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Stress Analysis of Connecting Rod of Two Wheeler Engine Using Finite Element Method

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ABSTRACT

The connecting rod is the intermediate member between the piston and the Crankshaft. Its primary function is to transmit the push and pull from the piston pin to the crank pin, thus converting the reciprocating motion of the piston into rotary motion of the crank. This project work is on designing and stress analysis of connecting rod. Currently existing connecting rod is manufactured by using Carbon steel. A parametric model of Connecting rod is modelled using CATIA V5 R19 software and to that model, finite element analysis is carried out by using ANSYS Software. Finite element method is used to determine parameters like various stresses & strain of a connecting rod and compare results with FFT analyzer. FFT analysis is done by hanging the connecting rod at small end and experimental results were compared with FEM. Linear static analysis is carried out & compared for both carbon steel and aluminium alloy to obtain the stress results by keeping the same boundary & loading conditions.

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I. INTRODUCTION

The connecting rod is a major link inside a combustion engine. It connects the piston to the crankshaft and is responsible for transferring power from the piston to the crankshaft and sending it to the transmission. There are different types of materials and production methods used in the creation of connecting rods. In modern automotive internal combustion engines, the connecting rods are most usually made of steel for production engines, but can be made of aluminium (for lightness and the ability to absorb high impact at the expense of durability) or titanium (for a combination of strength and lightness at the expense of affordability) for high performance engines, or of cast iron for applications such as motor scooters. The most common types of manufacturing processes are casting, forging and powdered metallurgy. Connecting rods are widely used in variety of engines such as, in-line engines, V-engine, opposed cylinder engines, radial engines and opposed-piston engines.

Fig. 1 Connecting Rod

G. Naga Malleshwara Rao [1] worked is to explore weight reduction opportunities in the connecting rod of an I.C. engine by examining various materials such as Genetic Steel, Aluminium, Titanium and Cast Iron. This study has dealt with two subjects, first, static load and stress analysis of the connecting rod and second, Design Optimization for suitable material to minimize the deflection. In the first of the study the loads acting on the connecting rod as a function of time are obtained. The relations for obtaining the loads for the connecting rod at a given constant speed of crank shaft are also determined. It can be concluded from this study that the connecting rod can be designed and optimized under a comprising tensile load corresponding to 360o crank angle at the maximum engine speed as one extreme load, and the crank pressure as the other extreme load. Furthermore, the existing connecting rod can be replaced with a new connecting rod made of Genetic Steel.

Ram bansal et al [2], in his paper a dynamic simulation was conducted on a connecting rod made of aluminium alloy using FEA. In this analysis of connecting rod were performed under dynamic load for stress analysis and optimization. Dynamic load analysis was performed to determine the in service loading of the connecting rod and FEA was conducted to find the stress at critical locations.

Vivek. C. Pathade et al [3], he dealt with the stress analysis of connecting rod by finite element method using pro-e wild fire 4.0 and ansys work bench 11.0 software. And concluded that the stress induced in the small end of the connecting rod are greater than the stresses induced at the bigger end, therefore the chances of failure of the connecting rod may be at the fillet section of both end.

Pushpendra kumar Sharma et al [4], performed the static FEA of the connecting rod using the software and said optimization was performed to reduce weight. Weight can be reduced by changing the material of the current forged steel connecting rod to crackable forged steel (C70). And the software gives a view of stress distribution in the whole connecting rod which gives the information that which parts are to be hardened or given attention during manufacturing stage.

K. Sudershan kumar, et al [5], described modeling and analysis of Connecting rod. In his project carbon steel connecting rod is replaced by aluminium boron carbide connecting rod. Aluminium boron carbide is found to have working factory of safety is nearer to theoretical factory of safety, to increase the stiffness by 48.55% and to reduce stress by 10.35%.

