International Journal of Research in Engineering, Science and Management
Volume-2, Issue-11, November-2019
www.ijresm.com | ISSN (Online): 2581-5792

Design and Analysis of a Connecting Rod

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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 describes designing and Analysis of connecting rod. Currently existing connecting rod is manufactured by using forged steel. In this drawing is drafted from the calculations. A parametric model of Connecting rod is modeled using CATIA V5 software and to that model, analysis is carried out by using ANSYS 14.0 Software. Finite element analysis of connecting rod is done by considering the materials, viz. Aluminium Alloy. The best combination of parameters like Von misses Stress and strain, Deformation, Factor of safety and weight reduction for two wheeler piston were done in ANSYS software. Aluminium Alloy has more factor of safety, reduce the weight, reduce the stress and stiffer than other material like Forged Steel. With Fatigue analysis we can determine the lifetime of the connecting rod.**

***Keywords*: connecting Rod, Analysis of connecting rod, four stroke engine connecting rod, forged steel connecting rod.**

1. Introduction

In a reciprocating piston engine, the connecting rod connects the piston to the crank or crankshaft. In modern automotive internal combustion engines, the connecting rods are most usually made of steel for production engines, but can be made of aluminum (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 small end attaches to the piston pin, gudgeon pin (the usual British term) or wrist pin, which is currently most often press fit into the con rod but can swivel in

the piston, a "floating wrist pin" design. The connecting rod is under tremendous stress from the reciprocating load represented by the piston, actually stretching and being compressed with every rotation, and the load increases to the third power with increasing engine speed. Failure of a connecting rod, usually called "throwing a rod" is one of the most common causes of catastrophic engine failure in cars, frequently putting the broken rod through the side of the crankcase and thereby rendering the engine irreparable; it can result from fatigue near a physical defect in the rod, lubrication failure in a bearing due to faulty maintenance or from failure of the rod bolts from a defect, improper tightening, or re-use of already used (stressed) bolts where not recommended. Despite their frequent occurrence on televised competitive automobile events, such failures are quite rare on production cars during normal daily driving. This is because production auto parts have a much larger factor of safety, and often more systematic quality control. When building a high performance engine, great attention is paid to the connecting rods, eliminating stress risers by such techniques as grinding the edges of the rod to a smooth radius, shot peening to induce compressive surface stresses (to prevent crack initiation), balancing all connecting rod/piston assemblies to the same weight and Magna fluxings to reveal otherwise invisible small cracks which would cause the rod to fail under stress. In addition, great care is taken to torque the con rod bolts to the exact value specified; often these bolts must be replaced rather than reused. The big end of the rod is fabricated as a unit and cut or cracked in two to establish precision fit around the big end bearing shell. Recent engines such as the Ford 4.6-liter engine and the Chrysler 2.0-liter engine have connecting rods made using powder metallurgy, which allows more precise control of size and weight with less machining and less excess mass to be machined off for balancing. The cap is then separated from the rod by a fracturing process, which results in an uneven mating surface due to the grain of the powdered metal. This ensures that upon reassembly, the cap will be perfectly positioned with respect to the rod, compared to the minor misalignments, which can occur if the mating surfaces are both flat. A major source of engine wear is the sideways force exerted on the piston through the con rod by the crankshaft, which typically wears the cylinder into an oval cross-section rather than circular, making it impossible for piston rings to correctly seal against the cylinder walls. Geometrically, it can be seen that longer connecting rods will reduce the amount of this sideways force, and therefore lead to longer engine life. However, for a given engine block, the sum of the length of the con rod plus the piston stroke is a fixed number, determined by the fixed distance between the crankshaft axis and the top of the cylinder block where the cylinder head fastens; thus, for a given cylinder block longer stroke, giving greater engine displacement and power, requires a shorter connecting rod (or a piston with smaller compression height), resulting in accelerated cylinder wear.

2. Literature survey

Pravardhan S. Shenoy and Ali Fatemi (2005) [1] carried out the dynamic load analysis and optimization of connecting rod.

