Analysis and Optimization of Connecting Rod used in Heavy Commercial Vehicles

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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 report describes designing and Analysis of connecting rod as well as finding alternate material for connecting rod. Currently existing connecting rod is manufactured by using Carbon steel. In this drawing is drafted from the calculations. A parametric model of Connecting rod is modelled using CATIA V5 R21 software and to that model, analysis is carried out by using ANSYS 18.1 Software. Finite element analysis of connecting rod is done by considering the materials, viz. SAE 4340, 42CrMo4 & Al 7075-T651. The best combination of parameters like Von misses Stress and strain, Deformation, Factor of safety and weight reduction for heavy duty vehicle piston were done in ANSYS software. The present work has been established to replace the existing connecting rod made of forged steel with the aluminium MMC connecting rod for weight optimization.

Keywords: Analysis, Connecting rod, FEA, Dynamic, Optimization, etc.

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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 blowholes which are detrimental from durability and fatigue points of view. The fact that forgings produce blow-hole- free and better rods gives them an advantage over cast rods (Gupta, 1993). Between the forging processes, powder forged or drop forged, each process has its own pros and cons. Powder metal manufactured blanks have the advantage of being near net shape, reducing material waste. However, the cost of the blank is high due to the high material cost and sophisticated manufacturing techniques (Repgen, 1998). With steel forging, the material is inexpensive and the rough part manufacturing process is cost effective. Bringing the part to final dimensions under tight tolerance results in high expenditure for machining, as the blank usually contains more excess material (Repgen, 1998). A sizeable portion of the US market for connecting rods is currently consumed by the powder metal forging industry. A comparison of the European and North American connecting rod markets indicates that according to an unpublished market analysis for the year 2000 (Ludenbach, 2002), 78% of the connecting rods in Europe (total annual production: 80 million approximately) are steel forged as opposed to 43% in North America (total annual production: 100 million approximately. In order to recapture the US market, the steel industry has focused on development of production technology and new steels. AISI (American Iron and Steel Institute) funded a research program that had two aspects to address. 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.

2. LITERATURE REVIEW

The connecting rod is subjected to a complex state of loading. It undergoes high cyclic loads of the order of 1089 to 10 cycles, which range from high compressive loads 3m 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.

G.M Sayeed Ahmed (2014), 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 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 present work has been undertaken to replace the existing connecting rod made of forged steel which is broken for LML Freedom with the aluminium connecting rod.

Leela Krishna Vegi (2013), 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 thesis describes designing and Analysis of connecting rod. Currently existing connecting rod is manufactured by using Carbon steel. In this drawing is drafted from the calculations. A parametric model of Connecting rod is modeled using CATIA V5 R19 software and to that model, analysis is carried out by using ANSYS 13.0 Software. Finite element analysis of connecting rod is done by considering the materials, viz... Forged steel. 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.Forged steel has more factor of safety, reduce the weight, increase the stiffness and reduce the stress and stiffer than other material like carbon steel. With Fatigue analysis we can determine the lifetime of the connecting rod.

B. Anusha (2013),Connecting rod is a major link inside of an internal combustion engine. Its primary function is to transmit the push and pull from the piston pin to the crank pin thus converting the reciprocating motion of piston in to rotary motion of the crank. In the present investigation a 4-stroke petrol engine of a specified model, market available connecting rod is selected for the investigation. For present investigation the designed connecting rod is modeled using solid modelling software i.e. PRO/E.The modeled connecting rod imported to the analysis software i.e. ANSYS. Static analysis is done to determine von-misses stresses, strain, shear stress and total deformation for the given loading conditions using analysis software i.e. 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

Somnath Chattopadhyay (2010), this activity centres on the courses of strength of materials and production design offered at a sophomore level Mechanical Engineering curriculum. A connecting rod is one of the most mechanically stressed components in internal combustion engines. The objective of the activity is to select the appropriate material for a connecting rod where the constraints are to make the product as light and cheap as possible and yet strong enough to carry the peak load without failure in high cycle fatigue. The fracture toughness also needs to be above a certain minimum value. A further requirement is that the connecting rod should not buckle during operation. These constraints are used to select an appropriate cross section and material for construction. The next phase involves the selection of manufacturing process for which the constraints are shape, mass, quality and economics. The selections of the material, the cross-sectional shape and the manufacturing processes involve the use of CES EduPack, which yields materials that meet the constraints. The current manufacturing processes for connecting rods by fracture split drop forging and fracture split powder forging are highlighted and compared based on current information.

Webster *et al.* (1983) performed three dimensional finite element analysis of a high-speed diesel engine connecting rod. For this analysis they used the maximum compressive load which was measured experimentally, and the maximum tensile load which is essentially the inertia load of the piston assembly mass. The load distributions on the piston pin end and crank end were determined experimentally. They modelled the connecting rod cap separately, and also modelled the bolt pretension using beam elements and multi point constraint equations.

In a study reported by Repgen (1998), based on fatigue tests carried out onidentical components made of powder metal and C-70 steel (fracture splitting steel), he notes that the fatigue strength of the forged steel part is 21% higher than that of the powder metal component. He also notes that using the fracture splitting technology results in a 25% cost reduction over the conventional steel forging process. These factors suggest that a fracture splitting material would be the material of choice for steel forged connecting rods. He also mentions two other steels that are being tested, a modified micro-alloyed steel and a modified carbon steel. Other issues discussed by Repgen are the necessity to avoid jig spots along the parting line of the rod and the cap, need of consistency in the chemical composition and manufacturing process to reduce variance in microstructure and production of near net shape rough part.

Park *et al.* (2003) investigated microstructural behaviour at various forging conditions and recommend fast cooling for finer grain size and lower network ferrite content. From their research they concluded that laser notching exhibited best fracture splitting results, when compared with broached and wire cut notches. They optimized the fracture splitting parameters such as, applied hydraulic pressure, jig set up and geometry of cracking cylinder based on delay time, difference in cracking forces and roundness. They compared fracture splitting high carbon micro-alloyed steel (0.7% C) with carbon steel (0.48% C) using rotary bending fatigue test and concluded that the former has the same or better fatigue strength than the later. From a comparison of the fracture splitting high carbon micro-alloyed steel and powder metal, based on tension-compression fatigue test they noticed that fatigue strength of the former is 18% higher than the later.

Sarihan and Song (1990), for the optimization of the wrist pin end, used a fatigue load cycle consisting of compressive gas load corresponding to maximum torque and tensile load corresponding to maximum inertia load. Evidently, they used the maximum loads in the whole operating range of the engine. To design for fatigue, modified Goodman equation with alternating octahedral shear stress and mean octahedral shear stress was used. For optimization, they generated an approximate design surface, and performed optimization of this design surface. The objective and constraint functions were updated to obtain precise values. This process was repeated till convergence was achieved. They also included constraints to avoid fretting fatigue. The mean and the alternating components of the stress were calculated using maximum and minimum values of octahedral shear stress. Their exercise reduced the connecting rod weight by nearly 27%. The initial and final connecting rod wrist pin end designs are shown in Figure 1.2.

Yoo *et al.* (1984) used variational equations of elasticity, material derivative idea of continuum mechanics and adjoin variable technique to calculate shape design sensitivities of stress. The results were used in an iterative optimization algorithm, steepest descent algorithm, to numerically solve an optimal design problem. The focus was on shape design sensitivity analysis with application to the example of a connecting rod. The stress constraints were imposed on principal stresses of inertia and firing loads. But fatigue strength was not addressed. The other constraint was the one on thickness to bind it away from zero. They could obtain 20% weight reduction in the neck region of the connecting rod.

Hippoliti (1993) reported design methodology in use at Piaggio for connecting rod design, which incorporates an optimization session. However, neither the details of optimization nor the load under which optimization was performed were discussed. Two parametric FE procedures using 2D plane stress and 3D approach developed by the author were compared with experimental results and shown to have good agreements. The optimization procedure they developed was based on the 2D approach.

El-Sayed and Lund (1990) presented a method to consider fatigue life as a constraint in optimal design of structures. They also demonstrated the concept on a SAE key whole specimen. In this approach a routine calculates the life and in addition to the stress limit, limits are imposed on the life of the component as calculated using FEA results.

Pai (1996) presented an approach to optimize shape of connecting rod subjected to a load cycle, consisting 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, with fatigue life constraint. Fatigue life defined as the sum of the crack initiation and crack growth lives, was obtained using fracture mechanics principles. The approach used finite element routine to first calculate the displacements and stresses in the rod; these were then used in a separate routine to calculate the total life. The stresses and the life were used in an optimization routine to evaluate the objective function and constraints. The new search direction was determined using finite difference approximation with design sensitivity analysis. The author was able to reduce the weight by 28%, when compared with the original component.

Sonsino and Esper (1994) have discussed the fatigue design of sintered connecting rods. They did not perform optimization of the connecting rod. They designed a connecting rod with a load amplitude Fa = 19.2 kN and with different regions being designed for different load ratios (R), such as, in the stem Fm = -2.2 kN and R = -1.26, at the piston pin end Fm = -5.5 kN and R = -1.82, at the crank end Fm = 7.8 kN and R = -0.42. They performed preliminary FEA followed by production of a prototype. Fatigue tests and experimental stress analysis were performed on this prototype based on the results of which they proposed a final shape. In order to verify that the design was sufficient for fatigue, they computed the allowable stress amplitude at critical locations, taking the R-ratio, the stress concentration, and statistical safety factors into account, and ensured that maximum stress amplitudes were below the allowable stress amplitude.

Serag *et al.* (1989) developed approximate mathematical formulae to define connecting rod weight and cost as objective functions and also the constraints. The optimization was achieved using a Geometric Programming technique. Constraints were imposed on the compression stress, the bearing pressure at the crank and the piston pin ends. Fatigue was not addressed. The cost function was expressed in some exponential form with the geometric parameters.

Folgar *et al.* (1987) developed a fibre FP/Metal matrix composite connecting rod with the aid of FEA, and loads obtained from kinematic analysis. Fatigue was not addressed at the design stage. However, prototypes were fatigue tested. The investigators identified design loads in terms of maximum engine speed, and loads at the crank and piston pin ends. They performed static tests in which the crank end and the piston pin end failed at different loads. Clearly, the two ends were designed to withstand different loads.

Balasubramaniam *et al.* (1991) reported computational strategy used in Mercedes-Benz using examples of engine components. In their opinion, 2D FE models can be used to obtain rapid trend statements, and 3D FE models for more accurate investigation. The various individual loads acting on the connecting rod were used for performing simulation and actual stress distribution was obtained by superposition. The loads included inertia load, firing load, the press fit of the bearing shell, and the bolt forces. No discussions on the optimization or fatigue, in particular, were presented.

Ishida *et al.* (1995) measured the stress variation at the column centre and column bottom of the connecting rod, as well as the bending stress at the column centre. It was also observed that the R ratio varies with location, and at a given location it also varies with the engine speed. The maximum bending stress magnitude over the entire cycle (0 to 720 degree crank angle) at 12000 rev/min, at the column centre was found to be about 25% of the peak tensile stress over the same cycle.

Athavale and Sajanpawar (1991) modelled the inertia load in their finite element model. An interface software was developed to apply the acceleration load to elements on the connecting rod depending upon their location, since acceleration varies in magnitude and direction with location on the connecting rod. They fixed the ends of the connecting rod, to determine the deflection and stresses. This, however, may not be representative of the pin joints that exist in the connecting rod. The results of the detailed analysis were not discussed, rather, only the modelling technique was discussed. The connecting rod was separately analysed for the tensile load due to the piston assembly mass (piston inertia), and for the compressive load due to the gas pressure. The effect of inertia load due to the connecting rod, mentioned above, was analysed separately.