Pravardhan S. Shenoy and Ali Fatemi [6] performed optimization to reduce weight and manufacturing cost of a forged steel connecting rod. The main objective of this study was to explore weight and cost reduction opportunities for a production forged steel connecting rod. The structural factors considered for weight reduction during the optimization process included fatigue strength, static strength, buckling resistance, bending stiffness, and axial stiffness. Cost was reduced by changing the material of the existing forged steel connecting rod to crackable forged steel (C-70). The shank region of the connecting rod offered the greatest potential for weight reduction. The optimized geometry is 10% lighter than the current connecting rod for the same fatigue strength, in spite of lower yield strength and endurance limit of C-70 steel compared to the existing forged steel

Pai [7] presented an approach to optimize the shape of a connecting rod subjected to a load cycle which consisted of the inertia load deducted from gas load as one extreme and peak inertia load exerted by the piston assembly mass as the other extreme. A finite element routine was first used to calculate the displacements and stresses in the rod, which were then used in another routine to calculate the total life. Fatigue life was defined as the sum of crack initiation and crack growth lives, with crack growth life obtained using fracture mechanics.

Sarihan and Song [8] optimized the wrist pin end of an engine connecting rod with an interference fit. They generated an approximate design surface and performed optimization of this design surface. The objective and constraint functions were updated in an iterative process until convergence was achieved. The load cycle that was used consisted of compressive gas load corresponding to a maximum torque and a tensile load corresponding to maximum inertia load. The modified Goodman equation with alternating and mean octahedral shear stress was used for fatigue analysis.

Serag et al. [9] developed approximate mathematical formulae to define connecting rod weight and cost as objective functions as well as constraints. The optimization was achieved using a geometric programming technique.

Yoo et al. [10] performed shape optimization of an engine connecting rod using variational equations of elasticity, material derivative idea of continuum mechanics, and an adjoint variable technique to calculate shape design sensitivities of stress. The results were then used in an iterative optimization algorithm to numerically solve for an optimal design solution.

II. METHODOLOGY

A connecting rod is a machine member which is subjected to alternating direct compressive and tensile forces. Since the compressive forces are much higher than the tensile force, therefore the cross-section of the connecting rod is designed as a strut and the Rankin formula is used.

A connecting rod subjected to an axial load W may buckle with x-axis as neutral axis in the plane of motion of the connecting rod, {or} y-axis is a neutral axis. The connecting rod is considered like both ends hinged for buckling about x-axis and both ends fixed for buckling about y-axis. A connecting rod should be equally strong in buckling about either axis.

A. Theoretical calculations of connecting rod:

Let A = cross sectional area of the connecting rod.
L = length of the connecting rod.
σ_c = compressive yield stress.
W_{cr} = crippling or buckling load.
I_{xx} and I_{yy} = moment of inertia of the section about x-axis and y-axis respectively.
K_{xx} and K_{yy} = radius of gyration of the section about x-axis and y- axis respectively.

B. Bajaj Discover 150 Specifications:

Engine type air cooled 4-stroke

Bore × Stroke (mm) = 56×58.8
Displacement = 144.8 cc
Maximum Power = 14.5bhp@8500rpm
Maximum Torque = 12.75Nm@6500rpm
Length of connecting rod (L) = 92.23 mm

C. Pressure calculation for 150cc engine:

Density of Petrol C_8H_{18} = 737.22 kg/m^3 = 737.22E-9 kg/mm^3
Temperature = 60 °F = 288.706 °K
Mass = Density × Volume = 737.22E-9×144.8E^3 = 0.107Kg
Molecular Weight of Petrol 114.228 g/mole
From Gas Equation, $PV = Mrt$

$$R = \frac{R^*}{Mw} = \frac{8.3143}{0.114228} = 72.786$$

$$P = \frac{0.107 \times 72.786 \times 288.706}{144.8\, E^3}, P = 15.50 Mpa.$$

D. Properties of standard I-Section

Thickness of flange & web of the section = t
Width of section $B = 4t$
Height of section $H = 5t$
Area of section

$$A = 2(4t \times t) + 3t \times t, A = 11\, t^2$$

MI of section about x axis:

