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MODELLING THE BLOWBY IN INTERNAL COMBUSTION ENGINE. PART 1: A MATHEMATICAL MODEL

The paper describes a mathematical model of gas flow through crevices between the piston, rings and cylinder in internal combustion engines. The model has been developed to study the influence of numerous design parameters and engine operation condition on piston ring pack behaviour and the exhaust gases blowby from the combustion chamber to the crankcase. The model integrates the gas flow and axial rings displacements in the grooves, and separately treats lands between piston, cylinder and adjacent rings and regions in the grooves behind the rings. The heat transfer between the gas and the surrounding surfaces is calculated and emphasis is placed on considering the influence of wear and thermal deformations of elements on ring pack performance.

1. Introduction

Piston with rings in the cylinder liner (PRC assembly) creates moveable, labyrinth seal of the engine combustion chamber, and as every labyrinth seal, does not guarantee full tightness. Losses of the charge resulting from the leakage are unfavourable for the engine, as they reduce engine performance and efficiency, increase toxicity of exhaust gases and reduce durability and reliability of the engine (e.g. by accelerated degradation of engine oil and reduction of start-up abilities of diesel engines).

Effectiveness of the PRC packing is influenced by a number of both design and operational factors. Rating the influence of individual factors including wear of PRC components only on the experimental basis is ineffective, mainly because of the dynamic nature of the ring pack operation and non-linear correlations between individual factors.

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In connection with the above, it was useful to develop a mathematical model of the piston ring pack, which would let to evaluate influence of design parameters and the wear of particular components on the tightness, in different engine operating conditions.

2. Review of existing models and foredesign of the developed model

In the first approximation, the PRC packing can be treated as a series of regions linked with each other by throttling crevices created by ring gaps (Fig. 1a). In the case of internal combustion engine, pressure in the sealed space (combustion chamber) changes to a large extent and with high rate. This produces an unsteady flow through the labyrinth, which intensity depends not only on the areas of crevice cross-sections but on the volume of individual stages, as well. Pressures in individual stages of the labyrinth are not constant, and periods can occur when pressures in these stages are higher than in the sealed chamber. This results in the change of gas flow direction. The scheme described above (Fig. 1a) was used in many mathematical models of the ring pack [1], [5], [6], [8], [10].

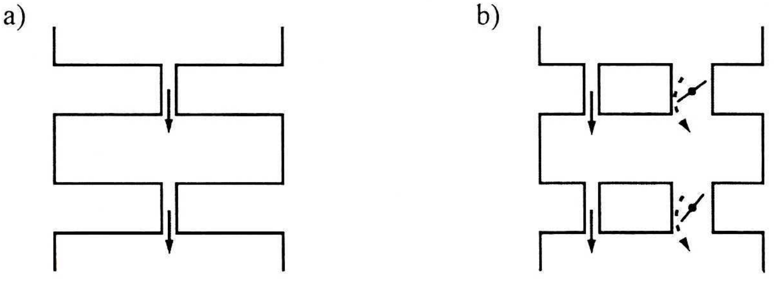


Fig. 1. Schematic diagram of labyrinth seal, considering gas flow only through the ring gaps (a) and gas flow through the ring gaps and through the grooves around the rings – the areas of the gas flow through the grooves are determined by axial position of the rings in the groove, but cumulative action of volumes behind the rings is not considered (b)

Piston ring should always adhere to the cylinder liner, even when its diameter along the generatrix is not constant and piston slaps in the cylinder. This is secured by clearance fit of the ring in the piston groove, which facilitates free radial movements of the ring. However, the clearance fit also allows for axial movements of the ring in the groove, and creates crevices between side surfaces of the ring and the groove, where gas flows. Maximum value of these crevices area can more than by order of magnitude exceed value of ring gap area [8].

