Article citation info:
WRÓBLEWSKI, P. The effect of the distribution of variable characteristics determining the asymmetry of the sealing rings sliding surfaces on the values of friction loss coefficients and other selected parameters of oil film. *Combustion Engines*. 2017, **171**(4), 107-116. DOI: 10.19206/CE-2017-418

Piotr WRÓBLEWSKI

CE-2017-418

The effect of the distribution of variable characteristics determining the asymmetry of the sealing rings sliding surfaces on the values of friction loss coefficients and other selected parameters of oil film

The article presents the results of research on the effect of the change in coordinates that determine the location of the common vertex of the parabolas and ellipses demonstrating the asymmetric shape of the rings sliding surfaces on the change in the thickness of oil film, the amount of scraped oil, and the oil left on the cylinder wall, once a set of selected sealing rings has passed, as well as the results concerning friction losses. Mathematical relationships of the asymmetric shapes of the rings sliding surfaces were presented, and the values of coefficients characterizing selected parameters of oil film and friction losses were determined. It was confirmed that there is a large dependence of the coordinate values of the vertices determining the asymmetric shapes of these surfaces on the condition and cooperation between piston rings and the cylinder bearing surface, and mainly on the amount of the oil scraped in the selected engine strokes.

Key words*: rings asymmetrical sliding surface, friction loss, oil film*

1. Introduction

Currently, sealing rings of a symmetrical barrel-shape sliding surface are most commonly used in the design of piston internal combustion engines [5, 6]. The continuous development of internal combustion engines aims at reducing the emission of toxic substances emitted into the environment by increasing the engine thermal and mechanical efficiency. In this case, it is important to reduce friction losses in the set of sealing rings, while ensuring the oil film continuity between the cooperating surfaces of kinematic pairs throughout the entire engine cycle [1]. An important structural factor for reducing friction loss in the piston ring set is the selection of the sliding surfaces of the lower and upper sealing rings [7]. The previous studies concerning modeling parameters for the piston-cylinder group have not paid major attention to modeling the shape of the sliding surface of a piston [2–5]. In most models, the authors accept a symmetrical shape of the sliding surfaces of sealing rings as the boundary condition. Therefore, there is then a justified need to examine the possibility of reducing friction loss by applying the asymmetric shape of these surfaces. The most important task is to determine the appropriate shape of the upper sealing ring, due to its specific working conditions, especially at the reduced thickness of the oil film just before TDC in the compression stroke and just after the TDC in the expansion stroke [7]. Each transition from the symmetrical shape of these surfaces to an asymmetric shape is related to a change in pressure distribution in the oil film throughout the entire engine cycle. The main purpose of this article is to present the influence of the distribution of variable characteristics determining the selected shapes of sealing rings on friction loss and the selected parameters of the oil film.

The pressure distribution in the oil film is also affected by the piston ring structural conditions, i.e. the force of elasticity, resulting from the type of material used, the geometrical parameters of the ring, the permeability and thermal expansion of the material, and the operating conditions of the engine, which results directly in the value of the gas force in a given period of engine work. Most of these parameters can be assumed theoretically, and then verified experimentally while designing the sealing rings. It is much more difficult to determine the distribution coverage of the sliding surfaces of sealing rings with the oil film in each engine stroke while taking into account the variable arrangement of the sealing rings and the scraper ring in the piston grooves, local changes in the temperatures of oil and materials, changes in oil viscosity and elasticity, and engine angular velocity [3, 4]. It is of particular importance to determine the oil film distribution between the sealing ring surfaces at low angular velocities and the low temperatures of lubricating oil [2]. Therefore, it is necessary to verify the influence of these parameters on the oil film coverage of the asymmetric shapes of the sealing rings sliding surfaces while taking into account the variable position of these surfaces due to the rings' rotation in the piston grooves. Such parameters strongly influence the pressure ratio resulting from extrusion and slip. Anticipating the effects of applying the asymmetry of the rings sliding surfaces for selected conditions of engine work and its design in terms of the functional needs of the lower and upper sealing rings may contribute to a significant reduction in friction losses. Unfortunately, the precise setting of all boundary conditions requires that a simulation be conducted for each selected engine design. However, the formation of a database or mathematical model determining the results of such adjustments in the ring shape under certain engine operating conditions and its structural parameters can greatly facilitate the engine design and construction work. The shape of sealing rings sliding surfaces directly influences the extrusion and slip effects, that is the amount of oil scraped into the next ring or the combustion chamber in the compression and exhaust strokes and the instantaneous thickness of the oil layer left on the cylinder bearing surface once the given ring has passed. In addition, the mathematical part of the author's model introduced to the model developed by A. Iskra can greatly accelerate the verification of the effects of the changes in the asymmetric shapes of the sealing rings on the distribution of the oil film on the cooperating surfaces, minimum thickness of oil film, friction loss and lubricant oil consumption.

2. Distribution of variable characteristics determining the asymmetry of parabolic and elliptical shapes of the sealing rings sliding surfaces

Expressing the effect of the changes in asymmetry of parabolic and elliptical shapes of the sealing rings sliding surfaces requires the formation of appropriate equations describing the distribution of variable features in the Cartesian coordinate system. Next, the coordinates of the points that are within specific ranges are introduced into the mathematical model. The ranges are sections of parabolas or ellipses with common coordinates of vertices or edge points coordinates. The mathematical presentation of relationships of shape asymmetry enables the expression of the effects of changes in the vertex coordinates of a set of parabolas or ellipses on the pressure distribution in oil film, and thus as a result of the friction loss, the parameters determining the oil film distribution on the cooperating surfaces and the lubricating oil consumption. An important feature of such mathematical presentation of these shapes and the adoption of certain coefficients for a given combination of shapes and their asymmetry is the ability to determine the influence of the change in coordinates of the common vertex on the most important parameters of the oil film distribution. Such modeling approach enables the selection of asymmetry of surface shapes for the required conditions of cooperating surfaces resulting from individual engine characteristics. It should be noted that the results of the illustrated simulations are unique for each given engine and are not a general indicator for all engines. By using the current model to approximate the results with polynomial functions of higher degrees or by interpolation, one can generate a significant error in certain engine areas, especially when there is a shift in the common peak of the parabola or ellipse expressing the given shape. In these areas of the engine cycle and for shapes with a large peak shift, an increased number of computational steps should be applied by generating a large number of co-ordinates describing the ring shape. Based on these assumptions, the change in friction loss and oil film parameters can be calculated for any engine design with a modified set of sealing rings.

Figure 1 shows the way of presenting asymmetric geometry of parabolic and elliptical shapes, and a diagram of velocity and pressure fields at the starting and separating points of oil film for two sealing rings.