$$I_{xx} = \frac{1}{12}\{4t\,(5t)^3 - 3t\,(3t)^3\} = \frac{419}{12}\, t^4$$

MI of section about y axis:

$$I_{yy} = \left\{2\,\frac{1}{12}\, t\,(4t)^3 + \frac{1}{12}(3t)\, t^3\right\} = \frac{131}{12}\, t^4$$

$$\frac{Ixx}{Iyy} = 3.2$$

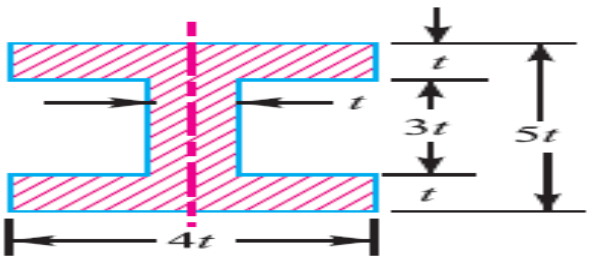


Fig 2: Standard dimension of I – Section

E. Design calculation for Steel (16MnCr5), without considering inertia force for standard I-section

Buckling load W_{cr} = maximum gas force on piston

$$W_{cr} = \frac{\sigma_c \times A}{1 + a\left(\frac{L}{K_{xx}}\right)^2} = 38176.64\, N$$

σ_c = Compressive yield stress = 415 MPa;
E = Young's modulus = 210 GPa

$$K_{xx} = \frac{I_{xx}}{A}, K_{xx} = 1.783\, t$$

$$a = \frac{\sigma_c}{\pi^2\, E}, a = 0.0002$$

By substituting $\sigma_c, A, a, L, K_{xx}$ *in* W_{cr},

Then $4565\, t^4 - 38176.64\, t^2 - 20428.32 = 0$

t^2 = 8.86; t = 2.97 mm, t ≈ 3.0 mm.
Width of section $B = 4t = 12\, mm$,
Height of section $H = 5t = 15\, mm$
Area $A = 11t^2 = 99\, mm^2$
Height at the small end (piston end) = $H_1 = 0.9H$ *to* $0.75H$, $H_1 = 13.5\, mm$
Height at the big end (crank end) $H_2 = 1.1H$ *to* $1.25H$, $H_2 = 16.5\, mm$
Weight of connecting rod per unit length,
w = volume x density = area x length x density = 7.77 x 10^{-3} kg/cm = 0.777 kg/m
Maximum bending moment,

$$M_{max} = \frac{w}{g} \times \omega^2 \times r \times \left(\frac{l^2}{9\sqrt{3}}\right) = 1.007\, kg.m.$$

Section modulus, $Z_{xx} = \frac{I_{xx}}{5t/2} = 0.377\, cm^3$

Maximum bending stress, $f_{b\,max} = \frac{M_{max}}{Z_{xx}} = 267.03\ kg/cm^2$

F. Design calculation for Aluminium LM9, without considering inertia force for standard I-section

Buckling load W_{cr} = maximum gas force on piston

$$W_{cr} = \frac{\sigma_c \times A}{1 + a\left(\frac{L}{K_{xx}}\right)^2} = 38176.64\, N$$

σ_c = Compressive yield stress = 250 MPa;
E = Young's modulus = 71 GPa

$$K_{xx} = \frac{I_{xx}}{A}, K_{xx} = 1.783\, t$$

$$a = \frac{\sigma_c}{\pi^2\, E}, a = 0.00036$$

By substituting $\sigma_c, A, a, L, K_{xx}$ *in* W_{cr},

Then $2750\, t^4 - 38176.64\, t^2 - 36764.10 = 0$

t^2 = 14.78;t = 3.85 mm, t ≈ 3.9 mm.
Width of section $B = 4t = 15.6\, mm$,
Height of section $H = 5t = 19.5\, mm$
Area $A = 11t^2 = 167.31\, mm^2$
Height at the small end (piston end) = $H_1 = 0.9H$ *to* $0.75H$, $H_1 = 17.55\, mm$
Height at the big end (crank end) = $H_2 = 1.1H$ *to* $1.25H$, $H_2 = 21.45\, mm$
Weight of connecting rod per unit length,
w = volume x density = area x length x density = 4.48 x 10^{-3} kg/cm = 0.448 kg/m
Maximum bending moment,

