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Evaluating Design of the Automotive Crankshaft for Fatigue Life using Finite Element Method

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***Abstract*—The project is about the study of static and Fatigue analysis of crankshaft. Crankshaft is most complicated and highly strained engine part, which converts the sliding motion of the piston to a rotary motion by slider crank mechanism. Crankshaft is subjected to cyclic bending and torsional loads due to gas pressure and inertial forces. Due to these forces crankshaft is subjected to bending and torsional stresses with high stress concentration at crankpin fillet and journal bearing fillet. There for it is necessary to study the crankshaft for static and Fatigue analysis. The crankshaft Fatigue and Static analyzed is carried out using commercial FEA software.**

Keywords—crankshaft,fatigue,Static,FEA

I. INTRODUCTION

In this project maximum gas force when piston is at TDC condition was taken to manipulate the stresses at critical area i.e. at crankpin and journal bearing fillet on single crank throw.Net force coming on the crankpin is the combination of gas force, centrifugal force and inertia force. This load is applied at upper portion of crankpin, and boundary conditions are applied according to the crankshaft mounting conditions. These stress induce high bending and torsional stress mainly concentrating at crankpin and journal bearing carried out using commercial software Creo Parametric 2.0, Hypermesh,MSC fatigue etc.

II. ENGINE SPECIFICATION

Type of engine	Air-cooled, 4-stroke single cylinder,
Bore(mm)	50.000
Stroke(mm)	49.500
Bore to stroke ratio	0.311
Compression ratio	9.0 : 1
Engine capacity	1.2 L
Maximum Power(Kw)	5.5 Kw @ 8000 rpm
Maximum torque(N-m)	7.95 N-m @ 5000 rpm.
Displacement(cc)	97.63 cc

III. CRANKSHAFT MATERIAL AND MANUFACTURING PROCESS

The crankshafts are subjected to shock and fatigue loads. Thus material of the crankshaft should be tough and fatigue resistant. The crankshafts are generally made of carbon steel, special steel or special cast iron.

In industrial engines, the crankshafts are commonly made from carbon steel and manganese steel such as 42CrMo4 are generally used for the making of crankshaft. The crankshafts are made by drop forging or casting process but the former method is more common hardening. The surface of the crankpin is hardened by case quenching and tempering or suitable heat treatment.

Material specification for 42CrMo4, equivalent grade Steel 4140, Q&T, BHN=293, Material Specification AISI 4130 (Material Reference SAE J1099 - June 1998)

IV. METHODOLOGY

- Pre processing
  Geometry creation using Creo Parametric 2.0
  Creation of Mesh model using Hyper mesh 11.0
  Applications of Field/Boundary conditions
  Assembling the system equations

- Solution
  Solution for the system equations using Nastran 5.0 for Static analysis
  For Fatigue Solution MSC fatigue Software is used

- Post processing
  Result of static analysis and Fatigue analysis using Hyper view, MSC fatigue etc.

V. MODEL CREATION USING CREO PARAMETRIC

Assembly of crankshaft includes crankpin; left web and right web (transmission side).The dimension are used from existing design.

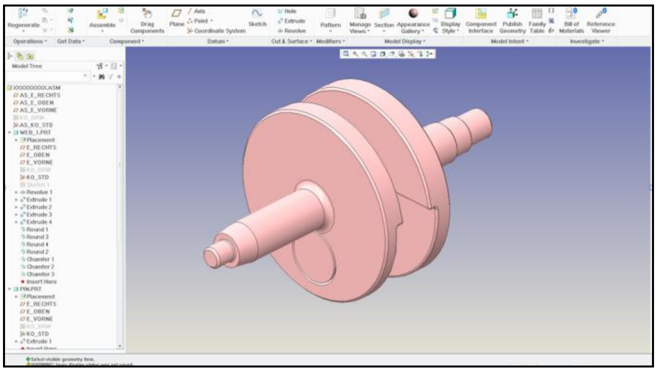

Fig. 1. Crankshaft 3D model using Creo Parametric.

