DESIGN & COMPARATIVE ANALYSIS OF TWO WHEELER CONNECTING ROD

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Abstract: Connecting rod act as an intermediate link between piston and crank. The push and pull from the piston pin to crank pin is carried out by connecting rod which results in the change of motion of the reciprocating piston to rotary motion of the crank. The work focus on experimental investigation of connecting rod for Pulsar 220 with development in existing design. In this work the attempt is to modify the existing design of connecting rod by satisfying the desired design and application constraint. The parameter of existing connecting rod like von misses stress, shear stress, total deformation is obtained from ANSYS software. In addition this work aims to carry out the test on UTM to find deflection and stress by satisfying the desired constraint. The hardness is determined along with the chemical composition of different elements. Modification of existing connecting rod is carried out. Different geometries are verified & optimum geometry is selected which is tested experimentally & compare with existing connecting rod. In case of buckling analysis, the buckling load factor obtained for optimized connecting rod from buckling mode and critical buckling load is calculated. Results obtained from experimental & FEA analysis are validated.

***Index Terms* - Connecting rod, chemical composition, desired constraints, hardness, optimized geometry, FEA, buckling analysis etc.**

I. INTRODUCTION

A Connecting rod is the link between the reciprocating piston and rotating crank shaft. Small finish of the connecting rod is hooked up to the piston by the use of gudgeon pin. The operation of the connecting rod is to convert the reciprocating movement of the piston into the rotary movement of the crankshaft. A blend of axial and bending stresses act on the rod in operation. The axial stresses are product due to cylinder fuel strain and the inertia force bobbing up on account of reciprocating motion. Whereas bending stresses are triggered due to the centrifugal effects. To furnish the highest stress with minimal weight, the move element of the connecting rod is made as I – part. Small finish of the rod is a superior eye or a break up eye, this end holds the piston pin. The big finish relaxation on the crank pin and is continuously cut up for heavy engines. In some connecting rods, a hole is drilled between two ends for carrying lubricating oil from the big finish to the small finish for lubrication of piston and the piston pin the intermediate component between crank and piston is known as connecting rod**.** The objective of CR is to transmit push & pull from the piston pin to the crank pin and then converts reciprocating motion of the piston into the rotary motion of crank. The components are big shank, a small end and a big end. The cross section of shank may be rectangular, circular, tubular, I Section, or ellipsoidal Section. It sustains force generated by mass & fuel combustion. The resulting bending stresses appear due to eccentricities, crank shaft, case wall deformation & rotational mass.

1.1 Problem Statement

The connecting rod is subjected to a complex state of loading. It undergoes high cyclic loads of the order of 108 to 109 cycles, which range from high compressive loads due to combustion, to high tensile loads due to inertia. This project work focus on the experimental investigation of connecting rod of Pulsar-220. In order to achieve weight reduction and to avoid buckling failure in automobile the connecting rod needs to be optimised by considering the cost and weight of the component**.**

1.2 Scope

It is very difficult to change the design as well as material of existing connecting rod. Failure is found in connecting rod due to buckling phenomenon. In order to avoid buckling, modifications is done for existing design of connecting rod.

