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Diesel Engine Crankshaft High Cycle Fatigue Life Estimation and Improvement Through FEA

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Abstract- **In an internal combustion engine, crankshaft is one of the most critical components. The main objective of this paper is to eliminate fatigue failure and improve fatigue life of crankshaft. FEA is carried out in ANSYS with the actual boundary conditions to simulate the physics. In this study a detailed stress analysis of crankshaft of single cylinder diesel engine is done. This paper describes the exact nature and magnitude of stresses at crankpin fillet and journal fillet during crankshaft operation. This paper also describes theoretical stress at critical locations and nature of stresses on crankshaft during engine operating condition and how this can be simulated in FEA. Despite huge amount of papers written on crankshaft stress analysis, detailed nature of stress and procedure to simulate the stress condition in FEA is lacking in literature. This paper also describes the possible modification in crankshaft to increase the fatigue life without modifying other interacting components of crankshaft.**

Keywords — crankshaft failure; fatigue life; FEA; PRO-E; ANSYS.

I. INTRODUCTION

Crankshaft is one of the most important components of internal combustion engine. It is a long component with complex geometry consisting of cylinder bearings, and plates with crank web. Geometry section in crankshaft causes stress concentration at fillet areas where bearings are connected to crank web. The crankshaft converts the reciprocating displacement of piston to a rotary motion with a four link mechanism. The most of crankshaft load is applied to perpendicular to its rotational axis, and reaction forces are thus transmitted through the rod and main bearing, thus the force acts as a bending load. However in addition to this load, torsional load also acts on crankshaft along its axis of rotation. Compared to bending load, twisting load is very small, and does not have much impact on stress analysis. Angled drilling in crankshaft carry pressurized lubricant from main bearing to each of the rod bearing. To avoid stress concentration these holes are chamfered at the end and sharp edges are broken that might damage the bearings [1].

A. Crankshaft Fillet Development

Design of a durable crankshaft starts with a single throw and the alternating stress across the web between the rod and main fillets. Fig. 1 shows loading when crankshaft is at TDC during firing of a particular cylinder, the top of the rod throw is loaded by force transmitted from the connecting rod, and reaction force acting at the bottom of main bearing is transferred into the main bearing cap. The resulting crankshaft deflection places the rod fillet in tension and the main fillet in compression as shown in Fig. 1. After one revolution of crankshaft, loading at TDC is shown in Fig 2. The inertia forces at piston assembly now predominate and connecting rod cap loads the underside of the rod bearing surface. The reaction force is now transmitted into the block along the upper surface of the crankshaft main bearing. This place the rod bearing fillet in compression and the main bearing fillet in tension as shown in Fig. 2. In summary each operation cycle of the engine creates an alternating stress component of simple bending across the crankshaft web. High cycle fatigue due to this bending mode is thus primary failure mechanism. For crankshaft durability, material properties and dimensions are most important. As the fillet radii are increased, the peak stress magnitude decreases, but bearing width decreases as well, in some cases the radii are undercut in order to increase the radii and simultaneously maintain the bearings area. As the crankshaft is being developed, it is important to determine the stress profile across the fillet radius and to identify the location of maximum stress along the radius [1].

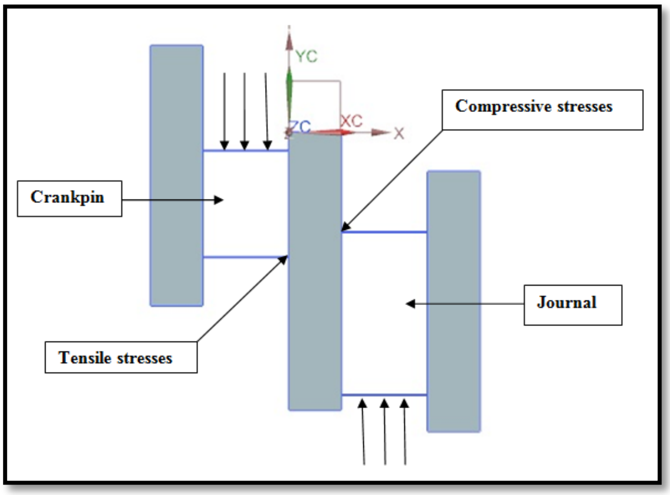

Fig. 1 Nature of stresses on fillet under compressive loading

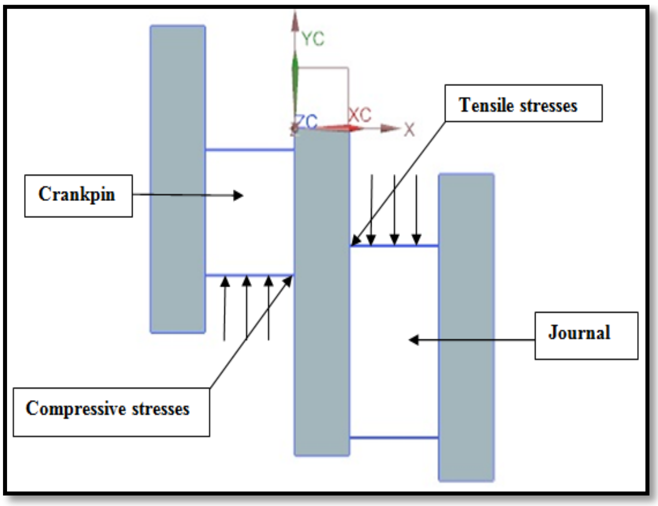

Fig. 2 Nature of stresses on fillet under tensile loading

II. LITERATURE SURVEY

Failure of the crankshaft makes an engine unworkable, resulting into costly procurement and replacement. An extensive research in the past shows crankshaft is subjected to multi-axial loading (Bending and Torsion), stress gradient and stress concentration and effect of variable amplitude loading. Effect of twisting load on the specified crankshaft is far less than the bending load, hence only bending load is considered for analysis for most of the crankshafts.