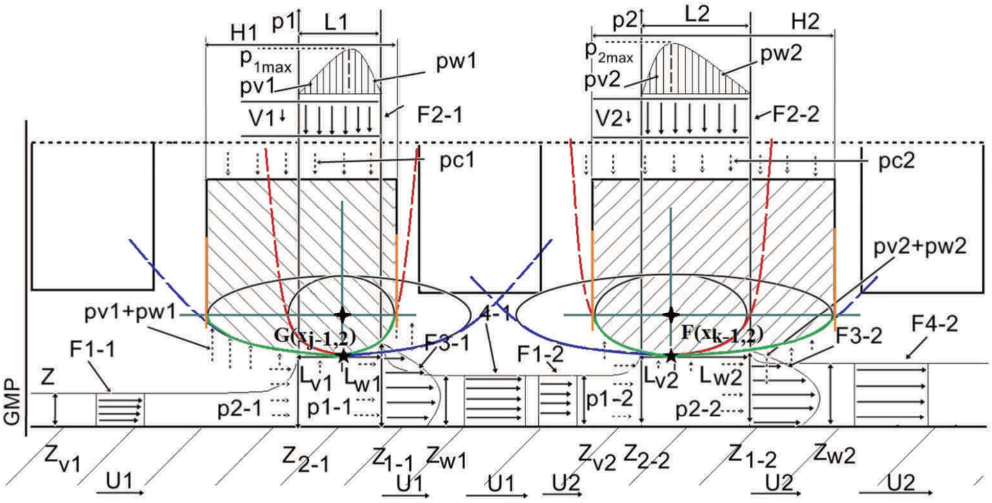

Fig. 1. Schematic representation of asymmetric geometry of parabolic and elliptical sliding surfaces, as well as velocity and pressure fields at the starting and separating points of the oil film for the lower and upper sealing rings where: ★ – $W_{j,k\,1-2}$ vertex coordinates of the parabolas and ellipses defining the given asymmetric shape of the sliding surface, ♦ – coordinates of the ellipses common center, green line – function expressing the asymmetrical shape of the sealing rings sliding surfaces

The presented mathematical model for which the corresponding mathematical relationships were selected and the results of the simulation do not take into account the changes in reference points coordinates, which represent the roughness and defects of the material structures existing on the sealing rings sliding surfaces and the cylinder walls. However, knowing the value of roughness Ra, which is determined experimentally for the given surfaces by approximating the coordinates of micro-roughness vertices, one can implement it into the current model. The implementation of the above consists in introducing a large number of coordinates of points representing the given roughness of the opposing surfaces. However, such action requires a considerable number of computational steps, up to over a dozen thousand of them, which is due to the number of coordinates of the two planes. It should be noted that by implementing only the coordinates of the points representing the roughness of the surface in an approximated manner, it is not possible to accurately represent the exact conditions that occur at the time of mixed or boundary friction. Then the current mathematical model for calculating selected oil film parameters and friction loss must be extended by additional blocks of mathematical equations expressing the co-operation conditions when the oil film thickness is reduced below the minimum value. Such a mathematical model should also take into account the distance criterion of the individual coordinate points describing the shape and directional position of the two planes and, in the case of mixed friction conditions, it should implement mathematical equations describing the conditions of the contact of micro-roughness vertices while separating the necessary parameters.

The asymmetric shape of the sealing rings parabolic sliding surfaces is to be expressed by four functions: $g(x_{j1}), g(x_{j2}), f(x_{k1}), f(x_{k2})$ with common parabola vertices coordinates $\mathbf{W_{j1} = W_{j2}}$ and $\mathbf{W_{k1} = W_{k2}}$. The vertex coordinates W_{jn} and W_{kn} are the coordinates of the critical transition point of the experimental piston rings' actual shape, which affects the ratio of the extrusion and slip coefficients. The functions defining the character of asymmetrical parabolic shapes can be expressed by the following set of equations:

$$g(x_{j1}) = a_{1j1}(x_{j1} - p_{j1})^2 + q_{j1}$$

$$g(x_{j2}) = a_{2j2}(x_{j2} - p_{j2})^2 + q_{j2}$$

$$f(x_{k1}) = a_{1k1}(x_{k1} - p_{k1})^2 + q_{k1}$$

$$f(x_{k2}) = a_{2k2}(x_{k2} - p_{k2})^2 + q_{k2}$$

where: $(a_{1j1}, a_{2j2}, a_{1k1}, a_{2k2}) \neq 0; x_{j1}, x_{j2}, x_{k1}, x_{k2} \in \mathbf{R}$

$$\mathbf{W_{j1}} = (p_{j1}, q_{j1}); \mathbf{W_{j2}} = (p_{j2}, q_{j2})$$

$$\mathbf{W_{k1}} = (p_{k1}, q_{k1}); \mathbf{W_{k2}} = (p_{k1}, q_{k1})$$

where: $p_{j1} = p_{j2} \neq 0; p_{k1} = p_{k2} \neq 0; q_{j1} = q_{j2} \neq 0; q_{k1} = q_{k2} \neq 0$

$$x_{j1}, x_{j2} \in (m_{j1}, k_{j1}); x_{k1}, x_{k2} \in (m_{j2}, k_{j2})$$

thus

$$G(x_{j1,2}) = \begin{cases} a_{1j1}(x_{j1} - p_{j1})^2 + q_{j1} \text{ for } x_{j1} \in (m_{j1,2}, p_{j1,2}) \\ a_{2j2}(x_{j2} - p_{j2})^2 + q_{j2} \text{ for } x_{j2} \in (p_{j1,2},, k_{j1,2}) \end{cases}$$

$$F(x_{k1,2}) = \begin{cases} a_{1k1}(x_{k1} - p_{k1})^2 + q_{k1} \text{ for } x_{k1} \in (m_{k1,2}, p_{k1,2}) \\ a_{2k2}(x_{k2} - p_{k2})^2 + q_{k2} \text{ for } x_{k2} \in (p_{k1,2},, k_{k1,2}) \end{cases}$$

where: $G(x_{j1,2})$, $F(x_{k1,2})$ – trigonometric functions in canonical form, expressing the asymmetrical parabolic shapes of the sliding surfaces of the upper and lower sealing rings – the green parabola.

Presenting the shapes of the rings' sliding surfaces by means of two parabolas with a common vertex W_{j1-2} enables to illustrate the influence of the vertex position on the friction loss and the selected oil film parameters. The selection of the number of coordinate points implemented into the simulation program from the two ranges of the set representing the selected shape of the sliding surface is of a unique character and depends on the required calculation accuracy. If one exclusively assumes the occurrence of liquid friction conditions, then a range of 50 to 200 coordinate points is considered sufficient for calculation. The number of coordinate calculation points depends on the complexity of the asymmetric shape of the sliding surfaces of the sealing rings and the axial height of the selected ring in the set. For the purpose of this simulation, it is assumed that H_1 = 1.50 mm and H_2 = 2.00 mm.

In order to determine the coordinates of any points defining a given shape, the following boundary conditions are assumed:

$$x_{j1-2} \in (m_{j1,2}, k_{j1,2}) \Leftrightarrow m_{j1,2} \leq x_{j1-2} \leq k_{j1,2}; m_{j1,2} = 1; k_{j1,2} = 50, x_{j1,2} \in \mathbb{N}_+$$

$$x_{k1-2} \in (m_{k1,2}, k_{k1,2}) \Leftrightarrow m_{k1,2} \leq x_{k1-2} \leq k_{k1,2}; m_{k1,2} = 1; k_{k1,2} = 50, x_{k1,2} \in \mathbb{N}_+$$

If $\mathbf{H_1 \neq H_2}$ then $|\mathbf{x_{j1,2}}| \neq |\mathbf{x_{k1,2}}|$

where:

$$|x_{j1,2(n+1)}x_{j1,2(n)}| = \sqrt{\left(x_{j1,2(n+1)} - x_{j1,2(n)}\right)^2 + \left(y_{j1,2(n+1)} - y_{j1,2(n)}\right)^2}$$

for $x_{j1} \in (m_{j1,2}, p_{j1,2}) \wedge x_{j2} \in (p_{j1,2}, k_{j1,2})$

$$|x_{k1,2(n+1)}x_{k1,2(n)}| = \sqrt{\left(x_{k1,2(n+1)} - x_{k1,2(n)}\right)^2 + \left(y_{k1,2(n+1)} - y_{k1,2(n)}\right)^2}$$

for $x_{k1} \in (m_{k1,2}, p_{k1,2}) \wedge x_{k2} \in (p_{k1,2}, k_{k1,2})$

On condition that: $H_1 \vee H_2 < 2.00$ mm, then $k_{j1,2} \wedge k_{k1,2} \geq 50$.

