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International Journal Of Engineering And Computer Science ISSN:2319-7242
Volume 3 Issue 5 May, 2014 Page No. 5579-5587

Stress Analysis and Optimization of Crankshafts Subject to Static Loading

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Abstract
Crankshaft is one of the critical components for the effective and precise working of the internal combustion engine. In this paper a dynamic simulation is conducted on a crankshaft from a single cylinder 4- stroke diesel engine. A three-dimension model of diesel engine crankshaft is created using SOLID WORKS software. Finite element analysis (FEA) is performed to obtain the variation of stress magnitude at critical locations of crankshaft. Simulation inputs are taken from the engine specification chart. The dynamic analysis is done using FEA Software ANSYS which resulted in the load spectrum applied to crank pin bearing. The analysis is done for finding critical location in crankshaft. Stress variation over the engine cycle and the effect of torsion and bending load in the analysis are investigated. Von-mises stress is calculated using theoretically and FEA software ANSYS. The relationship between the frequency and the vibration modal is explained by the modal and harmonic analysis of crankshaft using FEA software ANSYS.

Keywords – Diesel engine; Crank shaft in Ansys; finite element analysis; stress analysis;

1.0 INTRODUCTION

Crankshaft is one of the most important moving parts in internal combustion engine. Crankshaft is a large component with a complex geometry in the engine, which converts the reciprocating displacement of the piston into a rotary motion. This study was conducted on a single cylinder 4- stroke diesel engine. It must be strong enough to take the downward force during power stroke without excessive bending. So the reliability and life of internal combustion engine depend on the strength of the crankshaft largely. And as the engine runs, the power impulses hit the crankshaft in one place and then another. The torsional vibration appears when a power impulse hits a crankpin toward the front of the engine and the power stroke ends. If not controlled, it can break the crankshaft. Jian Meng et al. analyzed crankshaft model and crank throw were created by Pro/ENGINEER software and then imported to ANSYS software. The crankshaft deformation was mainly bending deformation under the lower frequency. And the maximum deformation was located at the link between main bearing journal, crankpin and crank cheeks.

Gu Yingkui et al. researched a three-dimensional model of a diesel engine crankshaft was established by using PRO/E software. Using ANSYS analysis tool, it shows that the high stress region mainly concentrates in the knuckles of the crank arm & the main journal and the crank arm & connecting rod journal ,which is the area most easily broken. Xiaorong Zhou et al. described the stress concentration in static analysis of the crank shaft model.

The stress concentration is mainly occurred in the fillet of spindle neck and the stress of the crankpin fillet is also relatively large. Based on the stress analysis, calculating the fatigue strength of the crankshaft will be able to achieve the design requirements. From the natural frequencies values, it is known that the chance of crankshaft resonant is unlike. This paper deals with the dynamic analysis of the whole crankshaft. Farzin H. Montazersadgh et al. investigated first dynamic load analysis of the crankshaft. Results from the FE model are then presented which includes identification of the critically stressed location, variation of stresses over an entire cycle, and a discussion of the effects of engine speed as well as torsion load on stresses.

2.0 FORCES ON THE CRANKSHAFT

Parameter	Specification
Capacity	395cc
Number of Cylinders	1
Bore x Stroke	86*68 mm
Compression Ratio	18:1
Maximum Power	8.1hp @ 3600 rpm
Maximum Torque	16.7 Nm @ 2200 rpm
Maximum gas pressure	25 Bar

Table 1: Specifications of single cylinder diesel engine

Calculation for forces on crankshaft

Force on the Piston,

F_p=Area of the bore * Max. Combustion pressure

= π/4 * D2 * Pmax

= π/4 * 862 * 10-6 * 25 *105

= 14.52 kN.

Thrust on connecting rod,

FQ= Fp/cos ϕ

ϕ = Angle of inclination of connecting rod with the line of stroke

Sin ϕ = sin θ / (L/R)

L = Stroke length = 68 mm

R = Crankshaft radius = 17 mm

θ = Maximum crank angle = 35°

ϕ = 8.24°

Thrust on connecting rod , FQ = 14.67 KN

Thrust on crankshaft is resolved into two components

1. Tangential force , FT = FQ sin (θ + ϕ)= 10.049 KN
2. Radial force , FR = FQ cos (θ + ϕ)= 10.69 KN

3.0.DYNAMIC LOAD AND STRESS ANALYSIS

The crankshaft is subjected to complex loading due to the motion of the connecting rod, which transforms two sources of loading, namely combustion and inertia, to the crankshaft. Optimization of the crankshaft requires a determination of an accurate assessment of the loading, which consists of bending and torsion. Dynamic loading analysis of the crankshaft results in more realistic stresses whereas static analysis provides results that may not reflect operating conditions.