R. Metkar [2] et al described finite element method as the most favorite method to solve stress and fatigue analysis and it is commonly used for analyzing engineering problems. He also studied stress life, strain life and LEFM methods to solve fatigue analysis. There are lot of softwares available for use in finite element analysis applications, such as, ANSYS, Abaqus, Nastran, and MSC.

F. Monte [3] et al studied two crankshafts of same material. The analysis was done to predict root cause of failure of shaft, both crankshafts were failed due to fatigue but crack on first crankshaft was at fillet and on second crankshaft was at oil hole.

Amardip Jadhav [4] et al studied the marine engine crankshaft and analyzed fatigue failure. They also performed experiment for evaluation of fatigue failure. Crack initiation due to the elliptical arrest lines, which comes in the plane or area between the crankpin and main journal, LEFM technique is used. Failure is not due to wear but due to significant magnitude of bending and torsional load.

Farzin H. Montazersadgh [5] et al suggested the modifications for improvement in fatigue life such as changing the main bearing radius, crank pin radius, fillet radius or main journal pin fillet and changing the type of material in crankshaft, which are very common modifications usually done in crankshaft geometry.

Amitpal Singh Punewale and Amit Chaudhari [6] et al studied torsional vibration with modified crankshaft geometry by adding some material at face inclination on crankshaft and found out improvement in results as compared to original geometry of crankshaft.

Vijaykumar Khansis [7] et al studied stress, balancing and fatigue life of crankshaft. They modified geometry by adding very small amount of material at bevel section of crankshaft. Maximum stresses generated at bevel section in original crankshaft were minimized in modified geometry.

Also none of the authors showed nature of stresses generated in crankpin and journal fillet explicitly, which is described in this paper.

III. GEOMETRY OF CRANKSHAFT

As per design dimensions, model has been created. Modeling of crankshaft is done using the PRO-E modeling software. Fig 3 shows crankshaft geometry with gear at one side and PTO at other side.

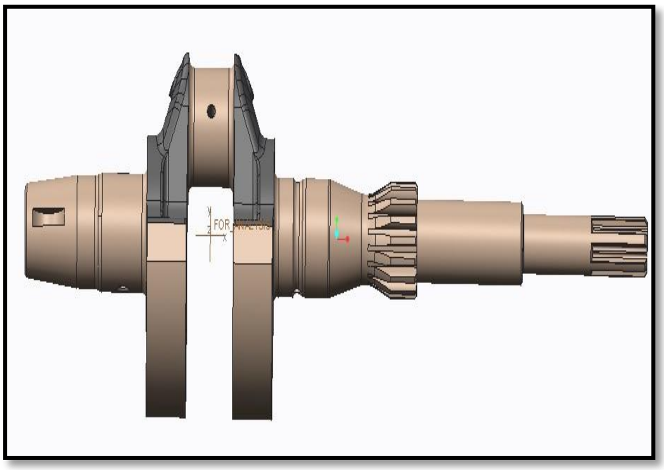

Fig. 3 Crankshaft geometry

IV. LOADING CONDITIONS

A. Force due to maximum gas pressure

After expansion of gas, force generated due to maximum pressure in cylinder is applied to crankshaft. The slider-crank mechanism converts the maximum gas pressure into vertical force which is applied on the piston head and then transmitted to the joint between crankshaft and connecting rod [5].

F_g = (Maximum Gas Pressure) × (C/s Area of Piston)

$$F_g = P_{max} \times \frac{\pi D^2}{4} \qquad (4.1)$$

$F_g = 39499.8$ N

B. Force due to maximum inertia force

Because of rotating as well as reciprocating components (e.g. connecting rod) the crankshaft is subjected to inertia force and this force increases with the increase of engine speed. Inertia force is a function of rotating speed and acceleration of rotating components [5].

F_i = (mass of Reciprocating components i.e. piston assembly.) × (Acceleration)

$$F_i = M_r \omega^2 R \left(\cos\theta + \frac{r\cos 2\theta}{L}\right) \qquad (4.2)$$

$F_i = 6112.34$N

V. FINITE ELEMENT ANALYSIS OF CRANKSHAFT

The FE model of the crankshaft geometry is meshed with tetrahedral elements. Mesh refinement are done on the crank pin fillet and journal fillet, so that fine mesh is obtained on fillet areas, which are generally critical locations on crankshaft [8]. The meshed crankshaft is shown in Figure 4.

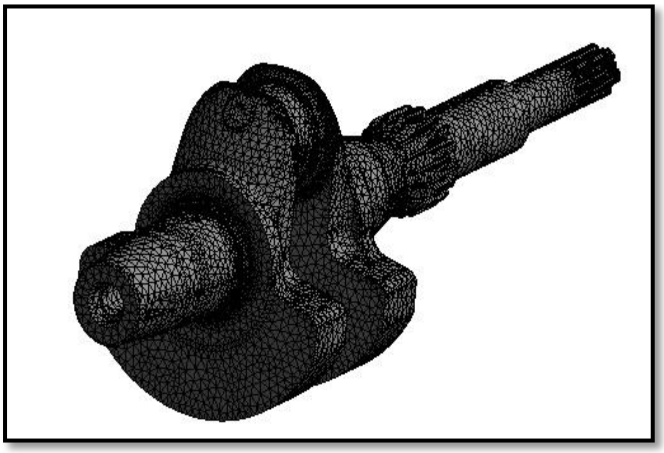

Fig. 4 Mesh model of crankshaft geometry.

