Influence of sliding surface roughness and oil temperature on piston ring pack operation of an automotive IC engine

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Abstract. In the paper a comprehensive model of a piston ring pack motion on an oil film has been presented. The local oil film thickness can be compared to height of the combined roughness of sliding surfaces of piston rings and cylinder liner. Equations describing the mixed lubrication problem based on the empirical mathematical model formulated in works of Patir, Cheng and Greenwood, Tripp have been combined and used in this paper. The developed model takes the following phenomena into account: hydrodynamic and contact forces, spring and gas forces acting on piston rings. The rings motion concerning low and high temperature of cylinder surface has been compared. These results concern cases of hydrodynamic and mixed lubrication. Changes of oil wetted area and contact zone of piston rings have been shown. In addition the oil film thickness distribution along cylinder liner and all the forces acting on piston rings have been analysed and discussed. The results have been presented in form of relevant diagrams. The developed model and software can be utilized for optimization of piston rings design.

1. Introduction
Piston rings are important part of internal combustion engines. Commonly a set of piston rings is used to form a dynamic gas seal between the piston and cylinder wall [4,5]. The sliding motion of the piston forms a thin oil film between the ring land and cylinder wall, which lubricates the sliding components [1,3,8]. The hydrodynamic force generated by this thin oil film is opposed by a combination of the gas pressure acting on the back side of each ring and the ring stiffness. Due to the dynamic nature of these forces, each individual ring is periodically compressed and extended as the piston runs through its cycle. The problem of studying this interaction is further complicated by the high temperatures involved, as these result in low oil viscosity and subsequently very low oil film thickness. The oil film is typically thick enough to expect the existence of mixed lubrication, so this phenomenon should also be taken into account [1,2,6,7,9]. The use of modern oil of low viscosity, working at a high temperature causes the existence of a very thin oil film thickness comparable to the value of the liner surface roughness. In such conditions, the possibility of direct contact between the ring and cylinder liner surface exists. Therefore the numerical simulation of these processes, which take place in a typical piston ring pack operation, is important from practical point of view.

2. Modelling of piston ring pack operation
2.1. Developed sub-models
A combined model of piston rings operation has been developed. It consists of two main models: a) model of gas flow through the labyrinth seal piston-rings-cylinder (PRC), b) model of oil flow in
the lubrication gap between the ring and cylinder liner. The two mentioned models are coupled. In addition, sub-models of the following mechanical phenomena have been used: a contact of rough surfaces, an axial movement of rings within piston grooves and an elastic torsional deformation of piston rings (Figure 1). But this time the effect of ring twist has not been taken into account.

**Figure 1.** Developed comprehensive model of the system: piston-ring-cylinder (PRC).

All the sub-models are described in detail in publications [10,11,13-16] of the author. In this paper only the sub-model of mixed lubrication [2,6,7] is shortly presented.

### 2.2. Model of mixed lubrication

Two main cases of oil flow in the system piston ring – cylinder liner are presented in Figure 2.

**Figure 2.** Scheme of gap between the ring face and cylinder liner in the case of: a) hydrodynamic, b) mixed friction.

A one dimensional form of the modified Reynolds equation developed by Patir and Cheng [6,7] has been used to calculate hydrodynamic forces in the case of rough gap surfaces. This equation is applicable to any general roughness structure and takes the following form:

\[
\frac{\partial}{\partial x} \left( \phi \frac{h^3}{12\mu} \frac{d}{dx} p \right) = \frac{U}{2} \frac{dh_T}{dx} + \frac{U}{2} \frac{d\phi}{dx} + \frac{dh_T}{dt}
\]

where: t - time; x – coordinate along cylinder liner; h – nominal oil film thickness; h_T – average gap (ring-cylinder); p – hydrodynamic pressure; U – axial ring velocity; \(\mu\) – dynamic oil viscosity; \(v=\partial h_T/\partial t\) – radial ring velocity, \(\sigma\) – composite root mean square roughness of sliding surfaces.

