FE-Analysis of crankshaft of I.C.Engine for increasing the breathing capacity

Mr. Jayesh Ramani[1], Mr. Hardik Ramani [2], Mr. Sumit Suhagiya[3]

Abstract

Crankshaft is a large volume of production for automobile industries. Day by day demand of highly efficient and optimized parts increasing with high rate. This paper presents a study of design work by upgrading 1.8 L four stroke inline cylinder engines. To upgrade engine capacity and to make powerful engine as 2.6 L four stroke engine diameter of crankshaft to be varied & to sustain high dynamic load coming from 2.6 L petrol engine, it needs to be cross checked against bending and torsional by FEA. FEA is performed by applying dynamic loads to crankshaft. FEA over crankshaft conducting stress calculations over both, entire crank & single throw. Boundary conditions apply to check maximum loading conditions for crankshaft. In FEA constraints are applied at Face and Bearing area over the crankshaft. The IC engine for the intended application is upgraded from 1.8 L to 2.6 L to leverage its objective for dispensing better performance as regards the power generated for it application in four wheelers.

Keyword- crankshaft, FEA, petrol engine, optimization

I. Introduction

Crankshaft is a large component with a complex geometry in the engine, which converts the reciprocating displacement of the piston to a rotary motion with a four link mechanism. Since the crankshaft experiences a large number of load cycles during its Service life, fatigue performance and durability of this component has to be considered in the design process. Design developments have always been an important issue in the Crankshaft production industry, in order to manufacture a less expensive component with the minimum weight possible and proper fatigue strength and other functional requirements. These improvements result in lighter and smaller engines with better fuel efficiency and higher power output. 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. The magnitude of the force depends on many factors which consist of crank radius, connecting rod dimensions, weight of the connecting rod, piston, piston rings, and pin. Combustion and inertia forces acting on the Crankshaft cause two types of loading on the crankshaft structure; torsion load and bending load.

There are many sources of failure in the engine. They could be categorized as operating sources, mechanical sources, and repairing sources. One of the most common crankshaft failures is fatigue at the fillet areas due to bending load caused by the combustion. Even with a soft case as journal bearing contact surface, in a crankshaft free of internal flaws one would still expect a bending or torsional fatigue crack to initiate at the pin surface, radius, or at the surface of an oil hole. Due to the crankshaft geometry and engine mechanism, the crankshaft fillet experiences a large stress range during its service life. It can be seen that at the moment of combustion the load from the piston is transmitted to the crankpin, causing a large bending moment on the entire geometry of the crankshaft. At the root of the fillet areas stress concentrations exist and these high stress range locations are the points where cyclic loads could cause fatigue crack initiation, leading to fracture.

II. Finite element analysis of crankshaft

There are two major approaches for stress calculation:

(a) Based on entire crank.
(b) Based on single throw.

The first procedure can be described as follows:

Run full crank reduced model (dynamic) to calculate main bearing reactions and torques. Model entire crankshaft with FEM. Constrain the model at the flywheel end. Run analysis applying all possible loads (at the pin and main bearing locations) (pressure distributed over bearing area) one at a time.
Another approach is published can be described as follows:
Run dynamic analysis on a reduced model.

Cut out one throw of the crank through the main journal middle cross-sections (detailed FE). Constrain one cross-section and apply the forces i.e. bending as well as torsional forces and obtain corresponding stress states. Another approach is to constrain the main bearings for all degree of freedom & applying the bending & torsional force at the crankpin end.