A. Boundary condition applied on crankshaft geometry

As alternating loads are acting on crankshaft which differ in magnitude as well as direction, so loading conditions are very difficult task for analysis. For fatigue analysis, combination of two solutions (compressive and tensile loading) is most suitable and simplest way to obtain the stresses and life. In FEA, boundary conditions are applied on crankshaft are based on engine configuration and component mounting conditions. This crankshaft is mounted on same type of bearings but length of bearing is different [8]. Table I shows the different boundary conditions used to simulate FEA and corresponding results, but out of these conditions force and remote displacement support gives good results. Fig 5 shows application of maximum compressive force and constraints on the bearing area. Fig 6 shows application of maximum tensile force and constraints on the bearing area.

Table-I: Different boundary conditions applied on geometry and corresponding results

Sr. No.	**Boundary conditions**	**Results**
1	Force at crank end and Fixed support at bearing.	Nature of stresses are incorrect.
2	Force at crank end and frictionless support (constraint in vertical direction) at bearing.	Nature of stresses are incorrect.
3	Force at crank end and displacement support (constraints Y and Z direction, X direction is free to deform) at bearing.	Nature of stresses are incorrect.
4	Force at crank end and remote displacement support (constraints Y and Z direction, X direction is free to deform, Rotation is free along X axis) at bearing.	Nature of stresses are correct at both fillets.

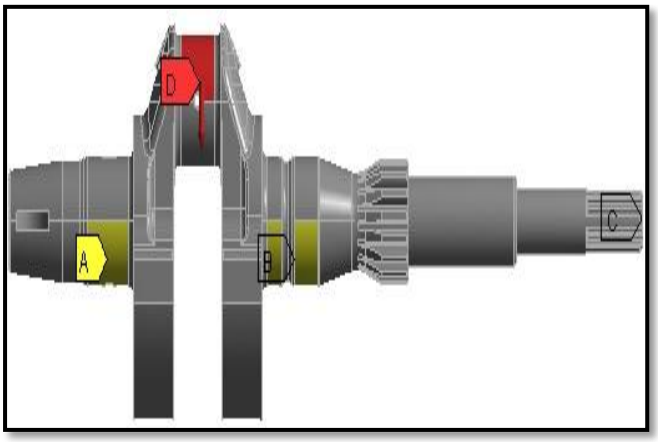

Fig. 5 Compressive loading

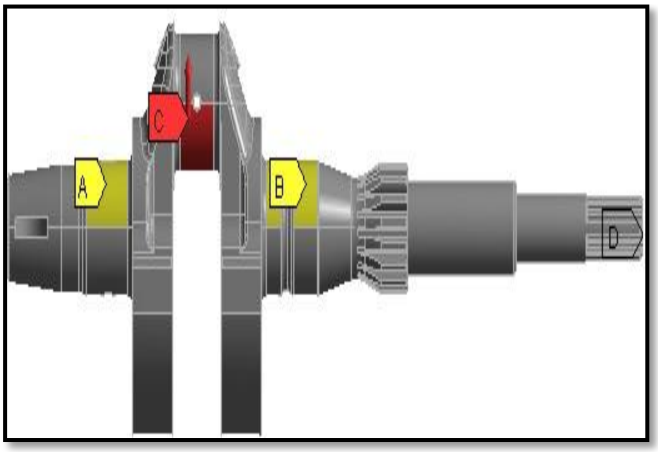

Fig. 6 Tensile loading

B. Results of Stress analysis

Crankpin fillet, journal fillet, oil hole region is considered for documentation. According to loading conditions applied on crankshaft geometry, results shows maximum and minimum stress locations [9]. Fig 7 shows normal stress of 484 MPa generated at crankpin fillet for compressive loading and Fig 10 shows 64 MPa compressive stress generated at crank pin fillet for tensile loading. Fig 11 shows equivalent stress plot for combustion loading. Fig 8 and 9 shows maximum and minimum stress locations at fillet.