Experimental research indicate a connection between ring axial movements in the groove and pressure changes in the adjacent inter-ring regions [3]. Some models [7], [12] include gas flow through the groove around the ring, during the ring displacement. However, assumed scheme of seal (Fig. 1b) does not permit modelling of dynamic effects related to cumulative action of volumes behind the rings, and moreover, because of short opening time of additional passages, considering them brings a little into the calculated blowby rate [7]. Axial position of the ring in the groove determines with which inter-ring region the behind-ring region is linked to. As the volume of the behind-ring region can be greater than the volume of inter-ring region, its cumulative action can have significant influence on the pressure courses in the inter-ring regions, and consistently, on the operation of the whole seal. So as to consider the above effects, models [3], [4], [9] separately treat inter- and behind-ring regions and fully integrate gas flow model with the model of axial movements of the rings in the piston grooves.

Models [3], [4], [9] assume isothermal flow (temperature of the gas in an individual region is constant) or quasi-isothermal (temperature of the gas is a function of piston position in the cylinder). Such assumption remarkably simplifies calculations (so as to calculate parameters of the gas in the region, the mass balance equation and the gas state equation are sufficient). However, real gas flow has intermediate character between adiabatic and isothermal, and according to some of the sources, e.g. [8], is much closer to the first. Conditions of heat transfer between the gas and the surrounding walls can change during long-term engine operation, for example as a result of deposits formation in the ring part of the piston, what can significantly influence operation of the ring pack. According to the above mentioned facts, it was decided that this effect should be included in the model, and isothermal flow is not assumed.

In consequence of inaccuracy of production and thermal and mechanical deformations, shapes of PRC components differ from those assumed during design and undergo current changes. The changes can be visible during long-term engine operation as a result of wear and accumulation of deposits. The changes are related to engine operating conditions (load, rotational speed, coolant and oil temperatures). There are also the changes which take place during single engine cycle, and which are connected with piston and rings movements relative to deformed (non-cylindrical) cylinder liner. The factors mentioned above cause constant changes of the ring gap areas and the inter-ring region volumes.

Taking into account planned application of the model and including the above analysis, it was assumed that the model should:

- integrate models of the gas flow and the axial movements of the rings in the grooves,
- consider independently the inter-ring and behind-ring regions (allow for flows between lateral surfaces of the ring and groove and for cumulative action of the behind-ring region),
- consider conditions of heat transfer between the gas in the labyrinth stages and the surrounding walls,

– take into account changes of the crevice cross-section areas where the gas flows and changes of the labyrinth stages volumes caused by deformations and wear of components,
– consider sealing effect of the oil ring (this ring is often passed by in models; results of experiments show, that pressure above this ring, in certain phases of the engine cycle, can differ significantly from the crankcase pressure).

3. Physical model

The model consists of a series of regions linked together by orifices (Fig. 2). Volumes V_2 and V_4 correspond to groove volumes behind rings, whereas volumes V_3 i V_5 correspond to volumes of inter-ring lands. Orifices with cross-section areas $A_{1,3}$, $A_{3,5}$ and $A_{5,7}$ correspond to the ring end-gaps, while orifices with cross-sections $A_{1,2}$ and $A_{2,3}$, $A_{3,4}$ and $A_{4,5}$ and $A_{5,6}$ correspond to ring-side crevices.