The main objective of this study was to explore weight and cost reduction opportunities for a production forged steel connecting rod. Typically, an optimum solution is the minimum or maximum possible value the objective function could achieve under a defined set of constraints. The weight of the connecting rod has little influence on the cost of the final component. Change in the material, resulting in a significant reduction in machining cost, was the key factor in cost reduction. As a result, in this optimization problem the cost and the weight were dealt with separately. The structural factors considered for weight reduction during the optimization include the buckling load factor, stresses under the loads, bending stiffness, and axial stiffness. Cost reduction is achieved by using C-70 steel, which is fracture crackable. It eliminates sawing and machining of the rod and cap mating faces and is believed to reduce the production cos t by 25%.

In the thesis by Shahrukh Shamim, Optimization was performed to reduce weight and manufacturing cost of a forged steel connecting rod subjected to cyclic load comprising the peak compressive gas load and the peak dynamic tensile load at 5700 rev/min, corresponding to 360o crank angle.

In the thesis by K. Sudershn Kumar, Dr. K. Tirupathi Reddy, Syed Altaf Hussain, for considering the parameters, the working factor of safety is nearer to theoretical factor of safety in aluminum boron carbide. Percentage of reduction in weight is same in Aluminum 360 and aluminum boron carbide. Percentage of increase in stiffness in aluminum boron carbide is more. Percentage of reducing in stress ALUMINIUM BORON CARBIDE and ALUMNUM is same than CARBON STEEL.

In the paper by Suraj Pal, Sunil kumar, Finite Element analysis of the connecting rod of a Hero Honda Splendor has been done using FEA tool ANSYS Workbench. It is concluded that the weight of the connecting rod is also reduced by 0.477g. Thereby, reduces the inertia force. Fatigue strength is the most important driving factor for the design of connecting rod and it is found that the fatigue results are in good agreement with the existing result

In the journal paper by Prof. Vivek C. Pathade, Dr. Dilip S. Ingole, From the theoretical, Finite Element Analysis and Photoelastic Analysis it is found that i) The stresses induced in the small end of the connecting rod are greater than the stresses induced at the big end. ii) Form the photoelastic analysis (from the fringe developed in the photoelastic model of connecting rod) it is found that the stress concentration effect exists at both small end and big end and it is negligible in the middle portion of the connecting rod. iii) Therefore, the chances of failure of the connecting rod may be at fillet section of both end

In the paper by Priyank D. Toliya, Ravi C. Trivedi, Prof. Nikhil J. Chotai, the objective of this research is to investigate the failure analysis of the connecting rod of the automotive engine. Apart from conventional material of connecting rod I choose the connecting rod of FM-70 Diesel engine which is made of Aluminium 6351. static analysis is done to determine the von Misses stress, elastic strain, total deformation in the present design connecting rod for the given loading conditions using the FEM Software Ansys 12.1. In the starting of the work, the static loads acting on the connecting rod, after that the work is carried out for safe design and life in fatigue. Fatigue Analysis is compared with the Experimental results The best combination of parameters like Von misses stress; Maximum shear stress and weight reduction for Four stroke diesel engine were studied in ProE software. In the paper by Tukaram S. Sarkate, Sachin P. Washimkar, Sachin S. Dhulekarsssss, it is concluded that The stress analysis of connecting used in engine has been presented and discuss in this paper. 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%. The factor of safety of connecting rod will reduce by 9.77% in case tensile load applied at crank end but it will increase in all other load conditions if Aluminum7068 alloy is used instead of AISI 4340

3. Methodology

A. Problem statement

The objective of the present work is to design and analyses of connecting rod made of Aluminium Alloy. Steel materials are used to design the connecting rod. In this project the material (Forged steel) of connecting rod replaced with Aluminium Alloy. Connecting rod was created in CATIA V5. Model is imported in ANSYS WORKBENCH 14.0 for analysis. After analysis a comparison is made between existing steel connecting rod viz., An Aluminium Alloy in terms of weight, factor of safety, stiffens, deformation and stress.

