Predicting the durability of piston engine crankshafts

A N Gots, Sh A Amirseyidov

Vladimir State University n.a. Alexander and Nikolay Stoletovs, Vladimir, Gorky street 87, Russia, 600000

E-mail: 1 shihamir@mail.ru

Abstract. A method for testing crankshafts of automobile and tractor engines under the action of variable bending moment is proposed. The test machine consists of a frame on which the crankshaft is installed through elastic springs on the extreme root necks. On the extreme necks, pendulums are installed, one end of which is rigidly fixed, and at the opposite ends of the pendulums, two inertial (unbalanced) vibrators are installed, creating two equal forces in the horizontal plane, directed in opposite directions and changing their magnitude. This pair of forces creates a variable bending moment relative to the axis of the crankshaft, acting on all its elements, and causes fatigue failure in the weakest sections. Based on the test results, a graph is plotted in the coordinates of maximum stress – the number of loading cycles before failure, which allows you to determine the parameters of the crankshaft fatigue life model.

1. Introduction

The main assumption, which is usually accepted in all theories for calculating the durability of a part under the action of a load spectrum, is that the effect of cyclic stresses of a given amplitude leads to fatigue damage, the value of which is determined by the number of cycles of stress of this amplitude, as well as the total number of such cycles before the destruction of the sample. As applied to crankshafts, it can be assumed that in the most loaded sections (usually this is the transition zone from the connecting rod neck to the cheek along the overlap line), the crystal lattice distortions occur, leading to the appearance of tears in some direction in which the greatest stresses act, which gradually pass under the influence of variable stresses into the crack. Under the influence of alternating stresses, the crack grows. When the total accumulated damage reaches a certain critical value, and the crack significantly weakens the cross-section, fatigue failure occurs.

2. Purpose of research

Development of a method for predicting the durability of piston engine crankshafts based on fatigue tests of full-scale samples

3. Materials and methods of research

In [1,3,4], the possibility of predicting the cyclic durability of crankshafts based on the results of fatigue tests is shown. Figure 1 shows an approximate graph of fatigue life for details [5,6]. Stress \( \sigma_{st} \)-No – the value of the endurance limit at the point of fracture A-for steel at \( N = 2 \times 10^6...5 \times 10^7 \) cycles. Its presence is associated with a
change in the mechanism of predominant development of fatigue. At a high stress level, fatigue failure occurs at \( N < 10^5 \) cycles as a result of the accumulation of plastic deformations along the shear planes. This is low-cycle fatigue, where the patterns of static destruction are more pronounced. If the destruction occurs when the number of cycles \( N = 2 \cdot 10^5 ... 5 \cdot 10^6 \), then in this case, the regularities of fatigue destruction appear [5,7,8].

![Figure 1. Approximate graph of fatigue life under regular loading](image)

At a low stress level (along the straight line AD), diffusion processes of dislocation movement develop. However, to obtain a section of the fatigue curve, time-consuming and lengthy tests are required [7,8,9].

The diagram of the installation for fatigue testing of crankshafts, developed under the guidance of the author, is shown in Figure 2. The establishment consists of a frame 1, on which the crankshaft 3 is installed on the extreme root necks through elastic springs 2.

On the same necks, pendulums 4 are installed, one end of which is rigidly fixed to the necks, and at opposite ends of the pendulums, two inertial (unbalanced) vibrators 5 are installed to create cyclic alternating loads, rotating with the same angular velocity \( \omega \) and creating two equal forces \( P = m \omega r^2 \) in the horizontal plane.

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Here \( m \) is the mass of the unbalanced balance weight of the vibrator located at the radius \( r \); \( \omega \) is the frequency of its rotation. The components of forces on the horizontal \( x \)-axis are equal to \( P_x = m \omega r^2 \sin \phi \), and on the vertical \( y \)-axis - \( P_v = m \omega r^2 \cos \phi \) (\( \phi = \omega t \) is the angle of rotation when the unbalanced mass \( m \) rotates around the vertical \( y \) axis, \( t \) is the current time).

In this case, the vertical component of the inertia force from the one balance weight load (Figure 2) \( P_v = P \omega \sin \phi \) it is balanced by a similar component from the second balance weight, and the forces from the vibrators 5 on the left and right create equal and oppositely directed forces \( 2P \sin \phi \). Thus, the crankshaft is loaded with a pure bend in a symmetrical cycle with a moment \( M = 2P l \sin \phi \), where \( l \) is the length of the pendulum 4 for the same type of fatigue testing of crankshafts, executive document RD 23.3.62-89 "Methodology for accelerated fatigue testing of crankshafts of tractor and combine engines" was developed [5,6].
To control the loading on the crankshaft rod journal in the section plane of the first crank pasted strain gauges 6, which are connected to the measuring instrument 7. Before the beginning of tests conducted calibration of the strain gages mounted on the crankshaft, which for both ends of the pendulum 4 (instead of system loading 5 – Figure 2) a static load was applied. The stand for calibration of strain gages consisted of a rigid frame, supports in the form of prisms, a hydraulic Jack and a compression dynamometer DOSM-3 [4].

