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Evaluation of FEM based fracture mechanics technique to estimate life of an automotive forged steel crankshaft of a single cylinder diesel engine

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Abstract

Crankshaft is one of the critical components of an IC engine, failure of which may result in disaster and makes engine useless unless costly repair performed. It possesses intricate geometry and while operation experiences complex loading pattern. In IC engines, the transient load of cylinder gas pressure is transmitted to crankshaft through connecting rod, which is dynamic in nature with respect to magnitude and direction. However, the piston along with connecting rod and crankshaft illustrate respective reciprocating and rotating system of components, the dynamic load and rotating system exerts repeated bending and shear stress due to torsion, which are common stresses acting on crankshaft and mostly responsible for crankshaft fatigue failure. Hence, fatigue strength and life assessment plays an important role in crankshaft development considering its safety and reliable operation. The present paper is based on comparative studies of two methods of fatigue life assessment of a single cylinder diesel engine crankshaft by using fracture mechanics approach viz. linear elastic fracture mechanics (LEFM) and recently developed critical distance approach (CDA). These methods predict crack growth, time required for failure and other parameters essential in life assessment. LEFM is an analytical method based on stress intensity factor which characteristics the stress distribution in the vicinity of crack tip, whereas CDA is a group of methods predicts failure using stress distance plot. The maximum stress value required for both the methods are obtained using finite element analysis. The present paper provides an insight of LEFM and CDA methods along with its benefits to the designers to correctly assess the life of crankshaft at early stage of design. This paper also gives a detailed overview of failure analysis process including theoretical methods and result integration for predicting life of components as compared to life estimation by means of software.

Keywords: Finite Element Method; Crack Propagation; Crankshaft; Fatigue; Fatigue Life Estimation

Nomenclature

| Symbol | Description |
|--------|-------------|
| a      | Crack half-length for a crack free to extend at both ends |
| C      | Coefficient in simple crack growth rate |
| K      | Stress intensity factor (Mode I) |
| L      | Critical distance |
| M      | Characteristic of slope of crack growth rate curve |

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1. Introduction

A comparative evaluation of fatigue assessment techniques on a forged steel crankshaft of a single cylinder diesel engine has been made by using fracture mechanics approach viz. linear elastic fracture mechanics (LEFM) and recently developed critical distance approach (CDA). This paper describes the mathematical modeling of crankshaft geometry for dynamic analysis using analytical and finite element method (FEM) to estimate the stress concentration regions to predict the probable location of the crack generation. Different crack lengths have been considered for the further life prediction for the failure to occur. The crankshaft being used is consisting of two web sections and one crankpin. In automotive industry, crankshaft analysis on single cylinder is done irrespective of size of crankshaft. Hence this study can be used for any number of cylinder crankshafts. This study also describes the accuracy of the model and explains the simplifications that were made to obtain an efficient FE model. Different mesh quality models and its convergence are compared to finalize on FE model to be used for the analysis. Identification of appropriate boundary conditions and loadings are also discussed and subsequent results of finite element analysis have been presented. The prediction of crack growth, time required for failure and other parameters essential in life assessment have been compared by both the methods. The stress values required for both the methods are estimated using Finite Element Analysis.

A forged steel crankshaft of weight 4.8 kg, designed for 480cc, single-cylinder diesel engine of power rating approximately 9.0 kW, has been used in the present fatigue life estimation study. It is evident that the critical locations of the crankshaft failure are nothing but the critical fillet areas of joining pin with web and web with main shaft, which are very much sensitive to the subjected repetitive fatigue loads. The figure 1 shows the theoretical model of the crankshaft with the critical locations considered for the evaluation of fatigue life. Crack lengths of various sizes have been assumed right from 0.5 mm to 8.0 mm with an interval of 0.5 mm and number of cycles of failure has been estimated. In LEFM method, the stress intensity factor has been evaluated first and compared with critical intensity factor of the material and at matching point the life is estimated. Similarly, the crankshaft has been evaluated using CDA method in which stress against distance plot from maximum stress area and characteristics length or critical length L have been found out for all the crack lengths from 0.5 mm to 8.0 mm with an interval of 0.5 mm and failure is predicted. In this study ABAQUS commercial software has been used for stress analysis to evaluate the critical stresses in the crack zone and then LEFM and CDA method have been used for fatigue life prediction and comparison. Also to evident the comparison an analytical approach has been developed to predict and compare the fatigue life.

