

## Finite Element Analysis of Connecting Rod of IC Engine

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**Abstract.** A connecting rod of IC engine is subjected to complex dynamic loading conditions. Therefore it is a critical machine element which attracts researchers' attention. This paper aims at development of simple 3D model, finite element analyses and the optimization by intuition of the connecting rod for robust design. In this study the detailed load analysis under in-service loading conditions was performed for a typical connecting rod. The CAD model was prepared taking the detailed dimensions from a standard machine drawing text book. Based on the gas pressure variation in the cylinder of an IC engine, the piston forces were calculated for critical positions. MATLAB codes were written for this calculation. Altair Hypermesh and Hyperview were used for pre-processing and post-processing of the model respectively. The finite element analyses were performed using Altair Radioss. The results obtained were compared to a case study for the field failure of the connecting rod. By comparing the induced stress result with the yield strength of the material, the component was redesigned. This was done to save some mass keeping in mind that the induced stress value should be well below the yield strength of the material. The optimized connecting rod is 11.3% lighter than the original design.

### 1 Introduction

The connecting rod is a critical component in internal combustion (IC) engine. The IC engine is the heart of automobiles. As the name suggests connecting rod is the intermediate member connecting the piston and the crankshaft of the IC engine. The piston is the input reciprocating element and the crankshaft is the output rotating element. Therefore the connecting rod is playing the vital role in transmitting the thrust of piston to torque of crankshaft. Thus it is subjected to complex loading such as combination of axial compression and flexural bending.

The connecting rod in IC engines consists of a long shank, a small end (pin-end) and a big end (crank-end) as shown in Figure 1. The upper end of the connecting rod is connected to the piston by the piston pin. As the lower end of the connecting rod revolves with the crankshaft, it is split to permit it to be clamped around the crankshaft. The bottom part, or cap, is made of the same material as the rod and is attached by two bolts. The cross-section of the shank may be rectangular, circular, tubular, I-section or H-section. Generally circular section is used for low speed engines while I-section is preferred for high speed engines [1].

Finite element analysis (FEA) has become common place in recent years, and is now the basis of a multibillion dollar per year industry. Numerical solutions to even very complicated stress problems can now be obtained routinely using FEA, and the method is so important that even introductory treatments of mechanics

of materials such as these modules should outline its principal features [2].

In recent years, more emphasis has been placed on higher vehicle fuel efficiency. Optimization of connecting rods in an engine is critical to fuel efficiency. Proper optimization of this component, however, necessitates a detailed understanding of the applied loads and resulting stresses under the in-service conditions.

In this work, based on the cylinder pressure variation of a typical engine, piston forces were calculated for the critical crank angles. The Finite Element Analyses (FEA) were carried out for different load cases and finally saved some material by iterative intuitive optimization.

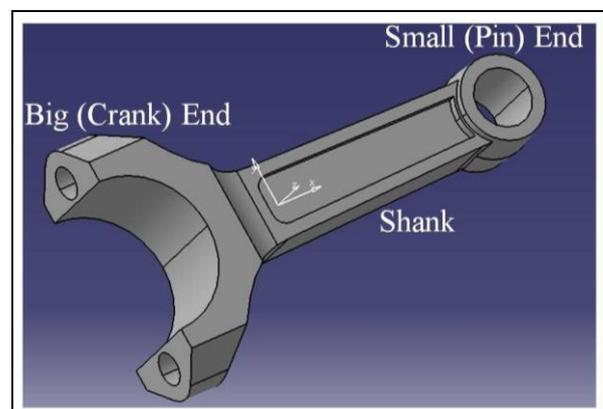


Figure 1. Details of Connecting Rod

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The calculation of piston forces for different crank angles and corresponding axial and normal forces on the connecting rod is given in the section ‘Load Analyses.’ In the section ‘Modeling and Analyses’, the detailed procedure for finite element modeling and analyses is described. Based on the resulting stress and displacement plots the optimization by intuition of the connecting rod is explained in the section ‘Results and Discussion.’ The concluding summary of this work is presented in ‘Conclusion’.

## 2 Load Analyses

As shown in Figure 2, the experimental data has been plotted in the cylinder pressure versus crank angle for one complete cycle of a typical engine [3].

