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Elastohydrodynamic Lubrication (EHL) of Piston Rings in the Internal Combustion Engine

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Abstract

The paper analyzes the numerical model of elastohydrodynamic lubrication (EHL) of piston rings in the cylinder liner of the internal combustion engine (ICE). Authors take into account reactions of the lubricating layer, the pressure force of the piston rings on the cylinder wall, forces of the gas pressure in the cylinder, the friction force between the upper edge of the piston ring and the piston groove. A simulation model was developed in the Fortran program and authors have analyzed characteristics and forms of piston rings in the ICE.

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Keywords: piston ring; Reynolds equation; equation for transverse vibrations of the ring; film thickness

1. Introduction

Friction losses in the piston assembly has been analyzed by Pinkus and Wikcock [1], Taylor and Coy [2], Rosenberg [3], Bartz [4], Tung and McMillan [5], Lang [6], Hoshi [7], Goto and Kai [8], Bartz [9], Enomoto [10], Taylor [11,12], Merlo [13]. According these researches the energy needed to overcome friction in these assembly is 45% (38–68%). The tribocontacts in the "piston rings-cylinder liner" pair are more complex than the other engine components and therefore it is important to create a correct model of the behaviour of the piston ring.

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2. Fundamentals

The main feature of the piston ring is its elasticity. This property is possessed of many elements of tribounits in an internal combustion engine. But this property is not taken into account with regard smallness displacements that occur in these elements by the action of external forces. For this reason, for example, piston skirt, main journals of the crankshaft considered absolutely rigid.

In addition, among the features of elastic-hydrodynamic lubrication of piston rings problem is the condition of the location of the center of gravity of the piston ring on the cylinder axis. Moreover, the center of gravity of the piston ring is located on the cylinder axis during the whole working cycle of ICE. This means that the ring does not move in a plane perpendicular to the cylinder axis. The clearance between the cylinder wall and the piston ring is changed only by the elastic deflection of the ring. Deflections are due to the hydrodynamic pressure in the "piston rings-cylinder liner" tribounit and normal force by the piston.

The calculated scheme is accepted, that the center of gravity of the piston ring located on the axis of the cylinder. The gap defined by equation of elastic displacements of points of the ring is in close relationship with the Reynolds equation that describes the hydrodynamic pressure field in the lubricating layer between the piston ring and the cylinder wall. These two equations are solved together, in which decisions are determined by the deflections of the ring and the hydrodynamic pressure field in the lubricating layer.

3. Main equations

We used equation describing the elastic bending of the piston ring [14]:

$$\frac{d^2 w(\theta)}{d\theta^2} + w(\theta) = -\frac{M_p(\theta) R^2}{EI}, \qquad (1)$$

where $M_p(\theta)$ – moment of the external forces acting on the ring; R – the radius of the median line; $w(\theta)$ – the radial displacement of the point of the ring; θ - circumferential coordinate of the ring; EI - flexural rigidity of the ring; E - Young's modulus; I - geometrical moment of inertia of the ring. Moment of the external forces acting on the ring is obtained:

$$M_p = \int_0^{2\pi} q(\theta) R^2 \sin(\theta\text{-}\varphi)\, d\theta, \qquad (2)$$

where φ – angle around the ring, in which the load is applied; $q(\theta)$ - distributed external load on the ring.

The distributed external load on the ring determines as sum of reactions of the lubricating layer $q_{hyd}(\theta)$, pressure forces of the piston ring on the cylinder wall $(q_r(\theta))$, gas pressure forces acting on the inside of the ring $(q_g(\theta))$, friction forces between the upper edge of the piston ring and the piston groove $(q_{fr}(\theta))$.

The reactions of the lubricating layer $q_{hyd}(\theta)$ is described by expression:

$$q_{hyd}(\theta) = \int_0^{\delta} p(\theta, z)\, dz, \qquad (3)$$

where δ - height of the ring; $p(\theta, z)$ – the hydrodynamical pressure in the lubricating layer of the"piston rings-cylinder liner" tribounit ; z- the cylinder axis.

Pressure forces of the piston ring on the cylinder wall can be determined by analytical equations, but the real pressure diagram can be obtained only by means of their experimental determination. In these work we used formulas from [15].

