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Fatigue Life Prediction Of Crankshaft Based On Strain Life Theories

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ABSTRACT

Fatigue analysis can be performed using one of the three basic methodologies such as stress-life theory, strain-life theory, and crack growth approach. These techniques are developed to determine the number of cycles to failure. Stress-life theory suitable when elastic stresses and strains are considered. However, for the components having nominal cyclic elastic stresses and plastic deformation, local strain-life theory is used for predicting the fatigue life. In the present work, fatigue behaviour of forged steel crankshaft, subjected to fully reversible cyclic loading, is analyzed using the strain-life theories. The analyses are aimed to identify the critical location through Finite Element Fatigue Analysis (FEFA) and, to predict the fatigue life of crankshaft. The modelling of crankshaft is carried out in parametric Pro/Engineer software whereas ANSYS workbench is used for the Finite Element Analysis (FEA). Maximum Von Mises stresses criterion is used for predicting the failure of crankshaft. Fillet area at crankpin is identified critical where stresses generated exceed the elastic limit. It is observed that Coffin-Manson strain-life theory is found to be conservative compared to Morrow and Smith-Watson-Topper (SWT) strain-life theories.

Keywords: *Crankshaft, Fatigue life, Cyclic loading, FEM, Strain-life theories*

1. Introduction

Crankshaft is a large component having complex geometry that converts linear reciprocating displacement of the piston to a rotary motion. Since the crankshaft experiences a large number of load cycles during its service life, its fatigue performance and durability 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 minimum weight, proper fatigue strength, higher fatigue life and satisfying other functional requirements.

Chatterley *et al.* [2] compared the fatigue performance of crankshafts made from ductile iron, austempered ductile iron (ADI), and forged steel. The experiments show that when standard fillet rolling forces are used, ADI had significantly lower fatigue strength than the forged steel. Park *et al.* [8] showed that without any dimensional modifications, the fatigue life of a crankshaft could be improved significantly by applying various surface treatments such as fillet rolling and nitriding. Mostly, the failure occurs due to the crack initiation and a conservative approach is to denote the component as failed when a crack has initiated [5]. This simplification allows designers to use linear elastic stresses, obtained from multi-body dynamic finite element (FE) simulations, for the prediction of fatigue life.

The crankshaft is subjected to fully reversible cyclic loading; consequently, exposed to fatigue damages. The fatigue life prediction is less accurate even under the controlled laboratory conditions. The numerical simulation is less expensive to perform; moreover, it provides insight to the failure mechanism. Rahman *et al.* [9] conducted FEFA of aluminium suspension arm subjected to variable amplitude loading conditions. They have identified the critical location and predicted the fatigue life using strain-life theory. The stress-life theory is found to have a better correlation at high cycle fatigue; however, the strain-life theory must be used if plastic overloads are observed, known as low cycle fatigue.

Tevatia *et al.* [13] performed FEFA of plus section connecting rod for three different materials and predicted fatigue life based on Coffin-Manson, Morrow and Smith-Watson-Topper (SWT) strain-life theories. They concluded that Coffin-Manson strain-life theory gives conservative results.

Similar to the connecting rod, crankshaft is also a complex component subjected to fully reversible cyclic loading. In the present work, FEFA of crankshaft has been carried out under the fully reversible cyclic loading. Coffin-Manson [3, 6], Morrow [7] and Smith-Watson-Topper (SWT) [12] strain-life theories are used for prediction of the fatigue life of forged steel crankshaft. It is observed that Coffin-Manson strain-life theory is found to be conservative for estimating the fatigue life as compared to Morrow and SWT strain-life theories; moreover, the optimized model of crankshaft possesses higher fatigue life.

2. Problem Formulation

Fatigue analysis based on stress-life theory is well suited only when elastic stresses and strains are considered. Crankshaft may have nominal cyclic elastic stresses but stress concentrations in the crankshaft may result into local cyclic plastic deformation. The strain-life theory includes technique for converting the loading history, geometrical and material properties (monotonic and cyclic) as input parameters for predicting the fatigue life. This theory is preferred when effect of local plastic strains (due stress concentrations) is used as an additional fatigue parameter along with the elastic strain; and loading history is irregular and mean stress and load sequence effects are thought to be of importance. The strain-life theories can be used proactively for a component during early design stages. These theories are found to be the best for explanation of complex fatigue phenomenon in components like crankshafts for the estimation of fatigue strength.

