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Dynamic analysis of gas flow through the ICE ring seal

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Abstract: (1) Purpose: The effect of axial movement of piston rings in the piston grooves is estimated by calculation and experimentally for the passage of gases in an internal combustion engine (ICE). This contributes to the development of practical recommendations for the further improvement of engine ring seal designs; (2) Methodology: When modeling the effect of axial movement of piston rings in the piston grooves for the passage of gases in the ICE, theoretical positions are used. It is based on the fundamental theory of heat engines, thermodynamics and hydraulics. Running was investigated using theoretical and theoretical research methods; (3) Results: The effect of axial movement of piston rings in the piston grooves on the passage of gases in the ICE is established. This creates prerequisites for a more accurate assessment of their sealing ability and the search for ways to further improve them. Calculated dependences are obtained for calculating the pass of gases depending on the relative position of the rings in the piston grooves. The dependences of gas escapes on the engine crankshaft rotation speed are 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 operational properties; (5) Practical implications: The practical method for estimating the dynamic stability of an annular seal based on gas escape dependencies on the crankshaft rotation speed in ICE is 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 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 [1-5]:

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 and

c) Improving the quality of used materials.

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

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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 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 its turn, increases the gas escape. This furthers the seizing of the piston rings, the increasing of oil consumption and fuel, and smoking. The final result is jamming and the engine trouble.

A wide variety of factors influencing the operation of the PR complicates analysis and generalization 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 should be based on the account of the totality of the main phenomena that determine the operability of CPG and PR parts.

Numerous papers cover experimental and theoretical study of the PR operation. In [6] 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 visualize 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 [7].

The diagnostic methodology can effectively determine the control of the condition of the PR in accordance with the characteristics of combustion [8].

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 are carried out on the assumption of forces balance [9].

Methods and devices to study mechanical friction losses are developed [10]. A simplified floating liner method is used and the test equipment is developed to fill the gap in between the full floating liner engine and the typical component bench test equipment.

The purpose of research [11] is to study the potential of the laser oil pockets new design to improve the piston rings lubrication. These pockets make it possible to achieve significant friction reduction by using appropriate geometric parameters.

At the moment 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 considered sufficiently. In particular, this is the PR movement in the piston grooves. This is connected with the engine running where all piston rings moving is 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. Materials and methods

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 and

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 papers.

The principal features of the adopted model are 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 equation:

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

here $\mu \cdot f$ – 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 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 (Figure 1) is the following: $\sum P_1 > 0$, $\sum P_2 > 0$ and $\sum P_3 > 0$ with $p_1 > p_2 > p_3$.

Then

$$\frac{dp_1}{d\bar{\tau}} = a_1 \cdot p_{cyl} \times \left[\overline{\mu \cdot f_0} \cdot \psi \cdot \left(\frac{p_m}{p_{cyl}} \right) \cdot \frac{k_{cyl}}{\beta_1} \sqrt{T_{cyl}} - k_1 \cdot \psi \cdot \left(\frac{p_2}{p_1} \right) \cdot \frac{p_1}{p_{cyl}} \cdot \sqrt{T_1} - \frac{\alpha \cdot F_1 \cdot \Delta t_1 \sqrt{R}}{\overline{\mu \cdot f_1} \cdot \beta_1 \cdot c_{cyl} \cdot p_{cyl}} \right] \tag{3}$$

here V_1 – volume of ring groove I [m^3]; t_n, t_w – average temperature of the piston head [K]; t_f – 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]; 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]; α – heat exchange coefficient from gases to surfaces of the cylinder and the piston [$\frac{W}{m^2 \cdot K}$]; 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 $[\frac{J}{kg \cdot K}]$; $a_1 = \mu_1 \cdot f_1 \sqrt{R_T} / V_1$; $\overline{\mu \cdot f}_0 = \frac{\mu_0 \cdot f_0}{\mu_1 \cdot f_1}$; $\overline{\mu \cdot f}_1 = \frac{\mu_1 \cdot f_1}{\mu_2 \cdot f_2}$; $\beta_1 = C_{V_1} / C_{V_{cyl}}$; $\Delta t_1 = 2 \cdot t_f - t_w - t_n$ and $t_f = \frac{t_{cyl} + t_1}{2}$.

