FEA based selection of Connecting Rod using ANSYS

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Abstract - *Connecting rod is integral component of internal combustion engine; it acts as a linkage between piston and crankshaft. There are three main zones, the piston pin end, the central shank and the big end. The piston pin end is the small end, the crank end is the big end and the central shank is of I-cross section. Material usually used to manufacture connecting rod in mass is steel, but it can also be made of Aluminum (for lightness and the ability to absorb high impact at the expense of durability) or titanium (for combination of strength and lightness at the expense of affordability) for high performance engines, or of cast iron. They can be produced either by casting, powder metallurgy or forging. However, connecting rods which are produced by casting have higher possibility of containing blow holes which are adverse from durability and fatigue points of view. Rods manufactured by powder metallurgy process have the advantage of being near net shape, reducing material waste. Once the manufacturing processes are studied design considerations were studied and it was used to reverse calculate the boundary condition for which steel connecting rod was designed. Various results for each material will be studied and comparison will be made to study the improvement that occurs with reduction in weight.*

Keywords: Connecting Rod, Finite element analysis, Optimization, Modal analysis, Materials etc.

1. INTRODUCTION

The automobile engine connecting rod is a high volume production, critical component. It connects reciprocating piston to rotating crankshaft, transmitting the thrust of the piston to the crankshaft. Every vehicle that uses an internal combustion engine requires at least one connecting rod depending upon the number of cylinders in the engine.

Automotive should be light in weight so as to consume less fuel and at the same time they should provide comfort and safety to passengers, which unfortunately leads to increase in weight of the vehicle. This tendency in vehicle construction led to the invention and implementation of quite new materials which are light and meet design requirements. Lighter connecting rods help to decrease load caused by forces of inertia in engine as it does not require big balancing weight on crankshaft. Application of composite material enables safety to increase and advances that leads to effective use of fuel and to obtain high engine power.

Any physical system can vibrate. The frequencies at which vibration naturally occurs, and the modal shapes which the vibrating system assumes are properties of the system which can be determined analytically using modal analysis. Detailed modal analysis determines the fundamental vibration modes shapes and corresponding frequencies. This can be relatively simple for basic components of a simple system, end extremely complicated when qualifying a complex mechanical device or a complicated structure exposed to periodic wind loading. These system required accurate determination of natural frequencies and mode shape using techniques such as Finite Element Analysis. Using Finite Element models to predict the dynamic properties of structures becomes more and more important in modern mechanical industries, such as the automobile industries.

2. DESIGN OF CONNECTING ROD [12]

To find Rankine formula numerical constant

$$W_B = \frac{\sigma_c \cdot A}{1+a(^{L}/_{k_{xx}})^2}$$

The buckling load W_B may be calculated by using the following relation,

W_B = Max. Gas force * Factor of safety

Where Max. Gas force = 4000 N

The factor of safety taken as 1.

$$W_B = \frac{\sigma_c \cdot 11t^2}{1+\frac{1}{7500}(^{L}/_{1.78t})^2}$$

$$4000 = \frac{\sigma_c \cdot 539}{1+\frac{1}{7500}(200/12.46)^2}$$

σ_c=7.67 N/mm^2

The dimensions of I-section of the connecting rod are

Width of the section, B = 4*t

= 4*7

B = 28mm

Depth or height of the section, H = 5*t

H = 5*7

H = 35mm

The depth near the small end, H_1 = 0.85*H

= 0.85*35

= 30mm

The depth near the big end, H_2= 1.2*H

= 1.2*35

= 42mm

To find bearing pressure at crank pin

Load on the crank pin = Projected area × Bearing pressure

$= d_c * l_c * P_{bc}$

$= 64 * 83.2 * P_{bc}$

P_{bc}= 0.7512N/mm^2

To find bearing pressure at crank pin

Load on the crank pin = Projected area × Bearing pressure

$= d_p * l_p * P_{bp}$

$= 24 * 48 * P_{bp}$

P_{bp}= 3.472 N/mm^2

To find allowable tensile force

The inertia force to the bolts, we have

$$F_I = \frac{\pi}{4}(d_{cb})^2 \sigma_t * n_b$$

$$4000 = \frac{\pi}{4}(8)^2 \sigma_t * 2$$

σ_t=39.78N/mm^2

The nominal or major diameter (d_b) of the bolt is given by

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

$$d_b = \frac{8}{0.84}$$

d_b = 9.524mm

To find allowable bending force

The maximum bending moment acting on the cap will be,

$$M_C = \frac{F_I \times x}{6}$$

Where,

x = Distance between the bolt centers.

