FINITE ELEMENT ANALYSIS OF CRITICAL COMPONENTS OF THE 2.6L GASOLINE ENGINE

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Abbreviations: FEA-Finite Element Analysis, CAE-Computer Aided Engineering
Keywords: FEA, Connecting rod, Crankshaft, Cylinder Block, Gasoline Engines

ABSTRACT

The Auto industry is faced with the growing challenges to fulfill the ever-growing demands of today's global customer, which puts a great pressure on the automotive R&D and test engineers to develop the critical components in the shortest possible time to minimize launch time for new products. This necessitates understanding of new technologies and quick absorption in the development of newer products. The main objective of this project is the development of 2.6 L capacity four stroke four cylinder inline petrol engine by up-gradation from the existing 1.8 L capacity engine. As the capacity of engine is increased, in accordance with the capacity of maximum pressure acting on the piston top is also increased. So to sustain high load coming from 2.6 L petrol engine to the same crankshaft of 1.8L engine with increased crank throw, it needs to be crosschecked against failure. The connecting rod which is proposed to be used in 2.6L engine is from the existing diesel engine, so it is also required to be crosschecked by FEA for proper functioning in the 2.6L petrol engine. The design modifications are also carried out in the combustion chamber profile of cylinder head, the water Jacket which surrounds the combustion chamber, height & bore of the cylinder block in order to increase the capacity of the engine. This paper also deals with the preprocessing of other critical component such as cylinder block in detail. The main concern is to perform structural dynamic analysis of the crankshaft and connecting rod to crosscheck its failure by FEA. The preprocessing and analysis is performed with Altair's HyperMesh as a pre-processor, Altair's OptiStruct as a solver & HyperView is used as post-processor.

Introduction

In the last years Finite Element Analysis (FEA) have evolved as a powerful tool to supporting engineers in various fields of product development and research. With the continuous increasing computational capabilities, these techniques become more and more important for the effective development of competitive products. Crankshaft is a large component with a complex geometry in the engine, which converts the reciprocating motion of the piston to a rotary motion with a four bar link mechanism. It is the most critically loaded component and experiences cyclic loads in the form of bending and torsion during its service life. Two main sources of loading exist for crankshafts; gas force due to the combustion process in the combustion chamber transmitted to the crankshaft by connecting rods, and inertia force due to the mass of the component and its attachments. Due to the dynamic nature of the system, both the gas and the inertia forces apply bending and torque to the crankshaft. Failures of crankshaft do occur after a fatigue process. The failure has begun at the sharp fillet region and the lubrication holes influenced the crack growing direction. [1]. 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. It undergoes high cyclic loads of the order of 10^8 to 10^9 cycles, which range from high compressive loads due to combustion, to high tensile loads due to inertia Ref [3]. Analytical approach was solved for a general slider crank mechanism which results in equations that could be used for any crank radius, connecting rod geometry, connecting rod mass, connecting rod inertia, engine speed, piston diameter, piston and pin mass, and any other variables of the engine. Vector Approach is performed to obtain the angular velocity, angular acceleration, linear acceleration of piston & to know the critical load acting on the crankshaft pin journal. Ref [4]. Calculation of dynamic forces, torque acting on the crankshaft and connecting rod are carried out. The FE models of the above parts are created by using Altair HyperMesh as preprocessor and boundary conditions are applied. The FE model is solved by OptiStruct and the results are evaluated by the postprocessor used as HyperView. The results are correlated with mechanical properties of the particular components and the conclusions are drawn accordingly. Moreover, by using 3D-Tetrahedral 4 noded elements the other critical components are meshed and the boundary conditions are identified. The FE analysis part will be carried out later.

