ISSN 2395-1621

Modeling, Analysis and Comparison of Crankshaft for Weight Optimization Using FEA

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ABSTRACT

Computer aided modelling and optimization analysis of crankshaft is done to study, evaluate and compare the fatigue performance of two competing manufacturing technologies for automotive crankshafts, namely forged steel and ductile cast iron. In this paper a dynamic simulation will be conducted on two crankshafts, cast iron and forged steel, from similar single cylinder four stroke engines. Finite element analysis will be performed to obtain the variation of stress magnitude at critical locations. The dynamic analysis will be done analytically and will be verified by simulations in ANSYS. Results achieved from aforementioned analysis will be used in optimization of the forged steel crankshaft. Geometry, material and manufacturing processes will be optimized considering different constraints, Manufacturing feasibility and cost. The optimization process will include geometry changes compatible with the current engine, fillet rolling and will result in increased fatigue strength, reduction in weight and cost of the crankshaft, without changing connecting rod and engine block.

ARTICLE INFO

Article History

Received :18th November 2015

Received in revised form :

19th November 2015

Accepted : 21st November , 2015

Published online :

22nd November 2015

Keywords— Crankshaft, ANSYS, Cast iron and forged steel, Finite Element analysis.

I. INTRODUCTION

A crankshaft is a highly stressed component in an engine that is subjected to bending and torsional loads. The crankshaft must be designed to last the life of the engine due to the catastrophic damage to the engine which would result if failure did occur. Considering the life of an engine in an automobile, for example, this results in requirement for an infinite life fatigue situation. Because of the long life and high stresses, as well as the need for weight reduction, material and manufacturing process selection is important in crankshaft design. There is competition among materials and manufacturing processes, due to cost, performance, and weight. This is a direct result of industry demand for components that are lighter, to increase efficiency, and cheaper to produce, while at the same time maintaining fatigue strength and other functional requirements. There are several ways to manufacture a crankshaft, including machining from a billet, forging, and casting. Due to cost and time, machining a crankshaft from a billet is seldom used except in very low production applications. The two most common types of crankshafts are made of cast iron and forged steel.

The objective being to outline the steps and various aspects considered and followed for the development of a weight optimized and reliable crankshaft. A benchmark

dynamic simulation was conducted and verified on the existing forged production crankshafts and compared against the various design variants. The procedure consisted of determining dynamic operating forces.

Results achieved from aforementioned analysis were used in optimization of the crankshaft. Geometry, material, and manufacturing processes were optimized considering different constraints, manufacturing feasibility, and cost. The optimization process included geometry changes compatible with the regular established materials and processes, resulting in weight reduction, increased fatigue strength and reduced cost of the crankshaft, without changing any other compressor parts.

II. LITERATURE REVIEW

YenettiSrinivasa Rao, GouthamanMurali&Udupa S. Srinivasa [1] had conducted study to investigate weight and cost reduction opportunities for a Transport Refrigeration Compressor Crankshaft. These Compressors are generally run by a diesel engine operating at non uniform speeds and unlike Hermetic compressor shafts they tend to act differently based on the engine characteristics.They found that Overall Weight reduction of 12 % and Cost reduction of 23% was achieved from the optimization process from original crankshaft without affecting the life or vibration characteristics.

Jonathan Williams and Ali Fatemi [2]conducted a study to compare the fatigue behavior of forged steel and ductile iron crankshafts from a one-cylinder engine as well as to determine if the fatigue life of a crankshaft can be accurately estimated using fatigue life predictions.They concluded with the findings

that ultimate strength and yield strength of ductile cast iron were 80% and 66% of forged steel, respectively. The forged steel has greater ductility than the ductile cast iron, with a percent reduction of area of 58% for forged steel and only 6% for ductile cast iron.

