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Design and Stress Analysis of Connecting Rod for Light Truck Engine

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Abstract: The purpose of this paper is to design and stress analysis of connecting rod for 3C light truck engine. The automobile engine connecting rod is a high volume production critical component. Every vehicle that uses an internal combustion engine requires at least one connecting rod depending upon the numbers of cylinders in the engine. The maximum gas pressure is first determined from the given specifications. All the dimension of the other parts was chosen from the empirical formula. The connecting rod design is calculated based upon bending stress. The rated power at the engine is 67KW at maximum engine speed of 4000 rpm. Connecting rod length is 174mm.Design Calculation for small end, big end, shank, cap and bolts of connecting rod are induced in connecting rod design. Connecting rod design calculations are analyzed by using MATLA program and details drawing and stress analysis are presented by using Solid works Software.

Keywords: Analysis, Bending Stress, Connecting Rod, Maximum Gas Pressure, SolidWorks.

I. INTRODUCTION

Today many types of automobile are moderately produced for many sites of the world .Light truck engines are widely used in many countries for transportation at present. The automobile engine connecting rod is a high volume production, critical component and it is classified under functional component. Its acts as a linkage between piston and crankshaft. The main function of connecting rod is to transmit the translational motion of piston to rational motion of crankshaft. The main function of connecting rod is also involves transmitting the thrust of piston to the connecting rod has three main zones. The piston pin end, the center shank and the big end. The piston pin end is the small end, the crank end is the big end and the center shank is of I cross section. The connecting rod is acted upon by gas loads and inertia loads during its operation. The forces included gas forces due to its own weight.

II. BACK GROUND OF DESIGN

Design procedures of connecting rod for internal combustion engines are selecting the materials for design, collecting the design data, selecting the engine for design parts, determining the basic design parameter, considering the required design parts, checking the design parts, analyzing strength, strain and safety factor of the design parts and resulting the design parts.

A. Design Specifications

Required design parameters are taken from UD Group Car Construction and these specifications are as follows

Types of engine = Four cylinder, four strokes, diesel engine
Application of engine = To drive the motor vehicle
Maximum Power Output =67kW at 4000rpm
Compression ratio =22:1
Cylinder Stroke =86mm
Cylinder Bore =94mm

B. Forces on the piston due to gas pressure and inertia of the reciprocating parts

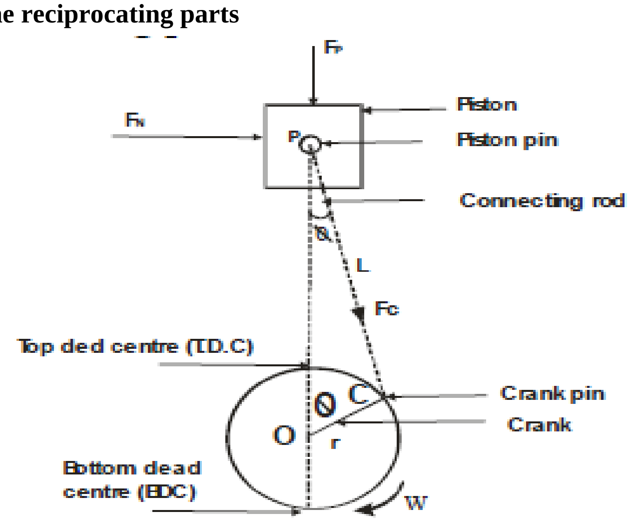

Fig1. Forces acting on connecting rod.

From fig 1 it is known that the force on the piston due to pressure of gas,

$$FL = \text{Pressure X Area} = P\times A = P\times\frac{\pi}{4}\times d^2 \quad (1)$$

And inertia force of reciprocating parts,

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FI = Mass x acceleration

$$= m_R \times \omega^2 \times r(\cos\theta + \cos 2\theta/n) \quad \textbf{(2)}$$

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)

Fp = Force due to gas pressure ± inertia force
= FL ± FI

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 $(W_R = m_R \times g)$ is to be taken in to consideration, then

FP = FL ± FI ± WR

The force FP gives rise to a force FC in the connecting rod And a thrust FN on the sides of the cylinder walls from fig.1 we see that force in the connecting rod at any instant,

$$F_c = \frac{F_p}{\cos\varphi} = F_p \times \sqrt{1 - \frac{\sin^2\theta}{n^2}} \quad \textbf{(3)}$$

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

C. 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 'ω' 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*w2 *CO
- Inertia force at D=m*w2*DO
- Inertia force at E=m*w2*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 (FC) 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.