B. Design of a connecting rod

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 rankine 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

According to Rankine formula

W_{cr} About x-axis = $[\sigma_c *A]/1+[L/k_{xx}]^2 = [\sigma_c *A]/1+[l/k_{xx}]^2$, [for both ends hinged L =l]

W_{cr} About Y-axis = $[\sigma_c *A]/1+[L/k_{yy}]^2=[\sigma_c *A]/1+a[l/2k_{yy}]^2$, [for both ends hinged L =l/2]

In order to have a connecting rod equally strong in buckling about both the axis, the buckling loads must be equal. i.e.

$[\sigma_c *A]/1+[L/k_{xx}]^2 = [\sigma_c *A]/1+a[l/2k_{yy}]^2$ (or) $[l/k_{xx}]^2 = [l/2k_{yy}]^2$

$k_{xx}^2 = 4k_{yy}^2$ (or) $I_{xx} = 4I_{yy}$

This shows that the connecting rod is four times strong in buckling about y-axis than about axis.

If $I_{xx} > 4I_{yy}$, then buckling will occur about y-axis and if $I_{xx} > 4I_{yy}$, then buckling will occur about x-axis. In Actual practice I_{xx} is kept slightly less than $4I_{yy}$. It is usually taken between 3 and 3.5 and the Connecting rod is designed for buckling about x-axis. The design will always be satisfactory for buckling about y-axis. The most suitable section for the connecting rod is I-section with the proportions shown mfg.

Area of the cross section = $2[4t \times t] + 3t \times t = 11t^2$

Moment of inertia about x-axis = $2[4t\times t] + 3t\times t = 11t^2$

Moment of inertia about x-axis $I_{xx} = 1/12\,[4t\{5t\}33-3t\,\{3t\}^3] = (419/12)[t]^4$

And moment of inertia about y-axis $I_{yy} = (2\times1/12)\times t\times\{4t\}^3+(1/12)\{3t\}t^3 = 131/12[t]^4$

$I_{xx} / I_{yy} = [419/12] \times [12/131] = 3.2$

Since the value of I I_{xx} / I_{yy} lies between 3 and 3.5 m therefore I section chosen is quite satisfactory. Precleaning is an essential first step in the penetrant process. The surface must be thoroughly cleaned to assure that all contaminants and other materials that may prohibit or restrict the entry of the penetrant into surface openings are removed. Thorough cleaning is essential if the examination results are to be reliable.

C. Pressure Calculation for 150 cc Engine Suzuki 150 cc Specifications

Engine type air cooled 4-stroke, Bore x Stroke (mm) = 57×58.6

Displacement = 149.5 CC, Maximum Power = 13.8 bhp @ 8500 rpm

Maximum Torque = 13.4 Nm @ 6000 rpm, Compression Ratio = 9.35/1,

Density of Petrol C8H18 = 737.22 kg/m^3 = 737.22_x001D_-kg/", Temperature = 60^0 c

F = 288.855 o K, Mass = Density × Volume

= 737.22E^{-9}- x149.5E^3 = 0.11kg

Molecular Weight of Petrol 114.228 g/mole from Gas Equation,

PV = Mrt R = (R_x / M_w) = 8.3143/114228

= 72.76P = (0.11×72.786×288.85)/149.5E^3

P = 15.5 Mpa

D. Design calculations for existing connecting rod

Thickness of flange & web of the section = t

Width of section B= 4t

The standard dimension of I - SECTION

Height of section H = 5t, Area of section A=2(4t×t) +3t×t, A = 11t²

M. O.I of section about x axis: $I_{xx} = 1/12\,[4t\,\{5t\}^3-3t\,\{3t\}^3] = 419/12[t^4]$

MI of section about y axis: $I_{yy} = 2\times1/12\times t\times\{4t\}^3+1/12\{3t\}t^3 = 131/12[t^4]$