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

$$\frac{dp_m}{d\tau} = a_m \cdot p_{m-1} \times \left[\overline{\mu f_{m-1}} \cdot k_{m-1} \cdot \psi\left(\frac{p_m}{p_{m-1}}\right) \cdot \frac{T_m}{\sqrt{T_{m-1}}} - k_m \cdot \psi\left(\frac{p_{m-1}}{p_m}\right) \cdot \left(\frac{p_m}{p_{m-1}}\right) \sqrt{T_m} \right] \tag{4}$$

here k_m – the adiabatic coefficient in *m*-th groove; T_m – gas temperature in *m*-th groove [K].

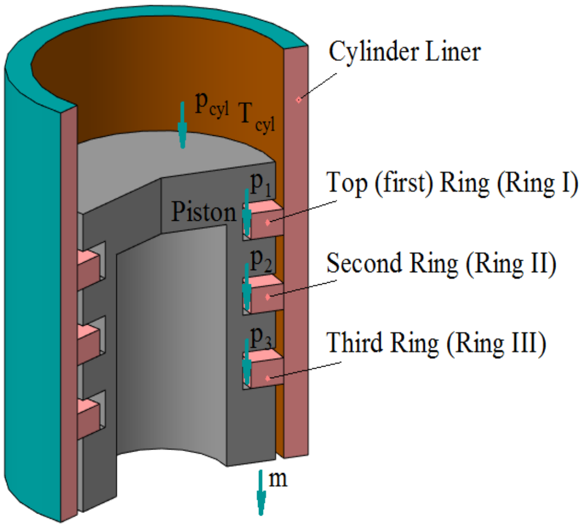


Figure 1. Case 1 of the positional relationship of the rings in the grooves of the piston, when $\sum P_1 > 0$, $\sum P_2 > 0$ and $\sum P_3 > 0$: m – mass gas escape; p_1, p_2, p_3 – gas pressure in annular piston cavities I, II and III.

Then Equation (2) is the following:

$$\frac{dm}{d\tau} = \mu_3 \cdot f_3 \cdot \psi \cdot \left(\frac{p_H}{p_3}\right) \cdot p_3 \sqrt{\frac{1}{R \cdot T_3}} \tag{5}$$

here $\mu_3 \cdot f_3$ – Discharge coefficient and flow section over PR III [m²]; p_3 and T_3 – pressure [Pa] and gas temperature in ring groove III [K].

Pressure in ring groove II has the following by Equation (6) if $p_{cyl} < p_1 < p_2$ and $p_2 > p_3$ (Fig. 2).

The Equation (6) is following:

$$\frac{dp_2}{d\tau} = -a_m \cdot k_2 \cdot p_2 \cdot \sqrt{T_2} \left[\overline{\mu f}_1 \cdot \psi\left(\frac{p_1}{p_2}\right) + \psi\left(\frac{p_3}{p}\right) \right] \tag{6}$$

here V_2 is a volume of ring groove II [m3]; $\frac{dp_3}{d\tau}$ is determined by the Eq. (4) with m = 3; V_3 – Volume of the ring groove III [m3]; $p_1 = p_{cyl}$; $a_2 = \frac{\mu_2 \cdot f_2 \cdot \sqrt{R}}{V_2}$; $\overline{\mu f_2} = \frac{\mu_2 f_2}{\mu_3 f_3}$; $a_3 = \frac{\mu_3 \cdot f_3 \cdot \sqrt{R}}{V_3}$; $\overline{\mu f_3} = \frac{\mu_3 f_3}{\mu_1 f_1}$ and $a_{23} = \frac{\mu_2 \cdot f_2 \cdot \sqrt{R}}{V_2 + V_3}$.

