A NUMERICAL MODEL FOR CALCULATION OF PISTON RINGS WETTED AREA IN A COMBUSTION ENGINE

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Abstract

Assumptions made while modelling the oil film between the ring and the cylinder liner in IC engines, especially concerning the boundaries of the wetted area of the ring face surface, were discussed and the assumptions adopted for the developed model were presented in this study. It was assumed that the model should take into account partially flooded lubrication, which meant that the boundaries of the ring wetted area had to be determined. Based on the adopted assumptions, a model for calculation of the oil film thickness between the cylinder and moving rings, and thickness of the oil film left on the cylinder by the ring pack was developed. A computer application operating in the Windows operation system was developed to carry out numerical calculations. The results of initial numerical calculations were also presented. The proposed model can be utilized to determine the effects of the ring pack geometry, especially the geometry of the ring surface, on parameters of lubrication. These parameters, including the oil film thickness, distribution of pressure in oil film or tangential force, are crucial for friction and wear of cooperating surfaces, oil consumption, and flow of gas from the combustion chamber to the crankcase, thus playing a role in the durability, fuel consumption and emission of an engine.

Keywords: *combustion engine, ring pack, lubrication, simulation, wear, mechanical losses*

1. Introduction

Development of any model requires certain simplifications to be made in the description of the phenomenon being investigated. It can be supposed that the more detailed a mathematical description is, the better it reproduces the phenomenon being modelled and the more precise are the results that it yields. However, the simplifications made to the main assumptions of a model may be so significant that a further increase in the detail of the description of the accompanying phenomena will fail to lead to the expected improvement in the model's accuracy. The intuitive approach taken by researchers who model phenomena that are particularly difficult to verify empirically translates into the multitude of proposed solutions.

In modelling the oil film formed during the interaction between the piston ring pack and the cylinder liner, of key importance is the way in which the geometry of the wetted area of ring faces is determined. The location of wetting boundaries x_a and x_b (Fig. 1a) strongly affects the distribution of pressure in a ring face and thereby the thickness of the lubricant wedge h_{min}.

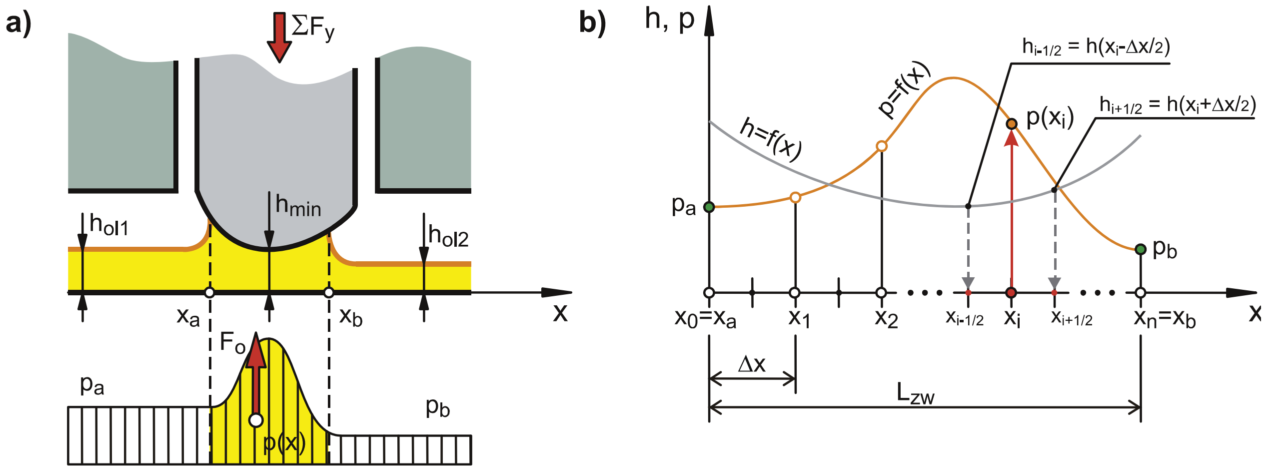

Fig. 1. Boundaries of ring face wetted area (a) and differential equation for determining oil pressure distribution (b)

