Scientific Journal of Silesian University of Technology. Series Transport
Zeszyty Naukowe Politechniki Śląskiej. Seria Transport

Volume 104 **2019**

p-ISSN: 0209-3324

e-ISSN: 2450-1549

DOI: https://doi.org/10.20858/sjsutst.2019.104.17

Journal homepage: http://sjsutst.polsl.pl

Article citation information:
Zarenbin, V., Kolesnikova, T., Sakno, O., Ollo, V., Klimenko, V. Impact evaluation of piston rings mobility on a gas passage in an internal combustion engine (ICE). *Scientific Journal of Silesian University of Technology. Series Transport*. 2019, **104**, 187-201. ISSN: 0209-3324. DOI: https://doi.org/10.20858/sjsutst.2019.104.17.

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IMPACT EVALUATION OF PISTON RINGS MOBILITY ON A GAS PASSAGE IN AN INTERNAL COMBUSTION ENGINE (ICE)

Summary. To estimate the effect of the axial movement of piston rings in the piston grooves on the blow-by in the internal combustion engine (ICE) by an experiment- calculated method. This contributes to the development of practical recommendations for the further improvement of the engine ring seal designs. Abstract theorems were used when modelling the effect of the axial movement of piston rings in the piston grooves on the blow-by in an ICE. They are based on the fundamental theory of heat engines, thermodynamics and hydraulics. The ICE running was analysed using design-theoretical research methods. The effect of the axial movement of piston rings in the piston grooves on the blow-by in the ICE was established. This creates prerequisites for a more accurate assessment of their sealing capacity and for ways to further improve them. Calculated dependences

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for computing the blow-by depending on the positional relationship of the rings in the piston grooves were obtained. The dependences of gas escapes on the engine crankshaft speed were obtained, which is especially important for idling modes by which one can judge the dynamic stability of the ring seal and solve the problems of improving its service properties. The calculated dependences for evaluation of the blow-by depending on the positional relationship of the rings in the piston grooves and their respective possible gas flows in the ring seal were obtained for the first time. The practical method for estimating the dynamic stability of the ring seal by decencies of gas escape on the crankshaft rotation speed in ICE was proposed.

Keywords: ICE ring seal, piston ring mobility, calculation of gas escape.

1. INTRODUCTION

The internal combustion engines (ICE) of trucks used in the mining industry are one of the critical units with expensive repairs. The ICE requires a preliminary diagnosis, which assesses the technical condition of both the engine and the truck as a whole.

The ICE generate vibrations and noise during operation. They can also be successfully used for diagnostic purposes. Such methods have been increasingly used in recent years [3-8,10,16,18,23,24].

The technical condition of the parts of the cylinder-piston group (CPG) can be determined by gas escape in the ICE.

The modern development of the high-speed ICE is the way to improve their technical, economic and environmental performance. This predetermines [1,2,9,15,20]:

a) the expansion of research and development projects on the further design and technological improvement of parts of the CPG of the ICE
b) the choice of optimal conditions for interfacing their contacting surfaces
c) improving the quality of used materials

Piston rings (PR) are the most high-wear parts of the CPG, thus, the issues of improving their performance and reliability are of current importance when creating prospective engines used in the mining industry.

The main factors that determine the normal running of the ICE are the condition of coupling of surfaces of the compression rings with the cylinder wall and their ends with the top and bottom planes of the piston grooves. This is connected with the sealing of the combustion chamber and prevention of a considerable blow-by in the engine case.

Gas escapes through the gaps in the parts mating of the CPG, break the oil film and increase wear which, in turn, increases gas escape. This furthers the seizing of the piston rings, increase in oil consumption and fuel, and smoking. The final result is jamming and engine trouble.

A wide variety of factors influencing the operation of the PR complicates analysis and generalisation of the experimental data and development of general principles of the theoretically substantiated choice of designs of the ICE ring seal. The solution of this issue could be based on the account of the totality of the main phenomena that determine the operability of CPG and PR parts.

Numerous papers cover the experimental and theoretical study of the PR operation. In [12] it was found that the hydrodynamic friction increased with the initial wear of the PR in conditions of increasing minimum thickness of the oil film. This contributes to the fact that the PR can remain operational during the entire service life. Hydrodynamic friction for high rings can be reduced using a narrow parabolic profile, which is impossible for narrow rings.

A laser fluorescence system was developed to visualise the thickness of the oil film between the PR and the cylinder wall of the running gasoline engine through a small optical window installed in the cylinder wall. The results show significant differences in the profiles of the thickness of the lubricant film for the ring seal if the lubricant deteriorates, which affects the ring friction and, ultimately, fuel economy [14].

The diagnostic methodology can effectively determine the control of the condition of the PR in accordance with the characteristics of combustion [13].

The calculation of the gas flow through the ICE ring seals with regard to the piston rings dynamics allows diagnosing the engine technical condition.