![Fig.1. Crankshaft model with critical locations.](image)

2. Methods of solutions

2.1 Linear elastic fracture mechanics

Linear elastic fracture mechanics can be used to describe ultimate static failure of low toughness high strength materials commonly used in aerospace, automobile and other specialised applications. Under fatigue loading, for a wide range of materials, crack growth rate can be correlated by the stress intensity factor. The basics used in LEFM is to predict the fatigue life of components based on the fact that a crack already pre-exists in the component and that the life is directly dependent on the stress intensity factor, which in-turn depends on initial crack length assumed or present.
In simple cases the function of fatigue load and the crank length is defined as stress intensity factor and is expressed as:

\[ \text{Stress Intensity Factor, } K = \sigma_0 \sqrt{\pi a} \]  

(1)

The linear relation between crack growth and threshold stress intensity can be represented by Paris law.

\[ \frac{da}{dN} = C(\Delta K)^m \]  

(2)

Fatigue life of component can be found out by integrating Eq.2, which in turn evaluates the number of cycles of failure that component withstand before failure. The number of cycles of failure \( N_f \) can be obtained as below,

\[ [N]_{N_f}^{N_f} = \frac{1}{C[\sigma_{\text{max}} - \sigma_{\text{min}}]^{\pi/2}} \left[ \frac{a}{1 - (m/2)} \right]^{b_1} \]  

(3)

2.2 Critical distance approach

The Theory of Critical Distances (TCD) is the name which describes a group of methods which is used for failure prediction of short cracks as well as for stress concentrations of arbitrary geometry, by using the results of finite element analysis (FEA) or any other computer-based methods. This method uses parameter taken in front of notch or stress concentration area. Accurate prediction of failure is possible when correct stress information is provided. In this paper Point Method (PM) is used, as this method is simple to use and can effectively predict failure in component. The PM predicts that failure will occur if the stress at a distance \( L/2 \) from the notch root is equal to the plain strength of the material. The same principle will be applies for fatigue, replacing the stress with a stress range and the plain strength with the plain fatigue strength. The critical distance represents the length ahead of notch and stress along this distance reduces as it moves away from notch. The stress distance plot is plotted in which stress along with focus path defined by the user is plotted against distance of that path. This curve is used to predict failure of engineering components. In point method stress at single point is considered and critical stress value is found out at a distance \( L/2 \) from notch or stress concentration region by using stress distance plot. If the critical distance and accurate estimation of stresses are known, point method is best to determine failure.

Fig. 2. Illustration of TCD (Point Method) using Elastic stress as function of distance, the fatigue strength of specimen \( \Delta\sigma_0 \) occurs at critical distance \( L/2 \).

Following relations are used to determine critical distance

\[ L = \frac{1}{\pi} \left( \frac{K_{\text{th}}}{\sigma_0} \right)^2 \]  

(4)

The above equation (4) allows the critical distance to be expressed as a function of the fracture toughness \( K_{\text{th}} \), and also linked with critical distances (\( L/2 \)). The PM calculates a stress value and equates it to a characteristic strength for the material to consider the propagation of a crack of finite size, and thus uses the material parameters \( K_{\text{th}} \). PM will be the most convenient, if the results of the FEA are available for the component.

2.3 Finite element analysis

2.3.1 Boundary and Loading Condition

The crankshaft is constraint with a ball bearing side (pump side) from one end and with a journal bearing (flywheel side) on the other end. The ball bearing is press fit to the crankshaft and does not allow the crankshaft to have any motion other than rotation about its main central axis. Since, only 180° of the bearing surfaces facing the load direction constraint the motion of the crankshaft, this constraint was defined as a fixed semi-circular surface as wide as the ball bearing width on the crankshaft. The other side of the crankshaft is on a journal bearing. Therefore, this side was modeled as a semi-circular edge facing the load at the bottom of the fillet radius fixed in a plane perpendicular to the central axis and free to move along central axis direction. Figures 3 show these defined boundary conditions in the FE model of crankshaft. Definition of a fixed
edge is based on the degrees of freedom in a journal bearing, which allows the crankshaft to have displacement along its central axis. The distribution of load over the connecting rod bearing is uniform pressure on 120° of contact area. The dynamic load is predicted from the engines pressure and crank-angle diagram. The dynamic load from the pressure and crank-angle diagram has been used to estimate the dynamic boundary condition for the stress analysis on the forged steel crankshaft shown in Figure 3. The stress analysis has been performed using FEM based ABAQUS commercial software.

