

# Optimization of a Forged Steel Crankshaft Subject to Dynamic Loading

**Farzin H. Montazersadgh**

Formerly Graduate Student at The University of Toledo

**Ali Fatemi**

The University of Toledo

Copyright © 2008 SAE International

## ABSTRACT

In this study a dynamic simulation was conducted on a forged steel crankshaft from a single cylinder four stroke engine. Finite element analysis was performed to obtain the variation of the stress magnitude at critical locations. The dynamic analysis resulted in the development of the load spectrum applied to the crankpin bearing. This load was then applied to the FE model and boundary conditions were applied according to the engine mounting conditions. Results obtained from the aforementioned analysis were then used in optimization of the forged steel crankshaft. Geometry, material, and manufacturing processes were optimized using different geometric constraints, manufacturing feasibility, and cost. The first step in the optimization process was weight reduction of the component considering dynamic loading. This required the stress range under dynamic loading not to exceed the magnitude of the stress range in the original crankshaft. Possible weight reduction options and their combinations were considered. The optimization and weight reduction were considered in an interactive manner and evaluated by manufacturing feasibility and cost. The optimization process resulted in an 18% weight reduction, increased fatigue strength, and a reduced cost of the crankshaft.

## INTRODUCTION

The crankshaft is one of the largest components in the internal combustion (IC) engine and has a complex geometry consisting of cylinders as bearings, and plates as the crank webs. Geometry section changes in the crankshaft cause stress concentration at fillet areas where bearings are connected to the crank webs. In addition, this component undergoes both torsional and bending loads during its service life. Therefore, fillet areas are locations that are subject to the most critical stresses during the service life of the crankshaft. As a result, these locations are a common fatigue failure site of crankshafts. Since the crankshaft has a complex

geometry for analysis, finite element models have been considered to provide an accurate and reasonable simulation. Because crankshafts are among large volume production components in the IC engine, weight and cost reductions of this component are very effective in increasing the fuel efficiency and reducing the overall cost of the engine.

Since fatigue crack initiation and fracture at the fillets is one of the primary failure mechanisms of automotive crankshafts, the fillet rolling process has been used to improve the fatigue life of crankshafts for many years. The fillet rolling process induces compressive residual stresses within the fillet surface. The compressive residual stress lowers the fatigue driving forces due to operating loads near the fillet surface and consequently increases the fatigue life of the crankshaft.

Kamimura (1985) [1] performed a study on the effect of fillet rolling on fatigue strength of a ductile cast iron crankshaft. Bench tests were conducted on crankshaft pin samples with a fatigue evaluation on test pieces in order to study the fatigue strength of fillet rolled crankshafts and specimens. This study showed that an optimum deep rolling method could increase the bending fatigue strength by 83% over conventional ductile iron crankshafts that were not fillet rolled.

Park et al. (2001) [2] showed that without any dimensional modification, the fatigue life of a crankshaft could be improved significantly by applying various surface treatments. Fillet rolling and nitriding were the surface treatment processes that were studied in this research. Their study showed that the standard base sample had a fatigue limit of 10 kN, while fillet rolled specimens with 500 kgf load exhibited a 14 kN fatigue limit (i.e. 40% increase in fatigue limit). With 900 kgf rolling load, the fatigue limit increased to more than 18 kN (i.e. 80% increase in fatigue limit). These experimental data clearly indicate that fillet rolling can dramatically increase the fatigue performance of crankshafts. Although higher rolling force results in

better fatigue strength as a result of inducing higher compressive stress on the fillet surface, the load should not be so high as to cause excess plastic deformation. The specimens prepared from the crankshaft in the Park et al. study found the optimum level of rolling force was experimentally to be between 700 and 900 kgf.

An extensive study was performed by Nallicheri et al. on material alternatives for the automotive crankshaft based on manufacturing economics [4]. Steel forgings, nodular cast iron, micro-alloy steel forgings, and austempered ductile iron (ADI) casting were considered as manufacturing options to evaluate the cost effectiveness of using these alternatives for crankshafts. It was concluded that the production volume of the crankshaft and the requirements of the engine were predominant factors in a cost effective production option for this application. The selection of the best alternative is dependent upon the production volume. At production volumes above 200,000 parts/year, microalloyed steel forgings offered the most cost effective high performance crankshaft. The ADI crankshafts were cost effective at low production runs (below 180,000 parts/year). Based on these assumptions, raw material was about 30% of the final cost, whereas machining cost was 47% for the forged steel crankshaft. The raw material cost of the microalloyed steel forging crankshaft was 38%, but the machining percentage cost was 43% which reduced the cost of the final component. In addition, the heat treatment process was eliminated for the microalloyed steel crankshaft, resulting in a further cost reduction.