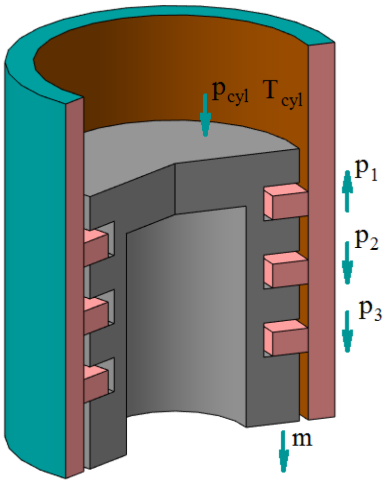


Figure 2. Case of the positional relationship of the rings in the grooves of the piston, when $p_{cyl} < p_1 < p_2$ and $p_2 > p_3$.

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 $p_1 > p_2 > p_3$.

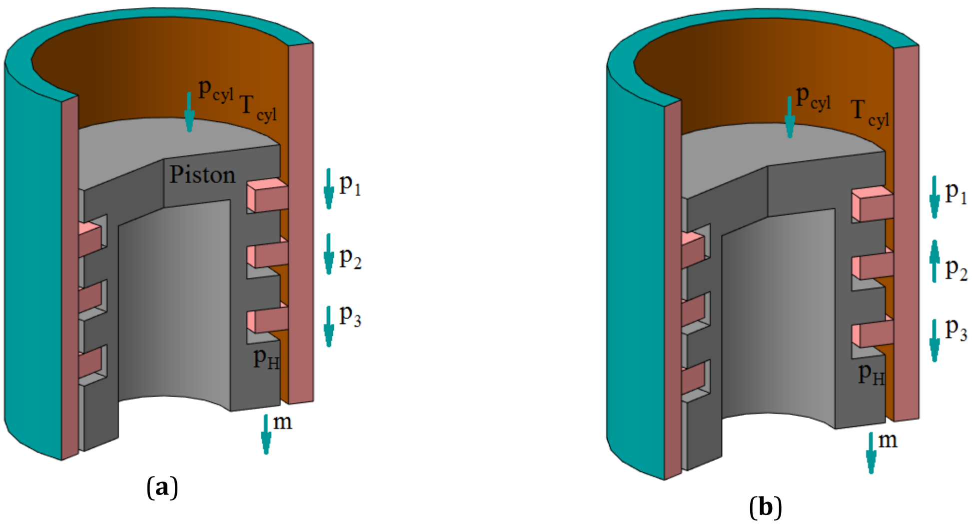


Figure 3. Case 2 of the positional relationship of the rings in the grooves of the piston, when: (**a**) $\sum P_1 > 0$, $\sum P_2 < 0$ and $\sum P_3 < 0$; (**b**) $p_1 < p_2 > p_3$.

Then $\dfrac{dp_1}{d\tau}$ is determined by the Equation (3); $\dfrac{dp_2}{d\tau}$ is determined by the Equation (4) with m = 2 and $p_3 = p_{cyl}$.

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

$$\frac{dp_2}{d\bar{\tau}} = -a_2 \cdot K_2 \cdot p_2 \cdot \left[\psi \cdot \left(\frac{p_1}{p_2} \right) + \frac{1}{\overline{\mu \cdot f_2}} \psi \cdot \left(\frac{p_H}{p_3} \right) \right] \sqrt{T_2} \tag{7}$$

here p_2 – gas pressure in the in the ring groove II [Pa]; $a_2 = \dfrac{\mu_2 \cdot f_2 \cdot \sqrt{R}}{V_2}$ and $\overline{\mu \cdot f}_2 = \dfrac{\mu_2 \cdot f_2}{\mu_3 \cdot f_3}$.

