EVALUATION OF FRICTION LOSSES ACCOMPANYING THE PISTON RING OPERATION AT RUPTURED OIL FILM

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Abstract

Conditions of engine piston and rings lubrication vary continuously at engine start and further run. Varying gas and inertia forces acting upon the elements of piston–cylinder assembly together with changeable temperature cause the changes in oil film distribution. When carrying out the computations of oil film distribution it is possible to point those regions where the oil film rupture occurs and the fluid friction becomes the mixed one. Beside the mentioned above conditions of collaboration also other factors affect the type of friction. They are related to the conditions of collaborating surfaces as well as the volume, grade and viscosity–temperature characteristics of lube oil.

Taking into account all those reasons the authors decided to define – using the computer modelling method – correlation between the characteristic parameters of compression ring operation and the friction losses when the ring runs in the border conditions between fluid and mixed lubrication.

1. Introduction

Formation of continuous oil film between the collaborating surfaces of piston ring and cylinder bore depends both on the design of these elements (dimensions, shape, micro- and macro geometry of rings and bore) and on conditions of their operation (speed and load). The quantity, grade and especially viscosity of the lube oil affect considerably the phenomena observed between those surfaces [7, 8].

The complex computations, most often using the sophisticated mathematical models of piston-cylinder assembly are needed for determination of conditions necessary for formation of the continuous oil film. Such models take into account changeability of numerous parameters related to the operation of engine and affecting the course and conditions of that cooperation. This changeability could lead to the exceeding of the momentary load capacity and rupture of the oil film. It means that even within the individual cycle of engine operation time periods and regions on bore can happen where the oil film remains continuous and such where the direct contact of asperities is observed.

The computer programs used so far by the authors assumed a certain critical oil film thickness h_{kr}, which surpassed cause an instantaneous change from the fluid lubrication to the border one ($h<h_o$, see Fig. 1). However, the experimental tests show that even a short period of collaboration of ring and bore in conditions of mixed lubrication could substantially affect the friction losses and wear of those surfaces. Due to that, this paper presents a trial towards taking into consideration the mixed lubrication in the piston-cylinder assembly model and achieved results.

2. Piston ring resistance in conditions of fluid and mixed lubrication

Fulfillment of several basic requirements resulting from the hydrodynamic theory of lubrication is necessary in order to achieve a correct collaboration of engine elements divided with oil film, such as journal and bush or piston ring and bore. To the most important one could classify the existence of convergent lubricating gap, relative motion of collaborating surfaces and sufficient oil supply between those surfaces. To achieve a full separation of the collaborating surfaces the oil film thickness should be such as to avoid the direct contact of both surfaces asperities.

The value of coefficient χ, called a specific oil film thickness [3] could be recognized as a measure of surfaces separation:

$$\chi = \frac{h_m}{R_{a,1} + R_{a,2}}, \tag{1}$$

where : h_m – oil film minimum thickness,
R_a – mean deviation from the medium roughness of both collaborating surfaces. Indexes “1’ and “2” relate to the collaborating surfaces, e.g. those of ring and bore.

It is being assumed that for a full separation of still not run-in collaborating surfaces a lubricant layer should be at least five fold the combined expression $R_{a,1} + R_{a,2}$. After the running-in period, when the asperities have been smoothed this thickness can be smaller, i.e. $h_m = R_{a,1} + R_{a,2}$.