The fatigue resistance of metals can be characterized by its strain-life curve. Coffin [3] and Manson [6] have established a mathematical relationship between the total strain amplitude $(\Delta\varepsilon / 2)$ and the reversals to failure cycles $(2N_f)$ as:

$$\frac{\Delta\varepsilon}{2} = \frac{\sigma_f^{'}}{E}\left(2N_f\right)^b + \varepsilon_f^{'}\left(2N_f\right)^c \qquad (1)$$

where, $(\Delta\varepsilon / 2)$ is the total strain amplitude, $\sigma_f^{'}$ the fatigue strength coefficient, $\varepsilon_f^{'}$ the fatigue ductility coefficient, E the Young's modulus of elasticity, $(2N_f)$ the fatigue life, b the fatigue strength exponent, and c the fatigue ductile exponent.

Morrow [7] and Smith *et al.* [12] studied the effect of mean stresses on fatigue behaviour of a component. Morrow [15] established a relationship between the mean stress (σ_{mean}) and fatigue life $(2N_f)$ as:

$$\varepsilon_a = \frac{\sigma_f^{'} - \sigma_{mean}}{E}\left(2N_f\right)^b + \varepsilon_f^{'}\left(2N_f\right)^c \qquad (2)$$

where, ε_a is the total strain amplitude, and (σ_{mean}) the mean stress.

Smith *et al.* [12] established another relationship, Smith-Watson-Topper (SWT) mean stress correction model, expressed as:

$$\sigma_{max}\,\varepsilon_a E = \left(\sigma_f^{'}\right)^2\left(2N_f\right)^{2b} + \sigma_f^{'}\varepsilon_f^{'}E\left(2N_f\right)^{b+c} \qquad (3)$$

where, σ_{max} represents the maximum stress.

Ramberg-Osgood [1] characterized the cyclic stress-strain behaviour of a component as:

$$\varepsilon = \frac{\sigma}{E} + \left(\frac{\sigma}{K'}\right)^{\frac{1}{n'}} \qquad (4)$$

where, ε is the total strain, σ the stress, $K^{'}$ the cyclic strength coefficient, and $n^{'}$ the cyclic strain hardening exponent.

3. Fatigue Life Estimation

Figure 1 shows the conventional fatigue life estimation procedure in which geometry, material properties and mechanical loading are regarded as three input parameters. Initially, the geometry and loading are used together to produce a stress-time $(\sigma - t)$ or strain-time $(\varepsilon - t)$ history at critical location. Next, the material fatigue properties are introduced for estimating the fatigue life. The only material properties needed in the first step are the Young's modulus, the elastic-plastic stress-strain curve, etc., which are not true fatigue properties.

In the present work, at first, the stress and strain at critical location are calculated. Next, finite element method (FEM) is used for converting reduced load-time history into the strain-time history, followed by the stress/strain calculations in the highly stressed (critical) area. Finally, three strain-life theories (Coffin-Manson, Morrow and SWT) are applied for the prediction of fatigue life.

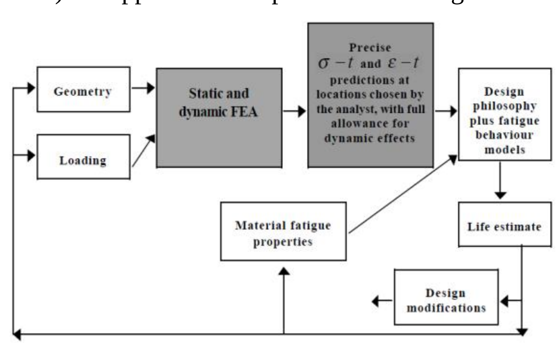

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Figure 1: Conventional fatigue life estimation procedure [1]

4. Finite Element Fatigue Analysis

The 3D model of crankshaft, termed original model, is generated in Pro/E software and FEFA is carried out on ANSYS workbench using 10-node tetrahedral SOLID 187 elements. Five design variables for the shape optimization of crankshaft model are: crankpin fillet radius (R_f), crankpin oil hole diameter (D_o), crank web thickness (W_t), depth(L_h) and diameter(D_h) of drilled hole at the back of crankshaft. The original crankshaft model is analyzed by assuming the critical values of dimensions as R_f = 2.38 mm, D_o = 18.29 mm, W_t = 20.32 mm, L_h = 34.29 mm and D_h = 8.64 mm. Seven critical locations on various fillet areas of the crankshaft are identified for the stress analysis. Figure 2 shows the Von Mises stresses distribution in different sectors of forged steel crankshaft in which maximum stresses are observed on crankpin fillet area.

