Computer Aided Stress and Modal Analysis of Connecting Rod

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Abstract-*Internal combustion engines components are highly complex and stressed components subjected to complex loading. Connecting rod is the link between piston and crankshaft which is acted upon by axial forces and moments at its ends, and converts the reciprocate motion to rotary motion. Yield, fatigue and buckling phenomenon are considered for the design and performance of connecting rod. In order to improve the performance and design, it is required to predict the critical locations of stresses and deformations and to determine the fatigue life. In this study, a finite element method is used to calculate the Von-Mises stresses and deformations under compressive loading on rod. The buckling of rod under compressive load is analyzed. The force on rod is calculated by numerical analysis for maximum combustion pressure in engine.*

Keywords-ANSYS, Buckling, Design, Finite element method, Stress, Von-Mises

I. INTRODUCTION

Connecting rods are widely used in variety of engines such as, oppose-piston engines, V-engines, opposed-cylinder engines, radial engines and In-line engines to transmit the thrust of the piston to the crankshaft, and results into conversion of the reciprocating motion of piston to the rotational motion of crankshaft. It consists of a pin-end, a shank section, and a crank-end. A connecting rod works in variably complicated conditions, and is subjected to not only the pressure due to the connecting rod mechanism, but also due to the inertia forces. Its behavior is affected by the fatigue phenomenon due to the reversible cyclic loadings. When the repetitive stresses are developed in the connecting rod it leads to fatigue phenomenon which can cause dangerous ruptures and damage. Bending stresses appear due to eccentricities, crankshaft, case wall deformation, and rotational mass force; therefore, a connecting rod must be capable of transmitting axial tension/compression and bending stresses caused by the thrust and pull on the piston and by the centrifugal force [1]. A primary design criterion for the connecting rod is endurance limit.The cyclic material properties are used to calculate the elastic-plastic stress-strain response and the rate at which fatigue damage accumulate due to each fatigue cycle [4]. Imahashi et al. [5] discuss the factors which affect the fatigue strength in powder forged (PF) connecting rod, i.e., hardness of the material, depth of decarburized layer, metallurgical structure, density, and surface roughness. Olaniran et al. [4] investigated a new crack able alloy of forged steel (FS) for connecting rod application. There are two practical buckling modes of connecting rod. One mode called 'side buckling' occurs in the direction parallel to the rotational axis of the connecting rod. The other mode called 'front-rear buckling' occurs in the direction perpendicular to side buckling. [8]. Webster et al., [12] discussed the loading criteria of connecting rod used in an IC engine. For tension loading the crank end and piston ends are found to have a sinusoidal distribution on the contact surface with pins and connecting rod.

II. FORCES ON ROD

The various forces acting on the connecting rod are as follows: Force on the piston due to gas pressure and inertia of the reciprocating parts, and Force due to inertia of the connecting rod or inertia bending forces, 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_G [7],

$$\boxed{F_C = (m_{piston} + m_{con-rod}).r\omega^2.(\cos\theta + \lambda\cos 2\theta) + F_G}$$

Force due to in-cylinder gas,

$$\boxed{F_G = p_{\max} \times A}$$

$\lambda = r/L$, Where r is crank radius and L is connecting rod length. Θ is crank angle and ω is angular velocity.

The connecting rod is subject to inertial bending forces (rod whip) as it swings through TDC. The gas force is determined by the speed of rotation, the masses of the piston, gudgeon pin and oscillating part of the connecting rod consisting of the small end and the shank. Axial loading on rod is due to gas pressure and rotational mass forces. Bending moments originate due to eccentricities, crankshaft, case wall deformation, and rotational mass force, which can be determined only by strain analyses in engine [4]. Loading on piston and then to connecting rod in an engine is the function

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of crank angle. It varies up to a maximum value from suction pressure and again lowers to exhaust pressure. The pressure in cylinder is the maximum at TDC. In this work, the maximum pressure $\boldsymbol{p_{max}}$ is obtained from engine configuration and is taken 7 times to mean effective pressure of the given engine [2]. The mean effective pressure is obtained by numerical analysis for given data of considered engine. The mean effective pressure is given as,

$$\boxed{Power = \frac{mep \times A \times L \times K \times N}{z \times 60000}}$$

Where, *A* is cross-section area, *K* is number of cylinders, *N* is rpm and $z = 2$

The mean effective pressure is calculated for the given engine configuration is 6.75 bar, thus the force on connecting rod is taken approximately 40 kN for the analysis.