Gas pressure forces acting on the inside of the ring we assumed constant over the entire circumference of the piston ring. Friction forces between the upper edge of the piston ring and the piston groove $q_{fr}(\theta)$ depend on the coefficients of friction between the ring and the cylinder liner f_1 and between the upper edge of the piston ring and the piston groove f_2 $q_{fr}(\theta)$ also depend on the pressure forces of the piston ring on the cylinder wall $(q_r(\theta))$.

Moreover, as a lubricant layer deforms the piston ring, the pressure force of the ring on the wall of the cylinder can be increased by the elasticity of the ring, which is given by the matrix of stiffness.

For this reason, in the expression for the sum of the forces acting on the ring, it is necessary to make a term which takes into account the increase of the elasticity forces of the piston ring $q_{el}(\theta)$. This term depends on the gap between the ring and the cylinder liner and determines by stiffness of ring and its fixed.

The final expression for the distributed external load on the ring can be written

$$q_{el}(\theta) = q_{hyd}(\theta) + q_r(\theta) + q_g(\theta) \pm q_{fr}(\theta) + q_{el}(\theta). \tag{4}$$

The term $q_{fr}(\theta)$ can be either the sign "+ " and with " - " as the piston presses the ring to the right side of the cylinder wall or to the left.

It is necessary to determine the fixing conditions. The fact is that in reality the ring is not fixed, and it is moved relative to the piston groove. Let us assume that the center of mass of the ring is on the axis of the cylinder and does not move during the operation cycle. The gap between the piston ring and cylinder wall is changed only by moving the points of the ring. Since the center of mass of the ring is in one position throughout the operation cycle after mathematical transformations we obtained expression of fixing conditions:

$$\begin{cases} \int\limits_{\alpha}^{2\pi-\alpha} w(\theta)\cos\theta \, d\theta = 2x_k(\alpha-\pi) - 2R\sin\alpha, \\ \int\limits_{\alpha}^{2\pi-\alpha} w(\theta)\sin\theta \, d\theta = 2y_k(\pi-\alpha). \end{cases} \tag{5}$$

Here α - a half angle of the ring's lock; x_k and y_k - the coordinates of the center of mass of the ring.

The reactions of the lubricating layer is found by integrating of the Reynolds equation, which is a partial differential equation:

$$\frac{\partial}{\partial z}\left(\left(h^3(\theta,z)\frac{\partial P(\theta,z)}{\partial z}\right)\right)\left(+ \frac{\partial}{\partial \theta}\left(\left(\frac{h^3(\theta,z)}{R^2}\frac{\partial P(\theta,z)}{\partial \theta}\right) = 6\mu V\frac{\partial h(\theta,z)}{\partial z} + 12\mu\frac{\partial h(\theta,z)}{\partial t}.\right)\right) \tag{6}$$

Here $h(\theta,z)$ - film thickness; $P(\theta,z)$ - hydrodynamic pressure field; μ - the coefficient of dynamic viscosity of the lubricating oil; *V*- the linear velocity of the ring along the cylinder liner, which is equal to the speed of the piston; *t*- time of the operation cycle.

As a result of the solution of Reynolds equation we find the field of hydrodynamic pressure $P(\theta,z)$ in the lubricating layer of the "piston rings-cylinder liner" tribopair. After that we find the reaction of the lubricating layer around the circumferential direction of the piston ring from the equation (3). Detail algorithm of integration of Reynolds equation in detail is considered in [16-20]. The main characteristics of the "piston rings-cylinder liner" tribopair are maximum hydrodynamic pressure $\sup P_{\max}$, the minimum film thickness $\inf h_{\min}$, flow rate *Q* and friction losses *N*.

4. Results and discussion

Joint solution of Reynolds equation (6) and equation describing the elastic bending of the piston ring (1) was carried out by the created software in Fortran programming.

For calculating example of the "piston rings-cylinder liner" tribopair we has been selected the diesel engine ChN 15/16. The initial data were accepted indicator diagram, a parabolic profile of the piston ring, the dynamic coefficient of viscosity of engine oil $\mu = 0,012$ Па • с , the ratio of the crank radius to the length of the connecting rod $\lambda = 0,75$ and the radius of the median line of the piston ring R = 73 mm. Excess hydrodynamic pressure on the edges of the rings are taken to be zero. These boundary conditions are not entirely correct for the first compression ring, but are adequate for the third ring.