Pressure distribution (Fig. 1b) in the oil film along the length L_{zw} of the wetted area of a ring face is commonly determined on the basis of a modified Reynolds equation (1) and approximated by a central difference operator [1, 2, 7, 8]:

$$\frac{\partial}{\partial x}\left(h^3 \frac{\partial p}{\partial x}\right) = 6\mu u \frac{\partial h}{\partial x} + 12\mu \frac{\partial h}{\partial t}, \quad \begin{matrix} p(x_a) = p_a, \\ p(x_b) = p_b, \end{matrix} \tag{1}$$

where:

p – oil pressure in the lubricant wedge,
h – height of the flow gap defined by the ring face profile,
u – oil flow rate resulting from the motion of the ring relative to the liner,
μ – oil dynamic viscosity.

The boundary value problem is solved within the boundaries of the ring wetted area x_a and x_b, whose modelling is so difficult that some researchers use simplifications burdened with a large error. The largest simplification adopted is the assumption that the ring face is entirely soaked with oil during a complete working cycle of the engine [1, 6]. This assumption is compared with a solution, which assumes fully flooded lubrication only on the oil inlet side relative to the moving ring face profile, while the wetted area boundary on the opposite side of the profile is assumed to lie at the point where oil pressure drops below saturated steam pressure [1, 6]. There are also those solutions in which the boundary on the outlet side is adopted at the point where the ring face is closest to the surface of the liner [2, 7]. Some authors assume that the thickness of the oil film covering the cylinder liner is constant, whereas the boundaries of the wetted area result from the geometric intersection of the ring profile with the surface of this layer [3]. The solutions cited above are open to doubt as they ignore the mass balance of the oil flowing through the ring pack and omit the distribution of oil along the liner. One of the consequences of this type of simplification would be a high oil consumption diverging from reality. To increase the accuracy of a model, it is necessary to determine more precisely the boundaries of the ring wetted area based on the balance of streams flowing and volumes of oil moving under the ring face.

2. Preliminary assumptions of the model

The model assumes that the piston, rings, and the cylinder liner have an axial symmetry. To facilitate the interpretation of diagrams and the analysis of results, the main axis of the geometric layout was assumed to be horizontal. The ring pack consists of two sealing rings and a double-lip oil control ring. The cylinder liner is coated with an oil film, the thickness of which results from the operation of the ring pack. Oil viscosity depends on oil temperature, which in turn, in accordance with the adopted assumption, is the same as the temperature of the surface of the cylinder liner along its height.

To determine oil film geometry, three types of accumulative space were distinguished (Fig. 2):
– the gap between the ring face and the cylinder liner,
– the space between rings,
– the space outside the ring pack – above or below the piston.