2D CFD model is used to study the effect of the ring seal design on the friction process, oil consumption and oil flow. Calculations of the piston rings dynamics were carried out on the assumption of forces balance [11].

Methods and devices to study mechanical friction losses were developed [19]. A simplified floating liner method was used and the test equipment was developed to fill the gap in between the full floating liner engine and the typical component bench test equipment.

The purpose of this research [17] was to study the potential of the laser oil pockets new design so as to improve the piston rings lubrication. These pockets make it possible to achieve significant friction reduction by using appropriate geometric parameters [7,8,10,24].

Presently, there is a wide range of solution of the PR reliability and operating life problems. However, the dynamics of the parts of the CPG are not sufficiently considered. In particular, this is the PR movement in the piston grooves. This is connected with the engine running, where all piston rings moving are difficult to measure; there are no theoretical dependencies that link the PR mobility in the piston grooves with gas escapes through the ring seal.

2. DYNAMIC ANALYSIS OF GAS FLOW THROUGH THE ICE RING SEAL

In general, the problem of the gases flow through the volumes on lands and piston ring grooves is quite complicated. However, it can be simplified if the following experimental and theoretical justifications of assumption are introduced:

a) the gas flow process is taken to be quasi-stationary
b) the areas of the flow passage between the PR, the piston and the cylinder liner should be replaced by the equivalent area of the flow passage of the piston-ring lock
c) geometric relationships in the ring seal can only be changed due to axial unloading of the rings and their subsequent separation from the bearing area of the groove

In this paper, the problem is considered on the example of the ring seal consisting of three rings in various cases of their relative position in the grooves, which received experimental confirmation in the paper.

The principal features of the adopted model were that it takes into account the throttling effect of the upper fascia of the piston and the change in the areas of the flow sections and the volume of the annular spaces due to the movement of the rings in the grooves.

It is accepted that the separation of the rings from the support surfaces of the groove in the direction of the piston axis occurs at the moments when the sum of the forces from the gas pressure P_G, the inertia of the ring P_j and the friction P_f are zero, which means

$$\sum P = P_G + P_j + P_f = 0 \tag{1}$$

The theoretical studies were based on the differential equations of mass and energy balances, as well as the criterion equation of heat exchange for the gases flow in micro-gap channels. For the second and third piston grooves, the gases flow was accepted to be isothermal with a gas temperature equal to the arithmetic average of the temperatures of the piston grooves and the cylinder liner.

As a result of the dynamic calculation the total forces that act on the piston rings, as well as various cases of their positional relationship in the grooves and their corresponding possible gas flows in the ring seal were identified.

Blow-by *m* through the PR leakiness was calculated by the formula:

$$dm = \mu \cdot f \cdot \psi \cdot p \cdot \sqrt{\frac{1}{R \cdot T}} d\tau, \tag{2}$$

where $\mu \cdot f$ – discharge coefficient and flow section between volumes on lands and the piston ring grooves [m^2]; ψ – speed function, which depends on the pressure ratio; p, T – pressure [Pa] and gas temperature in the grooves [K]; R – gas constant, R=287 [J/(kg·K)]; τ – time [s].

The following formulas to calculate the pressures and the blow-by in various cases of the positional relationship of the rings in the piston grooves were obtained.

2.1. Case 1 of the positional relationship of the rings in the grooves of the piston, when $\sum P_1 > 0$, $\sum P_2 > 0$ and $\sum P_3 > 0$

The initial equations for the calculation are the mass and energy balance Equations. Cylinder pressure ($P_{cyl} > P_1 > P_2$) is calculated

$$dm_1 = dm_{cyl} - dm_2 \tag{3}$$

$$d(m_1 u_1) = d(m_{cyl} i_{cyl}) - d(m_2 i_1) \pm dQ_1 \tag{4}$$

where dm_1 and $d(m_1 u_1)$ – change of elementary mass and internal energy of gases in the top groove during dt; dm_{cyl} and $d(m_{cyl} i_{cyl})$ – the elementary mass and enthalpy of gases which flow into the top groove out of the cylinder during dt; dm_2 and $d(m_2 i_1)$ – the elementary mass and enthalpy of gases which flow from the top groove to the second groove during dt; dQ_1 – the elementary quantity of heat that is transferred (or perceived) gases surrounding surfaces of CPG parts during dt.