The significance and mathematical description of empirical coefficients \(\phi_x, \phi_T\) and boundary conditions of equation (1) are presented in [6,7] and also in [10].
The effects of interacting asperities of piston ring and cylinder liner surfaces were modelled using the mathematical model developed by Greenwood and Tripp [2]. In this case the asperity contact force per unit circumference is given by

\[
F_A = 16 \sqrt{\frac{2}{15}} \pi (\eta \beta \sigma)^2 E' \int_{x_l}^{x_r} \left( \frac{\sigma}{\beta} \right)^{3/2} \frac{h}{\sigma} \, dx
\]

where the integration limits \( x_l \) and \( x_r \) define a continuous interval, \( x_l \leq x \leq x_r \), in which \( \frac{h}{\sigma} \leq 4 \)

\( \eta \) - asperity density; \( \beta \) - asperity radius of curvature;

\( \sigma = \sqrt{\sigma_1^2 + \sigma_2^2} \) composite roughness of sliding surfaces.

The model is also described in detail in publication [10] of the author of this article.

2.3. Viscosity as a function of oil temperature

The viscosity of the oil used for lubrication is a key factor influencing oil film thickness. Automotive engines have a relatively high temperature of oil film and due to that a relatively low oil viscosity (Figure 3). This explains the very thin oil film left by the periodically moving ring pack. In the analysis presented, it was assumed that the oil film temperature is equal to the liner temperature.

![Figure 3. Oil viscosity versus oil temperature.](image)

The variation of oil viscosity with temperature is determined by the commonly used Vogel equation:

\[
\mu_0(T) = k \exp \left( \frac{\theta_1}{\theta_2 + T} \right)
\]

where: \( k = 0.0352 \text{ cSt}\); \( \theta_1 = 1658.88 \text{ °C}\); \( \theta_2 = 163.54 \text{ °C} \) for the SA-10W50A multigrade oil and temperature \( T \text{ [°C]} \).

2.4. Experimental verification of the developed model

A verification of the simulation model has been done by the author for a two- and four-stroke marine engine [12,13,16]. The experimental verification of the model of gas flow through the labyrinth seal of piston rings was carried out using measurements of unsteady gas pressure in the cylinder, between the piston rings and under piston performed by piezoelectric sensors mounted in the piston. A satisfactory qualitative and quantitative compatibility of the analyzed pressure variations has been achieved. The
maximal relative differences between measured and calculated pressure values have not exceeded 15% [12,16]. On the other hand, the experimental verification of the hydrodynamic model of piston rings involved measurement results of scraped oil volumes by a gland-box of a two-stroke marine engine. Unfortunately, similar measurements for piston ring packs of tested engines have not been carried out. Examination of scraped oil volumes by the ring pack (of the gland-box of marine internal combustion engine) proves a satisfactory quantitative agreement between numerical and experimental results. The maximal relative differences between measured and calculated values have not exceeded 10% [12,16].

3. Computational results

3.1. Main data of chosen engine
The simulation investigations have been done for a four-stroke spark ignition engine of a middle class passenger car. The main data of the engine is presented in Table 1.

| Table 1. Main engine parameters |
|--------------------------------|
| Cylinder diameter $D_C = 80$ mm |
| Piston diameter $D_P = 79.92$ mm |
| Piston stroke $S = 79.5$ mm |
| Engine rotational speed $n = 3400$ rpm |

The type of ring set considered is common in car engines. It consists of three rings: a compression ring, a scraper ring and a two-lip oil ring. The package includes conventional straight ring end gaps. The surface geometry of the piston ring package, with vertical dimensions magnified by factor of 1000 relative to the horizontal ones, is depicted in Figure 4. All the rings are barrel shaped.