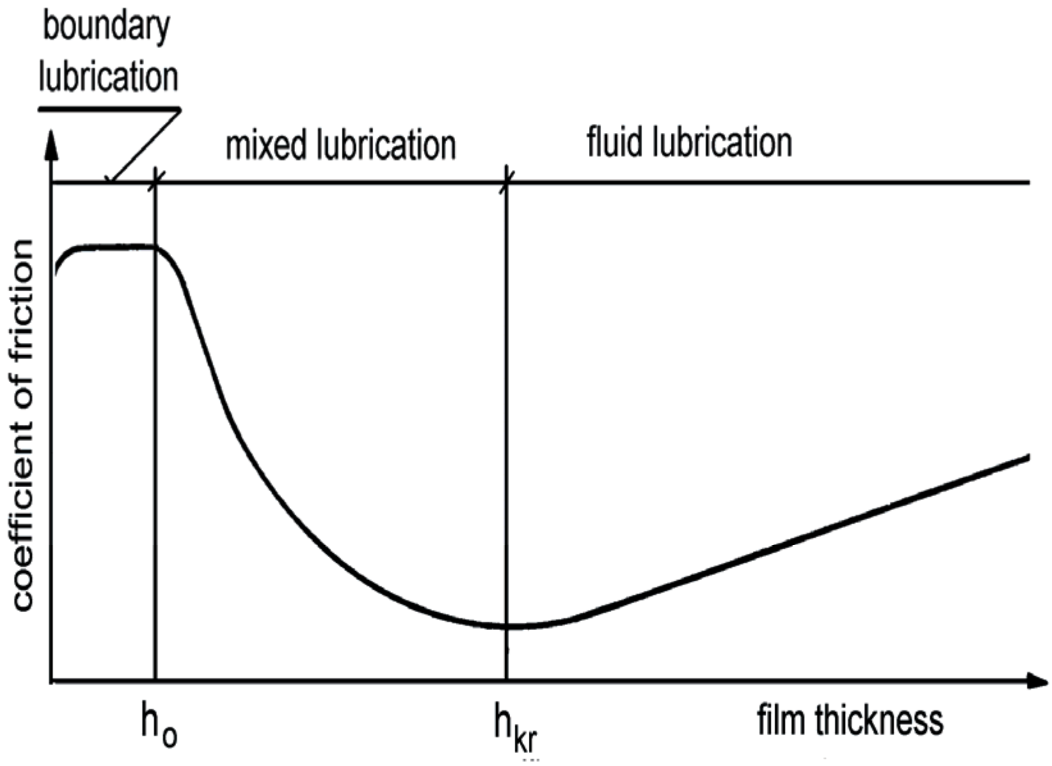

Fig. 1. Changes in coefficient of friction vs. oil layer thickness [2]

However, if these conditions are not fulfilled, the mixed or boundary lubrication can occur, depending on the layer’s thickness h (see Fig. 1).

The thickness of oil layer between collaborating surfaces depends on their relative velocity v, loading p and oil viscosity η. These quantities are taken into account in so called Hersey’s coefficient, that affects the course of lubricant layer thickness as it has been shown in Fig. 2.

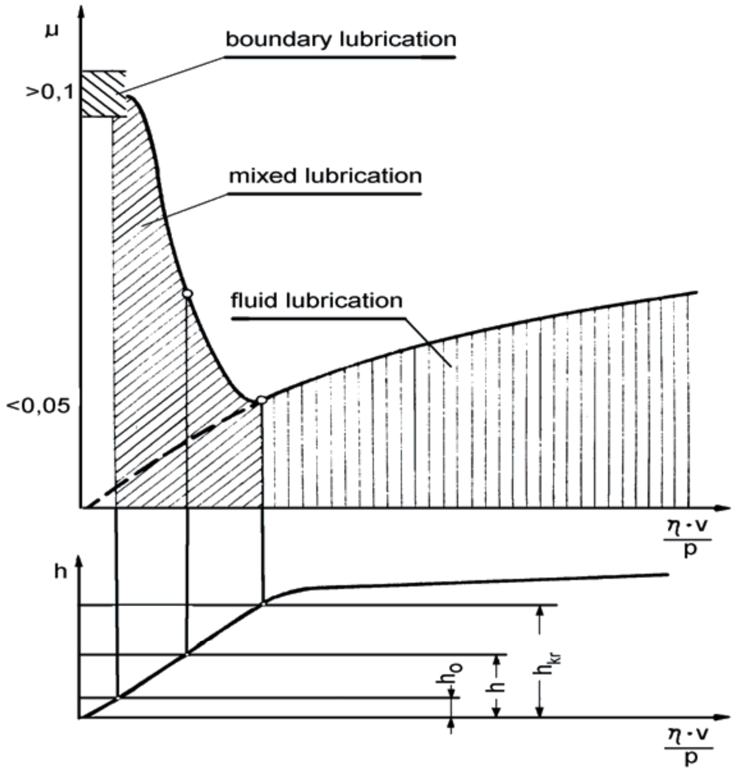

Fig. 2. Changes in the coefficient of friction μ vs Hersey's coefficient [2]