A grid independence test was performed on the FEM model with global size of 4 mm and 5 mm and the three layer mesh to capture the critical fillet regions of interest. It has been found that there were variations in the stress calculations of around 2 percent, which is very much under acceptable limits of numerical calculations. Hence, a decision was taken to use the global size of 5 mm mesh in the present study. The mesh design along with crack model for 8 mm crack is shown in Figure 4.

3. Results and discussion

The dynamic loaded stress analysis result from ABAQUS software predicts the maximum stress appears at location 4b during maximum loading situations at 360° of crank angle of the power stroke, which can be seen through figure 6. Hence, for further study on the crack stress analysis, it has been decide to create the crack at 4b location, ranging from 0.5mm crack to 8mm crack for LEFM and CDA method calculations. The crack models of various crack length along with mesh generation in and around of the crack at location 4b are shown in the Figure 5. It also shows the FEM analyzed true stress values at 360° of crank angle of the power stroke for LEFM and CDA technique for life prediction.

3.1 Life prediction of crankshaft using linear elastic fracture mechanics

In the present study to evaluate the linear elastic fracture mechanics theory for an automotive crankshaft specimen, the stress intensity factors have been calculated for every crack length in the specimen using principal stress range in Equation 2. The results obtained through LEFM method are presented in Table 1 and a graph plotted between stress intensity factor $K_i$ and crack length has been shown in Figure 7.

It is observed from the FEA analysis of the automotive crankshaft specimen that, with increase in crack length there is increase in critical stress concentration in and around the crack. The increase in stress concentration contributes in increasing the stress intensity factor $K_i$, which in turn depicts the reduction in life cycles of the crankshaft specimen. The

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**Figure 3.** Crankshaft studied under fatigue simulation.

**Figure 4.** Mesh design of crankshaft model with 8.0 mm crack length.

**Figure 5.** Stresses around the cracks for 0.5mm to 8mm.

**Figure 6.** Von-mises stress at bottom crankpin fillet area
comparison of calculated stress intensity factor $K_I$ with the experimental critical stress intensity factor $K_C$ value at every crank length shows that the crankshaft fails at the transition crack growth from 1.0 mm to 1.5 mm. The increase in stress intensity factor with respect to increase in crack length is also been depicted through Figure 7.

Table 1. Fatigue life of crankshaft specimen using LEFM.

| Sr. No | Crack length, mm | Estimated stress intensity factor, $K_I$ | Critical stress intensity factor, $K_C$ | Fatigue failure criteria |
|--------|------------------|------------------------------------------|----------------------------------------|------------------------|
| 1      | 0.0              | 0.00                                     | 36                                     | NO-FAILURE             |
| 2      | 0.5              | 18.34                                    | 36                                     | NO-FAILURE             |
| 3      | 1.0              | 28.84                                    | 36                                     | NO-FAILURE             |
| 4      | 1.5              | 37.57                                    | 36                                     | FAILURE                |
| 5      | 2.0              | 46.21                                    | 36                                     | FAILURE                |
| 6      | 3.0              | 60.80                                    | 36                                     | FAILURE                |
| 7      | 4.0              | 74.44                                    | 36                                     | FAILURE                |
| 8      | 5.0              | 87.64                                    | 36                                     | FAILURE                |
| 9      | 6.0              | 101.74                                   | 36                                     | FAILURE                |
| 10     | 7.0              | 117.26                                   | 36                                     | FAILURE                |
| 11     | 8.0              | 131.46                                   | 36                                     | FAILURE                |

The estimated number of reversal to failure of the automotive crankshaft specimen with respect to crack lengths is presented in the Figure 8. It can be observed that the life of the specimen without crack is found out to be around in the range of $7 \times 10^4$ number of reversal to failure, but with just initiation of crack in the specimen the life is reduced to the range of $6 \times 10^4$ number of reversal to failure, and further increase in the crack length in the specimen, it further decreases to around 1000 number of reversal to failure. Similarly, the stress curve derived for the same specimen along with number of reversal to failure is shown in Figure 9, which also shows the similar trend as shown in Figure 8.

