Research of the friction surfaces regular microgeometry parameters effect on the hydro-mechanical characteristics of the «piston-cylinder» tribounit

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Abstract. The article provides an overview of the friction surfaces micropofiling main types of the «piston-cylinder» tribounit. The calculation model is created and the program of the tribounit calculation analysis is developed. Calculations of the tribounit hydro-mechanical characteristics for different types of microgeometry are performed.

1. Introduction
At the new stage of development of tribology, the processes of friction and wear are presented as complex processes of physical and chemical mechanics, and it is impossible to describe them without certain simplifications. In this regard, various models are widely used, among which two directions are highlighted: a mathematical description of the microrelief of friction surfaces and a model of their contact interaction representation [1-6].

The interaction of rough surfaces, as well as surfaces with a technologically specified microrelief, determines many processes in electrical, heat and mechanical engineering, automotive industry, engine construction and other areas.

Micro-profiling is designed to reduce friction in tribounits, to increase their bearing capacity, reduce wear, improve the reliability and efficiency of friction units. Thus, the main types of micropofiling in the "piston-cylinder" connection are honing and texturing.

In this article we consider some types of regular microgeometry of friction surfaces (microgrooves) of tribounit "piston-cylinder" in the internal combustion engine (ICE) and their impact on the main hydromechanical characteristics (HMC): inf $h_{min}$ – minimum thickness of the lubricant layer, $sup p_{max}$ – maximum hydrodynamic pressure, $\bar{Q}$ – average lubricant consumption, $h_{min}^*$ – average thickness of the lubricant layer, $p_{max}^*$ – average hydrodynamic pressure, $N_f^*$ - average power of friction losses.

As the parameters of microgrooves were set their depth ($h_k$), the distance between them ($b_k$), and the inclination angle relative to the vertical axis of the piston ($\alpha_k$), that allows to simulate the honing of the friction surface.

2. Method of calculation
The field of hydrodynamic pressures was determined from the Reynolds equation:
\[
\frac{d}{d\varphi} \left[ \frac{h^3}{12\mu_E} \cdot \frac{d\tilde{p}}{d\varphi} \right] + \frac{1}{a^2} \cdot \frac{d}{dz} \left[ \frac{h^3}{12\mu_E} \cdot \frac{d\tilde{p}}{dz} \right] = \frac{\bar{w}_{21}}{2} \cdot \frac{dh}{dz} + \frac{d\bar{h}}{d\tau} .
\]

Where \( \tilde{p} \) – dimensionless pressure in the lubricant layer, \( \psi = h_0 / R \) – the relative characteristic of the lubricant layer thickness; \( \tilde{h} = h / h_0 \); \( h = h_0 + h_k - e \cdot \cos \varphi + Z \cdot \tan \gamma \cdot \cos \varphi \); \( \mu_E = \mu * / \mu_0 \); \( Z = z / R \); \( -a \leq z \leq a \); \( \varphi = \varphi \cdot R \); \( a = B / 2 R \); \( \bar{w}_{21} = (w_2 - w_1) / \omega_0 R \); \( \tau, \bar{h}, \mu_E \) – dimensionless time, the thickness and the effective viscosity of the lubricant, corresponding to the temperature \( T_E \); \( B, R \) – the height and the radius of the piston skirt; \( \mu_0, h_0, \omega_0 \) – lubricant viscosity film thickness at the central position of the spike and shaft speed; \( w_1, w_2 \) – speed of the translational motion of the movable elements; \( w_{21} \) – dimensionless translational velocity of the piston movement; \( e_c \) – the offset of the center of mass of the piston from the cylinder axis; \( \gamma \) – the angle of the piston; \( \varphi \) – circumferential coordinate.

Calculation of the piston trajectory on the lubricant layer in the engine cylinder (figure 1) was made while the coordinate system was fixed on a motionless cylinder and the beginning of the movable coordinate system was located in the center of the piston mass.

**Figure 1.** The dynamics of the piston on the lubricating layer in the diesel engine cylinder.

On the picture above \( P_G \) – pressure of gases; \( P_1, P_2 \) – the inertial forces projection \( P_i \) of translational moving of the piston (attached in the center); \( e_c \) – the offset of the piston’s center of mass; \( e_\rho \) – offsetting
of the piston; \( e_0 \) – offsetting of the engine; \( R_x, R_y \) – projection of the lubricating layer reaction, \( \alpha \) – angle of rotation of the crankshaft (ROC).

Complex movements of the piston consist of translational movements along and across the axis of the cylinder (respectively, with speeds \( w = w_2 \) and \( e \)), and of rotational movement around the axis of the piston finger with speed \( \gamma \). Gravity and friction forces were not considered due to their smallness. Thus, the piston has the ability to move in the general case in the \( XOZ, YOZ \) subspaces. The acceleration of the point C along the \( OZ \)-axis was assumed to be equal to the acceleration of the piston translational motion determined in the kinematics of the plane-parallel motion of the crank mechanism [7].

The force \( R \) was considered as given force in a subspace parallel to the subspace \( XOZ \), the cylinder is non-rotatable (\( w_1 = 0 \)).

3. Software complex

For the calculation analysis of the internal-combustion engine of the hydrodynamic tribounits, taking into account the described technique, program of tribological analysis "THE MICROGEOMETRY OF THE PISTON-CYLINDER TRIBOSYSTEM" was developed [8]. The initial data were: the indicator diagram of the working process in the cylinder, the macrogeometric parameters of tribounit and the parameters of regular microgeometry in the form of radial microgrooves, cross microgeometry with different inclination angles, and sinusoidal microgeometry.

