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Finite Element Analysis of a Diesel Engine Connecting Rod

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Abstract

The connecting rod forms an integral part of an internal combustion engine. It acts as a linkage between piston and crank shaft. The main function of connecting rod is to transmit the translational motion of piston to rotational motion of crank shaft. The function of the connecting rod also involves transmitting the thrust of the piston to the connecting rod. Connecting rod used in automotive engines is a critical component which comes under the influence of different types of loads in operation. Fatigue loading is one of the prime causes contributing to its failure. Failure and damage are also more in connecting rod, so stress analysis in connecting rod is very important. In this study, detailed load analysis was performed on connecting rod, followed by finite element method in Ansys . In this regard, in order to calculate stress in different part of connecting rod, the total forces exerted connecting rod were calculated and then it was modeled, meshed and loaded in Ansys software. The maximum stresses in different parts of connecting rod were determined by analysis.

Keywords: Connecting rod, FEA, Fatigue analysis, Stress concentration factor, Ansys.

Introduction

The internal combustion engine is an engine in which the combustion of a fuel (normally a fossil fuel) occurs with an oxidizer (usually air) in a combustion that is an integral part of the working fluid flow circuit. In an internal combustion engine (ICE) the expansion of the high-temperature and high-pressure gases produced by combustion apply direct force to some component of the engine. The force is applied typically to pistons, turbine blades, or a nozzle. This force moves the component over a distance, transforming chemical energy into useful mechanical energy. Some of the important components of the internal combustion engine are Cylinder, piston, piston rings, connecting rod, crankshaft etc.

Conversion of the piston's reciprocating motion into the rotational motion of the crankshaft is the major function of the connecting rod. Since the connecting rod has two ends, one of its ends is connected to the piston by the piston pin, and the other end moves in a circular shape or revolves with the crankshaft and is separated in a way that it allows it to get clamped around the crankshaft as shown in figure 1. There are different type of loads acting on connecting rod during operation, i.e. axial compressive load, bending loads and inertia loads due to reciprocating masses.

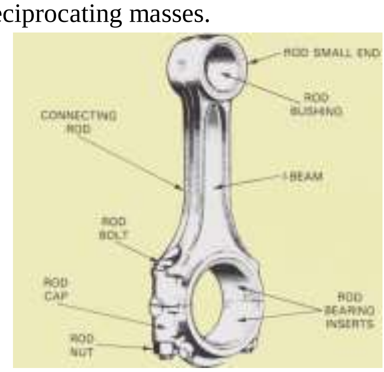

Figure 1 Connecting Rod of an I.C. Engine

Problem definition and methodology

The diesel engine connecting rod is a high volume production critical component. It connects reciprocating piston to rotating crankshaft, transmitting the thrust of piston to the crankshaft. Every engine requires at least one connecting rod depending upon the number of cylinders in the engine. For the analysis of I.C. engine connecting rod the most critical area is considered and accordingly

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the two dimensional model of connecting rod is formed.

Table 1 Major parameters in connecting rod assembly

Parameters	Value
Crankshaft radius	56.5 mm
Connecting rod length	232 mm
Piston diameter	114.3 mm
Mass of piston assembly	1.71 kg
Mass of connecting rod	1.875 kg
Distance of C.G from crank end center	71 mm

Table 2 Major specifications of a Diesel engine

Type of engine	Four stroke single cylinder , vertical, compression ignition engine
No. ofcylinders	one
Bore (mm)	114.3
Stroke (mm)	116
Rated RPM	1500
Swept volume(cc)	1230
Max Gas Pressure	8.044 MPa

Materials and methods

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Selection of materials in engine construction is rather complex and is based on trade off amongst conflicting requirement of high strength, low density and ease of fabrication or processing. The material used in various parts of vehicle structures generally are selected by different criteria. The material used in the connecting rod is alloy steel and it has the following properties as shown in Table 3.

Material	C -55 Hardened and Tempered Alloy Steel
Density	7.87 g/cm^3
Yield stress `	570 MPa
Ultimate tensile strength	850 MPa
Young's modulus	206*10^3 MPa

Finite Element Model of the Connecting Rod

The CATIA model was imported to the finite element analysis pre-processor software to prepare the the finite element model of the connecting rod.