C.M.Balamurugan,R.Krishnaraj,Dr.M.Sakthivel, K.Kanthavel, DeepanMarudachalam M.G, R.Palani [3] conducted study to evaluate and compare the fatigue performance of two competing manufacturing technologies for automotive crankshafts, namely forged steel and ductile cast iron. In this study a dynamic simulation was conducted on two crankshafts, cast iron and forged steel, from similar single cylinder four stroke engines.Since the forged iron crankshaft is able to withstand the static load, it is concluded that there is no objection from strength point of view also, in the process of replacing the cast iron crankshaft by forged crankshaft. They also reduced the cost of forged crankshaft by the mass production.

K. Thriveni, Dr.B. JayaChandraiah [5] in their paper studied the Static analysis on a crankshaft from a single cylinder 4-stroke I.C Engine. The modeling of the crankshaft is created using CATIA-V5 Software. Finite element analysis (FEA) is performed to obtain the variation of stress at critical locations of the crank shaft using the ANSYS software and applying the boundary conditions. They compared theoretical calculations with the ANSYS result and found 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.

C. AZOURY, A. KALLASSY, B. COMBES, I. MOUKARZEL, R. BOUDET [6] in their study presented the experimental and analytical modal analysis of a crankshaft. The effective material and geometrical properties are measured, and the dynamic behavior is investigated through impact testing. The three-dimensional finite element models are constructed and an analytical modal analysis is then performed to generate natural frequencies and mode shapes in the three-orthogonal directions. The finite element model agrees well with the experimental tests and can serve as a baseline model of the crankshaft.They concluded that the results from finite element model agree well with the experimental values. This model is suitable for the dynamic analysis of the crankshaft. The validated finite element model can be used for further dynamic analysis and evaluation of structural performance from loadings.

III.OBJECTIVES

In this project we will create model of crankshaft using modelling software CATIAV5 and imported in ANSYS 14 workbench for static analysis and Fatigue analysis. After analysis the comparison is made between cast iron and forged steel crankshafts. We will look for the opportunities of weight optimization of crankshaft in this process

IV.MATHEMATICAL MODEL FOR CRANKSHAFT

TABLE I

Crank Pin radius	22.6
Shaft diameter	34.925
Thickness of the crank web	21.336
Bore Diameter	53.73
Length of the crank pin	43.6
Maximum Pressure	35 bar

Force on the piston: Bore diameter (D) =53.73mm, FQ= Area of the bore ×Max. Combustion pressure

$=\pi/4 \times D^2 \times P$ max=7.93KN

In order to find the Thrust Force acting on the connecting rod (FQ), and the angle of inclination of the connecting rod with the line of stroke (i.e. angle Ø).

$\sin\phi = \sin\theta/(L/R) = \sin 35/4$

Which implies $=8.24^0$

We know that thrust Force in the connecting rod, FQ $=FP/\cos\varphi$

From we have

Thrust on the connecting rod, FQ =8.01KN

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

1. Tangential force on the crankshaft, FT = FQ sin (θ+ Ø) =5.48KN 2
2. .Radial force on the crankshaft, FR = FQ cos (θ+ Ø) =5.83KN
3. Reactions at bearings (1&2) due to tangential force is given by HT1=HT2 = FT/2

Similarly, reactions at bearings (1&2) due to radial force is given by

$H_{T1} = H_{T2} = F_T/2$

Design of crankpin:

Let d= diameter of crankpin in mm

We know that bending moment at the centre of the crankshaft $M_C=H_{R1}\times b_2$=156.62KN-mm

Twisting moment on the crankpin

(T_C)=61.94KN-mm

From this we have equivalent twisting moment

$Tc =(Mc^2 + Tc^2)^{1/2}$ = 168.42KN-mm

Von-misses stress induced in the crankpin

$Mev=((Kb+Mc)2)^{1/2} + ¾(KtxTc)2$ = 177.860KN-mm

$Mev= (\pi/32) \times d^3c \times \sigma v$

σv=19.6N/mm²

Shear stress:

$\tau e = (\pi/16) \times d3\ c \times \tau$

τ = 9.28N/mm

V. MATERIAL PROPERTIES

TABLE II

Chemical composition by percent weight [2]

