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Reliability prediction of the fatigue life of a crankshaft †

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Abstract

Crankshaft, the core element of the engine of a vehicle, transforms the translational motion generated by combustion to rotational motion. Its failure will cause serious damage to the engine so its reliability verification must be performed. In this study, the S-N data of the bending fatigue limit of a crankshaft are derived. To evaluate the reliability of the crankshaft, reliability verification and analysis are performed. For the purpose of further evaluation, the bending test of the original crankshaft is carried out, and failure mode analysis is made. The appropriate number of samples, the applied load, and the test time are computed. On the basis of the test results, Weibull analysis for the shape and scale parameters of the crankshaft is estimated. Likewise, the B_{10} life under 50% of the confidence level and the MTTF are exactly calculated, and the groundwork for improving the reliability of the crankshaft is laid.

Keywords: Acceleration factor; Crankshaft; Fatigue; Reliability; Weibull analysis

1. Introduction

Fatigue failures are breakdowns caused in components by the action of fluctuating loads. They are estimated to be responsible for 90% of all metallic failures since the loads on the components are usually not constant but instead vary with time. Fatigue failures occur when the components are subjected to a large number of cycles of the applied stress. With fatigue, the components fail under stress values much below the ultimate strength of the material and often even below the yield strength. What makes fatigue failures even more dangerous is the fact that they occur suddenly, without warning. The crack may be initiated by internal cracks in the component or irregularities in manufacturing. Once a crack has formed, it propagates rapidly under the effect of stress concentration until the stressed area decreases. This leads to a sudden failure [1, 2].

The crankshaft is the central part of the engine, and its failure would render the engine useless until costly repairs could be made or a replacement engine could be installed. The failure of a crankshaft can damage other engine components including the connecting rods or even the engine block itself. Therefore, when the failure of a crankshaft occurs, it often results to replacement of the engine or even scrapping of the equipment the engine was used in. Considering the ramifications of crankshaft failure, a crankshaft must be designed to last the lifetime of an engine [3].

Practical cases and some investigations revealed that bending stress is much severe than torsion stress. Therefore, bending stress is more often focused on, while torsion stress is often neglected [4].

The fillets have been identified as the highest stressed location of a crankshaft. The presence of a

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Table 1. Applied bending load and frequency.

Load, kN	Frequency, Hz
±98	15
±78.4	15
±61.74	15

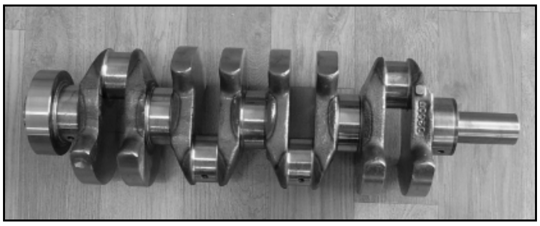

Fig. 1. A crankshaft sample for testing procedure.

fillet or notch in a crankshaft is virtually unavoidable. Any change in diameter results in a stress concentration. While sharp corners can be avoided with the use of fillets, other measures are often necessary to increase the fatigue performance of crankshafts. Compressive residual stresses have been shown to increase the fatigue performance not just of crankshafts but of other components as well. Often, in an attempt to induce compressive residual stresses at notches, the fillets are rolled. This compressive residual stress increases the fatigue strength at long life [5, 6].

2. Test setup and experimental procedure

Ductile cast iron crankshafts were used in this test (Fig. 1). The crankshaft is intended to be used in an engine with a displacement of up to 3000 cc, which is typical of those found in Sport Utility Vehicles or lightweight trucks.

A crankshaft was mounted into the fixture as shown in Fig. 2. Monotonic bending load was applied to the front main bearing journal through a special rod. The applied bending load and frequency levels are given in Table 1.

Crankshaft failures take place when the crankshaft cannot transmit torque any longer. Therefore, the occurrence of a two-piece failure would be the best match to an engine failure. However, this failure mode becomes limited in a practical laboratory testing since component fracture can damage the testing rig.