Figure 1 shows the digitized model of the forged steel crankshaft used in this study. The dynamic analysis of the engine that uses this type of crankshaft showed that as the engine speed increases the maximum bending load decreases. Therefore, the critical loading case for this engine is at the minimum operating speed of 2000 rpm [5]. This should not be misunderstood as to mean that the higher the engine rpm is the longer the service life because there are many other factors to consider in operation of the engine. The most important issue when the engine speed increases is wear and lubrication. As these issues were not considered in the dynamic load analysis study, further discussion of these issues is avoided. It was also shown that a consideration of the torsional load in the overall dynamic loading conditions has no effect on von Mises stress at the critically stressed location [5]. In addition, the effect of rsion on the stress range is also relatively small at other locations undergoing torsional load. Therefore, the crankshaft analysis could be simplified to an application of bending load only [5].

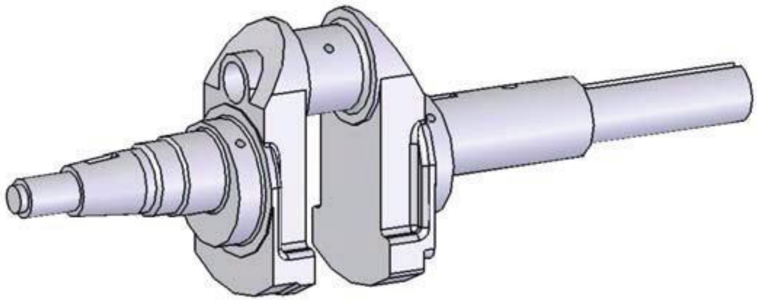

Figure 1: Digitized model of the original forged steel Crankshaft geometry used.

Convergence of stress at different locations was considered as the criterion for the selection of mesh size and number of elements for the finite element stress analysis. Satisfactory results were obtained using 119,337 elements for the crankshaft that corresponded to a global mesh size of 5.08 mm and a local mesh size of 0.762 mm in the model. This local mesh size resulted in five elements in the radius of the fillet areas.

The approach used to obtain stresses at different locations at different times during an engine cycle was by superposition of the basic loading conditions. This involves applying a unit load to each asic condition and then scaling the stresses from each unit load according to the dynamic loading. Then the corresponding stress components are added together.

Figure 3 shows the maximum von Mises stress and von Mises stress range at six locations on the crankshaft fillets at the engine speed of 2000 rpm for the crankshaft. The sign of von Mises stress was determined by the sign of the principal stress that has the maximum absolute value. As can be seen from the figure, the maximum von Mises stress occurs at location 2, while other locations experience stresses lower than location 2. Therefore, the other five locations were not considered o be critical in the analysis.

Local shape optimization techniques were applied to different locations of the crankshaft to lower the weight. After each optimization step the counter weights were balanced in order to achieve an accurate estimate of the weight reduction. Also, following each shape optimization iteration the optimized component was investigated for design feasibility and examined to determine whether the design was the best possible option.

4.0.OPTIMIZATION OBJECTIVES AND CONSTRAINTS

In this study, reducing the weight and manufacturing cost while improving or maintaining the fatigue performance of the original component were the main objectives. The stiffness has to be kept within permissible limits.

Since the original crankshaft used in the engine had acceptable fatigue performance, optimization was carried out in such a way that the equivalent local stress amplitude at any location of the optimized model did not exceed the equivalent stress amplitude at the critical location of the original model. In addition, the optimized crankshaft was expected to be interchangeable with the original crankshaft. Therefore, the following dimensions were not changed:

5.0. FORGED STEEL CRANKSHAFT DIMENIONS:

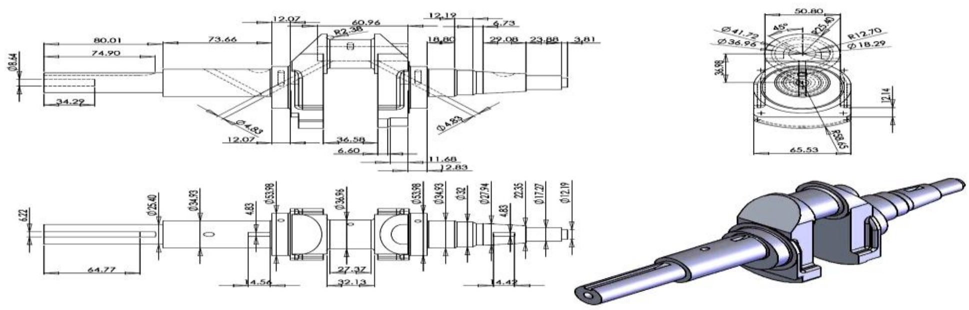

Figure 2: Dimensions of the existing crankshaft

X Outer diameters of different cylinders
x Crank radius
x Location of the main bearings
x Geometry of the main bearings
x Width and geometry of the connecting rod bearing .

According to FE analysis, the blue locations shown in Figure 3 have low stresses during the loading cycle and have the potential for material removal and weight reduction. It should be noted that the effect of mean stress on the results was negligible. Therefore, all of the stresses under consideration and discussed here are with reference to the stress ranges.

Several cases of geometric modifications were considered. Since maintaining dynamic

Another optimization step which does not require any complicated changes in the geometry is increasing the diameter of the crankpin hole (Case 2 in Figure 5). An Increase in the diameter of this hole will result in decreasing the moment of inertia of the cross section. Therefore, in order to avoid increasing the stress level at the fillet area, the fillet radius has to be increased. This

Considering the functions of the crankshaft and its constraints, the following design variables were then considered in the optimization study:
x Thickness of the crank web
x Geometry of the crank web
x increasing the inner hole diameters and depths
x Geometry changes on the outer section of the crankpin bearing.

balance is a key concern in the optimization of this component, the first step was to remove material symmetric to the central axis without compromising the dynamic balance of the crankshaft. On the far right side of the crankshaft shown in Figure 2 there is a threaded hole. The depth of this hole does not affect the function of the crankshaft. Therefore, this hole could be drilled as far as possible in the geometry (Case 1 in Figure 5).

increase in the fillet radius does not affect the connecting rod geometry since the connecting rod has sufficient clearance. Applying these changes to the crankshaft causes the centre of mass to move towards the counter weights. In order to balance the modified crankshaft, material has to be removed from the counter weight.

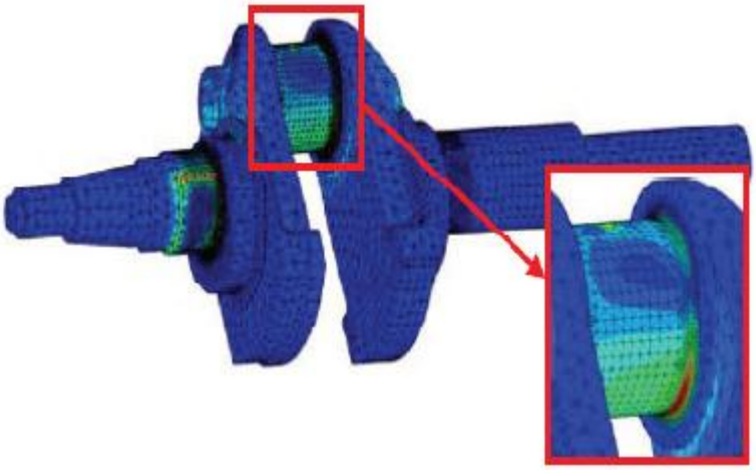

Figure 3:Stress distribution under critical loading condition at the crank angle of 5 degrees after TDC.

A significant percentage of the weight in the crankshaft is in the crank counterweight or web mass. Therefore, reducing the weight of this section could result in an improvement in weight reduction of the component. Reducing the web thickness is another optimization opportunity that was performed on the crankshaft web (Case 3 in Figure 5). As a result of this change in the crank web, the centre of gravity moves toward the crankpin bearing. In order to dynamically balance the crankshaft, material has to be added to the counter weights which can be accomplished by increasing the radius.

This option was restricted to the clearance between the piston and counter weights of the crankshaft. Since the current geometry is designed with a specific clearance between the counter weights and the piston, this optimization effort could not be implemented without additional changes to the piston. Redesigning the crank web and removing material from this section was the next optimization case under consideration while keeping the feasibility of the manufacturing process in mind. This requires not having negative slopes. The crank web was then modified such that no changes in the counter weights would be necessary (Case 4 in Figure 5).