	Forged steel	Ductile cast iron
C	0.45	3.44
Mn	0.81	0.48
P	0.016	0.019
S	0.024	0.004
Si	0.27	2.38
Al	0.033	0.01
Cr	0.1	0.09
Ni	0.05	0.06
Cu	0.13	0.31
N	0.008	---
O	13ppm	---

TABLE III

Summary of forged steel and ductile cast ironmaterial properties.[2]

Monotonic Properties	**Forged Steel**	**Cast Iron**	**Ratio**
Average Hardness, HRC	23	18	0.8
Average Hardness, HRB	101	97	0.96
Modulus of Elasticity, E, GPa	221	178	0.81
Yield Strength (0.2% offset), YS, MPa	625	412	0.66
Ultimate Strength, S_U, Mpa	827	658	0.80
Percent Elongation, % EL	54%	10%	0.19
Percent Reduction in Area, %RA	58%	6%	0.10
Strength Coefficient, K, MPa	1316	1199	0.91
Strain Hardening Exponent, n	0.152	0.183	1.20
True Fracture Strength , σ_f, MPa	980	562	0.57
True Fracture Ductility, ε_f	87%	6%	0.07

TABLE IV

Monotonic properties

Summary of forged steel and ductile cast ironmaterial properties. [2]

Cyclic Properties	**Forged Steel**	**Cast Iron**	**Ratio**
Fatigue Strength Coefficient, σ_f', MPa	1124	927	0.82
Fatigue Strength Exponent, b	-0.079	-0.087	1.10
Fatigue Ductility Coefficient, ε_f'	0.671	0.202	0.30
Fatigue Ductility Exponent, c	-0.597	-0.696	1.17
Cyclic Yield Strength YS', MPa	505	519	1.03
Cyclic Strength Coefficient, K', MPa	1159	1061	0.91
Cyclic Strain Hardening Exponent, n'	0.128	0.114	0.89
Fatigue Strength at 10^6 cycles, MPa	359	263	0.73
Note: - forged steel taken as the base for all ratio calculations.			

After doing the calculation the load on crankshaft will be obtained. The stresses obtained from static analysis would be utilized for the further fatigue analysis using S-N curve to obtain Fatigue life and fatigue Safety factor

VI. SCOPE OF WORK

The objective function is minimizing the weight, maximum stress at critical locations, and cost. The bounded constraint is maximum allowable stress of the material. The equality constraint is geometry limitations. And the design variables are upper and lower limit for size and geometry, material alternatives, and manufacturing processes. As discussed earlier, the optimization stages were considered based on judgment using the results of the FEA and dynamic service load. The judgment was based on mass reduction, cost reduction, and improving fatigue performance and bending stiffness using alternative

materials and considering manufacturing aspects. Manufacturing process and material alternatives were studied in a trial and error approach, after the geometrical optimization. Since the current crankshaft has proper 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. Since the optimized crankshaft was expected to be interchangeable with the current one, the dimensions of crank radius, location and Geometry of main bearings and thickness and geometry of connecting rod bearing were not changed.

VII. MODELLING AND MESHING OF THE CRANKESHAFT

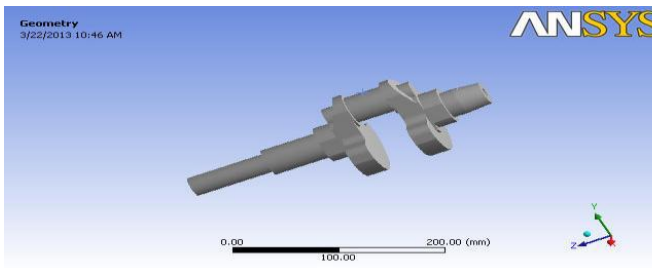

Fig 1.Model of the crank shaft [6]