III. Boundary conditions

Static FEA

The crank and piston pin ends are assumed to have a sinusoidal distributed loading over the

contact surface area, under tensile loading. This is based on experimental results .The normal pressure on the contact surface is given by:
P = Po cos φ
The load is distributed over an angle of 180o. The total resultant load is given by:
is given by:

$$P_t = \int_{-\pi/2}^{\pi/2} Po\ (\cos^2 \phi)\ r\ t\ d\phi = = Po\ r\ t\ \pi\ /\ 2$$

Where r is crank pin radius & t is the length of crank pin.
The normal pressure constant *Po* is, therefore, given by:
Po = Pt / (r t π/ 2)
The tensile load acting on the connecting rod, Pt, can be obtained using the expression from the force analysis of the slider cranks mechanism. For compressive loading of the connecting rod, the crank and the piston pin ends are assumed to have a uniformly distributed loading through 120^0 contact surface.
The normal pressure is given by:
p = po
The total resultant load is given by:

$$Pc = \int_{-\pi/3}^{\pi/3} Po\ (\cos\phi)\ r\ t\ d\phi = Po\ r\ t\ \sqrt{3}$$

The normal pressure constant is then given by:
Po = Pc / (r t π3)
Pc can be obtained from the indicator diagram, such as the one shown in Figure-1, of an engine. In this study four finite element models were analyzed. FEA for both tensile and compressive loads were conducted. Two cases were analyzed for each case, one with load applied at the crank end and restrained at the piston pin end, and the other with load applied at the piston pin end and restrained at the crank end. In the analysis carried out, the axial load was 55KN in compression & 11kN in Tensile Loading
Compressive Loading:
Crank pin End: Po = 62.5MPa
Piston pin End: Po = 74MPa
Tensile Loading:
Crank End:
P_o= 11000/ [24 x 17.056 x (π/2)]
= 13.6 MPa

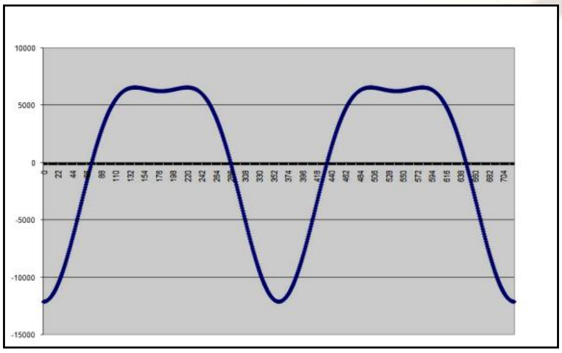

Figure-1 Linear Acceleration of Piston

IV. FEA of Crankshaft

In this FEA analysis, whole crankshaft finite element model is used, applying the bending force acting on the crank of four cylinder inline engine, reaction forces acting on the on all the main journal bearings & torsional moment is applied on each crank separately, The value of Bending & torsional force was obtained from the dynamic simulation of the crankshaft for whole 720^0rotation of the crankshaft as shown in Figure-2. Applying the respective forces on each cylinder as got from the firing order of the cylinders ie. 1-4-3-2.

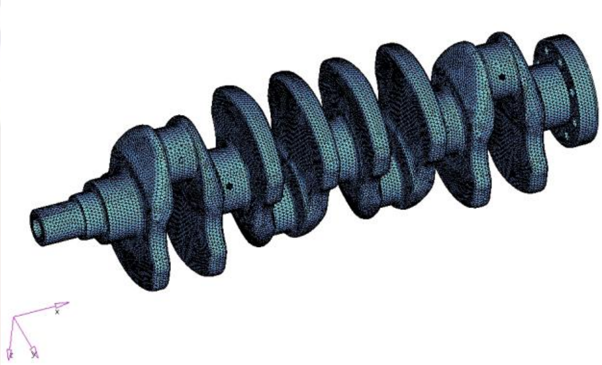

Figure-2 Crankshaft Meshed Geometry

Forces & moment are applied by making rigid element by selecting the nodes of the surface over which forces & moments are applied. The cylinder which is fired have maximum bending compressive force applied over 120^0 on the top surface of crank. All other all tensile force are applied over 180^0 over the surface in considering by same methodology of rigid elements.Torsional moment was obtain by multiplying the torsional force of the respective cylinder & crank throw. (47.5mm)Torsional moment is applied by selecting the node at the periphery where crank web & main journal are in contact of both end of crank & making the rigid with nodes .The FEA for the case of whole crankshaft with constrain at the flywheel end is shown in figure-3.The maximum Bending force applied on the is 55KN, other 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 of the crank having maximum torsional force & other moments on the respective crank as obtain from the graph.