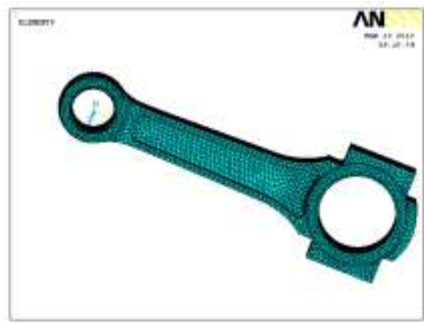

Figure 2; Meshed model of the connecting rod.

Figure 2 shows the meshed model of connecting rod considering the tetrahedral elements throughout the rod and without any hexahedron elements. The Connecting Rod consists of pin end, crank end, and rod. All these features are considered for the development of the finite element model.

Analytical load Calculations of the Connecting Rod

During the operation of the engine, the connecting rod undergoes a combination of axial and bending stresses acting on the rod in operation. The axial stresses are produced due to cylinder gas pressure (compressive only) and the inertia force arising in account of reciprocating action (both tensile as well as compressive), where as bending stresses are caused due to the centrifugal effects. The force acting on the small end of connecting rod is a combination of gas forces and inertia forces.

Forces on connecting rod small end = Gas forces - Inertial Forces of piston **(1)**

Gas force acting on the piston surface = Peak firing pressure × $\pi/4 \times D^2$ **(2)**

Using the values of bore and peak firing pressure from table 2 we get, Gas force = 82538.08 N

This inertia forces due to reciprocating parts can be calculated by multiplying the mass of the reciprocating parts and the acceleration of the reciprocating parts. Mass of the reciprocating parts includes mass of piston, piston rings, gudgeon pin and small end of the connecting rod.

Inertia forces of piston = $m_p \times a_p$ **(3)**

Acceleration of reciprocating parts = $r\omega^2$ (cos θ + cos (2θ/n)) **(4)**

Assuming that the maximum acceleration occurs corresponds to TDC position where the angle is 0° or 360°. The second term in acceleration equation is neglected considering the length of the connecting rod as infinite and in that case the motion will approach to simple harmonic type. The aim of calculating these forces is to find the maximum force with respect to fatigue. The magnitude of angular velocity used for calculating acceleration of reciprocating parts is taken with respect to top engine rated speed which is 1500 rpm = 157.07 rad/s. So, ω = 157.07 rad/s.

Using the values of crank shaft radius, connecting rod length from table 1 and ω value we get,

Acceleration of reciprocating parts = 1394.08 m/sec^2

Inertia forces of piston =3083.70 N

Therefore, from equation 1 Forces on connecting rod small end =79454.37 N

The net force considered for performing the static analysis was 79454.37 N. This was applied as axial force at the small end center of connecting rod.

Calculating forces exerted on pin end:-

Maximum Inertia force is considered:

$$F_j = -(m_j)\, r\omega^2 (\cos\theta + \cos(2\theta/n)) \quad \textbf{(5)}$$

By using the values from table 1&2, we get

F_j= - 3102.84

Calculating forces exerted on crank end:-

The combustion pressure force doesn't have effect on crank end, but it is affected by inertia force. F_{jr} max is the maximum inertia force exerted on crank end of connecting rods.

The inertia force exerted on crank end was calculated as

$$F_{jr} = -r\omega^2 [(m_p + m_{cr,p}) + (m_{cr,c} - m_c)] \quad \textbf{(6)}$$

Where **m_p** = Mass of the piston assembly (kg),

$m_{cr,c}$ =concentrated mass of connecting rod on the crank end,

$$\mathbf{m_{cr,c}} = 0.725 \times m_{cr} \quad \textbf{(7)}$$

$m_{cr,p}$= concentrated mass of connecting rod on the pin end.= $0.275 \times m_{cr}$ **(8)**

m_c = concentrated mass of crankshaft on crank end.

$$m_c = 0.25 \times m_{cr} \quad \textbf{(9)}$$

By using the table 1 and equations (7), (8), (9)