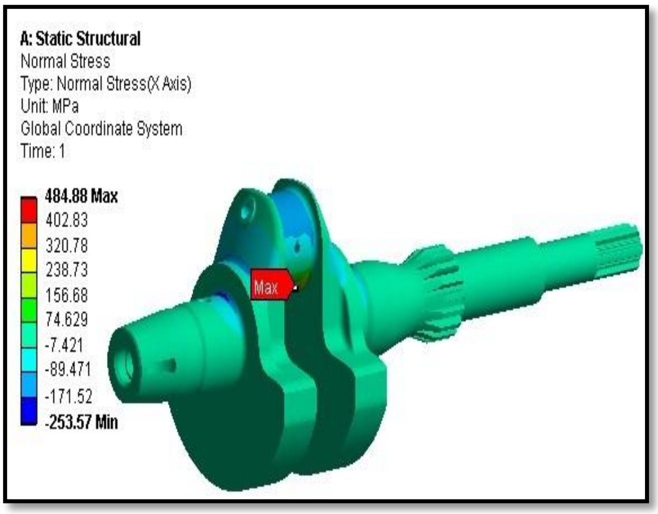

Fig. 7 Normal stress for compressive loading

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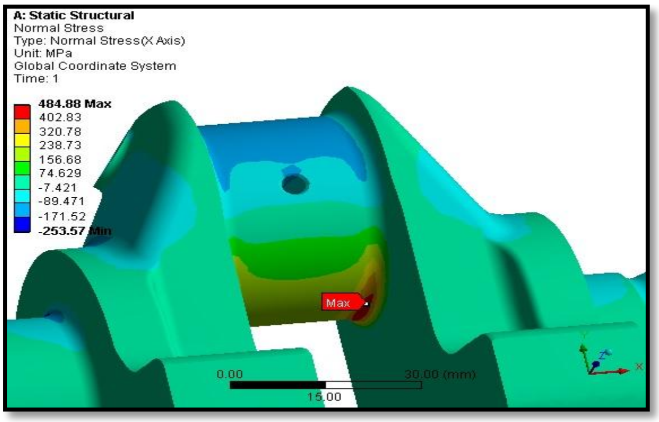

Fig. 8 Maximum stress location at crankpin fillet

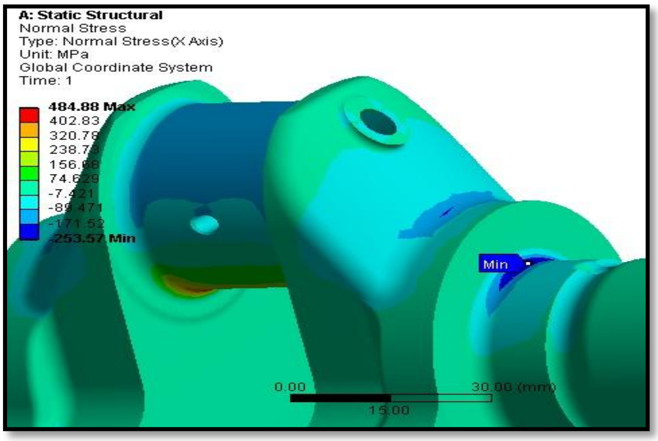

Fig. 9 Minimum stress location at journal fillet

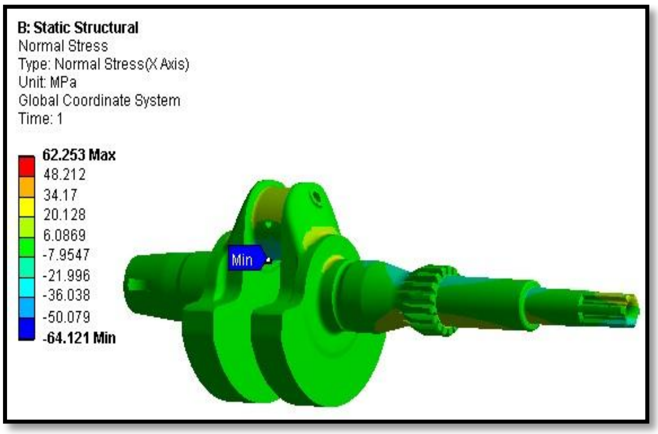

Fig. 10 Normal stress for tensile loading

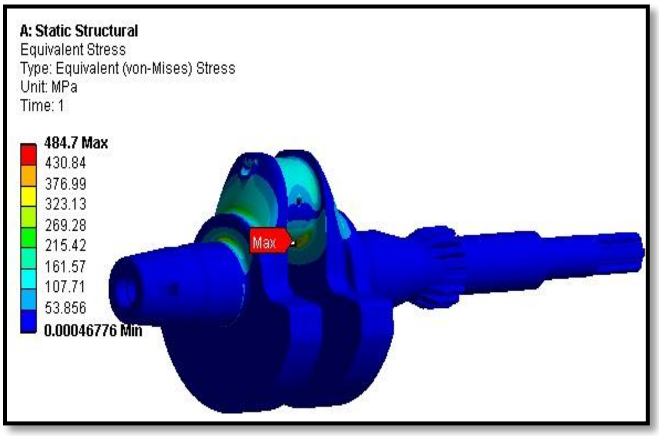

Fig. 11 Equivalent stress for compressive loading

C. Variation of stress along the geometry

Few graphs are plotted according to stress analysis. Fig. 12 shows the normal stress variation according to the angle on crankpin fillet diameter, maximum stress is approximately at 180^0 from the top and it is tensile in nature while at the same point it is compressive in nature when load is tensile. Fig 13 shows variation of stress along the y axis. Fig 14 shows variation of stress along x axis, it shows that there is sudden increment of stress at oil hole, which is located at a radial distance of 23mm thus indicating there is stress concentration near the oil hole. Also same variations are shown for journal fillet. From these figures it can be concluded that maximum stress is at outer layer of crankpin fillet, hence fatigue crack will originate from outer surface of fillet only.

