Numerical analysis of gas flow through the labyrinth seal of piston rings of an automotive IC engine

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Abstract. In the paper a model of transient gas flows through the labyrinth seal: piston-rings-cylinder (PRC) has been presented. The results of calculations for a ring pack of three piston rings operating on a cylinder liner of a spark ignition internal combustion engine have been shown. The analysis of the gas dynamics and a parametric study considering this phenomenon has been carried out. It allowed evaluating the influence of the most important parameters of this sealing on the gas pressure acting on the rings and blow-by to the crankcase. The simulation results show the necessity of taking into account the following parameters: uneven wear of cylinder liner and its thermal and mechanical deformation, thermal deformation of piston rings, heat transfer between the flowing gas and the labyrinth walls, dynamics of axial ring displacements in piston grooves.

1. Introduction
Studies of piston ring pack operation have drawn attention of many researchers because of their importance in determining the mechanical efficiency, wear, fuel economy and exhaust emissions of an internal combustion (IC) engine [2-4,7,10,11]. The system piston–rings–cylinder (PRC) is generally the greatest contributor to engine frictional losses - about 20 – 40% [5]. Piston rings are the most complicated tribological components in the IC engine to analyze because of large variations of speed, load, temperature and lubricant availability. Sliding surfaces of piston rings and cylinder liner may experience boundary, mixed and hydrodynamic lubrication in one single stroke of the piston [1,3,12]. Therefore the numerical simulation of these processes, which take place in a typical piston ring pack operation, is important from practical point of view. In contrast to the previous papers of the author [14-17] concerning mostly hydrodynamic and mixed lubrication of piston rings, this time a set of computational results characterizing gas flow process in the PRC system has been presented.

2. Modelling of piston ring pack operation
2.1. Developed gas flow model
In the paper a comprehensive model of transient gas flows through the labyrinth seal: piston-rings-cylinder (PRC) has been presented. The model consists of several volume regions V₁, V₂, ..., V₅, which are connected by orifices with cross-section areas A₁, A₂, ..., A₆ (Fig. 1). The volumes V₃, V₅ correspond to volumes between the piston rings, while volumes V₂, V₄ correspond to groove volumes behind the rings. Orifices with cross-section areas A₁, A₄ correspond to the ring end gaps, whereas orifices with cross-sections A₂, A₃, A₅ correspond to ring-side crevices [7-9].
Figure 1. Scheme of gas flow through the labyrinth seal: piston – rings – cylinder liner and the applied physical model for the ring pack of three piston rings

It has been also assumed that the gas flow through orifices is isentropic (depending on pressure ratio – subsonic or sonic). Leaks between piston rings and cylinder liner are defined by flow areas of ring end gaps and ring-side crevices [3,6-9,12]. In the mathematical model of these phenomena equations of the following physical laws are utilized (here given for a gas volume region number k):

**Equation of mass balance:**

\[
\sum_i dm_{in_i} - \sum_j dm_{out_j} = dm_k
\]  

**Equation of energy balance:**

\[
\sum_i dm_{in_i} \cdot h_{in_i} - \sum_j dm_{out_j} \cdot h_{out_j} - \delta Q_{\text{wall}} - \delta E_{\text{fric}} = d(m_k \cdot u_k) + p_k \cdot dV_k
\]  

**Gas state equation in differential form:**

\[
\frac{dV_k}{T_k} = \frac{dm_k}{p_k} + \frac{dV_k}{V_k} - \frac{dm_k}{m_k}
\]  

where: \(m\) – gas mass, \(p\) – gas pressure, \(T\) – gas temperature, \(u\) – internal gas energy, \(h\) – gas enthalpy, \(Q\) – heat transferred through cylinder walls, \(E\) – friction loss energy; Indexes: \(in\) – gas inflow, \(out\) – gas outflow, \(i\) – number of inflow channel, \(j\) – outflow channel number, \(k\) – analysed gas volume.