$$M_{max} = \frac{w}{g} \times \omega^2 \times r \times \left(\frac{l^2}{9\sqrt{3}}\right) = 0.580\, kg.m.$$

Section modulus, $Z_{xx} = \frac{I_{xx}}{5t/2} = 0.828\, cm^3$

Maximum bending stress, $f_{b\,max} = \frac{M_{max}}{Z_{xx}} = 70.05\ kg/cm^2$

G. Design calculation for Aluminium LM9, without considering inertia force for non standard I-section

Buckling load W_{cr} = maximum gas force on piston

$$W_{cr} = \frac{\sigma_c \times A}{1 + a\left(\frac{L}{K_{xx}}\right)^2} = 38176.64\, N$$

σ_c = Compressive yield stress = 250 MPa;
E = Young's modulus = 81.3 GPa

$$K_{xx} = \frac{I_{xx}}{A}, K_{xx} = 47.4$$

$$a = \frac{\sigma_c}{\pi^2\, E}, a = 0.00036$$

By taking $t = 4.2mm, B = 14mm, H = 19.5mm$ & substituting $\sigma_c, A, a, L, K_{xx}$ *in* W_{cr},
Then $W_{cr} = 41000\, N$

Width of section $B = 14\ mm$,
Height of section $H = 19.5\ mm$
Area $A = 164.22\ mm^2$
Height at the small end (piston end) $= H_1 = 14\ mm$
Height at the big end (crank end) $= H_2 = 14\ mm$
Weight of connecting rod per unit length,
w = volume x density = area x length x density = 4.40 x 10^{-3} kg/cm = 0.440 kg/m
Maximum bending moment,

$$M_{max} = \frac{w}{g} \times \omega^2 \times r \times \left(\frac{l^2}{9\sqrt{3}}\right) = 0.570\ kg.m.$$

Section modulus, $Z_{xx} = \frac{I_{xx}}{5t/2} = 0.741\ cm^3$

Maximum bending stress, $f_{b\,max} = \frac{M_{max}}{Z_{xx}} = 76.92\ kg/cm^2$

H. Weight reduction calculation:

Weight of Connecting rod of Steel (16MnCr5) existing product = 0.0956 kg = 95.6 gm

Weight of Connecting rod of Steel (16MnCr5) standard I-section from 3D model = Volume x Density
= 1.733 E^{-5} x 7850 = 0.136 kg = 136 gm

Weight of Connecting rod of Aluminium LM9 non standard I-section from 3D model = Volume x Density
= 2.284 E^{-5} x 2730 = 0.062 kg = 62 gm

TABLE 1
WEIGHT COMPARISON

Material	**Steel (16MnCr5)**	**Aluminium LM9**
Weight (gm)	95.6	62

I. Material composition:

TABLE 2
CHEMICAL COMPOSITION OF STEEL (16MNCR5)

C	**Si**	**Mn**	**P**	**S**	**Cr**	**Fe**
0.14 – 0.19	Max. 0.4	1 – 1.3	Max. 0.025	Max. 0.035	0.8 – 1.1	Balance

