A Numerical Model for Estimation of Service Life of Tribological Systems of the Piston Engine

K. Gavrilov\textsuperscript{a}, Y. Goritskiy\textsuperscript{b}, I. Migal\textsuperscript{b}, M. Izzatulloev\textsuperscript{a}

\textsuperscript{a}South Ural State University, Chelyabinsk, Lenin prospect 76, Russia, 
\textsuperscript{b} National Research University "MPEI", Moscow, Krasnokazarmennaya street 14, Russia.

**Keywords:**
Surface asperities interaction  
Markov process  
Tribological parameters

**Abstract**
This article describes, develops and applies approach of the interaction of rough surfaces for one of the tasks of simulation of tribological systems of the piston engine. In this paper we described the general approach to building a model of interaction between rough surfaces, leading to the analysis of the Markov process. Given the initial data and the method of calculating the trajectory of movable elements on the lubricating layer, we determined the tribological parameters defining the service life of tribological systems of the piston engine on the example of crankshaft bearings.

© 2017 Published by Faculty of Engineering

**1. INTRODUCTION**

For the development of components machines and mechanisms, where friction determines the durability and dimensions, there is a need for methods that enable to estimate the friction characteristics in time: friction force, coefficient of friction, contact area, run-in time, wear and so on [1,2].

Friction characteristics determined by such factors as surface microgeometry, physical characteristics of materials, velocity and the applied load. Most methods were based on the representation of the relief function for the random arguments [3-6]. Such models can estimate the area of contact, the friction force, coefficient of friction, wear, etc. However, this approach ignores the surfaces change and, consequently, contact and friction characteristics over the time. In this paper model of random change of surfaces heights over the time for "main bearing-crankshaft" tribosystem is supplemented by taking into account the presence of a lubricant in a mixed friction mode.

The "main bearing-crankshaft" tribosystem is mostly in the hydrodynamic lubrication regime, but at high loading levels, tribosystem is in the mixed or boundary lubrication regimes that are important for the determination of the service life.

**2. THE HYDROMECHANICAL CHARACTERISTICS OF THE "MAIN BEARING-CRANKSHAFT" TRIBOSYSTEM**

Reactions of a lubricant film were determined on the basis of results of numerical integration of
The modified equation Elrod for pressure in a lubricant film [7]. In this paper used modification, written relative to function $\Omega(\varphi, z)$ in form:

$$
\frac{\partial}{\partial \varphi} \left[ \frac{\hbar^3}{12 \mu_{\text{eff}}} \frac{\partial}{\partial \varphi} (g \Omega) \right] + 
+ \frac{1}{a^2} \frac{\partial}{\partial z} \left[ \frac{\hbar^3}{12 \mu_{\text{eff}}} \frac{\partial}{\partial z} (g \Omega) \right] = 
= \frac{\nu}{2} \frac{\partial}{\partial \varphi} \left[ \hbar [1 + (1 - g) \Omega] \right] + 
+ \frac{\partial}{\partial \varphi} \left[ \sigma [1 + (1 - g) \Omega] \right],
$$

where $\hbar = h / h_0$; $\mu_{\text{eff}} = \mu / \mu_0$; $-a \leq z \leq a$; $\varphi = x / R$; $\varphi = x / 2R$; $\tau = \omega d / \nu$; $\nu = (w_2 - w_1) / w_1$; $\hbar$, $\mu_{\text{eff}}$ - dimensionless film thickness and the effective viscosity of the lubricant; $B, R$ - the width and radius of the bearing; $\mu_{\text{eff}}$ - effective viscosity of the lubricant, the corresponding temperature to $T^*_e$; $\mu_0$, $h_0$, $w_0$ - respectively lubricant viscosity, typical film thickness at the center position of the shaft in the bearing and rotation speed of crankshaft; $\nu$ - the dimensionless rotating shaft velocity; $g$ - the switching function.

The main hydromechanical characteristics (HMC) of the tribosystem are: $h_{\text{min}}(\tau)$ - instantaneous values of the minimum oil film thickness; $p_{\text{max}}(\tau)$ - instantaneous values of the maximal hydrodynamic pressure; $h^*_{\text{min}}$ - average value of $h_{\text{min}}(\tau)$; $p^*_{\text{max}}$ - average value of $p_{\text{max}}(\tau)$; $N(\tau)$ and $N^*$ - instantaneous and average power loss of friction; $Q^*$ - the average flow rate of oil; $T^*_{\text{eff}}$ - the average effective temperature of the lubricating layer; $\alpha_{\text{lim}}\%$ - the areas, where minimal thickness of a lubricant film less than the permissible value $h_{\text{lim}}$ equal 1.5 $\mu$m, which we determined from profilograms of samples of the "main bearing-crankshaft" tribosystem of diesel engine.