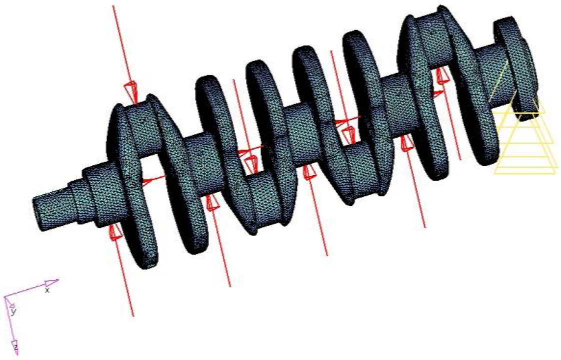

Figure-3 FEA of whole Crankshaft with Constrain at Flywheel End

Cases for analysis, when maximum bending force acting on crank, maximum torsional moment acting on cylinder .when maximum bending acts on cylinder torsional moment is zero, whereas when maximum torsion force act there was both bending as well as torsional force .

V. FEA of the Single Crank of the Crankshaft Constrain at Face.

In this FEA Analysis, Cut out one throw of the crank through the main journal middle cross-sections. Constrain one cross section for all Degree 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. The FEA model of Single crank is shown in figure-4 & figure-5

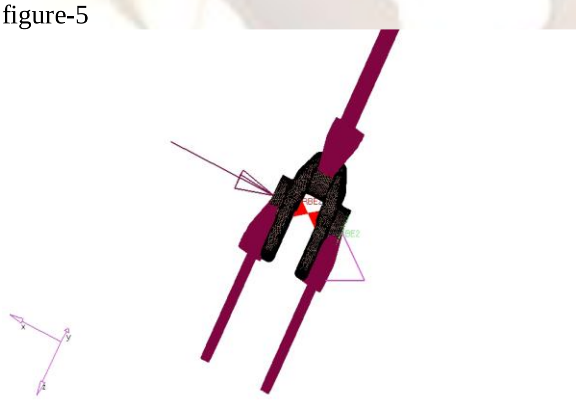

Figure-4 FEA of the single Crank Constrain at Face

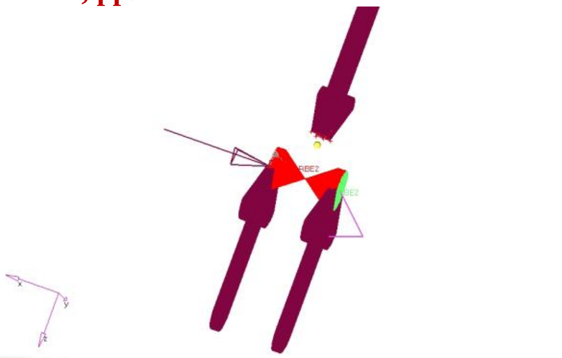

Figure-5 FEA of the single Crank Constrain at Face

VI. FEA of Single Crank Constrain at bearing Area

In this analysis constrain the crank in its bearing area for all degree of freedom by forming the rigids & applying the bending as well as torsional force as a pressure load in the crank pin. The FEA model is shown in figure-6

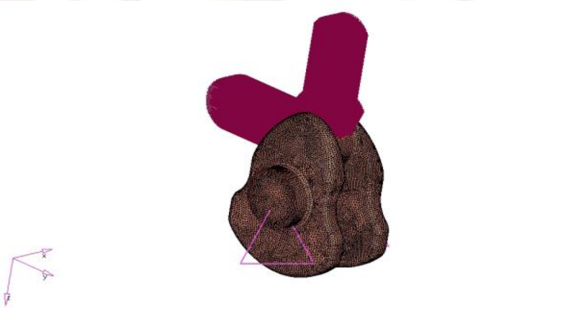

Figure-6 FEA of the single Crank Constrain at Bearing Area

VII. Finite Element Analysis Results and Discussion

The Finite Element Analysis is conducted on the crankshaft shows more stresses in the fillets and in the pin journal oil holes. Section changes in the crankshaft geometry result in stress concentrations at intersections where different sections connect together. Although edges of these sections are filleted in order to decrease the stress level, these fillet areas are highly stresses locations over the geometry of crankshaft. Therefore stresses were traced over these areas.