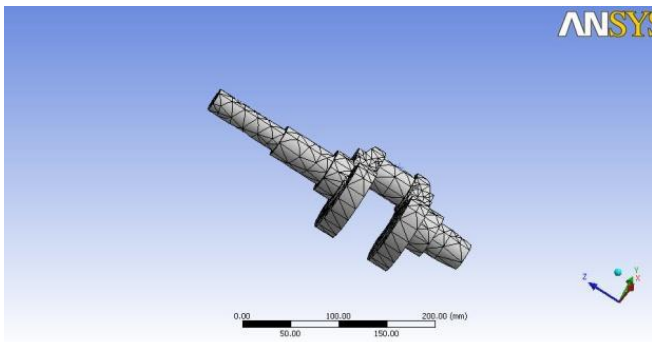

Fig.2 Model of crankshaft with meshing [6]

VIII. FINITE ELEMENT METHODOLOGY

Crankshaft is modeled in CATIA V5 Software and then imported to Ansys workbench 14. The FEA is divided in three different steps: - Preprocessor: Includes the 3D model preparation, loads and boundary condition definition, to select the appropriated element type and shape function and Finite Element (FE) mesh generation. - Solver: Definition of the numerical method to solve the linear system of equations, convergence criteria, error estimation and strain stress calculation. Post Processor: Engineers analysis and judgment phase based on stabilized design criteria.

IX. RESULTS & DISCUSSION

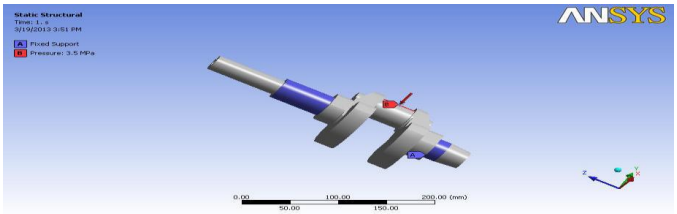

Fig.3 Apply Boundary conditioon the crankshaft The two ends of the crankshaft is To be fixed, the load 3.5 Mpa is applied on the top of the crankpin surface.[6]

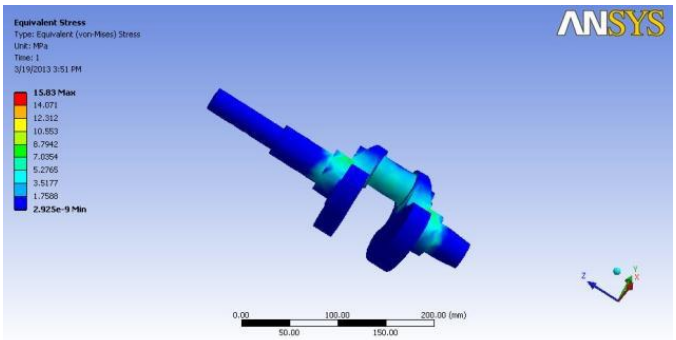

Fig.4 crankshaft voin-misses stress The maximum stress induced in the crankshaft is 15.83 Mpa at the crankpin neck surface. Minimum stress 2.925e-9 Mpa.[6]

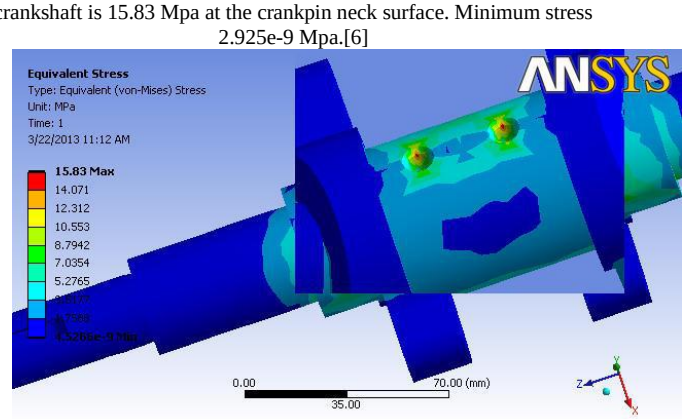

Fig.5 Maximum stress induced in the crankpin area Maximum stress induced in the Crankshaft is 15.83Mpa at the fillet areas. [6]