Table 1: Engine Specifications

Bore, *mm*	108
Stroke, *mm*	130
Connecting rod length *L*, *mm*	250
Swept Volume, *Litre*	4.8
No. of cylinders *K*	4
Engine speed *N*, *RPM*	1800
Compression ratio	17:1
Power, *kW*	48.47
Torque, *N-m*	289.1 @ 1250 RPM

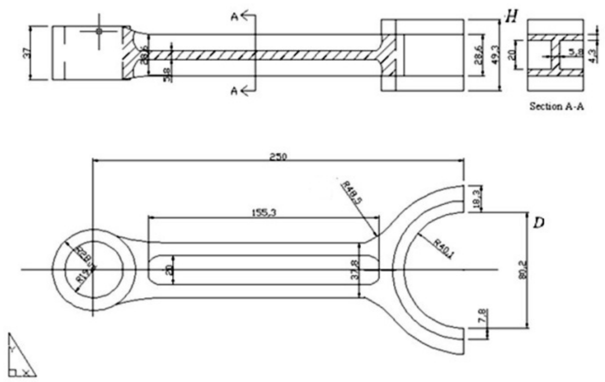


Figure 1: Drawing of Connecting Rod

For the given engine configuration, a typical connecting rod is taken as per the drawing [3]. The rod is open at crank end and the total length from pin to pin is 250 mm as in Fig. 1. The design parameters corresponding to given drawing are tabulated below.

Table 2: Design Parameters of Rod

Design Parameters	**Value, mm**
Piston pin end diameter	39
Crank pin end diameter	80
Effective length	250
Shank Length	155.4
Fillet radius at crank end	48.5
Total width	37.8

III. FE MESH MODEL

The 3-D solid model to the particular drawing is introduced to ANSYS software. The solid modelling is done on higher end CAD software as per design parameters given in Table 2 for I section. The meshing of rod is shown in Fig. 2 through the 10–node tetragonal SOLID 187 elements (300232 elements, 435444 nodes) of 2 mm length. The reason for selecting SOLID 187 elements is to make the geometrical parts of a complicated mechanical component to gain more authentic results based on the high techniques of stress calculation.

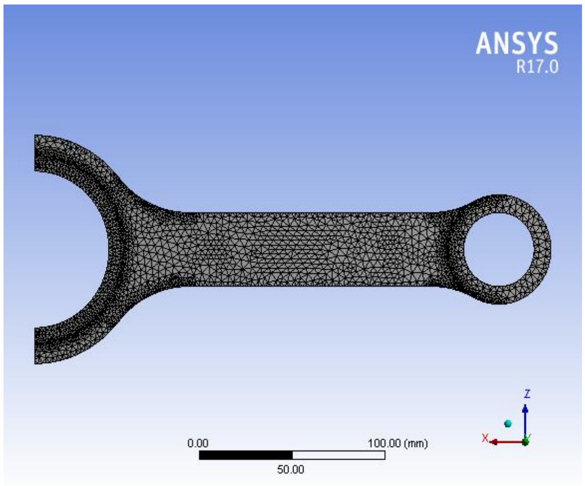


Figure 2: Mesh Model of Connecting Rod

The material properties are given below in Table 3 [14]

Table 3: Material Properties

Monotonic Properties	Forged Steel (FS)
Young’s Modulus, GPa	201
Yield Strength, MPa	700
Ultimate Tensile Strength, MPa	938
Poisson’s Ratio	0.30
Strain Hardening Exponent	0.122
Density, kg/m^3	7806

Boundary Condition and Loading

The stress and strain is the most serious when the explosive pressure of the fuel gas achieves maximum when the piston is at TDC. The maximum pressure is calculate using numerical calculation for the given engine parameters. The force is max when piston is at TDC and same force is transferred to the connecting rod. The displacement restrictions for the rod are restrained to let the rod in a static condition. The crank end face is fixed for the analysis. The crank pin end is kept fixed and load is applied on piston pin end towards stroke line of the connecting rod axis as shown in Fig. 3