The Equation (4) is differentiated. The Equation (3) is substituted into this equation. As a result, the following equation has the form:

$$m_1 d(u_1) = dm_{cyl}(i_{cyl} - u_1) - dm_2(i_1 - u_1) \pm dQ_1 \tag{5}$$

Internal energy of gases is determined $U = c_v \cdot T$ (c_v – mass isochoric heat capacity of gases at constant volume V in the ring groove); the enthalpy of gases is determined $i = c_p \cdot T$ (c_p – mass isobaric heat capacity of gases at constant volume V in the ring groove); gas state is determined $PV = mR \cdot T$. The following equations are used

$$dm = \mu f \psi \sqrt{P/V}\,dt \text{ and } dQ = \alpha \cdot (T - T_w)dt \qquad (6)$$

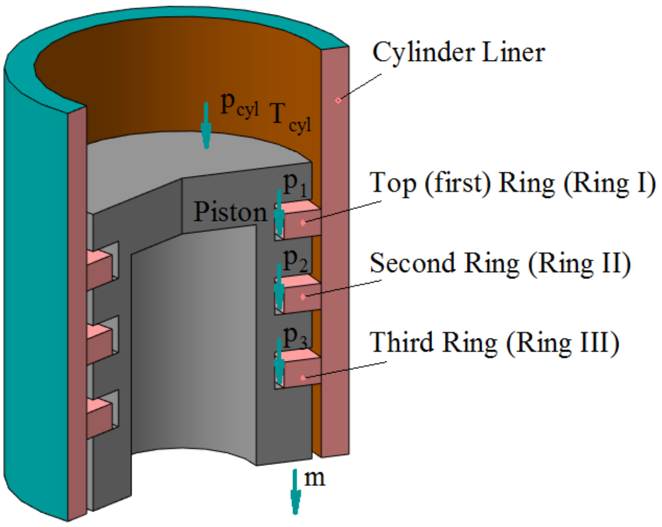
where α – heat exchange coefficient from gases to surfaces of the cylinder and the piston [$\frac{W}{m^2 \cdot K}$]; T_w – cooling surface temperature [K].

The equation is transformed and as a result has the following form

$$\frac{dT_1}{dt} = a_1 \cdot \frac{T_1}{p_1} \cdot \left[\overline{\mu_1 \cdot f_1} \cdot \psi_{cyl} \cdot \left(\frac{k_{cyl}}{\beta_1} \sqrt{T_{cyl}} - \frac{T_1}{\sqrt{T_{cyl}}} \right) - (k_1 - 1) \cdot \psi_1 \cdot \overline{p}_1 \sqrt{T_1} \pm \frac{\alpha_1 \cdot F_1 \cdot \Delta T_1 \sqrt{R}}{\mu_1 \cdot f_1 \cdot \beta_1 \cdot c_{V_{cyl}} \cdot p_{cyl}} \right] \qquad (7)$$

where $a_1 = \mu_1 \cdot f_1 \sqrt{R}/V_1$; $\overline{\mu_1 \cdot f_1} = \frac{\mu_{cyl} \cdot f_\Delta}{\mu_1 \cdot f_1}$; $\overline{p}_1 = \frac{p_1}{p_{cyl}}$; $\Delta T_1 = 2T_f - T_w - T_{pist}$; $\beta_1 = C_{V_1}/C_{V_{cyl}}$; $\psi_{cyl} = f(p_m, p_{cyl})$; $\psi_1 = f(p_2, p_1)$; V_1 – volume of ring groove I [m^3]; T_n, T_w – average temperature of the piston head [K]; $T_f = (T_{cyl} + T_1)/2$ – the determining gas temperature [K]; p_m – gas pressure in the minimum section of the jet [Pa]; p_{cyl} – gas pressure in the cylinder [Pa]; T_{pist} – average temperature of the piston head and cylinder liner [K].

The calculation of the blow-by when the sum of the forces acting on the rings $\sum P$ is positive and the gases pressure decreases from the top PR to the bottom (Fig. 1) is the following: $\sum P_1 > 0$, $\sum P_2 > 0$ and $\sum P_3 > 0$ with $p_1 > p_2 > p_3$.



m – mass gas escape; p_1, p_2, p_3 – gas pressure in annular piston cavities I, II and III

Fig. 1. Case 1 of the positional relationship of the rings in the grooves of the piston, when $\sum P_1 > 0$, $\sum P_2 > 0$ and $\sum P_3 > 0$

Then

$$\frac{dp_1}{d\tau} = a_1 \cdot p_{cyl} \cdot \left[\overline{\mu \cdot f_0} \cdot \psi \cdot \left(\frac{p_m}{p_{cyl}} \right) \cdot \frac{k_{cyl}}{\beta_1} \sqrt{T_{cyl}} - k_1 \cdot \psi \cdot \left(\frac{p_2}{p_1} \right) \cdot \frac{p_1}{p_{cyl}} \cdot \sqrt{T_1} - \frac{\alpha \cdot F_1 \cdot \Delta t_1 \sqrt{R}}{\overline{\mu \cdot f_1} \cdot \beta_1 \cdot c_{cyl} \cdot p_{cyl}} \right] \quad (8)$$

