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Finite Element Analysis of Connecting Rod to Improve Its Properties

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Abstract

The main objective of this study was to explore weight and cost reduction optimization for a production forged steel connecting rod. This has entailed performing a detailed load analysis. Therefore, this study has dealt with two subjects, first, Tensile &compressive load and stress analysis of the connecting rod, and second, optimization for weight and cost. In the first part of the study, the loads acting on the connecting rod as a function of stress were obtained. The relations for obtaining the loads for the connecting rod at a given constant speed of the crankshaft were also determined. The static FEA, Eigen buckling FEA was studied. Based on the observations of the static FEA and the load analysis results, the load for the optimization study was selected. The component was optimized for weight and cost subject to fatigue life and space constraints and manufacturability. It is the conclusion of this study that the connecting rod can be designed and optimized under a load range comprising tensile load corresponding to 360^o crank angle at the maximum engine speed as one extreme load, and compressive load corresponding to the peak gas pressure as the other extreme load. Furthermore, the existing connecting rod can be replaced with a new connecting rod made of C-70 steel that is 11% lighter and 25% less expensive due to the steel's fracture crack ability. The fracture crack ability feature, facilitates separation of cap from rod without additional machining of the mating surfaces. Yet, the same performance can be expected in terms of component durability.

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. Connecting rods for automotive applications are typically manufactured by forging from either wrought steel or powdered metal. They could also be cast. However, castings could have blow-holes which are detrimental from durability and fatigue points of view. The first aspect was to investigate and compare fatigue strength of steel forged connecting rods with that of the powder forged connecting rods. The second aspect was to optimize the weight and manufacturing cost of the steel forged connecting rod. The first aspect of this research program has been dealt with in a master's thesis entitled "Fatigue Behaviour and Life predictions of Forged Steel and PM Connecting Rods" (Afzal A., 2004). This current thesis deals with the second aspect of the study, the optimization part. Due to its large volume production, it is only logical that optimization of the Connecting rod for its weight or volume will result in large-scale savings. It can also achieve the objective of reducing the weight of the engine component, thus reducing inertia loads, reducing engine weight and improving engine performance and fuel economy. A Connecting rod is the link between the reciprocating piston and rotating crank shaft. Small end of the connecting rod is connected to the piston by means of gudgeon pin. The big end of the connecting rod is connected to the crankshaft.

1.1 Function: The function of the connecting rod is to convert the reciprocating motion of the piston into the rotary motion of the crankshaft.

1.2 Materials: The connecting rods are usually forged out of the open hearth steel or sometimes even nickel steel or vanadium steel. For low to medium capacity high speed engines, these are often made of duraluminium or other aluminium alloys. However, with the progress of technology, the connecting rods these days are also cast from malleable or spheroidal graphite cast iron. The different connecting rod steels are (4OC8, 37Mn6, 35Mn6 MO3, 35Mn6 Mo4, 40Cr4, 40Cr4 Mo3, 40NiCr4MO2) etc. In general, steel forged connecting rods are compact and light weight which is an advantage from inertia view point, whereas cast connecting rods are comparatively cheaper, but on account of lesser strength their use limited to small and medium size petrol engines.

1.3 Construction: A combination of axial and bending stresses act on the rod in operation. The axial stresses are product due to cylinder gas pressure and the inertia force arising on account of reciprocating motion. Whereas bending stresses are caused due to the centrifugal effects. To provide the maximum rigidity with minimum weight, the cross section of the connecting rod is made as and I – section end of the rod is a solid eye or a split eye this end holding the piston pin. The big end works on the crank pin and is always split. In some connecting rods, a hole is drilled between two ends for carrying lubricating oil from the big end to the small end for lubrication of piston and the piston pin.

1.4 Classification: The classification of connecting rod is made by the cross sectional point of view i.e. I – section, H – section, Tabular section, Circular section. In low speed engines, the section of the rod is circular, with flattened sides. In high speed engines either an H – section or Tabular section is used because of their lightness. The rod usually tapers slightly from the big end to the small end.