While investigating a connecting rod failure that led to a disastrous failure of an engine, Rabb (1996) performed a detailed FEA of the connecting rod. He modelled the threads of the connecting rod, the threads of connecting rod screws, the prestress in the screws, the diametral interference between the bearing sleeve and the crank end of the connecting rod, the diametral clearance between the crank and the crank bearing, the inertia load acting on the connecting rod, and the combustion pressure. The analysis clearly indicated the failure location at the thread root of the connecting rod, caused by improper screw thread profile. The connecting rod failed at the location indicated by the FEA. An axisymmetric model was initially used to obtain the stress concentration factors at the thread root. These were used to obtain nominal mean and alternating stresses in the screw. A detailed FEA including all the factors mentioned above was performed by also including a plasticity model and strain hardening. Based on the comparison of the mean stress and stress amplitude at the threads obtained from this analysis with the endurance limits obtained from specimen fatigue tests, the adequacy of a new design was checked.

Load cycling was also used in inelastic FEA to obtain steady state situation. In a published SAE case study (1997), a replacement connecting rod with 14% weight savings was designed by removing material from areas that showed high factor of safety. Factor of safety with respect to fatigue strength was obtained by performing FEA with applied loads including bolt tightening load, piston pin interference load, compressive gas load and tensile inertia load. The study lays down certain guidelines regarding the use of the fatigue limit of the material and its reduction by a certain factor to account for the as-forged surface. The study also indicates that buckling and bending stiffness are important design factors that must be taken into account during the design process. On the basis of the stress and strain measurements performed on the connecting rod, close agreement was found with loads predicted by inertia theory. The study also concludes that stresses due to bending loads are substantial and should always be taken into account during any design exercise.

3. OBJECTIVES AND OUTLINE

The objective of this work was to optimize the forged steel connecting rod for its weight and cost. The optimized forged steel connecting rod is intended to be a more attractive option for auto manufacturers to consider, as compared with its powder- forged counterpart.

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. Commercial software's such as Catia V5 and Ansys can be used to obtain the variation of quantities such as angular velocity, angular acceleration, and load. However, usually the worst case load is considered in the design process. Literature review suggests that investigators use maximum inertia load, inertia load, or inertia load of the piston assembly mass as one extreme load corresponding to the tensile load, and firing load or compressive gas load corresponding to maximum torque as the other extreme design load corresponding to the compressive load. Inertia load is a time varying quantity and can refer to the inertia load of the piston, or of the connecting rod. In most cases, in the literature the investigators have not clarified the definition of inertia load - whether it means only the inertia of the piston, or whether it includes the inertia of the connecting rod as well. Questions are naturally raised in light of such complex structural behaviour, such as: Does the peak load at the ends of the connecting rod represent the worst case loading? Under the effect of bending and axial loads, can one expect higher stresses than that experienced under axial load alone? Moreover, very little information is available in the literature on the bending stiffness requirements, or on the magnitude of bending stress. From the study of Ishida *et al.* (1995) reviewed in Section 1.2, it is clear that the maximum stress at the connecting rod column bottom does not occur at the TDC, and the maximum bending stress at the column centre is about 25% of the maximum stress at that location. However, to obtain the bending stress variation over the connecting rod length, or to know the stress at critical locations such as the transition regions of the connecting rod, a detailed analysis is needed. As a result, for the forged steel connecting rod investigated, a detailed load analysis under service operating conditions was performed, followed by a dynamic
FEA to capture the stress variation over the cycle of operation. Logically, any optimization should be preceded by stress analysis of the existing component, which should be performed at the correct operating loads. Optimization was performed to reduce the mass and manufacturing cost of the connecting rod, subject to fatigue life and yielding constraints. The material was changed to 42CrMo4 and Aluminium. A comparison between the various manufacturing processes and their costs is also presented.

PARAMETERS CONSIDER:

- Vehicle Model: Tata LPT 407 EX
- Engine: 2956 cc Diesel 4 in line cylinder (water cooled)
- Torque: 300Nm
- Power: 97.2 BHP
- Max Speed: 100 KMPH
- Pressure Calculation
- Density of Diesel: 832 *10 -9 Kg/mm3
- Operating Temperature: T = 210 C
- Volume per cylinder: 2956/4 = 739 cc =739 *103 mm3
- Mass = Density * volume = 832 *10 -9 * 739 *103 = 61.48 * 10-2 kg
- Molecular Weight of Diesel: 230 g/mole = 230 * 10-3 kg/mole
- Gas constant of diesel = R =8314.3 / 230 * 10-3 = 36.15 * 103 J/Kg.mole.K

4. DESIGN OF CONNECTING ROD

A connecting rod is a machine member which issubjected to alternating direct compressive andtensile forces. Since the compressive forces are much higher than the tensile force, therefore the cross-section of the connecting rod isdesignedas a strutandthe Rankine formula is used. A connecting rodsubjected to an axial load W may buckle with x-axis as neutral axis in the plane of motion of theconnecting rod, {or} y-axis is a neutral axis. The connecting rod is considered like both ends hingedfor buckling about x-axis and both ends fixed forbuckling about y-axis. A connecting rod should beequally strong in buckling about either axis.

According to Rankine formulae

$$W_{cr} \text{ about x axis} = \frac{[\,\sigma_c * A\,]}{1+a\left(\frac{L}{K_{xx}}\right)2} = \frac{[\,\sigma_c * A\,]}{1+a\left(\frac{l}{K_{xx}}\right)2}$$

$$[\therefore for both\ ends\ hinged L = l]$$

$$W_{cr} \text{ about x axis} = \frac{[\,\sigma_c * A\,]}{1 + a\left(\frac{L}{K_{yy}}\right)2} = \frac{[\,\sigma_c * A\,]}{1 + a\left(\frac{l}{2\,K_{yy}}\right)2}$$

$$[\therefore for both\ ends fixed L = l/2]$$

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

$$\frac{[\,\sigma_c * A\,]}{1 + a\left(\frac{l}{K_{xx}}\right)2} = \frac{[\,\sigma_c * A\,]}{1 + a\left(\frac{l}{2\,K_{yy}}\right)2}$$

Or

$$\left(\frac{l}{K_{xx}}\right)^2 = \left(\frac{l}{2\,K_{yy}}\right)^2$$

$$K^2_{xx} = 4.\ K^2_{yy}$$
$$I_{xx} = 4.\ I_{yy}\ [\ I = A*K^2\ ]$$

This shows that the connecting rod is four times strong in buckling about y-axis than about-axis. If I xx > 4Iyy, then buckling will occur about y-axis and if I xx < 4Iyy, then buckling will occur about x-axis. In Actual practice I xx is kept slightly less than 4Iyy. 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 x t] + 3t x t =$11t^2$

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

Moment of inertia about x-axis Ixx = $\frac{1}{12}[4t\ \{5t\}^3 - 3t\ \{3t\}^3] = \frac{419}{12}[t^4]$

And moment of inertia about y-axis

Iyy $= \frac{2*1}{12} \times t \times \{4t\}^3 + \frac{1}{12}\{3t\}t^3 = \frac{419}{12}[t^4]$

Ixx/I yy = [419/12]x [12/131] =3.2

Since the value of Ixx/Iyy lies between 3 and 3.5 m therefore I-section chosen is quite satisfactory.

4.1 PRESSURE CALCULATION

- Density of Diesel: 832 $*10^{-9}$ Kg/mm3
- Operating Temperature: T = 210 C
- Volume per cylinder: 2956/4 = 739 cc =739 $*10^3$ mm3
- Mass = Density * volume = 832 $*10^{-9}$ * 739 $*10^3$ = 61.48 * 10^{-2} kg
- Molecular Weight of Diesel: 230 g/mole = 230 * 10^{-3} kg/mole
- Gas constant of diesel = R =8314.3 / 230 * 10^{-3} = 36.15 * 10^3 J/Kg.mole.K
  Gas law equation

  PV=mRT

  P=mRT/V

  P = (61.48 * 10^{-2} * 36.15 * 10^3 * 210) / 739 * 10^3

  P = 6.3 MPa

  Gas Force = P * (π/4) * D^2

  F = 6.3 * (π/4) * 86^2

  F = 36595.5 N

4.2 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:

$$Ixx = \frac{1}{12}[4t\ \{5t\}^3 - 3t\ \{3t\}^3] = \frac{419}{12}[t^4]$$

MI of section about y axis:

$$Iyy = \frac{2*1}{12} \times t \times \{4t\}^3 + \frac{1}{12}\{3t\}t^3 = \frac{419}{12}[t^4]$$

I xx/I yy = [419/12]x [12/131] =3.2

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

L = 196 mm

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

Buckling load W_B = 36595.5 * 2

$$W_B = \frac{[\ \sigma_c * A\ ]}{1 + a\left(\frac{L}{K_{xx}}\right)2}$$

σ_c = compressive yield stress = **760 MPa**

$$Kxx = \frac{Ixx}{A}$$

Kxx = 1.78t

a = **0.0002**

Where, A = Cross-section area of connecting rod,
L = Length of the connecting rod
WB = Buckling load
σc = Compressive yield stress,
a = Constant
K_{xx} = Radius of gyration of section about x – x

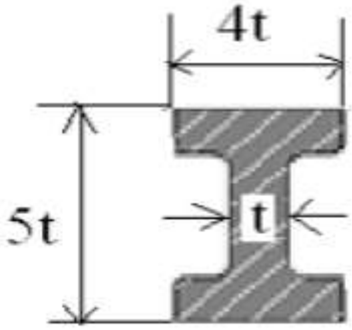
By substituting σc, A, a, L, Kxx on WB then

$$73191 = \frac{[\ 760 * \mathbf{11t^2}\ ]}{1 + 0.0002\left(\frac{196}{1.78t}\right)2}$$

8360 t^4 -73191 t^4 -177488.175 = 0

By solving above equation
We get
t = 3.276 mm = 4 mm

Design Calculations for Existing Connecting Rod



The standard dimension of I - SECTION.