By accepting the specified vertex coordinates $W_{j1,2} \wedge W_{k1,2}$ and the values $m_{j1,2}, k_{j1,2} \wedge m_{k1,2}, k_{k1,2}$,one can map the actual asymmetric shape of the sliding surfaces of the rings. If H_1 = 1.50 mm, the actual vertex height W_{j-k} is 10 μm, distance p_{j1-2} from the lateral side of the sliding surface conventionally defined as $m_{j1,2}$ = 60% H_1 and $k_{j1,2} = 50$, then one takes $p_{j1-2} = 30$ and q_{j1-2} = –0.10 as coordinates $W_{j1,2}$. Based on the accepted boundary conditions, the remaining coordinates of points from the range $x_{j1,2} \in (m_{j1,2}, k_{j1,2})$ can be calculated. For the calculation module of the simulation program, the obtained values $x_{j1,2-n}$ must be inserted into the equation:

$$Xsym_{j1,2-n} = (x_{j1,2-n} + |q_{j1-2}|) \times 0.0001$$

For the second set of functions $f(x_{k1}) \wedge f(x_{k2})$ representing the asymmetric parabolic shape of the lower sealing ring, all mathematical operations are identical and the final formula for these functions is equal to:

$$Xsym_{k1,2-n} = (x_{k1,2-n} + |q_{k1-2}|) \times 0.0001$$

Similar shapes of the asymmetric sliding surfaces of the sealing rings can also be expressed by using elliptical functions, i.e. several curves closed symmetrically with respect to their common center. One recommends expressing the selected asymmetric elliptic shapes by ellipses $E(x_{j1})$, $E(x_{j2})$, $E(x_{k1})$, $E(x_{k2})$. The eccentricity of the ellipses is called parameter e which constitutes the value expressing the ratio of the focal length to half the length of the transverse diameter. As a boundary condition, it is assumed that the common coordinates of the center of ellipse pairs are $E(x_{j1})$, $E(x_{j2})$, as well as $E(x_{k1})$, $E(x_{k2})$. Directional coordinates p_{j1} and p_{j2} determine the position of the common center of the two ellipses on the x axis. According to the accepted assumptions, the ellipse data in canonical form can be described in the Cartesian coordinate system by the following equations:

$$E(x_{j1}) = \frac{(x_{j1}-p_{j1})^2}{a_{1j1}^2} + \frac{y_{j1}^2}{b_{1j1}^2} = 1; E(x_{j2}) = \frac{(x_{j2}-p_{j2})^2}{a_{2j2}^2} + \frac{y_{j2}^2}{b_{2j2}^2} = 1$$

$$E(x_{k1}) = \frac{(x_{k1}-p_{k1})^2}{a_{1k1}^2} + \frac{y_{k1}^2}{b_{1k1}^2} = 1;\ E(x_{k2}) = \frac{(x_{k2}-p_{k2})^2}{a_{2k2}^2} + \frac{y_{j2}^2}{b_{2k2}^2} = 1$$

where:

$$a_{1j1} \neq a_{2j2};\ b_{1j1} = b_{2j2};\ a_{1k1} \neq a_{2k2};\ b_{1k1} = b_{2k2}$$

$$G(x_{j1,2}) \text{ and } F(x_{j1,2}) \leq 0$$

hence:

$$\frac{y_{j1}^2}{b_{1j1}^2} = 1 - \frac{(x_{j1}-p_{j1})^2}{a_{1j1}^2}$$

$$y_{j1}^2 = b_{1j1}^2 - \frac{b_{1j1}^2(x_{j1}-p_{j1})^2}{a_{1j1}^2}$$

where: $y_{j1}^2 = w \Leftrightarrow y_{j1} = \sqrt{w} \cup y_{j1} = -\sqrt{w}$ – the significant parameter is $-\sqrt{w} > 0$

$$y_{j1} = -\sqrt{b_{1j1}^2 - \frac{b_{1j1}^2(x_{j1}-p_{j1})^2}{a_{1j1}^2}}$$

$$y_{j2} = -\sqrt{b_{2j2}^2 - \frac{b_{2j2}^2(x_{j2}-p_{j2})^2}{a_{2j2}^2}}$$

$$G(x_{j1,2}) = \begin{cases} -\sqrt{b_{1j1}^2 - \dfrac{b_{1j1}^2(x_{j1}-p_{j1})^2}{a_{1j1}^2}} \text{ for } x_{j1} \in (m_{j1,2}, p_{j1,2}) \\ -\sqrt{b_{2j2}^2 - \dfrac{b_{2j2}^2(x_{j2}-p_{j2})^2}{a_{2j2}^2}} \text{ for } x_{j2} \in (p_{j1,2}, k_{j1,2}) \end{cases}$$

Similarly, an elliptical function $F(x_{k1,2})$ can be expressed as:

$$F(x_{k1,2}) = \begin{cases} -\sqrt{b_{1k1}^2 - \dfrac{b_{1k1}^2(x_{k1}-p_{k1})^2}{a_{1k1}^2}} \text{ for } x_{k1} \in (m_{k1,2}, p_{k1,2}) \\ -\sqrt{b_{2k2}^2 - \dfrac{b_{2k2}^2(x_{k2}-p_{k2})^2}{a_{2k2}^2}} \text{ for } x_{k2} \in (p_{k1,2}, k_{k1,2}) \end{cases}$$

On the basis of the above equations describing the relationships between two parabolas or ellipses, one can determine a specific number n of their coordinates expressing the given asymmetric shape of the rings sliding surface. By entering the above data into the mathematical model [1], it is possible to record the balance of velocity fields equivalent to the flow balance between the sliding surfaces of the sealing rings and the surface of the cylinder wall. Based on the accepted data, mathematical model and on the adoption of a certain number of broken curves depending on the complexity of the shape of these surfaces, it is possible to determine the tendency of change in friction loss and selected parameters of the oil film, depending on the displacement of the common peak of the given functions at the axial height of these rings.

3. Dimensional coefficients for friction losses and selected oil film parameters for parabolic asymmetric shapes of the sealing rings sliding surfaces

Based on the results of the simulation shown below, it can be observed that there is a close dependence of the location of the common point of the parabolas $\mathbf{W_{j1}} = (p_{j1}, q_{j1})$ and $\mathbf{W_{j2}} = (p_{j2}, q_{j2})$ on the friction power of the individual piston rings in the set. For the mathematical model in the simulation, it is assumed that the coordinates $p_{j1} = p_{j2}$ refer to the upper sealing ring. All simulation results were expressed for the change of parabolic coordinates p_{j1-2} and p_{k1-2} according to the above assumptions. This is justified by the positive results of the simulation which concern the shape formation of the sealing rings sliding surfaces. According to these assumptions, the shape of the upper sealing ring's sliding surface must favor the formation of the shape characterized by a high value of the slip effect coefficient and a low value of the extrusion effect coefficient. The contours with a high value of the slip effect coefficient should be located on the sliding surface of this ring from the crankcase side when the piston moves in the direction of the TDC in the compression and exhaust strokes. The contours with small values of extrusion effect coefficient should be implemented on the opposite side. Thus, changing the coordinate value p_{j1-2} for this ring denotes the displacement of the vertex $\mathbf{W_{j1-2}}$ in the crankcase direction at the axial height H_1. If $k_{j1-2} = 50$ then $p_{j1-2} = 25$ = the symmetrical ring, $p_{j1-2} = 30 = m_{j1,2}$ = 40% H_1, $p_{j1-2} = 35 = m_{j1,2}$ = 30% H_1, $p_{j1-2} = 40 = m_{j1,2}$ = 20% H_1 and $p_{j1-2} = 45 = m_{j1,2}$ = 10% H_1.