In the presented work, variation of the pressure inside the cylinder is obtained by numerical simulation of the physical processes taking place inside the cylinder during a single cycle. The pressure between all the piston rings was calculated using the above presented gas flow model. For this purpose a computer program GASFLOW was developed and applied. Due to the relatively high pressure differences between these volumes, the gas dynamic equation including both subsonic and sonic flows in orifices connecting volumes had to be used. For both mentioned cases the mass flow rate through an orifice can be calculated from the following expression [15]:

\[
\text{mass flow rate} = \frac{A \cdot \sqrt{2 \cdot \gamma \cdot \left(\frac{p_2}{p_1}\right) - 1}}{\left(\frac{p_2}{p_1}\right)^{\gamma / \gamma - 1}}
\]
\[ \dot{Q} = \frac{dm}{dt} = \begin{cases} C_C A \frac{p_0}{T_0} \left( \frac{2k}{k-1} \right)^{\frac{2-k}{k-1}} \left( \frac{p}{p_0} \right)^{\frac{2}{k}} \left( \frac{p}{p_0} \right)^{\frac{k+1}{k}} & \text{for } \frac{p}{p_0} > \left( \frac{2}{k+1} \right)^{\frac{k}{k-1}} \\ C_C A \frac{p_0}{T_0} \left( \frac{2k}{k-1} \right)^{\frac{k+1}{k-1}} \left( \frac{p}{p_0} \right)^{-\frac{k+1}{k}} & \text{for } \frac{p}{p_0} \leq \left( \frac{2}{k+1} \right)^{\frac{k}{k-1}} \end{cases} \]  

(4)

where: \( p \) - lower pressure, \( p_0 \) - higher (stagnation) pressure, \( T_0 \) - stagnation temperature, \( k \) - isentropic exponent, \( R \) - gas constant, \( C_c, C_v \) - contraction and velocity loss coefficient, \( A \) - cross section area of the considered ring end gap or ring-side crevice.

The heat flow \( dQ/dt \) transferred between a given volume zone Vol and the surrounding walls is [9]:

\[ \dot{Q}_{wall} = \frac{dQ_{wall}}{dt} = \alpha_c (T_{vol} - T_c) A_c + \alpha_p (T_{vol} - T_p) A_p + \alpha_r (T_{vol} - T_r) A_r \]

(5)

where: \( \alpha \) - heat transfer coefficient, \( T \) - temperature, \( A \) - heat transfer surface area (an appropriate part of the labyrinth surface), \( t \) - time; Indexes: \( c \) - cylinder, \( p \) - piston, \( r \) - ring, \( Vol \) - gas volume zone number.

The heat transfer coefficient \( \alpha_c \) between the gas and the cylinder (in inter-ring volumes) is calculated from the formula given in [9] and literature on heat transfer. It describes the heat transfer concerning the laminar flow over a flat plate, where the gas velocity relative to the cylinder liner is equal to the piston velocity:

\[ \alpha_c = 0.66 \frac{\lambda}{x_{br}} \text{Re}^{0.5} \text{Pr}^{0.33} \left( \frac{\text{Pr}_c}{\text{Pr}} \right)^{0.19} \]

(6)

where: \( \lambda \) - gas thermal conductivity, \( x_{br} \) - distance between rings, \( \text{Re} \) - Reynolds number (calculated for gas velocity relative to cylinder equal to piston speed), \( \text{Pr} \) and \( \text{Pr}_c \) - Prandtl number (thermophysical parameters calculated for gas or cylinder temperature, respectively).

Values of \( \alpha_c \) and \( \alpha_p \) coefficients were assumed to be constant (similarly as in [7-9]). A scheme of forces acting on a piston ring, action lines and distances between these forces and the centre of gravity \( S \) of the ring cross-section are shown in Fig. 2. All the force values are referenced to the unit of the ring circumference \( c_{mc} \) (unit forces [N/m]).