$I_{xx} / I_{yy} = 3.2$

Length of connecting rod (L) = 2 times the stroke, L = 117.2 mm

Buckling load W_b= maximum gas force × F.O.S

W_b = 37663N,

Stroke length (l)= 117.2 mm

Diameter of a piston (D) = 57 mm

P=15.5 N/mm^2

Radius of crank (r) =stroke length /2

=58.6/2 =29.3

Maximum force on the piston due to pressure

FI =$\Pi/4*D^2*P = \Pi/4*(57)^2*15.469$ =39473.16N

Maximum angular speed $w_{max} = [2\Pi N_{max}]/60 = \{[2\Pi*8500]/60\}$, $A=\Pi r^2$ = 768 rad /sec

Ratio of length of connecting rod to radius of crank N = 1/r = 112/29.3 =3.8

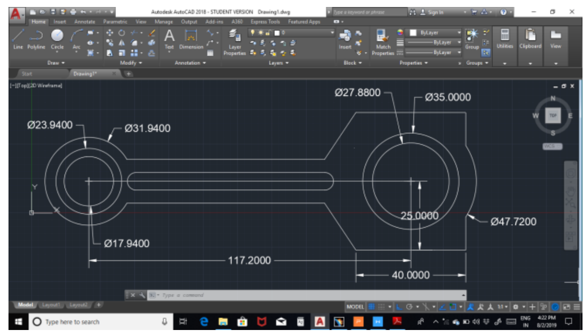

Fig. 1. 2D drawing for connecting rod

Maximum inertia force of reciprocating parts $F_{im} = Mr(w_{max})^2\, r\,(\cos\theta + \cos 2\theta/n)$ (or)

$F_{im} = Mr(w_{max})^2\, r\,(1+l/n) = 0.11*(768)^2*(0.0293)*(1+(1/3.8))$,

F_{im} =2376.26 N

Inner diameter of the small end $d_1 = F_g / pb_1 * l_1 =$ 6277.167/12.5*1.5d_1 =17.94 mm

σ_c = compressive yield stress = 415 Mpa,

$K_{xx} = I_{xx}/A$, K_{xx} =1.78t, a= σ_c/π^2, a= 0.0002

By substituting σ_c, A, a, L, K_{xx} on W_B then = $4565t^4-37663t^2-81639.46=0$

t^2 = 10.03, t = 3.167 mm,

t = 3.2 mm

Width of section B = 4t =4*3.2 = 12.8 mm

Height of section H = 5t = 5*3.2 =16 mm

Area A = $11t^2 = 11*3.2^2$

= 112.64 mm^2

Height at the big end (crank end) = H_2 = 1.1H to 1.25 H =1.1*16

H_2 =17.6 mm

Height at the small end (piston end) = H_1 = 0.9H to 0.75 H =0.9*16

H_1 = 12 mm

Where,

Design bearing pressure for small end Pb_1 = 12.5 to 15.4 N/mm^2

length of the piston pin l_1 = (1.5 to 2) d_1

outer diameter of small end = $d_1+2t_b+2t_m$

=17.94+[2*2]+[2*5] = 31.94 mm

Where, thickness of the bush (t_b) = 2 to 5 mm
Marginal thickness (t_m) = 5 to 15 mm
Inner diameter of a big end $d_2 = F_g / pb_1 * l_2$
$=6277.167/10.8*1.0d_1 = 23.88$ mm
Where, design bearing pressure foe big end pb_2 = 10.8 to 12.6 N/mm^2
length of the crank pin $l_2 = (1.0 \text{ to } 1.25)d_2$, root diameter of the bolt $= ((2F_{im})/(\pi *st))^{1/2}$
$= (2*6277.167\pi *56.667)^{1/2} = 4$ mm
outer diameter of a big end $= d_2+2t_b+2d_b+2t_m$
$= 23.88+2*2+2*4+2*5 = 47.72$ mm
Where, Thickness of the bush [t_b] = 2 to 5 mm
Marginal thickness [t_m] = 5 to 15 mm
Nominal diameter of bolt [d_b] = 1.2 * root diameter of the bolt = 1.2×4 = 4.8mm.