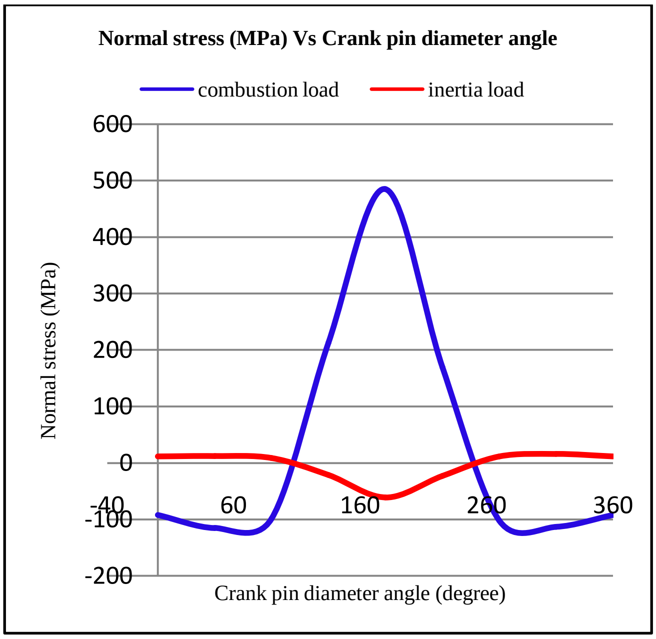

Fig. 12 Graph between Normal stress Vs Angle on Crank pin fillet

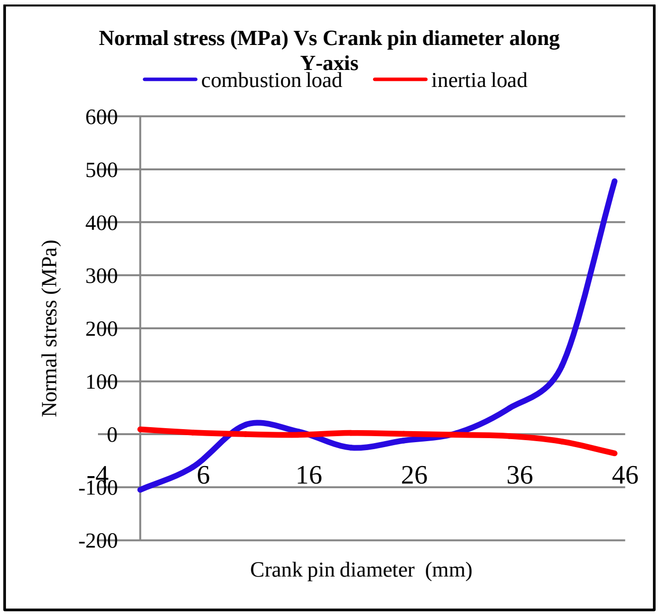

Fig. 13 Graph between Normal stress Vs Crank pin diameter fillet along Y-axis

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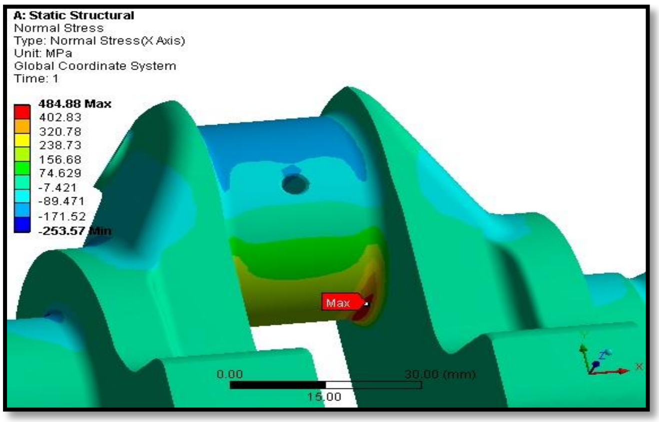