The program provides the following functions: calculation of external forces acting on the piston on the basis of the hodographs’ indicator diagram; determination of the field of hydrodynamic pressures in the lubricant layer, construction of the trajectory of the piston, calculation of tribological characteristics, determination of the friction mode and its duration by the angle of rotation of the crankshaft, as well as the optimization problem solution to determine the parameters of the macroprofile of the piston guide part.

4. The results of calculation

4.1. The adequacy check of the model

Using the developed program, comparative computational studies were performed on the example of a gasoline four-cylinder, in-line turbocharged engine. The initial parameters of the engine are presented in table 1.

| Parameter                                      | Value  |
|------------------------------------------------|--------|
| The radius of the crank, m                     | 0.045  |
| The length of the connecting rod, m            | 0.135  |
| The diameter of the piston, m                  | 0.084  |
| The area of the piston, m²                     | 0.0055 |
| The mass of the piston, kg                     | 0.434  |
| The height of the piston skirt, m              | 0.0305 |
| The nominal diameter clearance in «piston – cylinder» system, m | 0.0000145 |

Figure 2 shows the lateral force acting on the piston, and figure 3 presents the results of the calculation of friction losses in the tribounit obtained by the authors of [9] and calculated in accordance with the presented technique.
The results of the calculation indicate the qualitative and quantitative coincidence of the calculated parameters. With this in mind, it can be concluded that the developed calculation method is adequate.

### 4.2. The calculation results of a diesel engine

The calculation of the hydromechanical characteristics of the coupling is performed on the example of diesel type ChN 13/15 (4 stroke supercharged, piston diameter 130 mm, piston stroke 150 mm), the initial data for which are shown in table 2. In addition, the indicator diagram and three-constant viscosity-temperature characteristic of motor oil SAE 5W-50 with characteristic values of dynamic viscosity coefficients at a certain temperature were used as initial data: \( \mu_{40} = 0.074 \text{ Pa}\cdot\text{s}, \mu_{80} = 0.0237 \text{ Pa}\cdot\text{s}, \mu_{100} = 0.0154 \text{ Pa}\cdot\text{s} \).

| Table 2. Initial data for calculation of diesel ChN 13/15 |
|----------------------------------------------------------|
| Parameter                                                | Value  |
| The radius of the crank, m                               | 0.075  |
| The length of the connecting rod, m                      | 0.260  |
| The diameter of the piston, m                            | 0.130  |
| The area of the piston, m\(^2\)                          | 0.013  |
| The mass of the piston, kg                               | 5.035  |
| The height of the piston skirt, m                        | 0.084  |
| The nominal diameter clearance in «piston – cylinder» system, m | 0.00002 |

Table 3 presents calculations of HMC of diesel ChN 13/15.
Table 3. Results of HMC calculations

| Types of microgeometry          | inf $h_{\text{min}}$, $\mu$m | sup $p_{\text{max}}$, MPa | $h^*$, $\mu$m | $p^*$, MPa | $Q^*$, $s^{-1}$ | $N_{fr}^*$, W |
|--------------------------------|--------------------------------|----------------------------|---------------|------------|----------------|---------------|
| Without microgeometry          | 5.62                          | 7.86                       | 19.99         | 1.63       | 0.00169        | 298.99        |
| Circular microgeometry         | 4.84                          | 10.51                      | 18.92         | 2.05       | 0.00178        | 262.15        |
| Cross microgeometry            | 4.16                          | 13.80                      | 18.02         | 2.58       | 0.00186        | 236.09        |

The results indicate a decrease in the calculated friction losses in vehicles with regular microgeometry to 20% with non-critical changes in other HMC.

Graphs of the main HMC depending on the angle of rotation of the crankshaft for diesel are presented in figures 4-7.

Figure 4. Dependence of lubricant layer thickness on the angle of ROC.

Figure 5. Dependence of the power losses due to friction on the angle of ROC.

Figure 6. Dependence of the lubricant flow rate on the angle of ROC.

Figure 7. Dependence of the maximum hydrodynamic pressure on the angle of ROC.

From the graphs it can be seen that during the working cycle of a diesel engine there are no points in time where we can see the deterioration of such parameters as $h_{\text{min}}$ and $Q$. Moreover, power of friction...
losses are reduced throughout the entire working cycle, and the maximum hydrodynamic pressure increases at the moments of maximum load, which provides an increase in the carrying capacity of the coupling and prevents contact of friction surfaces.

Figure 8 shows hydrodynamic pressure diagrams in the lubricant layer for a smooth and textured piston skirt (radial microgrooves) and we see the positive effect of regular microgeometry of the piston guide surface, which consists in increasing the bearing capacity of the tribounit for the angle of crankshaft rotation corresponding to the greatest load acting on the piston (α=390 deg. ROC).

![Figure 8. Hydrodynamic pressure in the lubricant layer for α=390 deg. ROC: a) radial microgrooves, b) without texturing.](image)

5. Conclusion
The mathematical model and the program of the tribounit calculation analysis were developed. With their help, the study of influence of various kinds of regular microgeometry on the main HMC "piston-cylinder" tribounit was carried out. Its positive effect is shown, which consists in reducing friction losses (up to 20% depending on the type of microgeometry of the piston skirt surface), which are one of the main indicators of efficiency of the structure. As well as in increasing of the bearing capacity of the "piston-cylinder" tribounit at the moments of the maximum load on the piston and a possible transition to the mixed lubrication mode.

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