Table 1
Parameters of connecting rod

Sno	Parameters (mm)
1	Thickness of the connecting rod (t) = 3.2
2	Width of the section (B = 4t) = 12.8
3	Height of the section(H = 5t) = 16
4	Height at the big end = (1.1 to 1.125)H = 17.6
5	Height at the small end = 0.9H to 0.75H = 14.4
6	Inner diameter of the small end = 17.94
7	Outer diameter of the small end = 31.94
8	Inner diameter of the big end = 23.88
9	Outer diameter of the big end = 47.72

4. Analysis of the connecting rod

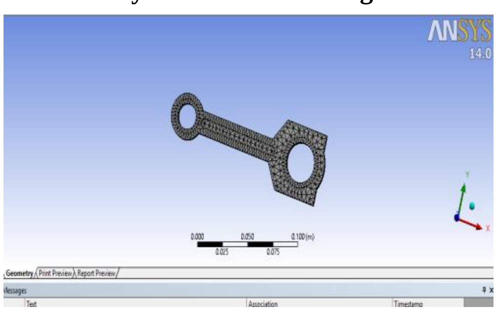

Fig. 2. Meshing of connecting rod

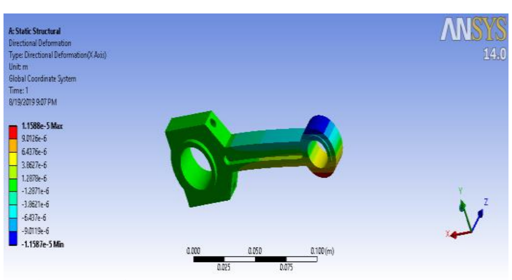

Fig. 3. Directional deformation (X-axis)

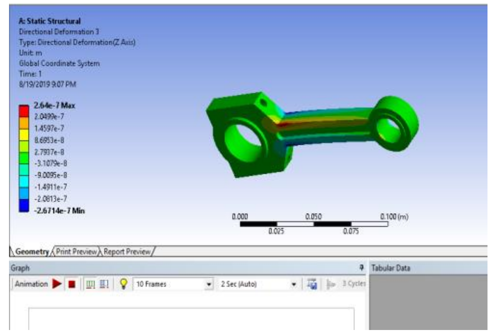

Fig. 4. Directional deformation (Z-axis)

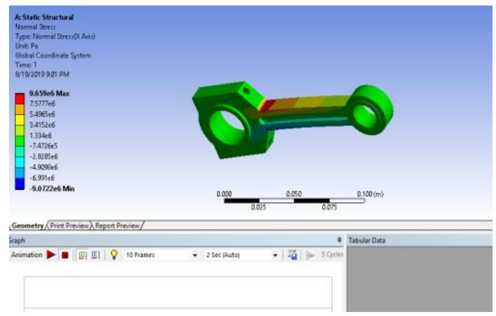

Fig. 5. Normal Stress-X-axis

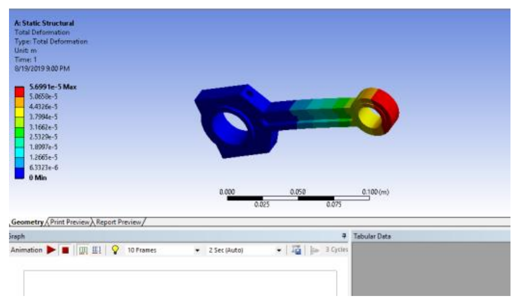

Fig. 6. Total deformation

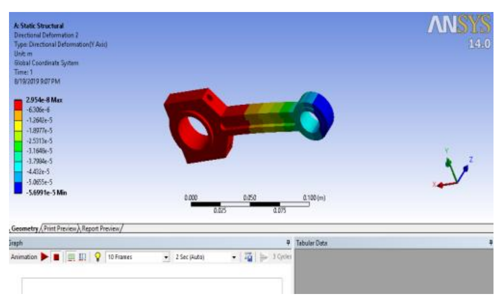

Fig. 7. Directional deformation (Y-axis)

