STUDY AND ANALYSIS OF CRANKSHAFT BELONGING TO A SINGLE CYLINDER SI ENGINE

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Abstract-

A crankshaft is a major functional part of the SI engine which is used to transfer the power from the engine to the various output devices such as the gearbox, lubrication oil pump, water pump etc. The life of the component is a major parameter for setting the maintenance schedules for repair and replacement. Fatigue is a major source of failure of the components that are subjected to cyclic loading. FEA is a most widely used tool to perform various analyses on any automotive component, designed in the recent days. It is customary to perform FEA on every crankshaft (re)designed. The current work focuses on finding the fatigue life of the component. A model of the component is discretized to FE model using a popular pre-processor: Hypermesh. The FE model is solved using ANSYS. A transient analysis of the crankshaft with the load varying (in magnitude and direction) throughout one cycle is carried out. The stress in the analysis is used as input for finding the fatigue life of the crank shaft. The result includes the stresses due to the crankshaft and the fatigue life factor of the crankshaft. The result obtained will be useful for finding the life of the component and to specify the schedule for maintenance for repair and/or replacement. The analysis procedure will be standardized to be performed in other optimized models of the crankshaft.

Key words: crankshaft, FEA, fatigue analysis, dynamic analysis, cyclic loading.

I. INTRODUCTION

The crankshaft, sometimes casually abbreviated to crank, is the part of an engine which translates reciprocating linear piston motion into rotation. To convert the reciprocating motion into rotation, the crankshaft has "crank throws" or "crankpins", additional bearing surfaces whose axis is offset from that of the crank, to which the "big ends" of the connecting rods from each cylinder attach.

The finite element method (FEM) (sometimes referred to as finite element analysis) is a numerical technique for finding approximate solutions of partial differential equations (PDE) as well as of integral equations. The solution approach is based either on eliminating the differential equation completely (steady state problems), or rendering the PDE into an approximating system of ordinary differential equations, which are then numerically integrated using standard techniques such as Euler's method, Runge-Kutta, etc.

Fatigue is the progressive and localized structural damage that occurs when a material is subjected to cyclic loading. The maximum stress values are less than the ultimate tensile stress limit, and may be below the yield stress limit of the material.

II. PROBLEM DEFINITION

The crankshaft is optimized to various proportions according to many parameters by several authors. The problem lies in the life of the component after optimization is done. It is required to verify the life of the part such that the maintenance of the part is taken care in time to reduce the possibility of failure of the component during working.

III. METHODOLOGY

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.

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.

Finite element analysis is necessary to obtain the stresses in the crankshafts due to the relatively complex geometry. The geometry led to a lack of symmetry at the top and bottom of the crankpin in the forged steel crankshaft in spite of cross-section symmetry, which could not be accounted for in the analytical stress calculations. The lack of symmetry at the top and bottom of the crankpin in the forged steel crankshaft was confirmed with experimental strain gage results. Experimental stress and FEA results showed close agreement, within 7% difference.

IV. INPUT DETAILS

The combustion pressure is input to the crankshaft. Thus, the cylinder pressure is converted into the force on the crank shaft [refer fig.1]. The variations in the curve with reference to the angle is converted to variation of force with respect to time by using the RPM of the crank shaft.

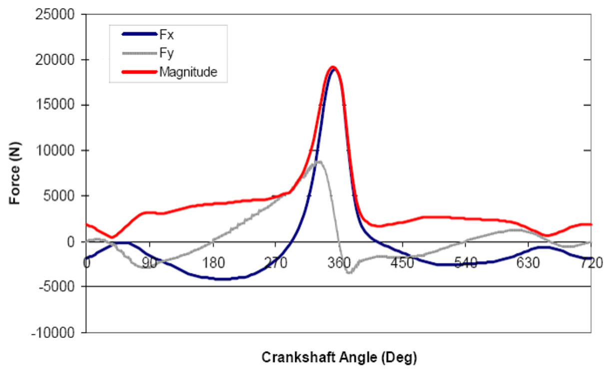

Fig. 1. Variation of the force components over one complete cycle at the crank end of the connecting rod defined in the local/rotating coordinate system at crankshaft speed of 2000 rpm.

The life of the component under scrutiny is dependant upon various parameters. The parameters that constitute to the fatigue performance of forged steel and cast iron are tabulated below.

Table 1. Basic fracture property.

Cyclic Properties	Forged Steel		Cast Iron	
Fatigue strength coefficient, σ_f', MPa (ksi)	1124	163	927	(134)
Fatigue strength exponent, b	-0.079		-0.087	
Fatigue ductility coefficient, ε_f'	0.671		0.202	
Fatigue ductility exponent, c	-0.597		-0.696	
Cyclic yield strength, YS', MPa (ksi)	505	73	519	(75)
Cyclic strength coefficient, K', MPa (ksi)	1159	168	1061	(154)
Cyclic strain hardening exponent, n'	0.128		0.114	
$S_f = \sigma_f'(2N_f)^b$ at $N_f = 10^6$, MPa (ksi)	359	(52)	263	(38)
Average E' Gpa (ksi)	204	(31,437)	174	(25,229)

Table 2. Fatigue property

Monotonic Properties	Forged Steel		Cast Iron	
Average Hardness, HRC	23		18	
Average Hardness, HRB	101		97	
Modulus of elasticity, E, Gpa (ksi)	221	(32,088)	178	(25,838)
Yield Strength (0.2%offset), YS, MPa (ksi)	625	(91)	412	(60)
Ultimate strength, S_u, MPa (ksi)	827	(120)	658	(95)
Percent elongation, %EL	54%		10%	
Percent reduction in area, %RA	58%		6%	
Strength coefficient, K, MPa (ksi)	1316	(191)	1199	(174)
Strain hardening exponent, n	0.152		0.183	
True fracture strength, σ_f, MPa (ksi)	980	(142)	658	(95)
True fracture ductility, ε_f	87%		6%	

V. CURRENT WORK

The modeling is done using Pro-Engineer and the decretization by using Hypermesh. These are commercially available software for the specified purposes.