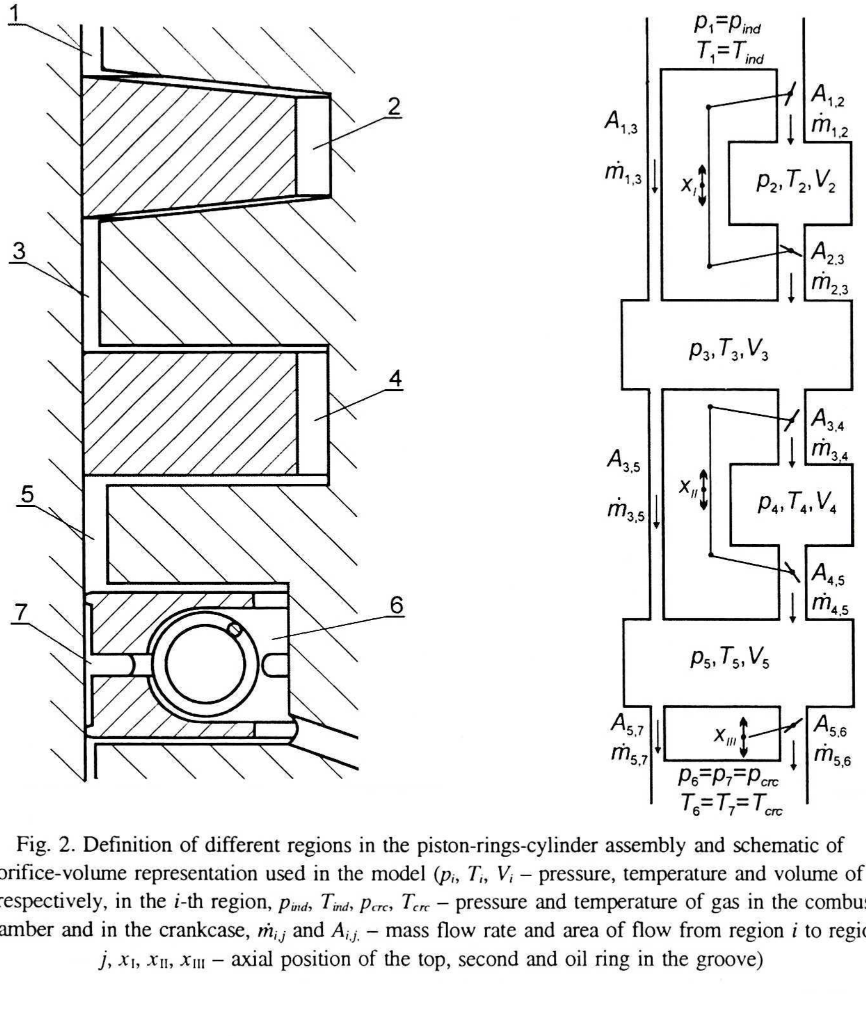


Fig. 2. Definition of different regions in the piston-rings-cylinder assembly and schematic of orifice-volume representation used in the model (p_i, T_i, V_i – pressure, temperature and volume of gas, respectively, in the i-th region, p_{ind}, T_{ind}, p_{crc}, T_{crc} – pressure and temperature of gas in the combustion chamber and in the crankcase, $\dot{m}_{i,j}$ and $A_{i,j}$ – mass flow rate and area of flow from region i to region j, $x_{\rm I}$, $x_{\rm II}$, $x_{\rm III}$ – axial position of the top, second and oil ring in the groove)

It was assumed that semi-ideal gas flows through the labyrinth, which internal energy u and specific heats c_v and c_p depend only on temperature. It was also assumed that the gas flow through orifices is isentropic, while heat transfer between gas and surrounding walls takes place in labyrinth stages. Moreover, kinetic energy of the gas in the stage was excluded; it was assumed that the energy of the gas incoming to the stage is completely exchanged to the internal energy.

The geometrical assumption was that the piston with rings are always centred in the bore and the piston rings always adhere to the liner (there are no leaks between ring and cylinder). Thermal expansion and wear of the PRC components were included, but it was assumed that temperatures and dimensions of the components (in steady-state conditions of engine operation) do not change during whole engine cycle. It was also assumed that the rings and groove plates are perfectly rigid (dynamic deformations were excluded), but the rings can move within the grooves. It means that all volumes of the regions and cross-section areas of the ring-end gaps depend on the crankshaft angular position, while cross-section areas of the ring-side crevices result from instantaneous positions of the rings in the grooves.

4. Mathematical model

4.1. Parameters of the gas in single stage of the labyrinth

Change of internal energy of the gas $\dot{U}$ in the region results from total enthalpy transferred into and out of the region by streams of the gas, heat exchange with the surroundings and work of the control volume change (Fig. 3a), what can be written as follows:

$$\dot{U} = \sum_i i_{di}\dot{m}_{di} - \sum_j i_{wj}\dot{m}_{wj} + \dot{Q} - p\dot{V}, \tag{1}$$

where:
i denotes specific total enthalpy, index d stands for flowing in (incoming), index w stands for flowing out (outgoing), while no index means that the quantity is related to the parameter of the gas in the region.