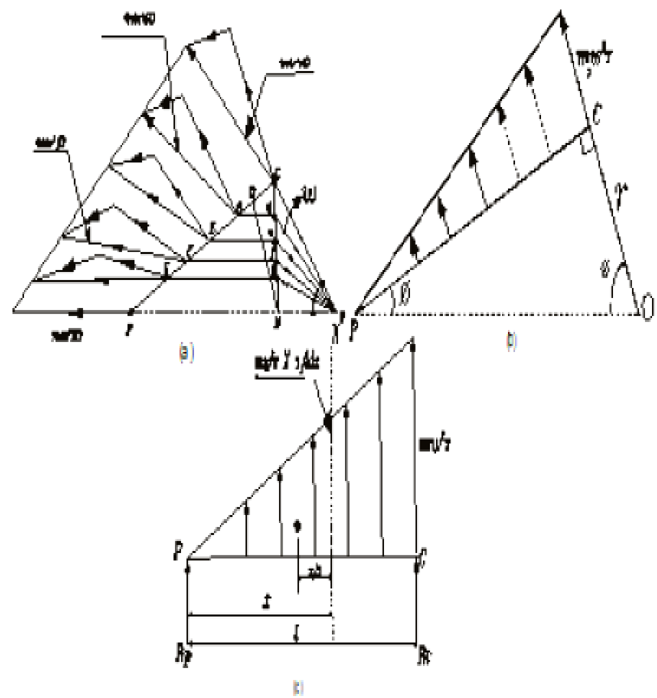

Fig2. Inertia and bending forces.

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=m1*w2r
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, dFI=m1×w2×r×x/l×dx
4. Resultant inertia force

$$F_1 = \int_0^L m_1 \times \omega^2 \times r \times \frac{x}{l} \times dx \quad (4)$$

$$= m_1 \times \frac{\omega^2 \times r}{l} \times [\frac{x^2}{l}]_0^l$$

This resultant inertia force acts at distance of $\frac{2 \times l}{3}$ from piston pin P. Since it has been assumed that $(1/3)^{rd}$ mass of the connecting rod is concentrated at piston pin P and $(2/3)^{rd}$ at the crank pin, therefore, the reaction at these two ends will be in the same proportion i.e.,

$R_p = F_1/3$ and $R_c = 2F_1/3$

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

$$M_x = R_p \times x - m_1 \times \omega^2 \times \frac{x}{l} \times \frac{l}{2} \times x \times \frac{x}{3} \quad (5)$$

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

$$M_x = \frac{F_1 \times x}{3} - F_1 \times \frac{x^3}{3 \times l^2}$$

$$M_x = \frac{F_1}{3}[x - \frac{x^3}{l^2}] \quad (6)$$

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

$$\frac{d_{Mx}}{dx} = 0 \quad \text{(or)} \quad \frac{F_1}{3} \times [1 - \frac{3 \times x^2}{l^2}] = 0$$

$$1 - \frac{3 \times x^2}{l^2} = 0 \;\text{(or)}\; 3 \times x^2 = l^2 \;\text{(or)}\; x = \frac{l}{\sqrt{3}}$$

Maximum bending moment,

$$M_{max} = \frac{F_1}{3} \times [\frac{l}{\sqrt{3}} - \frac{[\frac{l}{\sqrt{3}}]^3}{l^2}]$$

$$M_{max} = \frac{F_1}{3} \times [\frac{1}{\sqrt{3}} - \frac{1}{3 \times \sqrt{3}}]$$

$$M_{max} = \frac{F_1}{3} \times \frac{1}{\sqrt{3}} \times \frac{2}{3} = \frac{2 \times F_1 \times l}{9 \times \sqrt{3}}$$

$$= 2 \times \frac{m}{2} \times \omega^2 \times r \times \frac{l}{9 \times \sqrt{3}}$$

$$= m \times \omega^2 \times r \times \frac{l}{9 \times \sqrt{3}}$$

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

$$\sigma_{max} = \frac{M_{max}}{Z} \quad (7)$$

Where, Z=section modulus.