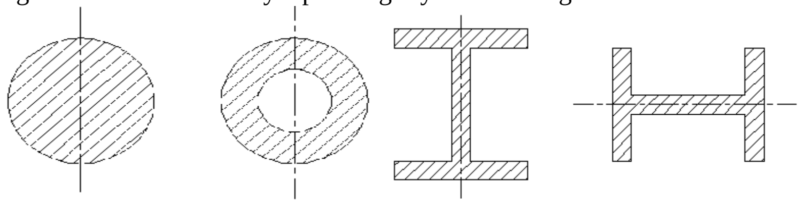

Figure 1.1: Different Cross sections of connecting rod

1.5 Forces acting on the Connecting Rod:

1. The combined effect (or joint effect) of,

a) The pressure on the piston, combined with the inertia of the reciprocating parts.

b) The friction of the piston rings, piston, piston rod and the cross head.

2. The longitudinal component of the inertia of the rod.

3. The transverse component of the inertia of the rod.

4. The friction of the two end bearings.

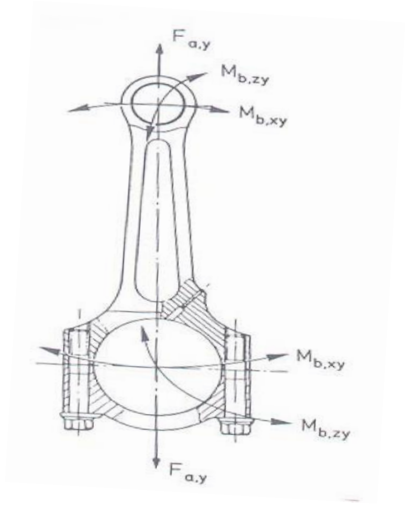

Figure 1.2: The origin of stresses on a connecting rod (Sonsino, 1996)

II. Literature Review

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. Therefore, durability of this component is of critical importance. Due to these factors, the connecting rod has been the topic of research for different aspects such as production technology, materials, performance simulation, fatigue, etc. For the current study, it was necessary to investigate finite element modelling techniques, optimization techniques, developments in production technology, new materials, fatigue modelling, and manufacturing cost analysis. This brief literature survey reviews some of these aspects. Pravardhan S. Shenoy May 2004 made research on 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. This has entailed performing a detailed load analysis. Therefore, this study has dealt with two subjects, first, dynamic load and quasi-dynamic stress analysis of the connecting rod, and second, optimization for weight and cost. In the first part of the study, the loads acting on the connecting rod as a function of time were obtained. The relations for obtaining the loads and accelerations for the connecting rod at a given constant speed of the crankshaft were also determined. Quasidynamic finite element analysis was performed at several crank angles. The stress-time history for a few locations was obtained. The difference between the static FEA, quasidynamic FEA was studied. Based on the observations of the quasi-dynamic FEA, static FEA and the load analysis results, the load for the optimization study was selected. The results were also used to determine the variation of R-ratio, degree of stress multiaxiality, and the fatigue model to be used for analyzing the fatigue strength. The component was optimized for weight and cost subject to fatigue life and space constraints and manufacturability. It is the conclusion of this study that the connecting rod can be designed and optimized under a load range comprising tensile load corresponding to 360o crank angle at the maximum engine speed as one extreme load, and compressive load corresponding to the peak gas pressure as the other extreme load.