II. Literature Review

Sarkate et.al. had carried out work on the aluminum & study concluded that the stress analysis of connecting used in engine has been presented. The results obtain by FEA for both Aluminum 7068 alloy and AISI 4340 alloy steel are satisfactory for all possible loading conditions. By using aluminum 7068 alloy instead of AISI 4340 alloy steel can reduce weight up to 63.95%. Also equivalent stresses for Aluminum 7068 alloy is less by 3.59%. [1].Anusha et.al. had carried out study on “Comparison of Materials For Two-Wheeler Connecting Rod Using Ansys”. In this analysis two materials are selected and analysed. The software results of two materials are compared and utilized for designing the connecting rod[2]. Kumar et.al. had analyzed two wheeler’s connecting rod. In this work connecting rod was replaced by aluminum reinforced with boron carbide for Suzuki GS150R motorbike. Compared to carbon steel, aluminum boron carbide and aluminum 360, aluminum boron carbide is found to have working factor of safety is nearer to theoretical factor of safety, 33.17% to reduce the weight, to increase the stiffness by 48.55% and to reduce the stress by10.35% and stiffer [3]. Reddy et.al. had carried out study to determine von misses stress and pressure, deformation, aspect of defense and weight discount for two wheeler pistons and concluded that fatigue strength is the principal driving factor for the design of connecting rod and he determined that the fatigue results are in good agreement with the present outcomes. The stress is determined maximum on the piston finish so the material is improved within the stressed portion to shrink stress [4].Singh had conducted study replace the conventional material of connecting rod i.e. steel with the composite material (E-Glass / Epoxy). In this study von misses stresses, deformations and other parameters are ascertained which are performed by FEA of the connecting rod. [5].Gupta & Nawajis had carried out work of existing connecting rod material, replaced by beryllium alloy and magnesium alloy. Maximum von misses strain and maximum displacement are minimum in connecting rod of beryllium alloy in comparison of rest of two materials comparing the different results obtained from the analysis. The stress induced in the Beryllium alloy is less than other for the present investigation [6].Ram et.al. carried out static analysis on the piston pin end and crank pin end of connecting rod then further study was shown to explore weight reduction possibility. It was concluded that The Peak stresses mostly occurred in the transition area between pin end, crank end and shank region. Forces at pin end are lower in comparison to the forces in crank end so strength of pin end should ideally be lower in comparison to the strength of crank region. Factor of safety was greater than 3.7 in both tensile as well as compressive loading cases for both original as well as optimized model. Percentage weight reduction was about 13% which will save material directly to reduce the manufacturing cost with increased engine efficiency [7]. Webster et.al. explained the loading of connecting rod in diesel engine. The tension and compression loadings were used based on experimental results. It is highest stress occurred at four location of connecting rod. The upper area of cap end on the axis of symmetry, the transition region of bolt section and the lower rib, the transition region of the lower rib and connecting rod’s bolt head[8]. Pranav et. al. carried out the FEA and optimization of connecting rod using ANSYS workbench. He studied two types of analysis static analysis and fatigue analysis. The weight reduction was achieved by 9.24% under static loading conditions of existing connecting rod [9]. Pravardhan et.al. presented the FEA procedure for optimization for connecting rod weight and cost reduction. Weight reduction forged steel connecting rod by iterative procedure. The result obtained in an optimized connecting rod 10% lighter and 25% less expensive as compared to existing connecting rod [10].

III. Experimental Methodology

In order to determine the deflection tension test is carried on UTM. From the design calculations the maximum force that may act on connecting rod is 38.87 KN. Following fig indicates the systematic arrangement in which the connecting rod is fixed between the fixture and the gradual load is applied to measure the deflection at the particular load.

Fig 3.1 Connecting Rod with set up on UTM

Fig.3.2 Connecting rod after failure

3.1 Dimensions of Connecting rod.

Connecting rod of bike’s engine, which is available in market, is selected for the present investigation.

TABLE 3.1 DIMENSIONS OF CONNECTING ROD

Sr. no.	Parameters	Values (mm)

1	Length of connecting rod.	144.66
2	Outer diameter of big end.	48.69
3	Inner diameter of big end.	37.92
4	Outer diameter of small end.	23.10
5	Inner diameter of small end.	16.88
6	Width	14.80
7	Thickness	3.87

3.2 Design Calculations

Connecting rod, subjected to pressure, gas pressure- which can be then used to find the maximum force that can act on connected rod. From the specification it can be observed that the maximum BP of the pulsar is 20 HP, while number of working stoke are 4250 RPM. In the subsequent section the maximum pressure is find out with other relevant parameters. To find the maximum pressure from actual specifications,

B.P = 20.8 BHP = 20.8 × 746 = 15516.8 Watt (1)