Considering mass balance:

$$\dot{m} = \sum_i \dot{m}_{di} - \sum_j \dot{m}_{wj} \tag{2}$$

and considering assumption of excluding kinetic energy of the gas in the region (than: $\dot{U} = u\dot{m} + m\dot{u}$), and assumption that the gas is semi-ideal ($\dot{u} = c_v\dot{T}$), the temperature change in the region can be described by the following equation:

$$\dot{T} = \frac{1}{c_v m}\left(\sum_i (i_{di} - u)\dot{m}_{di} - RT\sum_j \dot{m}_{wj} + \dot{Q} - p\dot{V}\right), \tag{3}$$

where R is the gas constant.

From the gas state equation in differential form:

$$\dot{p} = p\left(\frac{\dot{m}}{m} + \frac{\dot{T}}{T} - \frac{\dot{V}}{V}\right), \tag{4}$$

after substituting dependences (2) and (3), we get the pressure change of the gas in the region given by the equation:

$$\dot{p} = \frac{p}{mc_vT}\left(\sum_i i_{di}\dot{m}_{di} - (RT + c_vT)\sum_j \dot{m}_{wj} + \dot{Q} - \left(1 + \frac{c_v}{R}\right)p\dot{V}\right). \tag{5}$$

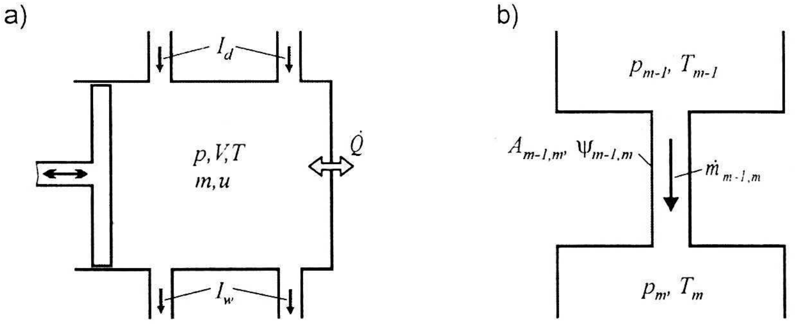


Fig. 3. Schematics of a single labyrinth stage (a) and an orifice linking two regions (b) (I_d and I_w – enthalpy of the gas incoming and outgoing in the region, respectively, $\dot{Q}$ – heat transfer rate over the boundary of a control volume, ψ – flow coefficient)

4.2. Gas flow in orifices

The flow through passages is modelled as isentropic expansion of compressible gas. Subcritical and critical flows are considered, and mass flow rate calculated in this manner is corrected using empirical flow coefficient ψ.

In the case of subcritical flow, taking place when the following condition is satisfied:

$$\frac{p_m}{p_{m-1}} > \left(\frac{2}{\kappa+1}\right)^{\frac{\kappa}{\kappa-1}}, \tag{6}$$

mass flow rate of gas $\dot{m}$ incoming to the region m from region m-1 (Fig. 3b) is calculated from the formula:

$$\dot{m}_{m-1,m} = \psi_{m-1,m} A_{m-1,m} \frac{p_{m-1}}{RT_{m-1}} \left(\frac{p_m}{p_{m-1}}\right)^{\frac{1}{\kappa}} \sqrt{2c_p T_{m-1}\left[1 - \left(\frac{p_m}{p_{m-1}}\right)^{\frac{\kappa-1}{\kappa}}\right]}, \tag{7}$$

where κ is specific heat ratio (c_p / c_v).