Process Methodology

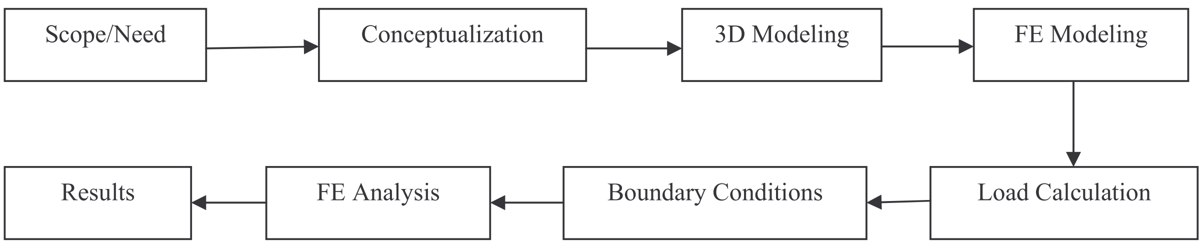

FIGURE 1: DESCRIPTION OF THE PROJECT

Scope/Need:

The main objective of this project is the development of 2.6 L capacity four stroke four cylinder inline petrol engine by up gradation from the existing 1.8 L capacity engine. Since this is an upgradition project, it was decided that instead of going for a new design starting from the basic concept, it is better to reengineer the critical components of the existing in-house engines. It drastically reduces cost and development time.

Conceptualization:
Cylinder bore and height of the 1.8L is varied to increase the capacity of engine to 2.6L. Accordingly the design modifications are carried out in combustion chamber profile of the cylinder head, reduction in wall thickness between two consecutive cylinders and changes in the water jackets in the cylinder head. Crank throw is increased in order to increase the stroke of the engine. The connecting rod of existing turbo- diesel engine is used without any design changes.

3D modeling:
All the components are modeled by using CATIA V5 R17. In case of connecting rod, by using respective density of material, the C.G and moment of inertia of the rod is found. The modeled components are exported to IGES format, which is able to retrieve by HyperMesh for the preprocessing of the part.

FE Modeling:
The accuracy of the FE analysis results solely depends upon the element size and meshing pattern of the parts. Hence it is essential to give more focus on the meshing of the components. Mesh size is decided based upon the dimension of the critical locations of the parts, which is more important for the analysis. FE meshing is started with the geometry cleanup operation, which is used to make the geometry more appropriate for meshing. It is a common practice used to combine a number of faces into a single smooth surface. This allows the elements to be created on the entire region at once, and prevents unnecessary artificial or accidental edges from being present in the final mesh. In order to mesh the geometry with smooth meshing, first the shell mesh (2D) of the component is done, and then it is converted to 3D- 4 nodded tetrahedral elements. During shell meshing, tria elements are used for global mesh size, while for critical locations number of elements is increased for accurate stresses at locations with high stress gradients. Average size of the elements of the crankshaft is 3mm, taking minimum size 1mm & maximum size is equal to 5mm. In case of connecting rod, the average size of the elements is 2mm, taking minimum size 1mm & maximum size is equal to 4 mm.
After shell meshing, checking of the model quality index is performed. The quality parameters such as aspect ratio, warpage, skew, jacobian and minimum, maximum angles for the tria elements, minimum and maximum length are

checked. It is essential that the particular model should qualify through all of these criteria's. The failure of anyone of the parameters leads to the failure during 3D meshing or FE analysis.

