Comparative Study of AISI 4340 and Al 7068 Connecting Rod

Vikrant K. Dhone[1]
Post Graduate Student
Department of Mechanical Engineering
Veermata Jijabai Technological Institute
Mumbai, India
Vikrantdhone992@gmail.com

A.S. Rao[2]
Assistant Professor
Department of Mechanical Engineering
Veermata Jijabai Technological Institute
Mumbai, India
asrao@vjti.org.in

Vinaay Patil[3]
Manager
Vaftsy CAE
Pune, India
vinaay.patil@vaftsycae.com

Abstract—The main objective of this paper is to study the possibility of replacing steel connecting rod with lightweight Aluminium connecting rod. In order to do so first original connecting rod made of Steel (AISI 4340) was analysed for different loading condition, this will be a baseline for comparison. Steel was replaced by Aluminium Alloy (AL 7068), this alloy is the strongest commercially available aluminium alloy. Aim was to reduce weight and study its effect on various property of Component. This has entailed performing a detailed maximum compressive and tensile load capacity, fatigue, stress, strain and natural frequency.

Keywords*- Finite element Analysis,Connecting rod, Boundery Condition,*

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I. INTRODUCTION

A connecting rod acts as a link between the piston assembly and crankshaft results in converting the reciprocating motion of piston into the rotary motion of crankshaft. Around the globe connecting rod is produced in large quantity and furthermore it works under high tensile and compressive loads. Automotive should be lighter as possible, should consume less fuel and at the same time they should provide comfort and safety to passengers. This unfortunately leads to increase in weight of the vehicle. Such tendency in vehicle construction led the invention and implementation of quite new materials which are light and meet design requirements. Lighter connecting rods help to decrease lead caused of forces of inertia in engine as it does not require big balancing weight on crankshaft. So a connecting rod should be designed in such a way that it can withstand high stresses that are produced in actual condition. Connecting rod as mainly three parts namely- a pin end, a shank region and a crank end. Pin end is connected to the piston assembly and crank end is connected to crankshaft. To perform the stress analysis on this parts any CAD software can be used to generate model and analyse it by using FEA software. Discovering new techniques and methods for weight Stress and strain reduction which can definitely increase the engine performance and economy is our objective.

II. METHADALOGY

First the basic study of designing method and manufacturing process for connecting rod were thoroughly studied from various handbooks and journals. This will include design of connecting rod under load condition and fulfilling safety norms. Also it will help in understanding problems arrived while maintenance and its remedy by varying design of assembly part. Those concept were applied to existing con rod which is part of existing system. By applying Boundary conditions various result like displacement stress and strain were obtained. This results were the baseline and for further compression it is used. Analysis were made in Ansys. Now steel is replaced by Aluminium. Same boundary conditions were applied and results were obtain for Aluminium. A comparative study was done between Steel and Aluminium. All this analysis will be done by using both Tetra and hex element.

III. PROBLEM DEFINATION

The objective of this work was to optimize the steel connecting rod for its weight. The optimization of steel rod is more intended to work with different material so as to have light weight and adequate strength. Optimization begins with identifying the correct load conditions and magnitudes. Overestimating the loads will simply raise the safety factors. The idea behind optimizing is to retain just as much strength as is needed. In this problem no such change in geometry is allowed. This is because the component is already a part of assembly, by doing reverse engineering same part is to be generated but with sufficient amount of load.

The design and weight of the connecting rod have influence on car performance. Hence, it effects on the car manufacture credibility. Change in the design and material results a significant increment in weight and also performance of the engine. The structural factors considered for weight reduction during the optimization include the buckling load factor, stresses under the loads, bending stiffness, and axial stiffness. Thus, the component can give the higher strength, efficient design and lighter that would create a major success in the automotive industry.

IV. BOUNDERY CONDITION

Connecting rod is subjected to both tensile and compressive load. But this load is applied at inner side of small end eye, which is distributed as a cosine distribution. This distribution occur because of contact non-linearity. In order to accommodate this separate shaft were place at both end. While applying load one shaft was kept fix and other is applied with either compressive or tensile load. For our application

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Maximum load that is to be applied was 12500 N tensile load and 40500 N as compressive load for both steel and aluminium were finally analysed at this load.

V. ANALYSIS OF AISI 4340

CAD model of connecting rod was already given but the boundary condition or the loading condition was unknown. To know the original boundary condition we are supposed to analyze the connecting rod for its maximum strength. In reference[1] detail of loads and designing method is explained. Using this method we would reverse engineer the condition for which it was designed. It was give that for analysis factor of safety of 2.5 is to be used. For AISI tensile yield strength is 710 MPa, thus the allowable tensile yield strength will be 284MPa and allowable compressive stress will be 454.4 MPa.