In the case of critical flow, which takes place when the inequality (6) is not satisfied, mass flow rate of gas incoming to the region is calculated from the equation:

$$\dot{m}_{m-1,m} = \psi_{m-1,m} A_{m-1,m} p_{m-1} \sqrt{\frac{\kappa}{RT_{m-1}}\left(\frac{2}{\kappa+1}\right)^{\frac{\kappa+1}{\kappa-1}}}, \tag{8}$$

Flow coefficients for ring-end gaps are calculated from an empirical equation, which makes allowance for ratio of pressures upstream and downstream of the orifice [9]:

$$\psi = 0{,}85 - 0{,}25\left(\frac{p_m}{p_{m-1}}\right)^2. \tag{9}$$

Flow coefficients for the crevices between rings and their grooves, whose geometric shapes change in a very wide range (crevice has constant length and width, but its height varies from zero to the value equal to the axial clearance of the ring in the groove), are calculated from the dependence which takes into account geometric shape of the crevice, ratio of upstream and downstream pressures and Reynolds number Re [2]:

$$\psi = 10^{-(10^{Y})}, \tag{10}$$

where:

$$Y = 0{,}0284X^2 - 0{,}459X - 0{,}1375, \tag{10a}$$

$$X = \log\left[2\left(\frac{h}{g}\right)\mathrm{Re}_k\left(1 - \frac{p_m}{p_{m-1}}\right)\right], \tag{10b}$$

$$\mathrm{Re}_k = \frac{p_{m-1}h}{\mu}\sqrt{\frac{\kappa}{RT_{m-1}}\left(\frac{2}{\kappa+1}\right)^{\frac{\kappa+1}{2(\kappa-1)}}}, \tag{10c}$$

μ – dynamic viscosity of the gas, h – crevice height, g – crevice length.

4.3. Crevice cross-section areas

The model assumes that temperatures in particular points of the piston, rings and cylinder liner are constant at given engine operating conditions. However, because of the changes of the cylinder diameter depending on the distance from the cylinder head (as a result of thermal loads and wear), flow areas through ring-end gaps depend on the position of the piston with the rings in the cylinder (crankshaft angel). Methodology of determining the area of flow through a ring-end gap is presented in Fig. 4.

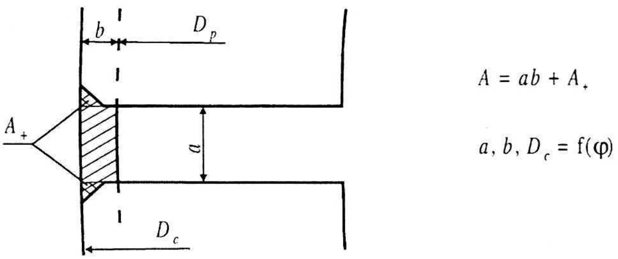


Fig. 4. A scheme to calculate the area of a ring-end gap (φ – crankshaft angle)

Cross-section area of the crevices between a ring and its groove results from instantaneous axial position of the ring in the groove x. A scheme of the area calculation is presented in Fig. 5. It was assumed that the side surfaces of the ring and groove are parallel.

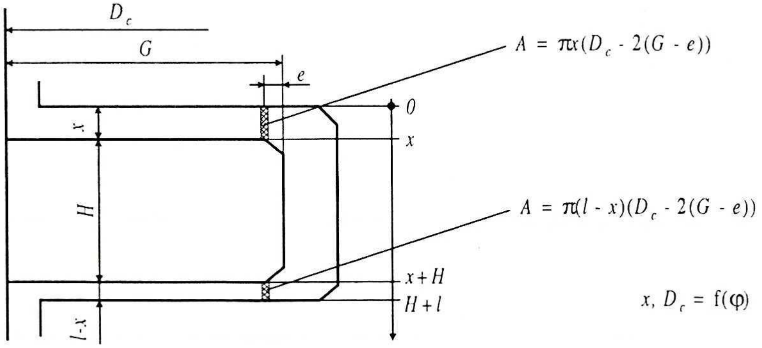


Fig. 5. A scheme to calculate the cross-section area of side-ring crevices

In the case of keystone compression ring, contrary to the plain compression ring, axial clearance l depends on the diameter of the cylinder liner (Fig. 6). Therefore, in the model the clearance value is calculated as a function of the crankshaft angular position φ.

