IOP Conf. Series: Materials Science and Engineering **148** (2016) 012068 doi:10.1088/1757-899X/148/1/012068

The problems of piston skirt microgeometry in combustion engines

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Abstract. Geometry of the slot between piston bearing surface and cylinder bore affects the friction losses of the IC engine to the far extent. It appears that these losses depend more on the area covered with oil than the thickness of oil layer separating collaborating parts. Barrel-shaped or stepwise piston bearing surface is the way to reduce the oil covered area. Turns out that the referred to friction losses contributes more to area covered by the oil film than the film thickness of the separation elements cooperating. The method to reduce the area covered by the oil film is a modification of the bearing surface of the piston by adjusting the profile. This paper presents the results of simulation leading to the reduction in friction losses and abrasive wear of piston bearing surface and cylinder bore. Covering the piston bearing surface with a thin layer of graphite one can get an extremely advantageous tribological properties of the piston assembly which means the expected parameters of oil film and in a case of film rupture-an ignorable abrasive wear of the graphite layer and/or cylinder bore.

1. Introduction

The advancement of modern engines is chiefly aimed at environment protection and low fuel consumption [3]. To meet the above-mentioned criteria, engines are designed based on the idea of downsizing characterized by a reduction of their displacement and increasing their effective pressure by the application of a variety of supercharging systems. The consequence of these changes is that engines operate at high thermal and mechanical loads. This is an adverse phenomenon for the piston-crankshaft assembly, particularly for pistons exposed to high temperatures and pressures inside the cylinder [4,6]. Modern engines are also expected to provide the highest overall efficiency. The way to ensure that is to reduce the friction loss in the piston-cylinder pair [7,15].

The methods to reduce the friction loss in the piston-cylinder pair are a subject of many publications that describe all types of design solutions, new materials or the application of surface layers of anti friction properties on the piston skirt [1,2].

The geometry of the gap between the piston skirt and the cylinder liner greatly affects the friction loss inside the engine. This means that the said friction loss is much more affected by the area covered by the oil film separating the mating elements than its thickness. A way to reduce the area covered by the oil film is the application of a stepped profile of the piston skirt [10,11,12]. According to theoretical considerations based on hydrodynamic theory of lubrication, the stepped shape of the gap reduces the friction loss.

The piston skirt microgeometry can be formed by the application of surface layers made from materials of good friction properties. The desired geometry of the profile can be achieved by applying thin coats on the cylindrical or conical surface of the piston skirt [5,8,14].
Modern coats are evenly applied on the surface of the piston skirt to modify the conditions of boundary friction. This, however, does not significantly affect the conditions of occurrence of liquid friction in the oil film. The most widespread coating technology is that based on graphite composites.
The results of investigations on the application of modifying layers applied on the piston skirt have confirmed a reduction in the friction loss in the piston-cylinder pair. A question arises, however, whether the microgeometry of the applied coat is optimal.
The paper presents the problems of microgeometry on the piston skirt of a combustion engine. The authors also present the results of a simulation research leading to a reduction of the friction loss and the improvement of the oil film parameters for the modified profile of the piston skirt.