During this test, a displacement shifting method was adopted. A computer monitors the displacement feedback signal, while the sample is under the bending load. As the crack initiates and propagates further, the displacement amplitude also increases, altering the feedback signal. In turn, the computer automatically stops the testing process.

Table 2. Exact failure times of the test samples.

Load, kN	Cycles-to-failure, N
±98	166.580
	155.437
	83.209
±78.4	184.297
	120.351
	227.982
±61.74	5.487891 (suspended)
	5.845.446 (suspended)
	5.907.819 (suspended)

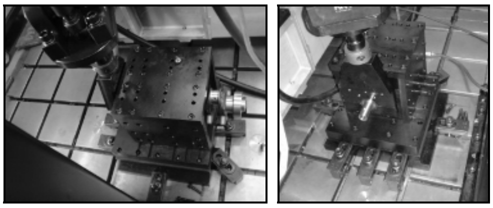

Fig. 2. Test rig and linear actuator.

The testing assembly was loaded by a high-capacity linear actuator together with the load cell and displacement sensors. In effect, the testing procedure is fully automated and precisely controlled by the computer system.

The first objective was to perform testing with the initial load level until three failure results will be obtained. If the results are reasonable, second-level testing is performed. The testing process is completed through third load-level testing. The fatigue and reliability theories were applied to analyze the test results.

3. Analysis of test results

The failure times of all samples were recorded and are presented in Table 2. The results of the third-level testing are not included in the analysis since no failure was observed.

Based on the customer's decision, the number of cycles over 5 million is considered as infinite life. This allows performing the analysis of failure data without nonfailed units (censored).

Usually, information about the nonfailed samples at accelerated stress conditions is more important than information about the failed samples, which are tested at much higher stress levels than the normal operating conditions. As such, the information about nonfailed units must be incorporated into the analysis of the data. However, there is a distinguishing point. The censored samples were tested in a different load level from the failed samples, and the testing time was not limited [7].

The likelihood function is given in Eq. (1) when Weibull distribution is assumed.

$$L(\alpha,\beta,t)=\left(\frac{\alpha}{\beta}\right)^{n}\prod_{i=1}^{n}t_i^{\alpha-1}e^{-t_i^{\alpha}/\beta} \tag{1}$$

where α is the shape parameter, β is the scale parameter, and t_i is the recorded failure times of n units.

The action of taking the logarithm of Eq. (1) and then considering the derivatives of the logarithmic function with respect to α and β will result in Eq. (2) and (3).

$$\frac{n}{\hat{\alpha}}+\sum_{i=1}^{n}\ln t_i-\frac{1}{\hat{\beta}}\sum_{i=1}^{n}t_i^{\hat{\alpha}}\ln t_i=0 \tag{2}$$

$$-\frac{n}{\hat{\beta}}+\frac{1}{\hat{\beta}^2}\sum_{i=1}^{n}t_i^{\hat{\alpha}}=0 \tag{3}$$

The maximum likelihood estimation of α and β can be obtained by solving Eqs. (2) and (3). When $\hat{\beta}$ as obtained from Eq. (3) is substituted into (2), the difference $D(\hat{\alpha})$ can be obtained as follows:

$$D(\hat{\alpha})=\frac{\sum_{i=1}^{n}t_i^{\hat{\alpha}}\ln t_i}{\sum_{i=1}^{n}t_i^{\hat{\alpha}}}-\frac{1}{\hat{\alpha}}-\frac{1}{n}\sum_{i=1}^{n}\ln t_i=0 \tag{4}$$

The Eq. (4) in $\hat{\alpha}$ can be solved numerically or by trial and error. Once $\hat{\alpha}$ is estimated, the value of $\hat{\beta}$ is computed as follows:

$$\hat{\beta}=\sum_{i=1}^{n}\frac{t_i^{\hat{\alpha}}}{n} \tag{5}$$

The probability plot for the bending test results is given in Fig. 3. The plot clearly shows that both ±98 kN and ±78.4 kN load-level test results have nearly the same shape α parameters. The Calculated Mean life, B_{10} life, and shape and scale parameters are given in Table 3.

Based on the data from Table 3, it is possible to calculate the acceleration factor by using Eq. (6).

