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Stress Analysis and Optimization of Crankshafts Subject to Dynamic Load

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Abstract: Stress Analysis and Optimization of Crankshafts Subject to Dynamic Loading The main objective of this study was to investigate weight and cost reduction opportunities for a forged steel crankshaft. The need of load history in the FEM analysis necessitates performing a detailed dynamic load analysis. Therefore, this study consists of three major sections: (1) dynamic load analysis, (2) FEM and stress analysis, (3) optimization for weight and cost reduction. In this study a dynamic simulation was conducted on two crankshafts, cast iron and forged steel, from similar single cylinder four stroke engines. Finite element analysis was performed to obtain the variation of stress magnitude at critical locations. The pressure-volume diagram was used to calculate the load boundary condition in dynamic simulation model, and other simulation inputs were taken from the engine specification chart. The dynamic analysis was done analytically and was verified by simulations in ADAMS which resulted in the load spectrum applied to crankpin bearing. This load was then applied to the FE model in ABAQUS, and boundary conditions were applied according to the engine mounting conditions. The analysis was done for different engine speeds and as a result, critical engine speed and critical region on the crankshafts were obtained. Stress variation over the engine cycle and the effect of torsional load in the analysis were investigated. Results from FE analysis were verified by strain gages attached to several locations on the forged steel crankshaft. Results achieved from aforementioned analysis were used in optimization of the forged steel crankshaft. Geometry, material, and manufacturing processes were optimized considering different constraints, manufacturing feasibility, and cost. The optimization process included geometry changes compatible with the current engine, fillet rolling, and the use of microalloyed steel, resulting in 18% weight reduction, increased fatigue strength and reduced cost of the crankshaft, without changing connecting rod and/or engine block. A 26% weight reduction is also possible considering changes in the main bearings and the engine block.

Keywords: FEM,ABAQUS, ADAMS, FE

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. This study was conducted on a single cylinder four stroke cycle engine. Two different crankshafts from similar engines were studied in this research. The finite element analysis was performed in four static steps for each crankshaft. Stresses from these analyses were used for superposition with regards to dynamic load applied to the crankshaft. Further analysis was performed on the forged steel crankshaft in order to optimize the weight and manufacturing cost. Figure 1.1 shows a typical picture of a crankshaft and the nomenclature used to define its different parts.

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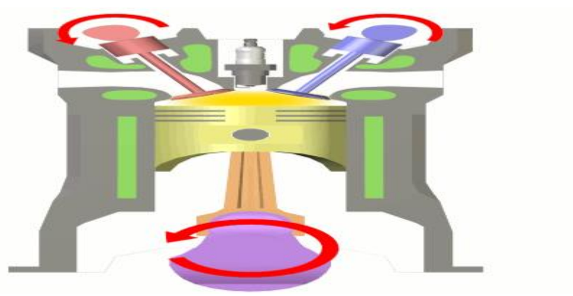
II. REVIEW

Their study presents a literature survey focused on fatigue performance evaluation and comparisons of forged steel and ductile cast iron crankshafts. In their study, crankshaft specifications, operation conditions, and various failure sources are discussed. Their survey included a review of the effect of influential parameters such as residual stress on fatigue behavior and methods of inducing compressive residual stress in crankshafts. The common crankshaft material and manufacturing process technologies in use were compared with regards to their durability performance. This was followed by a discussion of durability assessment procedures used for crankshafts, as well as bench testing and experimental techniques. In their literature review, geometry optimization of crankshafts, cost analysis and potential cost saving opportunities are also briefly discussed.



Side view of the engine block at the time of combustion

(http://www.wikipedia.org/, 2007

III. DYNAMIC LOAD ANALYSIS OF THE CRANKSHAFT

The crankshaft experiences a complex loading due to the motion of the connecting rod, which transforms two sources of loading to the crankshaft. The main objective of this study was the optimization of the forged steel crankshaft which requires accurate magnitude of the loading on this component that consists of bending and torsion. The significance of torsion during a cycle and its maximum compared to the total magnitude of loading should be investigated to see if it is essential to consider torsion during loading or not. In addition, there was a need for obtaining the stress variation during a loading cycle and this requires FEA over the entire engine cycle. The main objective of this chapter is to determine the magnitude and direction of the loads that act on the bearing between connecting rod and crankshaft, which was then used in the FEA over an entire cycle. An analytical approach was used on the basis of a single degree of freedom slider crank mechanism. MATLAB programming was used to solve the resulting equations.