For the lower sealing ring, the relationship is expressed by the coordinates $p_{k1} = p_{k2}$, which, along with the increase in values, are shifted towards the crankcase. For this ring, the contours with high extrusion effect coefficient values must be located on the sliding surface from the combustion chamber side, and the contours of the low slip effect coefficient values – on the opposite side. Thus the relation can be denoted as: $p_{k1-2} = 30 = m_{k1,2}$ = 60% H_2, $p_{k1-2} = 35 = m_{k1,2}$ = 70% H_2, $p_{k1-2} = 40 = m_{k1,2}$ = 80% H_2 and $p_{k1-2} = 45 = m_{k1,2}$ = 90% H_2. Accurate simulation and experimental results are needed to show changes in selected friction and oil film parameters for the specified coordinates p_{j1-2} and p_{k1-2} for both sealing rings.

According to the presented results, it can be seen that the reduction of the friction power and the friction force at 20° on the crankshaft after the TDC in the expansion stroke for the sealing ring occurs only for the selected coordinates of p_{j1-2} and p_{k1-2} and always runs in a nonlinear manner.

Consequently, the effect of the change of vertex coordinates for both sealing rings must be expressed by additional factors whose direction of change can be represented by a polynomial of higher degrees or by approximation of the results through interpolation. However, this method is subject to a significant error, for some special cases of shapes, especially at high values of the vertex offset $p_{j,k-1-2}$. For this reason, a more convenient method for approximating such results is to provide a comprehensive database for various engine designs, dynamic oil viscosities, and engine angular velocities.

It is also important to determine the effect of the changes in the asymmetrical shape, depending on the value of the gas pressure in the combustion chamber, especially in the expansion stroke. Below, there are only examples of simulation results for changes in $p_{j,k-1-2}$, for the design and performance of Cinquecento 700 engine, three selected crankshaft rotational speeds and two 10W/30 oil dynamic viscosities. All engine design parameters are shown in the simulations concerning the circumferential grooves placed on the sealing rings sliding surfaces [7]. The design features of this engine were selected for simulations solely because of the possibility to verify the obtained results with the results of bench tests. The impact of the change in the $p_{j,k-1-2}$ coordinate values on the percentage change in friction loss values and on oil film parameters selected for uncharged engines of similar geometric dimensions is comparable to most engines. As it can be expected, the simulation end values for these parameters are unique for each engine. However, their percentage difference in value from the simulation results that express the symmetry of the shape of standard rings used in any engine is already very similar to the presented results. It should be noted that the results are for a two-cylinder engine, whereas the results for engines with more cylinders must be properly corrected.

Figure 2 indicates that the increase in the coordinate value of parabola $p_{j,k-1-2}$ for both sealing rings simultaneously causes a reduction in the upper sealing ring friction force of 1–2% as compared to the value $p_{j,k-1-2} = 40$. Further displacement of the parabola common peak above this value leads to a significant increase in friction power of the entire ring set. For this range of the vertex shift, in some areas of engine operation, there occurs a discontinuity in the oil film between the upper sealing ring and the cylinder wall.

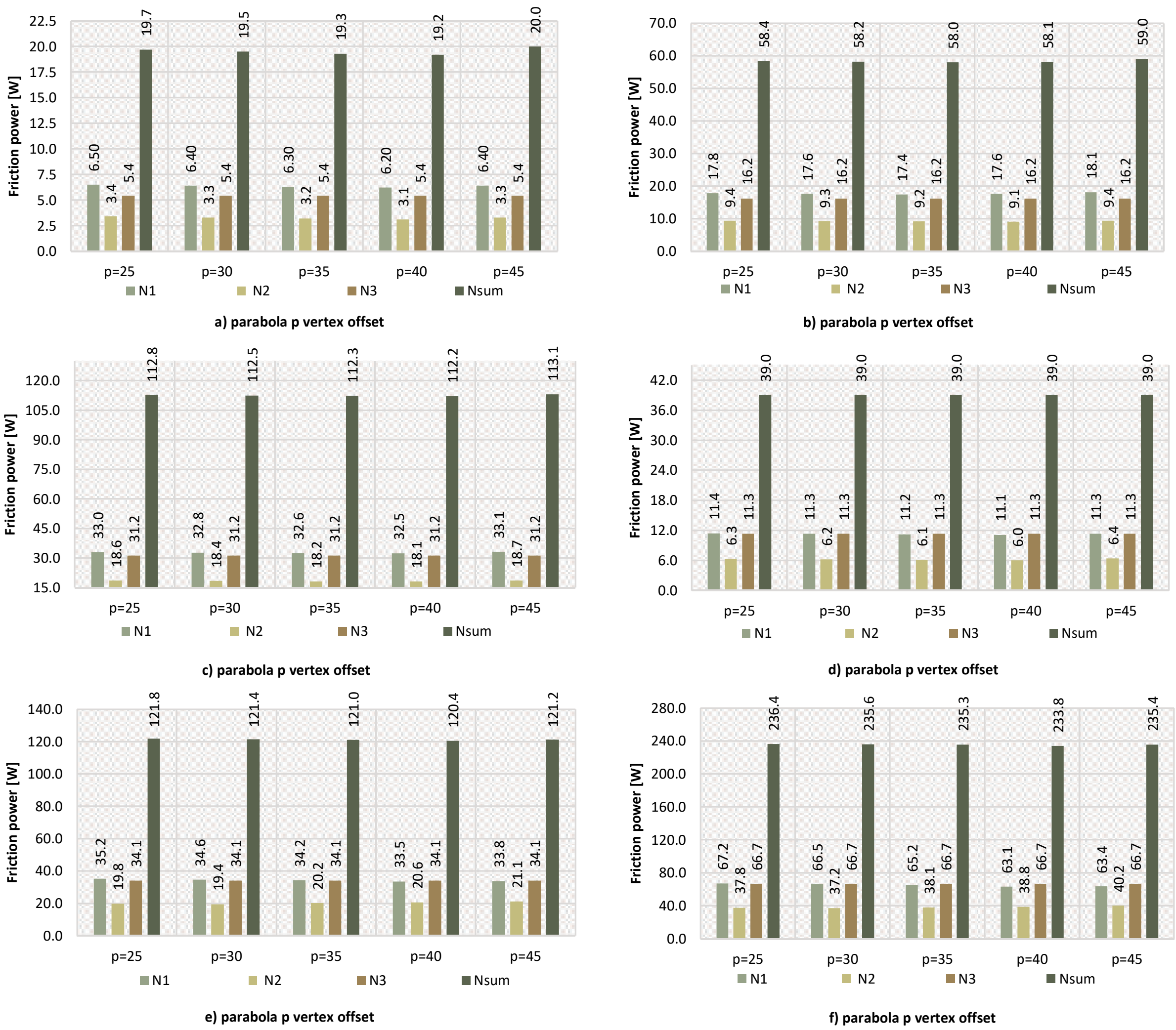

Fig. 2. Piston ring friction power for the engine velocity relative to the parabolic value of $p_{j,k-1-2}$, where: N_1 – Friction power of the upper sealing ring, N_2 – Friction power of the lower sealing ring, N_3 – Friction power of the scraper ring, N_{sum} – Friction power of a piston ring set, a) for 1000 rpm, b) for 2000 rpm, c) 3000 rpm – dynamic viscosity of lubricating oil 0.0120 Pa ×s, d) for 1000 rpm, e) for 2000 rpm, f) 3000 rpm – dynamic viscosity of lubricating oil 0.0386 Pa×s