- Width of section B = 4t
  B = 4*4
  B = 16 mm
- Height of section H = 5t
  H = 5*4
  H = 20 mm

- Height at the big end (crank end) = H_2
  H_2= 1.1H to 1.25H
  H_2= 1.2 * 20
  H_2= 24 mm

- Height at the small end (piston end) = 0.9H to 0.75H
  H_1 = 0.9*20
  H_1 = 18 mm

- Stroke length (l) = 98 mm
- Diameter of piston (D) = 86 mm
- Radius of crank(r) =stroke length/2
  = 98/2

= 49 mm

- Maximum angular speed Wmax = $\frac{2\pi N}{60}$ = = $\frac{2\pi * 6200}{60}$ = 648.93rad/sec

- Ratio of the length of connecting rod to the radius of crank
  N = $\frac{l}{r}$ = $\frac{198}{49}$ = 4.04

- Maximum Inertia force of reciprocating parts

  Fim = Mr $(\text{Wmax})^2$ r (cosθ + COS2θ/n) (Or)
  Fim = Mr $(\text{Wmax})^2$ r (1+1/n)
  Fim = 61.48 * 10^{-2} * $(648.93)^2$ * 0.049 (1+(1/4.04))
  Fim = 15857.53

- Inner diameter of the small end d_1 = $\frac{\text{F im}}{P_{b1} \times l_1}$

$$d_1 = \frac{15857.53}{12.5 \times 1.5 d_1}$$

d_1 =29.08 mm =29.50 mm

Where,
Design bearing pressure for small end Pb_1=12.5 to 15.4N/mm^2
Length of the piston pin l_1= (1.5to 2) d_1

- Outer diameter of the small end = $d_1+2t_b+2t_m$

  = 29.50 + (2*4) + (2 *6.5)

  = 50.5 mm

  Where,

  Thickness of the bush (t_b) = 2 to 5 mm

  Marginal thickness (t_m) = 5 to 15 mm

  Inner diameter of the big end $d_2 = \frac{2*F\ im}{P_{b2}\times l_2}$

  $$d_2 = \frac{2*15857.53}{10.8\times 1.0 d_1}$$

  d_2= 76.63 mm

  Where,

  Design bearing pressure for big end P_{b2} = 10.8 to 12.6N/mm^2

  Length of the crank pin l_2 = (1.0 to 1.25) d_2

- Root diameter of the bolt $d_{bc} = \sqrt{\frac{2*F\ im}{\pi\times\sigma_t}}$

  $$d_{bc} = \sqrt{\frac{2*15857.53}{\pi\times 80}}$$

  d_{bc} = 11.23 =12 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 x root diameter of the bolt

  = 1.2× 12 =14.4 mm

- Outer diameter of the big end = $d_2 + 2t_b + 2d_b + 2t_m$

  = 76.63 + 2*2 + 2*14.4 + 2*5

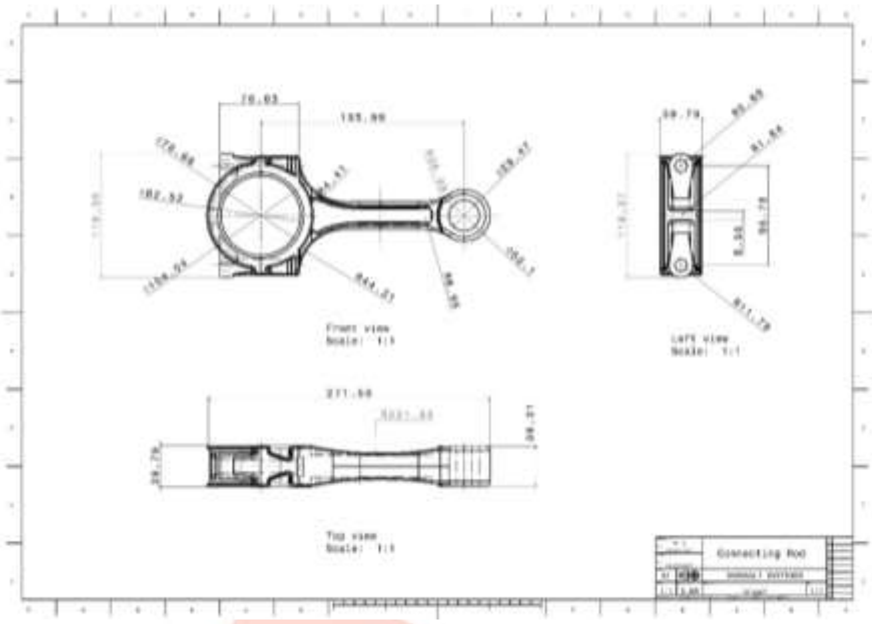
  = 119.43 mm

Specifications of connecting rod

Sr. No.	Parameters (mm)
1	Thickness of the connecting rod (t) = 4
2	Width of the section (B = 4t) = 16
3	Height of the section(H = 5t) = 20
4	Height at the big end = (1.1 to 1.125) H= 24
5	Height at the small end = 0.9H to 0.75H= 18
6	Inner diameter of the small end = 29.50
7	Outer diameter of the small end = 50.50
8	Inner diameter of the big end = 76.63
9	Outer diameter of the big end = 119.43

5. CAD MODELLING OF CONNECTING ROD

The connecting rod was digitized using a coordinate measuring machine. A solid model of the connecting rod, as shown in Figure 3.1, was generated using Catia V5. 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 1. 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.



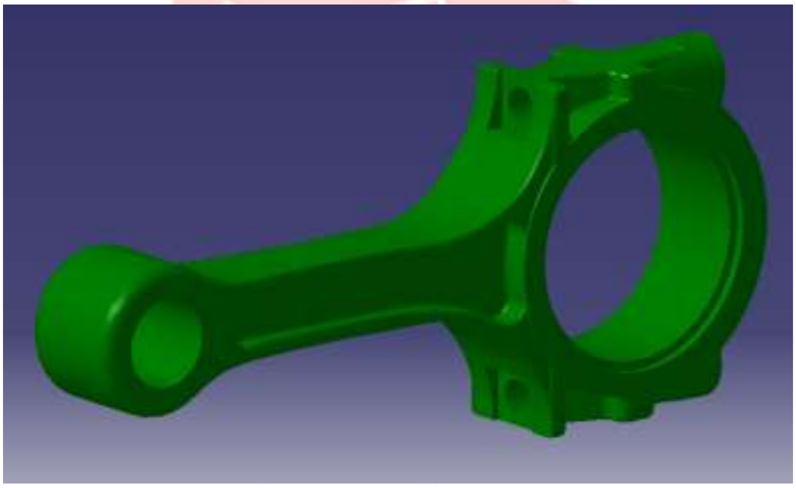


Figure 1: CAD Model of Connecting Rod

6. MESH GENERATION

Finite element mesh was generated using parabolic tetrahedral elements with various element lengths of 2.5mm. For most areas on the connecting rod convergence has been achieved with 1.5 mm uniform element length. 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 41431 elements. Further refinement was done locally by using element length of 0.8 mm. It can be seen that convergence has been achieved with 1 mm local mesh size.

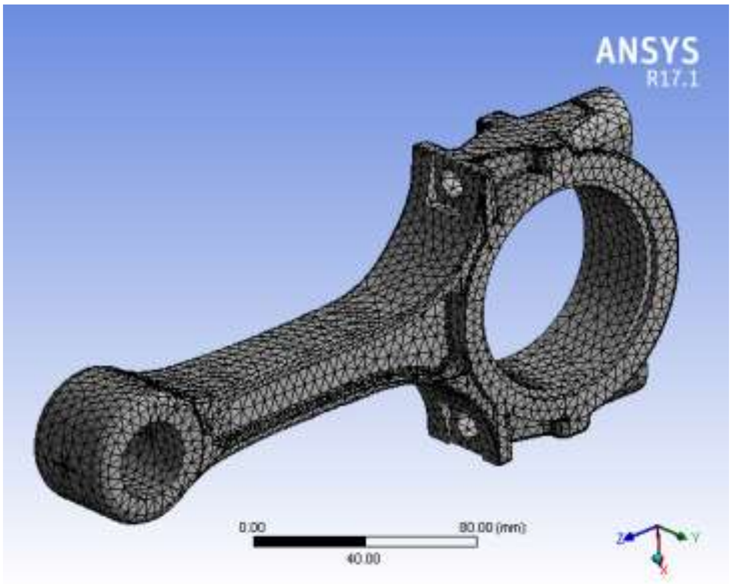


Figure 2: Mesh Model of Connecting Rod

MATERIAL PROPERTIES

	Steel 4340	Aluminium 7075-t651	42crmo4
Young's Modules	205	71.7	210
Poisson's Ratio	0.29	0.33	0.30
Density	7850	2810	7830
Yield strength	760	503	1034
Ultimate strength	1110	572	1200

7. DYNAMIC LOAD ANALYSIS OF THE CONNECTING ROD

The connecting rod undergoes a complex motion, which is characterized by inertia loads that induce bending stresses. In view of the objective of this study, which is optimization of the connecting rod, it is essential to determine the magnitude of the loads acting on the connecting rod. In addition, significance of bending stresses caused by inertia loads needs to be determined, so that we know whether it should be taken into account or neglected during the optimization. Nevertheless, a proper picture of the stress variation during a loading cycle is essential from fatigue point of view and this will require FEA over the entire engine cycle.
The objective of this chapter is to determine these loads that act on the connectingrod in an engine so that they may be used in FEA. The details of the analytical vector approach to determine the inertia loads and the reactions are presented as follows. This approach is explained by Wilson and Sadler (1993). The equations are further simplified so that they can be used in a spreadsheet format. The results of the analytical vector approach have been enumerated in this chapter. This work serves two purposes. It can used be for determining the inertia loads and reactions for any combination of engine speed, crank radius, pressure-crank angle diagram, piston diameter, piston assembly mass, connecting rod length, connecting rod mass, connecting rod moment of inertia, and direction of engine rotation.
In summary, this chapter enumerates the results of the analytical vector approach used for developing a spread sheet in MS EXCEL (hereafter referred to as DAP-Dynamic Analysis Program), verifies this DAP by using a simple model in Ansys, uses DAP for dynamic analysis of the forged steel connecting rod, and discusses how the output from DAP is used in FEA. It is to be noted that this analysis assumes the crank rotates at a constant angular velocity. Therefore, angular acceleration of the crank is not included in this analysis. However, in a comparison of the forces at the ends of the connecting rod under conditions of acceleration and deceleration with the forces under constant speed, the difference was observed to be less than 1%. The comparison was done for an engine configuration similar to the one considered in this study.