III. FE Modelling Of the Connecting Rod

3.1 Geometry of the connecting rod

The connecting rod is developed by using a Catia R-20 software. A solid model of the connecting rod, as shown in Figure 6.1 For FEA, the flash along the entire connecting rod length including the one at the oil hole was eliminated in order to reduce the model size. The flash runs along the length of the connecting rod and hence does not cause stress concentration under axial loading. The flash is a maximum of about 0.15 mm thick. Even under bending load the flash can be eliminated especially when we consider the fact that the solution time will increase drastically if we do model this feature, and very little increase in strength can be expected. This is due to the fact that the flash being 0.15 mm thick will drastically increase the model size, if it is modeled. The connecting rod geometry used for FEA can be seen in Figure 6.2. Note that the flash and the bolt-holes have been eliminated. The cross section of the connecting rod from failed components reveals that the connecting rod, as manufactured, is not perfectly symmetric. In the case of one connecting rod, the degree of non-symmetry in the shank region, when comparing the areas on either side of the axis of symmetry perpendicular to the connecting rod length and along the web, was about 5%. This non-symmetry is not the design intent and is produced as a manufacturing variation. Therefore, the connecting rod has been modeled as a symmetric component. The connecting rod weight as measured on a weighing scale is 765.9 grams. The difference in weight between the weight of the solid model used for FEA and the actual component when corrected for bolt head weight is less than 1%. This is an indication of the accuracy of the solid model.

3.2 Mesh generation- Static FEA

Finite element mesh was generated using parabolic tetrahedral elements with various element lengths of 1 mm (20917 elements), 2 mm (35373 elements), 1.5 mm (77412 elements). For most areas on the connecting rod convergence has been achieved with 1.5 mm uniform element length. Therefore, a finite element mesh was generated with a uniform global element length of 1.5 mm, and at locations with chamfers a local element length of 1 mm was used. This resulted in a mesh with 104471 elements. It can be seen that convergence has been achieved with 1 mm local mesh size. The maximum percentage difference between the stress values observed between the last two models (the one with 104471 elements and the one with 128954 elements) is 2.3%, which is small. Hence, the mesh with 104471 elements was used for FEA.

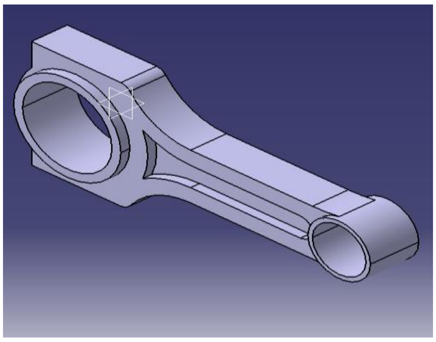

Figure 3.1:-A solid model of the existing connecting rod

3.3 Boundary conditions

The crank and piston pin ends are assumed to have a sinusoidal distributed loading over the contact surface area, under tensile loading, as shown in the Figure 3.2 & 3.5 This is based on experimental results (Webster *et al.* 1983). The normal pressure on the contact surface is given by: $p = p_o \cos\Theta$

The load is distributed over an angle of 180°. The total resultant load is given by:

$$P_t = \int_{-\pi/2}^{\pi/2} p_o (\cos^2\Theta)\, r\, t\, d\Theta = p_o\, r\, t\, \pi / 2$$

The normal pressure constant p_o is, therefore, given by: $p_o = P_t / (r\, t\, \pi / 2)$

The tensile load acting on the connecting rod, P_t, can be obtained using the expression from the force analysis of the slider crank mechanism.

For compressive loading of the connecting rod, the crank and the piston pin ends are assumed to have a uniformly distributed loading through 120° contact surface, as shown in Figure 3.3 & 3.4 (Webster *et al.* 1983). The normal pressure is given by: $p = p_o$

The total resultant load is given by:

$$P_c = \int_{-\pi/3}^{\pi/3} p_o (\cos\Theta)\, r\, t\, d\Theta = p_o\, r\, t\, \sqrt{3}$$

The normal pressure constant is then given by:

$$p_o = P_c / (r\, t\, \sqrt{3})$$

P_c can be obtained from the indicator diagram of an engine.