From above it can be seen that the maximum bending moment varies as the square of speed therefore, the bending stress due to high speed will be dangerous. It may be noted that the maximum axial force and maximum gas load occurs close to top dead centre whereas the maximum bending stress occurs when the crack angle $\theta = 65^o to 70^o$ from top depth center. 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 (FC) equal to the maximum force due to the pressure (FL), neglecting piston inertia effects and then checked for bending stress due to inertia force (i.e., whipping stress).

III. 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 crank pin at the big end and the piston pin at the small end
- Size of bolts for securing the big end cap
- Thickness of the big end cap

A. 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). A connecting rod should be equally strong in buckling about the axes.

$$\text{WB about X-axis} = \frac{\sigma_c \times A}{1 + a \times [\frac{L}{K_{xx}}]^2} = \frac{\sigma_c \times A}{1 + a \times [\frac{l}{K_{xx}}]^2} \quad (8)$$

(For both ends hinged, L=l)

$$W_B \text{ about Y-axis} = \frac{\sigma_c \times A}{1 + a \times [\frac{L}{K_{yy}}]^2} = \frac{\sigma_c \times A}{1 + a \times [\frac{l}{2 \times K_{yy}}]^2} \quad (9)$$

(For both ends fixed, L=l/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.,

$$\frac{\sigma_c \times A}{1 + a \times [\frac{L}{K_{xx}}]^2} = \frac{\sigma_c \times A}{1 + a \times [\frac{l}{2 \times K_{yy}}]^2}$$

$$[\frac{l}{K_{xx}}]^2 = [\frac{l}{K_{yy}}]^2$$

$$K_{xx}^2 = 4 \times K_{yy}^2 \quad \text{(or)} \quad I_{xx} = 4 \times I_{yy} \quad (10)$$

Where,
Ixx and Iyy = Moment of inertia of the section about X-axis and Y-axis respectively, and
Kxx and Kyy = Radius of gyration of the section about X-axis and Y-axis respectively

This shows that the connecting rod is four times strong in buckling about Y-axis than about X-axis. If Ixx > 4Iyy, then buckling will occur about y-axis and if Ixx < 4Iyy buckling will occur about X-axis. In actual practice, Ixx 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 cross-section of the shank may be rectangular, circular tubular, I-section or H-section as shown in figure. The length of the connecting rod (l) depends upon the ratio of l/r, where r is the crank radius fig. 3.

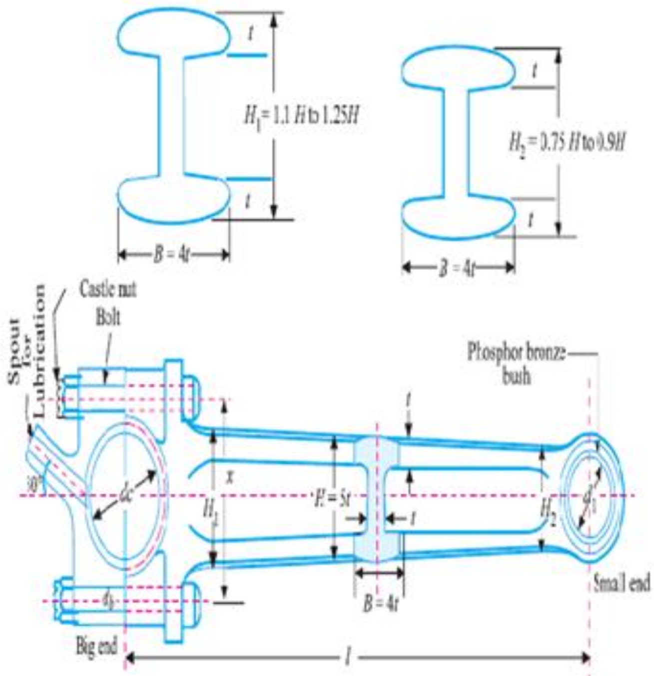

Fig3. Parameters of connecting rod.

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 in fig 4.