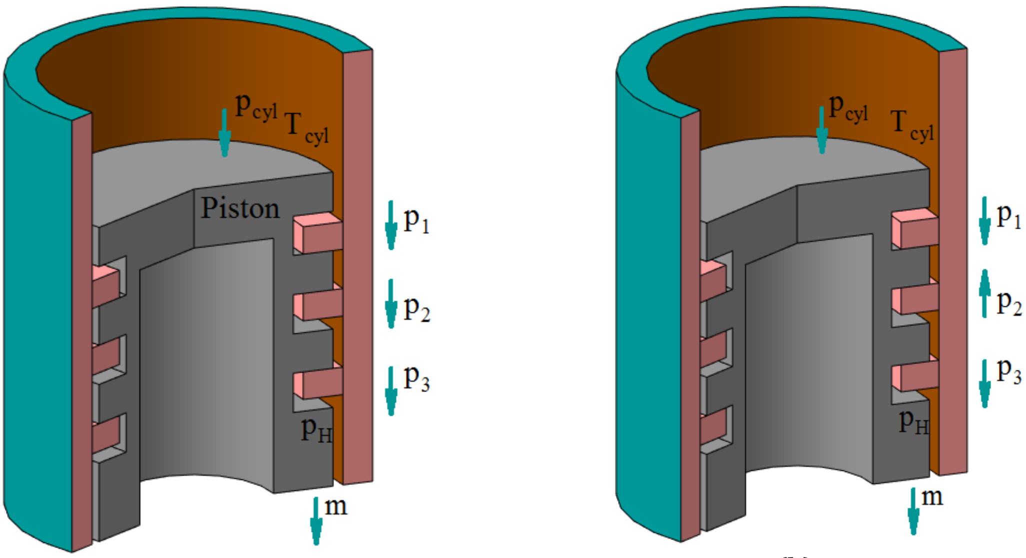
Then Eq. (2) is the following:

$$\frac{dm}{d\tau} = \mu_3 \cdot f_3 \cdot \psi \cdot \left(\frac{p_H}{p_2} \right) \cdot p_2 \sqrt{\frac{1}{R \cdot T_2}} \tag{8}$$

here 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)

Figure 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 $\dfrac{dp_1}{d\tau}$ is determined by the Eq. (3)

$$\frac{dp_2}{d\tau}=\frac{dp_3}{d\tau}=a_{23}\cdot K_2\cdot p_1\cdot\left[\frac{\psi\cdot\left(\frac{p_2}{p_1}\right)\cdot T_2}{\sqrt{T_1}}-\frac{1}{\mu\cdot f_2}\psi\cdot\left(\frac{p_H}{p_3}\right)\cdot\frac{p_2}{p_1}\cdot\frac{p_3}{p_1}\sqrt{T_2}\right] \quad (9)$$

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):

$$\frac{dp_2}{d\tau}=\frac{dp_3}{d\tau}=-a_{23}\cdot k_2\cdot p_2\left[\psi\left(\frac{p_1}{p_2}\right)+\frac{1}{\overline{\mu f}_2}\cdot\psi\left(\frac{p_H}{p_3}\right)\right]\sqrt{T_2} \quad (10)$$

Then Equation (2) is the following:

$$\frac{dm}{d\tau}=\mu_3\cdot f_3\cdot\psi\cdot\left(\frac{p_H}{p_3}\right)\cdot p_3\sqrt{\frac{1}{R\cdot T_3}} \quad (11)$$

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 (Figure 5): $\sum P_1<0$, $\sum P_2<0$ and $\sum P_3>0$ with $p_1 < p_2 > p_3$. Then Equation (11) is considered.