The appearance and growth of a fatigue crack in the test knee of the crankshaft is recorded by increasing the value of the signal received from the strain gages to the control and measuring equipment. The appearance of a crack in the cross section of the crankshaft reduces the frequency of its natural vibrations $\omega_n$ and since the tests were carried out in the resonant mode, the vibration amplitude increased. Tests are terminated if the signal value increases by 15% from the original value [10, 11].

Figure 2. Installation diagram for testing crankshafts

Figure 3. The crankshaft in the car for testing
The crankshaft can be installed in the test machine so that all the cranks are loaded simultaneously or each crank separately (Figure 3). The amount of bending moment acting in the \( n \)-th crank under test is monitored continuously throughout the entire test time. The control system consists of strain gages pasted on the connecting rod neck of the tested knee, a strain amplifier, and two oscilloscopes: electronic (for visual monitoring of the signal value from the strain gage) and loop (for periodic monitoring of the signal value after calibration under static loading). The drive of the vibrator 5 was carried out from a balancing machine.

The error in determining the controlled parameters during testing did not exceed the values indicated in Table 1.

**Table 1** Errors of controlled parameters

| Controlled parameters | Unit of measurement | Measurement errors, % |
|-----------------------|---------------------|------------------------|
| 1 Bending moment, \( M_{\text{ben}} \) | Nm | 2…3 |
| 2 Frequency of rotation of the drive shaft, \( n \) | min\(^{-1}\) | 1 |
| 3 Operating time in load mode | h | 0,1 |
| 4 Number of loading cycles, \( N \) | number | 0,01 |

At loading of each crank pure bending in its plane with time Mesh destruction must occur in the cross section of the least stiffness from the fillets interfaces connecting rod neck of crankshaft with a cheek with access to a fillet of the pairing crank journal and cheeks [12-14].

An example of the destruction of the crankshaft in this section is shown in Figure 4.

**Figure 4.** Example of the destruction of the crankshaft of a diesel D144

When testing, it is necessary to obtain at least three values of cycles \( N_1, N_2, N_3 \) on the straight line \( AB \) (Figure 1), at which the destruction occurs by loading with a net moment \( M_1, M_2, M_3 \). Since the value \( N_0 \) is selected in advance (usually taken \( N_0 = 2 \times 10^6...5 \times 10^7 \) cycles), such tests actually determine not only the maximum bending moment, but also the effect of the stress concentration in the fillet. In this case, it is quite simple to determine the effect of structural or technological measures on increasing the fatigue strength of the crankshaft [15-17].

**4. The results of the study and their discussion**

As deterministic models for the fatigue life of the crankshaft at the loading point (uniaxial stress state) can take it [7, 15]

\[
\sigma_{\text{a}} \sqrt{N} = C, \tag{1}
\]

where \( \sigma_{\text{a}} \) – the amplitude of the alternating voltage; \( \sqrt{N} \) – the average number of cycles to failure; \( C \) and \( m \) – parameters of the test to the crankshaft, depending on the material, manufacturing technology and other factors.

It is known that fatigue tests with the constant amplitude of variable stresses are characterized by a large variation in the number of cycles before failure. Therefore, in equation (1), we always mean the
average number of cycles $N$ to failure. In addition, in tests, equation (1) will be valid if the stress $\sigma_a$ is replaced by the moment at $N$.\cite{15,16,17}

Since the crankshaft tests are performed at different values of the amplitude stresses $\sigma_{ai}$, it is possible to determine the value of the parameters $C$ and $m$. Actually, using the model in the form of (1) and data on the cyclic durability of cranks for any two stress values $\sigma_{a1}$, $\sigma_{a2}$ and the corresponding average values of cycles before failure $N_1$ and $N_2$, we obtain after logarithm two equations for determining $C$ and $m$:

$$m \lg \sigma_{a1} + \lg N_1 = \lg C;$$

$$m \lg \sigma_{a2} + \lg N_2 = \lg C. \tag{2}$$

The obtained values $C$ and $m$ allow us to find the endurance limit for the base number of cycles $N_b = 2 \cdot 10^6$ or the value $\sigma_{aK} = K_a \sigma_a/\beta_a$. Since the rated voltage can be calculated, the value of the parameter ratio $K_a/\beta_a$ is determined based on experimental research data, which is taken into account when calculating at the design stage.\cite{18,19,20}

To determine the durability of the crankshaft in hours, assume that during the time $t_i$ of the engine in operation in one of the modes characterized by the value of the torque $M_i$ and the angular speed of rotation of the shaft $\omega_i$ during the entire service life (until destruction) $T$, it will be produced $n_i$ loading cycles, which for four-stroke engines is equal to

$$n_i = \frac{300 \omega_i 60 t_i}{2 \pi} = \frac{900 \omega_i}{\pi} f_i T , \tag{3}$$

where $t_i = f_i T$; $f_i$ is the probability density of operation in this mode.