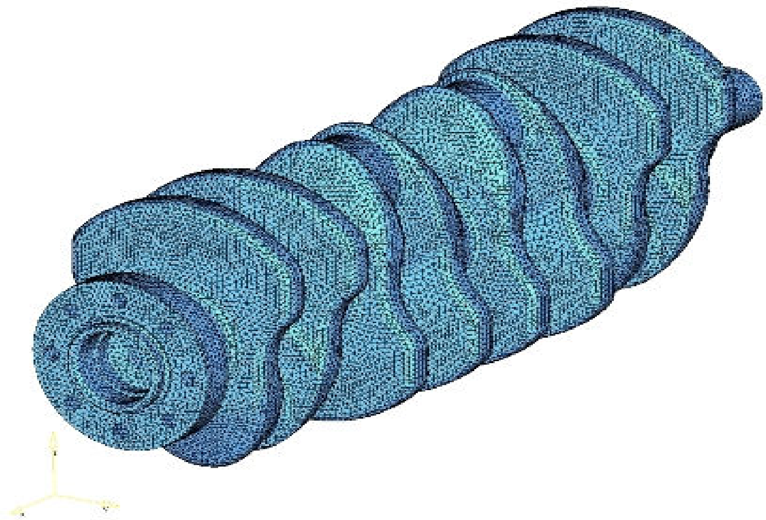

FIGURE.2: FE MODEL OF CRANKSHAFT

The meshed geometry of crankshaft is shown in figure 2, having tetrahedral elements with minimum tetra collapse of 0.22 & minimum interior angle of 16.60^{o}. The geometry has 79568 nodes and 353556 tetrahedral elements of 3.25mm as element size.

FIGURE.3: FE MODEL OF CONNECTING ROD

The geometry of bolts and threaded portions of connecting rod is represented by an equivalent RBE2 and CBAR element in HyperMesh. The CBAR element connects the two RBE2 elements one consist the nodes at periphery of bolts and the other RBE2 elements consist of the nodes at the threaded area of the nut. The connecting rod is meshed with average element size of 1.5mm, which has tetrahedral elements with minimum tetra collapse of 0.26 & minimum interior angle of 12.33^{o}. Total number of tetrahedral elements is 108202 and it has 26158 nodes which is shown in figure 3.

For cylinder block, only 2D meshing is carried out, the element size is around 3mm, it has 2D-Tria elements of 379880 and 190870 nodes. The FE model of cylinder block is shown in figure 4.

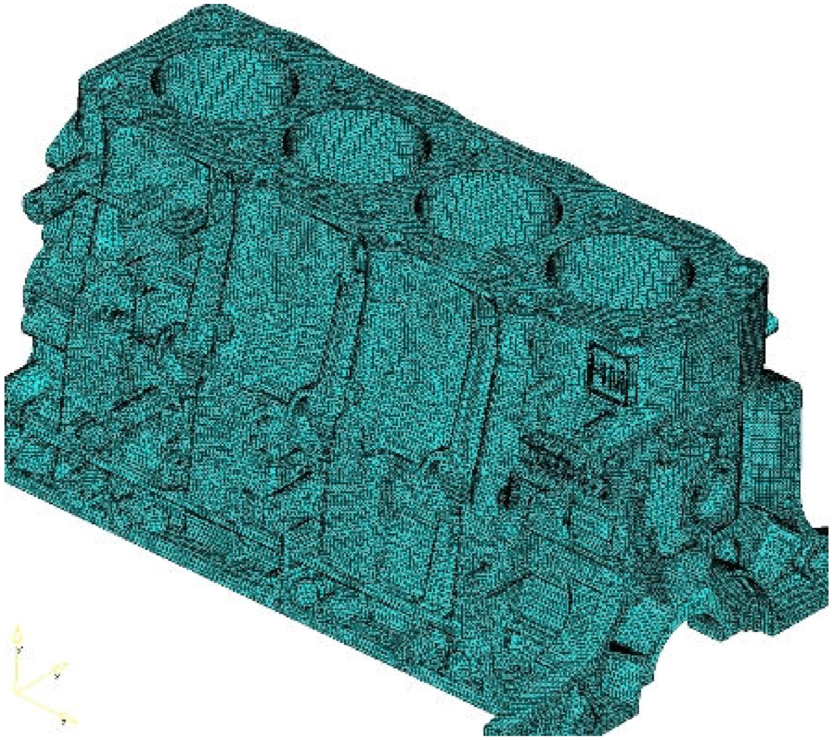

FIGURE.4: FE MODEL OF CYLINDER BLOCK

Load Calculation:

This chapter discuss the formulae, methods and magnitudes of forces which are acting over the connecting rod, piston and crankshaft constitutes the four bar slider-crank mechanism, which converts the sliding motion of the piston (slider in the mechanism) to a rotary motion as shown in figure.5. Crankshaft experiences large forces from gas combustion. This force is applied to the top of the piston and since the connecting rod connects the piston to the crankshaft, the force will be transmitted to the crankshaft. It undergoes high cyclic loads of the order of 10^8 to 10^9 cycles, which range from high compressive loads due to combustion, to high tensile loads due to inertia. The main objective of this chapter is to determine the magnitude and direction of the loads that act on the bearing between connecting rod and crankshaft, which was then used in the FEA over an entire cycle. An analytical approach was used on the basis of a single degree of freedom slider crank mechanism. The angle θ shown in figure.5 represents the crankshaft angle, which is used as the generalized degree of freedom in the mechanism; therefore every other dynamic property in this mechanism would be a function of this angle Ref [1].

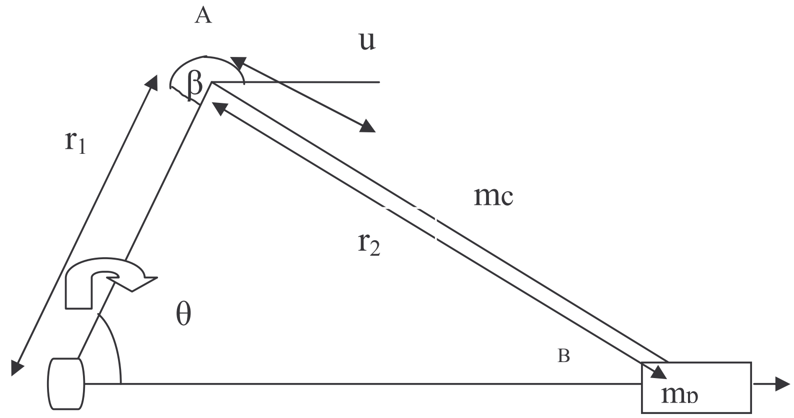

FIGURE.5: FOUR BAR SLIDER CRANK MECHANISM

Let,

- r -Radius of crankshaft pin
- r_1 -Radius of crank
- r_2 -Connecting rod length
- m_p, m_c -Mass of piston assembly, connecting rod
- u -Distance of C.G. of the connecting rod from crank end center
- ω_1 -Angular velocity of crankshaft
- ω_2 -Angular velocity of connecting rod
- α_2 -Angular acceleration of connecting rod
- θ -Crank angle, angular coordinate of polar coordinate system defined for the contact surface
- β -Connecting rod angle with positive direction of X axis
- I_{ZZ} -Moment of Inertia about Z axis and C.G. of the connecting rod
- a -Absolute acceleration of a point on the connecting rod
- A_{gx}, A_{gy} X, Y -components of the acceleration of the C.G. of the connecting rod
- A_A -Acceleration of point A of piston
- F_{AX}=Force in crankpin end in global X coordinate system
- F_{AY}=Force in crankpin end in global Y coordinate system
- F_{BX}=Force in piston pin end in global X coordinate system
- F_{BY}=Force in piston pin end in global Y coordinate system
- F_X =Bending Force in rotating coordinate system
- F_Y=Torsional Force in rotating coordinate system

The angular velocity and acceleration of the crankshaft are given by:

$$\omega_1 = d\theta/dt$$

The linkage can be described by the following vector equation:

$$r_1 + r_2 + r_3 = 0 \qquad (1)$$

$$r_1 e^{i\theta} + r_1 e^{i\beta} + r_3 = 0 \qquad (2)$$

Solving Equation (2) for both real & imaginary form gives the angle of the connecting rod, β, is related to θ by:

$$\mathrm{Sin}\beta = (- r_1.\mathrm{Sin}\theta)/ (r_2)$$

Differentiating Equation (1) w.r.t. time:

$$\omega_1 \times r_1 + \omega_2 \times r_2 - Vp=0 \qquad (3)$$

And the angular velocity of the connecting rod is given by the expression:

$$\omega_2 = - \omega_1 \cos\theta / [(r_2/r_1)^2 - \sin^2\theta]^{0.5} \qquad (4)$$