8.1 BOUNDARY CONDITION:

- Gas law equation

  PV=mRT

  P=mRT/V

$P = (61.48 * 10^{-2} * 36.15 * 10^{3} * 210) / 739 * 10^{3}$

$P = 6.3$ MPa

- Gas Force = $P * (\pi/4) * D^2$

  $F = 6.3 * (\pi/4) * 86^2$

  $F = 36595.5$ N

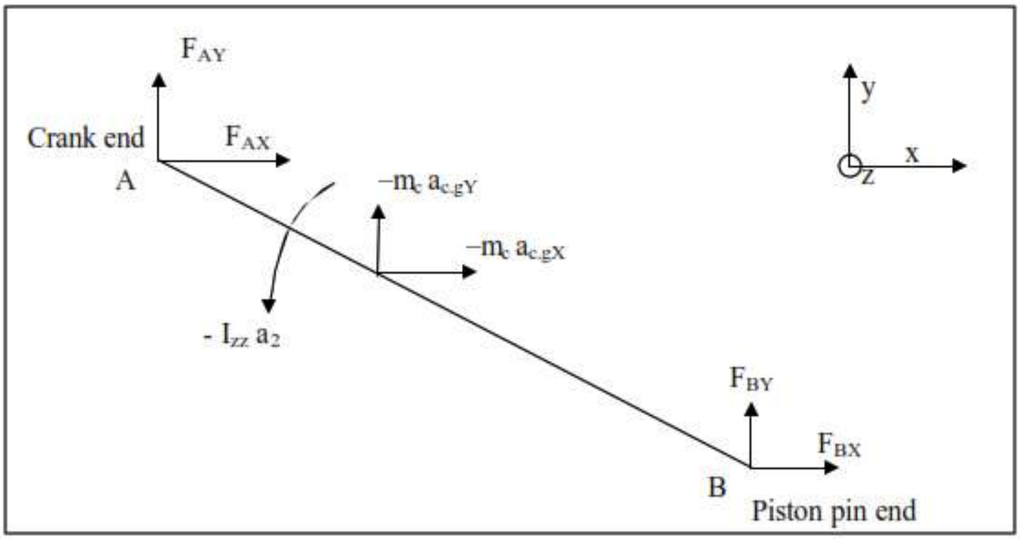


Figure 17: Free Body Diagram of connecting rod

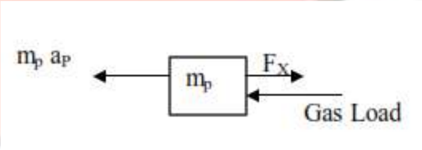


Figure 18: Free Body Diagram of Piston

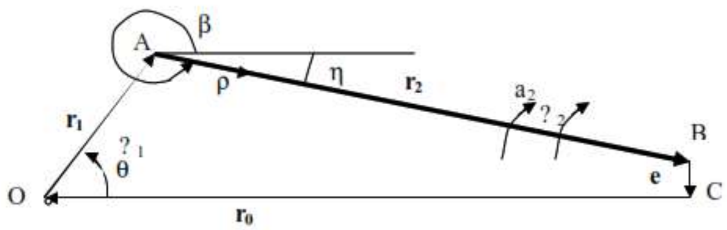
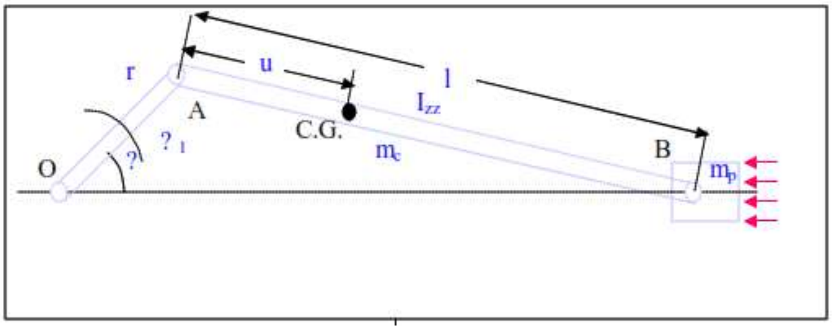


Figure 19: Vector representation of slider-crank mechanism.



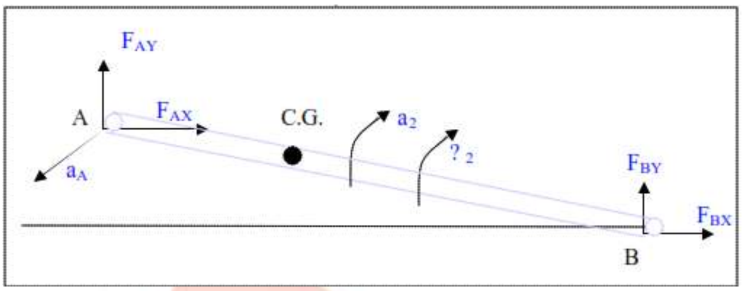


Figure 20: Typical input required for performing load analysis on the connecting rod andthe expected output.

With reference to Figure 1, for the case of zero offset (e = 0), for any given crank angle ϴ, the orientation of the connecting rod is given by:

$$\beta = \sin^{-1}\left\{\frac{-r1\ \sin\theta}{r2}\right\}$$

Angular velocity of the connecting rod is given by the expression:

$$\overline{\omega_2} = \omega_2\ \times k$$

$$\omega_2 = \frac{-\omega_1 \times \cos\theta}{[(r2/r1)^2 - \ \sin\theta^2]^{0.5}}$$

The angular acceleration of the connecting rod is given by:

$$\overline{a_2} = a_2\ \times k$$

$$a_2 = \left(\frac{1}{\cos\beta}\right)\left[{\omega_1}^2 \times \frac{r1}{r2} \times \sin\theta - \ {\omega_2}^2 \times \sin\beta\right]$$

Absolute acceleration of any point on the connecting rod is given by the following equation:

$$\bar{a} = \ (-r1 \times {\omega_1}^2 \times \cos\theta - \ {\omega_2}^2 \times u \times \cos\beta - \ a_2 \times u \times \sin\beta)i \\ + \ (-r1 \times {\omega_1}^2 \times \sin\theta - \ {\omega_2}^2 \times u \times \sin\beta + a_2 \times u \times \cos\beta)j$$

Acceleration of the piston is given by:

$$\bar{a} = \ (-r1 \times {\omega_1}^2 \times \cos\theta - \ {\omega_2}^2 \times r2 \times \cos\beta - \ a_2 \times r2 \times \sin\beta)i \\ + \ (-r1 \times {\omega_1}^2 \times \sin\theta - \ {\omega_2}^2 \times r2 \times \sin\beta + a_2 \times r2 \times \cos\beta)j$$

Forces acting on the connecting rod and the piston are shown in Figure 2.2. Neglecting the effect of friction and of gravity, equations to obtain these forces are listed below. Note that mp is the mass of the piston assembly and mc is the mass of the connecting rod. Forces at the piston pin and crank ends in X and Y directions are given by:

$$F_{BX} = -(m_p \,.\, a_p + Gas\ Load)$$

$$F_{AX} = m_c . a_{cgX} - F_{BX}$$

$$F_{BY} = (m_c . a_{cgY} . u . \cos\beta - m_c . a_{cgX} . u . \sin\beta + I_{zz} . a_2 + F_{BX} . r2 . \sin\beta)/(r2 . \cos\beta)$$

$$F_{AY} = m_c . a_{cgY} - F_{BY}$$

These equations have been used in an EXCEL spreadsheet. This program provides values of angular velocity and angular acceleration of the connecting rod, linear acceleration of the crank end centre, and forces at the crank and piston pin ends. These results were used in the FE model while performing quasi-dynamic FEA. An advantage of this program is that with the availability of the input as shown in Figure 17, the output could be generated in a matter of minutes. The loads required to perform FEA were obtained relatively quickly using this program.

	A	B	C	D	E	F	G	H	I	J	K	L	M	N	O
1	crank angle	Radius of crank	Connecting rod length	Connecting rod angle with positive direction of X axis	angular velocity of connecting rod	angular accelaration of connecting rod	absolute acc in x	absolute acc in y	angular acc of piston x	angular acc of piston y	angular accelarati on of piston	Fbx	Fax	Fby	Fay
2	θ	R1	R2	β	ω2	a2	acx	acy	apx	apy	ap				
3	0.00	0.05	0.20	0.00	-11.91	0.00	-114.07	0.00	-136.69	0.00	136.69	-36654.82	36561.17	0.00	0.00
4	45.00	0.05	0.20	-0.22	-6.44	493.58	-54.78	-74.78	-44.16	3.53	44.30	-36614.73	36569.75	8128.71	-8190.11
5	90.00	0.05	0.20	-0.22	5.05	499.86	50.46	-76.47	64.13	2.21	64.17	-36623.35	36664.77	8297.67	-8360.45
6	135.00	0.05	0.20	-0.02	11.44	50.50	100.49	-7.39	79.79	1.12	79.80	-36630.13	36712.64	800.38	-806.45
7	180.00	0.05	0.20	0.20	7.12	-450.80	64.66	68.16	70.99	-3.94	71.10	-36626.36	36679.45	-7399.59	7455.55
8	225.00	0.05	0.20	0.23	-4.16	-519.20	-35.06	79.70	-18.68	-1.56	18.74	-36603.63	36574.85	-8643.32	8708.76
9	270.00	0.05	0.20	0.04	-11.25	-100.49	-108.43	14.74	-127.91	-2.16	127.93	-36651.02	36562.00	-1595.15	1607.25
10	315.00	0.05	0.20	-0.19	-7.95	420.46	-69.89	-63.23	-67.42	4.57	67.58	-36624.83	36567.45	6859.13	-6911.04
11	360.00	0.05	0.20	-0.24	3.22	534.55	34.22	-82.28	52.85	0.97	52.86	-36618.44	36646.54	8933.09	-9000.64

Figure 21: DPA Sheet (Excel sheet of results)

CASE 1:

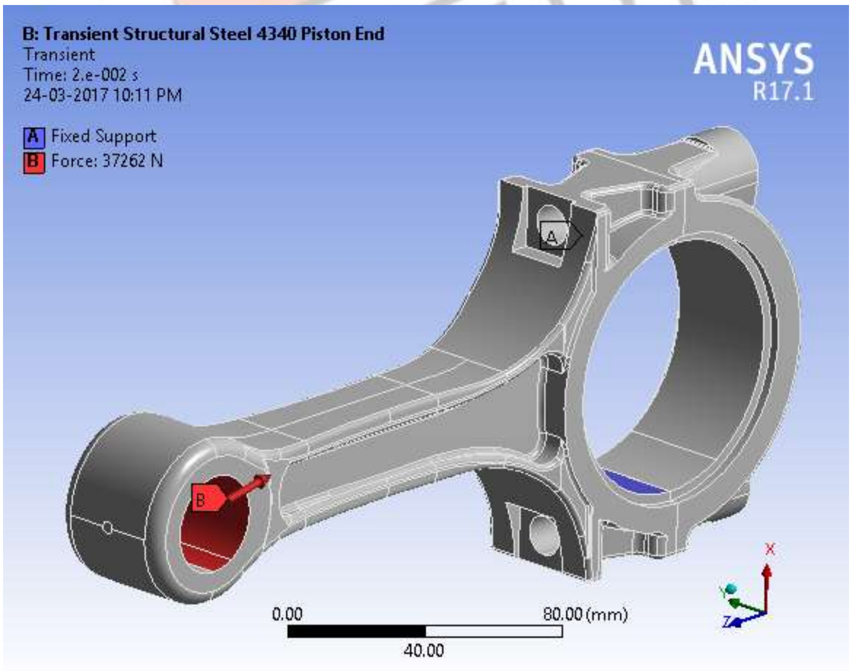


Figure 22: Boundary condition for Dynamic analysis in Ansys (Case 1: Load at Piston End)

Graph 1: Dynamic load of Case 1, Load at Piston end

Steps	Time [s]	X [N]	Y [N]	Z [N]
1	0.	0.	0.	0.
	2.5e-003			-36655
2	5.e-003	8128.7		-36615
3	7.5e-003	8297.7		-36623
4	1.e-002	800.38		-36630
5	1.25e-002	-7399.6		-36626
6	1.5e-002	-8643.3		-36604
7	1.75e-002	-1595.2		-36651
8	2.e-002	6859.1		-36625

CASE 2:

