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Original Research Article

Fatigue and Structural Analysis of Connecting Rod's Material Due to (C.I) Using FEA

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Abstract

In this research connecting rod is one of the most important part in engine assembly which transfers energy from piston to crankshaft and convert the linear, reciprocating motion of a piston into the rotary motion of a crankshaft. The connecting rod primarily undergoes tensile and compressive loading under engine cyclic process. The forces acting on connecting rod are:- forces due to maximum combustion pressure and force due to inertia of connecting rod and reciprocating mass[1]. From the viewpoint of functionality, connecting rods must have the highest possible rigidity at the lowest weight. This research addresses the computation of the strength and distortion characteristics of a connecting rod. Finite element method is used to analyze the connecting rod's stress and deformation using Ansys. For this case, a fatigue and structural analysis will be performed. The axial compressive load is greater than the axial tensile load. Therefore, the design is only analyzed for the axial compressive loads. This analysis shows the importance of the solution of the connecting rod big end distortions in view of the changes in the bearing clearance at the most important variants of the stress[9, 11]. This variant is frequently overlooked and primary importance is attached to the strength stress which however does not have to be limiting one and Fatigue analysis and determining if design of connecting rod is safe or not.

Key Words: Connecting Rod, Structural Analysis, steel, Gas load, fatigue, FEA

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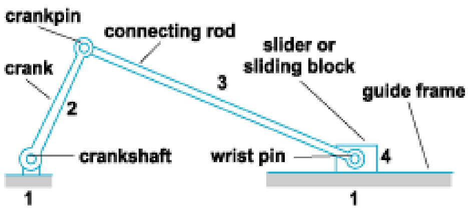
1. Introduction

Connecting rod is one of the most important part in engine assembly acting as a link between crankshaft and piston. The connecting rod primarily undergoes tensile and compressive loading under engine cyclic process. So Connecting rod has to be designed to withstand these cyclic loading conditions.Connecting rod is one of the inversions of slider crank mechanism by keeping cylinder fixed.In a single slider crank chain, links 1 and 2, links 2 and3, links 3 and 4 forms turning pair while link 4 and 1 form sliding pair[3].
The link 1 corresponds to frame of engine, link 2 corresponds to crank, link 3 corresponds to connecting rod and link 4 corresponds to piston.
Connecting rods transfer energy from pistons to crankshafts and convert the linear, reciprocating motion of a piston into the rotary motion of a crankshaft.

Fig 1.1 Single slider crank chain



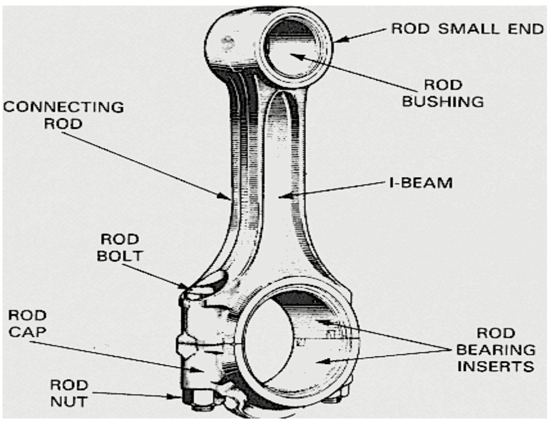


Fig 1.2 Connecting rod

The main aim of the research is to determine the Von Misses stresses, Maximum and Minimum Principle stress, Normal Stress, Directional Deformation, Vector Principle Stress, Stress Intensity, Total Deformation, Fatigue Analysis and determining if design of connecting rod is safe or not. If the existing design shows the failure, then suggest the minimum design changes in the existing Connecting rod. A lot has been done and still a lot has to be done in this field. In this research, only the structural FEA Analysis of the connecting rod has been performed by the use of the software (ANSYS).

2. Objectives

The main objective of this study is to investigate the strength behavior of the connecting rod during the engine operation. The step by step objectives are as following:

2.1 A geometrical model for connecting rod in Solidworks.

2.2 To check whether or not connecting rod material takes the structural stress induced due to gas load.

2.3 Investigate the maximum stress of connecting rod using Ansys for the worst case i.e. when maximum force is acting on connecting rod.

2.4 Is thermal stress severe?

This submission shows the implementation of the FEM software for the assessment of the strength and distortion characteristics of a connecting rod.