3.2. Calculation results for assumed oil temperature
An automotive internal combustion engine works where a wide range of loads and the resulting thermal conditions exist. In the paper two characteristic thermal states of engine operation have been analysed and compared. It has been assumed that the first characteristic engine load state is engine warm-up, when the cylinder liner temperature is low. The second characteristic engine state corresponds to the nominal thermal load, when the liner temperature is much higher.

Two calculations have been performed. The first one assumed that the oil temperature equals 93°C. Under this assumption, the oil film is thick enough in comparison with the mean value of the surface roughness. Therefore, the surfaces of rings and cylinder liner can be treated as smooth surfaces. The second calculation has been done under the assumption that the oil temperature equals 200°C. In this case, the possibility of mixed lubrication is expected and taken into account. The main parameters of the rough structure of the liner and ring surface are presented in table 2. An important calculation result is the oil film thickness distribution on the cylinder liner. The motion of the ring pack scraping and distributing oil on the cylinder liner leaves the oil film profile shown in Fig. 5 after a few cycles of operation. A comparison of results concerning low ($T_{oil}=93^\circ$C) and high ($T_{oil}=200^\circ$C) oil temperature is presented. An uneven oil film distribution along the cylinder liner can be clearly seen.
Table 2. Surface roughness parameters

| Surface data                              | Value               |
|-------------------------------------------|---------------------|
| Asperity density                          | $\eta = 1 \cdot 10^6 \text{ [m}^{-2}]$ |
| Asperity radius of curvature              | $\beta = 0.2 \text{ [\mu m]}$         |
| RMS roughness of cylinder liner           | $\sigma_1 = 0.22 \text{ [\mu m]}$     |
| RMS roughness of rings sliding surface    | $\sigma_2 = 0.044 \text{ [\mu m]}$    |

The low oil film thickness near the piston top dead centre (TDC) and peaks of accumulated oil near the leading ring lips should be noticed. The difference of the oil film thickness along the area of ring pack motion is very important. In the case of lower oil temperature, the minimum film thickness is higher than 1 $\mu$m in contrary to the case of higher temperature, where the oil film thickness decreases to 0.3 ÷ 0.4 $\mu$m. This value is comparable with the root mean square (RMS) roughness of the cylinder liner that equals 0.22 $\mu$m. The very low local film thickness values, in both cases, near the top dead centre can be explained by the existence of high gas pressure values existing in this area during the compression and working phases of engine operation.

Figure 5. Comparison of the oil film thickness distributions left by the ring pack along cylinder wall for the oil temperature of 93°C and 200°C.

In the following figures variations in some physical parameters as functions of the crankshaft rotation angle, beginning from the piston bottom dead centre (BDC) of the four-stroke engine operation (0°) are shown. In this case the end of compression phase is at 180° of crank angle (piston top dead centre - TDC).

The hydrodynamic friction forces as functions of the crankshaft rotation were calculated and presented in Figure 6. These forces significantly depend on piston velocity. For this reason the highest values of hydrodynamic friction forces should be noticed at crank angles, where the maximum piston velocity is reached. These forces could be neglected in the piston motion phases corresponding to low velocity near the reverse points.

In the case of a higher oil temperature (Figure 6b) the variation of hydrodynamic friction forces has a slightly different character. Changes in the amplitude of these forces, and thus the absolute extreme values are significantly lower (approximately 3 times for the compression ring).

Gas and ring elastic tensing forces must be compensated by hydrodynamic forces generated in the ring-liner gap and, additionally, in the mixed lubrication cases expected, by elastic contact forces.
In Figure 7a variations in radial components of elastic contact forces of piston rings with cylinder liner are visible. The force values are referenced to the unit of ring circumference. These forces occur in the case of a high oil temperature ($T_{oil}=200^\circ C$) near the piston top and bottom dead centre. The values of elastic contact forces are much lower than hydrodynamic forces acting on rings [10,13]. In the case of higher oil temperature and lower oil film thickness additionally tangential components of contact forces occur near piston reverse points (TDC and BDC - Figure 7b).