Mechanical efficiency is taken as 80% = ŋ = 0.8

P = (BP/ ŋ) = 19396 Watts. (2)

I.P = (P × L × A × N / 60) (3)

I.P = 1.246×10^6 N/m^2 = 1.246 N/mm^2

Usually maximum pressure is taken as 7 to 10 times of mean effective pressure so P_{max} will be from 8.82 N/mm^2 to 12.46 N/mm^2

- *To find N*

N = No of working stroke/minute

= N (2 stroke)

N = N / 2 (4 stroke) (4)

N = 8500 / 2

N = 4250 RPM (5)

- *Determination of load applied on CR*

Pressure = Force / Area (6)

The maximum pressure is 7 to 10 times we assume Pmax as 7 times mean effective pressure.

P_{max} = 1.26 × 7 = 8.82 N/mm^2. (7)

Force = Pressure × Area (8)

Force = 8.82 × 3525.65

Force = 31.096 KN.

Force = 31.096 × 1.25 (1.25 is taken as Factor of safety)

Force = 38.87 KN (9)

IV FEA OF EXISTING CONNECTING ROD

Connecting rod is modeled by taking design parameters of pulsar 220 & then by using CATIA software solid modeling is done which is shown in below fig. The model is imported in ANSYS15 workbench. The material properties were given. Meshing of the model is done. The crank end of connecting rod is kept fix and the load is applied on piston end at force of 38 KN. After meshing the model, boundary conditions such as loads and constrains are imposed. One of the important factors to get accurate results is to apply correct load & boundary conditions. Final stage of analysis is running a solver to get the desired results. The result for maximum von misses stress, deformed shape, Shear Stress are noted from ANSYS.

Fig. 4.1 3D model of connecting rod

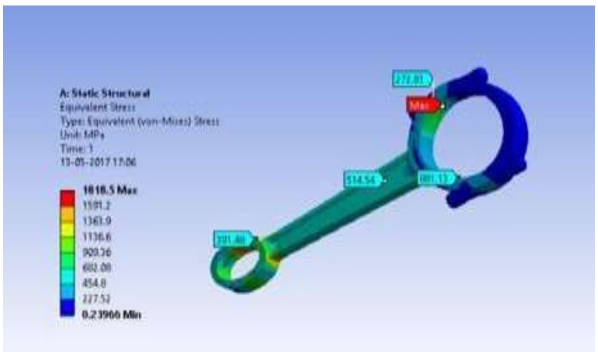

Fig. 4.2 Von-mises stress on connecting rod 1818.5 Mpa

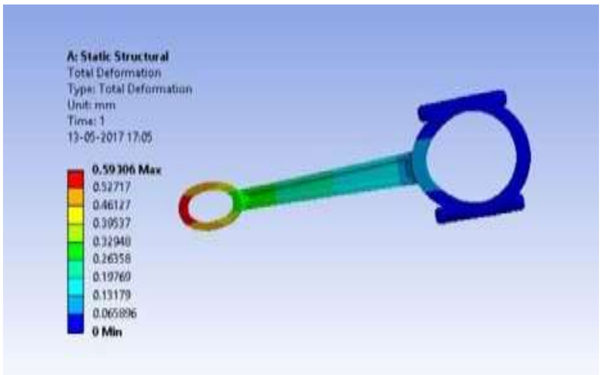

Fig. 4.3 Deformation on connecting rod is 0.593 mm

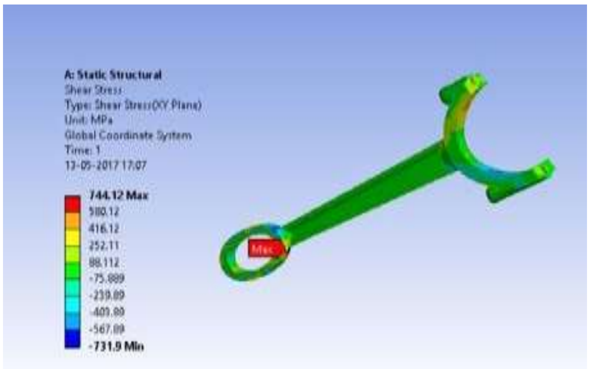

Fig. 4.4 Shear stress on connecting rod 744.12 Mpa

4.1 Modification in existing design of Connecting Rod

From literature survey it was found that failure in connecting rod occurs due to buckling effect. Different models of connecting rod with same dimensions but modifications at shank end was created using CATIA V5 R16. Solid modelling of connecting rod for four modifications is shown below.