Configuration of the engine to which the crankshaft belongs.

Crankshaft radius	36mm
Piston diameter	87mm
Mass of the connecting rod	0.281kg
Mass of piston assembly	0.416kg
Connecting rod length	120.76mm
Izz of connecting rod about the center of gravity	0.662kg-m^2
Distence of C.G.of connecting rod from crank end center	28.6mm
Maximum gas pressure	34Bar

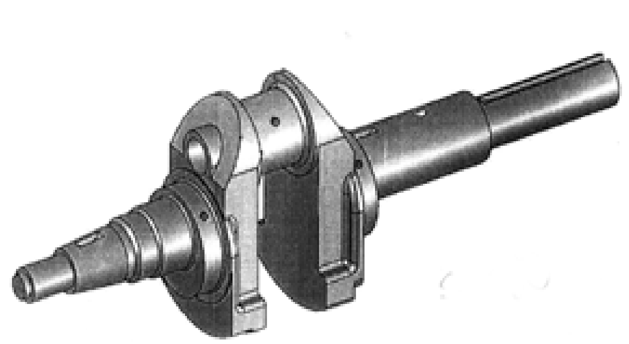


Figure 4.3 Generated geometry of the forged steel crankshaft.

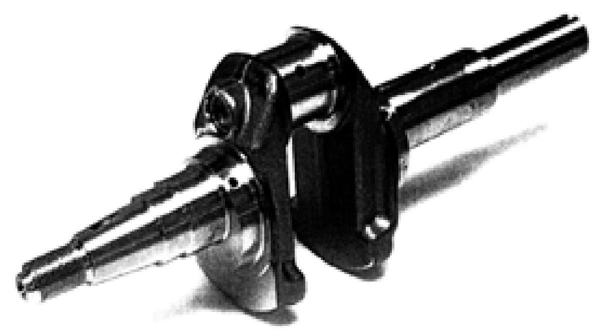


Figure 4.4 Picture of the actual forged steel crankshaft from which the digitized model was generated.

Table 4.2 Stress components in MPa at locations labeled in Figure 4.18 on the forged steel crankshaft, resulting from unit load of 1 kN.