The function $\Omega(\varphi, z)$ is related to the degree of filling of the clearance $\theta(\varphi, z)$ and is characterized by a function that determines the mass content of the liquid phase (oil) in the volume of the clearance of the journal bearing using the relationship $\theta = 1 + (1 - g) \cdot \Omega$.

Given the initial data and the methodology for calculating the trajectory of shaft on the lubricating layer [8, 9], we calculated the HMC for all main bearings of diesel engine, but only two of them (2nd and 3rd main bearings) had nonzero values of $\alpha_{\text{lim}}$ (Table 1).

Table 1 shows that the $h_{\text{min}}$ for the 2nd and 3rd main bearings reaches the values less than the total value of the sum of the maximum heights of microasperities for the contact surfaces.

For only the period of tribosystem's working time, where $h_{\text{min}}$ less $h_{\text{lim}}$ (contact area), we analyzed characteristics of the "main bearing-crankshaft" tribosystem using Markov chains.

3. METHODS

To analyse the process of friction in the contact area we used the discontinuous model where surfaces are represented by asperities of random height [10]. Asperities in contact deform and destruct each other that are why heights of contacting asperities may change. Total height change of all surface asperities leads to roughness transformations and variation of friction characteristics.

Let’s consider $K$ pairs of asperities positioned in one raw. Time is discontinuous and is measured in shift counts. After $n$ shifts heights of asperities of one surface are represented by vector $\xi_k(n), k = 1, ..., K$, and heights of asperities of the other surface are represented by vector $\eta_k(n)$. The asperity height change of the $k^{th}$ pair in one shift is described by general equations:

$$
\xi_k(n+1) = \Psi(\xi_k(n), \eta_{k-1}(n)), \quad (2)
$$
$$
\eta_k(n+1) = \Phi(\xi_k(n), \eta_{k-1}(n)), \quad (3)
$$

Table 1. HMC of the “main bearing-crankshaft” tribosystem of diesel engine.

<table>
<thead>
<tr>
<th>Num. bear.</th>
<th>$h^*_{\text{min}}$, $\mu$m</th>
<th>$p^*_{\text{max}}$, MPa</th>
<th>$N^*$, kg/$\text{s}$</th>
<th>$Q^*$,</th>
<th>$\alpha_{\text{lim}}$, %</th>
</tr>
</thead>
<tbody>
<tr>
<td>2</td>
<td>4.296</td>
<td>131.1</td>
<td>2056</td>
<td>0.067</td>
<td>7.8</td>
</tr>
<tr>
<td>3</td>
<td>4.697</td>
<td>131.6</td>
<td>1820</td>
<td>0.063</td>
<td>11.1</td>
</tr>
</tbody>
</table>
where \( k = \Gamma K \), \( \eta_0(n) = \eta_k(n) \),

where \( \Phi \) and \( \Psi \) determine stochastic mechanism of asperity interaction as a sphere interaction [11], with random radiusses. Presented model of asperity interaction takes into account elastic-plastic deformation and wear.

It is proposed to describe the trajectory of height changes of single asperity by means of Markov series [12]. The height \( \xi \) of single asperity changes when it interacts with asperities of other surface, having heights \( \eta(t) \) at the moment \( t \). It is supposed, that asperity with the height \( \xi \) interacts with the series of asperities of the other surfaces with heights \( \eta_1, \eta_2, ..., \eta_n \), distributed with density \( q_\eta(y) \). Thus series of heights \( \xi_1, \xi_2, ..., \xi_n, \) is obtained, height value \( \xi_{n+1} \) depends on \( \xi_n \) and \( \eta_{n+1} \):

\[
\xi_{n+1} = \Psi(\xi_n, \eta_{n+1}),
\]

where \( \Psi \) determines accepted mechanism of asperity contact interaction.