**Figure 2.** Scheme and definitions of forces acting on a piston ring. Nomenclature: \( F_h \) – hydrodynamic normal force, \( F_c \) – contact normal force, \( F_{spr} \) – ring spring force, \( F_{gas} \) – back ring gas force, \( F_{ygi} \) – leading edge gas force, \( F_{ygi+1} \) – trailing edge gas force, \( R_s \) – groove reaction force, \( F_{fr} \) – viscous...
friction force, $F_{cx}$ – contact friction force, $F_{gi}$ – leading side gas force, $F_{gi+1}$, $F_{gi+2}$ – trailing side gas forces, $m$ – ring mass, $g$ – gravitational acceleration, $a_p$ – piston acceleration

Typical equation in axial direction for the single-lip piston ring has the following form [15]:

$$\sum F_x = R_x - F_{pri} - F_{cx} + F_{gi} + F_{gi+1} + F_{gi+2} - \frac{m}{c_{inc}} (g + a_p) = 0 \quad (7)$$

Using the equation (7), one can calculate the reaction force $R_x$ between the ring and piston groove for every time step. If the sign of this force changes, the axial movement of the ring in the piston groove begins. At this point, the value of the reaction force $R_x = 0$ and the axial movement of the ring relative to the piston groove can be described by the following differential equation [15]:

$$\frac{m}{c_{inc}} \frac{d^2 x_r}{dt^2} = -F_{pri} - F_{cx} + F_{gi} + F_{gi+1} + F_{gi+2} - \frac{mg}{c_{inc}} \quad (8)$$

The ring movement $x_r$ completes when the ring reaches the opposite side of the piston groove. However, the initial ring lift may be followed by its axial oscillations within the piston groove (Section 3.6).

Significant forces are gas and inertia forces. The friction forces (due to hydrodynamic lubrication and elastic contact with surface roughness) play a much smaller role [3,6,9,12]. The sub-models of the hydrodynamic lubrication and elastic contact of rough surfaces between the ring and cylinder liner are described in detail in publications [13-17] of the author. In this paper only the problems concerning ring axial movement in the piston groove are shortly presented.

The developed program GASFLOW uses the simplified friction model between the ring and cylinder liner. Namely, the friction force (per unit of circumference) can be calculated as follows [8,9]:

$$F_i = f H (p_b + p_s) \quad (9)$$

whereas the coefficient of friction is described by the following relationship [8,9]:

$$f = 4.8 \left[ \frac{\mu_{ol}}{H (p_b + p_s)} \right]^{0.5} \quad (10)$$

where: $\mu_{ol}$ - dynamic oil viscosity, $H$ – axial height of the ring, $p_b$ - gas pressure behind the ring, $p_s$ - pressure resulting from the ring elastic tension force

2.2. 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 [13-15]. 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% [13-15].

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 Tab. 1.
Table 1. Main engine parameters

| Parameter               | Value          |
|-------------------------|---------------|
| 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 (Tab. 2).

Table 2. Main parameters of piston rings

| Parameter                           | Ring 1     | Ring 2     | Ring 3     |
|-------------------------------------|------------|------------|------------|
| Total axial height                  | 1.48 mm    | 1.98 mm    | 4.0 mm     |
| Axial height (upper land)           | --         | --         | 1.0 mm     |
| Axial height (lower land)           | --         | --         | 1.0 mm     |
| Radial width                        | 4.5 mm     | 4.5 mm     | 4.5 mm     |
| Nominal ring end gap                | 0.575 mm   | 0.50 mm    | 0.50 mm    |
| Mass                                | 12.5 g     | 16.7 g     | 25 g       |
| Elastic tension force (per unit of circumference) | 375 N/m | 350 N/m | 1545 N/m |

3.2. Calculation results for a basic case

The reference to the later described parametric study of the gas flow through the labyrinth seal of piston rings is the so-called basic calculation case. In this case, a fixed angle $\gamma = 3'$ (Fig. 3) is assumed for the cylinder liner conical shape resulting from its thermo-mechanical deformation and wear.