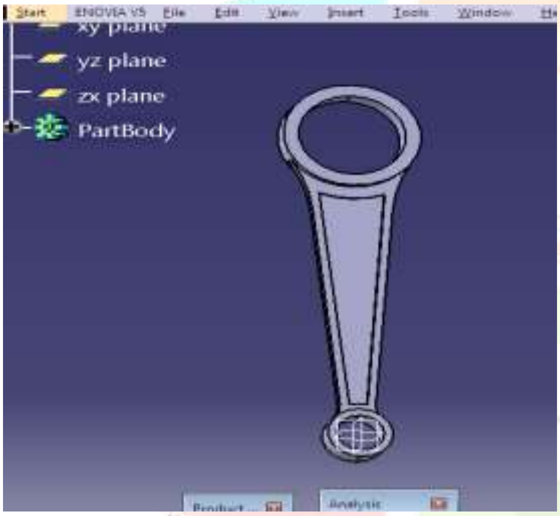

Fig. 4.1.1 modification in existing connecting rod model no 1

Fig. 4.1.2 modification in existing connecting rod model no 2

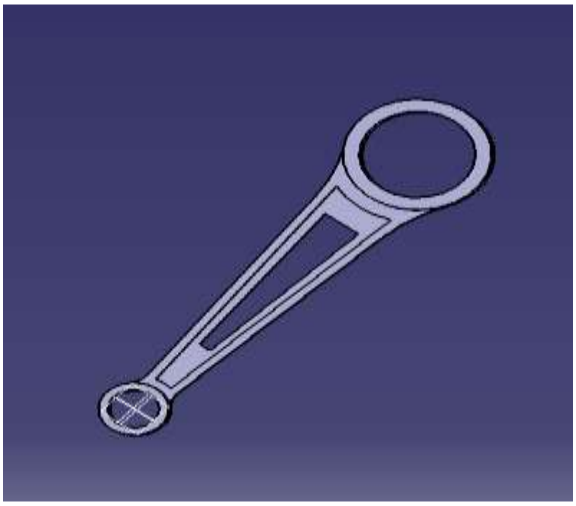

Fig. 4.1.3 modification in existing connecting rod model no 3

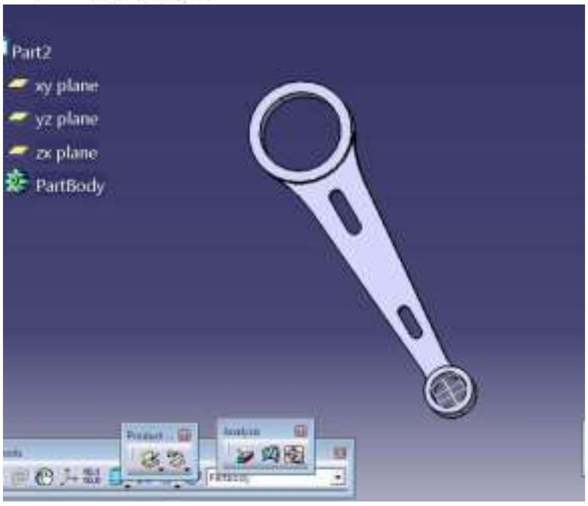

Fig. 4.1.4 modification in existing connecting rod model no 4

4.2 FEA analysis of Modified Connecting rod

Different modified geometry created using CATIA model are verified. The optimum geometry, will be tested & compare with existing connecting rod. From below FEA analysis it is found that stress induced for fig 4.2.3 is less compared to other modifications.