= dia. Of crankpin or big end bearing (d_c) + 2*Thickness of bearing liner (3mm) + Clearance (3mm)

= 64 + 6 + 9.52 + 3

= 82.52 mm

$$M_C = \frac{4000 \times 82.52}{6}$$

= 55013.33 Nmm

We know that section modulus for the cap,

$$Z_C = \frac{b_c(t_c^2)}{6}$$

$$= \frac{83.2(6.26^2)}{6}$$

= 543.40mm^3

Bending stress, $\sigma_b = \frac{M_C}{Z_C}$

$$\sigma_b = \frac{55013.33}{543.40}$$

$$\sigma_b = 101.24\text{N}/mm^2$$

3. FINITE ELEMENT ANALYSIS

3.1. SOLID187 Element Description

SOLID187 element is a higher order 3-D, 10-node element. SOLID187 has a quadratic displacement behavior and it is well suited to modeling irregular meshes & shapes (such as those produced from various CAD/CAM modules). The element is defined by 10 nodes having three degrees of freedom at each node: translations in the nodal x, y, and z directions as shown in figure 1.

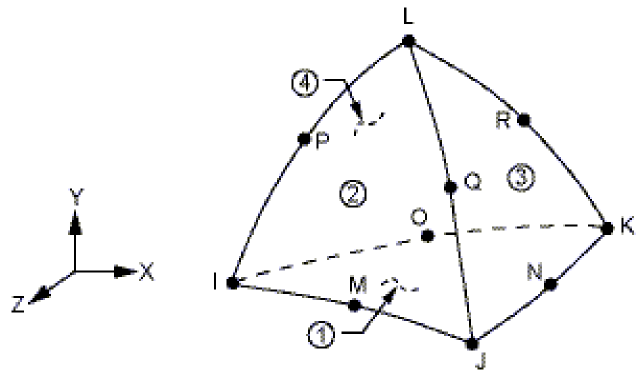

Fig -1: 3-D 10-Node Tetrahedral Structural Solid

3.2. Boundary Conditions of the Connecting Rod for axial loading:

In this case of axial loading, big end of connecting rod with titanium inserts is given tensile pressure of 3.514 MPa and small end is fixed.

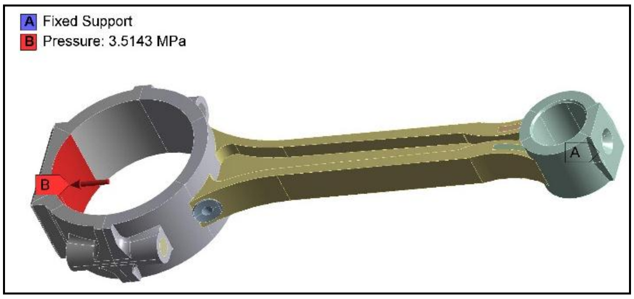

Fig -2: Boundary conditions of Connecting Rod with titanium inserts

3.3. Meshing of Tetrahedral Structural Solid

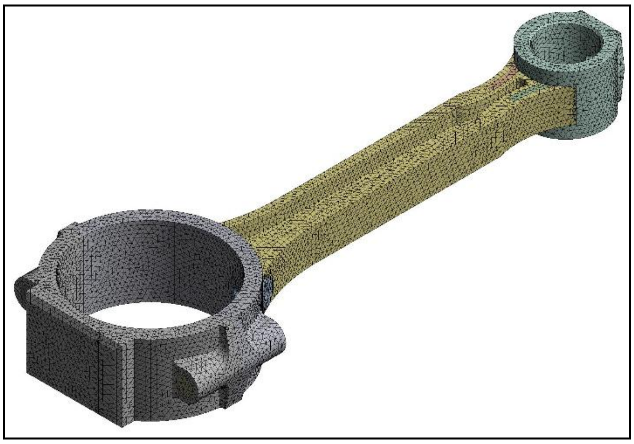

Fig -3: Meshed Model of connecting rod with titanium inserts at 200000 nodes

3.4. Total Deformation of Connecting Rod with titanium inserts

In loading condition

Total Deformation has been derived from the static analysis of structure using ANSYS as shown below

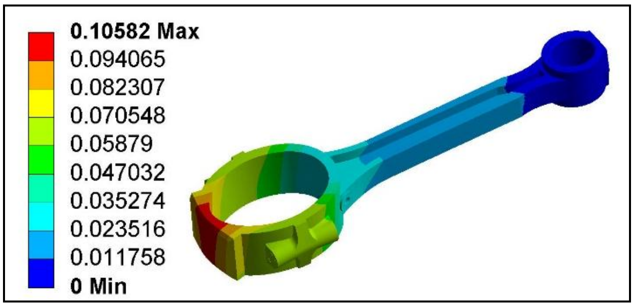

Fig -4: Total deformation for connecting with titanium inserts is 0.10582 mm (max) found at the big end.

3.5. Structural Static Analysis of Connecting Rod with titanium inserts

Equivalent (von-Mises) stresses has been derived from the static analysis of structure using ANSYS as shown below

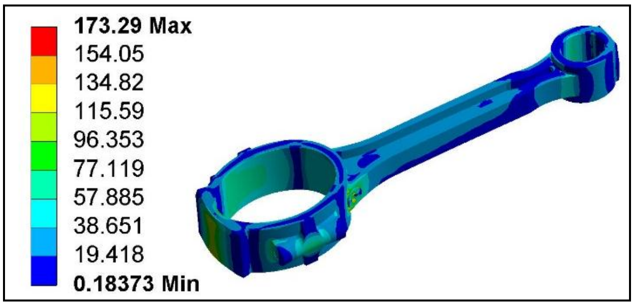

Fig -5: Equivalent (Von-Mises) stress for connecting with titanium inserts 173.29 MPa (max) found at the oil hole.