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 cross-section, A=2× (4×t×t) + (3×t×t)
5. Moment of Inertia of the section about x-axis,
   $I_{xx}=1/12[4t\times (5t)^3 -3t(3t)^3]= 419\ t^4/12$
6. Moment of inertia of the section about y-axis,

$$I_{yy}=[2\times\frac{t}{12}\times(4t)\times3+(\frac{3t^4}{12})]=\frac{131t^4}{12}$$

$$\frac{I_{xx}}{I_{yy}}=3.2$$

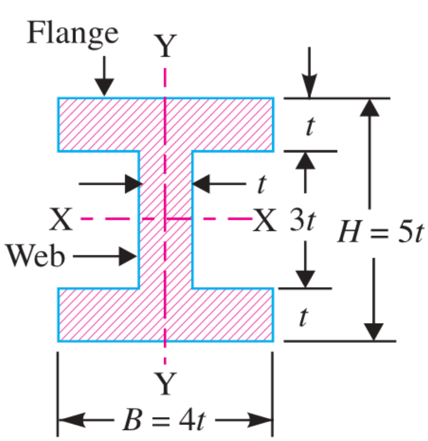

Fig4. Section of the connecting rod.

Since the value of Ixx/Iyy 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,

$$W_B=\frac{\sigma_c\times A}{1+a(\frac{L}{K_{xx}})^2} \quad \textbf{(11)}$$

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

C. Length of Connecting Rod

Ratio between connecting rod length and crank radius

Crank radius, $r=\frac{l}{2}$ **(12)**

Equivalent length of connecting rod, $L=\frac{R}{\lambda}$ **(13)**

D. Design Calculation of Crank Pin

The maximum gas force on the piston due to gas pressure,

$$F_L = \frac{\pi D^2}{4}\times P \quad \textbf{(14)}$$

The Load on the crank pin,

$W_{cp} = d_c \times l_{cp} \times P_{bc}$ **(15)**

L_{cp} =Length of crank pin in mm

P_{bc} = allowable bearing pressure of the crankpin in MPa

E. Design Calculation of Piston Pin

$W_p = d_p \times l_p \times P_{b1}$ **(16)**

L_p = Length of piston pin in mm

P_{b1} = allowable bearing pressure of the piston pin in MPa

F. Diameter of Small End

External Diameter of small end,

D_0=1.3 to 1.7d_p **(17)**

Internal Diameter of small end,

D_i=1.1to1.32d_p **(18)**

G. Design calculation of size of bolts for big end cap

The bolts securing the big end cap are subjected securing to tensile force which corresponds to the inertia force of the reciprocating parts at the top dead center on the exhaust stroke(θ=0),

$$F_I = m_R\times\omega^2\times r\times[\cos\theta+\frac{\cos 2\theta}{l/r}] \quad \textbf{(19)}$$

$$m_R = m_p + m_{rodpp}$$

m_p range is 200 to 300 N/mm^2

m_{rod} range is 250A to 300A.

$m_{rod.pp}$ =0.25m_{rod} **(20)**

Where,

m_R = mass of reciprocating parts(kg)

m_p = mass of piston (kg)

$m_{rod.pp}$ = mass lumped on the piston pin axis (kg)

ω = angular velocity in radian per second r = crank radius

$\frac{L}{r}$ = ratio between connecting rod length and crank Radius

A =cross-sectional area of connecting rod

The following figure 5 is relation of angle and inertia force of reciprocating parts. The design mass m_{rod} refer to a unit area of the piston. The bolts may be made of high carbon steel or nickel alloy steel. θ =0, 30, .720.

Force on the bolts,

$$F_b = \frac{\pi}{4} \times d_{cb}^2 \times \sigma_{tb} \times n_b \tag{21}$$

$$d_b = \frac{d_{cb}}{0.84} \tag{22}$$

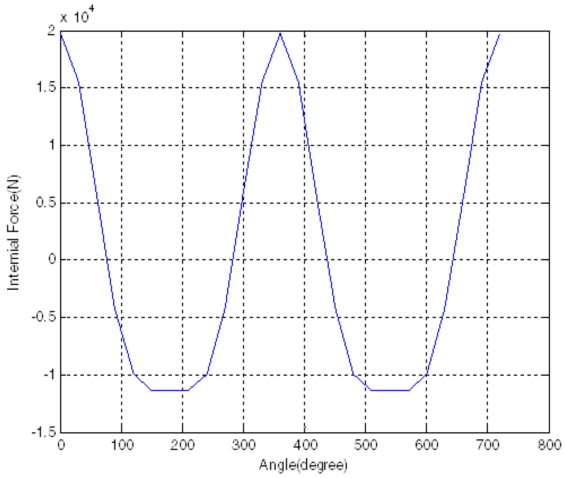

Figure 5. Relation of angle and Inertia Force.