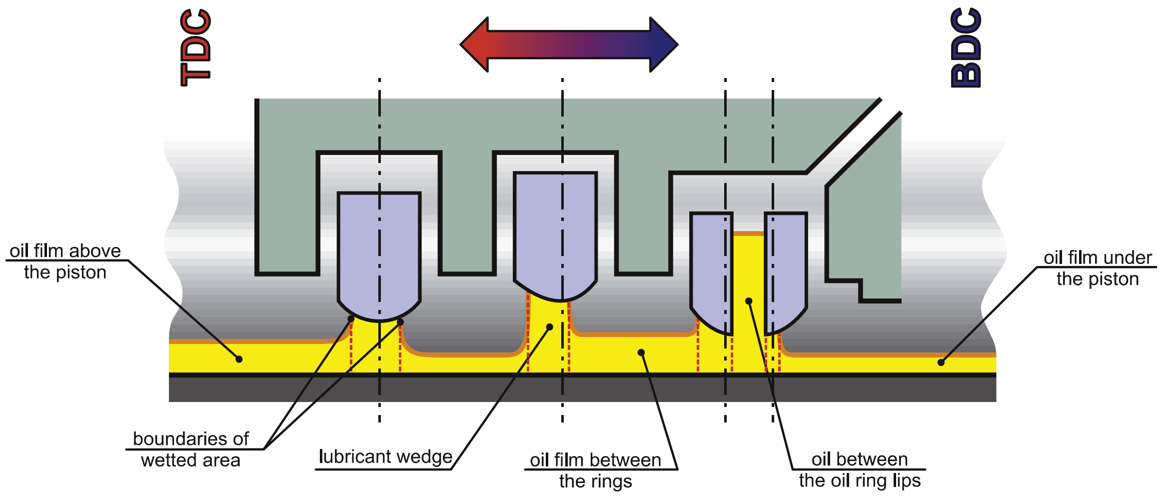

Fig. 2. Geometry of the oil film formed during the interaction between the piston ring pack and the cylinder liner

The boundaries of the wetted area of each ring face are variable and depend, among others, on the squeezing of oil from under a ring face as a result of radial displacement, the scraping of oil by the moving rings and the accumulation of oil in the spaces between rings.

3. Squeezing out and accumulation of oil under the ring face

A change in the height of the clearance between ring face and cylinder liner surface, associated with radial displacements of the ring, causes a change in the boundaries of the ring wetted area (Fig. 3a) [8]. At full flooding of the profile, a certain volume of the oil accumulated under the surface of the ring face is squeezed out and transported to an appropriate external space.

To simplify mathematical description, the ring face profile was approximated with a parabola and divided into two parts relative to the axis of this parabola. Further discussion concerns only one part of the profile, whose shape is described by the equation:

$$h(x) = \frac{x^2}{2R} + h_{min}, \quad x \in [0; x_z], \tag{2}$$

where:
h_{min} – minimum distance of the ring face from the cylinder liner,
R – radius of curvature of the ring face profile,
x – axial coordinate,
xz – coordinate of the edge of the discussed part of the profile:

$$x_z = x_{h\_min} \text{ or } x_z = C - x_{h\_min}, \tag{3}$$

where:

x_{h_min} – a coordinate of the point on the ring face profile corresponding to the minimum distance from the liner,

C – ring face width.

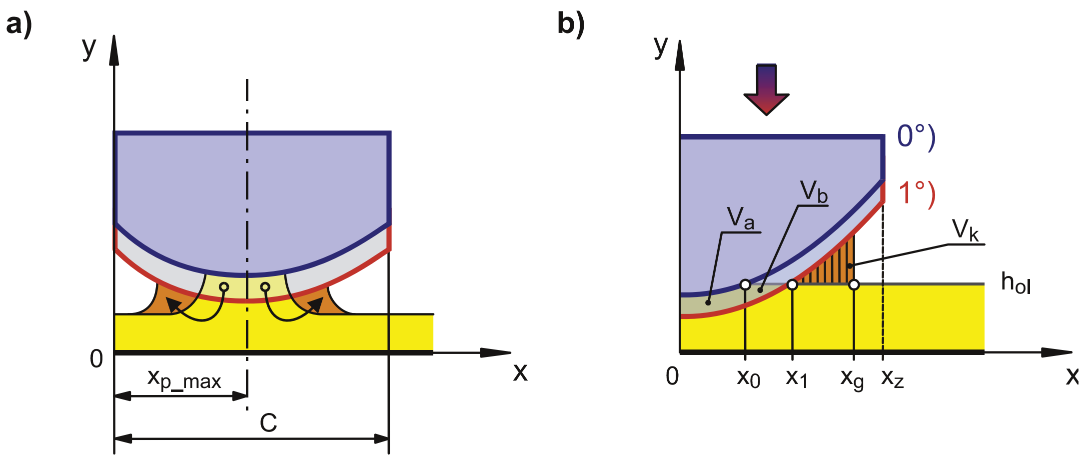

Fig. 3. Change of wetting boundaries (a) and accumulation of oil under the ring face (b)

A ring face wetting boundary (wetting location) x_g (Fig. 3b) is calculated on the basis of the volume of oil V_k accumulated in the space between the ring face and the oil film of thickness h_{ol} adhering to the cylinder liner. In a time interval between the time points 0 and 1, the accumulated volume Vk can increase, in accordance with the equation:

$$V_{k\_1} = V_{k\_0} + \Delta V, \; \Delta V = V_a + V_b, \tag{4}$$

where:

V_a and V_b – components of the volume of oil being squeezed out.