3.6. Boundary Conditions of the Connecting Rod for bending loading:

In this case of bending loading, big end of connecting rod with titanium inserts is given tensile pressure of 3.514 MPa at an angle of 30° and small end is fixed.

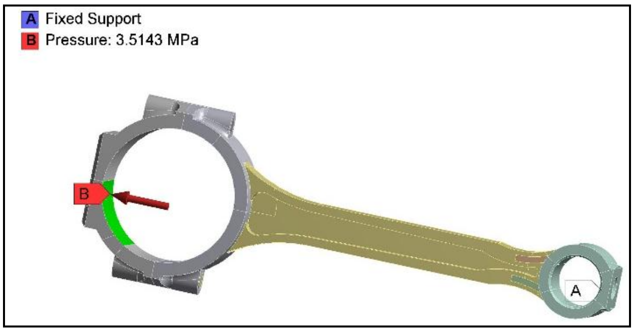

Fig -6: Boundary conditions of Connecting Rod

3.7. Total Deformation of Connecting Rod with titanium inserts

In loading condition

Total Deformation has been derived from the static analysis of structure using ANSYS as shown below

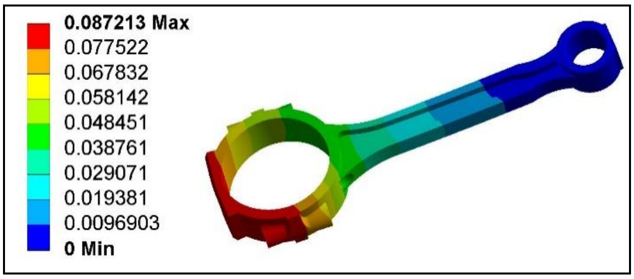

Fig -7: Total deformation for connecting with titanium inserts is 0.087213 mm (max) found at the big end.

3.8. Structural Static Analysis of Connecting Rod with titanium inserts

Equivalent (von-Mises) stresses has been derived from the static analysis of structure using ANSYS as shown below

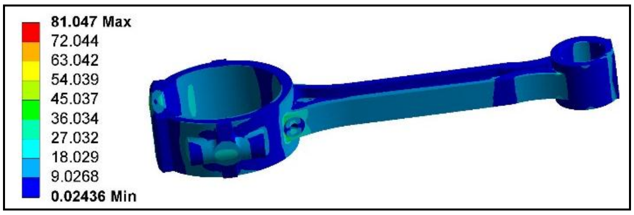

Fig -8: Equivalent (Von-Mises) stress for connecting with titanium inserts is 81.047 MPa (max) found at the oil hole.

3.9. Modal Analysis of the connecting rod with titanium inserts

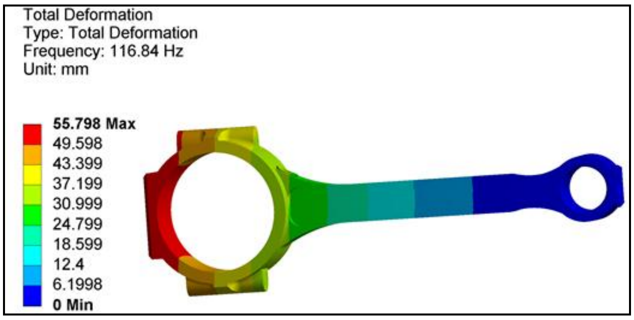

Fig -9: Mode shape of the connecting rod with titanium inserts at 1st Natural Frequency (116.84 Hz)

4. RESULT & DISCUSSION:

Similarly for connecting rod with steel inserts, we obtain:

Table -1: Static structural Analysis using ANSYS

Loading condition	Total (max) Deformation (mm)	Equivalent (Von-Mises) stress (MPa)
Axial load	0.1053	191.15
Bending load	0.086532	83.597

Table -2: Modal Analysis using ANSYS

Material type	Connecting rod with titanium inserts	Connecting rod with steel inserts
Natural Frequencies (Hz)	116.84 157.85 278.95 897.89 1064.7 2103.4	118.46 160.09 287.5 900.86 1064.6 2112.9
Weight (Kg)	0.83772 kg	0.84572 kg

5. CONCLUSIONS

1. The variation of maximum von-mises stresses for each loading condition in connecting rod with steel inserts with comparison to optimized model of connecting rod with titanium inserts was very less and it was found to be well below the limit of allowable stress.
2. Fillet given at oil hole has considerably reduced stress concentration occurring at the oil hole in tensile and compressive loading conditions.
3. Percentage weight reduction obtained by simulation in ANSYS Workbench based on stress observations was approximately 1%.
4. The maximum axial stress is reduced by 9.35% and maximum bending stress reduced by 3%.

Hence we can conclude that the connecting rod with titanium inserts is a better option than the connecting rod with steel inserts because it has lower weight and it has better life.

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