Table 3. Estimated parameters.

Load, kN	MTTF, cycles	B10 life, 50% CL	Shape parameter	Scale parameter
±98	136.183	92.487	4,8016	194.712
±78.4	178.360	100176	4,7328	148.793

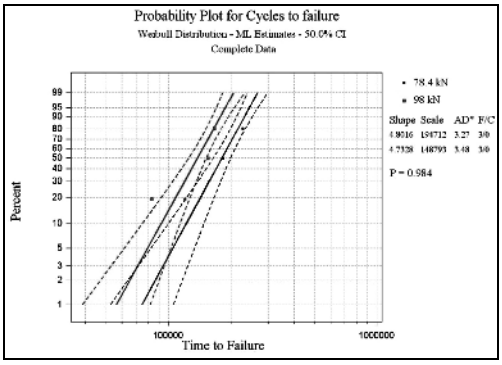

Fig. 3. Probability plot for the bending test results.

Fig. 4. Failure mode during bending test.

$$AF=\frac{B_{10}(78.4kN)}{B_{10}(98kN)}=\frac{100176}{92487}=1.083 \tag{6}$$

Accelerated life testing could be performed even faster if the load is increased and the shape parameters are matched.

As commonly known, most stresses in crankshafts are concentrated around the fillet area. Cracks are initiated around the fillet and propagate rapidly, causing destructive failure. During the conduct of this project, the observed failure mode was similar to most practical cases. It is shown in Fig. 4.

The S-N curve was developed based on the calculated MTTF value (Fig. 5).

The slope is evaluated using the developed S-N curve, and it is equal to *b=-0.99* for bending loading.

The supposed fatigue limit of 61.74 kN bending load can be used for correlating with the engine fa-

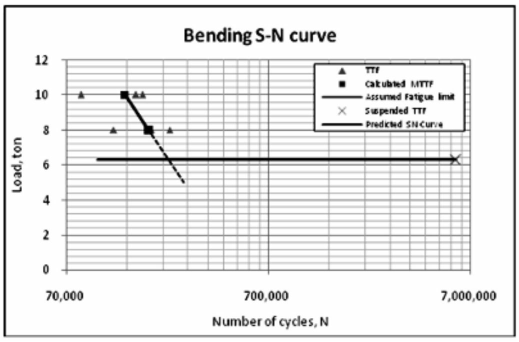

Fig. 5. S-N data after the bending fatigue test.

tigue life target. If some modifications will be made to the design target, then highly accelerated crankshaft life testing can be performed.

Adapting the equivalent damage Eq. (7) for the higher load-level testing will help predict the B_{10} life of the third step loading.

$$D = \sum_{i=1}^{n} \sigma_i^k N_i$$
$$\sigma_1^k N_1 = \sigma_2^k N_2 \tag{7}$$

where $k=-1/b$.

Thus, if the third-level loading will be decided for the highly accelerated life testing, then the next level could be assumed to be equal to ±150 kN. A few operations using Eq. (7) will result in a newly predicted B_{10} life of 52020 cycles. The acceleration factor by Eq. (6) shows that the test will be accelerated almost two times [Eq. (8)].

$$AF = \frac{B_{10}(78.4kN)}{B_{10}(150kN)} = \frac{100176}{52020} = 1.9 \tag{8}$$

4. Conclusions

The underestimated or overestimated fatigue limits of the component can make tremendous differences in safety issues and financial conditions. Perfectly designed components will last their intended lifetime and will help save resources, thus improving the efficiency of manufacturing companies.

In this study, the exact B_{10} life of a manufactured crankshaft is evaluated at stepped loading levels, the fatigue limit of *61.74 kN* is estimated, and reliability analysis of the data is performed. The fatigue limit of the component should be correlated with the design target, and necessary improvements will have to be made to the components design.

The acceleration factor between the two tested levels of 98 kN and 78.4 kN was equal to 1.083, which is not considerable. For future studies, a highly accelerated life testing of these crankshafts with a higher load level is suggested. If the load level of 150 kN will be chosen, then the acceleration factor is equals to 2. This simply means that the components testing time will be shortened 2 times.

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