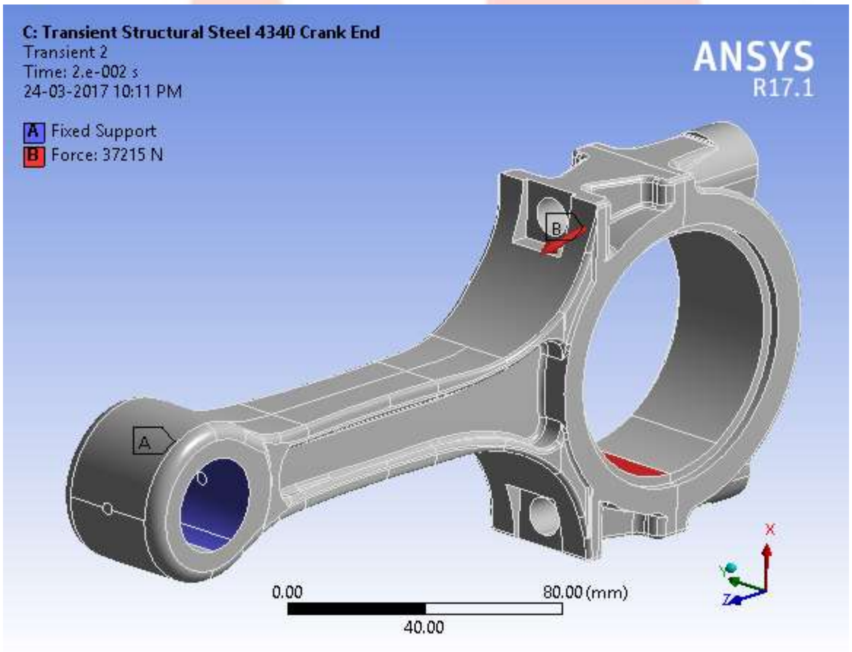
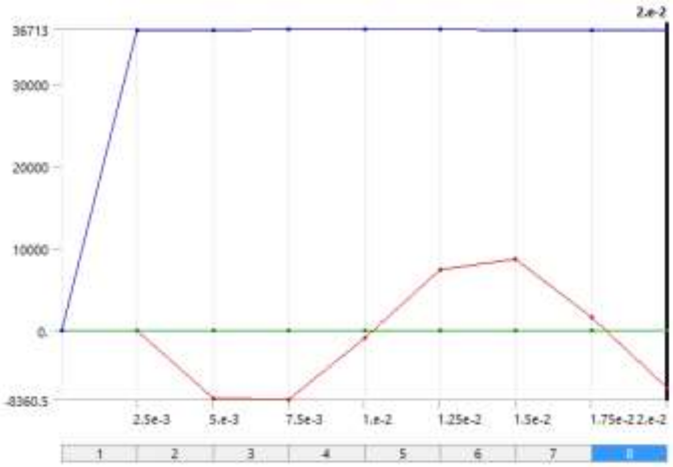


Figure 22: Boundary condition for Dynamic analysis in Ansys (Case 2: Load at Crank End)



Graph 2: Dynamic load of Case 2, Load at Crank end

Steps	Time [s]	X [N]	Y [N]	Z [N]
1	0.	0.	0.	0.
	2.5e-003			36561
2	5.e-003	-8190.1		36570
3	7.5e-003	-8360.5		36665
4	1.e-002	-806.45		36713
5	1.25e-002	7455.6		36679
6	1.5e-002	8708.8		36575
7	1.75e-002	1607.3		36562
8	2.e-002	-6911.		36567

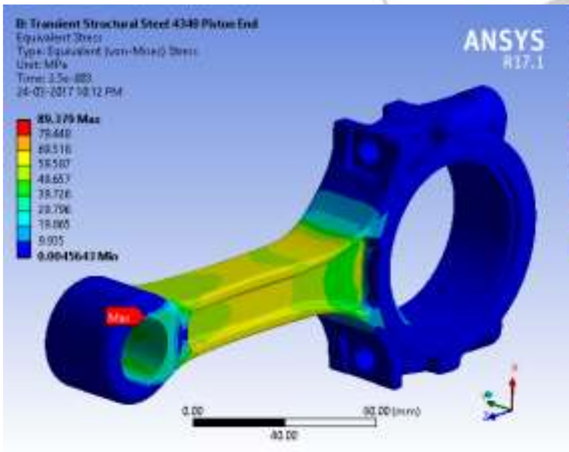
8.2 RESULT OF DYNAMIC ANALYSIS

Material 1: SAE 4340

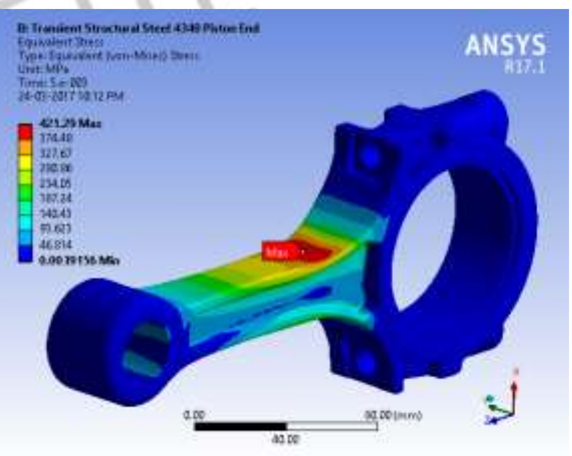
CASE 1

Von Mises Stress

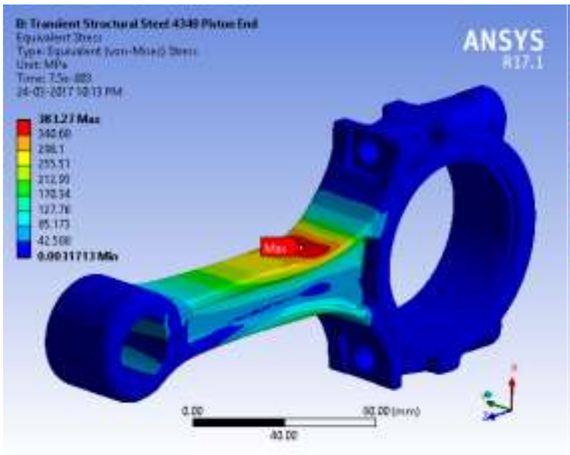
T = 0.0025



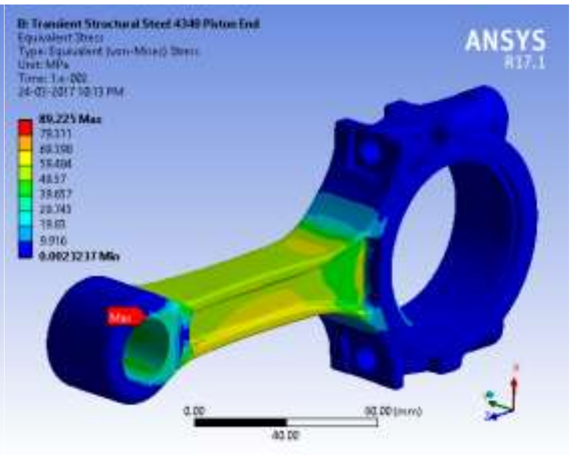
T = 0.005



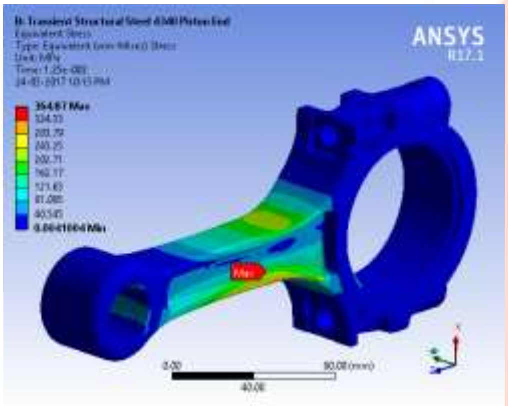
T=0.0075



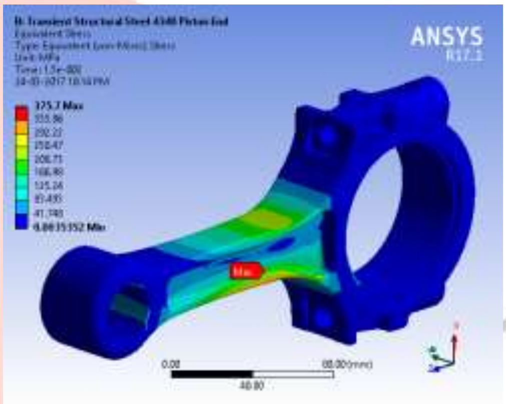
T=0.01



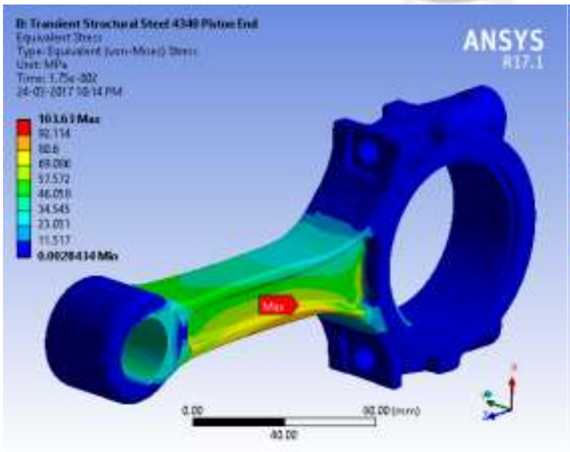
T=0.0125



T=0.0150



T=0.0175



T=0.02

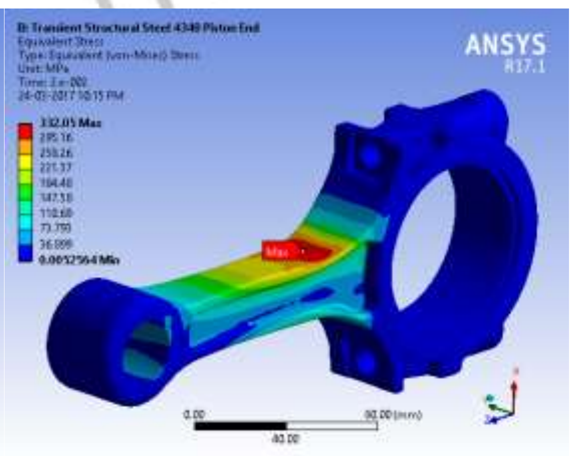
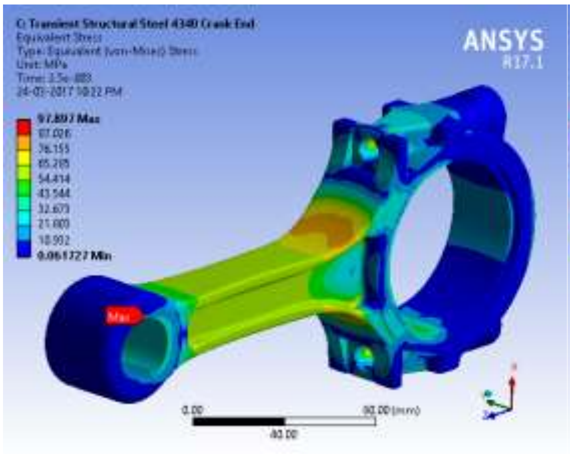
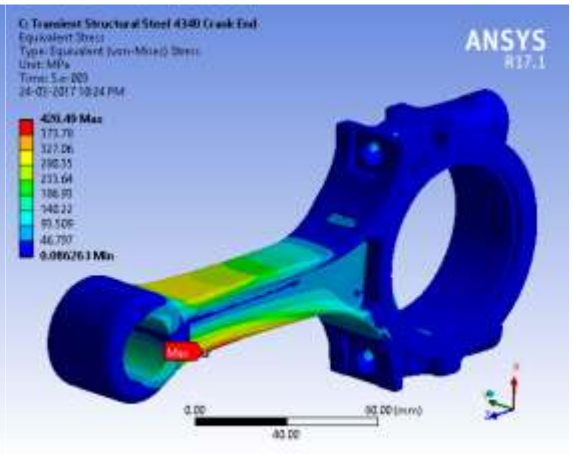