where p_H – gas pressure in the crankcase [Pa]; k_1 – the adiabatic coefficient of gas per ring I; k_{cyl} – the adiabatic coefficient of gas in the cylinder; T_{cyl} – gas temperature in the cylinder [K]; F_1 – heat receiving surface area [m^2]; c_{V_1} – mass heat capacity of gases at constant volume in the ring groove I [$\frac{J}{kg \cdot K}$]; c_{cyl} – mass heat capacity of gases at constant volume in the cylinder [$\frac{J}{kg \cdot K}$]

and

$$\overline{\mu \cdot f}_0 = \frac{\mu_0 \cdot f_0}{\mu_1 \cdot f_1} \qquad \overline{\mu \cdot f}_1 = \frac{\mu_1 \cdot f_1}{\mu_2 \cdot f_2} \qquad \Delta t_1 = 2 \cdot t_f - t_w - t_n \qquad t_f = \frac{t_{cyl} + t_1}{2} \quad (9)$$

Pressure difference in the heat zone of the piston, subject to maintaining the constancy of the gases velocity is determined

$$-\frac{dp}{dx} = \frac{\zeta_{cyl} \cdot \rho_{cyl} \cdot W^2}{2dE} \quad (10)$$

where p – gas pressure in the annular gap between the heat zone of a piston and cylinder; W – average consumed gases velocity; ρ_{cyl} – gas density; $dE = 2\Delta_g$ – the equivalent diameter of the channel in the form of gap; Δ_g – width of a gap; ζ_{cyl} – movement gas resistance coefficient.

The Equation (10) is integrated on condition that $\zeta_{cyl} = \frac{96}{R_e}$ (R_e is Reynolds number).

Then

$$\Delta p = p_m - p_1 = W_m \cdot \rho_m \cdot \frac{c_m}{2} \quad (11)$$

where ρ_m is gas density in the minimum section of the jet and $\rho_m = \rho_{cyl} \cdot \left(\frac{p_m}{p_{cyl}} \right)^{1/K_{cyl}}$.

Then

$$p_m - p_1 = W_m \cdot \rho_m \cdot \frac{c_m}{2} \quad (12)$$

Equations (11) and (12) are taken into consideration. As a result, the following equation is obtained

$$p_m = p_1 + \varphi_1 \cdot K_{cyl} \cdot c_m \sqrt{\frac{K_{cyl}}{2 \cdot (K_{cyl} - 1) \cdot R \cdot T_{cyl}} \cdot \left[\left(\frac{p_m}{p_{cyl}} \right)^{2/K_{cyl}} - \left(\frac{p_m}{p_{cyl}} \right)^{(K_{cyl}+1)/K_{cyl}} \right]} \quad (13)$$

where φ_1 – speed loss coefficient.

The equation for the second and third grooves (m = 2, 3, $p_4 = p_H$) is the following:

$$\frac{dp_m}{d\tau} = a_m \cdot p_{m-1} \cdot \left[\overline{\mu f}_{m-1} \cdot k_{m-1} \cdot \psi\left(\frac{p_m}{p_{m-1}}\right) \cdot \frac{T_m}{\sqrt{T_{m-1}}} - k_m \cdot \psi\left(\frac{p_{m-1}}{p_m}\right) \cdot \left(\frac{p_m}{p_{m-1}}\right) \sqrt{T_m} \right] \tag{14}$$

where k_m is the adiabatic coefficient in m-th groove; T_m is gas temperature in m-th groove [K].

Then Formula (2) is the following:

$$\frac{dm}{d\tau} = \mu_3 \cdot f_3 \cdot \psi \cdot \left(\frac{p_H}{p_3}\right) \cdot p_3 \sqrt{\frac{1}{R \cdot T_3}} \tag{15}$$

where $\mu_3 \cdot f_3$ – Discharge coefficient and flow section over PR III [m^2]; p_3 and T_3 – Pressure [Pa] and gas temperature in ring groove III [K].

Pressure in ring groove II has the following by Formula (16) if $p_{cyl} < p_1 < p_2$ and $p_2 > p_3$ (Fig. 2). The Formula (16) is following:

$$\frac{dp_2}{d\tau} = -a_m \cdot k_2 \cdot p_2 \cdot \sqrt{T_2} \left[\overline{\mu f}_1 \cdot \psi\left(\frac{p_1}{p_2}\right) + \psi\left(\frac{p_3}{p}\right) \right] \tag{16}$$

where V_2 is a volume of ring groove II [m^3]; $\frac{dp_3}{d\tau}$ is determined by the Formula (4) with m = 3; V_3 – Volume of the ring groove III [m^3]

and

$p_1 = p_{cyl}$ $\qquad a_2 = \frac{\mu_2 \cdot f_2 \cdot \sqrt{R}}{V_2}$ $\qquad \overline{\mu f}_2 = \frac{\mu_2 f_2}{\mu_3 f_3}$