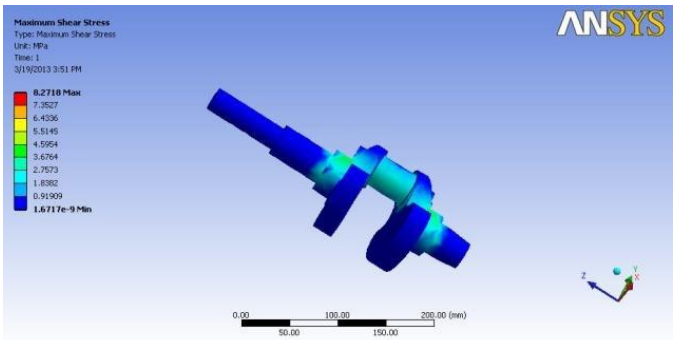

Fig.6 crankshaft shear stress The Maximum shear stress induced in the crankshaft is 8.2718Mpa, Minimum stress induced is1.6717e-91Mpa.[6]

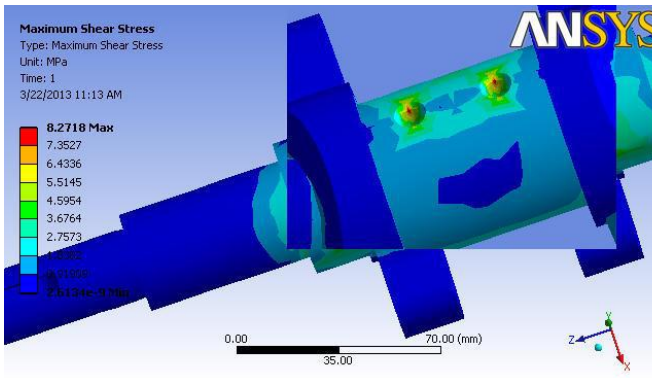

Fig.7 Maximum shear stress in the crankshaft. The Maximum shear stress is 8.271Mpa at the crankpin area.[6]

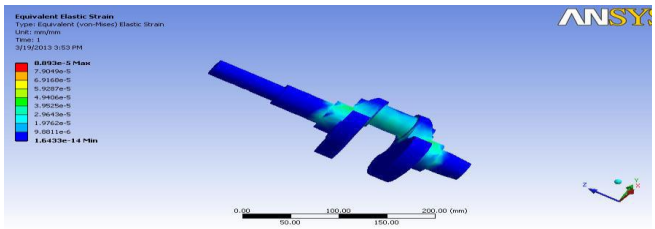

Fig.8 crankshaft (cast Iron) elastic strain Maximum Strain in the crankshaft is 8.893e-5.[6]

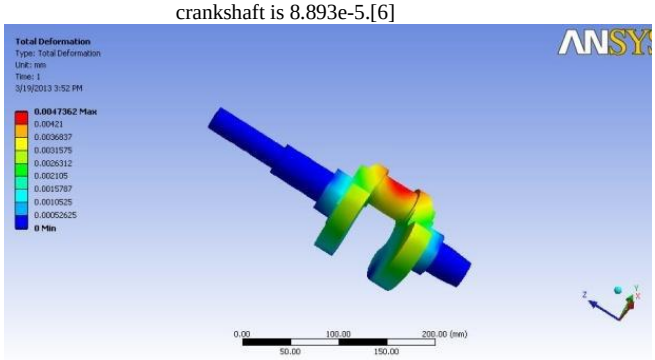

Fig.9 crankshaft total deformation [6]

X. CONCLUSION

Validated Results:-

S.No	Type of stress	Theoretical	ANSYS results
1	Von-misses stress(N/mm²)	19.6	15.83
2	Shear stresses (N/mm²)	9.28	8.271

- 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.

ACKNOWLEDGEMENT

We are thankful to staff of Department of Mechanical Engineering of DPCOE Wagholi, Pune and also thankful to Sinhgad Institute of Technology, Narhe for their constant support and guidance.

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