Coordinates x_0 and x_1 of the points of intersection of the ring face with the oil level line, corresponding to the previous time point 0 and the current time point 1, are specified by the equations:

$$x_0 = \sqrt{2R(h_{ol} - h_{min\_0})}, \; x_1 = \sqrt{2R(h_{ol} - h_{min\_1})}, \tag{5}$$

where:

h_{ol} – thickness of oil film adhering to the liner.

The components Va and V_b of the volume of oil squeezed out result from the respective geometric relationships:

$$V_a = (h_{min\_0} - h_{min\_1}) \cdot x_0, \tag{6}$$

$$V_b = h_{ol}(x_1 - x_0) - \int_{x0}^{x1} \left(\frac{x^2}{2R} + h_{min} \right) dx = (h_{ol} - h_{min})(x_1 - x_0) - \frac{x_1^3 - x_0^3}{6R}, \tag{7}$$

where:

h_{min} – stands for the smaller of the heights h_{min_0} or h_{min_1}.

If at time point 1, there exists point x_1 at which the ring profile and the oil level line intersect, then the wetting location x_g is determined relative to this point, according to the equation:

$$V_k = \int_{x1}^{xg} \left(\frac{x^2}{2R} \right) dx - h_{ol}(x_g - x_1) \quad \Rightarrow \quad x_g. \tag{8}$$

In a contrasting situation, when the ring profile at time point 1 is located above the oil level, the wetting location is determined relative to the coordinate x_{p_max}, at which oil film pressure reaches its maximum value:

$$V_k = \int_{xp\_max}^{xg} \left(\frac{x^2}{2R} \right) dx - h_{ol} (x_g - x_{p\_max}) \quad \Rightarrow \quad x_g , \tag{9}$$

where:

x_{p_max} – a coordinate of the maximum oil pressure point in the lubricant wedge.

The coordinates of wetting locations x_g of both parts of the profile are converted into coordinates x_a and x_b by performing a transformation, which consists in shifting the coordinate system by a distance equal to x_{h_min}.

4. Flow and accumulation of oil in the ring pack

The balance of oil flowing under the ring face is based on three streams: inlet stream q_d, stream q_p flowing through the cross-section at the point of maximum oil pressure, and outlet stream q_w. The flow streams per unit of ring circumference are determined from the following formulas:

$$q_d = h_d \cdot u \, , \; q_p = \frac{1}{2} h_p \cdot u \, , \; q_w = h_w \cdot u \, , \tag{10}$$

where:

h_d – thickness of oil film upstream of the ring face,
h_p – height of the clearance under the ring face at the point of maximum oil pressure,
h_w – thickness of the oil layer flowing out from under the ring face.

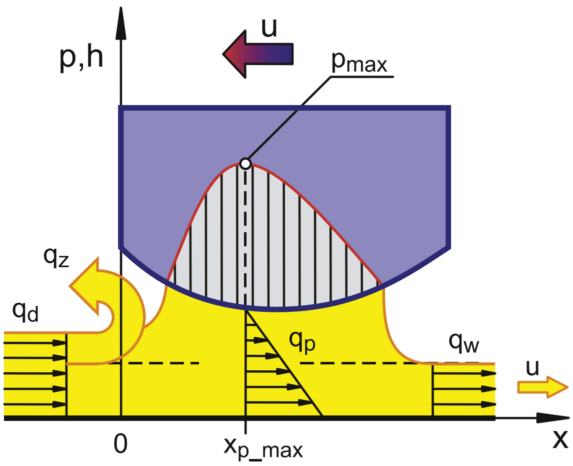

Fig. 4. Streams of oil flowing under the ring face

Under partially flooded conditions, the ring may only slide on the oil film, in which case the inlet stream q_d and the outlet stream q_w are equal. If the inlet stream q_d is larger than stream q_p, the excess oil is scraped. Then, the flow q_z of the oil scraped by the ring follows from the difference between those two streams.