Fig. 8 Maximum stress location at crankpin fillet

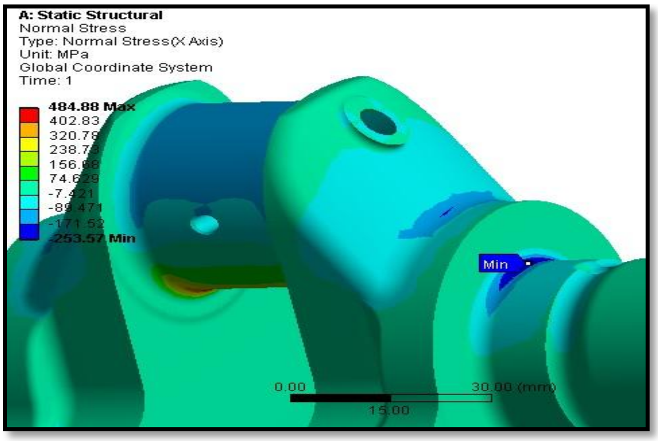

Fig. 9 Minimum stress location at journal fillet

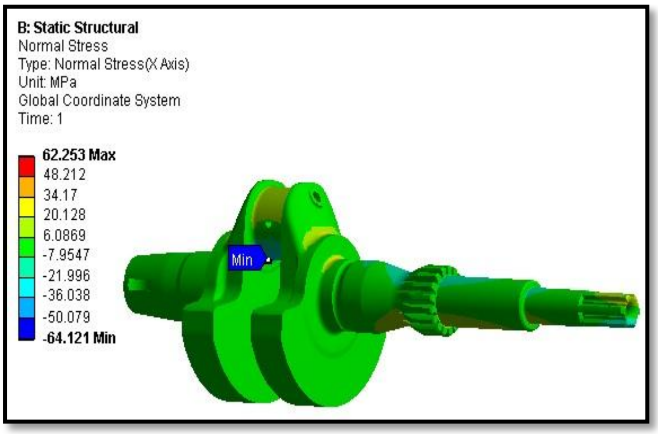

Fig. 10 Normal stress for tensile loading

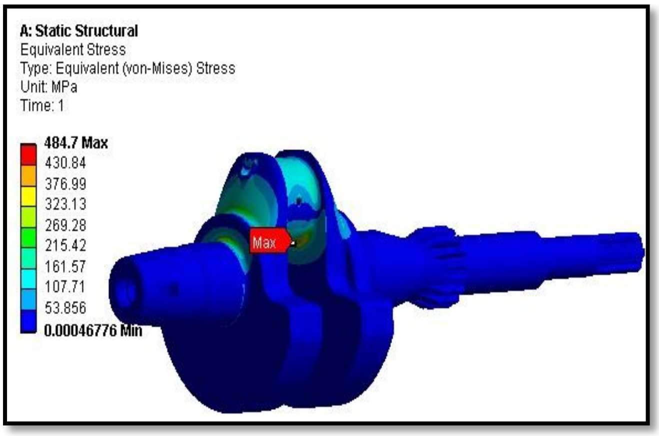

Fig. 11 Equivalent stress for compressive loading

C. Variation of stress along the geometry

Few graphs are plotted according to stress analysis. Fig. 12 shows the normal stress variation according to the angle on crankpin fillet diameter, maximum stress is approximately at 180^0 from the top and it is tensile in nature while at the same point it is compressive in nature when load is tensile. Fig 13 shows variation of stress along the y axis. Fig 14 shows variation of stress along x axis, it shows that there is sudden increment of stress at oil hole, which is located at a radial distance of 23mm thus indicating there is stress concentration near the oil hole. Also same variations are shown for journal fillet. From these figures it can be concluded that maximum stress is at outer layer of crankpin fillet, hence fatigue crack will originate from outer surface of fillet only.

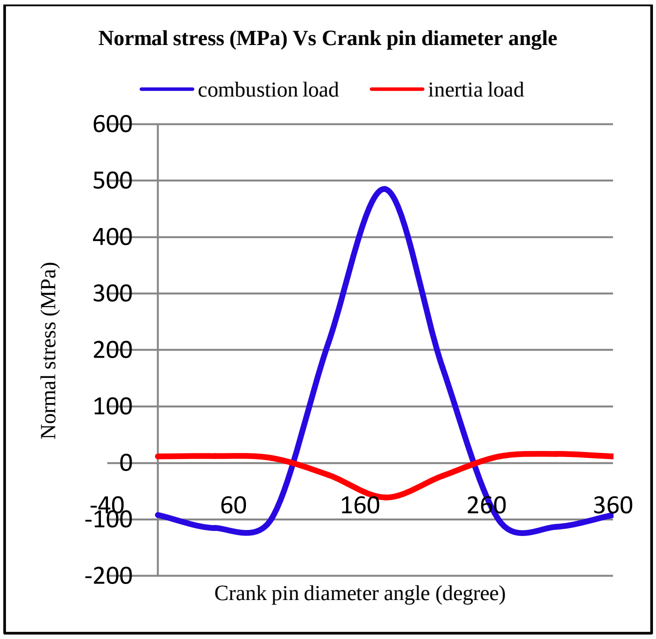

Fig. 12 Graph between Normal stress Vs Angle on Crank pin fillet

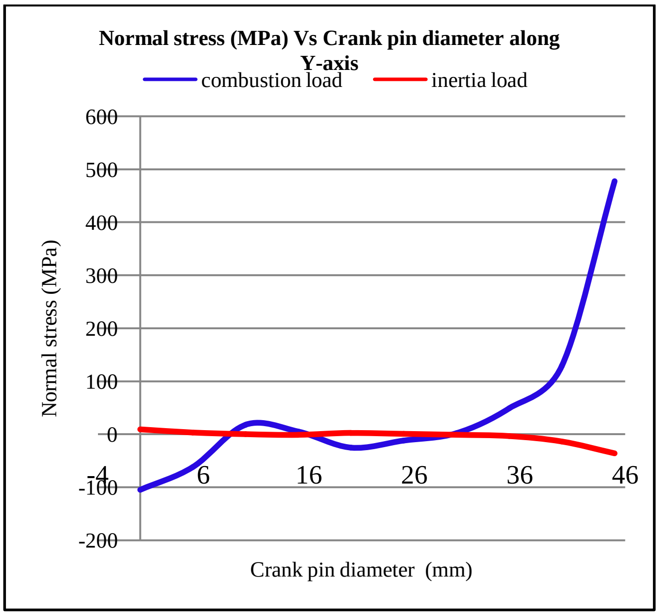

Fig. 13 Graph between Normal stress Vs Crank pin diameter fillet along Y-axis

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VIII. MODIFICATION IN CRANKSHAFT GEOMETRY

Original geometry has been modified as some material is added on the face inclination area and balancing is done according to weight added so that crankshaft geometry is completely optimized. Changing the main bearing radius, crank pin radius, fillet radius or main journal pin fillet as suggested by authors, requires change in dimension of different components of crankshaft assembly, sometimes requiring change in manufacturing process. But modification suggested in this paper does not require any change in mating components of the crankshaft and does not require different manufacturing process. Fig 19 shows the modified geometry of crankshaft. After modification in geometry same boundary condition as that of original were applied on the crankshaft geometry and verification was done in ANSYS. The results of original geometry and modified geometry were then compared.