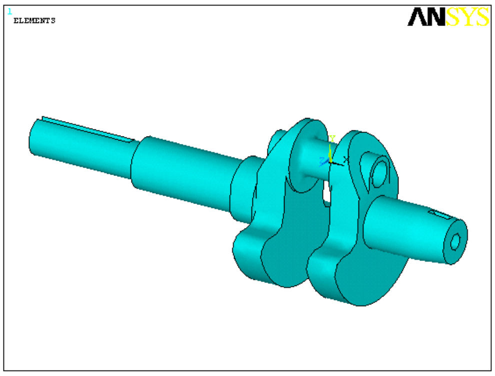

Fig. 2. Surface model of the cast iron crankshaft

A tetrahedral mesh with elements of second order is created for the model. The model is input into ANSYS and a transient analysis is performed by giving the load curve.

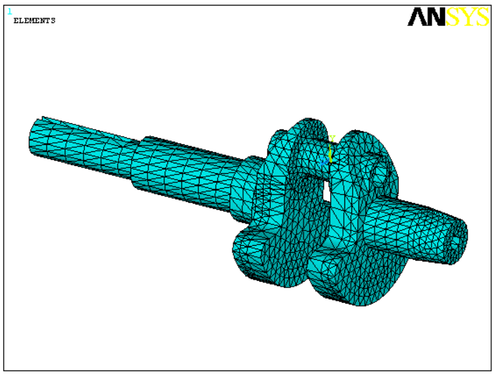

Fig. 3. The Tetrahedral mesh model.

The bearing areas are constrained in such a manner that the bearing that is connected to the timing gears is fully constrained and the bearing connected to the flywheel is constrained for radial motions and all rotations.

VI. RESULTS

The various stress and strain results are arrived from the model. The results across the section for better understanding are provided. The stresses are high at the radii of the shaft. The life at those locations is found to be 4.3285 E+07 cycles from the input S-N curves.

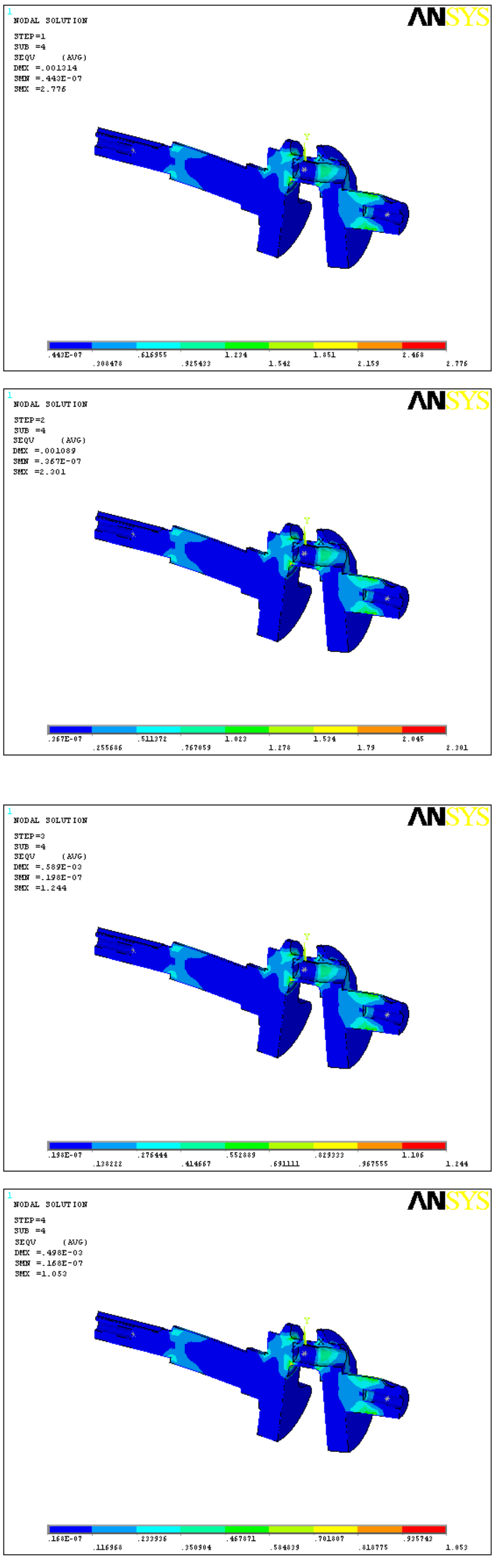

Fig. 4, 5, 6,7. The stress plots at various locations at various steps of the cycle.

VII. CONCLUSION

The life of the original model is found out in this work. The scope is extending the same procedure to a optimized model and to compare the life of both the models.

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