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Optimization of Crankshaft for Weight Reduction

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Abstract— *The main objective of this study was to investigate weight and cost reduction opportunities for a crankshaft. The need of load history in the FEM analysis necessitates performing a detailed analysis. Therefore, this study consists of two major sections: FEM and stress analysis and optimization for weight and cost reduction. In this study a simulation was conducted on crankshaft. Finite element analysis was performed to obtain the variation of stress magnitude at critical locations. The dynamic analysis was done analytically and was verified by simulations in ANSYS. The analysis was done and as a result, critical region on the crankshafts were obtained. Geometry, material, and manufacturing processes were optimized considering different constraints, manufacturing feasibility, and cost.*

Keywords- ansys; FEM; simulation; geometry.

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 consisting of shaft parts two journal bearings and one crankpin bearing. The Shaft parts which revolve in the main bearings, the crank pins to which the big end of the connecting rod are connected, the crank arms or webs which connect the crank pins and shaft parts. In addition, the linear displacement of an engine is not smooth; as the displacement is caused by the combustion chamber therefore the displacement has sudden shocks. The concept of using crankshaft is to change these sudden displacements to as smooth rotary output, which is the input to many devices such as generators, pumps and compressors. It should also be stated that the use of a flywheel helps in smoothing the shocks.

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 crank shaft, the force will be transmitted to the crankshaft. The magnitude of the forces depends on many factors which consist of crank radius, connecting rod dimensions, and weight of the connecting rod, piston, piston rings, and pin.

II. LITERATURE REVIEW

Farzin H. Montazersadgh and Ali Fatemi[1] conducted a study on a crankshaft for dynamic load and stress analysis in which they find that, 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 355 degrees for this specific engine. At this angle only bending load is applied to the crankshaft. Experimental stress and FEA results showed close agreement, within 7% difference. Critical (i.e. failure) locations on the crankshaft geometry are all located on the fillet areas because of high stress gradients in these locations, which result in high stress concentration factors. Geometry optimization resulted in 18% weight reduction of the forged steel crankshaft, which was achieved by changing the dimensions and geometry of the crank webs while maintaining dynamic balance of the crankshaft.

MS Shweta Ambadas Naik[2] conducted a study on Failure Analysis of Crankshaft by Finite Element Method in which she fined that the maximum deformation appears at the Centre of crankpin neck surface. The maximum stress appears at the fillets between the crankshaft journal and crank cheeks, and near the central point journal. The edge of main journal is high stress area. The crankshaft deformation was mainly bending deformation under the lower frequency. And the maximum deformation was located at the link between main bearing journal and crankpin and crank cheeks. She also fined that Dynamic analysis of a crank and slider can consider the combine effects of combustion pressure and inertia forces.

K. Thriveni, Dr. B.JayaChandraiah[3] conducted a study on Modeling and Analysis of the Crankshaft Using Ansys Software in which they find that the maximum deformation appears at the Centre of the crankpin neck surface. The maximum stress appears at the fillet areas between the crankshaft journal and crank cheeks and near the central point journal. The value of von-misses stresses that comes out from the analysis is far less than material yield stress so our design is safe.

Jaimin Brahmbhatt, Prof. Abhishek choubey conducted a study on design and analysis of crankshaft for single cylinder 4-stroke diesel engine in which they find that that FEA is a good tool to reduce time consuming theoretical Work. The maximum deformation appears at the centre of crankpin neck surface. The maximum stress appears at the fillets between the crankshaft journal and crank cheeks and near the central point Journal. The edge of main journal is high stress area. The Value of Von-Misses Stresses that comes out from the analysis is far less than material yield stress so our design is safe and we should go for optimization to reduce the material and cost.

III. SPECIFICATION OF ENGINE

Table 1 Splendor engine specification

Engine Type	**Air Cooled 4-Stroke**
Bore	50mm
Stroke	49.5mm
Displacement	97.2cc
Max. Power	5.74 KW (7.8 PS) @7500rpm
Max. Torque	8.04 N-m @4500 rpm
Compression Ratio	9:1
Density Of Petrol (C8h18)	737.22 kg/m^3
Flash Point Of Petrol	45F
Auto Ignition Temp.	220^{0}C=493K
Molecular Weight Of Petrol	114.228 g/mole

IV. SIMULATION WORK

A. Modeling

In modeling we first take the original crankshaft from the market. Then we take it’s measurement by hand held measuring devices. Then with the help of these measurements we create a single-single part namely crank pin, LHS side of crankshaft and RHS side of crankshaft in CATIA V5. Then we assembled those parts in assembly mode. Then we save the assembly with the extension required for meshing. So we complete modeling by using CATIA V5.