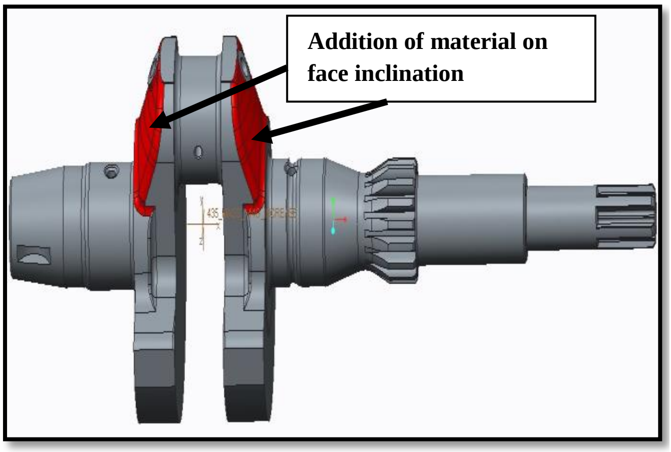

Fig. 19 Modification done on crankshaft geometry

A. *Results of modified geometry*

According to the applied loading conditions on modified crankshaft geometry, the results show maximum and minimum locations of stresses. Fig 20 shows normal stress of 393 MPa generated at crankpin fillet for compressive loading. Fig 21 shows 48 MPa compressive stresses generated at crank pin fillet for tensile loading. Fig 22 shows fatigue life of the component as 2.0998×10^{8}cycles.

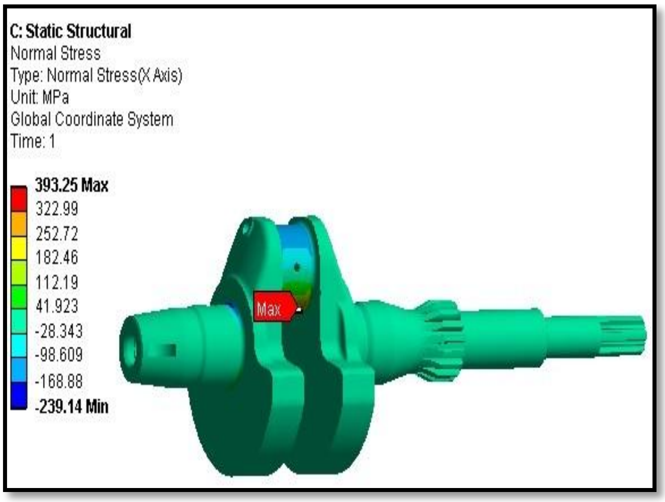

Fig. 20 Normal stress on modified crankshaft for compressive loading

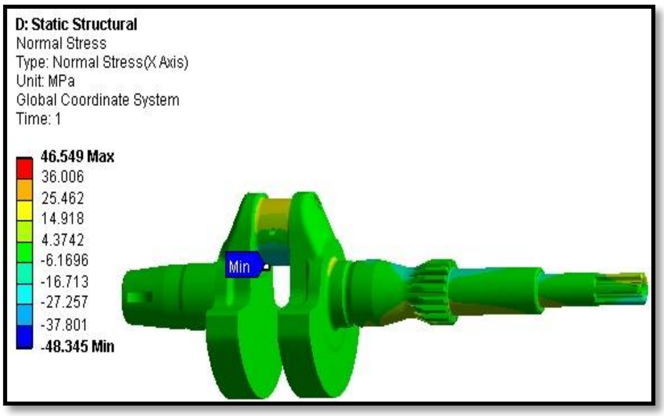

Fig. 21 Normal stress on modified crankshaft for tensile loading

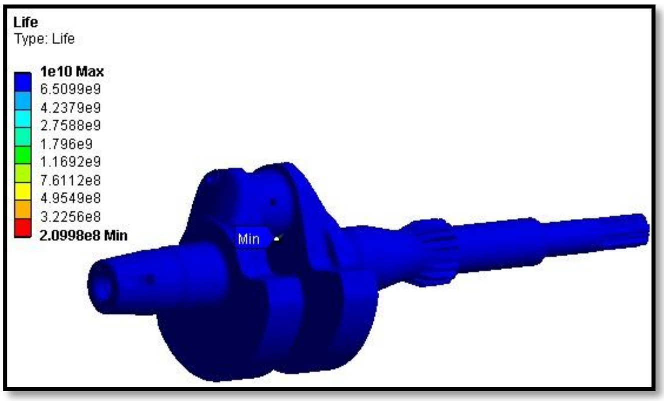

Fig. 22 Fatigue life for modified crankshaft

B. *Comparison of results*

Comparison between two results clearly shows that the stress is reduced by significant amount on modified crankshaft. Fig 24 and 25 shows comparison between normal stress at crankpin fillet and normal stress at journal fillet respectively on original as well as modified crankshaft. Maximum reduction in stress is almost 20%. Bar chart describes comparison of life at crankpin fillets on original as well as modified geometry and also indicates that maximum 15 times improvement in fatigue life of crankshaft.