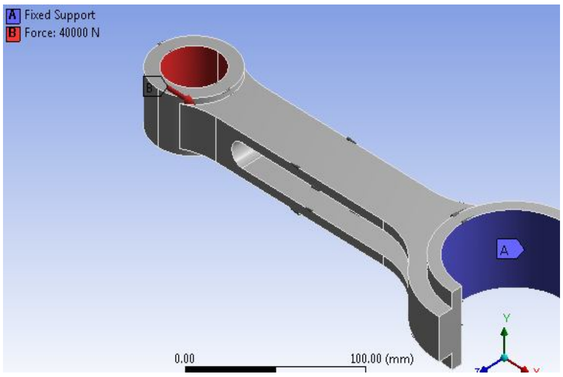


Figure 3: End Constraints

IV. BUCKLING OF ROD

The elastic term (Euler equation) of the classical formula used in the buckling criterion of connecting rod is valid when the column is an ideal slender column with uniform cross-section under specific boundary conditions such as pined–pined, fixed–fixed and pined–fixed joints. The actual connecting rod is however incompatible with the assumptions of the Euler formula in three aspects. First, the shank of connecting rod has non-uniform cross-sections along the length. Secondly, the real buckling constant is different from the classical constant because both ends of connecting rod are connected with the pin and crank shaft which have relatively big radii. Thirdly, it is difficult to define the effective length of connecting rod for front-rear buckling [6]. Consequently, an enhanced formula is needed in the shape design of connecting rod shank. There are two practical buckling modes of connecting rod. One mode called 'side buckling' occurs in the direction parallel to the rotational axis of the connecting rod. The other mode called 'front-rear buckling' occurs in the direction perpendicular to side buckling. Modal analysis has been used to study the inherent frequencies and vibration mode shapes of mechanical systems, which only relate with the stiffness specialty and mass distribution of structure, and it has nothing to do with external factors [9].

The movement differential equation of the N freedom degrees linear system can be express as

$$[M]\{\ddot{X}\}+[C]\{\dot{X}\}+[K]\{X\}=\{F(t)\}$$

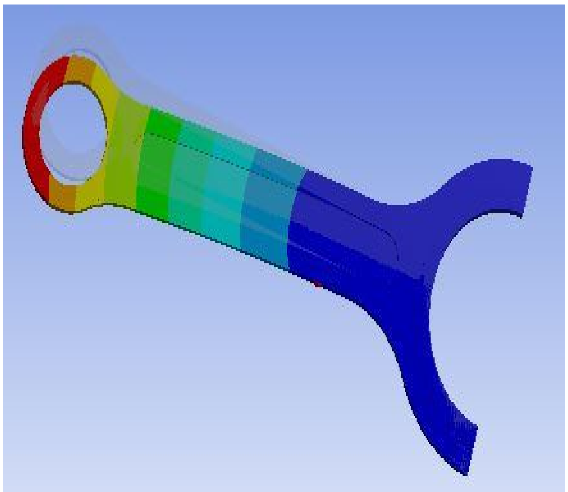
[M] is the mass matrix, [C] is the damping matrix, [K] is the stiffness matrix, $\{\ddot{X}\}$ is acceleration vector, $\{\dot{X}\}$ is velocity vector, $\{X\}$ is the displacement response column vector of node, $\{F(t)\}$ is the load column vector.
The damping and external load were not considered when studying the inherent frequencies and vibration mode shapes in the process of modal analysis, so the vibration differential equations could be further simplified to get the undamped free vibration differential equations of system as [10]

$$[M]\{\ddot{X}\} + [K]\{X\} = 0$$

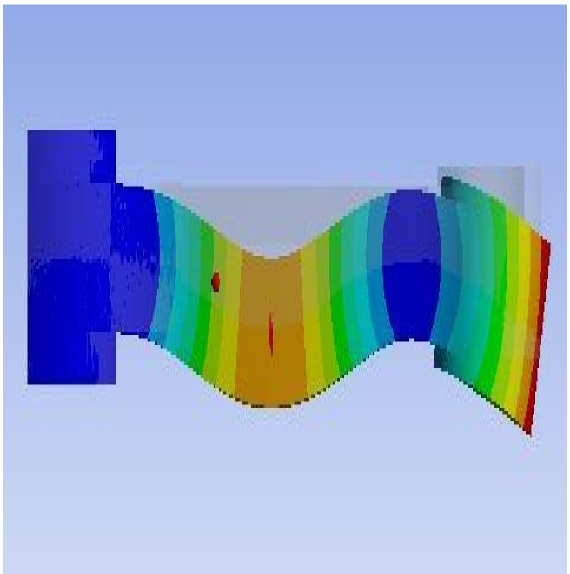
A characteristic of modal for the connecting rod by three-dimensional finite element model is established above, and frequency of vibration modes is calculated out using FEM in ANSYS. [11] Connecting rod is prone to cause too large dynamic stress produce the fatigue cracks in its working process. The actual bending of connecting rod [13] according to loading and shapes were examined.