Other optimization cases considered in this study, but not implemented due to either a small decrease in weight and/or manufacturing difficulty or cost, were:

x Redesigning the crankpin geometry

x Removal of material from the centre of the crank web, symmetric to the central axis

x Removal of semi-circular material from the centre of the crank web, symmetric to the central axis

x Eccentric crankpin hole

Since each optimization case was studied individually, further analysis was needed by considering a combination of these cases. Options for a redesigned crankshaft were developed such that as many optimization cases as possible could be applied.

FE models of possible combinations were created and FE analysis with dynamic load was considered for each combination. Results of the stress range for each of these combination models showed that the critical (location 2 in Figure2) does not change as a result of these geometry modifications. A comparison plot of stress range ratios for the different case combinations of the optimized crankshaft, as compared to the stress range in the original crankshaft at the critical location, is shown in Figure 6. The horizontal line at 1 stands for the original crankshaft

PARAMETERIC CASES TO BE CHANGED AND OPTIMISED:

1. Increasing the depth of the drilled hole at the back of the crankshaft
2. Increasing the hole diameter of the crankpin oil hole
3. Redesigning the geometry of the crankpin
4. Rectangular material removal from the center of the crank web symmetric to the central axis
5. Semi-circle material removal from the center of the crank web symmetric to the central axis
6. Reducing the thickness of the web
7. Modification of the crank web design

6.0 REDESIGNED MODEL AND ITS STRESSES:

S.No	MODEL	VON-MISES STRESS $(N\backslash m^2)$
1	Original design	**Min :** 588.723 **Max :** 2.9375e+009
2	Increasing the depth of the drilled hole at the back of the crankshaft	**Min :** 77.9113 **Max :** 2.67439e+008
3	Increasing the hole diameter of the crankpin oil hole	**Min :** 768.277 **Max :** 2.63655e+009
4	Redesigning the geometry of the crankpin	**Min :** 327.008 **Max :** 2.58568e+008
5	Rectangular material removal from the center of the crank web symmetric to the central axis	**Min :** 306.435 **Max :** 2.46762e+008
6	Semi-circle material removal from the center of the crank web symmetric to the central axis	**Min :** 372.931 **Max :** 3.98568e+008
7	Reducing the thickness of the web	**Min :** 577.128 **Max :** 2.83763e+008
8	Modification of the crank web design	**Min :** 464.6 **Max :** 2.79902e+008