$a_3 = \frac{\mu_3 \cdot f_3 \cdot \sqrt{R}}{V_3}$ $\qquad \overline{\mu f}_3 = \frac{\mu_3 f_3}{\mu_1 f_1}$ $\qquad a_{23} = \frac{\mu_2 \cdot f_2 \cdot \sqrt{R}}{V_2 + V_3}$

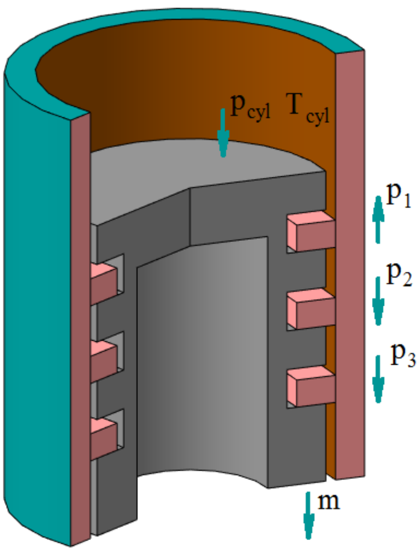


Fig. 2. Case of the positional relationship of the rings in the grooves of the piston, when $p_{cyl} < p_1 < p_2$ and $p_2 > p_3$

2.2. Case 2 of the positional relationship of the rings in the grooves of the piston, when $\sum P_1 > 0$, $\sum P_2 < 0$ and $\sum P_3 < 0$

The calculation of the blow-by if the sum of the forces acting on the ring I is positive and on the rings II and III is negative (Fig. 3, a) is the following:

$\sum P_1 > 0$, $\sum P_2 < 0$, $\sum P_3 < 0$ with $p_1 > p_2 > p_3$.

Then

$\frac{dp_1}{d\tau}$ is determined by the Formula (8);

$\frac{dp_2}{d\tau}$ is determined by the Formula (14) with $m = 2$;

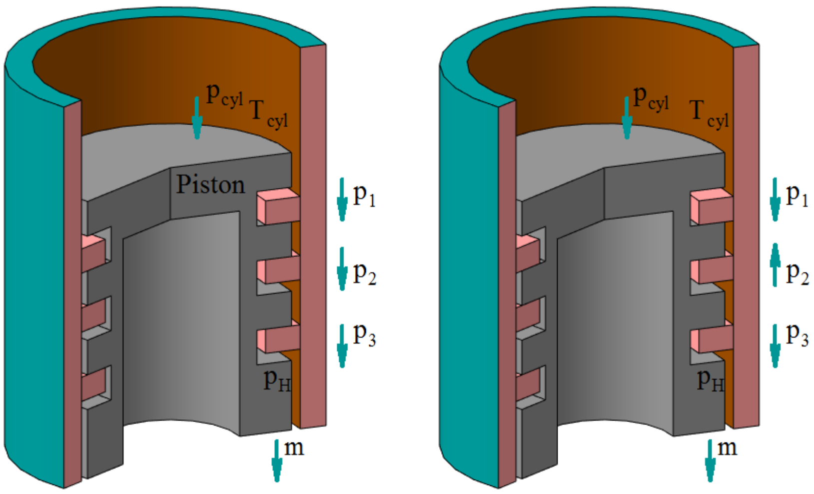
$p_3 = p_{cyl}$.

When $p_1 < p_2 > p_3$ (Fig. 3, b):

$$\frac{dp_2}{d\bar{\tau}} = -a_2 \cdot K_2 \cdot p_2 \cdot \left[\psi \cdot \left(\frac{p_1}{p_2} \right) + \frac{1}{\overline{\mu \cdot f_2}} \psi \cdot \left(\frac{p_H}{p_3} \right) \right] \sqrt{T_2} \tag{17}$$

where p_2 is gas pressure in the in the ring groove II [Pa]

$$a_2 = \frac{\mu_2 \cdot f_2 \cdot \sqrt{R}}{V_2} \qquad \overline{\mu \cdot f}_2 = \frac{\mu_2 \cdot f_2}{\mu_3 \cdot f_3}$$



a) b)

Fig. 3. Case 2 of the positional relationship of the rings in the grooves of the piston, when: a) $\sum P_1 > 0$, $\sum P_2 < 0$ and $\sum P_3 < 0$; b) $p_1 < p_2 > p_3$

Then Formula (2) is the following:

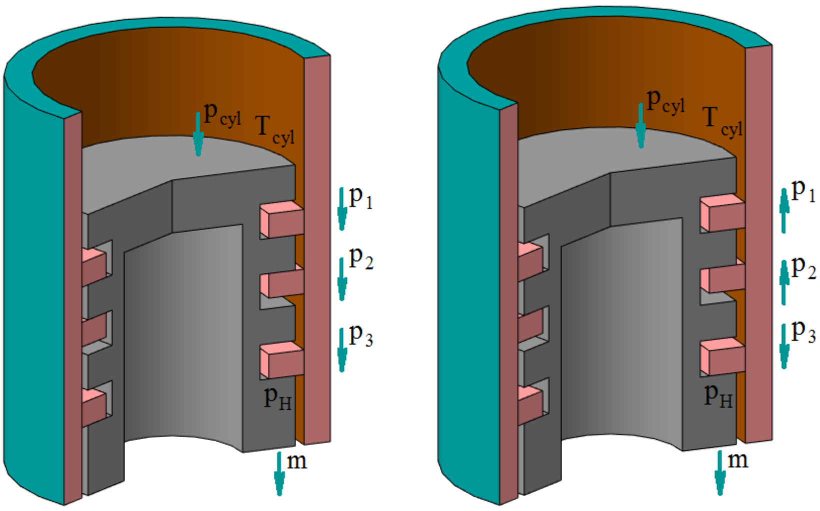
$$\frac{dm}{d\tau} = \mu_3 \cdot f_3 \cdot \psi \cdot \left(\frac{p_H}{p_2} \right) \cdot p_2 \sqrt{\frac{1}{R \cdot T_2}} \tag{18}$$

where T_2 is gas temperature in the ring groove II [K].

2.3. Case 3 of the positional relationship of the rings in the grooves of the piston, when $\sum P_1 > 0$, $\sum P_2 < 0$ and $\sum P_3 > 0$

The calculation of the blow-by if the sum of the forces acting on the ring I and III is positive and on the ring II is negative (Fig. 4, a) is the following:

$\sum P_1 > 0$, $\sum P_2 < 0$ and $\sum P_3 > 0$ with $p_1 > p_2 = p_3$.



a) b)

Fig. 4. Case 3 of the positional relationship of the rings in the grooves of the piston, when: a) $\sum P_1 > 0$, $\sum P_2 < 0$ and $\sum P_3 > 0$; b) $p_2 = p_3 > p_1$

Then $\frac{dp_1}{d\tau}$ is determined by the Formula (8)

$$\frac{dp_2}{d\bar{\tau}} = \frac{dp_3}{d\bar{\tau}} = a_{23} \cdot K_2 \cdot p_1 \cdot \left[\frac{\psi \cdot \left(\frac{p_2}{p_1}\right) \cdot T_2}{\sqrt{T_1}} - \frac{1}{\mu \cdot f_2} \psi \cdot \left(\frac{p_H}{p_3}\right) \cdot \frac{p_2}{p_1} \cdot \frac{p_3}{p_1} \sqrt{T_2} \right] \tag{19}$$

where $a_{23} = \frac{\mu_2 \cdot f_2 \cdot \sqrt{R}}{V_1 + V_3}$

When $p_2 = p_3 > p_1$ and $p_1 = p_{cyl}$ (see Fig. 4, b):

$$\frac{dp_2}{d\tau} = \frac{dp_3}{d\tau} = -a_{23} \cdot k_2 \cdot p_2 \left[\psi\left(\frac{p_1}{p_2}\right) + \frac{1}{\overline{\mu f}_2} \cdot \psi\left(\frac{p_H}{p_3}\right) \right] \sqrt{T_2} \tag{20}$$

Then Formula (2) is the following:

$$\frac{dm}{d\tau} = \mu_3 \cdot f_3 \cdot \psi \cdot \left(\frac{p_H}{p_3}\right) \cdot p_3 \sqrt{\frac{1}{R \cdot T_3}} \tag{21}$$

2.4. Case 4 of the positional relationship of the rings in the grooves of the piston, when $\sum P_1 < 0$, $\sum P_2 < 0$ and $\sum P_3 > 0$

The calculation of the blow-by if the sum of the forces acting on the ring I and II is negative and on the ring III is positive is the following (Fig. 5):

$\sum P_1 < 0$, $\sum P_2 < 0$ and $\sum P_3 > 0$ with $p_1 < p_2 > p_3$.

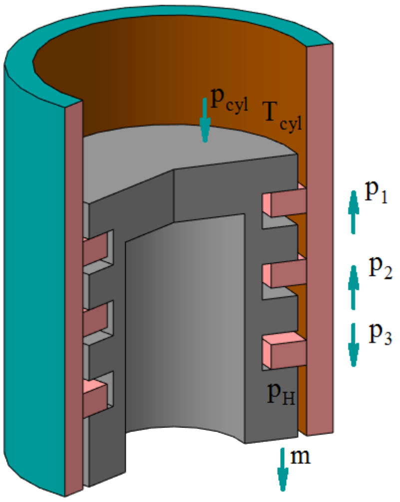


Fig. 5. Case 4 of the positional relationship of the rings in the grooves of the piston, when $\sum P_1 < 0$, $\sum P_2 < 0$ and $\sum P_3 > 0$