TABLE 3
CHEMICAL COMPOSITION OF ALUMINIUM LM9

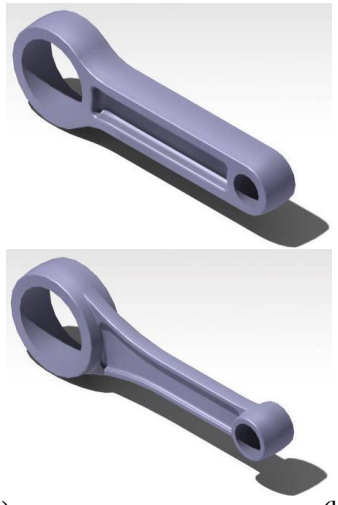
Si	**Fe**	**Cu**	**Mn**	**Mg**	**Ni**	**Zn**	**Sn**	**Pb**	**Ti**	**Al**
10 – 13	0.6	0.2	0.3 – 0.7	0.2 – 0.6	0.1	0.1	0.05	0.1	0.2	Balance

III.FINITE ELEMENT METHOD

Analysis done with pressure load applied at the piston end and restrained at the crank end or other load applied at the crank end and restrained at the piston end. The finite element analysis is carried out on carbon steel (16MnCr5) connecting rod as well as on aluminium LM9.

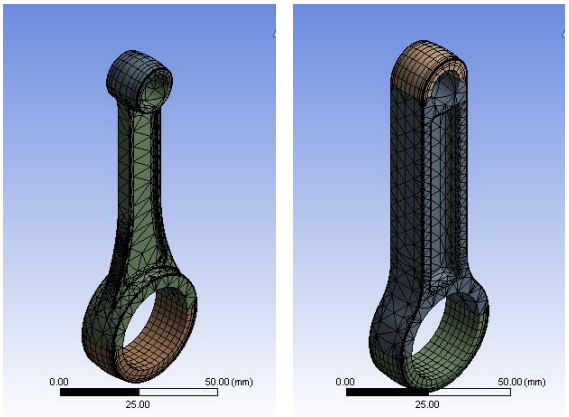
The CAD model of connecting rod of Carbon steel (existing product) and Aluminium LM9 is developed in CATIA V5R19 is shown in fig. 3. These meshed models of connecting rod of Carbon steel (existing product) and Aluminium LM9 is shown in fig. 4.

Analysis is carried out on these meshed models of connecting rod in Ansys 14. From the analysis the equivalent stress (Von-mises stress), displacements were determined and are shown in figure 5-8. Table 4 shows the comparative results for different materials.



(a) (b)

Fig. 3 CAD Model of connecting rod (a) Steel (16MnCr5) Existing product (b) Aluminium LM9



(a) (b)

Fig. 4 Mesh Model of connecting rod (a) Steel (16MnCr5) Existing product (b) Aluminium LM9

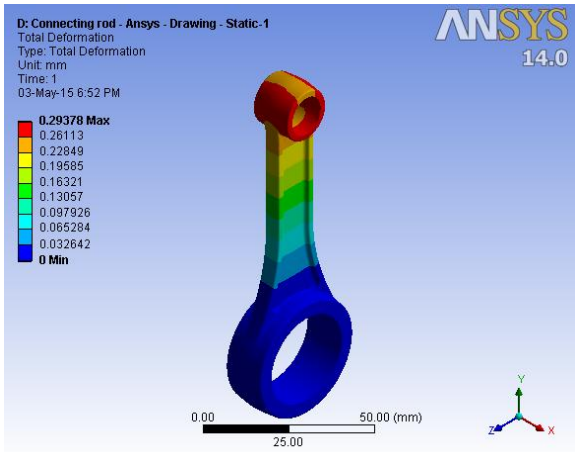


Fig. 5 Analysis of connecting rod (16MnCr5) Existing product (Deflection)

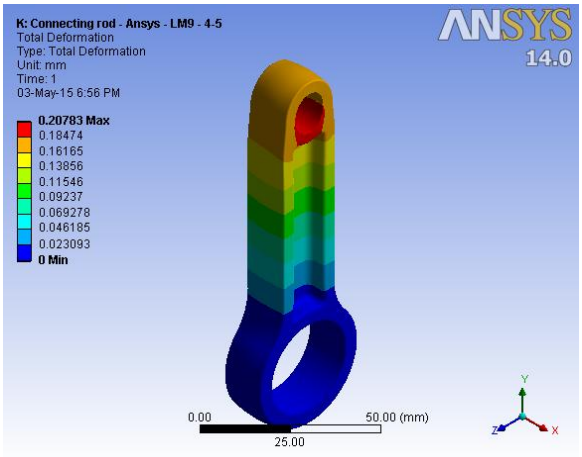


Fig. 6 Analysis of connecting rod (Aluminium LM9) (Deflection)