IV. RESULT

The result that is calculated for piston and connecting rod are shown in table 1.

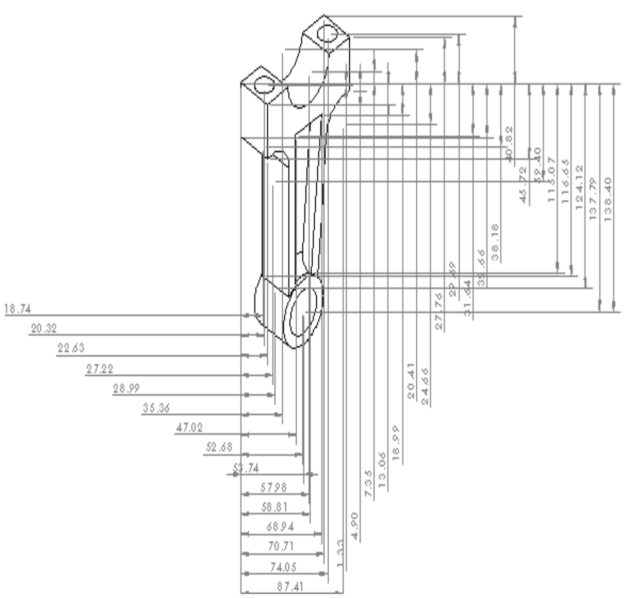

Fig6. Isometric View of Connecting Rod.

TABLE I: Calculated Result for Piston and Connecting Rod

No	Description	Symbol	Value
1	Buckling Load on the connecting rod	W_B	231771.31N
2	Break mean effective pressure	P_b	0.923MPa
3	Thickness of the cylinder wall	t	5.5mm
4	Length of connecting rod	L	174mm
5	Length of piston	L	99.05mm
6	Mass of Piston	m_p	1.425kg
7	Mass of connecting rod	m_R	1.8895kg
8	Maximum gas force on the piston	F	38628.55N

The detail drawing of connecting rod design is as shown in figure 6.

V. STRESS ANALYSIS OF CONNECTING ROD SIMULATION

The stress distributions of connecting rod at the maximum stretch condition are shown in Figure 7.The stresses are high at the connecting rod shank, but stress distribution is relatively uniform. The maximum stress value is 171.99 MPa. The strain and displacement distributions of connecting rod at the maximum stretch condition are shown in figure (8,9). The maximum strain and displacement values are5.597e-004 MPa and 9.851e-002mm. Stress and strain distributions of connecting rod at maximum compression condition are shown in Figs. (**7**,8). The results show that the critical location is at the transition region between the small end and connecting shank at maximum compression condition. Safety factor of CR is shown in Figure (9). The results show that safety factor of CR is 1.5. Fatigue resistance of connecting rod is good. Safety factor of the whole CR is 1.5 which is only a little greater than design safety factor. In order to increase the reliability of CR, some improvement is carried out.

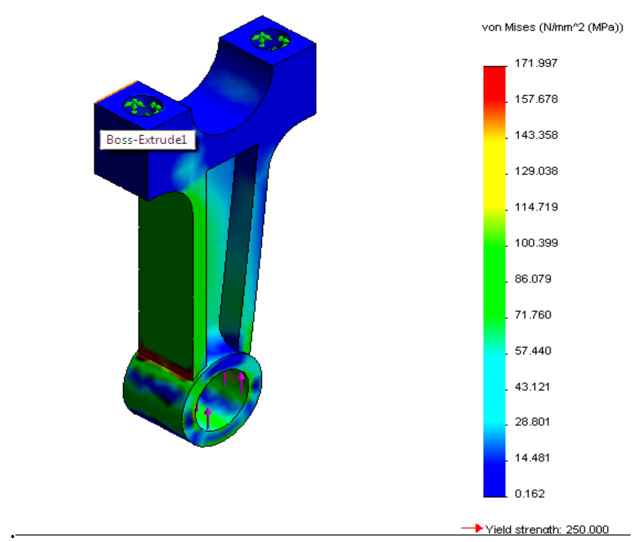

Fig7. Stress distribution of connecting rod.