3. Literature Review

Webster *et al.* (1983) performed three dimensional finite element analysis of a high-speed diesel engine connecting rod. For this analysis they used the maximum compressive load which was measured experimentally, and the maximum tensile load which is essentially the inertia load of the piston assembly mass. The load distributions on the piston pin end and crank end were determined experimentally. They modeled the connecting rod cap separately, and also modeled the bolt pretension using beam elements and multi point constraint equations.
Folgar *et al.* (1987) developed a fiber FP/Metal matrix composite connecting rod with the aid of FEA, and loads obtained from kinematic analysis. Fatigue was not addressed at the design stage. However, prototypes were fatigue tested. The investigators identified design loads in terms of maximum engine speed, and loads

at the crank and piston pin ends. They performed static tests in which the crank end and the piston pin end failed at different loads. Clearly, the two ends were designed to withstand different loads.

Athavale and Sajanpawar (1991) modeled the inertia load in their finite element model. An interface software was developed to apply the acceleration load to elements on the connecting rod depending upon their location, since acceleration varies in magnitude and direction with location on the connecting rod. They fixed the ends of the connecting rod, to determine the deflection and stresses. This, however, may not be representative of the pin joints that exist in the connecting rod. The results of the detailed analysis were not discussed, rather, only the modeling technique was discussed. The connecting rod was separately analyzed for the tensile load due to the piston assembly mass (piston inertia), and for the compressive load due to the gas pressure. The effect of inertia load due to the connecting rod, mentioned above, was analyzed separately.

In a published SAE case study (1997), a replacement connecting rod with 14%weight savings was designed by removing material from areas that showed high factor of safety. Factor of safety with respect to fatigue strength was obtained by performing FEA with applied loads including bolt tightening load, piston pin interference load, compressive gas load and tensile inertia load. The study lays down certain guidelines regarding the use of the fatigue limit of the material and its reduction by a certain factors into account for the as-forged surface. The study also indicates that buckling and bending stiffness are important design factors that must be taken into account during the design process. On the basis of the stress and strain measurements performed on the connecting rod, close agreement was found with loads predicted by inertia theory. The study also concludes that stresses due to bending loads are substantial and should always be taken into account during any design exercise.

Table 5.1 Design Specifications of connecting rod

1.	*Length of connecting rod*	*380 mm*
2.	Thickness of flange	7 mm
3.	Width of section	28 mm
4.	Depth of section	35 mm
5.	Diameter of bolt	12 mm
6.	Length to diameter ratio at piston end	1.3
7.	Length to diameter ratio at crankshaft end	2.0
8.	Young's modulus	2.1 X 10^5 MPa
9.	Poisson's ratio	0.3
10.	Density of material	8000kg/m^3

4. Materials used In connecting rod

Connecting Rods can be made from various grades of structural steel, aluminum, and titanium. Steel rods are the most widely produced and used type of connecting rods. Their applications are best used for daily drivers and endurance racing due to their high strength and long fatigue life[7]. The only problem with using steel rods is that the material is extremely heavy, which consumes more power and adds stress to the rotating assembly. Performance steel rods can be made from 4340 and even 300M grade steel. The tensile strength, yield strength, and hardness of 4340 steel depends on the temperature at which the steel is forged, and how the steel is heat treated. Variations in the tempering

temperature and quenching procedure can produce extremely different results with tensile strength and yield strength.
Various types of structural steel are :

4.1. Carbon steels

4.2. High strength low alloy steels

4.3. Corrosion resistant high strength low alloy steels

4.4. Quenched and tempered alloy steel

5. Design and calculation

5.1 Design specifications

6. Determination of forces on connecting rod

6.1 Functional specification of connecting rod

Table 6.1 Forces On Connecting Rod

1.	*Speed of IC Engine*	*1800 r.p.m*
2.	Bore Diameter	100mm
3.	Mass of reciprocating parts	2.25 kg
4.	Factor of safety	6
5.	Young's modulus	2.1 X 10^5 MPa
6.	Poisson's ratio	0.3
7.	Density of material	8000kg/m^3
8.	Wall pressure for piston rings(oil rings)	0.137 MPa
9.	Number of rings	3
10.	Coefficient of friction	0.05
11.	Explosion pressure	3.15 MPa
12	Piston pin diameter	29 mm
13.	Crank pin diameter	44 mm

7. Forces acting on connecting rod

Following are the forces acting on connecting rod

(i) Force on the piston due to gas pressure.