In the case of fluid lubrication (for $h>h_{kr}$ – Fig. 1), when the collaborating surfaces are fully separated with the oil film, the friction force t accompanying the ring motion is given in following form resulting from the hydrodynamic theory of lubrication:

$$t = \pi \cdot d \cdot h_m \cdot \left[(p_b - p_a) T_p + \frac{\eta \cdot u \cdot b_f}{h_m^2} T_u + \frac{\eta \cdot v \cdot b_f^2}{h_m^3} T_v \right], \qquad (2)$$

where: p_a, p_b - outer pressure loading the ring,
u, v - ring speeds, axial and radial, respectively,
η - lubricating oil viscosity,
h_m - minimum oil film thickness,
t - friction force of ring against bore,
d - bore,
T_i - coefficients depending on shape,

When the oil film thickness is lower than the critical one ($h<h_{kr}$) the oil film rupture occurs and further collaboration proceedes in the conditions of mixed lubrication. At this moment a part of loading is still carried by so called microwedges of oil (as for fluid lubrication) and another part – by the direct contact of asperities on both collaborating surfaces.

Taking into account the mixed lubrication in the model of piston-cylinder assembly requires a knowledge of mathematical description of phenomena present at the microregions of contact including the model of contact of rough surfaces in particular. An ample literature of this problem contains description of a number of such models [1, 2, 6]. Most of them assumes that:

- microroughness on the collaborating surfaces can be modeled with simple geometric figures,
- asperities contact each other with the highests points,
- the rough surface can be described directly with teh profile parameter values.

The earliests models of contact it has been assumed that the tops are cylinders of equal diameter fixed in rigid basis. On the other hand, other models assumed the asperities as spherical caps of different statistical distribution depending on surfce close-up. However, as the authors of models underline, there is no method of evaluation of asperities distribution density.

It is of comparable importance for the description of surface collaboration to define the form of contact deformations (elastic, plastic or elasto-plastic) which accompany the contact of rough surfaces, because their form decisively affect the value of frictional resistance and wear. As the experimental tests prooved, the range of presence of individual deformations depend on material stiffness and asperities height. Alas, there is no general solution of the problem of elasto-plastic deformations for the range of R_a = 0,1 - 5 µm. It turns out that a number of solutions based upon the experimental data give results burdened with substantial errors.

Also the methods of computations of real contact surface and of individual collaborating surfaces' share in loading are burdened with simmilar errors but of bigger magnitude.

Taking all of this into account the authors decided not to enrich the model of piston-cylinder assembly with earlier mentioned models of rough surface collaboration but introduce approximate calculations which make use of the relation between the oil film thickness and the type of lubrication to the algorithm of friction losses calculation. Following principles of calculations have been assumed:

- a critical thickness of lubricant layer h_{kr} separates the regions of fluid and mixed lubrication,
- the friction power is calculated according to the formula (2) resulting from the hydrodynamic theory of lubrication over the region of fluid lubrication ($h > h_{kr}$) where the coefficient of friction increases with the increase in oil film layer thickness,
- for the oil film layer thickness $h < h_{kr}$ the value of friction coefficient μ increases along with the drop in this thickness,
- the share of load carried by the oil microwedges decreases with the drop in oil layer thickness from 100% for $h > h_{kr}$ to 0% for $h = h_o$.

The value of oil layer critical thickness h_{kr} depends on, among others, microgeometry of collaborating surfaces of ring and bore and on dimensions of debris in lubricating oil. For a changeable oil film thickness h the selection of its value will affect the beginning of collaboration in conditions of mixed lubrication and on frictional resistance. For example, the increase in h_{kr} from the value in point 1 to that of point 2 (Fig. 3) causes an earlier increase in coefficient of friction as well as different course of change in its value.

Using the presented principles (taken into consideration in the simulation program describing the operation of piston-cylinder assembly) the relevant computations concerning the course of frictional losses in various conditions of collaboration between the compression ring and bore have been carried out.

3. Results and their analysis

The technical data of the International Harvester DT-466 engine parts had been used as the input data for the simulations. The research included the collaboration of the bore with a new ring and, for comparative reasons, with a worn one (after 7 000 hours of operation). Ring characteristic parameters of major importance have been collected in Table 1.