As shown in Figure 2, for dynamic oil viscosity of 0.0386 Pa×s, a significant reduction of approximately 5% in the friction power of the upper sealing ring is observed, with the permissible vertex shift $p_{j,k-1-2}$, and the increase in friction power of the lower sealing ring at the engine velocity of 3000 rpm. The increased friction power of the lower sealing ring is determined by the local increase in the oil film thickness between the sliding surface of the ring and the cylinder wall. Under favorable conditions, this may also be due to the rupture in the oil film continuity as a result of the reduction in thickness of the oil layer left on its return passage. On the basis of the data presented, it can be assumed that the most advantageous range for the sealing ring set, due to the reduction in friction power, is the offset of the vertex coordinates $p_{j,k-1-2}$ in the range 35 to 40, i.e. 70% to 80% of the ring axial height. It is highly probable that independently of the selection of the values $p_{j,k-1-2} \in$ <26;40> one can obtain a reduction in the friction force of a piston ring set with respect to the symmetrical shape of the sealing rings. It should be noted that the symmetrical sliding surface of the sealing rings $q_{j,k-1-2} = -0.10$ has been accepted for simulation studies, which is equivalent to the vertex offset of 10 μm extending from the outer edge of the ring. In the actual reference rings, the value of this extension does not, on average, exceed 5 μm for the molybdenum and chromium coatings after reaching the piston rings. The asymmetric shapes of the sealing rings, on the other hand, are produced with a vertex offest of about 20 μm, and after running-in it is within the range of 8 to 12 μm. This is of high importance due to the value of the sealing rings friction power because the rings with larger offsets have a significantly lower friction power. This is justified by the reduced axial oil film coating of the sliding surfaces of these rings. In this case, the minimum oil film thickness for the upper sealing ring is also reduced. This is, however, sufficient to ensure the conditions for liquid friction throughout the entire engine cycle. For the engine in question, the minimum oil film thickness is greater than 0.201 μm at a dynamic viscosity of 0.0120 Pa×s. With the required roughness of the sliding surfaces of both the rings and the cylinder wall there is a negligible risk of breaking the continuity of the oil film for such rings in a longer period of engine operation.

Figure 3 shows the change in the friction force at 20° on the crankshaft after TDC in the expansion stroke for the upper sealing ring, depending on the displacement of the vertex $p_{j,k-1-2}$. On the basis of the data presented, it can be clearly stated that the shift of the vertex $p_{j,k-1-2}$, results in a significant reduction in friction force in the most loaded engine area. The friction force in this engine cycle period decreases with each modification in engine velocity and oil viscosity. The force of friction is influenced by the velocity gradient caused by slip.

For symmetrical shapes of rings, velocity gradients generated by the extrusion and slip effects, as well as differential pressure under and above the ring slightly affect friction force. However, for asymmetric shapes with a significant vertex shift $p_{j,k-1-2}$ the effect of the velocity gradient caused by the extrusion coefficient is significant. The velocity gradient caused by the difference in pressures below and above the ring for asymmetric ring shapes in supercharged engines is also of great importance. Knowing the unit friction force tangential to the cylinder liner with respect to Newtonian liquids, one can determine the total sealing ring friction force and the power absorbed to overcome the internal friction resistance in the oil film.

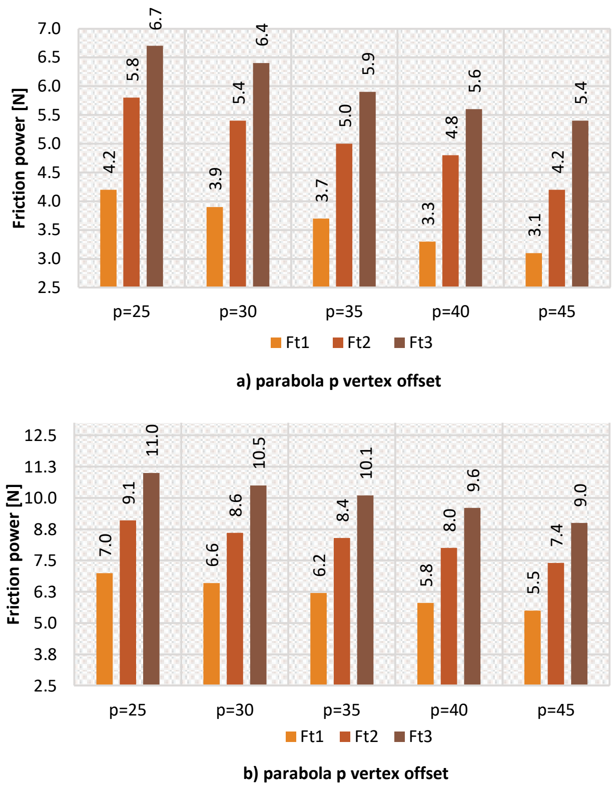

Fig. 3. Friction force at 380° on the crankshaft in the expansion stroke according to the parabolic vertex value of $p_{j,k-1-2}$ describing the sliding surface shape of the sealing rings, where: F_{t1} = 1000 rpm, F_{t2} = 2000 rpm, F_{t3} = 3000 rpm; a) for the dynamic viscosity of lubricating oil 0.0120 Pa×s, b) for the dynamic viscosity of lubricating oil 0.0386 Pa×s

By analyzing Figure 4, it is possible to observe the change in the oil film thickness depending on the vertex shift $p_{j,k-1-2}$. Figures 4a and b indicate that for the oil dynamic viscosity of 0.0120 Pa×s the vertex shift $p_{j,k-1-2}$ causes variation in the oil film thickness. This is due to the change in crankshaft angle in the expansion stroke assigned to the minimum oil film thickness, depending on the vertex displacement $p_{j,k-1-2}$. This means that the crankshaft angle, for which the oil film thickness between the ring's sliding surface and cylinder wall is minimal, depends on the dynamic viscosity of the oil. If $p_{j,k-1-2} = 25$, then $z_{min} = 364°$ on the crankshaft, $p_{j,k-1-2} = 30$, then $z_{min} = 366°$ on the crankshaft, $p_{j,k-1-2} = 35$, then $z_{min} = 370°$ on the crankshaft, $p_{j,k-1-2} = 40$, then $z_{min} = 372°$ on the crankshaft and $p_{j,k-1-2} = 45$, then $z_{min} = 374°$ on the crankshaft. For dynamic viscosity of oil at 0.0386 Pa×s, the oil film thickness can be reduced by increasing the value of $p_{j,k-1-2}$. This is especially noticeable for higher engine speeds – from 1000 rpm upwards. The change in the oil film thickness, depending on the change in the vertex position and dynamic viscosity of the oil, is determined by a variety of thickness of the scraped oil layers left by the ring on its return passage after the TDC and the thickness of the oil film scraped by the lower sealing ring. It should also be noted that the displacement of the vertex $p_{j,k-1-2}$ for the upper sealing ring towards the engine crankcase results in its encounter with the scraped oil layers in the expansion stroke in the further angles of the crankshaft in this stroke. For the value of $p_{j,k-1-2} \in <26;40>$ this is a beneficial effect that affects the reduction of the friction force and power in that ring. When the value of $p_{j,k-1-2} = 40$ is exceeded, a significant reduction in film thickness is achieved, which may not only form conditions for mixed friction, but also for boundary friction and irreversible engine damage. It should be mentioned here that the value of $p_{j,k-1-2} = 40 =$ limit, may vary for each engine. Thus, the limit value of parameter $p_{j,k-1-2}$ depends on the engine geometry and especially on the thickness of the oil layers scraped by each ring in the set and on the thickness of the oil film scraped by the lateral surface of the piston.