In the following figures variations of 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 the crankshaft rotation (piston top dead centre - TDC).

![Figure 3. The clearance between the piston and cylinder depending on the position of the piston in the cylinder (for the conical cylinder profile – angle $\gamma$)](image)

![Figure 4. Gas pressure $p_i$ in all volume zones versus crank angle rotation: $p_1$ – in combustion chamber, $p_2$ – behind the 1st piston ring, $p_3$ – among the 1st and 2nd ring, $p_4$ – behind the 2nd ring, $p_5 \approx p_0$ – in crankcase)](image)

Fig. 4 presents the pressure variation between piston rings, calculated on the basis of known cylinder pressure variation and simulated gas leakage through the labyrinth seal of the piston ring pack (orifices corresponding to ring end gaps and ring-side crevices – Fig. 1). Generally, the gas pressure in the cylinder and all the inter-ring gas pressures increase during the piston upstroke (primarily compression
phase) and decrease during the piston downstroke (expansion phase). In Fig. 4 the following maximum gas pressure values can be noticed: nearly 4.5 MPa (45 bar) in combustion chamber and about 1.2 MPa (12 bar) between the 1st and 2nd piston ring. In some cases during the expansion phase the gas pressure under the first piston ring can reach a higher value than the gas pressure over this ring. For this reason axial ring lifts in piston grooves are anticipated (Fig. 6). On the piston underside a constant pressure gas in the crankcase (equal approximately to ambient pressure) is assumed. The higher is the gas pressure the stronger is the radial gas force acting to increase the ring diameter. It means that the radial gas force can be many times greater than the natural force due to ring stiffness acting in the same direction. The 1st piston ring (top ring) is strongly pressed against the cylinder liner surface. Certainly the 2nd piston ring is significantly less loaded.

Then in Fig. 5 the calculated gas flow rates concerning the ring end gaps of the 1st and 2nd piston ring are presented. Positive values of these flow rates mean the flows towards the crankcase, and their negative values mean the return flows. In the compression stroke (0-180° of the crankshaft rotation), the flow rate \( Q_1 \) (through the 1st ring end gap) is clearly higher than the flow rate \( Q_4 \) (through the 2nd ring end gap).

In Fig. 6 axial ring lifts in piston grooves as a function of crank angle for analysed engine are shown. Due to big changes of piston acceleration the axial movements of the 1st and 3rd (Fig. 1) piston rings within their grooves can be observed. This phenomenon takes place twice during the whole cycle (0°–720°) of the four-stroke engine. The 2nd piston ring (Fig. 1) is strongly pressed by the gas force against the lower flank of piston groove and does not move during the compression and expansion strokes. But then during the exhaust and suction strokes the mentioned gas force is less important and two lifts of the 2nd piston ring within its groove can be observed. Gas pressure variations near the crank angle of 540° are followed by oscillations of axial movement of the 1st and 2nd piston ring.

![Figure 5. Gas flow rates \( Q_i \) (i = 1, 4) through the ring end gaps of piston rings versus crank angle rotation (\( Q_1 \) - for the 1st ring, \( Q_4 \) - for the 2nd ring)](image)

![Figure 6. Axial ring lifts \( X_{ri} \) (i = 1, 2, 3) in piston grooves versus crank angle (\( X_{r1} \) - for the compression ring, \( X_{r2} \) - for the scraper ring, \( X_{r3} \) - for the two-lip oil ring)](image)

The nominal axial clearance of the first and second ring in piston grooves equals 0.20 mm and for the third piston ring - 0.25 mm. Rapid ring lifts are accompanied by short lasting, but big changes of gas flow areas between rings and piston grooves. They are much bigger than flow areas of ring end gaps, because the opening ranges include the whole piston circumference. In this case temporary rapid changes of gas flow rates should be expected. This phenomenon has been taken into account, but is not shown in this paper. Then the engine cycle-averaged mass flow rate (blow-by) \( G_{sp} = 0.257 \text{ g/s} \) to the crankcase has been calculated.