Then

$$\frac{dm}{d\tau} = \mu_3 \cdot f_3 \cdot \psi \cdot \left(\frac{p_H}{p_3}\right) \cdot p_3 \sqrt{\frac{1}{R \cdot T_3}} \tag{22}$$

2.5. Case 5 of the positional relationship of the rings in the grooves of the piston, when $\sum P_1 < 0$, $\sum P_2 < 0$ and $\sum P_3 > 0$ with $p_1 < p_2 > p_3$ and $p_1 = p_{cyl}$

The calculation of the blow-by if $\sum P_1 < 0$, $\sum P_2 < 0$ and $\sum P_3 > 0$ with $p_1 < p_2 > p_3$ and $p_1 = p_{cyl}$ (Fig. 6) is the following.

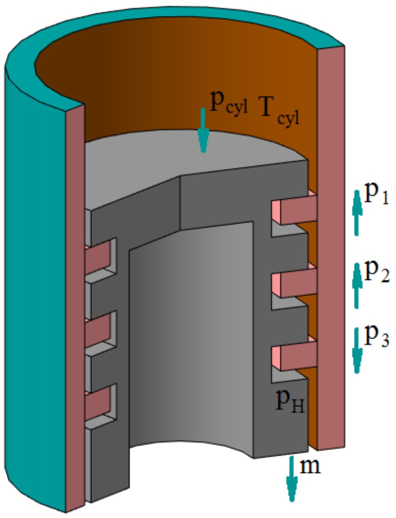


Fig. 6. Case 5 of the positional relationship of the rings in the grooves of the piston, when $\sum P_1 < 0$, $\sum P_2 < 0$ and $\sum P_3 > 0$ with $p_1 < p_2 > p_3$ and $p_1 = p_{cyl}$

Then $\frac{dp_2}{d\tau}$ is determined by the Formula (15) with $a_{23} = a_2$; $p_H = p_3$ and $p_3 = p_2$.

2.6. Case 6 of the positional relationship of the rings in the grooves of the piston, when $\sum P_1 < 0$, $\sum P_2 > 0$ and $\sum P_3 > 0$ with $p_1 < p_2 > p_3$, $p_1 = p_{cyl}$ and $p_2 = p_1$

The calculation of the blow-by if $\sum P_1 < 0$, $\sum P_2 > 0$ and $\sum P_3 > 0$ with $p_1 < p_2 > p_3$, $p_1 = p_{cyl}$ and $p_2 = p_1$ (Fig. 7) is the following.

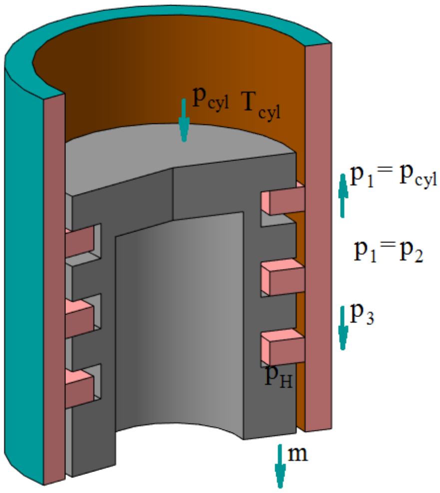


Fig. 7. Case 6 of the positional relationship of the rings in the grooves of the piston, when $\sum P_1 < 0$, $\sum P_2 < 0$ and $\sum P_3 > 0$ with $p_1 < p_2 > p_3$, $p_1 = p_{cyl}$ and $p_2 = p_1$

Then $\frac{dp_3}{d\tau}$ is determined by the Formula (14) with m = 3, $p_4 = p_H$.

For further analysis, it is convenient to consider the relative magnitude of gas escapes:

$$\bar{m}(\varphi) = \frac{m}{m_{cyl}} \tag{23}$$

where m is current gas escape in the crank angle [kg]; m_{cyl} is total gas escape per cycle [kg].

3. THE RESULTS OF THE CALCULATIONS

The dependencies $\bar{m}(\varphi)$ and m_{cyl} for the diesel with the main initial data are shown in Fig. 8 and 9:

- engine power Ne = 155 kW
- engine speed n = 2600 min^{-1}
- value of flow section $\mu_2 \cdot f_2 = \mu_3 \cdot f_3$ = 0.3·10^{-6} m^2
- volume on lands and the piston ring grooves: $V_2 = V_3$ = 1.73·10^{-6} m^3

The initial and boundary conditions were set according to the results of indexing and thermometry of the diesel engine at the rated duty; discharge coefficient μ =0.85.