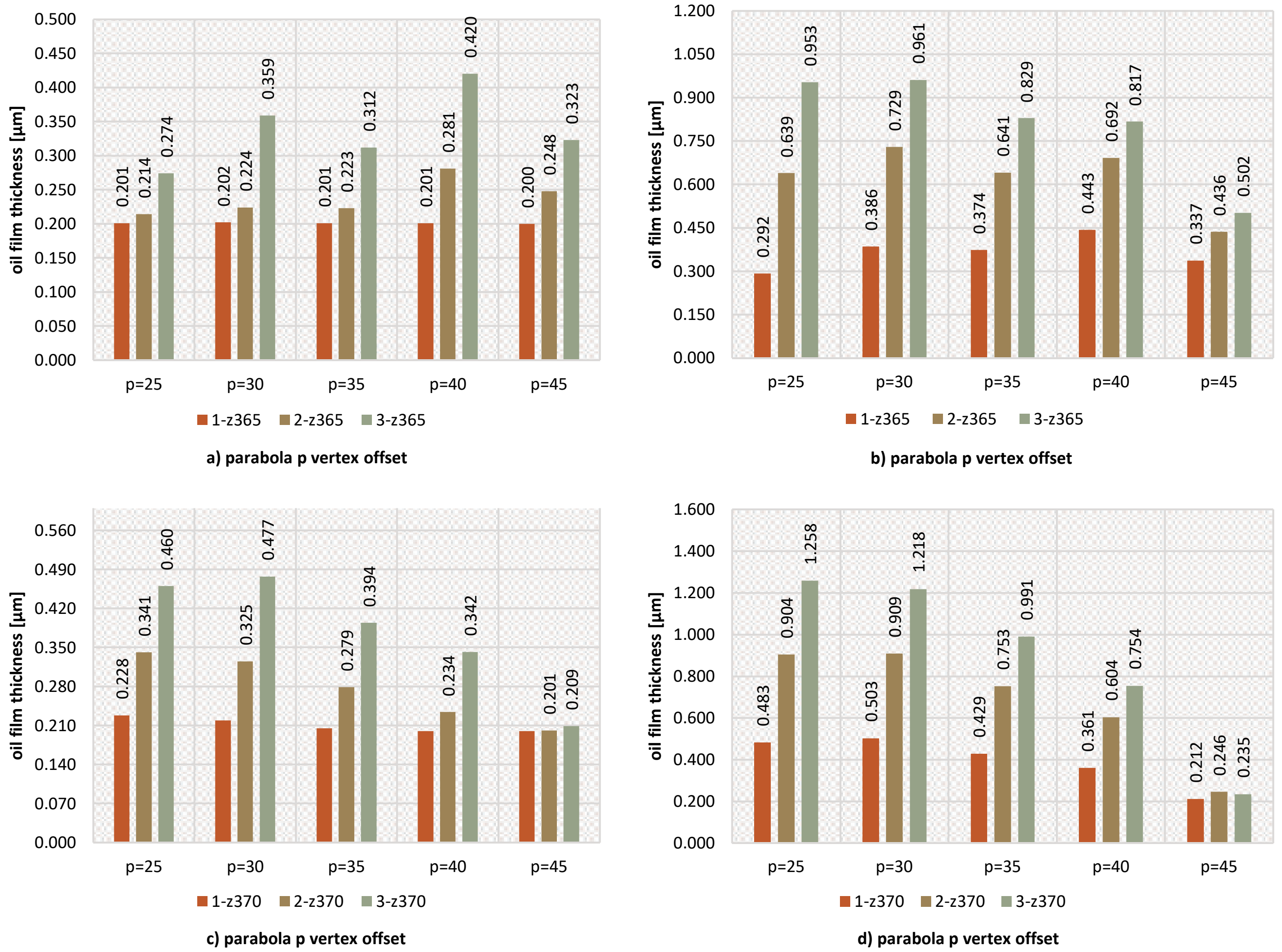

Fig. 4. Oil film thickness on the upper sealing ring slide surface depending on the parabolic vertex coordinate value $p_{j,k-1-2}$, where: 1 – 1000 rpm, 2 – 2000 rpm, 3 – 3000 rpm, z365 – the thickness of the oil film at 5° after TDC in the expansion stroke, z370 – the thickness of the oil film at 10° after TDC in the expansion stroke; a), c) for the dynamic viscosity of lubricating oil 0.0120 Pa×s, b), d) for the dynamic viscosity of lubricating oil 0.0386 Pa×s

4. Quantitative oil consumption analysis for parabolic asymmetric shapes of sealing rings sliding surfaces

The dynamic viscosity of the lubricating oil is one of the factors determining oil consumption. However, a multiple increase in the dynamic oil viscosity, which is due to the change in engine operating temperature and for the symmetrical sliding surfaces of the rings, does not significantly alter the quantitative oil consumption. As it is shown in Figure 5, the most important parameter determining oil consumption is the volume of oil scraped in the compression stroke V2 and the exhaust stroke V4 into the combustion chamber. The engine velocity also substantially influences the amount of scraped oil in the combustion chamber. Despite the large changes in both parameters for the symmetrical surfaces, volumes V2 and V4 are very similar. As can be seen in the graphs (Fig. 5), the displacement of the $p_{j,k-1-2}$. coordinates is quite significant for the volume of scraped oil in the combustion chamber throughout the entire engine cycle. An increase in this parameter to the value between 35 and 40 results in a very large reduction in the amount of consumed oil. When the value of parameter $p_{j,k-1-2} > 40$, there is a slight increase in the volume of oil scraped into the combustion chamber, with oil dynamic viscosity of 0.0120 Pa×s. For larger oil dynamic viscosities, the amount of oil scraped into the combustion chamber for this range of values also decreases.