In this study four finite element models were analyzed. FEA for both tensile and compressive loads were conducted. Two cases were analyzed for each case, one with load applied at the crank end and restrained at the piston pin end, and the other with load applied at the piston pin end and restrained at the crank end. In the analysis carried out, the axial load was 27.8 kN in both tension and compression. The pressure constants for 27.8 kN are as follows:

Compressive Loading:
Crank End: p_o = 27800
Piston pin End: p_o = 27800
Tensile Loading:
Crank End: p_o = 27800
Piston pin End: p_o = 27800

Since the analysis is linear elastic, for static analysis the stress, displacement and strain are proportional to the magnitude of the load. Therefore, the obtained results from FEA readily apply to other elastic load cases by using proportional scaling factor.

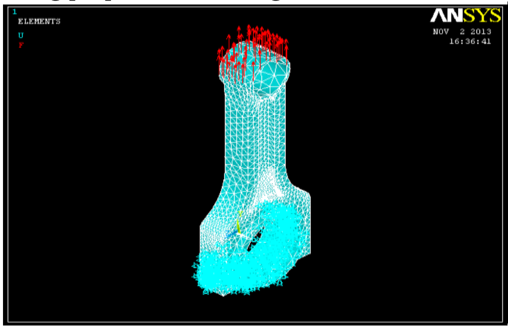

Fig 3.2:-Tensile load on piston pin end of con rod

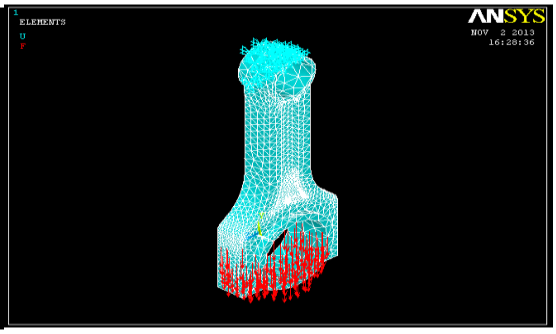

Fig 3.3:-Tensile load on crank end of con rod.

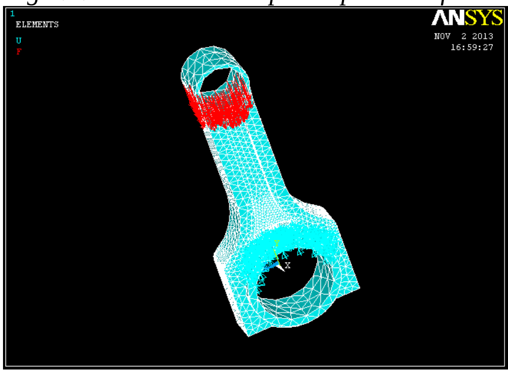

Fig 3.4:-Compressive load on piston pin end of con rod.

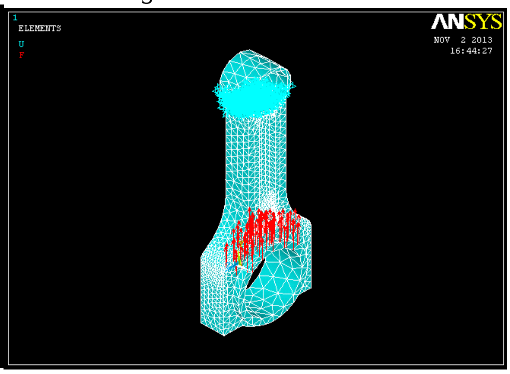

Fig 3.5:-Compressive load on crank end of con rod.