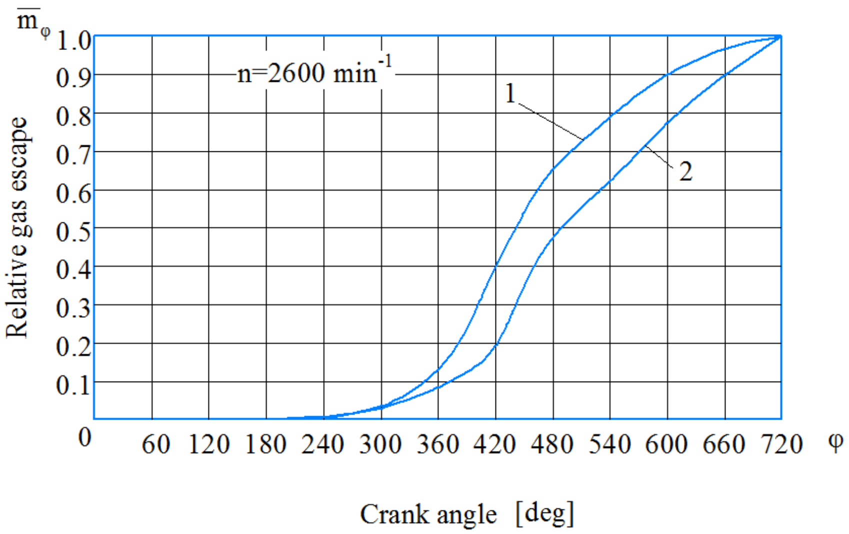


Fig. 8. Changes of the relative gas escape $\overline{m}(\varphi)$ depending on the crank angle (n = 2600 min^{-1}) subject to: 1 – the movement of the rings in the grooves; 2 – the fixed rings

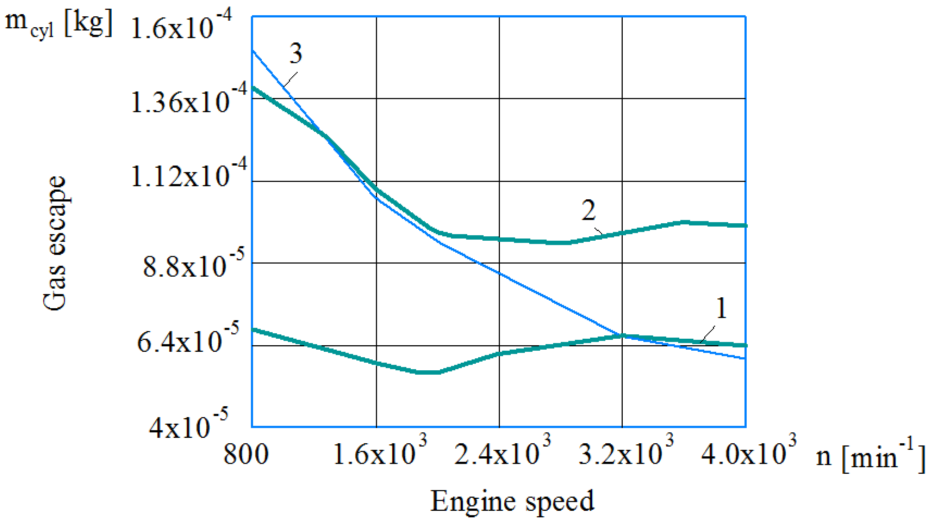


Fig. 9. Changes of the gas escape m_{cyl} depending on the engine speed: 1 – idling conditions; 2 – load conditions; 3 – the fixed rings

Movements of the rings in the grooves noticeably affect the gas escape into the crankcase, especially when the pressure in the cylinders is the biggest, that is, at 360-660° crank angle (Fig. 8). After 660° crank angle, the difference in gas escape values can be neglected.

The analysis of dependences $m_{cyl} = f(n)$ confirms the significant influence of the dynamics of rings on the course of cyclic gas escape curves (Fig. 9). The deterioration of the sealing properties of the piston rings with increasing the engine speed was noted earlier in other studies [21,22].

This fact especially manifests itself in idling modes due to the reduced pressure in the combustion chamber.

Thus, according to the gas escape flow nature dependence on the engine speed, it is possible to assess the compression ability of the diesel ring seal, the quality operation of the piston rings. This enables the determining of the technical condition of the CPG parts which is especially important for mining industry trucks.

4. CONCLUSIONS

1. The effect of the axial movement of the piston rings in the piston grooves on the blow-by in the ICE was established. This creates prerequisites for a more accurate assessment of their sealing ability and the search for ways to further improve them.
2. The calculated dependences to compute the blow-by depending on the positional relationship of the rings in the piston grooves was obtained
3. The dependencies of the gas escapes on the engine speed allow judging upon the dynamic stability of the ring seal and solving the issues of evaluating the technical condition of the CPG parts and improving their operational properties, especially for idling modes.

Acknowledgements

The authors would like to express their gratitude to the referees for their constructive and valuable suggestions.

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Received 19.05.2019; accepted in revised form 11.08.2019