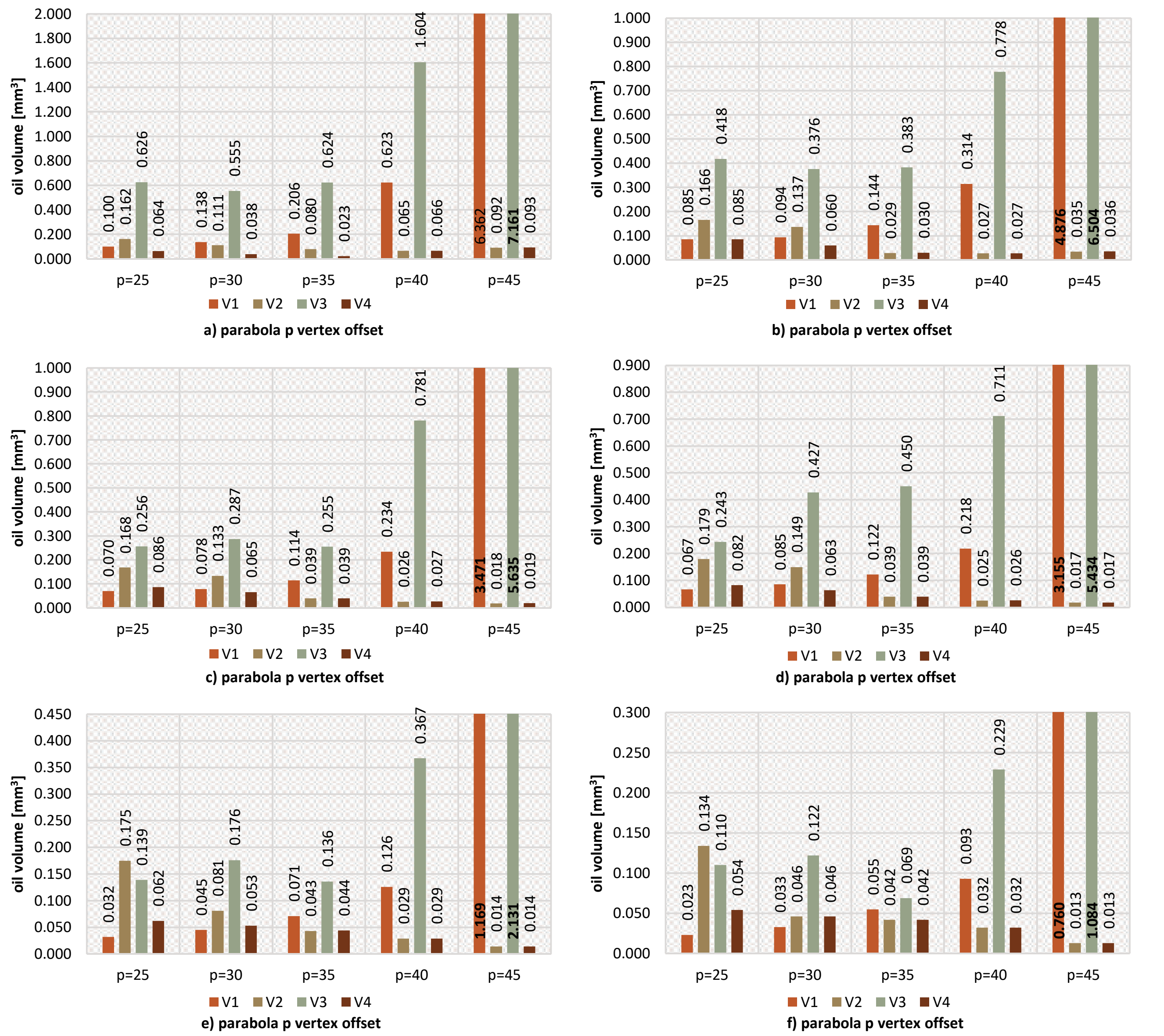

Fig. 5. The volume of lubricating oil scraped into the combustion chamber and crankcase in selected engine running strokes depending on the vertex offset value of the parabola $p_{j,k-1-2}$, where: V_1 – intake stroke, V_2 – compression stroke, V_3 – expansion stroke, V_4 – exhaust stroke; a) for 1000 rpm, b) for 2000 rpm, c) 3000 rpm – for the dynamic viscosity of lubricating oil 0.0120 Pa×s, d) for 1000 rpm, e) for 2000 rpm, f) 3000 rpm – for the dynamic viscosity of lubricating oil 0.0386 Pa×s

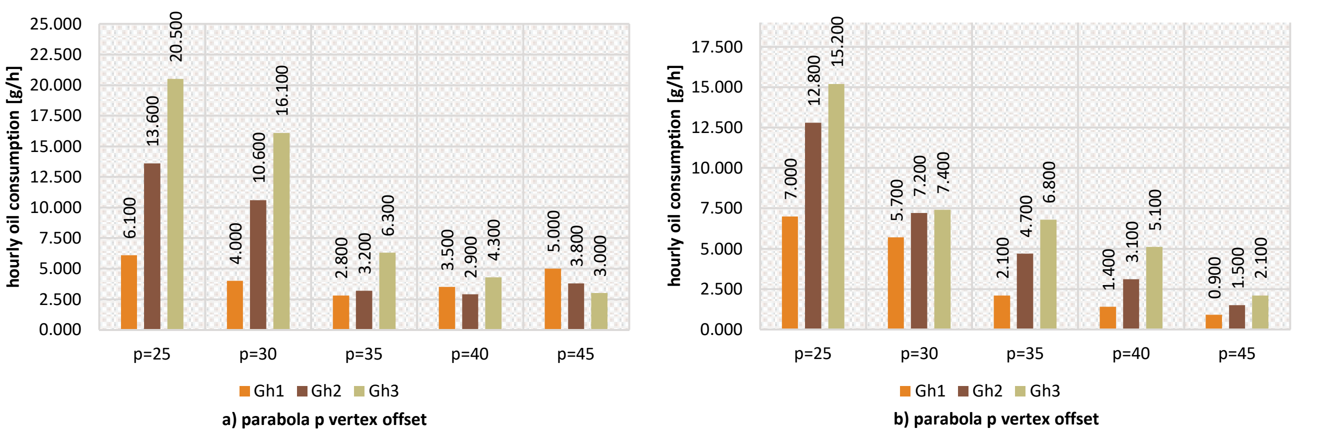

Fig. 6. The hourly consumption of lubricating oil according to the parabolic vertex value of $p_{j,k-1-2}$, where: Gh_1 = 1000 rpm, Gh_2 = 2000 rpm, Gh_3 = 3000 rpm; a) for the dynamic viscosity of lubricating oil 0.0120 Pa×s, b) for the dynamic viscosity of lubricating oil 0.0386 Pa×s

Table 1. Values of the coefficients determining the change in parameter $p_{j,k-1-2}$ expressing the asymmetric shape of the sealing rings sliding surfaces on the volume of scraped oil in the individual engine running strokes into the crankcase in strokes V1 and V3, and into the combustion chamber in strokes V2 and V4

$p_{j,k-1-2}$*	25	26	27	28	29	30	31	32	33	34	35
V1 [mm^3]	0.100	0.107	0.113	0.121	0.129	0.138	0.145	0.155	0.167	0.180	0.213
V1 [%]	100	107	113	121	129	138	145	155	167	180	213
V2 [mm^3]	0.162	0.146	0.134	0.125	0.117	0.111	0.104	0.098	0.092	0.086	0.080
V2 [%]	100	90.12	82.72	77.16	72.22	68.52	64.20	60.49	56.79	53.09	49.38
V3 [mm^3]	0.626	0.605	0.583	0.570	0.560	0.555	0.561	0.570	0.584	0.604	0.624
V3 [%]	100	96.65	93.13	91.05	89.46	88.66	89.62	91.05	93.29	96.49	99.68
V4 [mm^3]	0.064	0.057	0.051	0.046	0.042	0.038	0.034	0.030	0.027	0.025	0.023
V4 [%]	100	89.06	79.69	71.88	65.63	59.38	53.13	46.88	42.19	39.06	35.94
$p_{j,k-1-2}$*	------	36	37	38	39	40	41	42	43	44	45
V1 [mm^3]		0.259	0.311	0.382	0.491	0.623	0.934	1.560	2.765	4.430	6.362
V1 [%]		259	311	382	491	623	934	1560	2765	4430	6362
V2 [mm^3]		0.076	0.072	0.069	0.066	0.065	0.067	0.071	0.076	0.083	0.092
V2 [%]		46.91	44.44	42.59	40.74	40.12	41.36	43.83	46.91	51.23	56.79
V3 [mm^3]		0.687	0.802	0.978	1.234	1.604	2.130	2.801	3.750	5.102	7.161
V3 [%]		109.74	128.12	156.23	197.12	256.23	340.26	447.44	599.04	815.02	1143.93
V4 [mm^3]		0.026	0.033	0.041	0.053	0.066	0.074	0.081	0.086	0.089	0.093
V4 [%]		40.63	51.56	64.06	82.81	103.13	115.63	126.56	134.38	139.06	145.31

* for the lubricating oil dynamic viscosity value of 0.0120 Pa×s and engine speed 1000 rpm

As shown in Figure 5, the vertex shift $p_{j,k-1-2}$ for the upper sealing ring in the crankcase direction, and for the lower sealing ring towards the combustion chamber within the range of 51% to 80% results in a significant reduction in the volume of oil scraped into the combustion chamber. Similar conclusions can be reached following the analysis of Figure 6, concerning hourly oil consumption. It can be stated that the most advantageous reduction of oil consumption for 10W/30 oils can be obtained for the $p_{j,k-1-2}$ vertex coordinate shift to a maximum value of about 37. When the maximum value is exceeded at a normal engine running temperature of about 90°C, one observes an increase in the hourly oil consumption. At lower temperatures of this oil, the oil consumption is reduced at the higher values of the vertex coordinates $p_{j,k-1-2}$.