A. *FEA MODEL*

For the analysis of crankshaft mesh generation on the geometry of crankshaft is performed in Hyper mesh 11. Mesh generation for crankshaft is done with tetrahedral second order elements. Where the mesh size on the critical areas is between 0.25mm to 2mm and for non-critical areas it is above 2 mm up to 4.8mm. This meshing of the system with second order tetra elements discretized the physical continuum into 63800 elements with 85494 nodes.

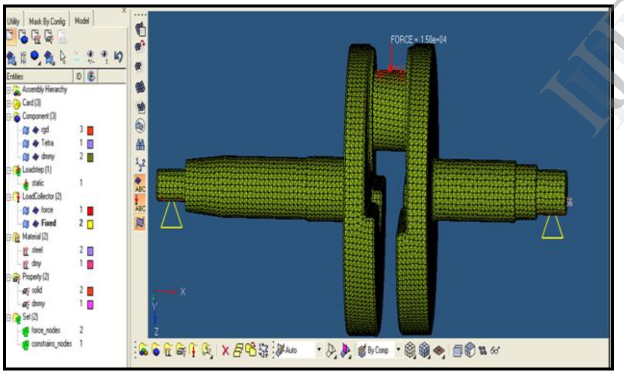

Fig. 2. Crankshaft Mesh genaration using Hypermesh

B. *Boundry conditions*

Boundary conditions represent parts and an effect that is not or cannot be explicitly modeled makes tremendous assumptions that the effect of these un-modeled entities can truly be simulated .The bearing support at both ends is kept as fixed. The crankpin supposed to have load of 15KN Compressive acting downward direction. This is assumed to be at TDC of piston.

In this case maximum gas force is applied on the top 180 degree portion of crankpin. Bottom half portion of journal bearing shaft is fixed as a constrain.

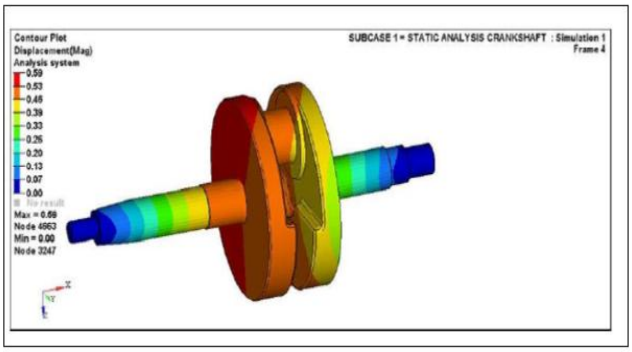

Fig. 3. Displacement Plot of crankshaft

Description: The Displacement of crankshaft is observed in the range of 0.00-0.59 mm.The maximum Displacement is 0.59 mm and Minimum Displacement is 0.00 mm

