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 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.
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.

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.
Figure 3: General flow chart of forged steel crankshaft optimization procedure.

The general flow chart of the optimization process is shown in Figure 3. Objective function, design variables, and constraints are summarized in this figure and it is shown that the optimization process consisting of geometry modifications, manufacturing process considerations, and material alternatives was performed simultaneously.

Local shape optimization techniques were applied to different locations of the crankshaft to lower the weight. After each optimization step the counter weights were balanced in order to achieve an accurate estimate of the weight reduction. Also, following each shape optimization iteration the optimized component was investigated for design feasibility and examined to determine whether the design was the best possible option. In a feasible design, the component stresses fall within the stress limits and bending stiffness does not change significantly, while the manufacturability is still economical.

According to FE analysis, the blue locations shown in Figure 4 have low stresses during the loading cycle and have the potential for material removal and weight reduction. It should be noted that the effect of mean stress on the results was negligible. Therefore, all of the stresses under consideration and discussed here are with reference to the stress ranges.

Figure 4: Stress distribution under critical loading condition at the crank angle of 5 degrees after TDC.

Several cases of geometric modifications were considered. Since maintaining dynamic balance is a key concern in the optimization of this component, the first step was to remove material symmetric to the central axis without compromising the dynamic balance of the crankshaft. On the far right side of the crankshaft shown in Figure 1 there is a threaded hole. The depth of this hole does not affect the function of the crankshaft.
Therefore, this hole could be drilled as far as possible in the geometry (Case 1 in Figure 5).

Another optimization step which does not require any complicated changes in the geometry is increasing the diameter of the crankpin hole (Case 2 in Figure 5). An increase in the diameter of this hole will result in decreasing the moment of inertia of the cross section. Therefore, in order to avoid increasing the stress level at the fillet area, the fillet radius has to be increased. This increase in the fillet radius does not affect the connecting rod geometry since the connecting rod has sufficient clearance. Applying these changes to the crankshaft causes the center of mass to move toward the counter weights. In order to balance the modified crankshaft, material has to be removed from the counter weight.

A significant percentage of the weight in the crankshaft is in the crank counterweight or web mass. Therefore, reducing the weight of this section could result in an improvement in weight reduction of the component. Reducing the web thickness is another optimization opportunity that was performed on the crankshaft web (Case 3 in Figure 5). As a result of this change in the crank web, the center of gravity moves toward the crankpin bearing. In order to dynamically balance the crankshaft, material has to be added to the counter weights which can be accomplished by increasing the radius. This option was restricted to the clearance between the piston and counter weights of the crankshaft. Since the current geometry is designed with a specific clearance between the counter weights and the piston, this optimization effort could not be implemented without additional changes to the piston.

Redesigning the crank web and removing material from this section was the next optimization case under consideration while keeping the feasibility of the manufacturing process in mind. This requires not having negative slopes. The crank web was then modified such that no changes in the counter weights would be necessary (Case 4 in Figure 5).

Other optimization cases considered in this study, but not implemented due to either a small decrease in weight and/or manufacturing difficulty or cost, were:

- Redesigning the crankpin geometry
- Removal of material from the center of the crank web, symmetric to the central axis
- Removal of semi-circular material from the center of the crank web, symmetric to the central axis
- Eccentric crankpin hole

Since each optimization case was studied individually, further analysis was needed by considering a combination of these cases. Options for a redesigned crankshaft were developed such that as many optimization cases as possible could be applied.

Figure 5: Optimized crankshaft geometry.

FE models of possible combinations were created and FE analysis with dynamic load was considered for each combination. Results of the stress range for each of these combination models showed that the critical location (location 2 in Figure 2) does not change as a result of these geometry modifications. A comparison plot of stress range ratios for the different case combinations of the optimized crankshaft, as compared to the stress range in the original crankshaft at the critical location, is shown in Figure 6. The horizontal line at 1 stands for the original crankshaft.

![Stress Range Deflection Weight Difference with Original Crankshaft](chart)

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Figure 6: Stress range ratio at critical location, radial deflection at mid section of the crankpin, and weight comparison with the original crankshaft. The horizontal line at 1 stands for the original crankshaft with 188 MPa for stress range, with $2.36 \times 10^{-2}$ mm for radial deflection, and with 3.72 kg for weight.

The counterweight radius for Case 3 is larger than the original crankshaft which causes interference between the counterweight and the piston. However, a combination of Cases 1, 2, and 4 results in increased clearance between the counter weights and the piston allowing for the increased counter weight radius in Case 3. The material removal from the counter weights in the combination Cases 1, 2, and 4 will move the center of
gravity toward the counter weights. Since reducing the web thickness is not symmetric with respect to the central axis and results in more material removal from the side of the counter weights, the center of gravity will move toward the crankpin. At a certain reduction in web thickness, the center of gravity will be located on the main crankshaft axis, which means the crankshaft becomes dynamically balanced. The combination of Cases 1, 2, 3, and 4 results in a 12% weight reduction in comparison with the original crankshaft. As can be seen in Figure 6, the stress range at the critical location for the combination of Cases 1, 2, 3, and 4 are reduced slightly. Crankshaft deflection does not increase more than 15% for the combination of these cases. Therefore, the stiffness of the optimized crankshaft is within the acceptable limit. Deflections of different optimized crankshaft cases are shown in Figure 6.