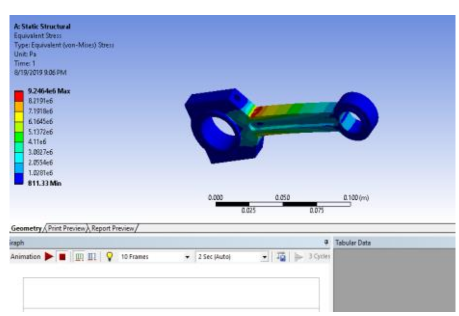

Fig. 8. Equivalent stress

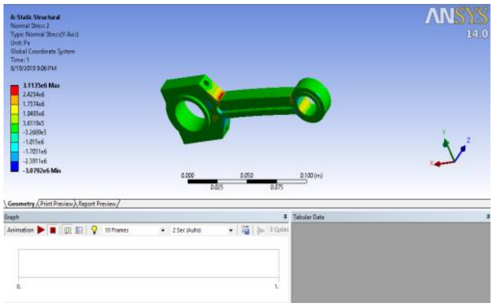

Fig. 9. Normal Stress-Y-axis

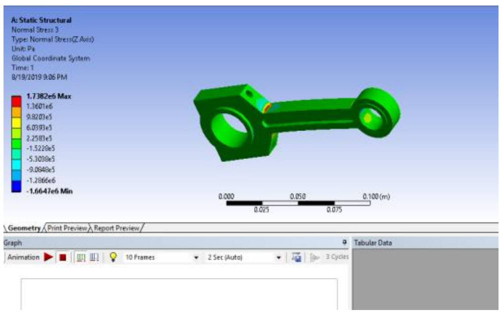

Fig. 10. Normal Stress-Z-axis

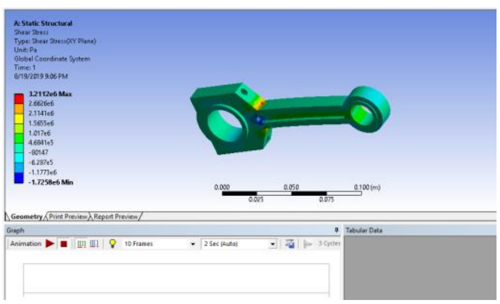

Fig. 11. Shear stress XY

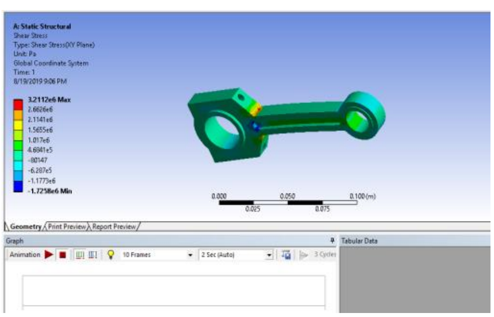

Fig. 12. Shear stress XZ

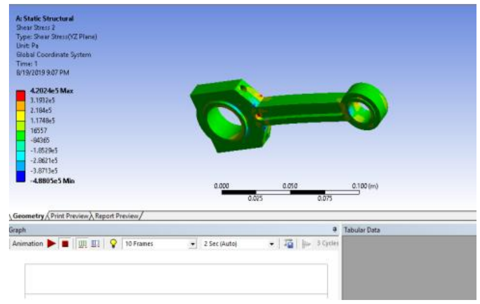

Fig. 13. Shear stress YZ

5. Conclusion and future scope

Solid modeling of connecting rod were made according to production drawing specification and analysis under the effect of tensile and compressive loads in terms of pressure is done in ANSYS Workbench 14.0. In the present design and analysis of connecting rod using aluminum alloy have been done with the help of CATIA V5 and ANSYS 14.0. Here Analysis is done for the Normal stress as well as Shear stress in x-y plane. From modeling and simulation, CATIA V5 is good but for the Analysis, it is observed that Ansys is better than other software. Here we can find minimum stresses among all loading conditions, were at crank end cap as well as at piston end. So the material can be reduced from those portions, thereby reducing material cost.

For further optimization of material dynamic analysis of connecting rod is required. After considering dynamic load condition, once again finite element analysis will have to be perform. It will give more accurate results than existing results.

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