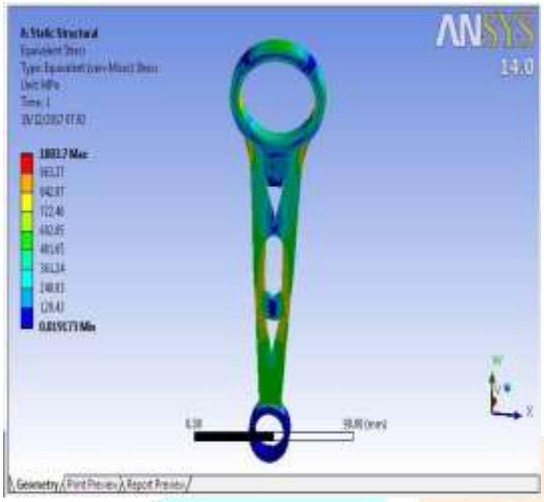

Fig. 4.2.1 Maximum Von mises Stress – 1083.7 Mpa

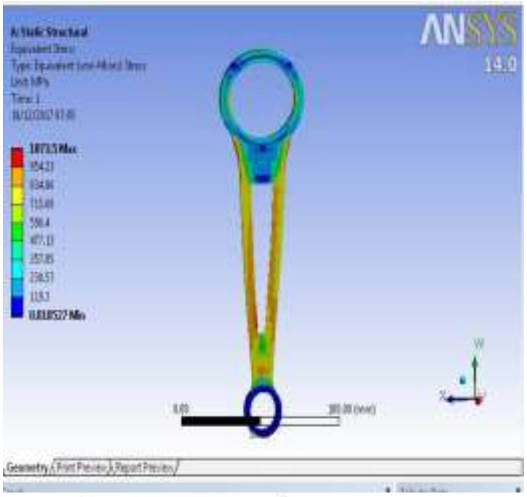

Fig. 4.2.2 Maximum Von mises Stress – 1073.5 Mpa

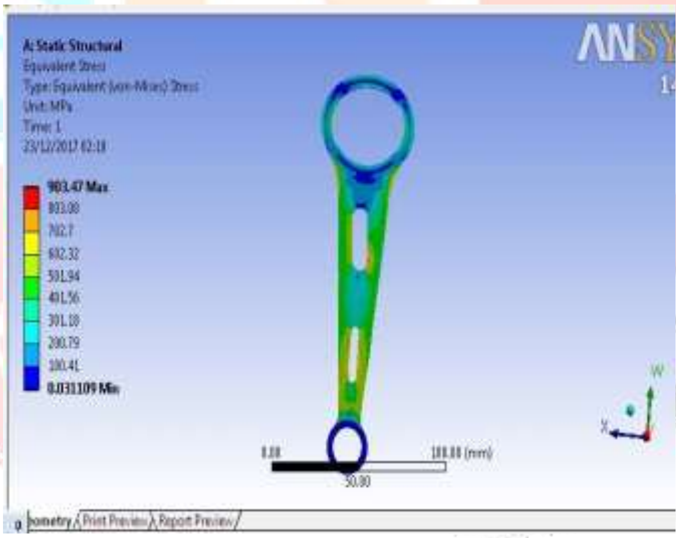

Fig. 4.2.3 Maximum Von mises Stress – 903.47 Mpa

V BUCKLING ANALYSIS OF CONNECTING ROD

Buckling analysis is a technique used to determine buckling loads-critical loads at which a structure becomes unstable and buckled mode shapes the characteristic shape associated with a structure's buckled response. In linear static analysis, a structure is normally considered to be in a state of stable equilibrium. As the applied load is removed, the structure is assumed to return to its original position. However, under certain combinations of loadings, the structure may become unstable. When this loading is reached, the structure continues to deflect without an increase in the magnitude of the loading. In this case, the structure has actually buckled or has become unstable; hence, the term “instability” is often used interchangeably with the term “buckling”.