To obtain the angular acceleration, consider Equation 3. Differentiating the equation w.r.t. time, for constant angular velocity of the crank, and after substitution, we get:

$$\alpha_2 = (1/ \cos\beta) [\omega_1^2 (r_1/r_2) \sin\theta - \omega_2^2 \sin\beta] \qquad (5)$$

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

$$a = (-r_1 \omega_1^2 \cos\theta - \omega_2^2 u \cos\beta - \alpha_2 u \sin\beta) i + (-r_1 \omega_1^2 \sin\theta - \omega_2^2 u \sin\beta + \alpha_2 u \cos\beta) j \qquad (6)$$

Acceleration of Piston is given by:

$A_p = (- r_1\ \omega_1^2\cos\theta - \omega_2^2\ r_2\ \cos\beta - \alpha_2\ r_2\sin\beta)\ i + (- r_1\ \omega_1^2\sin\theta - \omega_2^2\ r_2\ \sin\beta + \alpha_2\ r_2\ \cos\beta)\ j$ (7)

Forces at the crank end and piston end of connecting rod is derived as below,

By applying dynamic Equilibrium conditions to the piston we get:

F_X – mp A_{px} - Gas Load = 0

The corresponding force in the X direction at the pin end is given by:

F_{BX} = – (mp A_{px} + Gas Load) (8)

Application of dynamic equilibrium conditions to the connecting rod results in the following equations:

$F_{AX} + F_{BX}$ – mc A_{gx} = 0 (summing forces in the X direction)

$F_{AY} + F_{BY}$ – mc A_{gy} = 0 (summing forces in the Y direction)

Taking moment about point in the crank pin end:

$F_{BX}\ r_2\ \sin\eta + F_{BY}\ r_2\cos\eta + (-\ mc\ A_{gx})\ u\ \sin\eta + (-\ mc\ A_{gy})\ u\ \cos\eta + (-\ I_{zz}\ \alpha_2) = 0$

Solving the above three equations gives:

F_{AX} = mc A_{gx} - F_{BX} (9)

F_{BY} = [mc A_{gy} u cosß - mc A_{gx} u sinβ + $I_{zz}\ \alpha_2$ + $F_{BX}\ r_2$ sinß] / $r_2$cosβ (10)

F_{AY} = mcA_{gy} - F_{BY} (11)

F_{AX} and F_{AY} are expressed in the global coordinate system, which is not rotating with the crankshaft. Forces expressed in a coordinate system attached to the crankshaft, better explain the loading history applied to the crankshaft. These forces are given by:

$F_X = F_{AX}\text{Cos}\theta + F_{AY}\text{Sin}\theta$ (12)

$F_Y = F_{AY}\ \text{Cos}\theta - F_{AX}\ \text{Sin}\theta$ (13)

Equations derived from analytical approach are used to plot the curves for the load acting on the crankshaft for 720^o rotation of the crankshaft with the help of Microsoft Excel Programming. It should be pointed out that in this analysis it was assumed that the crankshaft rotates at a constant angular velocity, which means the angular acceleration was not included in the analysis. However, in a comparison of forces with or without considering acceleration, the difference was found to be less than 1%. Ref [4].It should be noted that the pressure versus volume of the cylinder graph changes as a function of engine speed, which is used during the calculation of forces. The bending force and torsional moment in rotating coordinate system, which are obtained by Microsoft excel programming, is plotted in the graph and the magnitudes of forces are referenced from this for FE analysis. The variation of magnitude of forces for each cylinder depends upon the firing order of the engine.

Boundary conditions:

Identification of correct boundary conditions determines the accuracy of the FEA results. Hence more emphasis is given to this part of the analysis. The applied loading patterns should depict the actual working condition of the components/systems as in the field. In case of connecting rod, the crank and piston pin ends are assumed to have a sinusoidal distributed loading over the contact surface area, under tensile loading (The load is distributed over an angle of 180°). This is based on experimental results.Ref [2]. The tensile load acting on the connecting rod 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. Ref [2]. The compressive load, Pc can be obtained from the indicator diagram of an engine.