Location Number	von Mises	S11	S22	S33	S12	S13	S23
Load direction: +F_y							
1	3.58E-02	2.37E-03	-7.09E-03	-2.76E-02	-1.07E-03	3.84E-03	1.29E-02
2	[illegible]	4.81E-01	3.50E 01	1.11261	3.68E-03	-6.22E-05	-6.05E-01
3	5.99E-01	1.78E-01	[illegible]	[illegible]	[illegible]	[illegible]	1.27E-01
4	2.07E-01	1.23E-01	2.51E 02	2.21E 01	[illegible]	[illegible]	-7.90E-02
5	2.59E-01	-1.96E-01	[illegible]	[illegible]	[illegible]	[illegible]	5.07E-02
6	5.78E-01	1.77E-01	6.18E-02	5.96E-01	[illegible]	[illegible]	1.79E-01
7	[illegible]	[illegible]	[illegible]	-4.16E-01	[illegible]	6.96E-03	2.09E 01
Load direction: -F_y							
2	8.90E-02	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
3	6.51E-01	1.99E-03	[illegible]	1.69E-04	-7.01E-02	-3.69E-01	-6.98E-04
4	[illegible]	[illegible]	8.37E-05	3.07E-02	7.30E 03	1.86E 01	1.17E 02
5	3.04E-01	2.41E-02	2.79E-05	[illegible]	[illegible]	-1.70E-01	[illegible]
7	6.58E-01	-4.19E 03	5.50E 03	[illegible]	[illegible]	3.78E-01	[illegible]
Load direction: F_x							
2	[illegible]	[illegible]	[illegible]	[illegible]	-5.23E-03	1.90E 03	3.56E 01
3	8.57E-02	-8.51E-03	6.18E-09	-2.51E-02	-2.35E-03	[illegible]	[illegible]
4	[illegible]	6.09E-02	4.54E 03	2.61E 01	[illegible]	-4.63E-03	[illegible]
5	2.59E-01	7.15E-02	6.88E-03	2.76E-01	[illegible]	[illegible]	[illegible]
7	4.88E 01	1.29E 01	1.15E 01	[illegible]	5.75E-03	-3.26E-03	-2.11E-01
Load direction: +F_x							
2	[illegible]	[illegible]	[illegible]	[illegible]	1.88E 02	1.72E 02	1.09E-02
3	6.46E-01	1.34E-02	-2.19E-03	[illegible]	6.59E-02	[illegible]	[illegible]
4	[illegible]	1.28E-03	5.53E-03	3.10E 02	7.36E 02	[illegible]	1.13E-03
5	3.01E-01	2.13E-02	[illegible]	[illegible]	[illegible]	[illegible]	-4.82E-01
7	6.31E 01	6.81E 02	-2.49E-03	-3.51E-03	[illegible]	-6.75E-01	[illegible]

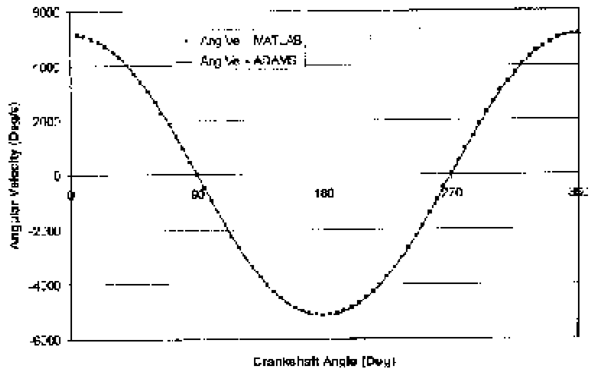


Figure 3.3 Angular velocity of link BC for the slider-crank mechanism. A comparison of the results obtained by MATLAB programming and ADAMS/View software at 2500 rpm crank speed

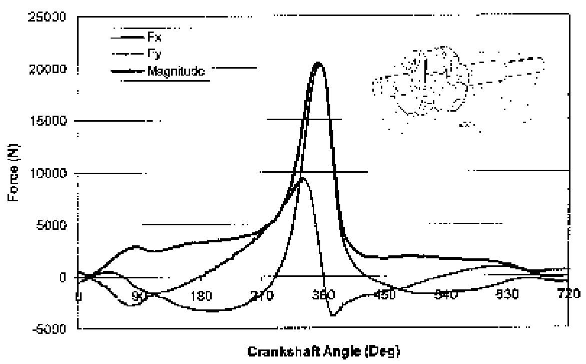


Figure 3.15 Variation of the force components over one complete cycle at the crank end of the connecting rod defined in the local/rotating coordinate system at crankshaft speed of 2000 rpm.

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.

Considering torsional load in the overall dynamic loading conditions has no effect on von Mises stress at the critically stressed location. The effect of torsion on the stress range is also relatively small at other locations undergoing torsional load. Therefore, the crankshaft analysis could be simplified to applying only bending load. Superposition of FEM analysis results from two perpendicular loads is an efficient and simple method of achieving stresses for different loading conditions according to forces applied to the crankshaft from the dynamic analysis. Experimental stress and FEA results showed close agreement, within 7% difference. These results indicate non-symmetric bending stresses on the crankpin bearing, whereas using analytical method predicts bending stresses to be symmetric at this location. The lack of symmetry is a geometry deformation effect, indicating the need for FEA modeling due to the relatively complex geometry of the crankshaft. 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.

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