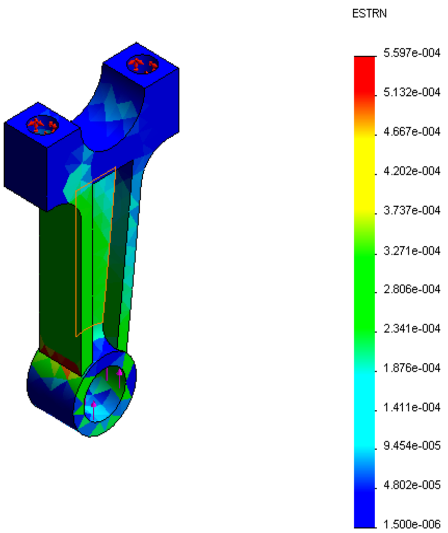

Fig8. Strain distribution of connecting rod.

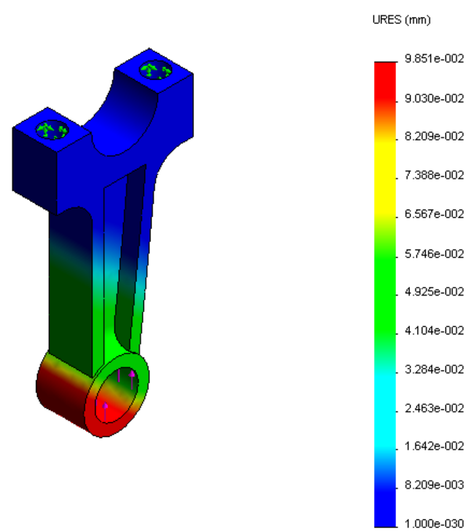

Fig9. Displacement distribution of connecting rod

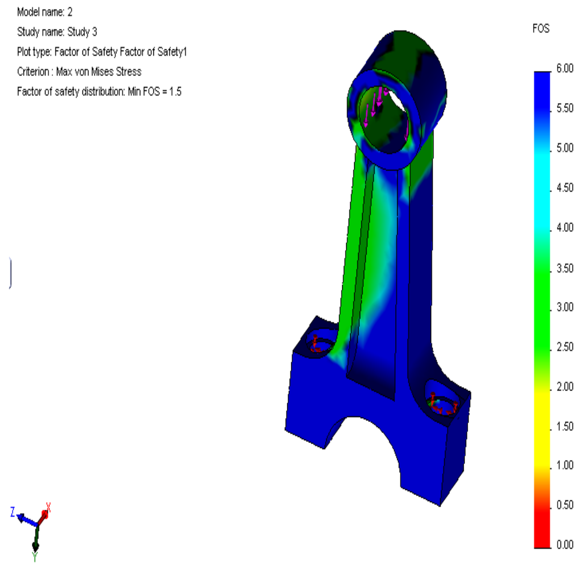

Fig10. Safety factor of connecting rod.

According to the analysis results, the main stress concentration locations are the transition region between small end and connecting rod shank and the I-shaped cross-section at big end. So it is necessary to improve the structure of the two positions.

VI. CONCLUSION

In this paper, not only the connecting rod in Light Truck engine design but also the analyses of the stress effect are applied to connecting rod. In calculating the parameters of the engine take the assumed values, i.e. the length of the connecting rod (L) is taken as 1 to 1.5 times the cylinder bore. In this paper, most of the values are taken as the mean within the range. In this paper, when a connecting rod is designed; it should be taken to have a maximum weight with strength to withstand pressure and inertia forces. In designing the connecting rod, the length of connecting rod has been taken as 174mm for Light Truck engine. The thickness of flange and web of I-section is 8mm.For connecting rod design, I-beam section is considered when the buckling load is evaluated. The stresses are high at the small end, but stress distribution is relatively uniform. The maximum stress and strain values are 171.99 MPa and 5.597e-004 respectively. The critical location is at the transition region between the small end and connecting shank at maximum compression condition.

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