From the equation of the actual endurance curves $\lg N - \lg M_{\max}$ of the crankshaft, we determine the number of loading cycles required for its destruction

$$N_i = \left(\frac{M_{\max}}{M_{\max,i}}\right)^m N_0, \tag{4}$$

where $m$ is an exponent numerically equal to the cotangent of the slope of the fatigue curve in logarithmic coordinates ($m = \text{ctg} \alpha$); $\alpha$ is the angle of inclination of the left branch of fatigue; $M_{\max,i}$ – the amplitude of the maximum bending moment under the $i$-th loading mode.

Thus, taking into account (3) and (4), a certain amount of damage will be accumulated by the crankshaft over time $t_i$:

$$\frac{n_i}{N_i} = \frac{900 \omega_i f_i T}{\pi \left(\frac{M_{\max}}{M_{\max,i}}\right)^m} N_0 . \tag{5}$$

According to the corrected linear hypothesis of fatigue damage accumulation, shaft failure occurs when, taking into account (5), the sum of accumulated damage is equal to:

$$\sum_{i=1}^{k} \frac{n_i}{N_i} = \frac{900 T}{\pi N_0} \sum_{i=1}^{k} f_i \left(\frac{M_{\max,i}}{M_{\max}}\right)^m = a_p , \tag{6}$$

where $k$ – the operation of the engine in operation.

Following [5] and considering that the stresses are replaced by limiting bending moments, the value $a_p$ for bench accelerated tests [1] is also determined by the formula:

$$a_p = \frac{0,5 M_{\max}}{M_{\max} - 0,5 M_{\max}} , \tag{7}$$

$$\zeta = \sum \frac{M_{\max,i}}{M_{\max}} \cdot \frac{v}{v_n} \left( M_{\max,i} > 0,5 M_{\max} \right) ; \tag{8}$$
where \( v_a = \sum_{i=1}^{n_k} v_i \) – the total number of loading cycles. \( v_i \) – the number of cycles of repeating amplitudes in the loading block.

When using a limited endurance limit, you should assume:

\[
a_p = \zeta = \sum_{i} \frac{M_{\max i}}{M_{\text{max}0}} v_i, \tag{9}
\]

where the summation extends over all loading amplitudes without discarding small amplitudes.

According to the linear hypothesis of fatigue damage accumulation, crankshaft failure occurs when \( a_p = 1 \).

From (6), the crankshaft durability can be found by the fatigue strength resistance condition:

\[
T = \frac{nN_i a_p}{900} \sum_{i=1}^{n_k} f_i \left( \frac{M_{\max i}}{M_{\text{max}0}} \right)^{p_i}, \tag{10}
\]

Therefore, to predict the durability of the crankshaft, it is necessary to have information about the fatigue strength limit \( M_{\text{max}0} \), the probabilistic and statistical load \( f_i \) of engines in operation, as well as the loads \( M_{\max i} \) on the crankshaft.

The relationship between the level of maximum bending moment \( M_{\max i} \) and the number of loading cycles \( N_i \) before failure, even with the strictest observance of the identity of the crankshaft fatigue tests, due to the heterogeneity of the material itself, has a pronounced random character. To take this into account, we must assume that the parameters of the fatigue curve are random variables that obey some statistical distribution. In this regard, the durability of the crankshaft will be determined by the expression [6,21]:

\[
T = \overline{T} \chi^n, \tag{11}
\]

where \( \overline{T} \) is the average life of the crankshaft, calculated by the formula (8); \( \chi \) is a random parameter whose distribution density has the form:

\[
f(\chi) = \frac{1}{\nu (1-\varepsilon)} \left( \frac{\chi - \varepsilon}{1 - \varepsilon} \right)^{\nu'} \exp \left[ - \left( \frac{\chi - \varepsilon}{1 - \varepsilon} \right)^{\nu'} \right],
\]

where \( \nu \) is the coefficient of variation; \( \varepsilon = 0.6 \) is the fraction of the average fatigue limit \( M_{\text{max}0} \) below which no fatigue damage accumulates.

Taking into account these factors, the probability of failure of the crankshaft due to fatigue failure during operation \( T \) is determined from the expression

\[
F(T) = 1 - \exp \left[ - \left( \frac{T}{\overline{T}} - \varepsilon \right) \frac{1}{1 - \varepsilon} \right]. \tag{11}
\]

5. Conclusions

A method for predicting the durability of the crankshaft comprises the following stages: the experimentally determined endurance limit of the crankshaft \( M_{\text{max}0} \); and according to the loading conditions of the crankshaft load in the use – values of \( M_{\max i} \) and the number of blocks of loading, the formula (8) calculated the average durability of \( T \); in the formula (9) for a given resource \( T \) is the probability of failure of crankshaft due to fatigue fracture.

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