The explanation of load distribution in the Webster et al. study is for connecting rods, but since the crankshaft is in interaction with the connecting rod, the same loading distribution of both compressive loading and tensile loading will be transmitted to the crankshaft Ref [5]. The value of Bending & torsional force was obtained from the dynamic simulation of the crankshaft for whole 720^o rotation of the crankshaft as in graph. Applying the respective forces on each cylinder as got from the firing order of the cylinders i.e. 1-3-4-2. The cylinder which is fired has maximum bending compressive force applied over 120^o on the top surface of crank and other tensile force is applied over 180^o over the surface of the crank. Torsional moment was obtained by multiplying the torsional force of the respective cylinder and the crank throw. Torsional moment is applied by selecting the node at the periphery where crank web, main journal are in contact of both end of crank and making the rigid with nodes.

FE Analysis:

FEA of Crankshaft:

There are two major approaches for stress calculation of crank shaft:

(a) Based on entire crank

(b) Based on single crank

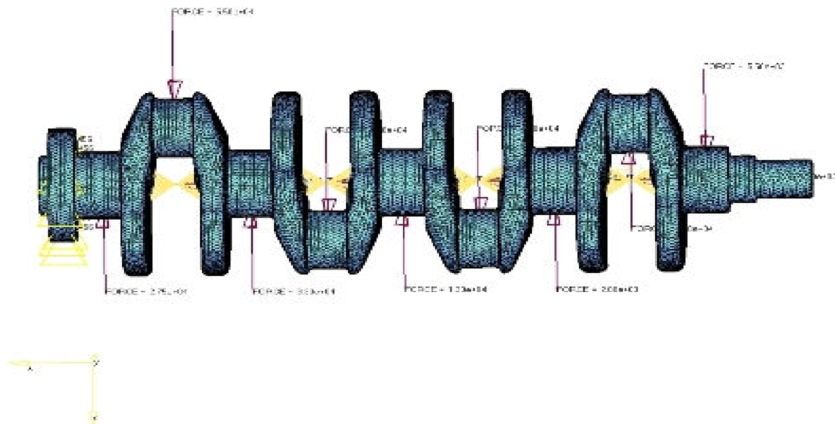

FIGURE.6: FE MODEL OF CRANKSHAFT WITH FIRST LOAD CASE

In the first case of FEA analysis, the whole crankshaft is constrained at the flywheel end is shown in figure.7.The maximum bending force applied is 55KN, the forces on other cranks are 11.205KN, 15KN, 11.4KN. Maximum torsional moment applied on the crank is 1425000 N-mm on one of the crank having maximum torsional force other moments on the respective cranks are obtained from the graph.

In the second case of FEA Analysis, Cut out one throw of the crank through the main journal middle cross-sections. Constrain one cross section for all degrees of freedom. Applying maximum Compressive load at the crank pin & bearing area as a pressure load, same boundary conditions are used as above. Torsional moment was applied on the rigid formed by selecting the nodes on the surface of the main journal of both the end. Maximum pressure load acting is 62.5Mpa whereas torsional moment is 1425000N-mm.

FEA of connecting rod:

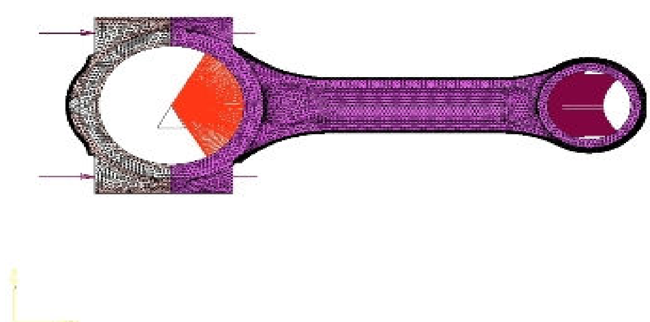

FIGURE.7: FE MODEL OF CONNECTING ROD WITH FIRST LOAD CASE

In this, three load cases exists, such as (i) Constrain the crank pin end for all degrees of freedom of the connecting rod and applying compressive load of 90Mpa at 120° in piston pin end as shown in figure 8. (ii) Constrain the piston pin end for all degrees of freedom of the connecting rod and applying compressive load of 62.5Mpa at 120° in crank pin end.(iii) Constrain the piston pin end for all degrees of freedom of the rod and apply tensile load of 14.3 Mpa at 180° in crank pin end. Bolt Pretension force applied on beam element of 32500N to equalize the bolt tightening torque. Moreover, the crankpin and piston pin ends are applied with the temperatures of 150°C and 300°C respectively and the FE analysis is carried out.

Results & Discussions

The following conclusions can be drawn from the analysis conducted during this project:
1. Dynamic loading analysis of the crankshaft results in more realistic stresses whereas static analysis provides overestimated results. Accurate stresses are critical input to optimization of the components. 2. There are two different load sources in an engine; inertia and combustion. These two load sources cause both bending and torsional load on the crankshaft. The maximum load occurs at the crank angle of 360 degrees for this specific engine. At this angle only bending load is applied to the crankshaft. 3. Torsional force is maximum when crank is at 25^{o} from the top dead centre. Critical (i.e. failure) locations on the crankshaft geometry are all located on the pin fillet and main fillet because of high stress gradients in these locations, which result in high stress concentration factors. The Finite Element Analysis is conducted on the crankshaft shows more stresses in the fillets and in the pin journal oil holes. Although edges of these sections are filleted in order to decrease the stress level, these fillet areas are highly stressed locations over the geometry of crankshaft. Therefore stresses were traced over these areas.

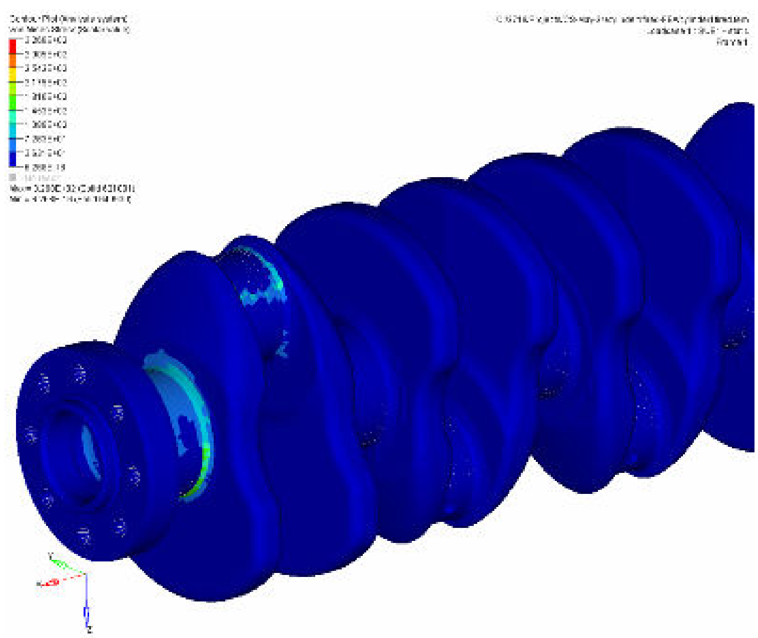

FIGURE.8: VONMISES STRESS CONTOUR PLOT OF CRANKSHAFT

The maximum vonmises stress acting on the crankshaft is 326Mpa by taking both maximum torsional and bending together as shown in the figure.9, where as the yield strength of the material of crankshaft is 584Mpa and FOS is coming about 1.76. The maximum vonmises stress acting on the connecting rod is 413 Mpa and yield strength of connecting rod material is 500Mpa. The FOS of this case is coming about 1.21. Figures 10, shows that the vonmises stress results of the connecting rod for the load first load case. Since the FOS in both the cases is greater than 1, we can conclude that the modified design is under safer working condition.