In the further part of the article, a parametric study of the gas flow process through the piston ring pack is made using the developed mathematical model. The purpose of the simulation tests is to assess the impact of the main parameters of the analyzed process on the gas blow-by to the engine crankcase. The basic calculation case described above is always used as a reference.
3.3. Influence of the shape of cylinder liner surface

Calculations have been made for perfectly cylindrical cylinder liner (nominal clearance between the piston and cylinder), as well as for the assumed conicity of the cylinder surface resulting from its thermo-mechanical deformation and wear (basic case). Then, the selected calculation results have been compared and shown in Figs. 7 - 8.

In Fig. 7 pressure variations in the main volume zones, i.e. in combustion chamber, between the 1st and 2nd ring and in the zone between the 2nd and 3rd ring (connected with the channel to the crankcase) are shown. The comparison of the pressure variation in function of crank angle can only concern the zone between the 1st and 2nd ring (Fig. 1), i.e. the pressure $p_3$. This is due to the fact that in all the cases described in this paper, the gas pressures $p_1$ in the combustion chamber have been assumed to be the same. In addition, a constant pressure gas $p_5$ in the crankcase equal to ambient pressure has been always assumed. In the case of the ideally cylindrical shape of the cylinder liner, the pressure $p_3$ reaches a lower peak value (in the expansion stroke - around 220° of crank angle), than by assuming its approximately conical shape (caused by wear and thermo-mechanical deformation).

It can be explained in the following way. For a perfectly cylindrical cylinder liner the clearance between the piston and the ring is smaller than in the case of the conical cylinder. As a result, cross-section areas of the ring end gaps are constant for the first of these cases. In the second case, they are larger and change in the following way. They increase in the compression and exhaust strokes (0-180° and 360-540°) and decrease in the intake and expansion strokes (180-360° and 540-720°).

For the above reasons, the mass flow rates through the ring end gaps are greater when considering the conical shape of the cylinder liner compared to the case of the perfectly cylindrical surface. This can be proved by the variation of the relevant gas flow rates shown in Fig. 8. A larger gas flow rate through the 1st ring end gap causes the gas pressure $p_3$ to increase in the zone between the rings 1 and 2 (Fig. 7).

If a perfectly cylindrical cylinder liner were assumed, the cycle-averaged blow-by to the crankcase would be smaller ($G_{sp} = 0.205$ g/s) than in the case of a real conical shape of cylinder liner ($G_{sp} = 0.257$ g/s).

![Figure 7. Comparison of gas pressure variations $p_i$ ($i = 1, 3, 5$) in selected volume zones versus crank angle. The "cs" index when specifying the pressure $p_i$ means taking into account the conical shape of cylinder liner in the calculations](image1)

![Figure 8. Comparison of gas flow rates $Q_i$ ($i = 1, 4$) through the ring end gaps of piston rings versus crank angle rotation ($Q_1$ - for the 1st ring, $Q_4$ - for the 2nd ring). The "cs" index when specifying the flow rate $Q_i$ means taking into account the conical shape of cylinder liner in the calculations](image2)
Subsequent results of simulation tests (Figs 9 - 10) indicate that it is important to take into account the thermal expansion of piston rings causing the increase of their circumferential length, and thus reducing the crevices of ring end gaps.

If this phenomenon were neglected, it would lead to an overestimation of the gas pressure $p_3$ in the volume zone between the 1st and 2nd ring (Fig. 9). This is due to the overestimation of cross-section areas of the ring end gaps, and consequently the mass flow rates through these gaps (Fig. 10). This would also lead to the calculation of increased cycle-averaged blow-by to the engine crankcase. It would equal in this case $G_{sp} = 0.271$ g/s.