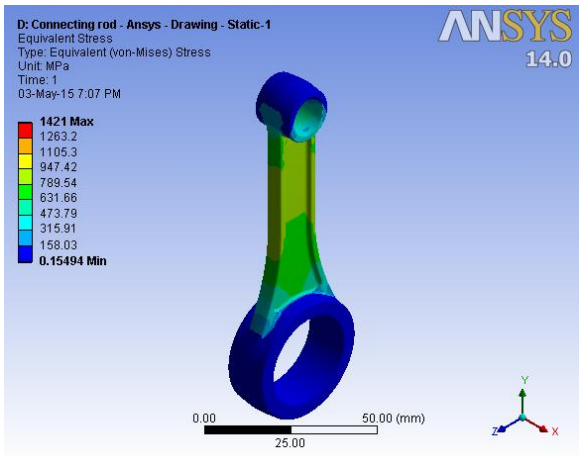


Fig. 7 Analysis of connecting rod (16MnCr5) Existing product (Eq. Stress)

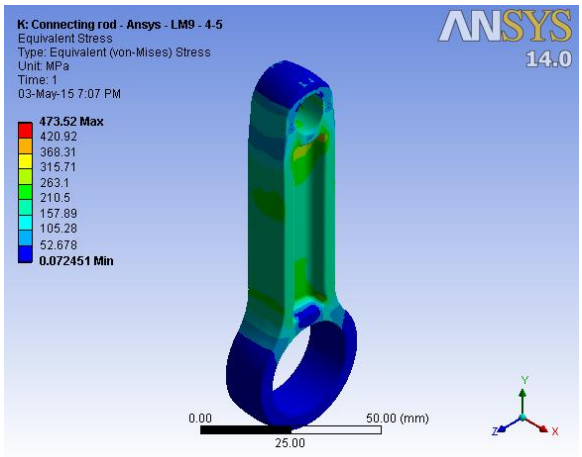


Fig. 8 Analysis of connecting rod (Aluminium LM9) (Eq. Stress)

TABLE 4
COMPARISON OF FEA RESULTS FOR STRESSES & DEFORMATION

Material	**Carbon Steel 16MnCr5**	**Aluminium LM9**
Total deformation	0.29378 mm	0.20783 mm
Equivalent stress	1421 MPa	473 MPa
Max Principal stress	603 MPa	417 MPa

IV. EXPERIMENTAL MODAL ANALYSIS

Experimental modal analysis, also known as modal analysis or modal testing, deals with the determination of natural frequencies, damping ratios, and mode shapes through vibration testing. Two basic ideas are involved:

- When a structure, machine, or any system is excited, its response exhibits a sharp peak at resonance when the forcing frequency is equal to its natural frequency when damping is not large.
- The phase of the response changes by 180° as the forcing frequency crosses the natural frequency of the structure or machine, and the phase will be 90° at resonance.

Experimental modal analysis is done by hanging the connecting rod at small end and experimental results were compared with FEM for Carbon steel & Aluminium LM9. From the finite element analysis and experimental modal analysis the natural frequencies were determined. Fig. 13 & 14 shows the comparative results for different materials.