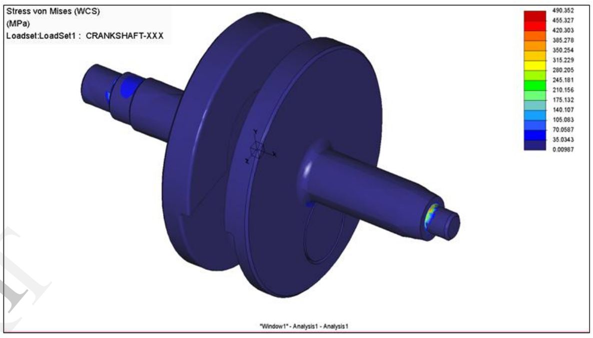

Fig. 4. Von misses Stress for crankshaft of crankshaft

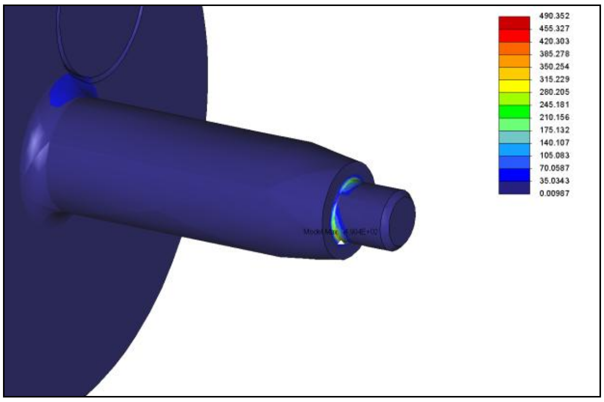

Fig. 5. Von misses Stress at critical location of crankshaft

Description: The Von Misses stress of component are observed in the range of the 0.00987 N/mm^2 to 490.352 N/mm^2.The maximum stress value observed 490.352N/mm^2 at bearing fillet location as shown in Figure 5.

C. *Fatigue analysis*

Various methods are available and some are developed by researchers. Three methods which are used to predict fatigue life include stress life(S-N), strain Life (E-N) and Linear Elastic Fracture Mechanics (LEFM). S-N method is based on nominal stress life using rain flow cycle counting. This method can be helpful to test fatigue life but only disadvantage is that plasticity effect is not considered and provides poor accuracy for low cycle fatigue (Solid work Corporation 2005).The strain life method provides more detailed analysis involving plastic deformation at a localized region and is useful for low cycle fatigue. Linear elastic fracture mechanics assume that crack is already available and detected and predicts crack growth by considering stress intensity factor.

Because of easy implementation and various data available, stress life method is most commonly used where S-N curve data for particular material is commonly available. A new method called, cracks modeling, is developed by (Taylor and Ciepalowicz, 1997) to predict fatigue failure of crankshaft. This technique uses a Liner Elastic Finite Element Analysis to derive stress intensity factor for component under load. In this method, stress intensity factor is calculated without introducing a crack into a component; stress field around the maximum stress point is examined and compared to that of standard center cracked plate. The material property is threshold stress intensity range. This research predicted the failure of crankshaft under constant amplitude loading. Failure occurs from stress concentration whose location is varied with loading type basically due to bending and torsion In order to describe the severity of stress natural frequency, bearing load and axial displacement which is required for fatigue life estimation

Figure 6 shows the conventional fatigue life estimation procedure in which geometry, material properties and mechanical loading are regarded as three input parameters. Initially, the geometry and loading are used together to produce a stress-time *(–t)* or strain-time

(ε –t) history at critical location. Next, the material fatigue properties are introduced for estimating the fatigue life. The only material properties needed in the first step are the Young's modulus, the elastic-plastic stress-strain curve, etc., which are not true fatigue properties. In the present work, at first, the stress and strain at critical location are calculated. Next, finite element method (FEM) is used for converting reduced load-time history into the strain-time history, followed by the stress/strain calculations in the highly stressed (critical) area. Finally, three strain-life theories (Coffin-Manson, Morrow and SWT) are applied for the prediction of fatigue life.

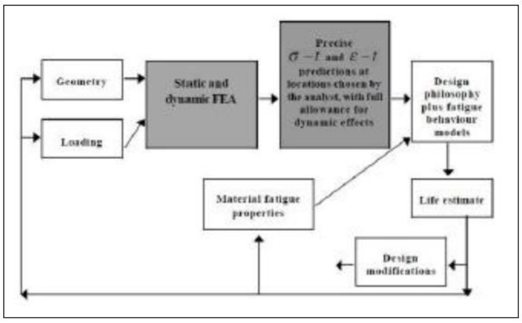

Fig. 6. The conventional fatigue life estimation procedure

The component is analyzed for infinite life cycles for S-N type analysis. The Cyclic loads are fully reversed type 490.352 N/mm^2 from static analysis. The Caffon-Massons theory is used and results are plotted in Sinusoidal pattern. Solver used for fatigue MSC fatigue.