(ii) Force due to inertia of the connecting rod and reciprocating mass.

(iii) Force due to friction of the piston rings and of the piston

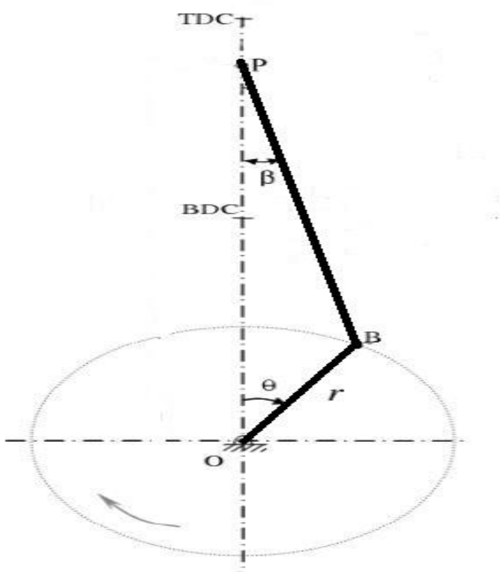


Fig 7.1 Forces Acting On Connecting Rod

7.1 Calculation of forces

7.1.1 Force due to gas pressure

Maximum force due to gas pressure, $F_a = \pi d^2 p_e/4$
Where,
P_e = explosion pressure.
d = bore diameter
F_a = 24,740 N

7.1.2 Inertia Force Due To Reciprocating Mass

$F_i = M\,\omega^2 r\,(\cos\Theta + r\cos\Theta/l)$

Where,
M =mass of (piston and rings + Piston pin + 1/3 rd of connecting rod)
d =bore diameter, mm
ω= angular speed, rad/s
r = crank radius, mm
l= length of connecting rod, mm

Maximum inertial force is at ϴ=0 (at top dead centre)
F_i = 1756 N

7.1.3 Frictional Force

The force of friction due to piston rings and piston is :
$F_f = h\pi d i p_r \mu$.
where
h= axial width of rings.
i = number of rings.
P_r = pressure of rings.
µ.= Coefficient of friction.
F_f= 4099 N

7.1.4 Force acting on piston

$F = F_{gas} + F_{inertia} - F_{friction}$
F=22396 N

Force Acting On Connecting Rod

$F_c = F/\cos\beta$
At top dead centre β=0
F_c = 22,396 N

8. Material properties of connecting rod.

8.1 Material properties of connecting rod

9. Structural analysis results

9.1 Boundry condition here section A represents small end of connecting where piston will be attached and section B represents the big end where crankshaft will be attached. Compressive load is acting along X axis on the surface highlighted by red colour.

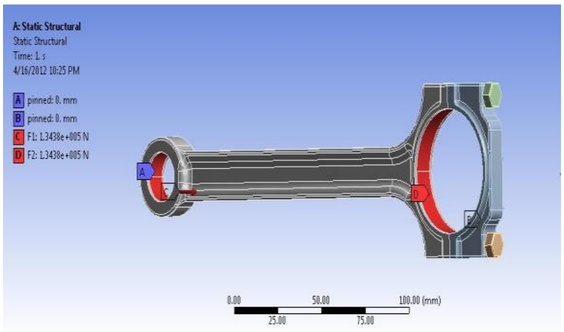


Fig 9.1 Boundry Condition section A and section B.

9.2 Meshed model in ansys

Mesh generation is one of the most critical aspects of engineering simulation. Too many cells may result in long solver runs, and too few may lead to inaccurate results. ANSYS Meshing technology provides a means to balance these requirements and obtain the right mesh for each simulation in the most automated way possible [8].