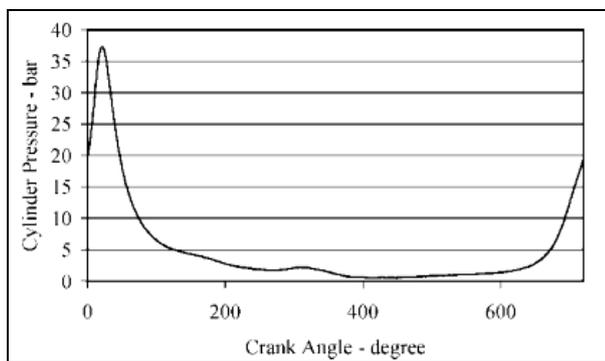


Figure 2. Cylinder Pressure versus Crank Angle

It is found that maximum gas pressure is 37.3 bar for crank angle of  $30^\circ$  from Figure 2. In general the connecting rod is subjected to maximum axial load at the crank angle of  $0^\circ, 180^\circ, 360^\circ, 540^\circ$  and  $720^\circ$ , and maximum bending load at the crank angle of  $90^\circ, 270^\circ, 450^\circ$  and  $630^\circ$ . The piston forces at these crank angles are calculated based on the pressure variation on the piston from the plot as shown in Figure 2. Out of these angular positions of the crank, the pressure on the piston is high at  $0^\circ$  or  $720^\circ, 30^\circ$  and  $90^\circ$  or  $630^\circ$ . The corresponding cylinder pressures are 18 bar, 37.3 bar and 8 bar respectively. Table 1 represents the dimensions of the engine considered. From Table 1, the piston forces were calculated for above mentioned values of cylinder pressures. The relationship between crank angle ‘ $\theta$ ’ and the connecting rod angle ‘ $\phi$ ’ is obtained and the corresponding axial forces and normal forces on the connecting rod were calculated using MATLAB.

Table 1. Dimensions of the Engine.

Particulars	Dimensions [mm]
Crank Length (R)	50
Length of Connecting Rod (L)	130
Diameter of Piston	75

The detailed calculation was done based on the load diagram as shown in Figure 3.

For Crank angle  $\theta = 0^\circ$ , the cylinder pressure = 18 bar, the piston force,  $P = 8 \text{ kN}$ , the axial force  $P_A = 8 \text{ kN}$  and the normal force  $P_N = 0$ .

For Crank angle  $\theta = 30^\circ$ , the cylinder pressure = 37.3 bar, the piston force,  $P = 16.56 \text{ kN}$ , the axial force  $P_A = 16.2 \text{ kN}$  and the normal force  $P_N = 3.17 \text{ kN}$ .

For Crank angle  $\theta = 90^\circ$ , the cylinder pressure = 8 bar and the piston force,  $P = 3.5 \text{ kN}$ , the axial force  $P_A = 3.26 \text{ kN}$ , the normal force  $P_N = 1.36 \text{ kN}$ .

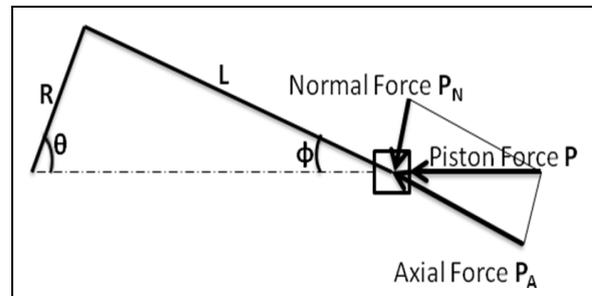


Figure 3. Load Diagram

## 3 Modeling and Analyses

The 3D CAD model of the connecting rod was prepared using the commercial software CATIA-V5 for the IC engine whose dimensions are given in Table 1, and for the detailed dimensions, a machine drawing text book was referred [4]. The 3D CAD model is shown in Figure 1. The finite element modeling and analyses were carried out for different load cases at critical crank angles by using commercial package Altair Hypermesh and Radioss [5].

In pre-processing, the CAD model was meshed using tetrahedron elements. The total number of nodes is 18509 and total number of elements is 75900. The big end of the connecting rod was constrained whereas the forces, as calculated in the previous section, were applied at the small end for three different load cases as shown in Figure 4 to 6.