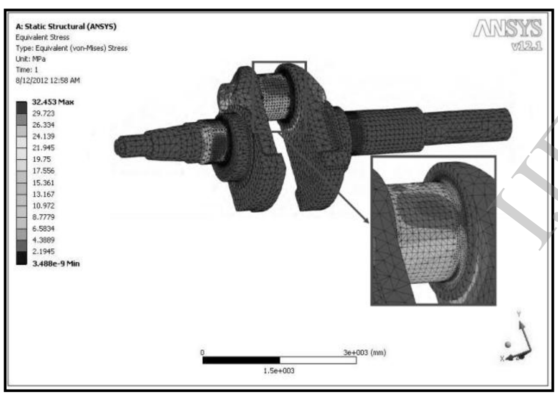

Figure 2: Von Mises stresses generated in various sectors of forged steel crankshaft (maximum stress point occurs near the crankpin fillet)

5. Results And Discussion

Material properties play an important role in the interpretation of finite element results. The cyclic material properties are used for the calculation of the elastic/plastic stress-strain response and the rate at which fatigue damage accumulates during each cycle. The fatigue results are obtained for the forged steel material with monotonic and cyclic mechanical properties listed in Table 1.

Figure 3 shows *S-N* curves obtained from the fatigue analysis of crankshaft. Stresses corresponding to the critical location are based on stress-life (*S-N*) theory as well as strain-life (*e-N*) theory. The maximum stresses at critical location are calculated by continuously increasing the force cycles up to 10^6. For N_f = 20 cycles, the stresses at critical location (fatigue strength) are found to be 24.77 MPa and 14.43 MPa based on stress-life theory and strain-life theory, respectively. As expected, the stresses at critical location are monotonically decreasing with increasing the number of force cycles. The crankshaft appears to have nominal cyclic elastic stresses but stress concentrations may result into local cyclic plastic deformation. Under these conditions, Coffin-Manson strain-life theory is used for estimating the fatigue life. The fatigue life (in seconds) at critical location is found to be conservative for the design; alternatively, strain-life theory estimates lower fatigue life.

Table 1: Monotonic and Cyclic mechanical properties of forged steel [14]

Monotonic Properties	**Forged Steel**
Young's Modulus (E), GPa	221
Yield Strength (S_Y), MPa	625
Ultimate Tensile Strength (S_u), MPa	827
Strength Coefficient (K), MPa	1316
Strain Hardening Exponent (n)	0.152
Density, kg/m^3	7833
Poisson's Ratio	0.30
Fatigue Properties	
Fatigue Strength Coefficient ($\sigma_f^{'}$), MPa	1124
Fatigue Strength Exponent (b)	-0.079
Fatigue Ductility Coefficient ($\varepsilon_f^{'}$)	0.671
Fatigue Ductility Exponent (c)	-0.597
Cyclic Yield Strength ($S_Y^{'}$) MPa	505
Cyclic Strength Coefficient ($K^{'}$), MPa	1159
Cyclic Strain Hardening Exponent ($n^{'}$)	0.128

The shape optimization of crankshaft has been carried out by successively varying the five critical design parameters and their optimized values are calculated as: crankpin fillet radius R_f = 3.00 mm, crankpin oil hole diameter D_o = 20.20 mm, crank web thickness W_t = 18.10 mm, depth of drilled hole L_h = 74.30 mm and diameter of drilled hole D_h = 10.64 mm at the back of crankshaft. Figure 4 shows S-N curves based on Coffin-Manson, Morrow and Smith-Watson-Topper (SWT) strain-life theories corresponding to the optimized values of design parameters. For given force cycle, the maximum induced stresses at the critical location

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are found to be in the following order: $(\sigma_{max})_{Coffin-Manson} < (\sigma_{max})_{Morrow} < (\sigma_{max})_{SWT}$. Thus, SWT strain-life theory exhibits higher fatigue strength compared to Morrow and Coffin-Manson strain-life theories but Coffin-Manson theory gives conservative results; hence safe for the design.