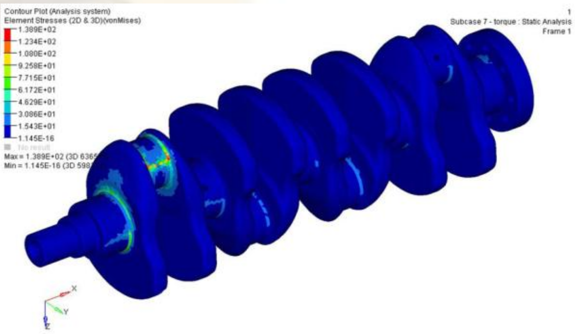

Figure-7 Stress Contour for Bending Load

(Cylinder 1 is Fired)

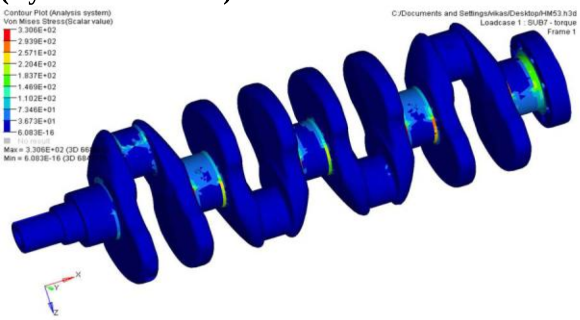

Figure-8 Stress Contour for Torsional Load (Cylinder 1 is fired)