One important computational problem is the definition of the boundaries of the ring wetted area (see Figure 8). After a series of piston operation cycles, each ring scrapes and accumulates excessive oil, leaving behind an oil film not sufficient for full lubrication. The variation of wetted area has an essential influence on the hydrodynamic bearing force of the ring and the resulting radial ring velocity. Oil film thickness, piston velocity, ring stiffness and ring surface roughness parameters have significant influence on changes of wetted area boundaries and areas of direct contact of the ring and cylinder liner surfaces in the case of mixed lubrication. Most often rings are only partially wetted in phases of high piston velocity. Rings are fully wetted in the piston motion phases corresponding to low velocity near the reverse points. Figure 8a depicts changes of the wetted area boundaries of the compression ring for low oil temperature. In this case the full lubrication can be observed. Oil is mostly concentrated in the central part of the ring profile. In the rear part the hydrodynamic pressure drops and this area is ventilated by the gas outside the ring. In the case of very thin oil film, the wetted area is also reduced in the upstream part of the ring surface. Additionally, in Fig. 8b corresponding to the case of high oil temperature and mixed lubrication, boundaries of the contact area are shown.

Figure 6. Variation in hydrodynamic friction force $F_{hi,i}$ for each piston ring lip ($i=1,2,3,4$) versus crank angle. Results for oil temperature: a) $93^\circ C$, b) $200^\circ C$.

Figure 7. Variation in: a) radial $F_{c,i}$ and b) tangential $F_{cx,i}$ component of contact force for each piston ring lip ($i=1,2,3,4$) versus crank angle. Results for rough surfaces and oil temperature of $200^\circ C$. 
to very low oil film thickness and assumed contact limit of \( h/\sigma = 4 \) [2,6,7], an arrival of elastic contact area is noticeable. In both cases (Figures 8a and 8b) the shapes of the wetted areas look very similar. However, in the second case of high oil temperature, the wetted area is a little bit smaller.

**Figure 8.** Oil wetted area (\\\\) and contact zone (///\/) of the 1st ring (compression ring) versus crank angle. Results for the oil temperature: a) 93°C, b) 200°C.

The oil wetted area and contact zone of the 2nd (scraper) ring, but for the higher surface roughness \( \sigma_1 = \sigma_2 = 0,4 \mu m \), has been presented by the author in article [10].

The two-land oil ring shows the same pattern (Figures 9a and 9b) of wetted area distribution in both cases. It can be seen that depending on the direction of the piston motion, only the leading ring land is wetted with starvation on the trailing land. It should be noted that the ring twist effect has been neglected. The hydrodynamic lubrication case is presented in Figure 9a, and the mixed lubrication case in Figure 9b. In addition, a small area of contact in the central ring part is observed in the case of high oil temperature.

The calculations showed that at higher oil temperatures, its viscosity is so low that it causes the movement of the ring in the distance of the cylinder liner compared to the size of medium asperities characterizing this surface. In this case, the model of the oil flow in the lubricating gap described by the Reynolds equation is replaced by the model developed by Patir and Cheng [6,7] (equation (4)). In this case, there are also mechanical interactions between the mating rough surfaces. It is therefore necessary to take these forces into account, which allows the model of Greenwood and Tripp [2].

**Figure 9.** Oil wetted area (\\\\) and contact zone (///\/) of the 3rd ring (oil ring with 2 lands) versus crank angle. Results for the oil temperature: a) 93°C, b) 200°C.
3.3. Calculation results for the chosen surface roughness of cylinder liner

Simulation investigations were carried out for four selected surfaces of cylinder liner characterized by the following root mean square (RMS) roughness: \( \sigma_1 = 0.1 \, \mu m, \sigma_1 = 0.22 \, \mu m, \sigma_1 = 0.3 \, \mu m \) and \( \sigma_1 = 0.4 \, \mu m \). Every time a constant value of \( \sigma_2 = 0.044 \, \mu m \) was assumed for piston rings. In addition, for comparison calculations concerning perfectly smooth surfaces of cylinder liner and piston rings (\( \sigma_1 = 0 \, \mu m \) and \( \sigma_2 = 0 \, \mu m \)) were performed.