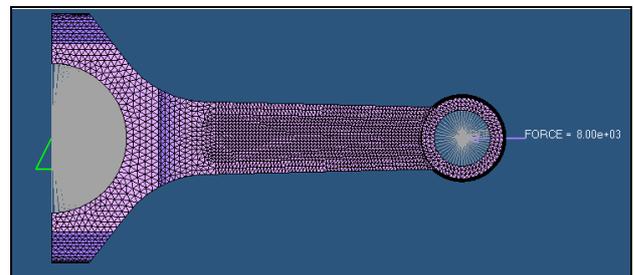


Figure 4. Finite Element Model for Crank angle of  $0^\circ$

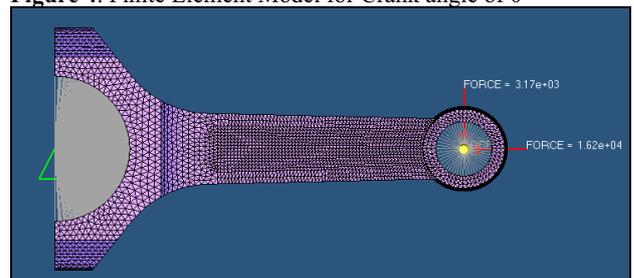


Figure 5. Finite Element Model for Crank angle of  $0^\circ$

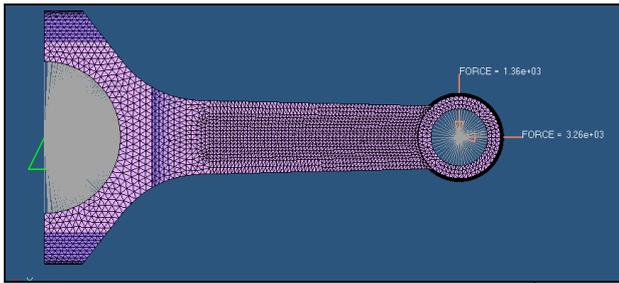


Figure 6: Finite Element Model for Crank angle of 90°

### 4 Results and Discussion

The results were observed in the post-processor, Altair Hyperview. The stress and displacement plots for three different load cases at respective crank angles are shown in Figure 7 to 12.

At the crank angle of 0°, the connecting rod is subjected to only axial force of 8 kN. From the stress plot as shown in Figure 7, the maximum tensile stress is 17.55 MPa and maximum compressive stress is 38.82 MPa. From Figure 8, the maximum displacement is  $9.547 \times 10^{-3}$  mm.

At the crank angle of 30°, the connecting rod is subjected to axial force of 16.2 kN and normal force of 3.17 kN. From the stress plot as shown in Figure 9, the maximum tensile stress is 134.3 MPa and maximum compressive stress is 227.3 MPa. It has a maximum displacement of 0.304 mm as shown in Figure 10. In this case the bending stress is dominating. The fillet region near big end of the connecting rod is experiencing maximum stress.