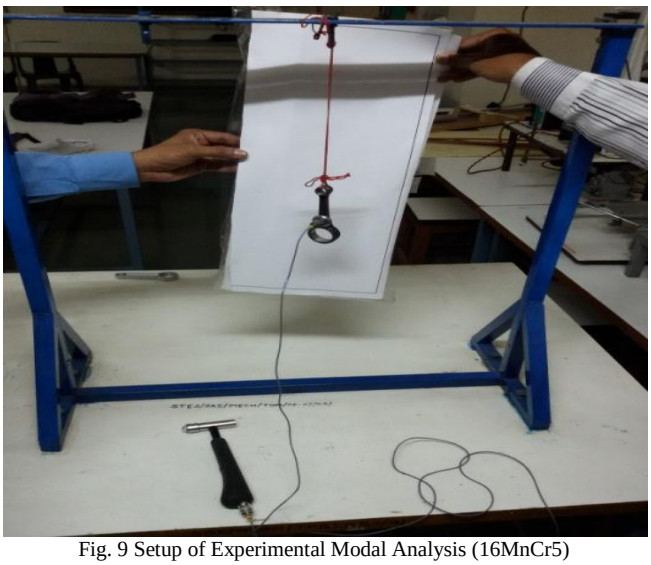


Fig. 9 Setup of Experimental Modal Analysis (16MnCr5)

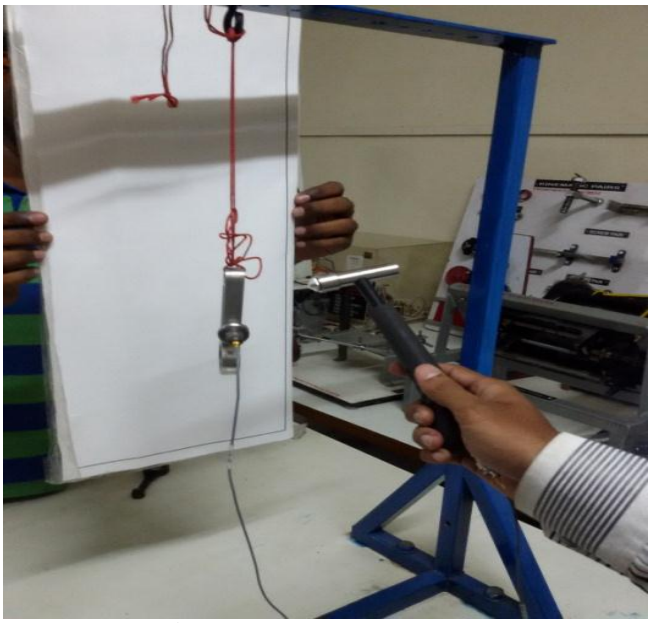


Fig. 10 Setup of Experimental Modal Analysis (Aluminium LM9)

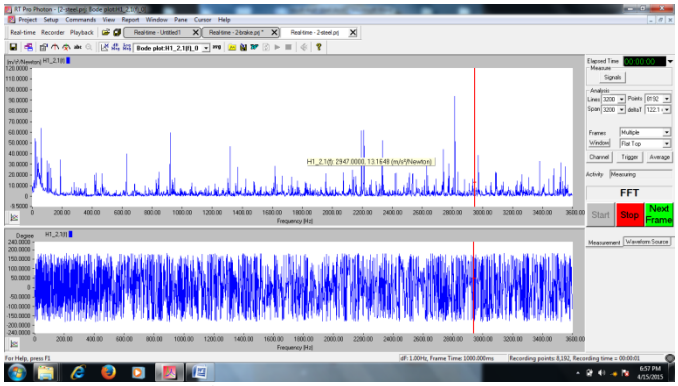


Fig. 11 Graph of Experimental Modal Analysis (16MnCr5)

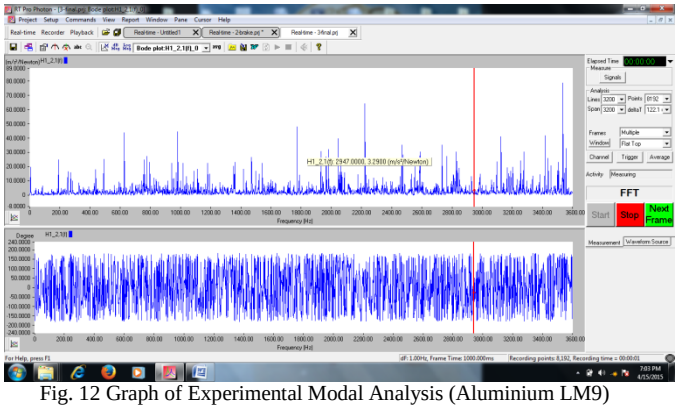


Fig. 12 Graph of Experimental Modal Analysis (Aluminium LM9)