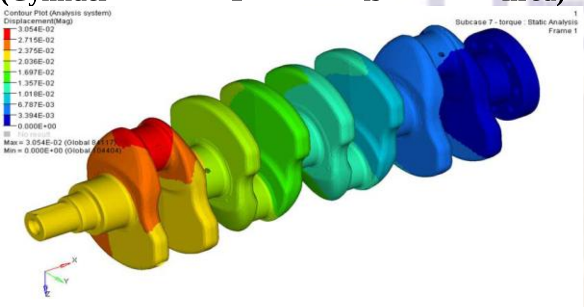

Figure-9 Displacement Contours

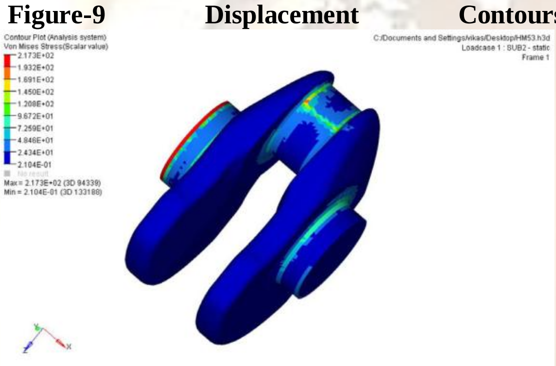

Figure-10 Stress Contour for Bending Load

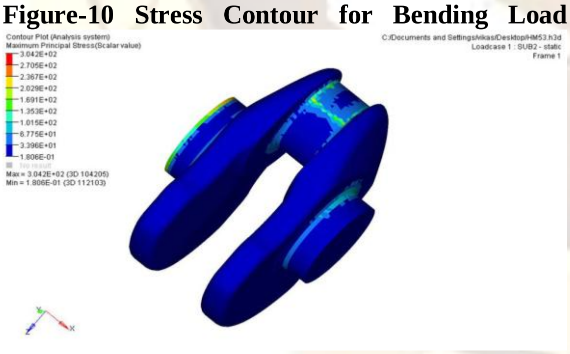

Figure-11 Stress Contour for Torsional Load

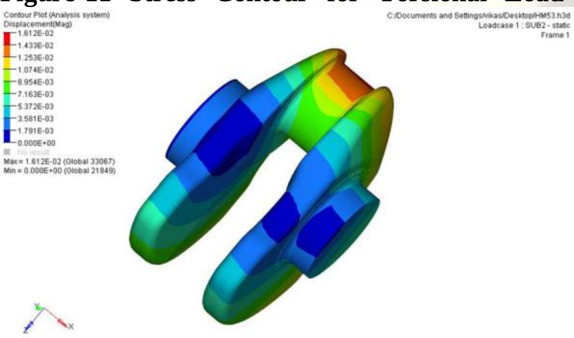

Figure-12 Displacement Contours

The maximum Bending stress acting on the crankshaft is 330Mpa by taking both maximum torsional & bending together. Yield strength of the material of crankshaft is 584. FOS is coming about 1.76.

VIII. Conclusion

The IC engine for the intended application is upgraded from 1.8 L to 2.6 L to leverage its objective for dispensing better performance as regards the power generated for it application in four wheelers.

FEA is performed by applying dynamic loads to crankshaft. The following conclusions can be drawn from the analysis conducted in this study:

1. Dynamic loading analysis of the crankshaft results in more realistic stresses whereas static analysis provides overestimated results. Accurate stresses are critical input to fatigue analysis and optimization of the crankshaft.
2. There are two different load sources in an engine; inertia and combustion. These two load source 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 25^0 from the top dead centre.
4. Critical (i.e. failure) locations on the crankshaft geometry are all located on the Pin fillet & main fillet because of high stress gradients in these locations, which result in high stress concentration factors.
5. Maximum Stress acting on the crankshaft is 330Mpa approx by taking both maximum torsional & bending force together, Factor of safety of approx 1.76.

IX. Future Scope of Work

The critical components are modeled and analyzed in FEA software to evaluate to maximum stresses and the points and conditions for failure of the same. But to design the critical components the designer should bear in mind that considerable heat is generated during the working of the critical components like the crankshaft and the connecting rod. In addition, the dynamic analysis should also be carried out to compute the natural frequencies of the crankshaft to estimate the vibration characteristics of the same. Moreover the critical components are designed taking into consideration the properties of one particular material stainless steel. The design of the critical components can also be accomplished by considering materials apart from the one considered in this dissertation.

References

[1] H.Bayrakceken, S. Tasgetiren, F. Aksoy, Failures of single cylinder diesel engines crank shafts, Afyon Kocatepe University, Technical Education Faculty, Afyon,

Turkey, Engineering Failure Analysis 14 (2007) 725–730, Available online 19 May 2006.

[2] Zhiwei Yu, Xiaolei Xu, Failure analysis of a diesel engine crankshaft, Engineering Failure Analysis vol-12, Issue 11 November 2004, Institute of Metal and Technology, Dalian Maritime University, Dalian, 116026, PR China

[3] F.S. Silva, Analysis of a vehicle crankshaft failure, Engineering Failure Analysis vol10, Issue 5,October 2003, Department of Mechanical Engineering, University of Minho, Azure'm, 4800-058 Guimara'es, Portugal.

[4] Paul Spiteri, Simon Ho, Yung-Li Lee, Assessment of bending fatigue limit for crankshaft sections with inclusion of residual stresses, International Journal of Fatigue 29 Issue 5 May 2006, Stress Lab and Durability Development, DaimlerChrysler, CIMS 484-05-20, 800 Chrysler Drive, Auburn Hills, MI 48326, United States, Powertrain CAE, DaimlerChrysler, United States

[5] Osman Asi, Failure analysis of a crankshaft made from ductile cast iron, Engineering Failure Analysis vol13,Available online 7 February 2006, Department of Mechanical Engineering, Usak Engineering Faculty, Afyon Kocatepe University, 64300 Usak, Turkey

[6] M. Zorufi & A. Fatemi, A Literature review on durability evaluation of crankshafts including comparison of competing manufacturing processes & cost analysis. The University of Toledo, Toledo Ohio.

[7] Zissimos P.Mourelatos, A Crankshaft system Model for structural Dynamic Analysis of Internal Combustion Engine.Computers & structures vol79 issue 6 june 2001, Vehicle Analysis & dynamics lab, General Motors research & development & planning.