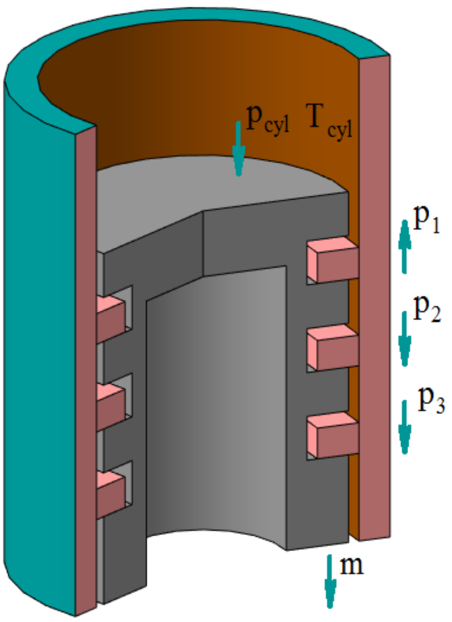


Figure 5. Case 4 of the positional relationship of the rings in the grooves of the piston, when $\sum P_1<0$, $\sum P_2<0$ and $\sum P_3>0$.

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 $p_1 = p_{cyl}$ is the following Fig. 6.

Then $\frac{dp_2}{d\tau}$ is determined by the Equation (5) with $a_{23} = a_2$; $p_H = p_3$ and $p_3 = p_2$.

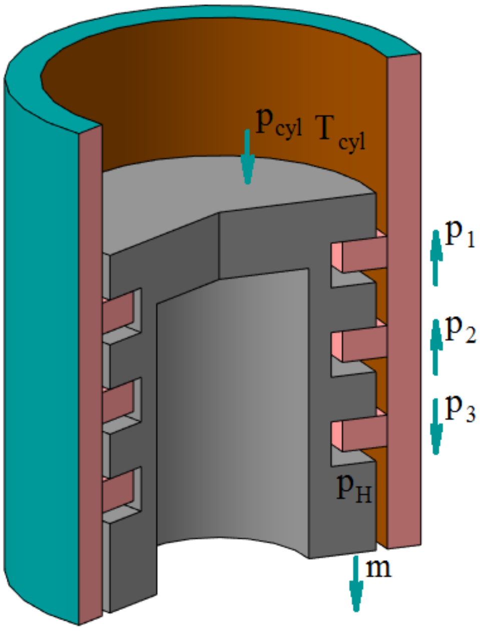


Figure 6. Case 5 of the positional relationship of the rings in the grooves of the piston, when $\sum P_1 < 0$, $\sum P_2 < 0$ and $\sum P_3 > 0$ with $p_1 < p_2 > p_3$ and $p_1 = p_{cyl}$.

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 $p_2 = p_1$ is the following Figure 7.

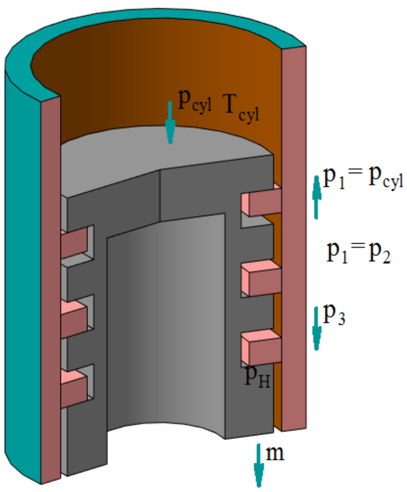


Figure 7. Case 6 of the positional relationship of the rings in the grooves of the piston, when $\sum P_1 < 0$, $\sum P_2 < 0$ and $\sum P_3 > 0$ with $p_1 < p_2 > p_3$, $p_1 = p_{cyl}$ and $p_2 = p_1$.

Then $\frac{dp_3}{d\tau}$ is determined by the Equation (4) with $m = 3$, $p_4 = p_H$.

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

$$\overline{m}(\varphi) = \frac{m}{m_{cyl}} \tag{12}$$

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

3. Results

The dependencies $\overline{m}(\varphi)$ 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 $\mu_2 \cdot f_2 = \mu_3 \cdot f_3$ = $0.3 \cdot 10^{-6}$ m^2;
- volume on lands and the piston ring grooves: $V_2 = V_3$ = $1.73 \cdot 10^{-6}$ m^3.