Figure 23: Result of Dynamic analysis of case 1 for material SAE 4340

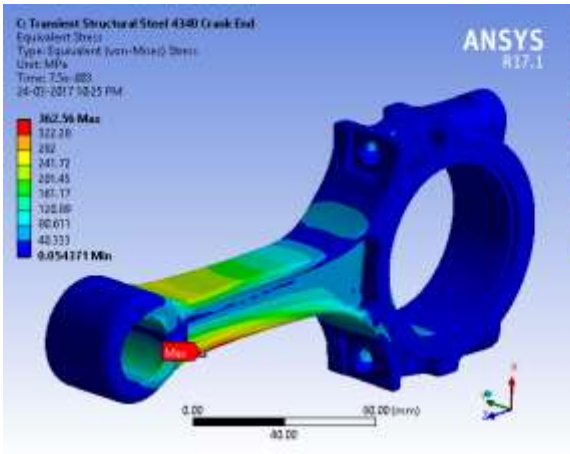
CASE 2 T=0.0025



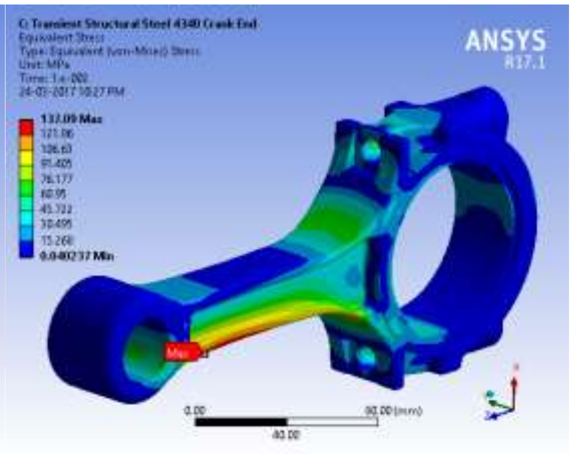
T=0.005



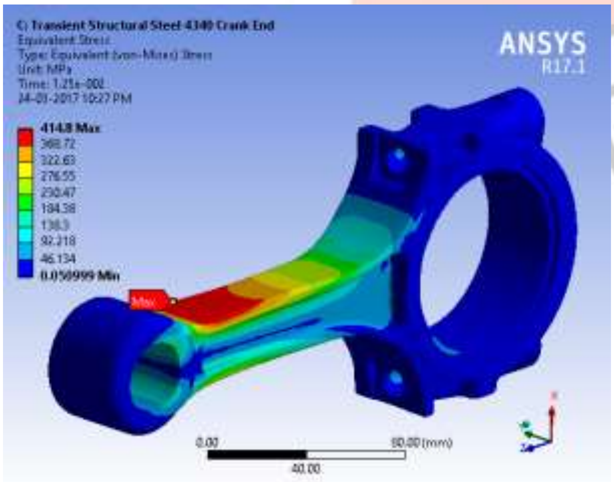
T=0.0075



T=0.010



T=0.0125



T=0.0150

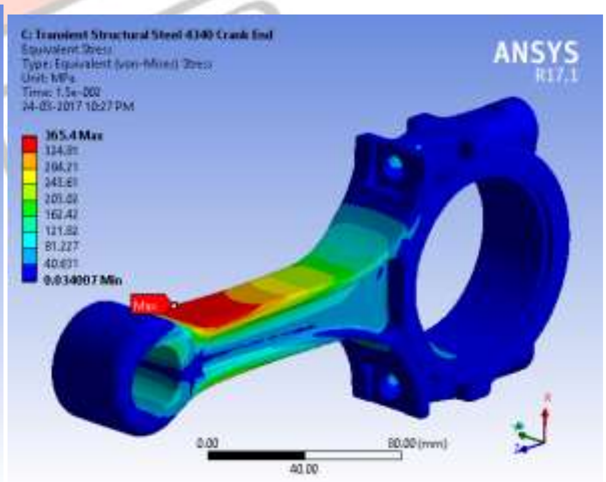
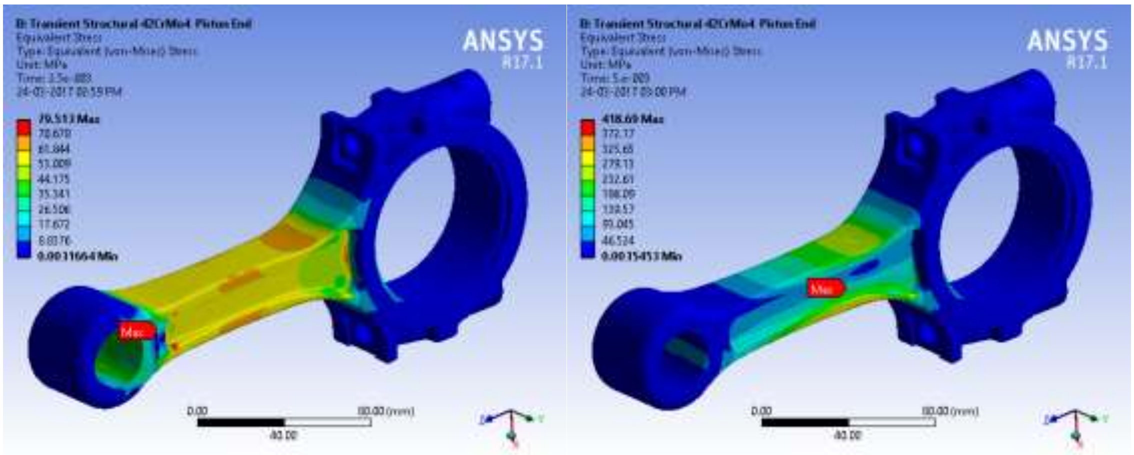