Table 2 :Redesigned models and its stress

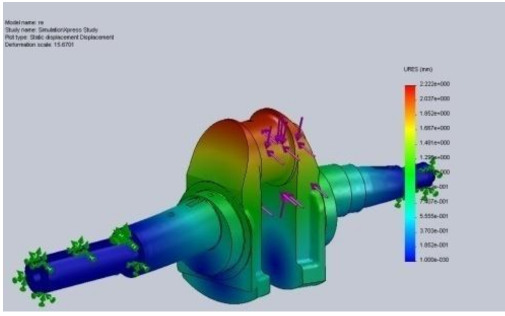

Fig. 4 : Original model

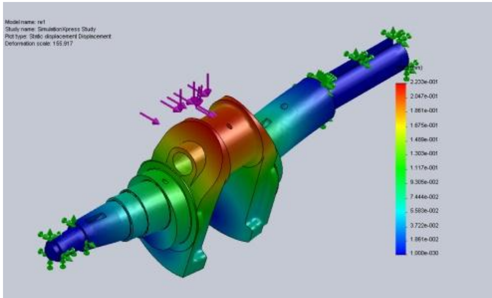

Fig. 5 : Increasing the depth of the drilled hole at the back of the crankshaft

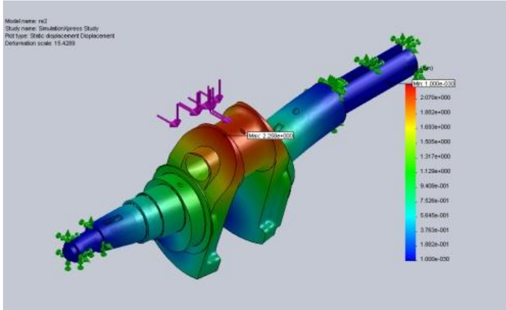

Fig. 6 : Increasing the hole diameter of the crankpin oil hole

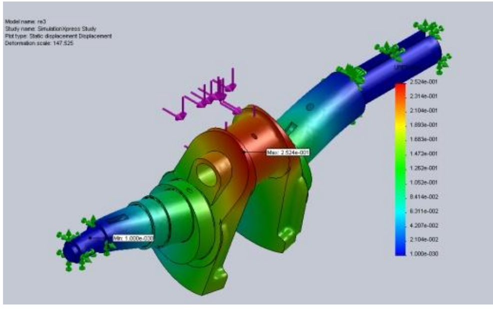

Fig. 7 : Redesigning the geometry of the crankpin

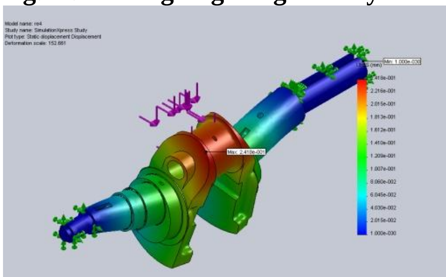

Fig. 8: Rectangular material removal from the center of the crank web symmetric to the central axis

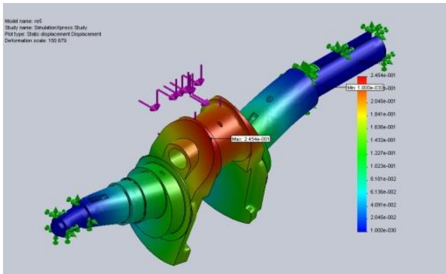

Fig. 9:Semi-circle material removal from the center of the crank web symmetric to the central axis

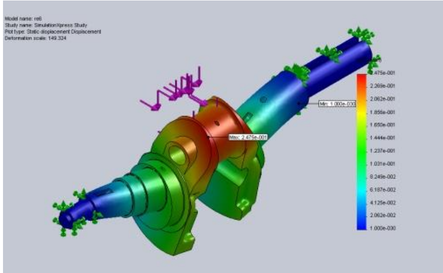

Fig. 10:Reducing the thickness of the web

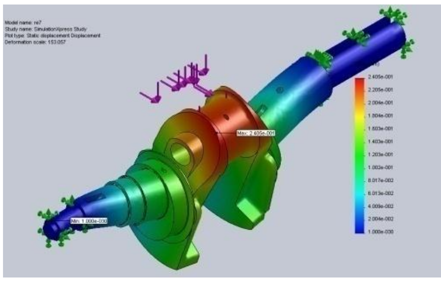

Fig. 11: Modification of the crank web design

7.0 OPTIMISATION FLOWCHART:

The investigation of the stress contour of the crankshaft FEA model during an engine cycle showed that some locations of the crankshaft, such as the counter weights and crank webs, are subjected to low stresses. The crankshaft has to be dynamically balanced in which the counter weights serve this purpose. Therefore, although stresses applied to these sections are low, these sections cannot be removed, but can only be changed according to other modifications made to the component.

The general flow chart of the optimization process is shown in Figure 12. Objective function, design variables, and constraints are summarized in this figure and it is shown that the optimization process consisting of geometry modifications, manufacturing process considerations, and material alternatives was performed simultaneously.