Therefore from equation (6), we get , F_{jr} = -4343.00 N

Results and discussion

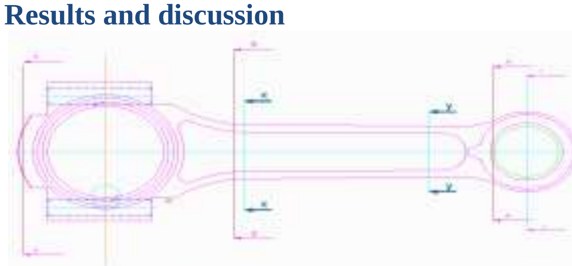

Figure 3: Various sections consider for static stress analysis

A few geometric locations were identified on the connecting rod at which stresses were traced over the entire load cycle to obtain the stress at

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various locations. Some of these locations are shown in Figure 3 and include high stressed regions of the crank end, the pin end at the oil hole, the shank and at transitions to the shank at the crank and piston pin ends.

Compression loading condition

The compression loading for the crank end and piston end is assumed to have a loading over the 180^0 contact arc. A force of 79.454 KN was also used in compression. The compression pressure applied on crank end while for the pin end is constrained for compression loading in the crank end, the piston pin end is constrained in all six directions (three for translational and three for rotational) through an 180^0 contact arc. Uniform loading was applied at crank end through 180^0 contact arc.

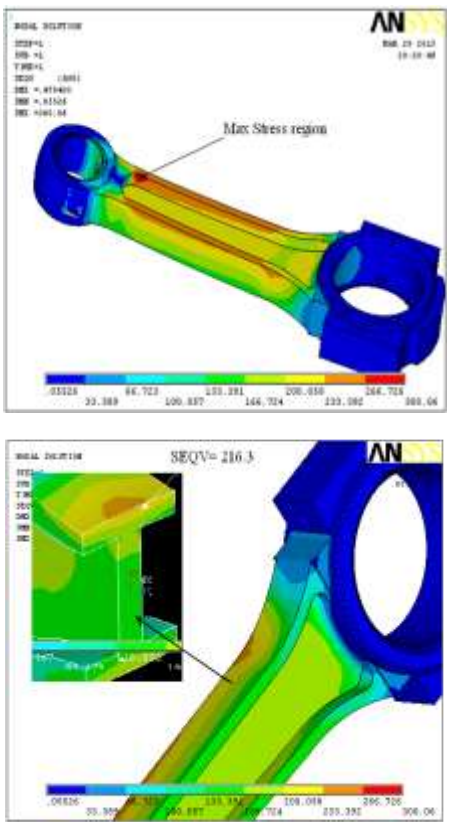

Figure 4; Von misses stress contours for pin end compression and crank End restrained

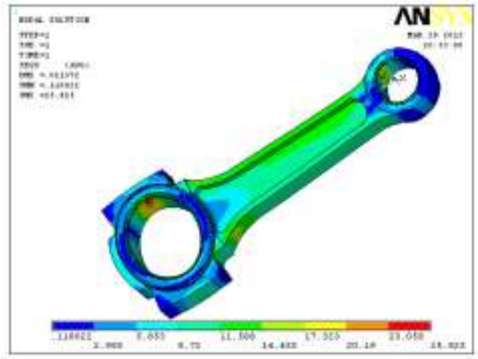

Figure 5; Von misses stress contours for crank end tension and pin End restrained

Table 4 various stress acting on connecting rod design sections

Design Parameter	Stress value
Small end	
I-I section	6.607 N/mm^2
A-A section External fibers Internal fibers	 8.4 N/mm^2 10.34 N/mm^2
Big end	
II-II section	10.57 N/mm^2
Connecting rod shank	
σ_{max} in rocking plane	216.3 N/mm^2

Fatigue Life Prediction of Connecting Rod

In materials science, fatigue is the progressive and localized structural damage that occurs when a material is subjected to cyclic loading. A mechanical component loaded with a periodic stress that oscillates between some limits is subjected to stresses called repeated, alternating, or fluctuating stresses. Often the machine members fail under the action of repeated or fluctuating stresses. The most distinguishing characteristic of these failures is that the stresses have been repeated a very large number of times. Hence, the failure is called a fatigue failure. A fatigue failure begins with a small crack. Once the crack is initiated, the stress concentration effect becomes greater and the crack progress more rapidly.