ANSYS Meshing technology has been built on the strengths of stand-alone, class-leading meshing tools

Unit system used	*Metric*
Material used	Structural steel
Compressive ultimate strength	610 MPa
Compressive yield strength	530 MPa
Reference temperature	22 ºC
Young's modulus	2.1 x 10^5 MPa
Bulk modulus	1.75 x 10^5 MPa
Shear modulus	80769 MPa
Poisson's ratio	0.3
Factor of safety	6
Total load acting	22396 X 6 = 134377 N

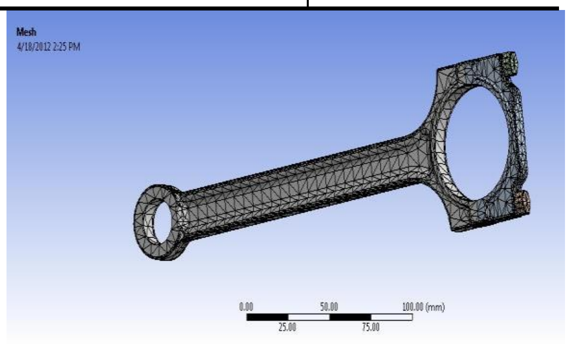


Fig 9.2.1 Meshed Model have element 413465 and nodes 511298.

9.3 Normal stress distribution

Normal stress refers to the stress caused by forces that are perpendicular to a cross-section area of the material.

Maximum value of normal stress = 227.03 MPa

Minimum value of normal stress = -215.71 MPa

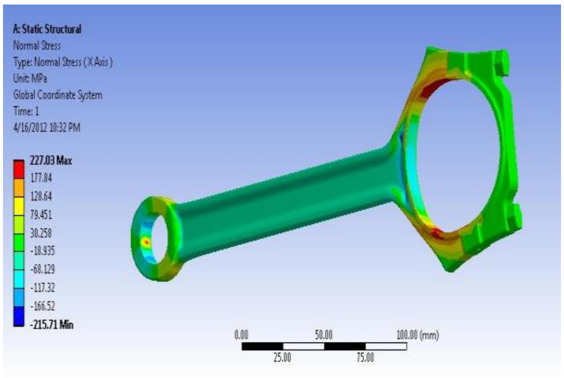


Fig 9.2.2 Normal Stress Distribution perpendicular to a cross-section area of the material.

9.3 Maximum principal stress distribution

Within stressed body, there always exists three mutually perpendicular planes on each of which the resultant stress is normal stress.

Max. value of Maximum principle stress = 411.32 MPa

Min. value of Maximum principle stress = -12.904 MPa

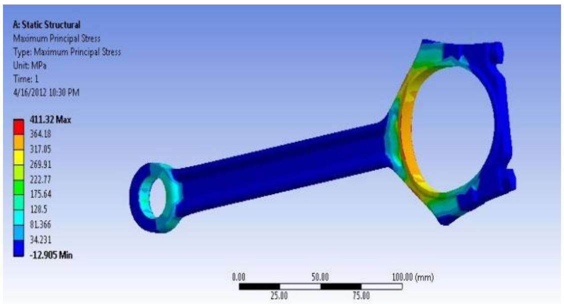


Fig 9.2.3 Maximum Principal Stress Distribution in connecting rod.

9.4 Minimum principle stress distribution in connecting rod

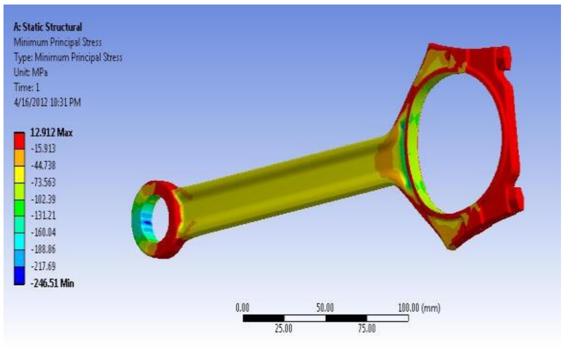


Fig 9.2.4 Minimum Principle Stress Distribution in connecting rod is -246.51

9.5 Von misses stress

Maximum distortion energy theory or Maximum shear strain energy theory or Von Misses theory states that, the failure occurs when Maximum shear strain energy when exceeds the shear strain energy in a simple tensile test, very good results for ductile materials and gives answers close to experimental values[14].