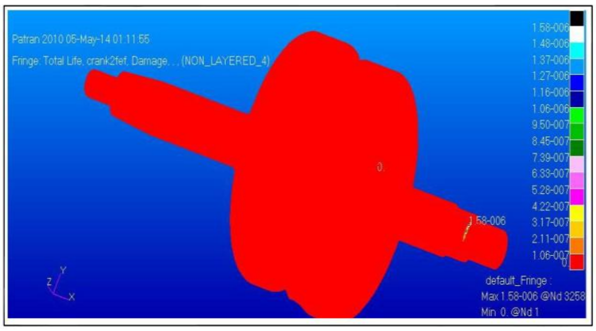

Fig. 7. Critical locations for damages of crankshaft

The Damage of crankshaft ranges from 0 to 1.58x10^{-6} for cycle of infinite life. The overall component damages shows closer to Minimum range whereas at bearing support fillet and critical stressed area shows damages of 1.58x10^{-6}.The Initial crack origination starts at critically stress area shown in above figure. Also the Crack propagation starts at fillet areas of bearing support.

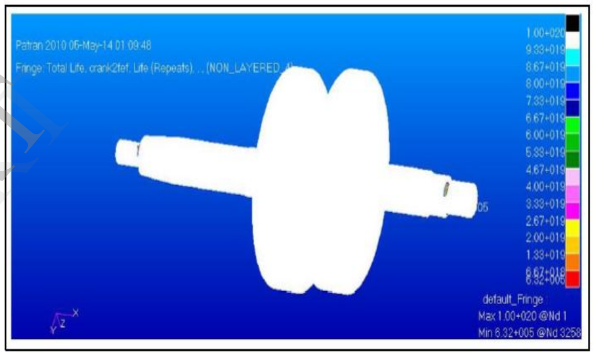

Fig. 8. Fatigue life of crankshaft

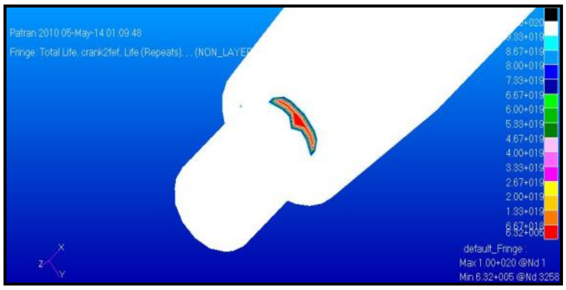

Fig. 9. Fatigue life at critical location of crankshaft

The fatigue life of crankshaft ranges from 6.32x10^5 to 1x10^{20} .The component shows overall fatigue life closer to 1x10^{20}.The area at fillet region of bearing support shows fatigue life 6.32x10^5.The critical locations for fatigue life are located near both bearing support ends where it shows minimum fatigue life of 6.32x10^5

VI. RESULT TABLE

Parameter	FEA
Von Misses Stress(MPa)	490.352
Displacement(mm)	0.59
Fatigue life (in Cycle)	$6.35x10^5$
Damage (mm or cycle/total life)	1.58×10^{-6}

VII. CONCLUSION

The Critical stress areas of component are analyzed .The bearing supports fillets are stressed area in crankshaft. The Damage and fatigue life of crankshaft is within permissible limit and hence Design is Safe. The Crack intimation process will starts at critical fillet areas and propagates to other areas after infinite number of cycle.

The fatigue life and damage are within permissible limits but need to improve. The crankshaft show maximum stress at support ends by help of variable radius or geometry simplification may reduce stress at particular location.

VIII. FUTURE SCOPE

- The change of Crankshaft Fatigue life prediction by the optimization method.
- The Geometry at critical failure and highly stress location can be optiomsed.
- The Material for the crankshaft needs to increase Fatigue endurance limit.
- The Heat treatment like case depth hardened or any other suitable method need to identify.
- The quenching and tempering heat treatment condition need to modify (e.g.temparature, quenching time etc.)
- Geometry optimization may reduce the bending and torsional forces or by mass balancing

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