The calculation results show that hydrodynamic pressure field in the lubricating layer, the minimum film thickness in the circumferential direction of the ring and shape of the piston ring depend on the linear velocity of the ring along the cylinder liner. The velocity of the piston reaches the highest modulus values in the middle of the operation cycle. The hydrodynamic pressure in the lubricating film between the piston ring and the cylinder liner at this moment achieves the maximum value, which compresses the ring and forming its largest flexure.

Hydrodynamic pressure fields and forms of the piston ring matching the 180° and 360° of crankshaft position are presented in Fig. 1, 2. It was under these position the crankshaft, piston's velocity is minimum, and as a consequence the minimum thickness of the lubricant layer.

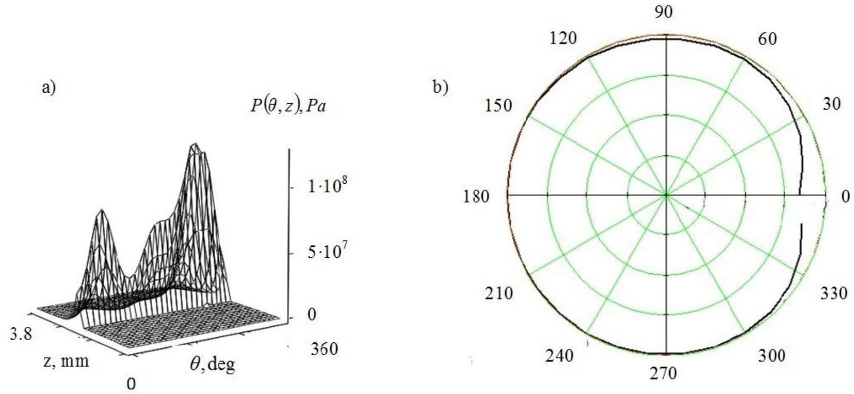

Fig.1. Diagrams of hydrodynamic pressure in the lubricating layer (a) and shape of the piston ring (b) (180° of crankshaft position, $\inf h_{\min} = 0,020 \mu m$)

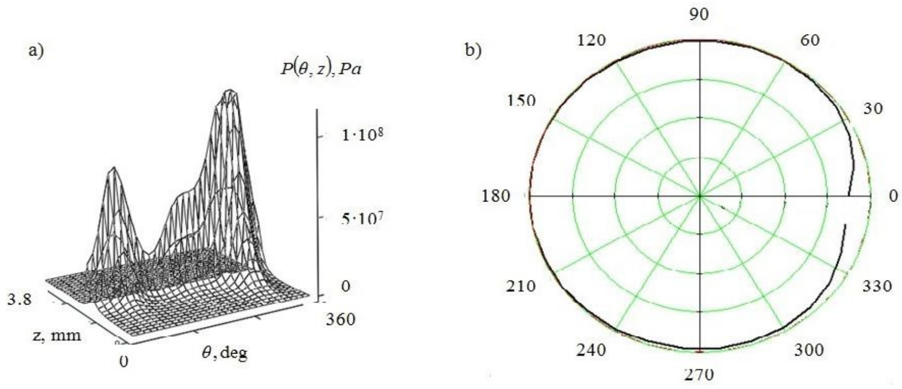

Fig.2 Diagrams of hydrodynamic pressure in the lubricating layer (a) and shape of the piston ring (b) (360° of crankshaft position, $\inf h_{\min} = 0,024 \mu m$)

Analyzing the results, it can be concluded that at top dead center (TDC) and bottom dead center (BDC) we deal with a non- hydrodynamic friction mode. The minimum film thickness, according to the calculations is 0,02μm, while the height of roughness of contact surfaces according to experimental measurements of samples reaches 1,5 - 2 μm.

5. Conclusion

1. Authors developed a numerical model of elastohydrodynamic lubrication of piston rings in the cylinder liner took into account complex of external operating factors, including reactions of the lubricating layer in the gap between the piston ring and cylinder liner.
2. Characteristics of the "piston ring - cylinder liner" tribopair were calculated for diesel engine. It allowed us to get the shapes of the ring and estimate the values of the film thickness during operation cycle of engine, which were about 0,02 μm. This indicates the mixed or boundary lubrication conditions.
3. Estimations agree with experimental observations.

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