Oil is accumulated in the spaces between rings. Changes in the oil volume in this type of space are a result of the difference in the amounts of oil flowing through the neighbouring rings. In the present study, also the volume of oil that may be squeezed from under the ring face into an appropriate space as a result of radial displacement was taken into account. The thickness of the oil film filling a given space is determined on the basis of a flow balance for this space determined in the way described above.

5. Computer program

The computer program used was written in its entirety in the C++ programming language using the object-oriented technique. This provided broad possibilities of modifying and extending the numerical model. The program had a module structure in which the model of oil film formation during interaction between the ring pack and the cylinder liner represented only one of the integrated parts. Inputting of the discussed model required specification of such parameters as the resultant of the radial forces loading the individual rings and the pressures in the spaces between rings. The current values of these parameters were calculated in the basic module, which simulated the action of the piston-rings-cylinder assembly and ring dynamics [4, 5]. The model of the oil film, in turn, allowed calculation of lubricant wedge heights, boundaries of the ring wetted area, and the oil film thickness in the spaces between rings and outside of the ring pack. Also calculated were the friction force and the pressure field coordinate in the lubricant wedge.

6. Results of simulations

The diagrams shown in Fig. 5–7 represent ring face wetting history for the individual rings at successive strokes of one engine cycle. It was assumed that the range of the vertical axis corresponded to the profile width, while the relevant values were measured relative to the ring face axis.

The wetted area was situated between the upper and lower wetting boundaries. In places where these boundaries contacted the ring edge, oil scraping occurred. The model of the wetting of the top ring (Fig. 5) showed that oil was not scraped in either direction. This type of situation is beneficial as it involves reduced oil consumption, but it may be a result of the fact that considerable ring face rounding was assumed for this ring.

