	Smn	Dmx	Smx	Smn	Dmx	Smx	Smn
1	0.0017	21.77	0.002	0.0032	21.264	0.0026	0.0034	22.933	0.0023	0.005	21.092	0.002
2	0.5105	399.4	0.031	0.9725	397.82	0.0626	1.021	404.91	0.0238	1.513	397.20	0.638
3	0.0038	37.16	0.002	0.0074	36.315	0.0026	0.0077	38.394	0.0023	0.011	36.068	0.002
4	0.5107	399.3	0.032	0.9725	397.82	0.0342	1.021	404.91	0.0464	1.513	397.20	0.355
5	0.3632	285.8	0.090	0.6832	285.79	0.0193	0.7260	290.72	0.0816	1.063	285.34	0.193

6. Conclusions

This work investigated weight reduction and the suitable better material for minimizing deflections in connecting rod. First the connecting rod was digitized. Load analysis was performed which comprised of the connecting rod, small and big ends of connecting rod using analytical techniques and computer based mechanism simulation tools. FEA was then performed using the results from load analysis to gain insight on the structural behavior of the connecting rod and to determine the design loads for optimization. The following conclusions can be drawn from this study.

- There is considerable deference in the structural behavior of the connecting rod between axial fatigue loading and static loading. There are also differences in the analytical results obtained from fatigue loading simulated by applying loads directly to the connecting rod.
- Bending Stresses are significant and should be accounted for tensile bending stresses are about 266.86333 N/mm^2
- In this FEA, the two model analysis Solid 92, Solid 95 is used to estimate the bending stresses and deflection. From this study, it is found that solid 95 gives accurate measurements of stress compared to Solid 92
- It is also found that the connecting rod made of genetic steel shows less amount of deflection and stresses than other material like Titanium, Cast Iron and Aluminum which are also studied in this study.

7. Nomenclature

P = Maximum pressure of gas
D = Dia. Of Piston
A_P = Cross-sectional area of piston = $\pi D^2/4$
m_R = mass of reciprocating parts
w = angular speed of crank
$\emptyset$= Angle of inclination of the connecting rod with the line of stroke
θ= Angle of inclination of the crank from Top Dead Centre
r = radius of crank
L = Length of the connecting rod and
n = ratio of length of connecting rod to radius of crank = L/r
F_L = Force due to gas pressure on connecting rod
F_I = Force due inertia on connecting rod
F_p = Net force acting on the piston or piston pin
σ_C = Compressive yield stress
Wb = Buckling load,
I_{xx} and I_{yy} = Moment of inertia of the section about X-axis and Y-axis respectively
K_{xx} and K_{yy} = Radius of gyration of the section about X-axis and Y-axis respectively

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