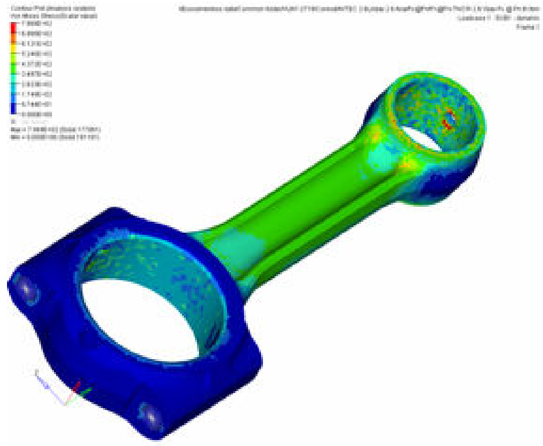

FIGURE.9: VONMISES STRESS CONTOUR PLOT OF CONNECTING ROD

Benefits Summary

New product development time is getting reduced by taking the advantage of existing design of the components and necessary modifications are carried out. All the possible load cases are analyzed by CAE, since there is no physical part available. This drastically saved the development time and testing material. Various load patterns are applied over the parts and the results are validated by the available data. The FE analysis gives a real insight over the critical stress areas by which it leads to carry out optimization of components in near future. As a whole, this analysis gives a clear picture about the stresses and displacements developed during real time working condition of the components.

Challenges

Due to the complexity in geometry of connecting rod, crankshaft, cylinder block the meshing part took most of the time. It was decided that, the FE analysis is to be carried out by using 3D tetrahedral elements; even these elements are stiffer than the hexahedral elements. These hexahedral elements took very long time to solve the problem and since it is essential to save the CPU running time in other words cost of the analysis. In case of cylinder block, because of its complexity, the geometry cleanup became a tedious operation and it took very long time. The process of conversion of 2D tria elements to 3D tetrahedral elements was carried out by doing number of iterations of meshing. During this iteration process, the element quality check is done and the neceaasry modifications are carried out accordingly. If auto meshing of parts with 3D hexahedral elements/brick meshing are available, then it will be helpful to get good representation of FE model and it will give more accurate results, instead of going by manual meshing.

Future Plans

The interpretation of FE results showed that the parts are under safe working condition, hence there is a scope for the optimization of these components without compromising the FOS as 1. The optimization might be design changes of components other than in critical areas which are indentified by FE analysis. The reduction in cost by changing the material of the part can also be taken into account. In case of connecting rod, the peak stresses occurred mostly in the transition area between the small end, big end and shank portion. The value of stress at the middle of the shank portion is well below the allowable limit. Since forces at the pin end are lower in comparison to the forces at the crank end, the strength of the pin end region should ideally be lower, in comparison to the strength at the crank end region for optimum material utilization. The choice of different locations will definitely show a different picture in terms of available scope for weight reduction. Stresses at these locations still give a general idea of the scope and direction for optimization. While performing optimization, the stresses at all the nodes are taken into account rather than stresses at just a few locations.

Conclusions

This project investigated design and FE analysis of 2.6L gasoline engine critical components from the base design of in-house existing engine. The cost advantage of existing proven design is used instead of going for a new development process. First of all, the critical parts are digitized by 3D modeling software. Load analysis was performed based on the input from proposed engine performance data, which comprised of the power output, maximum torque delivered, crank radius, piston diameter, the piston assembly mass, and the pressure-crank angle diagram, using analytical techniques by considering the assembly as a slider crank mechanism. The boundary conditions of the connecting rod and crankshaft are applied over the respective FE model which was created by using HyperMesh and the processing part was done by OptiStruct. In case of cylinder block, the preprocessing part was finished and the boundary conditions (temperature during combustion) are indentified, the FE analysis part will be carried out later. The FE results are interpreted with the yield strength of the component material and FOS was calculated. The outcome of this analysis predicts the safer working of the parts under the stated operating conditions.

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