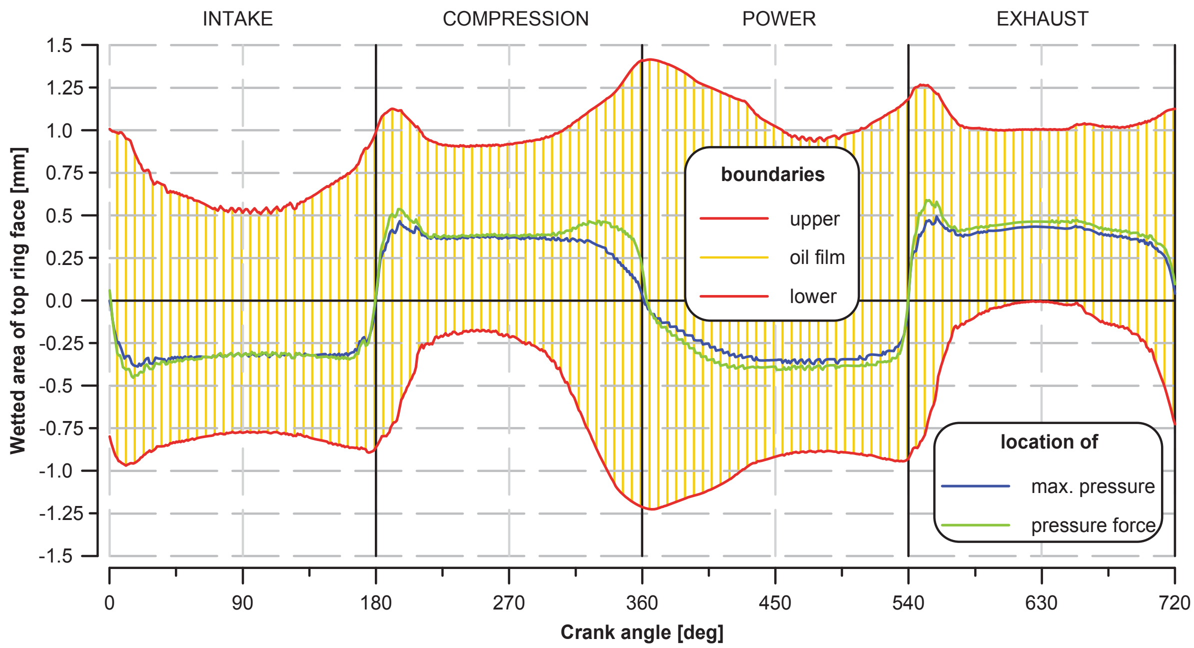

Fig. 5. Plot of top ring wetting

The asymmetric shape of the profile of the second ring (Fig. 6) led to the scraping of oil into the space dividing it from the third ring over a crank angle range of 360–540°. This phenomenon was also likely to occur within the crank angle range of 0–180° if the ring face axis was shifted further downwards. Full soaking of the lower part of this ring over the crank angle range of 540-585° was caused by a periodic increase in the oil film thickness in the second inter-ring space.

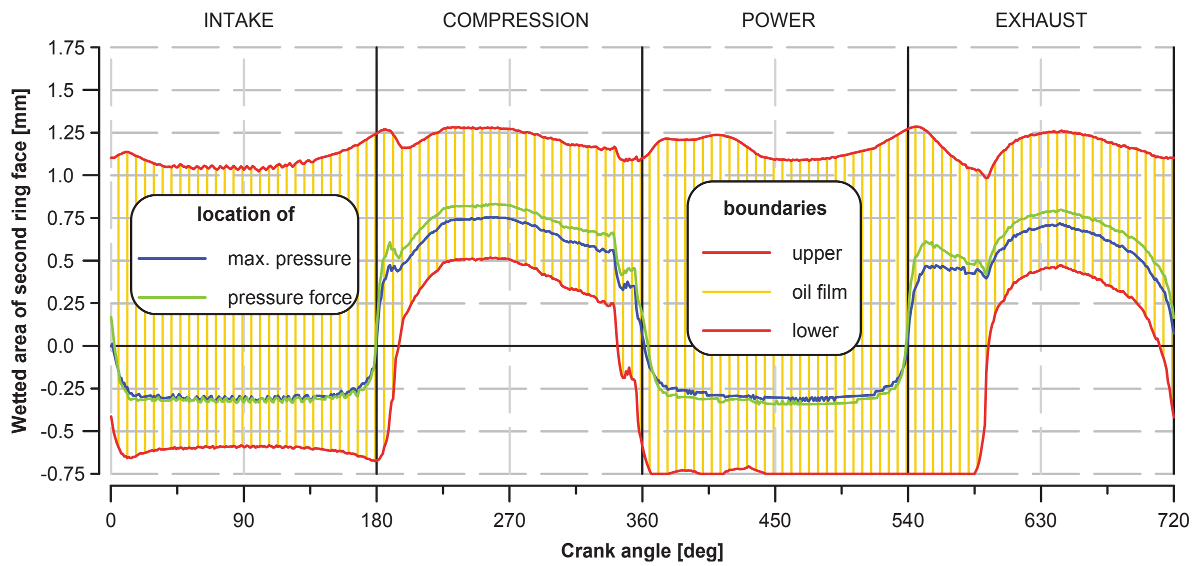

Fig. 6. Plot of second ring wetting

The lips of the oil control ring (Fig. 7) were fully soaked on the inner side, which was a consequence of the fact that excess oil was removed through the clearance between the lips. The lower lip of this ring scraped oil also during those strokes in which the piston moved towards the crankcase. The above observations may show that the oil control ring worked correctly. The upper lip was fully wetted over the crank angle range of 410–585°, which, similarly as in the case of the second ring, was connected with the increase in the oil film thickness in the second inter-ring space.