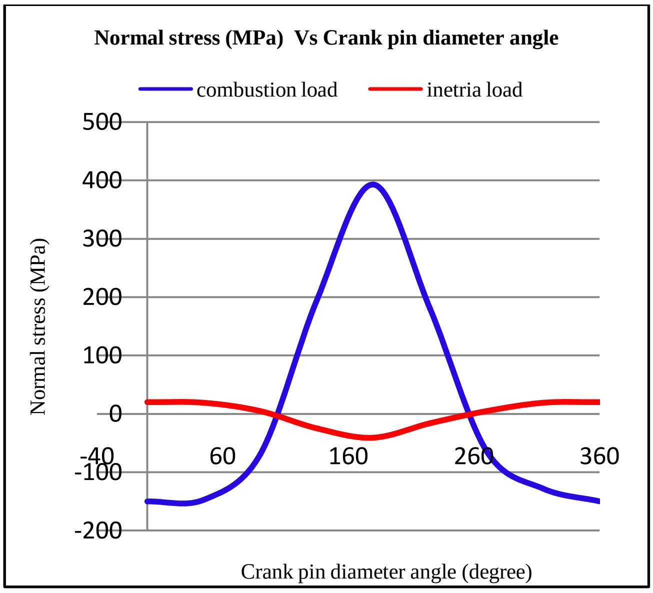

Fig. 23 Graph between Normal stress Vs Angle on Crank pin fillet on modified geometry

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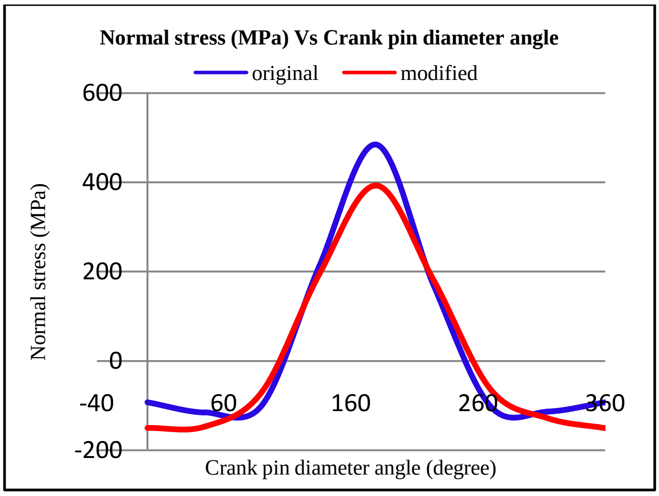

Fig. 24 Comparison of stresses between original and modified geometry on crankpin fillet

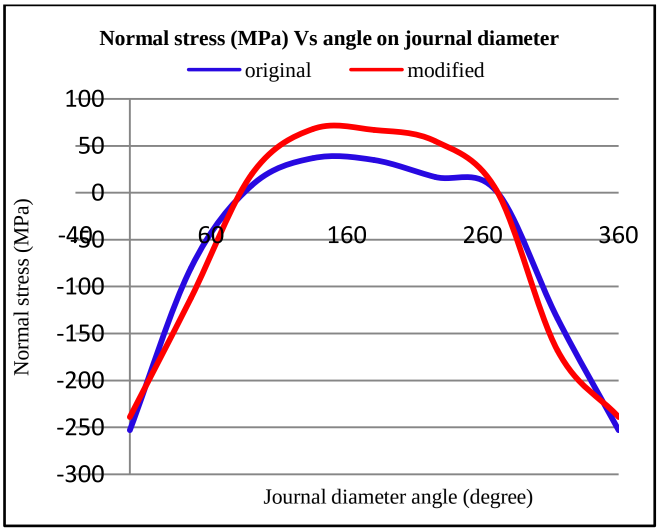

Fig. 25 Comparison of stresses between original and modified geometry on journal fillet

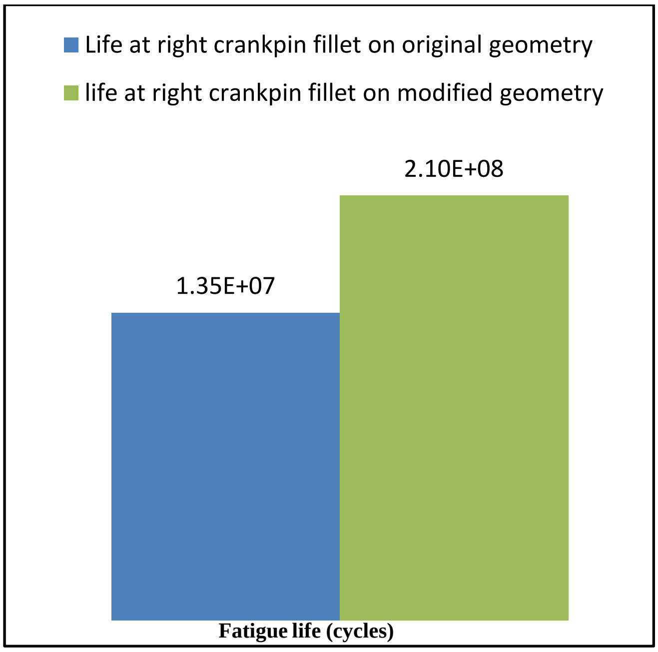

Fig. 26 Comparison of life at fillet locations

CONCLUSION

Detailed structural and fatigue analysis procedure for crankshaft analysis is described. Nature and magnitude of stresses at crankpin and journal fillet location are clearly shown. Also crankshaft geometry has been successfully modified and results are compared and verified using ANSYS.

Maximum stress location is observed at the same location as that of actual failure location for baseline crankshaft, with maximum tensile stress at crank pin fillet and minimum compressive stress at journal fillet.

Slight addition of material on crankshaft face inclination, gives 20% reduction in stresses and 15 times increment in fatigue life for modified configuration.

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