2. The piston skirt profile

Piston skirt is to ensure axial displacement of the piston inside the cylinder and a transfer of lateral forces on the cylinder liner.
In reality, however, the cooperation of the piston skirt with the cylinder occurs on a much smaller area, which results in a reduction of stresses between these surfaces. Taking this fact into account, it is easily observable that the piston skirt operates in much worse conditions than those of generously lubricated sliding bearings. The lubrication of the piston skirt is much scanter. Ensuring the required durability of the piston skirt under such conditions is a very difficult task requiring a careful selection of materials and design solutions [13].
The distribution of temperatures on the piston circumference is uneven. The presence of the kingpin hubs and parts connecting the hubs with the piston makes the cross-section of this spot greater than the rest of the circumference. Parts of greater cross-section have lower thermal resistance, which makes the temperature on the piston skirt in these regions much higher than on the plane perpendicular to the kingpin axis. This results in greater thermal deformations towards the kingpin axis, which is why the piston skirt must be oval with a high ovality axis perpendicular to the axis of the kingpin and a low ovality axis alongside the kingpin to ensure that under operating temperatures the piston skirt becomes cylinder-shaped or close to it.
The operating force acting on the piston influences not only the friction conditions between the piston and the cylinder liner but also deforms the piston skirt, which is why, when cold, the guiding element should be shaped in such a way as to ensure cooperation with the cylinder liner on the largest possible area upon thermal deformation under operating temperatures and under the influence of the lateral forces.
The above-mentioned factors render the geometry of the piston skirt very complex, particularly in engines of high power ratio. When the piston reaches its operating temperatures, the play between the piston and the cylinder liner changes. Changes in the play also occur after any change of the operating conditions following the stabilization of the thermal equilibrium.
Such play variations are disadvantageous for a variety of reasons. One of them is the difficulty to ensure optimum play in terms of friction and lubrication under a sufficiently wide range of conditions. An increase in the play under lower operating temperatures results in an increase in dynamic forces exerted by the piston on the cylinder, vibration of the cylinder sleeve as well as engine noise and may lead to cavitation damage of cylinder liners in water-cooled engines.
The shape of the piston skirt in the axial cross-section, perpendicularly to the kingpin axis, and in the transverse cross-section should be designed in such a way as to ensure the maximum area of cooperation between the piston skirt and the cylinder liner under operating temperatures and under forces exerted on the piston skirt.
A modification of the geometry of the piston skirt consists in choosing such a shape of the surface that will ensure the continuity of the oil film during the piston reciprocating motion with a possibly low value of friction loss in the piston-cylinder pair.
The adopted microgeometry of the piston skirt is a result of a continuation of former research. The authors assume an application of a layer of graphite of the shape of letter H on the piston skirt. Thus

IOP Conf. Series: Materials Science and Engineering **148** (2016) 012068 doi:10.1088/1757-899X/148/1/012068

obtained stepped profile is characterized by good tribological properties under liquid friction and owing to the application of graphite, one can expect good properties under boundary friction.
In order to ascertain the friction loss in the piston-cylinder pair, 6 variants of piston microgeometry have been developed as shown in Figure 1-2.

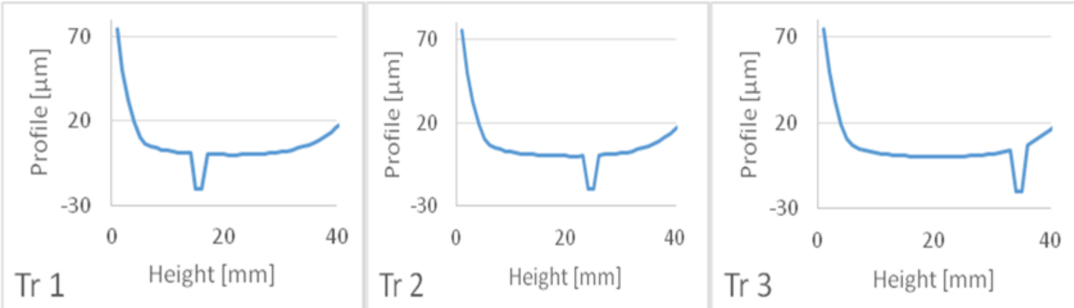

Figure 1. Tr1, Tr2, Tr3 variants of the piston bearing surface profile.

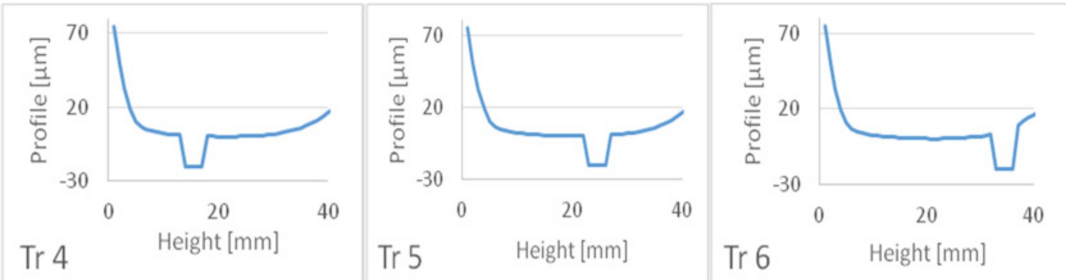

Figure 2. Tr4, Tr5, Tr6 variants of the piston bearing surface profile.

The horizontal bar of the letter H allows extending the area of high hydrodynamic pressure at the same time obtaining the effect of reduction of the oil film internal friction forces. The problem consists in determining the thickness proportion of the H bar guaranteeing high oil film bearing capacity at a limited internal friction force. Because the correctness of the computer model was validated experimentally in earlier research, the results confirm the positive results of the modification of the piston skirt microgeometry of a specified thickness of the H bar, which proves the validity of the replacement of classic barrel-shaped profile of the piston skirt with the stepped one by the application of a layer of graphite of a specified thickness.