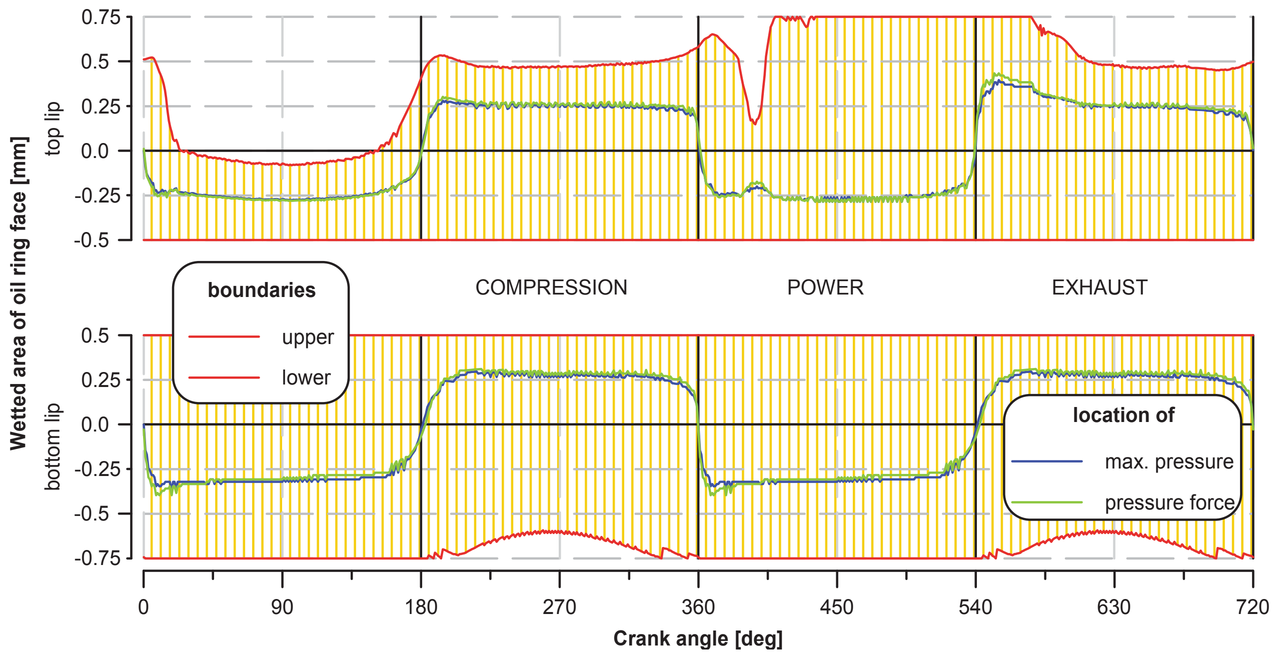

Fig. 7. Plot of third ring wetting

The diagrams additionally show s plot for the maximum pressure coordinate, which, in accordance with the assumptions of the model, is of essential significance to the volume distribution of the oil accumulated in the lubricant wedge, and the plot of the oil pressure force coordinate, which determines the length of the moment arm of this force. Worth noticing is the cyclic nature and similarity of the plots for those coordinates.

7. Conclusions

The literature mentions various ways of determining the wetted area of a ring face, which, however, do not always make it possible to increase the level of detail of a model. Cyclic displacements of the ring wetted area lead to changes in radial force loading on the ring, thereby affecting the calculated thickness of the lubricant wedge and the length of the moment arm of the pressure force in the lubricant wedge.

The model presented here allows determination of the boundaries of the ring wetted area, taking into accounting the balance of oil flowing through the ring pack. Therefore, the model can be used to calculate engine oil consumption and oil distribution along the cylinder liner.

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