3.2 Life prediction of crankshaft using critical distance approach

In critical distance point method the material is assumed to possess a characteristic material length parameter, L. The value of L can be found out by using Equation 5, utilizing two material properties such as, applied stress amplitude and the crack-propagation threshold $\Delta K_{th}$, whereas, the applied stress amplitude is defined as the range of stress (i.e. the difference between maximum and minimum stress in the cycle) at which failure occurs in a specified number of cycles. Once the critical parameters (L/2 and applied stress amplitude) are known, fatigue failure of the specimen containing stress concentration feature is estimated by examining the stresses along the path drawn normal to the underlying surface at the point of maximum stress concentration. The relevant stress parameter used is the maximum principal stress. If the stress at a distance L/2 along this path is greater than the critical stress i.e. plain strength (yield stress) of the material, the CDA predicts that a crack will propagate from that defect and the body will fail. In the present study of life prediction of automotive crankshaft specimen, the applied stresses and yield stresses are presented in Table 2 and the result shows that the plate fails at the transition crack growth from 5.0 mm to 6.0 mm.

The Figure 10 shows a graph obtained by plotting the stress as a function of distance from the notch root, taken along a line drawn normal to maximum stress point. The plot is known as stress–distance curve. The line is called as focus path; to consider the effect of stress field in the vicinity of crack. To illustrate this curve, the value of principal stress is taken from crack tip running across the specimen, normal to loading axis. It can be observed from the figure that with increase in distance from the crack there is decrease in stress value and achieves asymptotic lower equilibrium value.
Table 2. Fatigue life of crankshaft specimen using CDA.

| Sr.No. | Crack Length, mm | Distance L/2, mm | Stress at L/2, (MPa) | Yield Stress, (MPa) | Fatigue failure criteria |
|--------|------------------|-----------------|---------------------|-------------------|-------------------------|
| 1      | 0.0              | 0.000           | 267.00              | 625.0             | NO-FAILURE              |
| 2      | 0.5              | 0.058           | 411.77              | 625.0             | NO-FAILURE              |
| 3      | 1.0              | 0.118           | 457.06              | 625.0             | NO-FAILURE              |
| 4      | 1.5              | 0.176           | 485.07              | 625.0             | NO-FAILURE              |
| 5      | 2.0              | 0.235           | 515.44              | 625.0             | NO-FAILURE              |
| 6      | 3.0              | 0.352           | 550.50              | 625.0             | NO-FAILURE              |
| 7      | 4.0              | 0.475           | 579.50              | 625.0             | NO-FAILURE              |
| 8      | 5.0              | 0.603           | 604.95              | 625.0             | NO-FAILURE              |
| 9      | 6.0              | 0.706           | 634.59              | 625.0             | FAILURE                 |
| 10     | 7.0              | 0.859           | 669.27              | 625.0             | FAILURE                 |
| 11     | 8.0              | 1.000           | 692.65              | 625.0             | FAILURE                 |

Fig. 10. Stress - Normal distance plot for various crack length.

Conclusion

The dynamic loaded stress analysis of the crankshaft has been performed to predict and compare the fatigue life of the crankshaft by LEFM and CDA methods, which are based on the fracture mechanics approach and evidenced by analytical method. Rather, Ansys and nCode commercial software, which are also been used for predicting the fatigue life but are based on the stress and strain method, hence are not been used in the present study of comparative fatigue life prediction. The study finally depicts the following pertinent features.

1. Critical 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.

2. Analytical and FEA results does not show close agreement, because FEA results indicate non-symmetric bending stresses on the crankpin bearing, whereas analytical method predicts bending stresses to be symmetric at this location. Also, there is no provision of incorporating explicitly the crack effects into the analytical calculations. The lack of symmetry is a geometry deformation effect, indicating the need for FEA modeling due to relatively complex geometry of the crankshaft. Also the lack of crack modeling for the purpose of capturing the crack and its growth, indicating the need for FEA modeling.

3. It has been observed that the predictions from LEFM under predict the life of the component as compared to CDA approach in predicting the life of the crankshaft at the transition from one crack length to other.

4. The present study provides an insight of LEFM and CDA methods along with its benefits to the design engineers to correctly assess the life of crankshaft at early stage of design. This study also gives a detailed overview of failure analysis process including analytical methods and result integration for predicting life of components as compared to life estimation by means of analysis tools.

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