Table 2 Material properties

Sr. no	Property	Value
1	Young's modulus	2.06E+11Pa
2	Poisson's Ratio	0.3
3	Bulk modulus	1.72E+11Pa
4	Shear modulus	7.92E+10Pa
5	Yield strength	9.80E+08Pa
6	Ultimate strength	1.10E+09Pa

B. Meshing

Mesh generation is the practice of generating a polygonal or polyhedron. Mesh that approximate geometric domain. The term 'grid generation' is the often use interchangeably. Typical uses are for rendering to a computer screen or for physical simulation such as finite element analysis or computational fluid dynamics. The input model form can vary greatly but common sources are CAD, STL(file format) or a point cloud. The field is highly interdisciplinary, with contribution found in mathematics, computer science, and engineering.

Three-dimensional meshes created for finite element analysis need to consist of tetrahedral, pyramids, prisms or hexahedra. Mesh is other piecewise discretization of domain existing in one, two or three dimensions.

Table 3 Meshing statistics

Sr. no	Content	Description
1	Element type	Tetrahedron
2	Number of nodes	373311
3	Number of Elements	236190

V. BOUNDRY CONDITIONS

Fixed supports at ball bearings. And pressure of 3MPa pressure at crankpin.

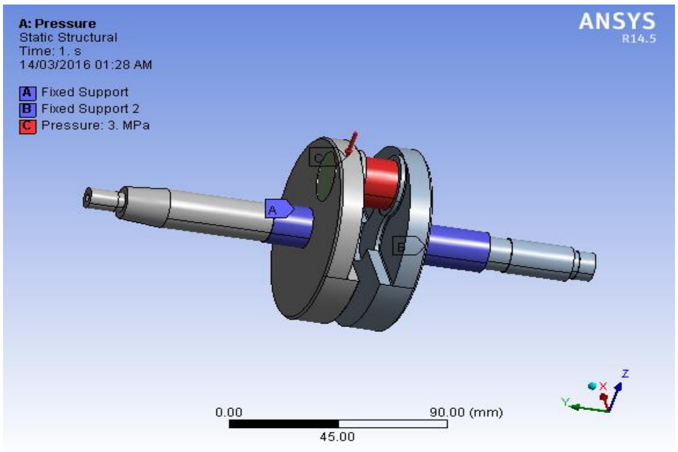

Figure 1 Boundary Condition

VI. CALCULATION FOR VON-MISSES STRESESS

P_{max}=30bar (For splendor 100)

Design of crankshaft when the crank is at an angle of maximum twisting Moment

Force on the Piston

F_p = Area of the bore x Max. Combustion pressure

$$=\frac{\pi}{4}*D^2*P_{max}$$

$$=\frac{\pi}{4}*(.050)^2*3000000$$

$$=5.89KN$$

As $\Theta=35^o$;

$$\sin\phi=(\sin\Theta)/(N)$$

N=Obliquity ratio

=sin35/4

Which implies $\phi=8.24^o$

We know that thrust Force in the connecting rod,

$$F_Q=F_P/\cos\phi$$

$$=5.89/\cos 8.24$$

$$=5.95KN$$

From we have

Thrust on the connecting rod,

$$F_Q=5.95KN$$

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Thrust on the crankshaft can be split into tangential component and radial component. 1. Tangential force on the crankshaft,

$$F_T = F_Q \sin(\theta + Ø) = 4.076KN$$

2 .Radial force on the crankshaft,

$$F_R = F_Q \cos(\theta + Ø) = 4.33KN$$

Reactions at bearings (1&2) due to tangential force is given by

$$H_{T1} = H_{T2}$$

$$= F_T/2$$

$$= 2.038KN$$

Thrust on the crankshaft can be split into tangential component and radial component.

1. Tangential force on the

Similarly, reactions at bearings due to radial force is given by

$$H_{R1} = H_{R2} = F_R/2$$

$$= 2.165KN$$

Design of Crankpin

We know that bending moment at the center of the crankshaft

twisting moment $Tc = \sqrt{(Mc^2 + Tc^2)}$

$$= 148.66 \text{ kN-mm}$$

Von-misses Stresses

$$M_{ev} = \sqrt{\left((Kt \cdot Mc)^2 + \frac{3}{4}(Kt \cdot Tc)^2\right)}$$

$$= 142.03 \text{KN-mm}$$

Kb=combine shock & fatigue Factor for Bending

Kb=2

Kt = combine shock & fatigue Factor for Torsion

Kt=1.5

$$Mc = \frac{\pi}{32} \cdot d^3 \cdot \sigma$$

$$= 56.49 \text{N/mm}^2$$

Shear stress:

$$\tau_c = \frac{\pi}{16} d^3 c \times \tau$$

$$\tau = 54.85 \text{N/mm}^2$$

$$\tau_c = 37.2 \text{MPa}$$

VII. ANALYSIS RESULTS

A. Von-misses stress

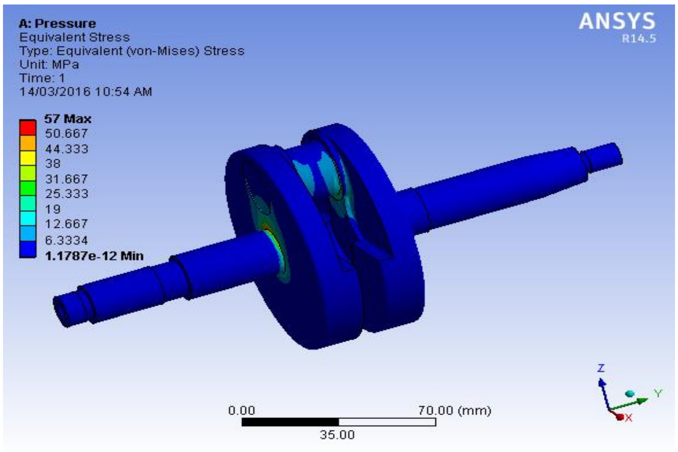

Figure 2 Von misses stresses

B. Total deformation

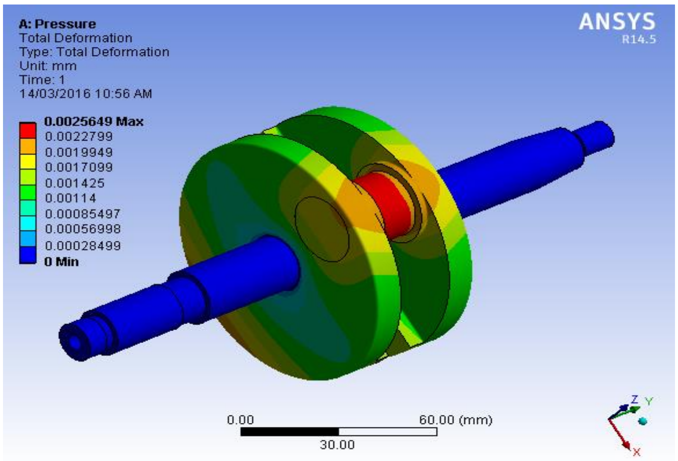

Figure 3 Total Deformation

VIII. WEIGHT REDUCTION

For reducing weight localized stress area and also where stresses are minimum. After finding these places from these places we can remove material considering counter weight.

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On that approach some of the cases are considered those cases are as follows.

Case I) Reducing Crank web thickness on each flange

Reduced Crank web thickness by 1 mm on each web of and then material removed by maximum of 3%.

Result of this changing crank web thickness the center of gravity moves towards crank pin bearing.

This optimizing step did not change the stress compared original crankshaft.

Case II) Increasing The diameter of the drill hole at the back of crankshaft.

By using this case not disturbs the dynamic balancing of crankshaft. Weight is reduced by 1%.

Case III) Semicircle material removal from the center of the crank web symmetric to central axis.

Optimization the stresses slightly with applied pressure. This case also not disturbs the dynamic balancing of crankshaft.

VIII. COMBINED CASES FOR WEIGHT REDUCTION

Applying All Cases combined after that the total weight reduced by 5%.

Also stresses are reduced by some extent.

For this also fatigue analysis is done as well and it resulted safe. So this optimization is useful in weight reduction of crankshaft.