The reduction in the volume of oil scraped into the combustion chamber in strokes V2 and V4 and in the hourly oil consumption as the vertex position $p_{j,k-1-2}$ changes is determined by the lower thickness of the oil layer on which the return movement of the upper sealing ring is made. This is explained as follows: the higher the vertex offset of the $p_{j,k-1-2}$, the closer the turning point of the upper sealing ring is to the turning point of the lower sealing ring. The selected coefficients expressed as a percentage change in the volume of the scraped oil for the range of the vertex offests $p_{j,k-1-2} \in$ <25;45> are summarized in Table 1. A similar comparison can be made for any engine design and any engine velocity and dynamic lubricating oil viscosity.

5. Conclusions

On the basis of the data obtained from the simulations, one can state that the simultaneous opposing offset of the vertex $p_{j,k-1-2}$ for the lower and upper sealing rings with parabolic and elliptical asymmetrical sliding surfaces results in:

- reduction in the lower and upper sealing ring friction power from 1% to 5% depending on the dynamic viscosity of the lubricating oil as the vertex coordinates $p_{j,k-1-2}$ increase
- reduction in the friction force at 20° crankshaft after the TDC in the expansion stroke from several to more than 20% for the range of $p_{j,k-1-2} \in$ <26;45> depending on the engine velocity and oil dynamic viscosity,
- reduction in oil film thickness from 5 to 10° on the crankshaft after TDC in the expansion stroke whose value decreases with the increase of the vertex coordinates $p_{j,k-1-2}$,
- displacement of the crankshaft angle after TDC in the expansion stroke into higher values for which the minimum film thickness is recorded as the value of the vertex coordinates $p_{j,k-1-2}$ increases,
- significant reduction in the volume of oil scraped to the combustion chamber in the V2 compression stroke and V4 exhaust stroke and the reduction in hourly oil consumption for $p_{j,k-1-2} \in$ <26;40>. The percentage change in the value of these parameters affects the engine velocity and the dynamic viscosity of the oil. When $p_{j,k-1-2} > 40$, there is a rapid increase in the volume of oil scraped into the combustion chamber in each engine stroke.

Some additional conclusions can be drawn:

- determining the value of coefficients on the basis of simulation studies for individual oil film parameters and friction losses at different engine performance and different oil viscosities can contribute to a rapid verification of the effects of changes in symmetrical shapes of the reference rings into asymmetrical shapes adapted to individual engine geometric parameters.
- the asymmetric shapes of the sealing rings sliding surfaces make it possible to improve the engine mechanical efficiency while maintaining its durability and reducing oil consumption at the same time.

At present, the reduction in friction losses of a piston ring set is most frequently achieved by reducing their axial height and maintaining symmetrical shapes. Such actions result in serious consequences, i.e. a significant increase in the consumption of lubricating oil and a drastic reduction in the oil film thickness between the upper sealing ring's sliding surface and the cylinder wall. On the basis of the conducted simulations, the use of asymmetrically shaped sealing rings seems to be more advantageous. Their design does not significantly reduce the minimum thickness of the oil film between the cooperating surfaces but reduces the total thickness of the oil film and the total oil film coverage of these surfaces throughout the entire engine cycle. This is a very advantageous solution due to the fact that the only period of engine operation most prone to rupturing the oil film between cooperating kinematic pairs is in the range of 5 to 20° on the crankshaft after TDC in the expansion stroke for the upper sealing ring. In the remaining crankshaft angles throughout the engine cycle, outside the area just before the TDC, the thickness of the oil film in the compression stroke is several times higher, which means that there is no increased risk of oil film failure in the crankshaft angle range. This only occurs if the minimum oil film thickness at the beginning of the expansion stroke for the upper sealing ring is fulfilled. This means that the reduction in the thickness of the oil film in the remaining crankshaft angles throughout the whole engine cycle is a very desirable phenomenon. Such distribution of film thickness and oil film coverage of the sealing rings sliding surfaces can be achieved primarily by using the asymmetric shapes of those rings with a suitably selected individual displacement of the vertex $p_{j,k-1-2}$.

Nomenclature

F_1	oil film speed gradient at the oil film entrance point
F_2	oil velocity gradient due to mutual approach of the surfaces limiting the oil film
$F_3 = F_4$	oil speed gradient at the outlet from the lubricant slot
H_1, H_2	axial height of the sealing rings
L_x	the length of the lubrication gap covered with the oil film
p_1, p_2	external pressure acting on the oil film on the divergent and convergent side
p_v	pressure in the oil film caused by the extrusion effect
p_w	oil pressure caused by the slip effect
U_1, U_2	relative sliding speeds
Z_{1-x}, Z_{2-x}	distance of divergent and convergent oil boundaries
Z_v	thickness of oil film applied on the cylinder liner
Z_w	thickness of oil film left on cylinder surface after ring passage

Bibliography

[1] ISKRA, A. Rozkład filmu olejowego na gładzi cylindrowej silnika spalinowego, *Rozprawy nr 181*, Politechnika Poznańska 1987.

[2] KOSZAŁKA, G. Application of the piston-rings-cylinder kit model in the evaluation of operational changes in blowby flow rate. *Maintenance and Reliability*. 2010, **4**, 72-81.

[3] TIAN, T. Dynamic behaviors of piston rings and their practical impact – part ii: oil transport, friction, and wear of ring/liner interface and the effects of piston and ring dynamics. *Proc. Inst. Mech. Eng., Part J: Journal of Engineering Tribology*. 2002, **216**.

[4] WOLFF, A. Numerical analysis of piston ring pack operation. *Combustion Engines*. 2009, **2**, 128-141.

[5] WOLFF, A. Modelowanie i symulacja numeryczna funkcjonowania pakietu pierścieni tłokowych dwusu-wowego silnika okrętowego. *Zesz. Nauk. Inst. Pojazd., Wydz. Sam. i Masz. Robocz., Warsaw University of Technology*. 2011, **1**(82), 5-17.

[6] WOLFF, A. Numerical analysis of piston ring pack operation of a marine two-stroke engine. *Combustion Engines*. 2011, 146(**3**).

[7] WRÓBLEWSKI, P., ISKRA, A. Geometry of shape of profiles of the sliding surface of ring seals in the aspect of friction losses and oil film parameters. *Combustion Engines*. 2016, **167**(4), 24-38.

Piotr Wróblewski, MEng. – Faculty of Machines and Transport at Poznan University of Technology.

e-mail: *Piotr.Je.Wroblewski@doctorate.put.poznan.pl*