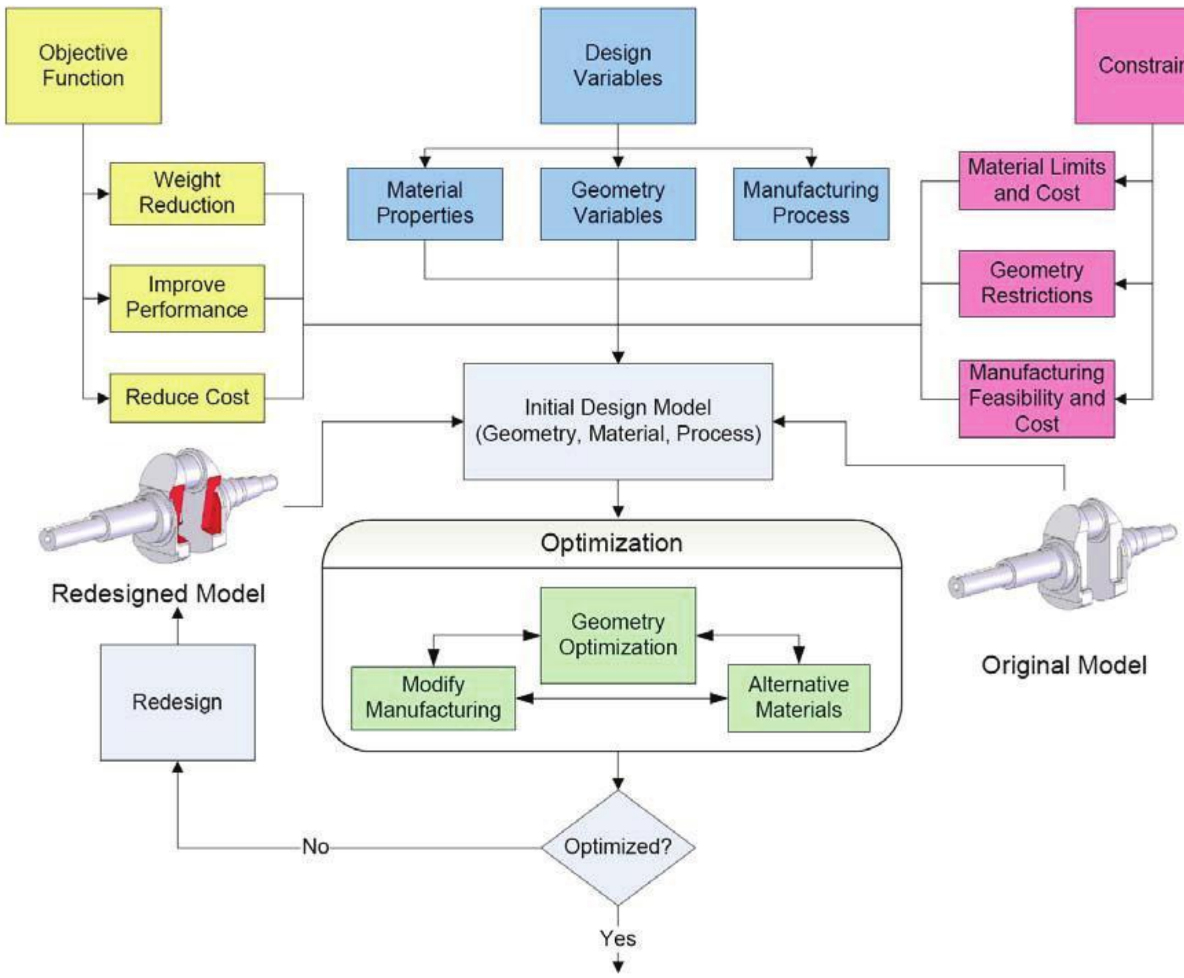

Fig.12 : Optimmisation process Flow chart

8.0 FINAL OPTIMIZED CRANKSHAFT:

Considering the manufacturing processes, the geometry of the crankshaft could be modified further to take advantage of the results of improved fatigue strength due to fillet rolling and/or the use of micro alloyed steel. Further modification to the crankpin geometry is possible. Increasing the crankpin hole is an option which does not influence the manufacturing process and is not an expensive process. Increasing the hole diameter from the original 18.3 mm (0.72 inch) to 25.4 mm (1 inch) and reducing the crank web thickness in order to maintain dynamic balance of the crankshaft, will cause the stress range at the critical location to increase by 7%. This increase is easily compensated for by the beneficial effect of the compressive residual stress from fillet rolling. This modification is shown as Case 1, 2, 3, and 4 in Figure 6. Since the wall thickness in the crankpin area is limited, further increasing the hole diameter to larger than 25.4 mm was not possible, because sufficient material is needed to restrict plastic deformation in the rolling process to produce residual stress. These modifications reduce the weight of the original crankshaft by 18%. The final optimized geometry is shown in Figure 5. It should be noted that the fatigue strength of the optimized crankshaft is significantly higher than the original crankshaft due to a slight increase in the stress range of 7% at the fillet and a significant increase on the order of 40% to 80% in fatigue strength due to fillet rolling, as discussed earlier. Figure 7 shows the stress range comparisons at different locations shown in Figure 2, between the original and the optimized crankshafts. As can be seen in this figure, the critical location is still at the fillet and the increase in stress at other locations is not significant.