3. Results of the simulation

The calculations of the parameters of the oil film were performed for different microgeometries of the piston skirt, starting from the barrel-shaped profile of the reference pistons and ending with the stepped piston skirt profile [9]. The thickness of the H bar is variable as well as its height in the bottom or top position against the symmetrical one. In the form of diagrams, figures 3-6 show the results of the simulation of friction loss for barrel-shaped reference pistons and the pistons of a modified piston skirt. Based on the performed analyses one can observe that the developed variants of microgeometry of the piston skirt have given the expected reduction of the friction loss in the piston-cylinder mating pair, compared to the reference pistons.

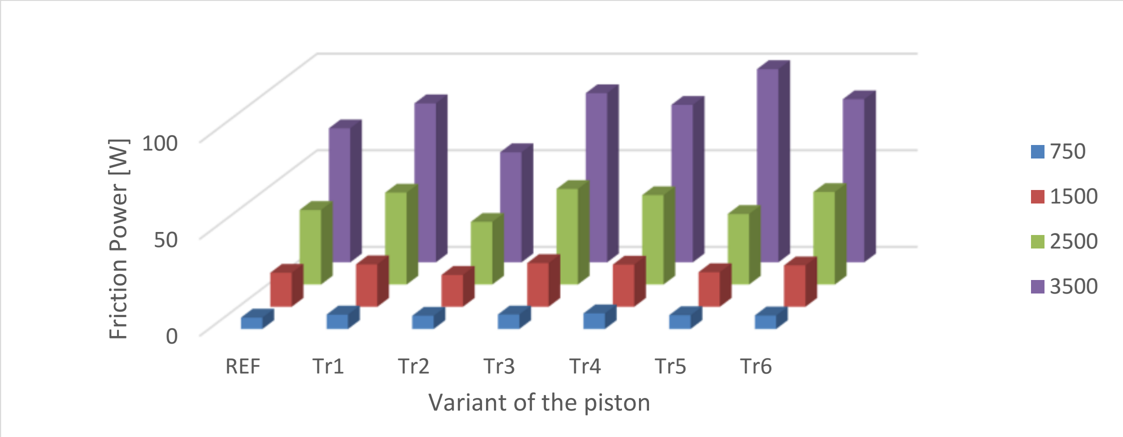

Figure 3. Friction loss in the piston skirt-cylinder liner pair as a function of engine speed and oil temperature of 100 °C.

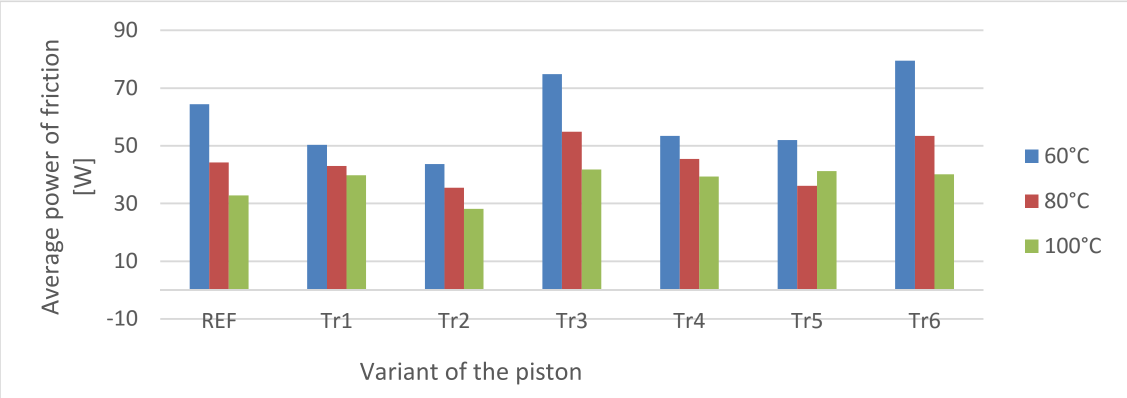

Figure 4. Results of average simulation of the friction loss in the piston skirt-cylinder liner pair for different oil temperatures.