Figure 4: Buckling Shapes [13]

The buckling analysis done for this work is shown in Fig 5. Two mode analyses are carried out, one for side buckling and other is front buckling.



(a)

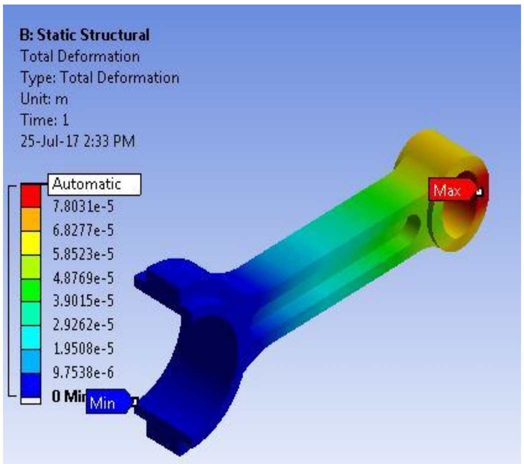


(b)

Figure 5: (a) First Mode (b) Second Mode

V. STRESS ANALYSIS

Finite element meshes are generated using ten-node tetrahedral elements (SOLID 187) with various elemental lengths from 5 mm to 1 mm in a step of 0.5 mm. The material properties are gieven in Table 3 for which the analysis is carried out in ANSYS. The Von-Mises stresses, when stresses generated exceed the allowable limit, are checked for the convergence at critical locations within rod. The results show that the convergence has been achieved for the entire range of elemental length and constant values of stresses generated if the size of elements 2 mm is considered. The FEA results are shown in given Fig. 6 for Von-Mises stress, strain and deformations. The maximum stress location is on piston pin end side.



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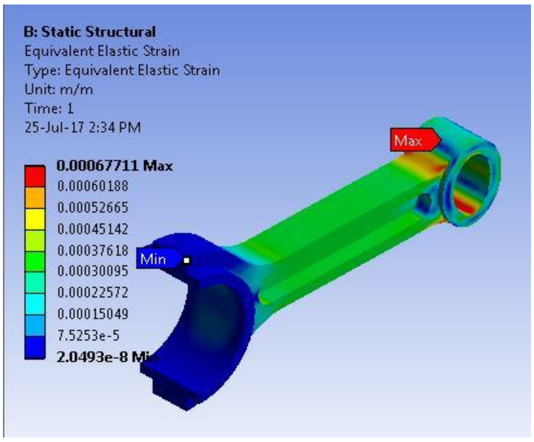

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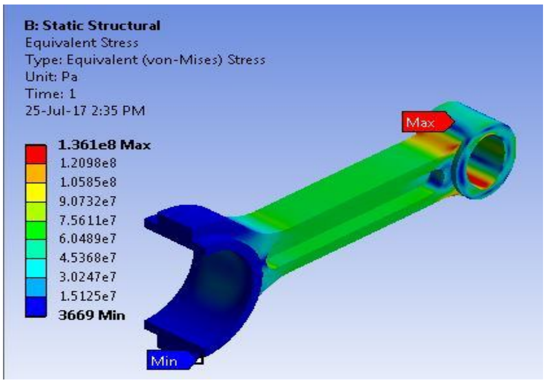

(c)

Figure 6: (a) Total Deformation, (b) Strain and (c) Stress of Connecting Rod

VI. CONCLUSIONS

The finite element analysis is very popular method to deal the problem of stress analysis when geometry of the object is complicated and loading conditions are complex. In the present work modeling of connecting rods has been carried out on CATIA software and ANSYS workbench software is used for the analysis. The ANSYS software is able to analyze the different components from varied aspects such as fatigue analysis and consequently save the time and the cost. The buckling analysis is done on two mode, one for side buckling and two for front buckling as per the practical situation. The maximum stress obtained is 136 MPa near the small end and minimum stress near the big end. The strain value is 0.006 and the total deformation 2.049e-8 m at the top of small end.

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