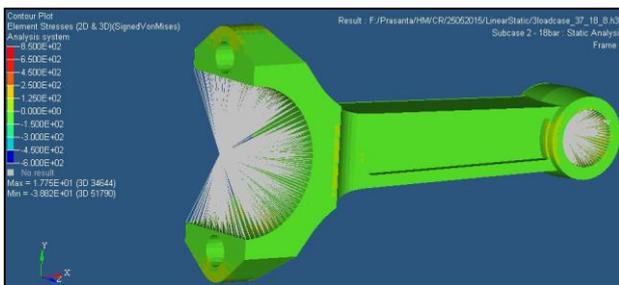


Figure 7: Stress plot for Crank angle of 0°

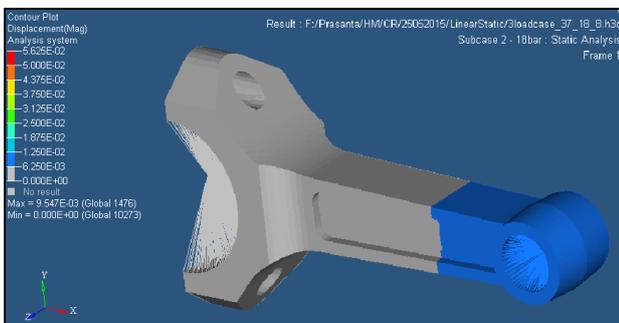


Figure 8: Displacement plot for Crank angle of 0°

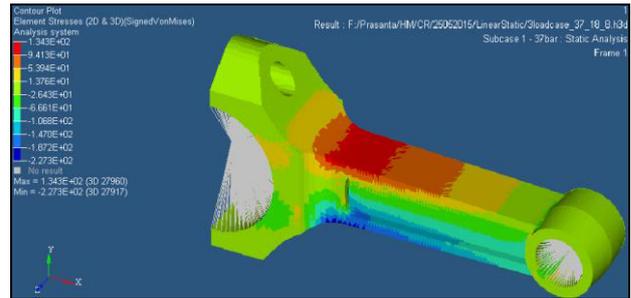


Figure 9: Stress plot for Crank angle of 30°

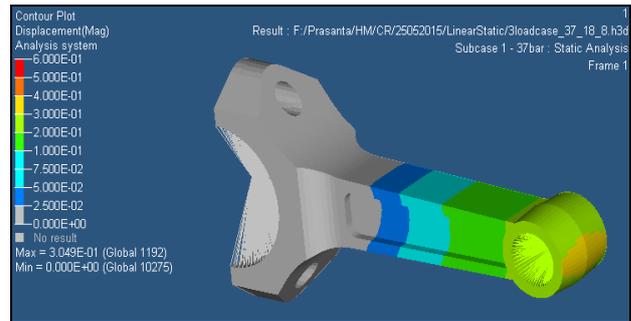


Figure 10: Displacement plot for Crank angle of 30°

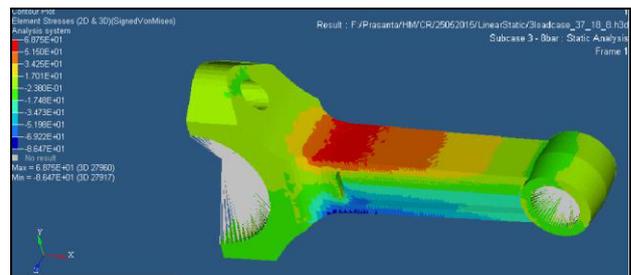


Figure 11: Stress plot for Crank angle of 90°

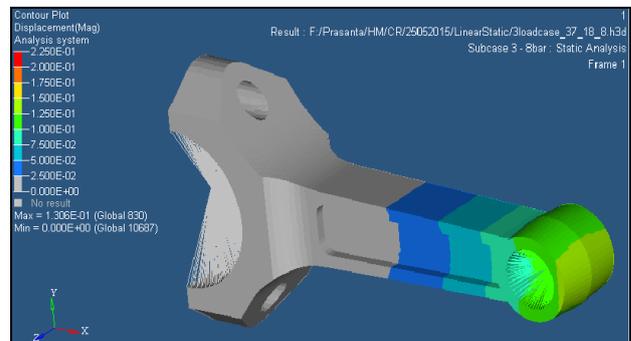


Figure 12: Displacement plot for Crank angle of 90°

At the crank angle of 90°, the connecting rod is subjected to axial force of 3.26 kN and normal force of 1.36 kN. The maximum tensile stress is 68.75 MPa and the maximum compressive stress is 86.47 MPa in this case from the stress plot as shown in Figure 11. It has a maximum displacement of 0.136 mm from the displacement plot as shown in Figure 12. Here also the bending is the dominating one.

The cast steel (AISI 1045) was considered as the connecting rod material [2]. Its properties are shown in Table 2.

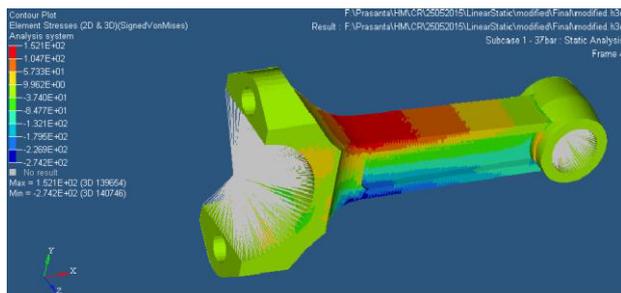
**Table 2.** Material Properties of Cast Steel (AISI 1045)

Material	Yield [Mpa]	Ultimate [Mpa]	% Ductility
AISI	449.32	648.03	225

It has been observed that out of these three load cases, second case, that is, at the crank angle of  $30^0$  where the cylinder pressure is maximum of 37.3 bar, is the critical one. In this case the induced maximum stress is almost half of the yeild strength of the material. Hence there is an opportunity to optimize the size of connecting rod to save some mass ultimately to increase the vehicle fuel efficiency.

#### 4.1 Optimization

The modified CAD model was prepared, meshed into tetrahedron elements, the loads and boundary conditions were applied and then FE analysis was carried out for the critical load case only. The thicknesses of both web and flanges have been reduced by 4 mm in two iterations. The stress plot for final proposed model is shown in Figure 13.



**Figure 13.** Stress plot for Optimized Model at Crank angle of  $30^0$

By iterative, intuitive optimization, the mass of the connecting rod is reduced from 0.592 kg to 0.532 kg. The induced maximum tensile and compressive stresses are 152.1 MPa and 274.2 MPa. Which are well below the yield strength of the material.

#### 5 Conclusion

The 3D CAD model of the connecting rod was prepared using the modeling software CATIA-V5 for a typical IC engine, and referring to a machine drawing text book for detailed dimensions. The piston forces were calculated, using MATLAB, for the critical crank angles based on the cylinder pressure variation for the engine. The finite element modeling and analyses were carried out for different load cases at critical crank angles by using commercial package Altair Hypermesh and Radioss. Out of three load cases, the maximum stress induced is 227.3 MPa in the second load case at the crank angle of  $30^0$  where the connecting rod is subjected to highest cylinder pressure of 37.3 bar. This stress is much lower than the yield strength of the material. Therefore, the connecting rod was redesigned to save some material. The final optimized connecting rod is 11.3% lighter than the

original design. The maximum stress induced is 274.2 MPa which is also well below the yield strength of the material.

#### References

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