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RESEARCH

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

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A B S T R A C T

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.

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 modified equation Elrod for pressure in a lubricant film [7]. In this paper used modification, written relative to function $\Omega(\varphi, z)$ in form:

$$\frac{\partial}{\partial\varphi}\left[\frac{\bar{h}^3}{12\mu_{eff}^*}\frac{\partial}{\partial\varphi}(g\Omega)\right]+ \\ +\frac{1}{a^2}\frac{\partial}{\partial z}\left[\frac{\bar{h}^3}{12\mu_{eff}^*}\frac{\partial}{\partial z}(g\Omega)\right]= \\ =+\frac{w}{2}\frac{\partial}{\partial z}\{\bar{h}[1+(1-g)\Omega]\}+ \\ +\frac{\partial}{\partial\tau}\{\bar{h}[1+(1-g)\Omega]\}, \quad (1)$$

where $\bar{h}=h/h_0$; $\mu_{eff}^*=\mu_{eff}^*/\mu_0$; $-a \le z \le a$; $z=z/R$; $\varphi=xR$; $a=B/2R$; $\tau=w_0 t$; $w=(w_2 - w_1)/w_0$; $\bar{h}$, μ_{eff} - dimensionless film thickness and the effective viscosity of the lubricant; B, R – the width and radius of the bearing; μ_{eff}^* – effective viscosity of the lubricant, the corresponding temperature to T_{eff}^*; μ_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; w – the dimensionless rotary shaft velocity; g – the switching function.

The main hydromechanical characteristics (HMC) of the tribosystem are: $h_{\min}(\tau)$ – instantaneous values of the minimum oil film thickness; $p_{\max}(\tau)$ – instantaneous values of the maximal hydrodynamic pressure; $h_{\min}^*$ – average value of $h_{\min}(\tau)$; $p_{\max}^*$ – average value of $p_{\max}(\tau)$; $N(\tau)$ and N^* – instantaneous and average power loss of friction; Q^* – the average flow rate of oil; T_{eff}^* – the average effective temperature of the lubricating layer; $\alpha_{h_{\lim}}$,% – the areas, where minimal thickness of a lubricant film less than the permissible value $h_{\lim}$ equal 1.5 μm, 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 $\alpha_{h_{\lim}}$ (Table 1).

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

Num. bear.	$h_{\min}^*$, μm	$p_{\max}^*$, MPa	N^*, W	Q^*, kg/s	$\alpha_{h_{\lim}}$, %
2	4.296	131.1	2056	0.067	7.8
3	4.697	131.6	1820	0.063	11.1

Table 1 shows that the $h_{\min}$ 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 $h_{\min}$ less $h_{\lim}$ (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 $\xi_k(n), (k=1,\ldots,K)$, 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:

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

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

$$k = \overline{1,K}\,,\ \eta_0(n) = \eta_K(n)\,,$$

where Φ and Ψ determine stochastic mechanism of asperity interaction as a sphere interaction [11], with random radiuses. 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 ξ 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 ξ interacts with the series of asperities of the other surfaces with heights $\eta_1,..\eta_n...$, distributed with density $q_\eta(y)$. Thus series of heights $\xi_1,..\xi_n...$is obtained, height value ξ_{n+1} depends on ξ_n and η_{n+1}:

$$\xi_{n+1} = \Psi(\xi_n, \eta_{n+1}), \tag{4}$$

where Ψ determines accepted mechanism of asperity contact interaction.

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

$$\eta_{n+1} = \Psi(\eta_n, \xi_{n+1}). \tag{5}$$

Once two functions Ψ and Φ are elaborated the system of equations for stationary distributions p^* and q^* is obtained:

$$\begin{cases} p^* = p^* \cdot P_\xi\left(q^*, \Psi\right), \\ q^* = q^* \cdot P_\eta\left(p^*, \Phi\right), \end{cases} \tag{6}$$

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

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

$$\begin{aligned} &\xi_{n+1} = \Psi(\xi_n, \eta_n) = \\ &= \xi_n - D_\xi(\delta_n) - W_\xi(\delta_n) + (V_N/2)\Delta t\,, \end{aligned} \tag{7}$$

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 ξ is the surface of the bearing, surface η - the surface of the shaft.

For period of time, where h_{min} less h_{lim} we estimated next parameters: the distribution of the heights of roughness of surfaces (p_ξ, q_η); dependence of the roughness of the surfaces $(\sigma_\xi, \sigma_\eta)$ on the friction distance; the average contact area; the friction force F_{fr} 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.

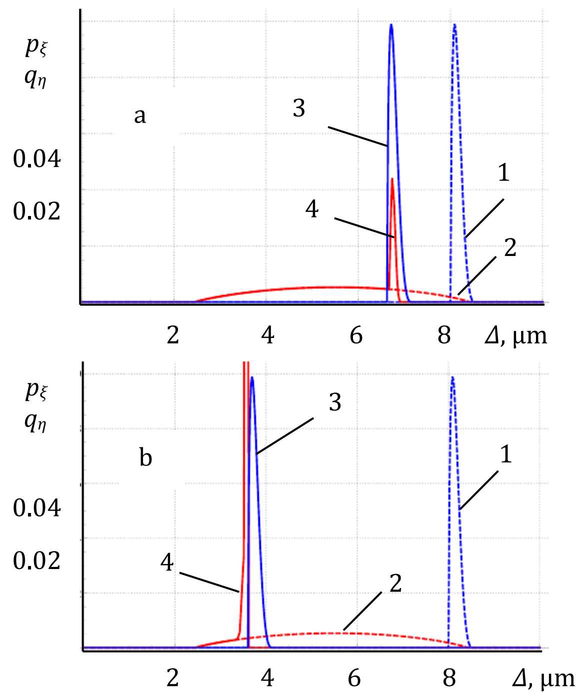

Fig. 1. The distribution of the heights of roughness of contact surfaces: after run-in time (a) and after friction distance 1380 km (b): 1, 2 - initial curves for shaft and bearing respectively; 3,4 - curves after interaction.