IV. Results of Finite Element Stress Analysis

4.1 Static axial stress analysis

Figures 4.3 show the von Mises stress distribution of the connecting rod under static axial loading. Figure 4.1 & 4.2 shows the von Mises stress distribution with tensile load at the crank end, while the piston pin end is restrained. Figure 4.4 shows Stress concentration at pin end shank portion of a connecting rod due to tensile load at crank end. Figure 4.4 shows the von Mises stress distribution with tensile load at the piston pin end, while crank end is restrained with Front Buckling of connecting rod. The differences between the four FEA models are now discussed. In order to do so, we observe & analyse the result for all four FEA models. In the first case Tensile load is applied at crank end while piston pin end is restrained, it shows front buckling of connecting rod and also shows Stress concentration at pin end shank portion of a connecting rod in the slot of shank. In the second FEA model Tensile load is applied at piston pin end while crank end restrained, it shows front buckling of connecting rod and also shows Maximum Stress at pin end of connecting rod. In the third case compressive load is applied at crank end while piston pin end is restrained, it shows front buckling of connecting rod near the piston pin end. In the fourth FEA model compressive load is applied at piston pin end while crank end restrained, it shows rear buckling of shank portion of connecting rod near piston pin end. Figure 4.1 shows the von Mises stress at a few discrete locations at the mid plane along the length of the connecting rod. This plot gives a general idea of the stress variation along the length of the connecting rod. The static loads for which these stresses are plotted, are a tensile load of 27.8 kN (load at the crank end at 360^o crank angle and at 5700 rev/min). The crank end region in Figure7.1 especially the region near the bolt holes, shows very low stresses. The highest von Mises stress in the region is about 525 MPa and 6177.78mm maximum deformation. However, it should be noted that the bolt hole and the bolt pre-tension are not included in the finite element model. If this region is to be optimized, the bolt hole and the bolt pre-tension should be modeled and considered during the optimization. Figure 4.1 indicates that the stresses at the small end transition are in the neighborhood of 525

MPa. The stresses at nodes in this web region, baring node 247, are below 53 MPa. The oil hole is a region that experiences very high local stresses in tension. FEA results indicate locations with local stresses in excess of the yield strength (700 MPa).

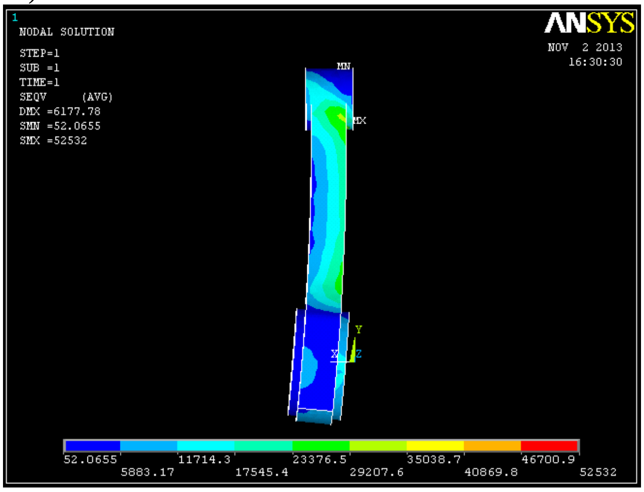

Fig 4.1:- Rear buckling of con rod due to tensile load at crank end

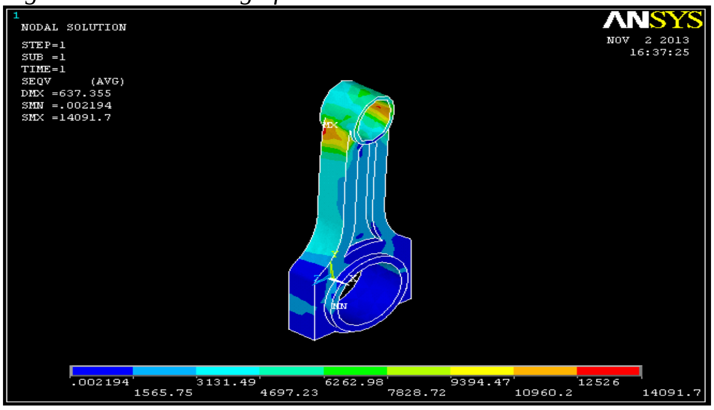

Fig 4.2:- Front Buckling of con rod due to Tensile load at pin end

However, it should be noted that the stresses at the oil hole may not be accurate. This is because the oil hole is very close to the boundary condition (loading). Moreover, during fatigue testing of the connecting rod, no failures were observed in the oil hole region (Afzal, 2004).