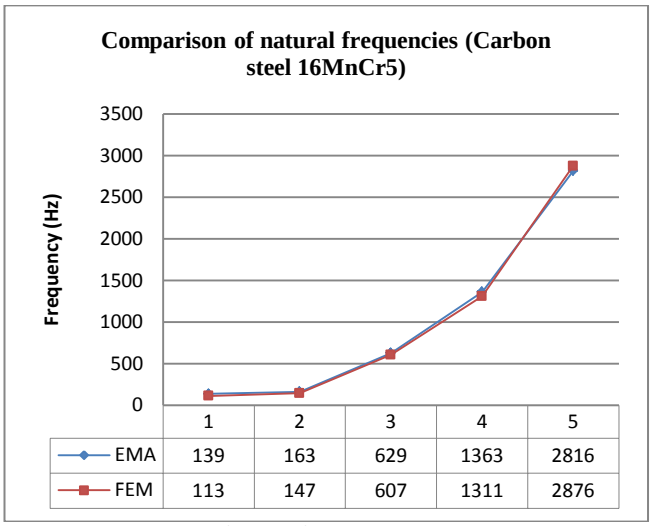


Fig. 13 Comparison of natural frequencies (Carbon steel 16MnCr5)

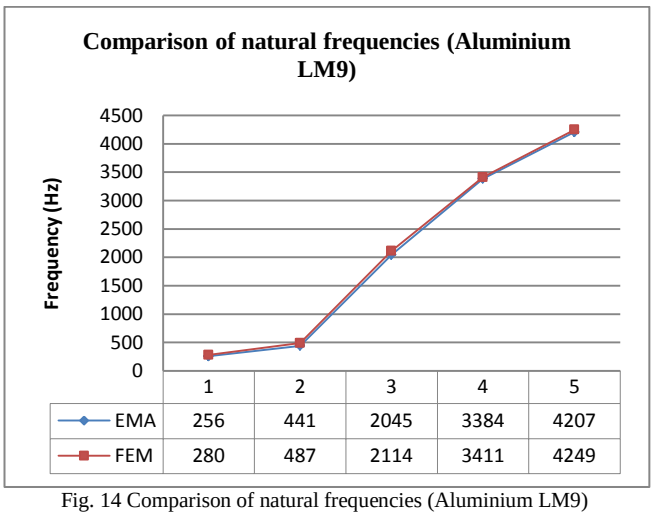


Fig. 14 Comparison of natural frequencies (Aluminium LM9)

V. Conclusion

Analysis done with pressure load applied at the piston end and restrained at the crank end or other load applied at the crank end and restrained at the piston end. The finite element analysis is carried out on carbon steel (16MnCr5) connecting rod as well as on aluminium LM9.

This work investigated suitable better material for minimizing deflections in connecting rod. Load analysis was performed which comprised of the connecting rod, small and big ends of connecting rod using analytical techniques and computer based mechanism simulation tools. FEA was then performed using the results from load analysis to gain insight on the structural behaviour of the connecting rod and to determine the design loads for optimization. The following conclusions can be drawn from this study.

The deflection of Aluminium LM9 connecting rod is less than the deflection of existing carbon steel (16MnCr5) connecting rod.

The Aluminium LM9 connecting rod shows less amount of stresses than existing carbon steel (16MnCr5) connecting rod.

It is also found that the Aluminium LM9 connecting rod is light in weight than existing carbon steel (16MnCr5) connecting rod.

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