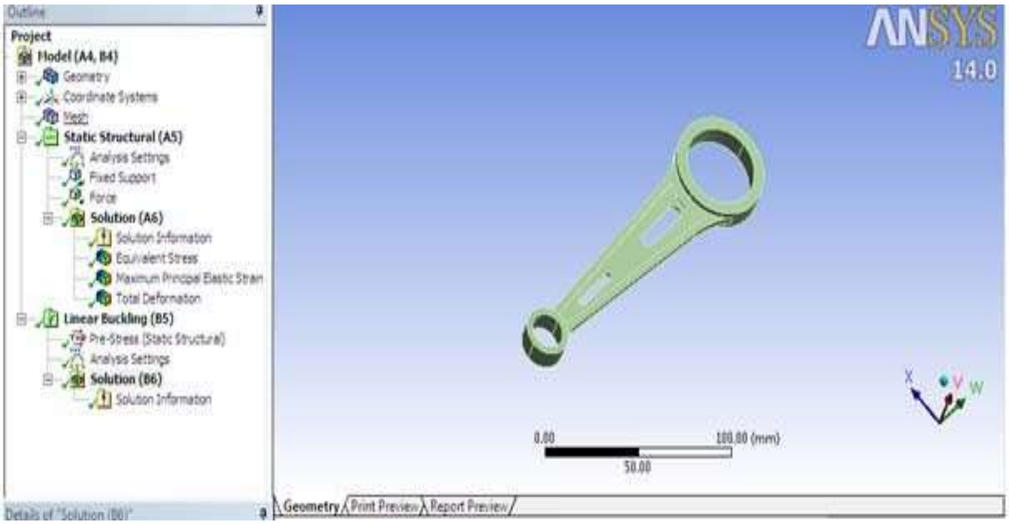

Fig. 5.1 Buckling load factor = 6.7610

5.1 Max. Buckling load P_{cr} for connecting rod using FEA results

The maximum buckling load (the estimated load that induces buckling) as calculated using the methodology of linear buckling analysis is equal to the second mode eigen value multiplied by the applied load. Therefore, the Max. /Critical Buckling load for Connecting rod is given by,

Buckling Load = Applied Load x Buckling Load factor

= 31034.29 x 6.7610

= 209822.83N

The critical buckling load obtained for modified connecting rod using FEA is 209822.83 N.

5.2 Calculations to find critical buckling load for connecting rod

Buckling of connecting rod in a plane of location is given by, following formula

$$P_{cr} = \frac{SycxAc}{1+a(\frac{Lc}{Kyy})2} \tag{1}$$

P_{cr}= Critical/Max. Load in N.

Kyy =Radius of gyration in mm.

α = 1/7500

L = Actual length in mm.

Kyy^2 = Iyy/A

Iyy = $1/12BH^3$-1/12 $[(B-t)\ (H-2t)^3]$

=1/12(5t) x $(6t)^3$-1/12$[(5t-t)(6t-2t)^3]$

= 50.6667 t^4 mm^4

Kyy^2 = Iyy/A = $3.619t^4$

Kyy = 1.9023 t mm = 1.9023x5 = 9.5115 mm

Le = equivalent length

= 0.5 x l = 0.5 x 120 = 60 mm

$$P_{cr}= \frac{600x\ 14t2}{1+1/7500(\frac{60}{9.5115})2}$$

Pcr = 209823.7480 N (Maximum critical buckling load)

As compare to all cases buckling load factor and buckling load value for optimized connecting rod is maximum so we can conclude that the connecting rod with modified design will not fail for buckling load.

VI CONCLUSION

i. Experimental analysis of existing material is carried out which is in good agreement with finite element analysis.

ii. By comparing the results obtain from the above analysis it can be conclude that the stress induced in modified connecting rod is less than the stress induced in existing connecting rod.

iii. The theoretical calculated critical buckling load for modified connecting rod is 209823.74N closely meets with the FEA critical buckling load for optimized connecting rod is 209822.83N.

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