Series \( \xi_1, \xi_2, ..., \xi_n, \) is Markov series. Moreover, heights \( \xi_1, \xi_2, ..., \xi_n, \) are discontinuous, series \( \xi_1, \xi_2, ..., \xi_n, \) becomes Markov chain. Similarly, \( \xi \) is distributed with density \( p_\xi(x) \) and next equation is obtained:

\[
\eta_{n+1} = \Psi(\eta_n, \xi_{n+1}).
\]

Once two functions \( \Psi \) and \( \Phi \) are elaborated the system of equations for stationary distributions \( p^* \) and \( q^* \) is obtained:

\[
\begin{align*}
p^* &= p^* \cdot P_\xi(q^*, \Psi) \\
q^* &= q^* \cdot P_\eta(p^*, \Phi)
\end{align*}
\]

where \( P_\xi(q^*, \Psi) \) and \( P_\eta(p^*, \Phi) \) are matrices of transition probabilities for series of \( \xi_n \) and \( \eta_n \). Elements of matrices are determined by distributions \( p^*, q^* \) and by two functions \( \Psi \) and \( \Phi \).

We can write the interaction function for the discrete model. If \( \xi(t) \) - the height of the fixed element at the moment \( t \), \( \xi_n = \xi(t_n) \), \( \xi_{n+1} = \xi(t_{n+1}) \) then for fixed element of the surface \( \xi \):

\[
\xi_{n+1} = \Psi(\xi_n, \eta_n) = \\
\xi_n - D_\xi(\delta_n) - W_\xi(\delta_n) + (V_N/2)\Delta t,
\]

where \( \delta_n = \xi_n - \eta_n \) - contact intersection at the moment \( t_n \), \( D_\xi(\delta_n) \) - the magnitude of the decrease of the height of the protrusion due to plastic deformation, \( W_\xi(\delta_n) \) - change in height as a result of fatigue failure, \((V_N/2)\Delta t\) - the half value of convergence of surfaces under normal load during the time \( \Delta t = \tau \). More detail in [13,14].

4. RESULTS AND DISCUSSION

Initial data were determined on base of profilograms of specimens of the shaft and bearing: the surface \( \xi \) is the surface of the bearing surface \( \eta \) - the surface of the shaft.

For period of time, where \( h_{\text{min}} \leq h_{\text{lim}} \) we estimated next parameters: the distribution of the heights of roughness of surfaces \( (p_\xi, q_\eta) \); dependence of the roughness of the surfaces \( (\sigma_\xi, \sigma_\eta) \) on the friction distance; the average contact area; the friction force \( F_f \) and friction coefficient \( f \); the linear wear \( I_h \). Figures 1a and 1b present the height distribution of surfaces in function of the friction distance.
Figure 1a shows the result of the surfaces change after running-in. It can be seen that after the run-in, the surface of the bearing was deformed. The process of run-in was observed until the pressure on the projections of the surface became less than twice true fracture stress for the bearing.

Then there is the process of fatigue failure. High protrusions are destroyed, the elastic force of mechanical contact increases due to convergence of the surfaces, increasing the proportion of mechanical contact and to reduce the proportion of liquid contact.

Figure 2 depicts the dependence of the surface roughnesses on the friction distance \((L)\). Here you can also see the transition between the mixed and hydrodynamic regimes: the curve break at 430 km of the friction distance.

![Figure 2](image)

**Fig. 2.** The heights of surface roughness of bearing (1) and shaft (2) in function of friction distance.

![Figure 3](image)

**Fig. 3.** The dependence of wear (the convergence of surfaces) for the 2nd (curve 2) and 3rd (curve 1) main bearings on rev.

Figure 3 represents the dependence of wear (the convergence of surfaces) for the 2nd and 3rd main bearings on the number of revolutions of the crankshaft. As a result the mixed friction regime for one revolution for the 3rd main bearing is longer than for the 2nd main bearing (see Table 1), and the wear for the 3rd main bearing is more intensive.

5. CONCLUSION

1. The HMC of the "main bearing-crankshaft" tribosystem were calculated for diesel engine, which allowed us to estimate the duration of the surfaces contact interaction for the 2nd and 3rd main bearings of diesel engine.

2. We proposed a Markov model of the interaction of rough surfaces, which was supplemented by taking into account the presence of a lubricant in a mixed friction mode. It allows us to estimate the friction characteristics over the time for contact area of tribosystem taking into account the transition between the mixed and hydrodynamic regimes.

3. The proposed model can be used to estimate the service life of different tribosystems operating under different friction modes.

Acknowledgement

This work was financially supported by grants of the Russian Foundation for Basic Research (project № 16-08-00990, project № 16-08-01020).

REFERENCES