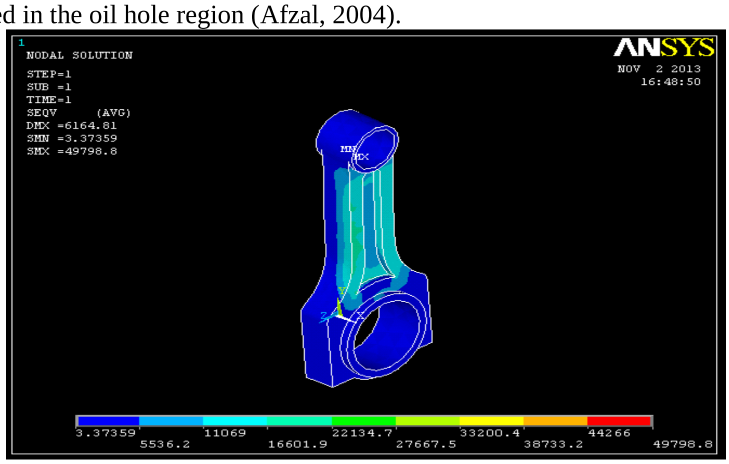

Figure 4.3:- Von Mises Stress distribution of connecting rod due to Compressive load at crank end

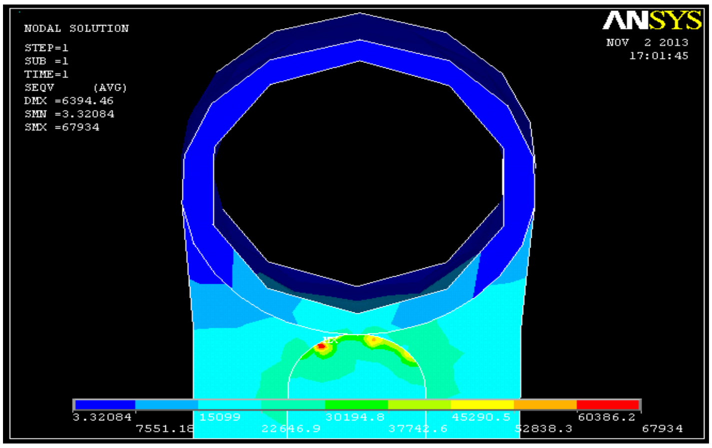

Figure 4.4:- Stress Concentration along shank portion rod due to Compressive load at pin end

V. Conclusions

This research project investigated weight and cost reduction opportunities that steel forged connecting rods offer. The connecting rod chosen for this project belonged to a mid-size sport utility vehicle. First, the connecting rod was modelled. Load analysis was performed, using analytical techniques and computer-based mechanism simulation tools (CATIA R20 and ANSYS 14.0). Furthermore the optimization of connecting rod is performed with manufacturing & economic aspects which offer some geometrical changes in connecting rod design. The following conclusions can be drawn from the optimization part of the study:

1) The connecting rod was optimized under a load range comprising the dynamic load at 360^{o} crank angle at maximum engine speed and the maximum gas load. This connecting rod satisfied all the constraints defined and was found to be satisfactory at other crank angles also.

2) At locations like the cap-rod outer edge, the extreme end of the cap, and the surface of the piston pin end bore, the stresses were observed to be significantly lower when compared to stresses predicted by cosine loading (tensile load).

3) Fatigue strength was the most significant factor (design driving factor) in the optimization of this connecting rod. It improves the properties of connecting rod yield strength, ultimate tensile stress and buckling factor.

4) The optimized geometry is 11% lighter and cost analysis indicated it would be 25% less expensive than the current connecting rod, in spite of lower strength of C-70 steel compared to the existing forged steel. PM connecting rods can be replaced by fracture splitable steel forged connecting rods with an expected cost reduction of about 15% or higher, with similar or better fatigue behaviour.

5) By using other facture crackable materials such as micro-alloyed steels having higher yield strength and endurance limit, the weight at the piston pin end and the crank end can be further reduced. Weight reduction in the shank region is, however, limited by manufacturing constraints.

VI. References

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