Figure : Result of Dynamic analysis of case 2 for material SAE 4340

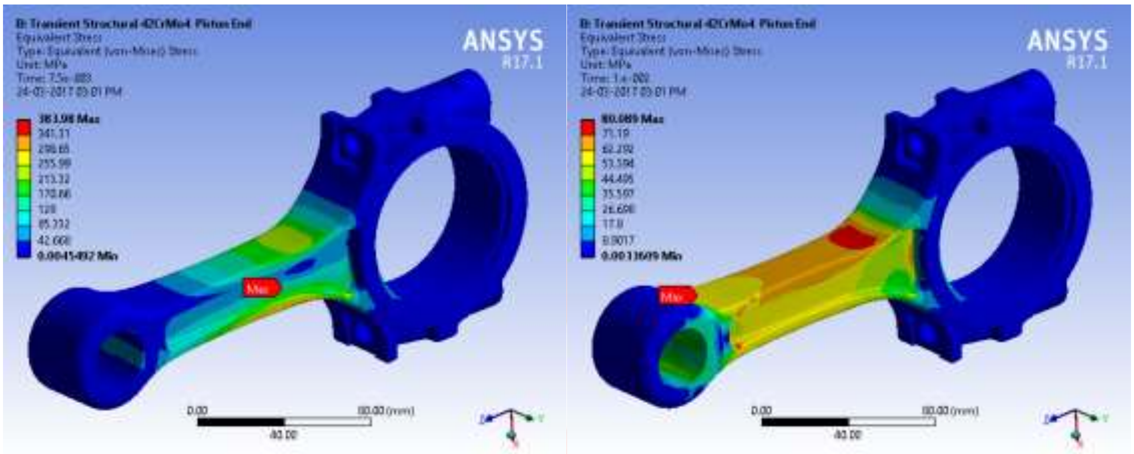
Material 2: 42CrMo4
Von Mises Stress

CASE 1

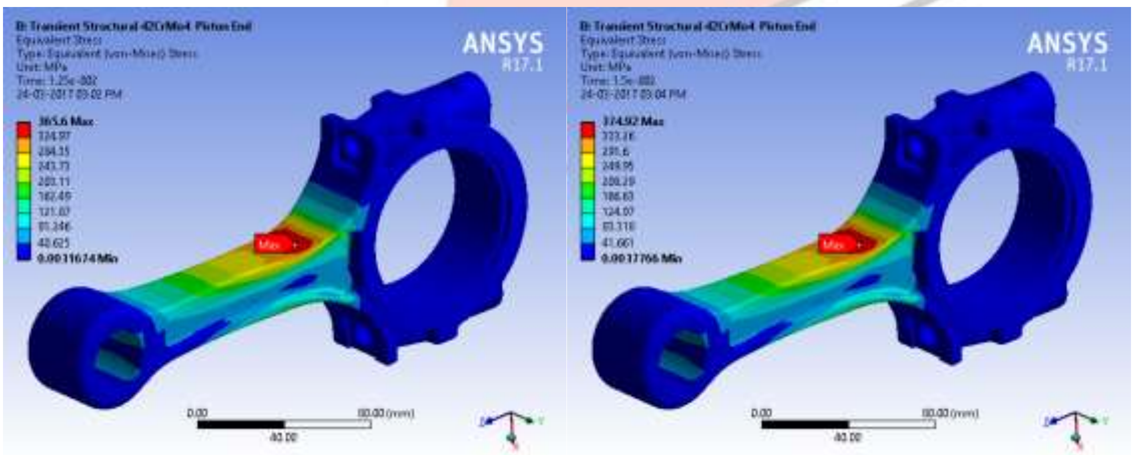
T = 0.0025 **T = 0.005**



T=0.0075 **T=0.01**



T=0.0125 **T=0.0150**

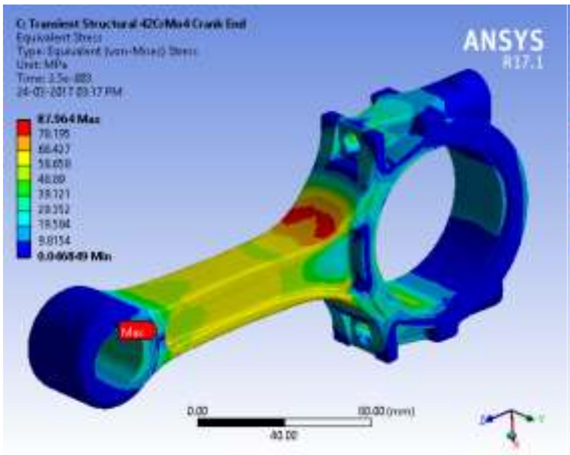


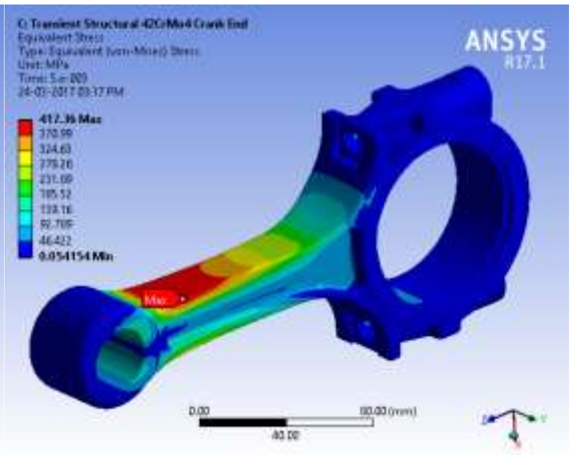
T=0.0175

Figure 25: Result of Dynamic analysis of case 1 for material 42CrMo4

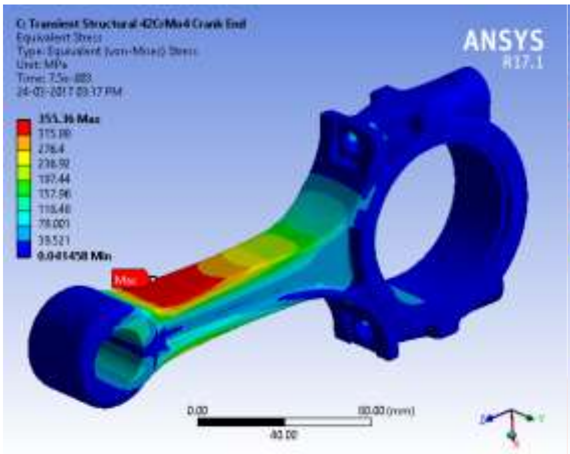
CASE 2

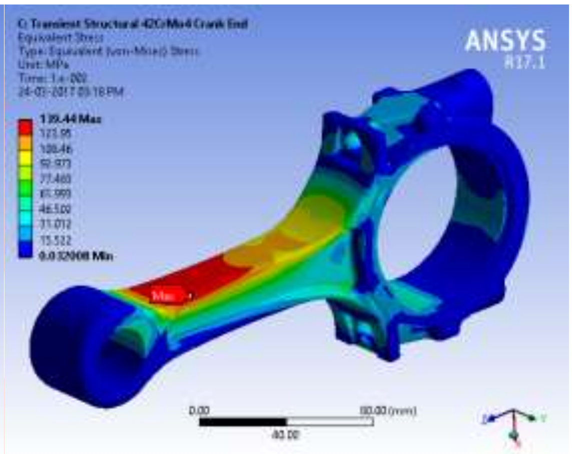
T=0.0025 T=0.0050



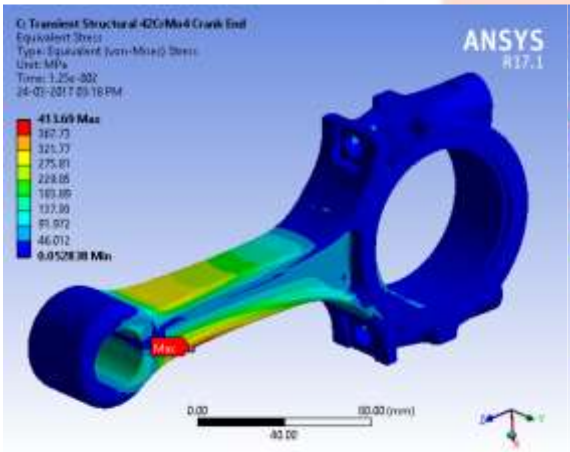


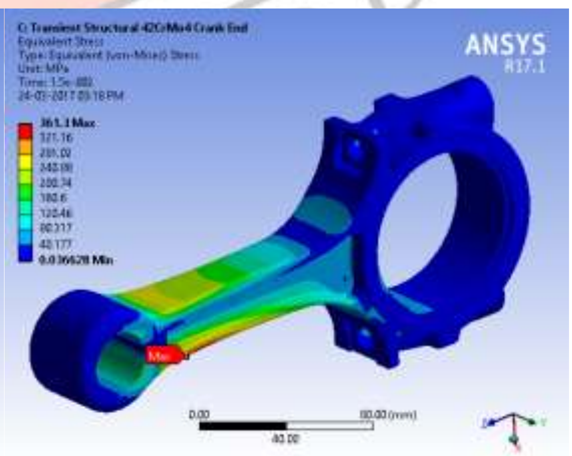
T=0.0075 T=0.010





T=0.0125 T=0.0150





T=0.0175 T=0.020

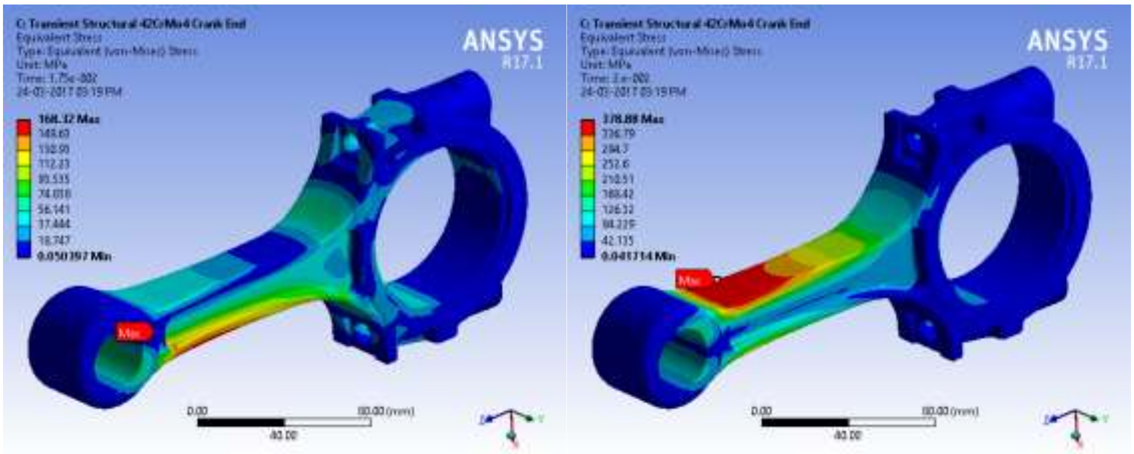
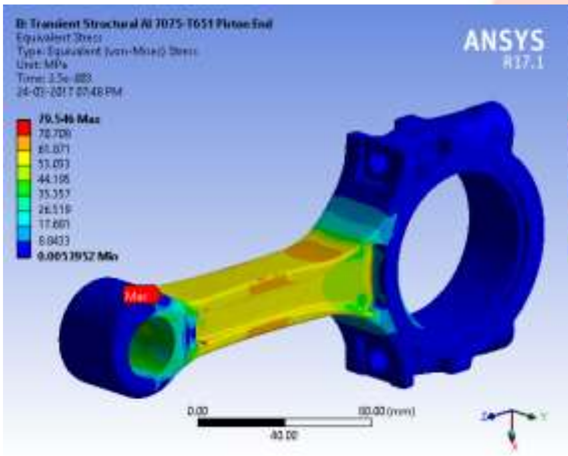


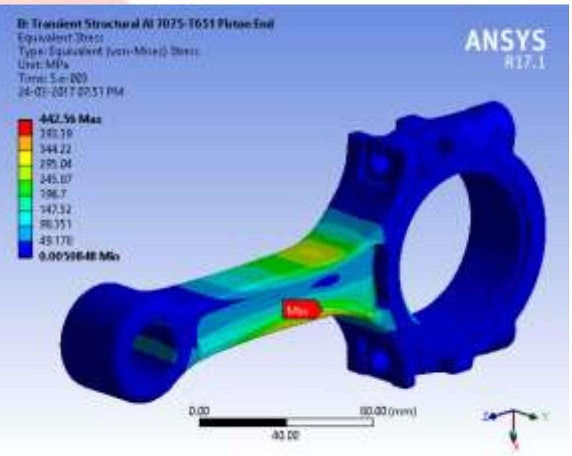
Figure 26: Result of Dynamic analysis of case 2 for material 42CrMo4

Material 3: Aluminium 7075-T651
Von Mises Stress

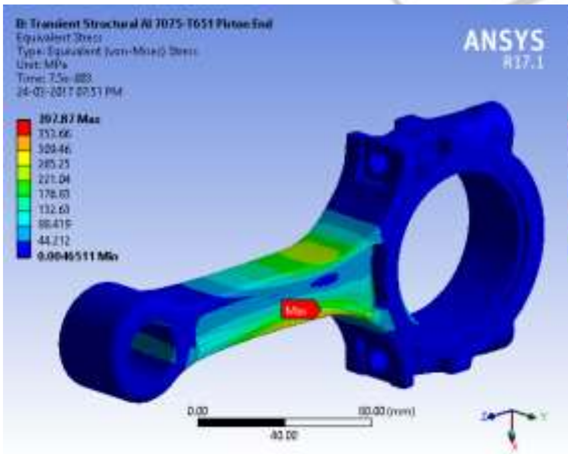
CASE 1

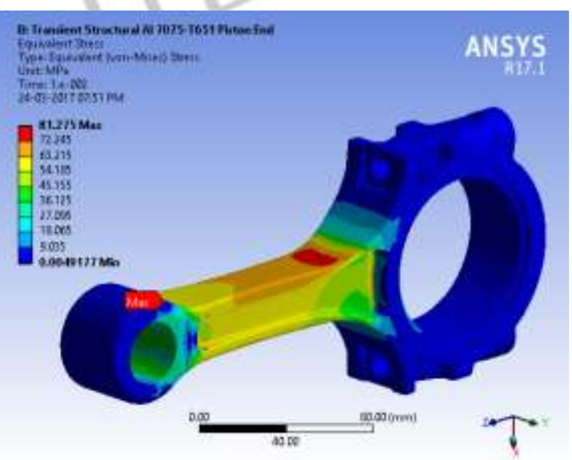
T = 0.0025 **T = 0.005**





T=0.0075 **T=0.01**





T=0.0125 **T=0.002**

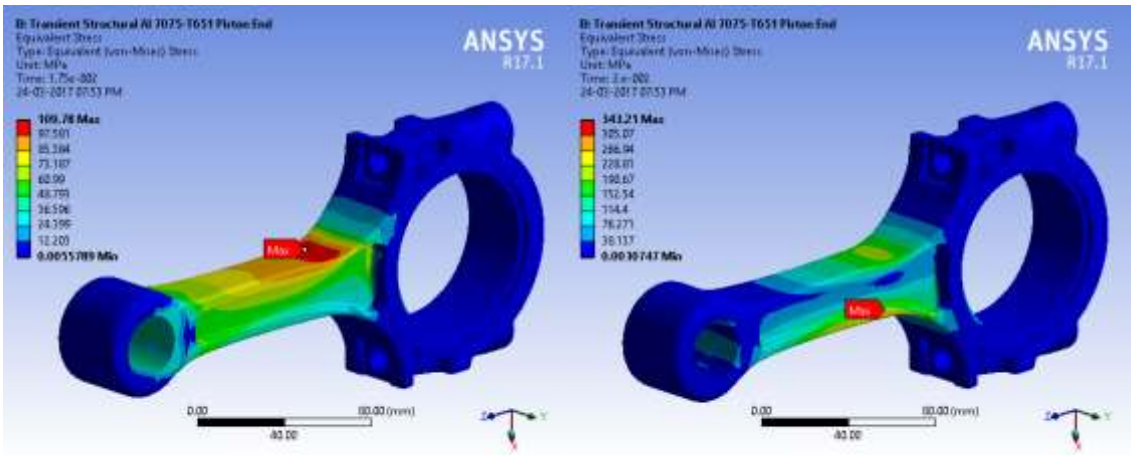
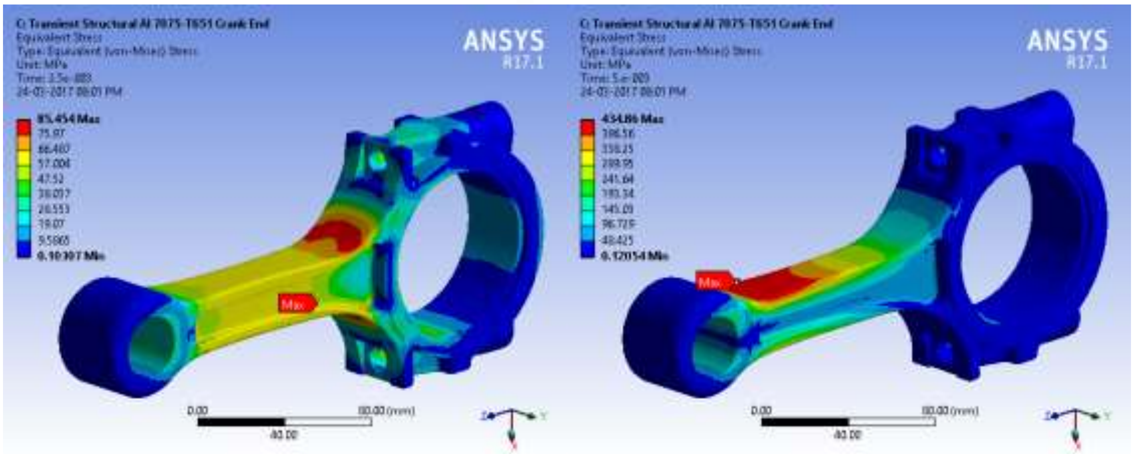