$$\sigma_1^2 + \sigma_2^2 - \sigma_1 \times \sigma_2 <= (S_{yt}/N)^2$$

Maximum Von Misses Stress is 411.41 MPa

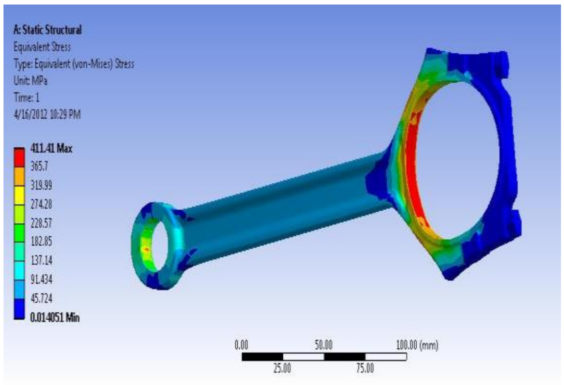


Fig 9.2.5 Von Misses Stress is 411.41 MPa very good results for ductile materials and gives answers close to experimental values.

9.6 Deformation along X-Axis

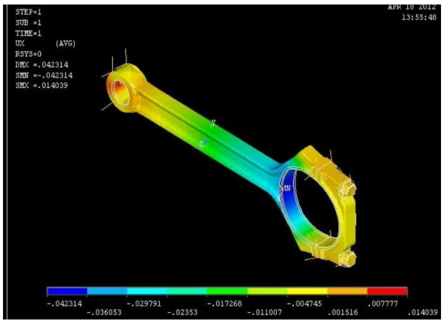


Fig 9.2.6 Maximum Deformation along X-axis is 0.0042314 mm. Simalarly for Y-Axis is 0.028846 mm and Z-Axis is 0.002935 mm

9.6 Total deformation

Maximum total deformation is 0.042314 mm

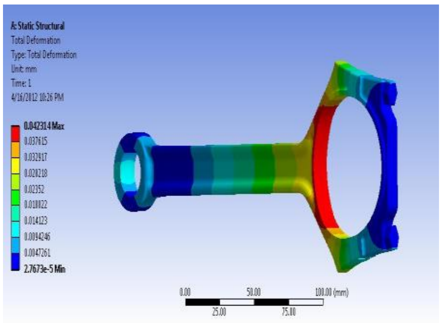


Fig 9.2.7 Total Deformation is 0.042314 mm

10. Results obtained of connecting rod using FEM

10.1 Table result for structural and fatigue

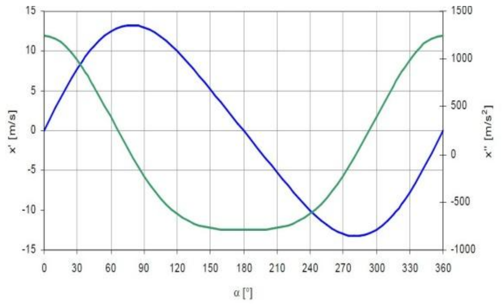
S. No.	*Different stresses*	*Results*
1.	Maximum value of normal stress	227.03 MPa
2.	Maximum value of Maximum principle stress	411.32 MPa
3.	Maximum value of Von Misses Stress	411.41 MPa
4.	Maximum total deformation	0.042314 mm
5.	Maximum Deformation along X- axis	0.0042314 mm
6.	Maximum Deformation along Y- axis	0.028846 mm
7.	Maximum Deformation along Z- axis	0.002935 mm

11. Conclusion

Maximum Principal stress comes out to be 411.32 MPa which is less than yield compressive strength that is 530 MPa hence the design is safe.

The maximum deformation, calculated with the classic method, has a value of 0.053mm but maximum deformation as calculated from ANSYS V14.0 is 0.041mm. This can be explained by the fact that many simplification hypotheses were considered for the classic calculation.

These results came at Inner Dead Centre as maximum force is acting on that point which is evident from the graph shown below–



The pattern of the piston speed and acceleration dependent on the crankshaft angular displacement

Graph 1. Inner Dead Centre as maximum force.

From the verification's calculations point of view, it is obvious that using a finite element analysis software(ANSYS V14.0, in this case) for the stresses and deformations calculations, saves a lot of time, comparative to the classic method's calculation. Even more, results are obtained in all of the structure's nodes, not only in certain sections.

By such analysis on ANSYS we will got stress and deformation for all finite elements in X, Y and Z direction.

Both compressive and tensile forces are acting on connecting rod but compressive forces are much greater than tensile forces, therefore we had designed according to compressive forces. Since, connecting rod is hinged at both ends by piston pin and crank pin and experiences compressive forces, therefore we can say that it will behave like a strut.

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