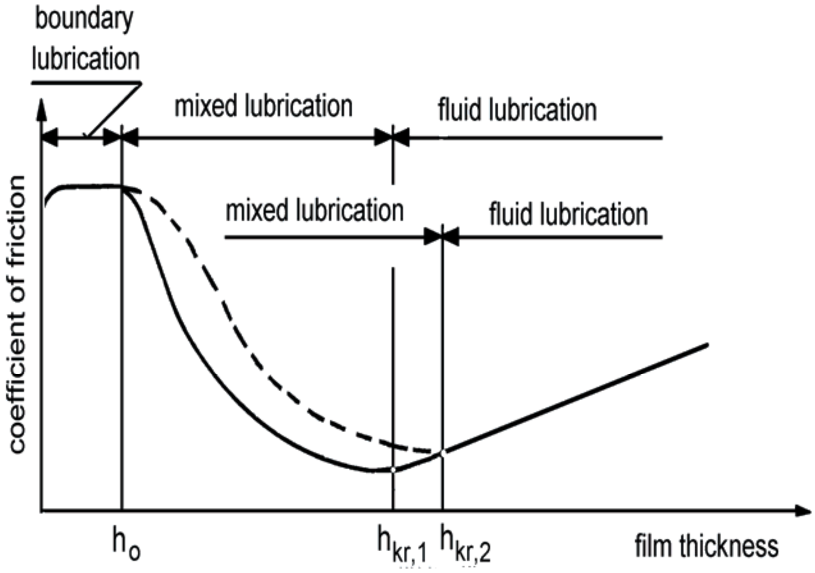

Fig. 3. The effect of selection of oil film critical thickness h_{kr} on the coefficient of friction value

Table 1. Results of measurements and calculations of the DT-466 engine first compression ring characteristic data vs. the run time [3]

Ring designation	Run time	Bore	Axial height	Ring face rise	Pressure
	h	mm	mm	µm	MPa
A	0	109	2,9	16	0,165
B	7000	109	2,4	2,2	0,123

The preliminary assumption was that the further described research would be carried out at the load corresponding to the engine full power characteristics for the speed within the range from 25 to 250 rad/s. Such assumption allowed for a crucial to the research provisional evaluation of oil film temperature distribution over the bore between the both dead centers and following assessment of viscosity along the bore generatrix [4, 5]. The mineral lube oil SAE 15W/40 grade made by the Gdańsk Refinery has been chosen for lubrication.

As mentioned earlier, the fundamental aim of the research was to determine the correlation between some characteristic parameters of ring shape and its operation in engine, and the magnitude of frictional resistance accompanying the ring motion in conditions of fluid and mixed lubrication.

In order to achieve this, a number of simulation tests have been carried out. It is worth noting that the introductory assumptions concerning the operation were often different from those in real engine. For instance, it has been assumed that piston had only one compression ring and the oil layer thickness is constant and equals 1 µm. However, such simplifications were necessary, because it allowed for a full answer to many important questions that could not be answered in case of a real engine.

As it results from the courses presented in Fig. 4, for a new ring (A) the maximum oil film layer depends on the crankshaft velocity to the minor extent, though the variations of the thickness are noticeable close to the TDC. This implies the fluctuations in friction force distribution, especially remarkable around the TDC.