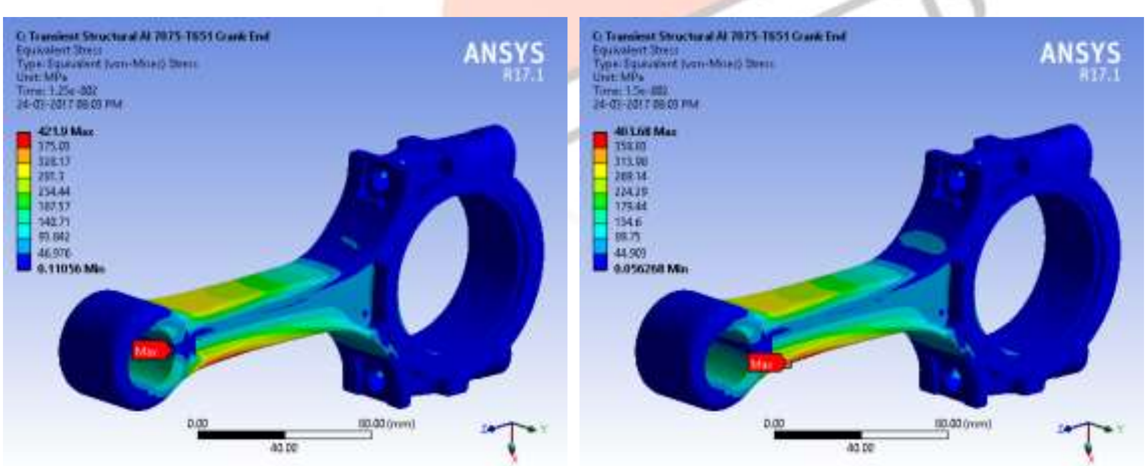
Figure 27: Result of Dynamic analysis of case 1 for material Al 7075-T651

CASE 2

T=0.0025 **T=0.0052**



T=0.0125 **T=0.0150**



T=0.0175 **T=0.020**

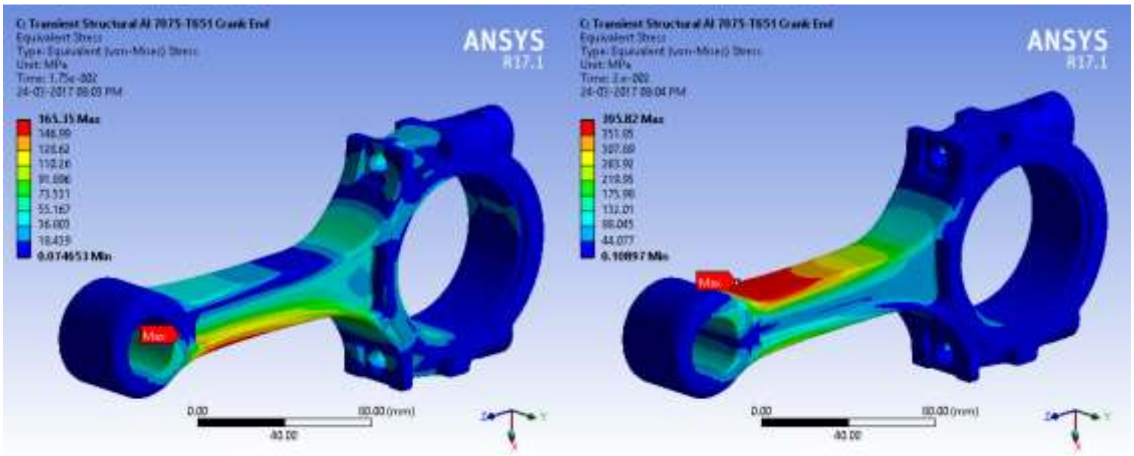


Figure 28: Result of Dynamic analysis of case 2 for material Al 7075-T651

- Result Table of Dynamic analysis

Result Table of Dynamic analysis of connecting rod for Case 1

Case 1 : Load at Piston End												
	Steel 4030				42CrMo4				Aluminium 7075-T651			
Time	Max Stress	Max Strain	Max Deformation	Weight	Max Stress	Max Strain	Max Deformation	Weight	Max Stress	Max Strain	Max Deformation	Weight
0.001250	44.629	0.000218	0.019382		39.706	0.000192	0.018907		39.692	0.000559	0.055216	
0.002500	89.379	0.000438	0.038902		79.513	0.000384	0.037944		79.546	0.001120	0.110830	
0.003750	216.15	0.001054	0.450480		215.8	0.001028	0.440600		223.31	0.003115	1.349700	
0.005000	421.29	0.002055	1.074600		418.69	0.001994	1.046900		442.56	0.006173	3.257700	
0.006250	402.58	0.001964	1.017200		400.07	0.001906	0.990780		423.06	0.005901	3.086600	
0.007500	383.27	0.001870	0.957870		383.98	0.001829	0.942040		397.87	0.005550	2.866800	
0.008750	288.26	0.001406	0.670590		286.64	0.001365	0.653060		303.53	0.004234	2.049400	
0.010000	89.225	0.000437	0.047072	2.3031 kg	80.089	0.000382	0.046962	2.2972 kg	81.275	0.001144	0.139350	1.920 kg
0.011250	186.78	0.000911	0.360330		185.15	0.000882	0.347590		195.36	0.002725	1.110900	
0.012500	364.87	0.001780	0.900280		365.6	0.001741	0.883170		375.01	0.005231	2.679200	
0.013750	423.17	0.002064	1.078100		420.74	0.002004	1.048100		445.06	0.006209	3.291700	
0.015000	375.7	0.001833	0.932240		374.92	0.001786	0.910150		389.47	0.005433	2.801900	
0.016250	314.12	0.001532	0.747390		313.95	0.001495	0.730760		324.99	0.004534	2.245200	
0.017500	103.63	0.000506	0.113070		102.13	0.000486	0.106160		109.78	0.001531	0.375660	
0.018750	171.61	0.000837	0.315900		169.74	0.000809	0.304060		181.83	0.002536	0.990530	
0.020000	332.05	0.001620	0.802300		331.61	0.001580	0.785870		343.21	0.004788	2.389600	

Result Table of Dynamic analysis of connecting rod for Case 2

Case 2 : Load at Crank End												
	Steel 4030				42CrMo4				Aluminium 7075-T651			
Time	Max Stress	Max Strain	Max Deformation	Weight	Max Stress	Max Strain	Max Deformation	Weight	Max Stress	Max Strain	Max Deformation	Weight
0.001250	48.602	0.000238	0.030601		43.683	0.000209	0.029851		42.259	0.000590	0.087238	
0.002500	97.897	0.000480	0.062136		87.964	0.000421	0.060587		85.454	0.001193	0.177220	
0.003750	167.76	0.000818	0.608900		167.31	0.000797	0.600130		170.06	0.002373	1.802900	
0.005000	420.49	0.002051	2.095900		417.36	0.001989	2.051400		434.86	0.006066	6.283200	
0.006250	520.38	0.002539	2.719600		512.6	0.002443	2.635300		553.73	0.007729	8.439300	
0.007500	362.56	0.001769	1.730500		355.36	0.001694	1.667600		398.02	0.005552	5.639800	
0.008750	212.65	0.001037	0.873770		213.68	0.001018	0.870490		219.6	0.003065	2.647300	
0.010000	137.09	0.000669	0.487360	2.3031 kg	139.44	0.000664	0.490760	2.2972 kg	127.17	0.001775	1.219500	1.920 kg
0.011250	162.97	0.000795	0.598060		166.62	0.000794	0.608870		162.1	0.002261	1.707800	
0.012500	414.8	0.002024	2.076400		413.69	0.001970	2.034400		421.9	0.005885	6.077800	
0.013750	475.52	0.002320	2.447100		466.54	0.002222	2.349300		512.76	0.007158	7.712900	
0.015000	365.4	0.001783	1.745600		361.3	0.001721	1.692500		403.68	0.005631	5.713300	
0.016250	274.42	0.001339	1.251700		278.71	0.001329	1.256200		278.88	0.003893	3.678700	
0.017500	169.27	0.000826	0.672820		168.32	0.000802	0.652450		165.35	0.002307	1.853700	
0.018750	163.78	0.000799	0.617420		169.31	0.000806	0.640630		156.2	0.002180	1.643800	
0.020000	384.92	0.001878	1.896100		378.88	0.001806	1.835400		395.82	0.005522	5.651800	

The load analysis was carried out to obtain the loads acting on the connecting rod at any given time in the loading cycle and to perform FEA. Most investigators have usedstatic axial loads for the design and analysis of connecting rods. However, lately, some investigators have used inertia loads (axial load varying along the length) during the design process. A comparison between the two is needed and is discussed in this chapter. Connecting rods are predominantly tested under axial fatigue loading, as it was the case for the connecting rod investigated in this project (Afzal, 2004). The maximum and minimum static loads can simulate the fatigue testing range. As a result, FEA was carried out under axial static load with no dynamic/inertia loads. In order to capture the structural behaviour of the connecting rod under service operating condition, dynamic FEA was also performed. Dynamic FEA results differ from the static FEA results due to time varying inertia load of the connecting rod which is responsible for inducing bending stresses and varying axial load along the length.

9. FUTURE SCOPE

The FEA of the connecting rod has been performed by the use of the software Catia V5 for cadmodelling and ANSYS WORKBENCH 18.1 for Finite Element Analysis. This work can be extended to study the effect of vibration on the connecting rod under dynamic conditions. Experimental stress analysis (ESA) can also be used to calculate thestresses which will provide more reasons to compare the different values obtained.
Now days a lot is being said about vibration study of mechanical component important role in its failure. So thestudy can be extended to the vibration analysis of the connecting rod. We can change the manufacturing process of the connecting rod forbetter result. Changing the geometry, but as it was the restriction from customer end, this is not covered in this project. Wecan notice from the design that the factor of safety considered for the design is too large which results in the wastage of thematerial and also increases its cost. So the need of the hour is the optimization of the connecting rod which will lead to arevolution in the manufacturing sectors of the automobile industry

10. CONCLUSION

The present work was aimed at evaluating alternate material for connecting rod with lesser stresses and lighter weight. This work found alternate material for minimizing stresses in connecting rod. FEA analysis performed using ANSYS 18.1 software for determining stresses & deformation. The following conclusions can be drawn from this study.
The Aluminium 7075-T651 connecting rod shows nearly same amount of stresses than existing carbon steel connecting rod.
From the above Dynamic analysis we seen that the maximum stress generated in case 1 i.e. when load applied to piston end is almost same in all material (In SAE 4340 = 423.17MPa, 42CrMo4 =420.74MPa & Al 7075-T651 = 445.06MPa) also in case 2 i.e. load applied to crank end, the maximum stress generated in all material is nearly same (In SAE 4340 = 523.38 MPa, 42CrMo4 =512.60 MPa & Al 7075-T651 = 553.73 MPa) but the weight of aluminium connecting rod is very less than that the other two materials. The deflection of Aluminium 7075-T651 connecting rod is same compared to deflection of existing carbon steel (42CrMo4) connecting rod. It is also found that the Aluminium 7075-T651connecting rod is light in weight (ie.35%) than existing carbon steel connecting rod approximately.

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