Due to the presence of relatively thin oil layers the contact between the rough surface of the cylinder and piston rings occurred (mixed friction). It was important to obtain a response of the models of Patir and Cheng [6,7] and Greenwood and Tripp [2] to the change of the surface roughness. However, the main objective of the study was to evaluate the influence of the surface roughness of cylinder liner on friction losses. The computational results are shown in Figs. 10 and 11. The first one depicts the total power loss (due to hydrodynamic and mixed lubrication) versus crank angle for all of the aforementioned surface roughness of the cylinder liner.

In turn, Fig. 11 shows average values of previously presented friction power losses associated with the selected values of surface roughness. Generally it can be concluded that the smallest friction power loss occurs in the case of perfectly smooth surface. The higher RMS of cylinder surface roughness is applied the more friction power loss is detected.

![Figure 10. Total power loss (due to hydrodynamic and mixed lubrication) versus crank angle. Results for the chosen RMS surface roughness of cylinder liner: \( \sigma_1 = 0 \, \mu m \), \( \sigma_1 = 0.1 \, \mu m \), \( \sigma_1 = 0.22 \, \mu m \), \( \sigma_1 = 0.3 \, \mu m \), \( \sigma_1 = 0.4 \, \mu m \).](image-url)

A significant increase of friction power loss at the end of the compression stroke and at the beginning of expansion can be seen (Figure 10), i.e. when a high gas pressure in cylinder exists and an elastic contact of piston rings with surface roughness of cylinder liner occurs. In this range of operating conditions an increase of wear intensity of the cylinder liner can be predicted. It is noteworthy that at piston reverse points the axial velocity of piston rings decreases to zero, which means the simultaneous drop of friction power loss to zero value.

Although the model of Patir and Cheng [6,7] and Greenwood and Tripp [2] allows to perform a number of interesting analysis, it is difficult to directly choose the optimum surface roughness. For this purpose it would be necessary to examine more tribological factors such as the microstructure of surface treatment, etc. It would be also good to collect an extensive experimental data.
4. Conclusions

The major conclusions that may be drawn from the results are as follows:

1. The developed mathematical model and simulation programme give a lot of practical information that would be more complicated and expensive to obtain using experimental methods;
2. Elastic radial contact forces due to surface roughness occur in the area of piston TDC and BDC (Figure 7). These forces are relevant to high gas pressure and low oil viscosity caused by high temperature. However, the values of elastic contact forces are much lower than hydrodynamic forces acting on piston rings [10,13];
3. The hydrodynamic forces are generated by relatively low pressure acting on a large surface in contrast to high local contact pressure concentrated on a very small area of elastic contact. Due to that fact, the elastic contact seems to be responsible for wear process despite the low speed of the piston close to TDC and BDC;
4. The most severe lubrication conditions occur close to TDC towards the end of the compression and beginning of expansion stroke, especially for the first (compression) ring (Figure 7). Due to high temperature, the oil viscosity is very low and, consequently, the oil film along the cylinder liner is very thin: 0.3 ÷ 0.4 μm at TDC (Fig. 5). In this case it is essential that a model of mixed lubrication accounting for the surface roughness is used (RMS = 0.22 μm for the cylinder liner of analyzed engine);
5. The higher RMS of cylinder surface roughness is applied the more friction power loss is detected (Figure 11);
6. Although the model of Patir and Cheng [6,7] and Greenwood and Tripp [2] allows to perform a number of analysis concerning mixed lubrication, it would be difficult to choose the optimum surface roughness. For this purpose it would be necessary to examine more tribological factors such as the microstructure of surface treatment, etc. It should be also good to collect an extensive experimental data.

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