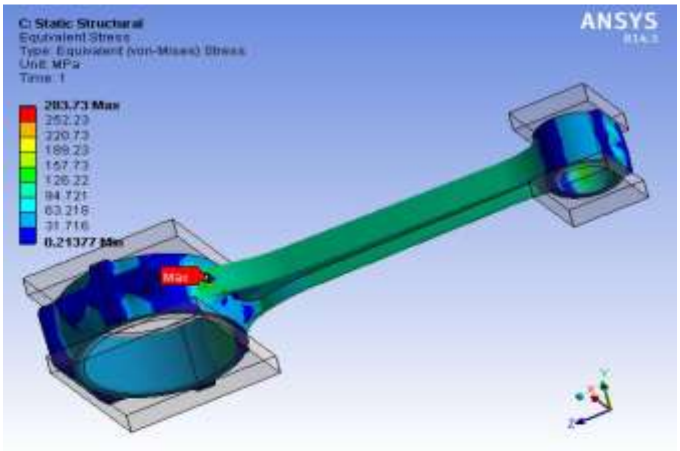

Figure 1 – Stress for Tensile loading

Tensile analysis was done with element size of 3.7mm, the value of force was increased until the stress in connecting rod reach to allowable tensile strength. Total no. of node that were obtained after meshing was 67856 nodes. When the load of 13500 N was applied stress was 283 MPa. Nature of stress along the rod is presented in fig 1. Total deformation at this load was about 0.18391mm. In similar way maximum compressive strength was obtained. Element size of 3.7 is used and analysis was performed. Total of around 75000 nodes were made and used for analysis. Variation of stress for compression is in fig. 2. Stress point can be seen at base of I section where I beam and big end merge. It is appoint of max stress bit at same time it is also the point for stress riser. Thus the stress at that point will be always predicted higher than the

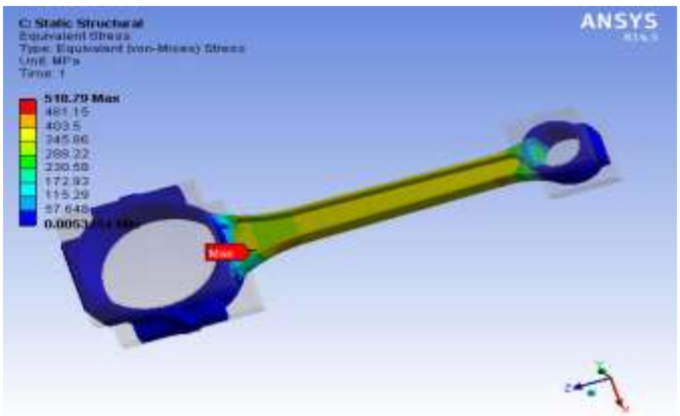

Figure 2 – Stress for Compressive load

Table 1 – Maximum load for AISI 4340

Item	AISI 4340	
	Tetra	Hex
Compressive strength	50000 N	49600 N
Tensile strength	13500 N	13500 N
Buckling load	37220 N 144590 N	37190 N 144600 N

actual value. To find actual stress, variation of stress around the stressed point is studied and it was found that at around 50000 N load component reach the allowable compressive stress. This above table 1 contain two buckling load, this load are related to the load in order to buckle about a particular axis. Second buckling load is the load which will be used for study because it is the load that will buckle I section about axis perpendicular to I section.

VI. REVERSE CLACULATION OF BOUNDERY CONDITION

In this section the calculation will be done in order to find the final boundary condition that is to be sustained by connecting rod. Which will then be used further. The ration of connecting rod length to crank radius is generally kept in between 4 to 5. Let us take this ration as 4 for our problem. Connecting rod length is 200mm which means our crank radius will be 50mm.
It was give that piston diameter is 3.2 times the connecting rod small end eye diameter.
Small end eye diameter is 24 mm
Thus piston diameter is 76.8 mm ≈ 77 mm
Thus we can say stroke is of 90 mm
In Table 1 buckling load is given, which is Euler buckling load where as in designing of connecting rod Rankin buckling load is used. Conversion between both of this is given bellow.

$$\frac{1}{P_{Rankine}} = \frac{1}{P_{Euler}} + \frac{1}{F_c * A}$$

P_{Rankin} = Rankin buckling load
P_{Euler} = Euler buckling load
F_c = Allowable compressive stress
A = Cross section area

$$\frac{1}{P_{Rankine}} = \frac{1}{144600} + \frac{1}{454 * 125}$$

$P_{Rankine}$ = 40496.639 N ≈ 40500 N
This is the maximum compressive load that is to be sustained by the new connecting rod which will be made of Aluminum. From cad model the size of bolt to secure big end is measured, if we study that we can say that if the diameter of bolt is known then the inertial load of reciprocating parts can be found out. Therefore we can write the following equation.