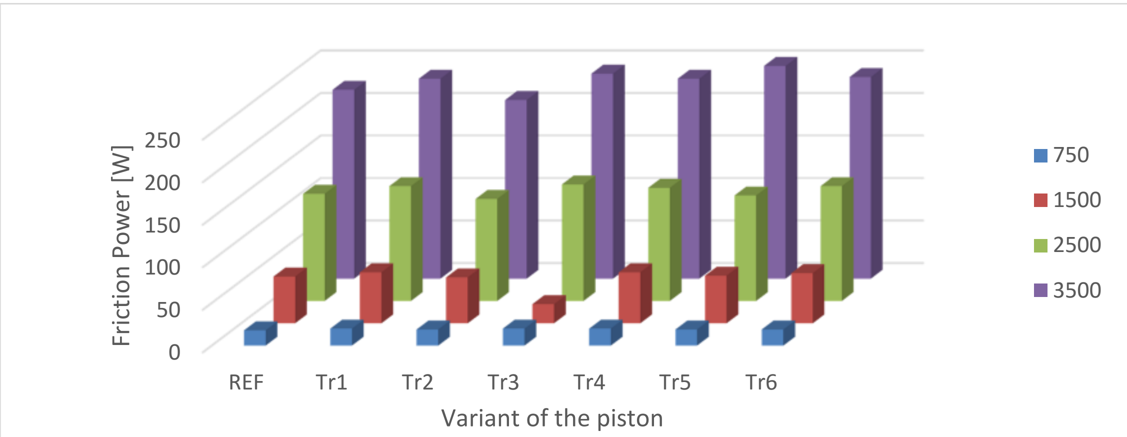

Figure 5. Friction loss in the pair (piston with a set of rings-cylinder liner) as a function of engine speed and oil temperature of 100 °C.

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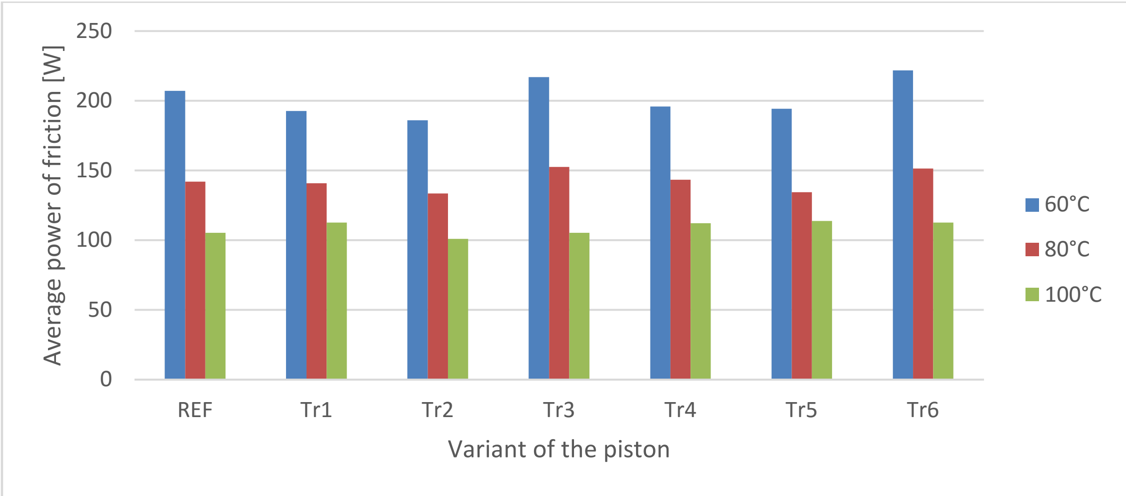

Figure 6. Results of average simulation of friction loss in the pair (piston with a set of rings-cylinder liner) for different oil temperatures.

As it turns out, the most advantageous solution is the symmetrical position of the horizontal H bar, as the friction loss decreases by approx. 10% compared to pistons of a standard profile. One could also observe that the reduction in the friction loss is greater for higher engine speeds. This is an important observation because, due to increased friction loss converted into heat when increasing the engine speed, the engine elements warm up faster and the oil viscosity drops, which may lead to boundary friction. This phenomenon may reduce the friction loss assuming that boundary friction in the piston skirt - cylinder liner pair does not occur.

From the equations of hydrodynamic lubrication theory we know that in order to form a hydrodynamic oil film one needs a narrowing oil gap. As results from the performed simulations, the microgeometry of the piston skirt significantly influences the friction loss in the piston - cylinder pair. The obtained results are explained by the graphs (figures 7-9) presenting the coverage of the piston skirt with oil film. The graphs have been made for the reference piston and the TR2 and TR5 variants for the engine speed of 2500 rpm with the following parameters:

- Nr [W] - friction loss in the piston skirt - cylinder liner pair,
- Nc [W] - total friction loss of the piston with a set of rings.