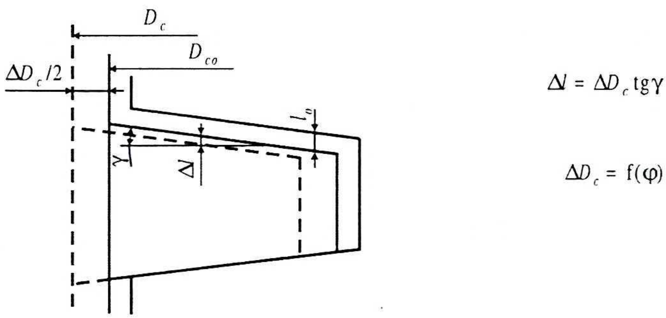


Fig. 6. Effect of the cylinder diameter on the axial clearance of a keystone ring

4.4. Heat transfer

In the case of gas inside a behind-ring region, heat transfer rate between the gas and surrounding walls is a sum of heat flux between the gas and the piston and between the gas and the ring:

$$\dot{Q} = S_p\alpha_p(T_p - T) + S_r\alpha_r(T_r - T), \tag{11}$$

where:
S – heat transfer area, α – heat transfer coefficient.

In the case of an inter-ring region, it was assumed that heat is exchanged between the gas and the cylinder and between the gas and the piston (heat exchange with the ring was excluded, as ring surfaces for these regions are much smaller than the other two surfaces):

$$\dot{Q} = S_c\alpha_c(T_c - T) + S_p\alpha_p(T_p - T). \tag{12}$$

The heat transfer between the gas in the inter-ring region and the cylinder is modelled as forced convection heat transfer for a fluid flowing parallel to a isothermal flat plate with laminar boundary layer. The influence of temperature on the thermo-physical properties of the fluid is considered by using an experimental coefficient given by Zukauskas [11]. According to the analysis given in [8], it is assumed that gas flow velocity relative to the cylinder liner equals to the piston speed v_p. With the above assumptions, the heat transfer coefficient between the gas and the cylinder is calculated from the formula:

$$\alpha_c = 0{,}664\pi\lambda \frac{\Pr^{0{,}52}}{\Pr_c^{\,0{,}19}} \sqrt{\frac{v_p p}{\mu RTL}}, \tag{13}$$

where:
λ – gas thermal conductivity, L – distance between rings, Pr i $\Pr_c$ – Prandtl numbers determined at gas and cylinder surface temperatures, respectively.

Considerations presented in [8] show that heat flux between gas and piston and gas and rings is smaller than between gas and cylinder liner. Taking this fact into account, and considering the difficulties with estimation of gas velocity in the regions (gas velocity depends on relative position of gaps of adjacent rings, which theoretical determination is actually impossible), we assume in the model that values of heat transfer coefficients between gas and piston α_p and gas and rings α_r are constant.

4.5. Volume of labyrinth stages

The model considers volume changes of regions during single engine work cycle, as a result of the piston and rings displacement relative to the cylinder with varying diameter, and as a result of the axial movements of the rings within the grooves. A scheme of volume calculation is presented in Fig. 7.

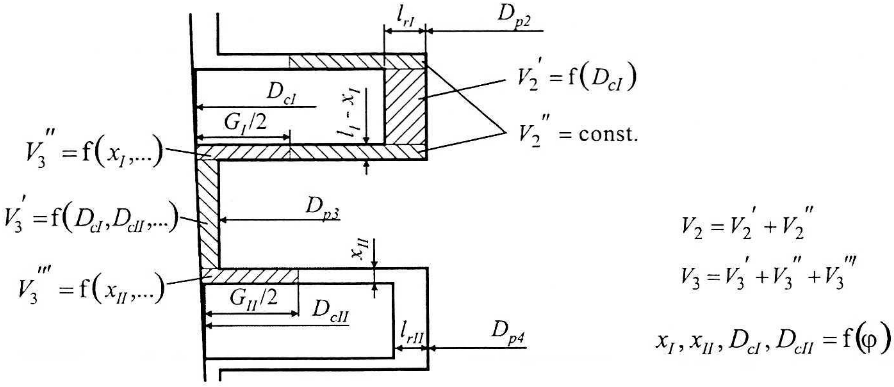


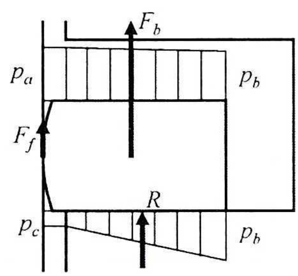
Fig. 7. A scheme to calculate the volumes of regions (for example the regions: behind the top ring and between the top and the second ring)