a) *b)*

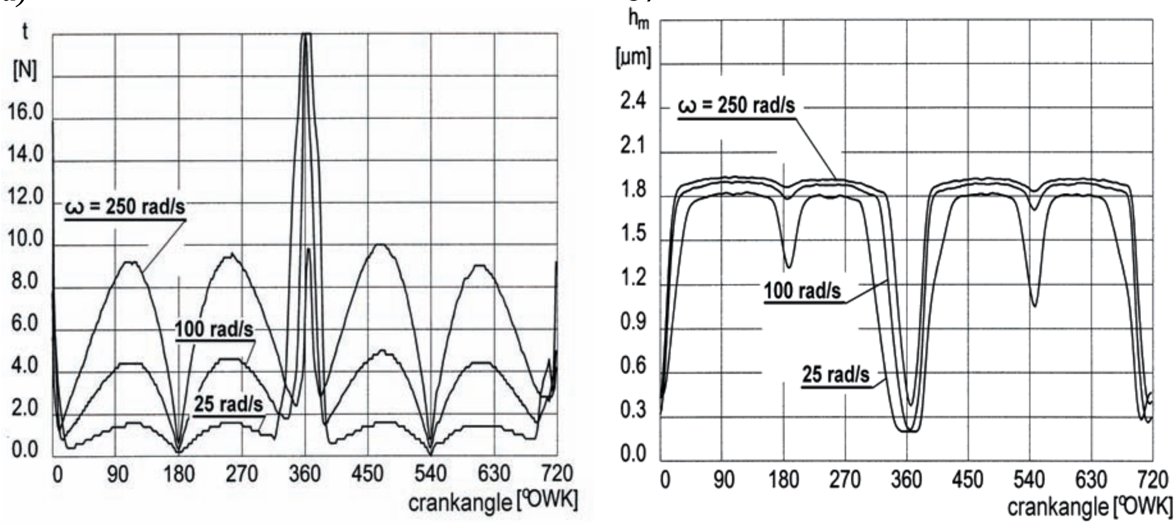

Fig. 4. Exemplary courses of minimum oil film thickness h_m (a) and friction force t (b) under a new compression ring; SAE 15W/40 grade lube oil, layer h_{kr} – 1 μm

The presented calculations have been repeated for an already used compression ring – B in Fig. 5.

a) *b)*

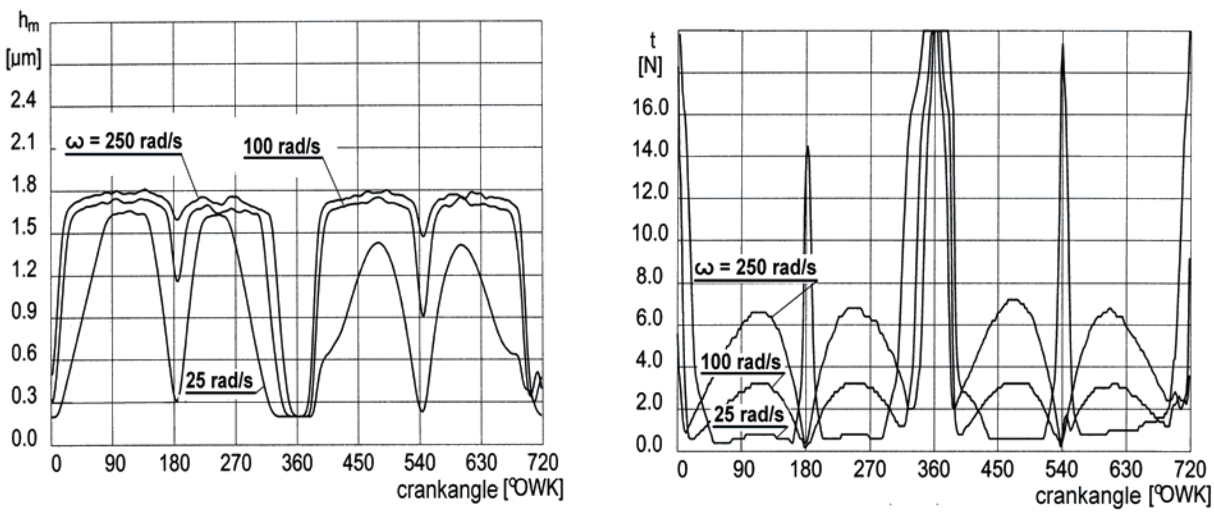

Fig. 5. Exemplary courses of minimum oil film thickness h_m (a) and friction force t (b) found out for an used compression ring; SAE 15W/40 grade lube oil, layer h_{kr} – 1 μm

Changes in the ring face geometry brought about by its wear cause a different distribution of oil film, and eventually the friction forces. One can notice significant drop in oil film thickness not only around the TDC, but also near the BDC which leads to the relative changes in oil film distribution in the vicinity of dead centers.

As the detailed calculations revealed, the average values of friction force and friction power are smaller for a new ring, except the low velocities when the collaboration is performed in conditions of mixed or even boundary lubrication (see Fig. 6). Although due to the decrease in ring axial height the momentary values of friction force are lower outside the dead centers but concurrently the ring load capacity and oil film thickness close to the dead centers also decreased which decide about the increase in friction resistance average value.

a) b)