3.5. Influence of heat transfer between the flowing gas and the labyrinth walls

If the phenomenon of heat transfer were neglected, it would lead to an overestimation of the gas pressure $p_3$ in the zone between the 1st and 2nd piston ring (Fig. 11). The reason is an increase of gas temperature, and thus the lowering of its density in the mentioned volume zone. Consequently, the increase of gas pressure in this zone occurs. It can also be observed that the mass flow rate through the labyrinth crevices decreases, thus reducing the cycle-averaged blow-by to the engine crankcase. It equals in this case $G_{sp} = 0.190$ g/s. The increased pressure $p_3$ also results in some changes in the start and end times of the axial lifts of the 1st and 2nd ring in piston grooves (Fig. 12).
account the heat transfer between flowing gas and the labyrinth walls in the calculations

3.6. Influence of engine rotational speed

One of the main objectives of the computational study has been to evaluate the influence of the engine rotational speed on gas pressure between the piston rings (Fig. 13). The reduction of the engine speed to 2000 rpm in relation to the basic calculation case \( n = 3400 \text{ rpm} \) results in the increase of the gas pressure \( p_3 \) in the volume region between the 1\textsuperscript{st} and 2\textsuperscript{nd} piston ring (Fig. 13). The reason is the longer duration of filling the mentioned gas zone by the flowing gas.

![Figure 13](image_url)

**Figure 13.** Comparison of gas pressure variations \( p_i \) \((i = 1, 3, 5)\) in selected volume zones versus crank angle. The indices "n\textsubscript{1}" and "n\textsubscript{2}" when specifying the pressure \( p_i \) mean the results for the engine rotational speed of 2000 and 3400 rpm, respectively.

Increasing the engine speed in relation to the basic calculation case \( n = 3400 \text{ rpm} \) results in rapid fluctuations in the gas pressure \( p_3 \) in the zone between the 1\textsuperscript{st} and 2\textsuperscript{nd} ring during the compression and expansion strokes – not graphically shown in this paper. This is caused by the lifts of the 2\textsuperscript{nd} ring relative to the bottom flank of the piston groove starting at approximately 120° of crank angle. The ring vibrates causing subsequent fluctuations in the gas pressure \( p_3 \). The reason for initiating the lifts of this ring is that at higher engine speed the inertia forces of the rings increase significantly due to the piston acceleration. For \( n = 5200 \text{ rpm} \), the cycle-averaged blow-by to the crankcase is \( G_{sp} = 0.307 \text{ g/s} \).

Taking into account the phenomenon of piston rings vibrations (flutter) has a clear justification for high-speed engines. The final result of described analysis is shown in Fig. 14, i.e. the influence of engine rotational speed on cycle-averaged blow-by \( G_{sp} \) through the labyrinth seal of piston rings. It has been proved that the higher rotational speed is applied the more gas flow rate is detected.

**Figure 14.** Influence of engine rotational speed on cycle-averaged gas flow rate \( G_{sp} \) through the labyrinth seal of piston rings

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. The simulation model characterizes well the piston ring pack operation of different kinds of internal combustion engines (two- and four-stroke) [13-17]. It concerns mainly variations of gas pressure between the rings as functions of crank angle and exhaust gas flow rates (blow-by) through the labyrinth seal of a piston ring pack;
3. The simulation results have shown the necessity of taking into account the following parameters: uneven wear of cylinder liner and its thermal and mechanical deformation, thermal deformation of
piston rings, heat transfer between the flowing gas and the labyrinth walls, engine rotational speed and dynamics of axial ring displacements in piston grooves;

4. If a perfectly cylindrical cylinder liner were assumed, the cycle-averaged blow-by to the crankcase would be smaller than in the case of a real conical shape of cylinder liner (Section 3.3);

5. If the thermal deformation of piston rings were neglected, it would lead to the calculation of increased gas flow rate to the engine crankcase (Section 3.4);

6. If the heat transfer between the flowing gas and the labyrinth walls were neglected, the cycle-averaged blow-by to the crankcase would be reduced (Section 3.5);

7. The higher rotational speed is applied the more gas flow rate through the labyrinth seal of piston rings is detected (Fig. 14);

8. The developed model and software can be utilized for optimization of piston rings design.

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