4.6. Ring dynamics model

Radial position of the ring in the groove is determined by the local diameter of the cylinder (it is assumed that the ring always contacts the cylinder). Axial position of the ring in the groove is calculated from the equation of equilibrium between forces acting on the ring axially. It was assumed that the forces that act on the ring in axial direction are: pressure force F_p, inertia force F_b and friction force F_f (Fig. 8). So, in piston co-ordinate system, we obtain the following equation:

$$F_p + F_b + F_f = m_r \frac{\mathrm{d}^2 x}{\mathrm{d}t^2}, \tag{14}$$

where:
m_r – mass of the ring, x – ring displacement relative to the piston.



$$F_p = B_r \frac{p_a - p_c}{2}$$

$$F_b = -m_r a_p$$

Fig. 8. Assumed pressure distribution and forces acting on the ring in axial direction (F_p – pressure force, F_b – inertia force, F_f friction force, a_p – piston acceleration, B_r – ring-side surface area)

The pressure force F_p is a resultant of forces generated by the pressure acting on the upper and lower surfaces of the ring. The force was calculated assuming commonly used linear pressure distribution – see equation on fig. 8. Fig. 8 also defines the inertia force F_b.

The friction force between the ring and the liner F_f is calculated using empirical relationship [4], [7]:

$$F_f = -f\pi D_c H (p_b + p_s), \tag{15}$$

where:
f – friction coefficient defined as:

$$f = 4{,}8\left(\mu_{oil}\frac{v_p}{H(p_b + p_s)}\right)^{\frac{1}{2}}, \tag{16}$$

μ_{oil} – dynamic viscosity of oil, H – height of the ring, p_b – pressure behind the ring, p_s – ring pressure on the liner, resulting from self-elasticity of the ring.

Reference [4] presents the analysis of how the method of calculation of the friction force F_f influences the calculated ring displacements and pressure courses in inter-ring regions. The authors of the cited work compared results obtained for empirical equation (15) and friction force calculated using a hydrodynamic model of lubrication. Differences were considered as insignificant (friction forces calculated with these two methods differed considerably in certain ranges of crankshaft positions but the friction force is the smallest of all the considered forces acting on the ring). According to this, the empirical dependence is also used in the developed model.

5. Concluding remarks

The developed model describes phenomena essential for the sealing operation of the PRC assembly, using physically based representations. Empirical relationships were used only for the description of effects of less significant meaning (e.g. friction forces) or insufficiently explained (e.g. flow coefficients). It is expected that the model will allow for better recognition of ring pack operation mechanisms, and so for predicting the influence of design and operational parameters on the blowby rate.

The second part of the presented paper will concentrate on preliminary results of simulation research of the model and verification research.

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Modelowanie przedmuchów spalin do skrzyni korbowej w silniku spalinowym. Część 1: Model matematyczny

Streszczenie

W pracy przedstawiono matematyczny model uszczelnienia tłok-pierścienie-cylinder umożliwiający lepsze rozpoznanie mechanizmów działania uszczelnienia, m.in. związków pomiędzy przemieszczeniami pierścieni w rowkach pierścieniowych tłoka a przepływem gazu przez uszczelnienie. Model pozwoli przewidywać wpływ różnych czynników konstrukcyjnych i eksploatacyjnych na natężenie przedmuchów spalin do skrzyni korbowej.

Opracowany model jest zintegrowanym modelem przepływu gazu przez szczeliny pomiędzy tłokiem, pierścieniami i cylindrem i przemieszczeń pierścieni w rowkach pierścieniowych tłoka. W modelu przestrzenie między- i zapierścieniowe rozpatrywane są niezależnie. Ponadto uwzględniono deformacje cieplne i zużycie elementów zespołu TPC oraz uszczelniające działanie pierścienia zgarniającego, a także wymianę ciepła pomiędzy przepływającym gazem a otaczającymi go ściankami.