With regard to the cost of the optimized crankshaft, this is affected by the geometry changes and weight reduction, modification in the manufacturing process, and the use of MA steel. The optimized geometry requires redesign and remanufacturing of the forging dies used. The geometry parameters that influence machining and the final cost of the component include an increase in the drilling process. This is because the drilled holes in Cases 1 and 2 are redesigned to have larger diameters and the bore in Case 1 is modified to have more depth than the original bore. The application of compressive residual stress by a fillet rolling process is a parameter in the manufacturing process that will add to the cost of the finished component.

Although a micro alloy steel grade is somewhat more expensive than hot-rolled steel bar, the heat treatment cost savings are significant enough to offset this difference (Wicklund, 2007 [8]). In addition, micro alloyed steel has 5% to 10% better machinability than quenched and tempered steel, resulting in reduced machining costs due to enhanced production rates and longer tool life (Nallicheri et al. [3]). A consideration of these factors, along with the reduced material cost due to the weight reduction, indicates a reduction in the total cost of the forged steel crankshaft.

9.0. SUMMARY

An optimization study was performed on a forged steel crankshaft that considered the geometry, performance, manufacturing process, and cost. A major constraint of this optimization was for the optimized crankshaft to replace the original crankshaft in the engine without any changes to the engine block or the connecting rod. An optimization in the geometry included local changes at different locations on the crankshaft, which were then combined to obtain the final optimized geometry.

Adding fillet rolling was considered in the manufacturing process. Fillet rolling induces compressive residual stress in the fillet areas, which results in a significant increase in fatigue strength of the crankshaft, and in turn, significantly increases the fatigue life of the component. The use of a micro alloyed steel as an alternative material to the current forged steel composition results in the elimination of the heat treatment process. In addition, when considering the improvement in machinability of the micro alloyed steel along with the reduced material cost because of an 18% weight reduction, the overall cost of the forged steel crankshaft can be reduced.

The optimization resulted in weight reduction of the forged steel crankshaft. This was achieved by changing the dimensions and geometry of the crank counterweights while maintaining dynamic balance of the crankshaft. The optimization that was developed did not require any changes to the engine block or connecting rod.

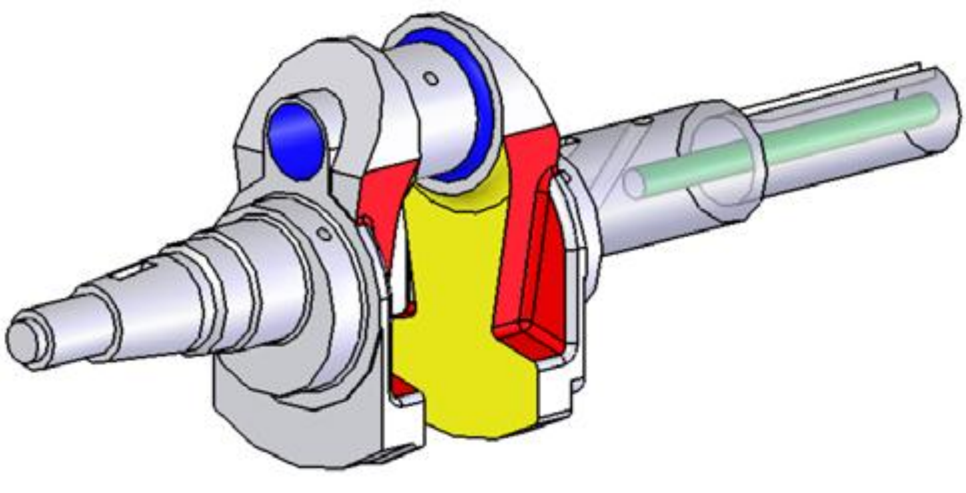

Fig. 13: Final Optimised model of Crankshaft

10.0 CONCLUSIONS :

In this paper, the crankshaft model was created by Solid works 2009 software. Then, the model created by Solid works was analyzed solid works simulation software. This project focused on the optimization possibilities in the crankshaft. A crankshaft made from AISI 1035 Steel was selected to utilize the form flexibility of the forging process and the cost benefits of using steel. The results illustrate that even a production crankshaft possesses enough potential for further optimization in friction and weight reduction. According to the outputs of the simulations, the goals defined by the optimization are realizable, without encountering fatigue problems. Additional reductions in bearing diameters or larger inner diameters would lead to a loss in durability, which can cause the crankshaft to operate under the safety limit, even at lower speeds.

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