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.

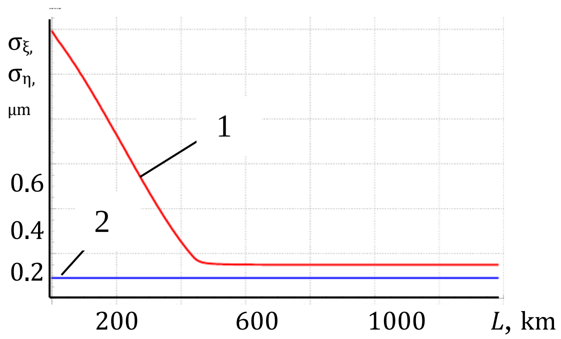

Fig. 2. The heights of surface roughness of bearing (1) and shaft (2) in function of friction distance.

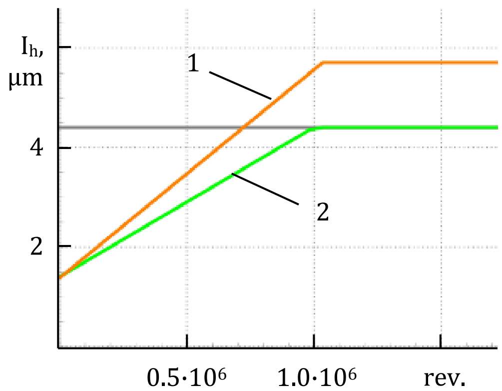

Fig. 3. The dependence of wear (the convergence of surfaces) for the 2nd (curve 2) and 3rd (curve 1) main bearings on rev.

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.

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REFERENCES

[1] A. Vencl and A. Rac, 'Diesel engine crankshaft journal bearings failures: Case study', *Engineering Failure Analysis*, vol. 44, pp. 217–228, 2014.

[2] V.E. Lazarev, K.V. Gavrilov A.A. Doikin, J. Sequrd-Base, G. Vorlaufer, 'Estimation of the tribotechnical parameters of the “piston skirt-cylinder liner” contact interface from IC-engine for decreasing the mechanical losses', in *1st International Conference on Energy Production and Management in the 21st Century: The Quest for Sustainable Energy*, Ekateringburg, Russian Federation, WIT Transactions on Ecology and the Environment, 23–25 April, 2014, pp. 625-635.

[3] P.R. Nayak, 'Random Process Model of Rough Surfaces', *Journal of Lubrication Technology*, pp. 398 – 407, 1971.

[4] J.A. Greenwood and J.B.P. Williamson, 'Contact of Nominally Flat Surfaces', *Proceedings of Royal Society, Series A*, vol. 295, pp. 300 – 319, 1966.

[5] M.B. Amor, S. Belghith and S. Mezlini, 'Finite Element Modeling of RMS Roughness Effect on the Contact Stiffness of Rough Surfaces', *Tribology in Industry*, vol. 38, no. 3, pp. 392–401, 2016.

[6] J. Javorova, A. Mazdrakova, I. Andonov and A. Radulescu, 'Analysis of HD Journal Bearings Considering Elastic Deformation and NonNewtonian Rabinowitsch Fluid Model', *Tribology in Industry*, vol. 38, no. 2, pp. 186-196, 2016.

[7] Y. Rozhdestvensky and K. Gavrilov, 'Numerical and experimental investigations of tribosystems of piston en-gines taking into account boundary lubrication regime, macro and microgeometry of contact interface', in *STLE Annual Meeting & Exhibition*, May 18-22, 2014, Disney's Contemporary Resort Lake Buena Vista, Florida, USA.

[8] Y. Goritskiy, K. Gavrilov, Y. Rozhdestvensky and A. Doikin, 'Numerical model of mechanical interaction between rough surfaces of tribosystem of the high forced diesel engine', *Procedia Engineering*, vol. 129, pp. 518-525, 2015.

[9] Y. Rozhdestvensky, K. Gavrilov, A. Doykin and I. Levanov, 'Numerical and Experimental Investigations of Piston - Cylinder Tribosystem of Diesel Engine', in *Proceeding of 5th World Tribology Congress*, WTC, pp. 1255-1258, 2013.

[10] Y. Goritskiy, Y. Ismailova, K. Gavrilov, Y. Rozhdestvensky and A. Doikin, 'A Numerical Model for Mechanical Interaction of Rough Surfaces of the “Piston-Cylinder Liner” Tribosystem', *FME Transactions*, vol. 43, pp. 249-253, 2015.

[11] K.L. Johnson, *Contact Mechanics*. Cambridge University Press, 1987.

[12] J.L. Doob, *Stohastic Processes*. New York – Jonh Willey & Sons, 1953.

[13] D. Tigetov and Y. Goritskiy, 'Markov model of mechanical interaction of rough surfaces in the process of friction', *Friction and lubrication in machines and mechanisms*, no. 3, pp. 4-13, 2010. (in Russian)

[14] Y. Goritskiy, S. Glavatskih and Y. Brazhnikova, 'Markov model of interaction of rough surfaces', *Friction and lubrication in machines and mechanisms*, no. 2, pp. 11-20, 2014. (in Russian)