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As the stressed area decreases in size, the stress increases in magnitude until, and the remaining area finally fails suddenly.

Fatigue Life Prediction

The Stress Life (SN) theory was employed to evaluate the connecting rod fatigue life. In order to perform the fatigue study, the finite element results should be combined to obtain the alternate and mean stresses for each operating condition.

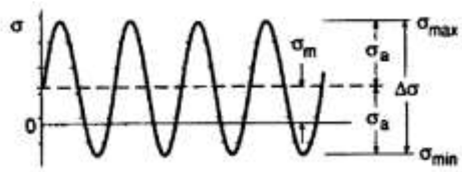

Figure 6; Nomenclature for constant amplitude cyclic loading.

The alternate and mean stresses were calculated for each operating condition, combining the finite element results as explained below

$$\sigma_a = (\sigma_{compression} - \sigma_{tensile}) / 2 \qquad (10)$$

$$\sigma_m = (\sigma_{compression} + \sigma_{tensile}) / 2 \qquad (11)$$

Goodman Diagram

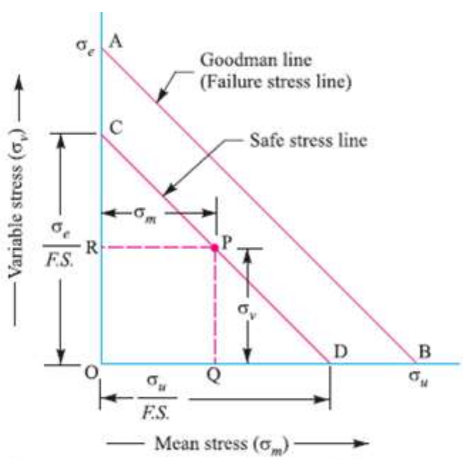

Figure 7; Good man diagram

Line AB connecting endurance limit and ultimate strength is called Goodman's failure stress line. If a suitable factor safety applied to endurance and ultimate strength, a safe stress line CD may be drawn parallel to the line AB.

The value of notch sensitivity factor (q) can be calculated using the notch root radius 13 mm from the design data hand book by Dr K Lingaiyya reference figure 4-13 (6) Notch sensitivity factor is 0.8.

Fatigue concentration factor K_f can be calculated by equation

$$\frac{1}{F.S.} = \frac{\sigma_m}{\sigma_u} + \frac{\sigma_v \times K_f}{\sigma_e \times K_b \times K_{sur} \times K_{sz}}$$

By using the design factors values k_b , k_{sur} , k_{sz} from design data hand book and equations (11),(12) , we get ,

Factor of safety = 1.34

From the Good man diagram, the factor of safety obtained was 1.34. The fatigue factor acceptable criterion is 1.34. By analyzing the numerical results and established acceptable criteria, we can conclude that no connecting rod fatigue failures are expected for these load levels.

Predicting the Life of the Connecting Rod Material from the S-N Curve

Stress Ratio is -1 for reversed load i.e. maximum compressive stress to the maximum tensile stress.

From standard alloy steel S-N diagram, using the alternate stress (σ_a) and Stress Ratio, we can predict the life of the component.

Table 6 Fatigue cycles calculation of a connecting rod

Parameter	Value
Load	79.45 KN
Static stress , N/mm²	300.06 N/mm²
Fatigue life , cycles	10^7

By analyzing the results and established acceptable criteria, we can conclude that no

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connecting rod fatigue failures are expected for these life cycles of the component.

Conclusion

The following conclusions are drawn from the work:

1. The maximum compressive stress was obtained between small end and shank of connecting rod.
2. Fatigue life of the connecting rod was determined to be around 10^7 cycles, with a factor of safety of 1.34, which is satisfactory for mechanical components.
3. The software also reveals the importance of the varying I - cross section which is provided for uniform stress distribution over the entire web of the connecting rod.

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