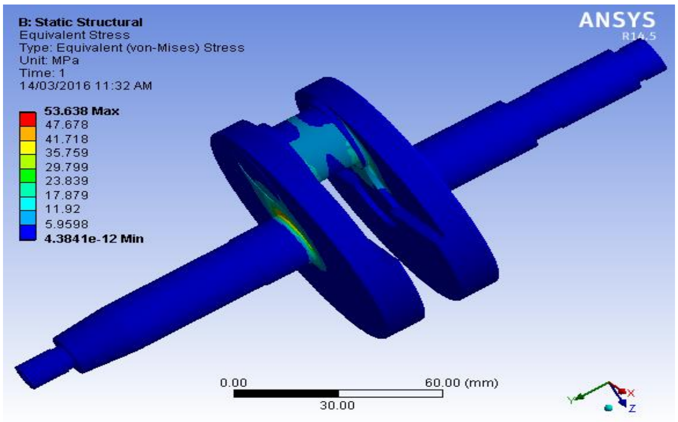

Figure 4 Von misses stresses

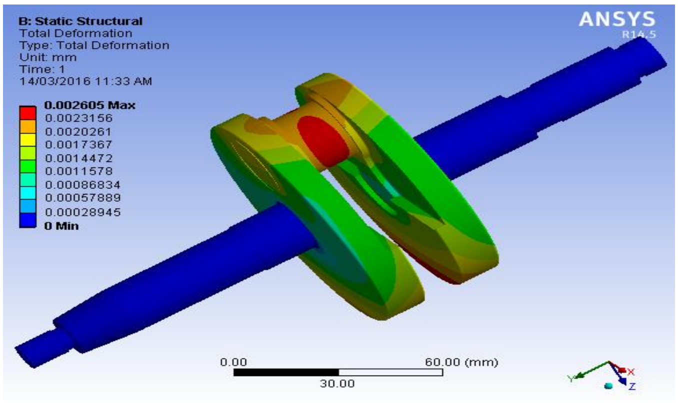

Figure 5 Total Deformation

IX. FLOWCHART

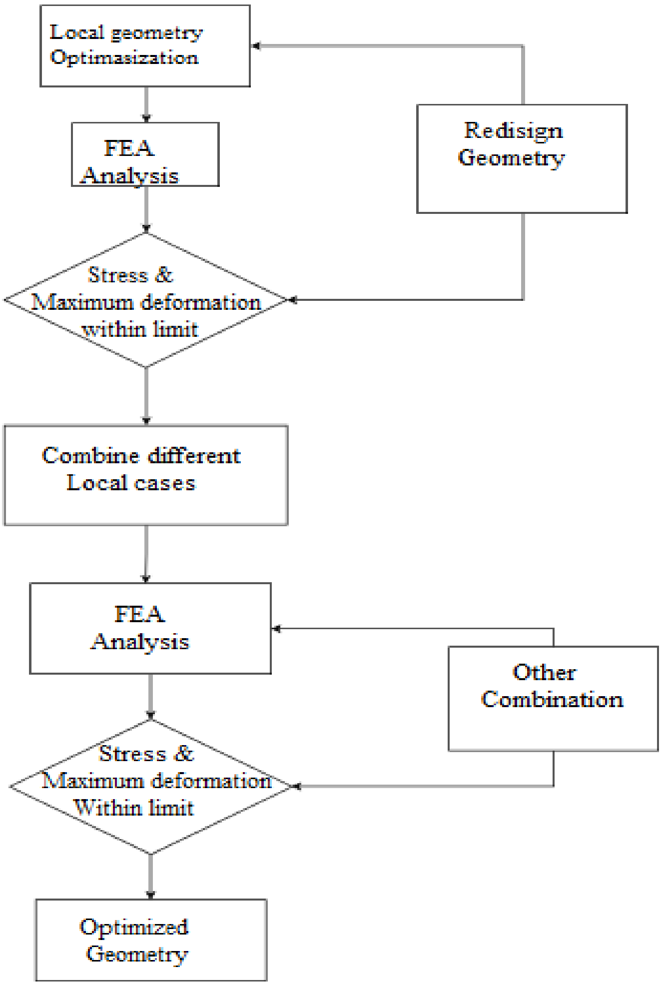

Figure 6 Flowchart

X. WEIGHT REDUCTION

Table 4 Weight reduction

	Total Weight Reduction(kg)	%Weight Reduction
Original weight(kg)	1.68509	-
Case1	1.60571	4.71
Case2	1.64588	2.32
Case3	1.6455	2.34
Combination case	1.60095	5

XI. CONCLUSION

Geometry optimization resulted in 5% weight reduction of crankshaft.

Further we go for the fatigue testing.

Acknowledgment

It gives us immense pleasure in presenting the report on “Optimization Of Crankshaft For Weight Reduction”. The joy and satisfaction that come along with successful completion of any work would be incomplete unless we mention the people who made it possible .There are many people, we like to thank for their contribution in completing this report successfully.

First we would like to mention a special thanks to our respected Project Guide Kodarkar Nakul M. who has really taken lot of efforts to get this project at individual level and also given us a continuous and immense support, suggestions and courage to complete this task. he has applied a constant guiding force to help us prepare and complete the project on schedule .We feel extremely fortunate to have worked under our project guide Kodarkar Nakul M.

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