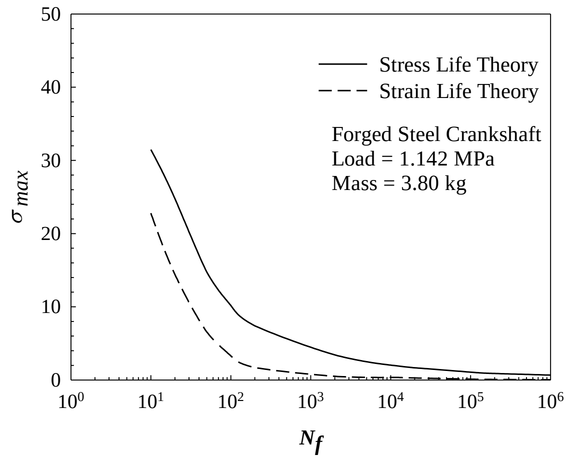

Figure 3: *S-N* curves for crankshaft based on stress-life theory and strain life theory

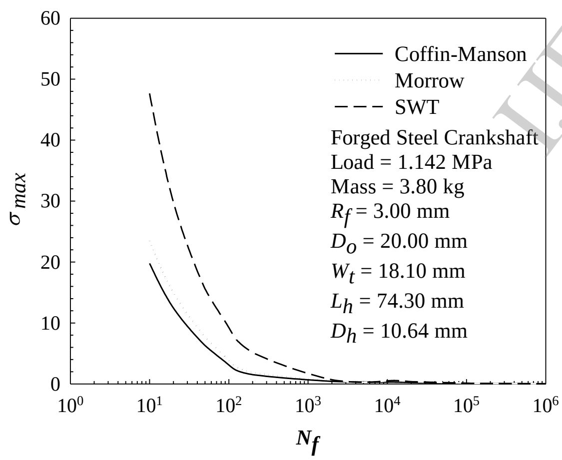

Figure 4: *S-N* curves for optimized crankshaft model based on three different strain life theories

Since Coffin-Manson strain-life theory is safe; hence, *S-N* curves based on Coffin-Manson are plotted (Figure 5) for the original and optimized crankshaft models. For N_f = 20 force cycles, it is observed that maximum stresses at the critical location are $(12.44)_{\text{optimized}} < (14.27)_{\text{original}}$. Thus, the critical stresses (σ_{max}) generated for the optimized model are low compared to the original model.

Table 2 presents the maximum stresses at critical/failure location and fatigue life based on the three strain-life theories for original and optimized forged steel crankshaft models. The critical location is elected as the point for fatigue failure. As far as fatigue life is concerned, the optimized model possesses higher fatigue life (in sec) irrespective of the strain-life theories. Moreover, for a given model, Coffin-Manson theory gives conservative results; consequently, estimates the lowest fatigue life, hence safe. Finally, it is concluded that the forged steel optimized shape crankshaft model is the best when both elastic and plastic strains are considered, i.e., Coffin-Manson theory is used for estimating the fatigue life.

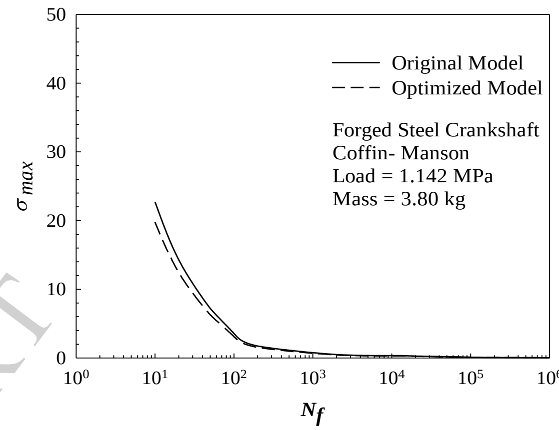

Figure 5: *S-N* curves for original and optimized crankshaft models based on Coffin-Manson theory

Table 2: Maximum stresses and fatigue life for original and optimized crankshaft

Crankshaft	**Maximum stresses at critical location σ_{max} (MPa)** *	**Fatigue life $\times 10^6$ (Sec)***		
		Coffin-Manson	**Morrow**	**SWT**
Original model	32.453	15.34	16.21	17.70
Optimized model	28.071	21.73	22.56	24.05

* Mass = 3.80 kg (approximately constant),
Load = 1.142 MPa (fully reversible cyclic)

6. Conclusions

In the present work, FEFA of forged steel crankshaft are carried out using different strain-life theories. It is observed that crankpin fillet area is critical as far as maximum stresses generated are concerned. Coffin-Manson strain-life theory is found to be conservative for estimating the fatigue life as compared to Morrow and SWT theories. Moreover, the optimized crankshaft model possesses higher fatigue life.

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