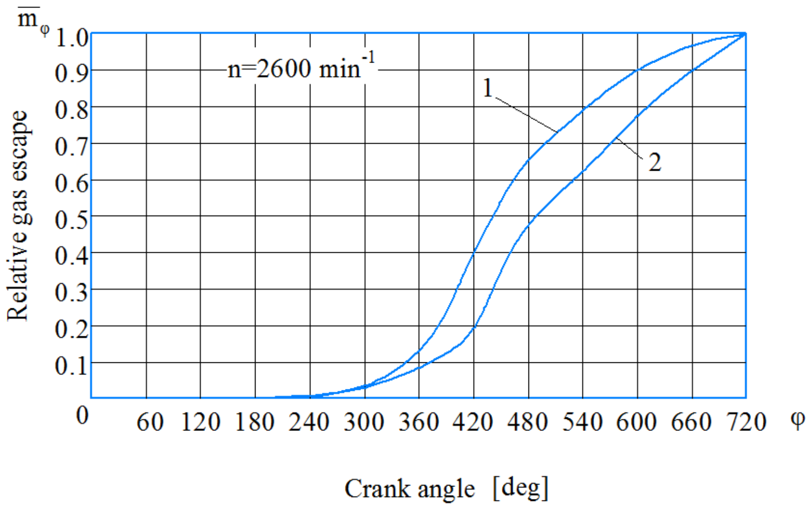


Figure 8. Changes of the relative gas escape $\overline{m}(\varphi)$ depending on the crank angle (n = 2600 min^{-1}) subject to: 1 – the movement of the rings in the grooves; 2 – the fixed rings.

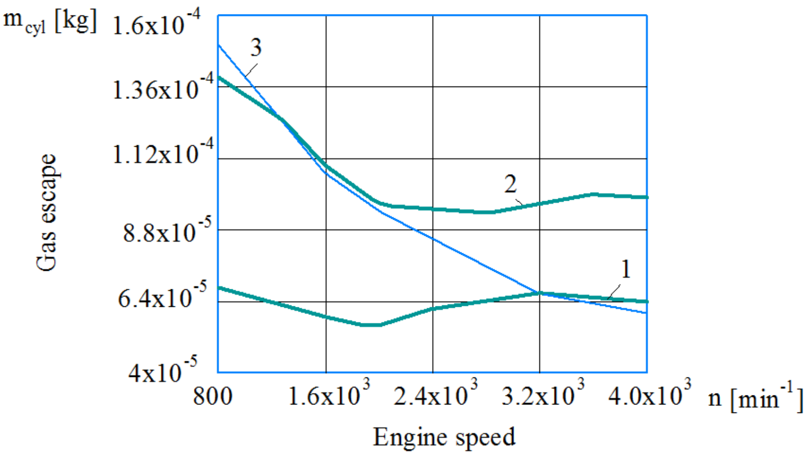


Figure 9. Changes of the gas escape m_{cyl} depending on the engine speed: 1 – idling conditions; 2 – load conditions; 3 – the fixed rings.

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.

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 (see Figure 8). After 660° crank angle, the difference in gas escape values can be neglected.

4. Discussion

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 (see Fig. 9). The deterioration of the sealing properties of the piston rings with increasing the engine speed was noted earlier in other studies [12-13].

This fact especially manifests itself in the 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 allows determining the technical condition of the CPG parts which is especially important for mining industry trucks..

5. Conclusions

1. The effect of the axial movement of the piston rings in the piston grooves on the blow-by in the ICE is established. This creates prerequisites for a more accurate assessment of their sealing ability and the search for the ways to further improve them.

2. The calculated dependences to calculate the blow-by depending on the positional relationship of the rings in the piston grooves are 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.

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