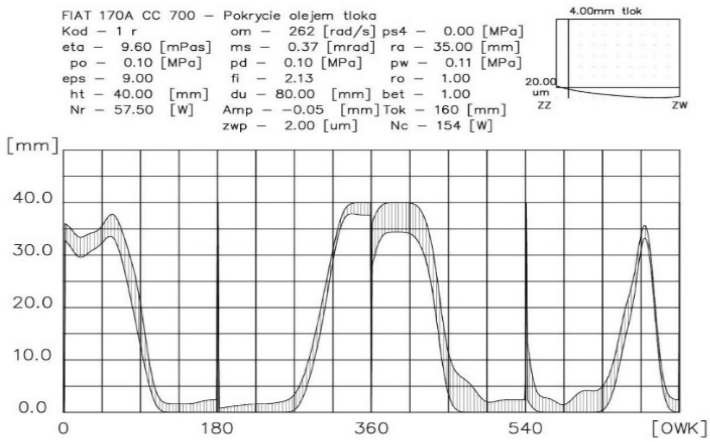

Figure 7. Oil film height on the reference piston skirt.

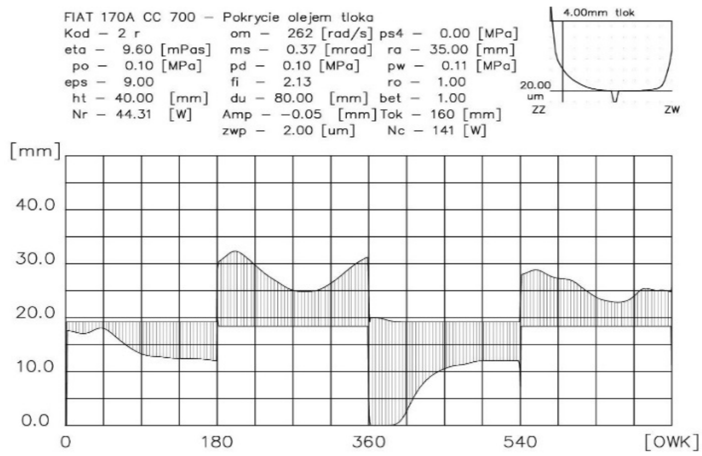

Figure 8. Oil film height on the TR2 piston skirt.

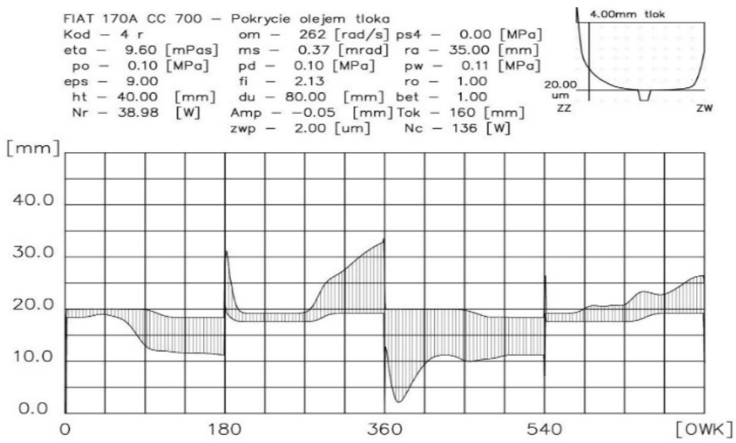

Figure 9. Oil film height on the TR5 piston skirt.

Comparing the oil films on the piston skirts one can observe that intense oil film results in an increased friction loss. If the differences in the oil film level on the piston skirt are small, even minimum variations are impactful on the friction loss. The oil film thickness is greater for the stepped profile.

4. Conclusions

- Replacing the barrel-shaped piston skirt with the stepped one may yield a reduction of the friction loss of approx. 10%.
- The stepped surface can be formed by applying coats such as graphite.
- The coats are particularly needed when lack of continuous oil film occurs i.e. the engine is off and its temperature equals the ambient temperature.
- Replacing the barrel-shaped piston skirt with the stepped one reduces the cost of the sliding surface processing of the piston and allows preserving greater tolerance at constant cost of the piston production.

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