**FILLET ROLLING AND MATERIAL ALTERNATIVES**

The next step in the optimization process was an attempt to modify the production steps in order to reduce the cost or improve the performance of the original crankshaft. Further improvement of the performance could result in further geometry changes and weight reduction. A potential modification for this improvement and subsequent weight reduction is the addition of compressive residual stress to the fillet area of the crankpin which is the critical (i.e. failure) area.

Due to lack of experimental information, the precise magnitude of the residual stress that could be induced in the crankshaft geometry under investigation could not be determined. Studies by Kamimura [1], Park et al. [2], and Chien et al. [6] showed that inducing compressive residual stress increases the fatigue strength of the crankshaft significantly. Based on these studies, the application of compressive residual stress in the fillet area of the crankshaft can increases the fatigue strength by 40% to 80%, depending on the material properties, crankshaft geometry, and the applied rolling force.

With regard to material options, one of the most common alternatives for forged steel is microalloyed (MA) steel. Pichard et al. [7] performed a study on a modified microalloyed steel with a titanium addition for the production of forged crankshafts. The use of MA steel enables elimination of any further post-forging heat treatment, resulting in shorter manufacturing process with a subsequent increase in the forged crankshaft productivity. The choice of this MA steel for a crankshaft application was based on the 35MV7 steel grade, with a typical composition of 0.35C, 1.8Mn, 0.25Si, 0.12V with an addition of the microalloying element Ti. Based on the results of their study, the 35MV7 control-cooled microalloyed steel showed similar tensile and rotating bending fatigue behavior to an AISI 4142 quenched and tempered steel. In addition, the machinability of the microalloyed steel could be improved by about 40% in turning and about 160% in drilling [7]. Therefore, the use of microalloyed steel can reduce the final cost of the crankshaft by eliminating the heat treatment process and increasing machinability.

**FINAL OPTIMIZED CRANKSHAFT**

Considering the manufacturing processes, the geometry of the crankshaft could be modified further to take advantage of the results of improved fatigue strength due to fillet rolling and/or the use of microalloyed steel. Further modification to the crankpin geometry is possible. Increasing the crankpin hole is an option which does not influence the manufacturing process and is not an expensive process. Increasing the hole diameter from the original 18.3 mm (0.72 inch) to 25.4 mm (1 inch) and reducing the crank web thickness in order to maintain dynamic balance of the crankshaft, will cause the stress range at the critical location to increase by 7%. This increase is easily compensated for by the beneficial effect of the compressive residual stress from fillet rolling. This modification is shown as Case 1, 2, 3, and 4 in Figure 6. Since the wall thickness in the crankpin area is limited, further increasing the hole diameter to larger than 25.4 mm was not possible because sufficient material is needed to restrict plastic deformation in the rolling process to produce residual stress.

These modifications reduce the weight of the original crankshaft by 18%. The final optimized geometry is shown in Figure 5. It should be noted that the fatigue strength of the optimized crankshaft is significantly higher than the original crankshaft due to a slight increase in the stress range of 7% at the fillet and a significant increase on the order of 40% to 80% in fatigue strength due to fillet rolling, as discussed earlier. Figure 7 shows the stress range comparisons at different locations shown in Figure 2, between the original and the optimized crankshafts. As can be seen in this figure, the critical location is still at the fillet and the increase in stress at other locations is not significant.

With regard to the cost of the optimized crankshaft, this is affected by the geometry changes and weight reduction, modification in the manufacturing process,
and the use of MA steel. The optimized geometry requires redesign and remanufacturing of the forging dies used. The geometry parameters that influence machining and the final cost of the component include an increase in the drilling process. This is because the drilled holes in Cases 1 and 2 are redesigned to have larger diameters and the bore in Case 1 is modified to have more depth than the original bore. The application of compressive residual stress by a fillet rolling process is a parameter in the manufacturing process that will add to the cost of the finished component.

Although a microalloy steel grade is somewhat more expensive than hot-rolled steel bar, the heat treatment cost savings are significant enough to offset this difference (Wicklund, 2007 [8]). In addition, microalloyed steel has 5% to 10% better machinability than quenched and tempered steel, resulting in reduced machining costs due to enhanced production rates and longer tool life (Nallicheri et al. [4]). A consideration of these factors, along with the reduced material cost due to the 18% weight reduction, indicates a reduction in the total cost of the forged steel crankshaft.

SUMMARY

An optimization study was performed on a forged steel crankshaft that considered the geometry, performance, manufacturing process, and cost. A major constraint of this optimization was for the optimized crankshaft to replace the original crankshaft in the engine without any changes to the engine block or the connecting rod. An optimization in the geometry included local changes at different locations on the crankshaft, which were then combined to obtain the final optimized geometry.

The optimization resulted in an 18% weight reduction of the forged steel crankshaft. This was achieved by changing the dimensions and geometry of the crank counterweights while maintaining dynamic balance of the crankshaft. The optimization that was developed did not require any changes to the engine block or connecting rod.

Adding fillet rolling was considered in the manufacturing process. Fillet rolling induces compressive residual stress in the fillet areas, which results in a significant increase in fatigue strength of the crankshaft, and in turn, significantly increases the fatigue life of the component. The use of a microalloyed steel as an alternative material to the current forged steel composition results in the elimination of the heat treatment process. In addition, when considering the improvement in machinability of the microalloyed steel along with the reduced material cost because of an 18% weight reduction, the overall cost of the forged steel crankshaft can be reduced.

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