A study was performed by Hoffmann and Turonek to examine the cost reduction opportunities associated with forged steel in which the raw material and machinability were the primary factors evaluated [5]. Materials evaluated in their study included medium carbon steel SAE 1050 (CS) and medium carbon alloy steel SAE 4140 (AS) grades using a sulfur level of 0.10%, (CS-HS and AS-HS), and two micro-alloy grades (MA1 and MA2). The material selection was based on fatigue strength requirements and potential cost benefits. The microalloy grades that were evaluated offered cost reduction opportunities over the original materials. The micro-alloy grade could reduce the finished cost by 11% to 19% compared to a quenched and tempered alloy steel (SAE 4140), and by 7% to 11% compared to a quenched and tempered carbon steel (SAE 1050). In addition, the micro-alloy grades met or exceeded the fatigue strength of the original materials for the applications studied and had better machinability characteristics.

Detailed dynamic load and stress analysis of the crankshaft investigated in this study was the subject of another paper [5]. Finite element analysis was used to obtain the variation in stress magnitude at critical locations. The dynamics of the mechanism was solved using analytical techniques and the results were verified by simulation in ADAMS, which resulted in the load

spectrum applied to the crank pin bearing. The load was applied to the FE model in ABAQUS, and the boundary conditions were defined according to the engine mount design. The analysis was performed over different engine speeds and as a result the critical engine speed and critical locations on the crankshaft were obtained. Stress variation over the engine cycle and the effect of torsional load in the analysis were also investigated. Results from FE analysis were verified by attaching strain gages to several locations on the crankshaft [5].

The optimization carried out in this work was not based on only the typical geometrical optimization techniques. This is because variables such as manufacturing and material parameters could not be organized in a mathematical function according to the set of constraints such that the maximum or minimum could be defined. Instead, each optimization step was approximated based on improving fatigue resistance while considering manufacturing feasibility and maintaining dynamic balance with an aim of reducing the weight and the final cost of the component.

In this paper, a brief summary of the dynamic loading and stress analysis of the forged steel crankshaft is first considered with details provided in [5]. Then the optimization objectives, constraints, and procedure that have been used are described. This is followed by a discussion of material alternatives and the application of compressive residual stress in the fillet area of the crankpin. Finally, a brief discussion of cost reduction of the optimized crankshaft is provided.

## 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. Accurate assessment of stresses is critical to the input of fatigue analysis and optimization of the crankshaft design.

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.

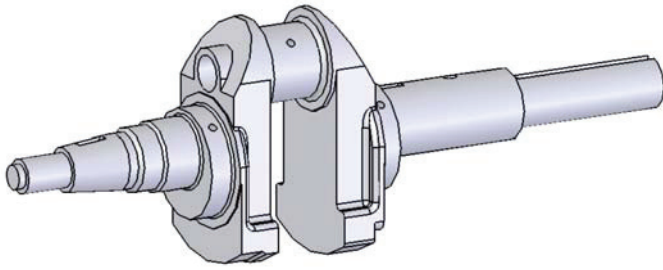


Figure 1: Digitized model of the original forged steel crankshaft geometry used.

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 torsion 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].

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 basic condition and then scaling the stresses from each unit load according to the dynamic loading. Then the corresponding stress components are added together.

Figure 2 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 to be critical in the analysis.

## 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. In addition, the bending stiffness has to be kept within permissible limits.

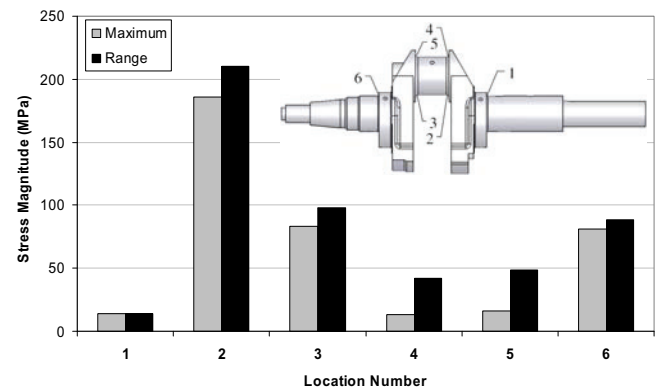


Figure 2: Maximum and range of von Mises stress at an engine speed of 2000 rpm at different locations on the crankshaft.

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:

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

Considering the functions of the crankshaft and its constraints, the following design variables were then considered in the optimization study:

- Thickness of the crank web
- Geometry of the crank web
- Increasing the inner hole diameters and depths
- Geometry changes on the outer section of the crankpin bearing

## GEOMETRY OPTIMIZATION PROCEDURE

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 subject 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.