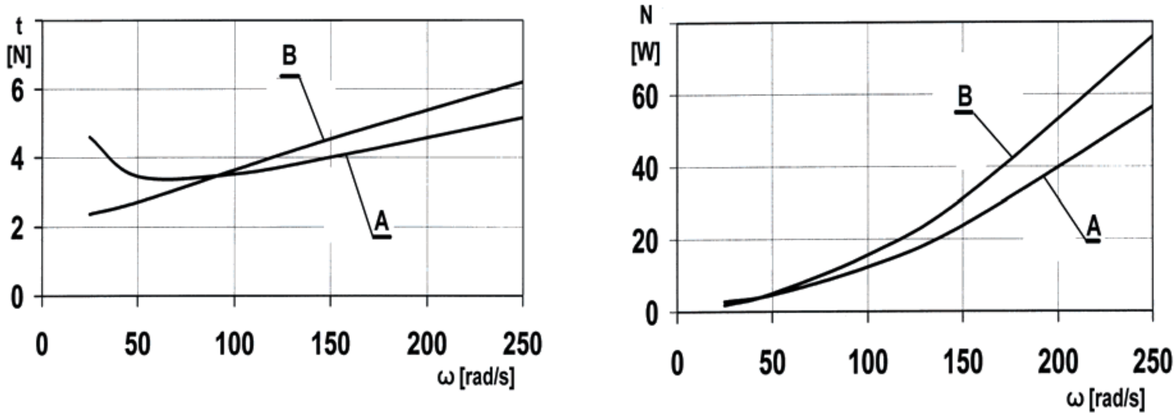

Fig. 6. Courses of friction force (a) and friction power (b) for new ring (A) and used one (B) vs. crankshaft speed ω, for h_{kr} = 1 μm

a) b)

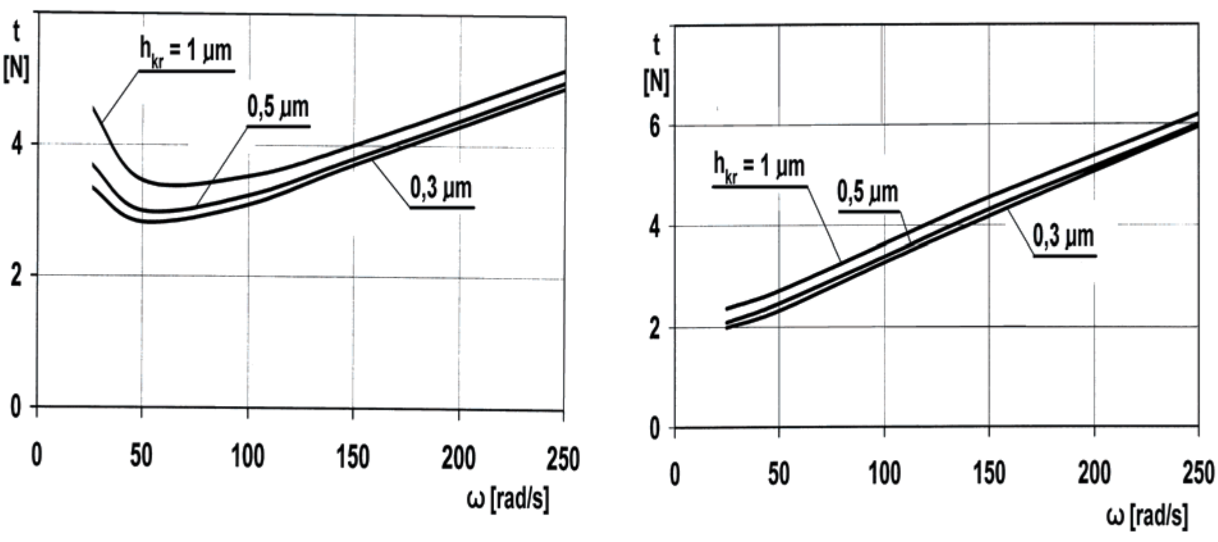

Fig. 7. Courses of friction force t for new ring (a) and used one (b) vs. crankshaft speed ω, for selected values of oil film critical thickness

As mentioned above, the moment of oil film rupture has been connected with its thickness assuming that it happens when the thickness falls below the critical value h_{kr}. At the successive stage of simulation tests the effect of film thickness on friction power has been investigated, assuming the following values: 0,3, 0,5 and 1,0 μm (Fig. 7).

As it could be expected, the increase in friction resistance (due to the earlier rupture of oil film and mixed lubrication) accompany the increase in thickness critical value apart from the condition of tested ring (ring face wear).

Summarizing this stage of investigation, it can be stated that though in correctly designed piston-cylinder assembly only on a limited section of bore the collaboration of ring and bore occurs in conditions of mixed lubrication, the accompanying increase in resistance substantially affects the combined value of friction losses.

At the next stage of tests the authors tested the effect of oil grade (viscosity and elasticity) on motion resistance, also in conditions of mixed lubrication. However, due to the limited volume of this paper the achieved results will be presented in further papers.

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