F_I = force on bolt = $\frac{\pi}{4} d^2 * \sigma_t * N$
F_I = Inertia force due to reciprocating part
d = diameter of bolt = 8 mm
σ_t = allowable tensile stress = 125 N/mm^2
N = number of bolts = 2
Using cad model of connecting rod diameter of bolt was measured and it came out to be of 8mm.
$F_I = \frac{\pi}{4} * 8^2 * 125 * 2$
F_I = 12566.37 N ≈ 12570 N
This load of 12570 N is the force that will be applied as a tensile load while performing analysis.

Fatigue calculation

Steel (AISI 4340)

We will estimate S_e (endurance limit) based on ultimate strength using equations as below.

$S_{e'}$ = 0.5*S_{ut} (for S_{ut} < 1400 MPa) (1)

$S_{e'}$ = 700 MPa (for $S_{ut} \geq$ 1400 MPa) (2)

We will use equation (1) since ultimate strength of steel is less than 1400 MPa.

$S_{e'}$ = 0.5*1110 = 555 MPa

Loading is axial so the loading factor is,

C_{load} = 0.7

The part is larger than the test specimen and is not round, so an equivalent diameter based on its 95% stressed area (A_{95}) must be determined and used to find size factor.

A_{95} = 0.1*b*t = 0.1*16*20 = 32 mm^2

$$d_{equiv} = \sqrt{\frac{A_{95}}{0.0766}} = \sqrt{\frac{32}{0.0766}} = 20.43 \text{ mm}$$

Size factor is,

$C_{size} = 1.189*(d_{equiv})^{-0.097}$

$C_{size} = 1.189*(20.43)^{-0.097}$

C_{size} = 0.887

The surface factor is found from chart given in reference [5] Pg 328

C_{surf} = 0.9

Temperature Factor,

C_{Temp} = 1

The reliability factor is taken as 99%

$C_{reliable}$ = 0.814

The corrected endurance limit can now be calculated as bellow

$S_e = C_{load}*C_{size}*C_{surf}*C_{Temp}*C_{reliable}*S_{e'}$

S_e = 0.7*0.887*0.9*1*0.814*555

S_e = 252.45 MPa

To create S-N curve we also S_m

S_m = 0.75*S_{ut}

S_m = 0.75*1110

S_m = 832.5 MPa

Fatigue strength at different Life cycle

S(N) = aN^b

log S(N) = log(a) + b*log(N) (1)

at N = 10^3; S(N) = S_m = 832.5 MPa. Substitute in equation (1)

log 832.5 = log(a) + b*log(10^3) (2)

at N = 10^6; S(N)= S_e = 252.45 MPa. Substitute in equation (1)

log 252.45 = log(a) + b*log(10^6) (3)

Solve equation (2) and (3), we get

a = 2745.2015

b = -0.17273

Final equation for steel is,

Log S(N) = log(2745.2015) – 0.17273*log(N)

S-N equation for Al 7068 is give as

Log N = 9.101-3.498*log (S-19.71)

Table 2 – S-N values for different cycle

Sr. No.	No. of cycle	Steel	Aluminum(MPa)
1	10^3	832.4918	518.410
2	10^4	559.3066	333.9468
3	10^5	375.7681	238.4365
4	10^6	252.4585	188.9859
5	10^7	252.4585	163.3829

VII. AL 7068

FOS of 2.5 was give thus the allowable tensile strength was found out to be as 236 MPa and allowable compressive stress is 378 MPa. By using element size of 3.7 analysis for both tensile and compression were performed. In this maximum load that can be carried without exceeding the allowable range was determined. Following fig.3 shows stress distribution over rod.

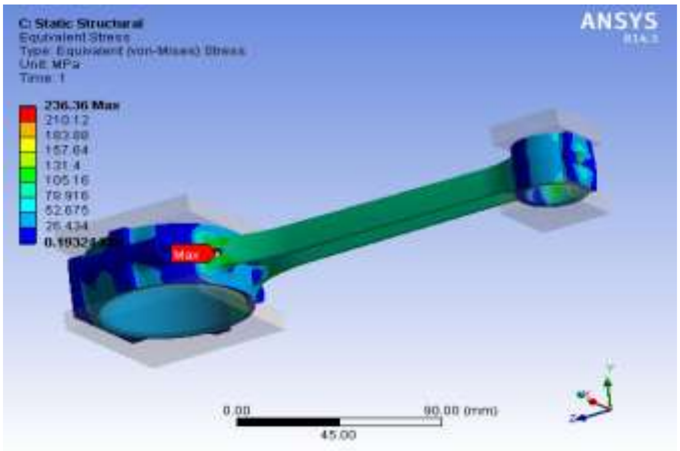

Figure 3 – Stress for Tensile loading.

This distribution will be similar to that of steel rod. We can see that in above fig.3 that the stress value have reached to its allowable value when the load of 12000 N was applied to it. In section VI we have seen that the tensile load that connecting rod should sustain was of 12570 N. this show that connecting rod will fail at oil hole that is present near big end for repeated tensile loading. On same line it was also loaded with compressive loading and stress were studded due to the presence of stress riser point.

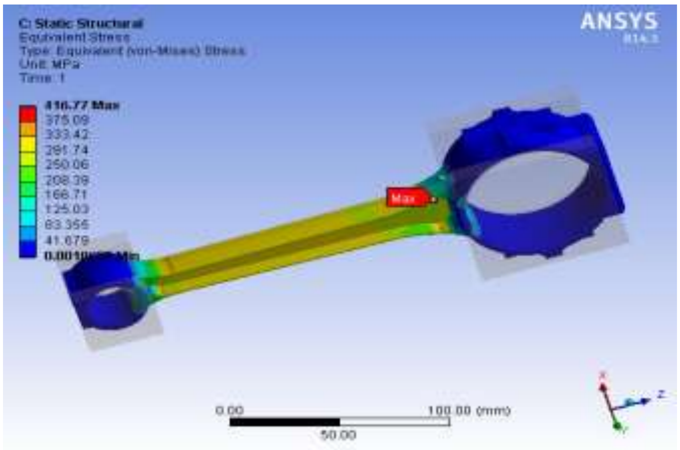

Figure 4 – Stress variation for compressive loading.

Fig.5 is the plot that will be used to identify the correct stress in a stress riser point. When we are encountered with a stress riser point the surrounding element around that point and some points little far from that point which will act like a reference to study that point.

When the stress values from this point are collected they are placed in this graph and an approximate graph is plot leaving those point which are affected by stress riser element. Once we know at what point we need to know stress that point is selected and a vertical line is dropped so that it will intersect the interpreted line. This intersection point will be the point which will give the actual value of stress for given load.

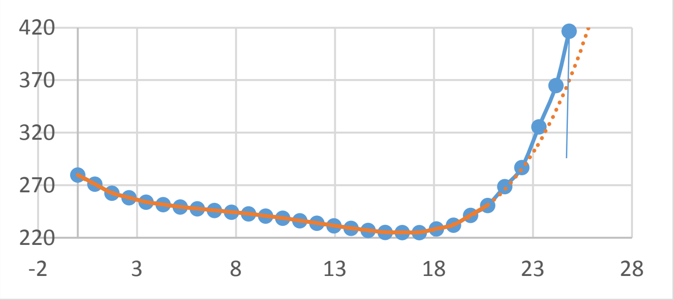

Figure 5 – Graph for stress riser study

At a compressive load of 39700 N stress reach its allowable stress. This little difference in strength of rod and require strength of rod leaves a scope for further work in material. For this we would look for some kind of composite work since we know that composite are light in weight but still have high strength. We will try to apply some concept of this composites in our connecting rod. Final results for analysis are given in table below.

Item	AL 7068	
	Tetra	Hex
Compressive strength	39700 N	39300 N
Tensile strength	12000 N	12100 N
Buckling load	12957 N	12944 N

VIII. Results

When tensile load is applied to small end maximum stress is generated in the area near oil hole which is placed above big end this is clearly visible in fig. 3. Similarly when compressive load is applied stress region are visible in fig. 4. Maximum stress occur at the interface where I section and big end merge with each other. Following table display various other properties of connecting rod with different material.

Item	AISI 4340		AL 7068	
	Tetra	Hex	Tetra	Hex
Compressive strength	50000	49600	39700	39300
Tensile strength	13500	13500	12000	12100
Buckling load	37220	37190	12957	12944
Natural frequency	217.56 448.15 711.45 1727.3 2921.5 4484.1	217.26 447.89 703.38 1725.5 2917.1 4448.0	213.24 438.69 685.73 1691.4 2856.4 4342.3	212.98 438.45 678.15 1689.8 2851.9 4308.5
weight	0.67028 Kg		0.24335 Kg	

IX. Conclusion

Following are the changes that were observed when material change was made for connecting rod.

- 63.69 % reduction in weight.
- 10.37 % of reduction in Tensile strength.
- 65.19 % of reduction in buckling load.
- No much changes were observed in natural frequency.

It is clear from the result that when material is changed from steel to Aluminium strength reduces. But as AL 7068 have higher tensile strength compared to other aluminium alloy we find that the tensile strength is close to Steel. Buckling load is reduced by 65 % this shows that AL 7068 connecting rod can be used in engine by having some slight modification in material like use of titanium inserts at high stress point. Weight reduction was of 63.69% thus in high rpm work environment inertial load will be eliminated this helps in reduction of vibration that is produce and weight of overall engine.

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