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THE EFFECT OF CHANGES IN CYLINDER LINER GEOMETRY ON THE OPERATION OF PISTON COMPRESSION RING

WPŁYW ZMIAN GEOMETRII CYLINDRA NA PRACĘ USZCZELNIAJĄCEGO PIERŚCIENIA TŁOKOWEGO

A b s t r a c t

Presented paper deals with cylinder deformations which could be most often encountered and tries to explain the causes of such situations. The essential part of the paper presents relations which allow to describe cylinder shape of any geometry and the attached example explains better the principles of their application. The obtained relations will be applied to the constructed mathematical model, in particular in a module describing the link between cylinder deformations and blowby during engine run. It should be added that the presented study is a part of a series concerning the collaboration of compression ring with the cylinder wall observed on a running engine.

Keywords: combustion engine, piston ring, ring wall pressure

S t r e s z c z e n i e

W niniejszym opracowaniu opisano najczęściej spotykane deformacje cylindra i dokonano próby wyjaśnienia przyczyn ich powstawania. W zasadniczej jego części zaprezentowano związki pozwalające opisać kształt cylindra o dowolnej, złożonej geometrii, a załączony przykład obliczeniowy pozwala lepiej wyjaśnić zasady ich stosowania. Uzyskane zależności zostaną wykorzystane w budowanym modelu matematycznym, w module dotyczącym wpływu odkształceń cylindra na przedmuchy gazów spalinowych w czasie pracy silnika. Należy dodać, że niniejsze opracowanie jest częścią większego cyklu dotyczącego współpracy uszczelniającego pierścienia tłokowego z gładzią cylindra w czasie pracy silnika spalinowego.

Słowa kluczowe: silnik spalinowy, pierścień tłokowy, nacisk pierścienia

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1. Introduction

The operation of piston compression ring is being regarded as correct if it prevents effectively the blow by to the crankcase. Fulfillment of this condition requires the full contact of ring face to the liner surface along the entire circumference (which means that the ring pressure against the liner should be higher than naught). The value of the pressure is selected by constructor at the initial stage of ring construction and it depends on ring geometry and material.

The wear of cylinder liner surface should be pointed out as one of the causes of this phenomenon. As the measurements carried out on new and worn cylinders prove their geometry differs from the designed one (see comments in [10]), sometimes to the far extend. Cylinder deflections might appear both at the stage of engine assembly and during its operation. Category of deflections that could emerge before engine run (called the assembly ones) include deformations caused by the errors in construction of cylinder block and head as well as improper assembly of these elements. Often a simple modification in engine block construction leads to the reduction in amplitude of deformations or to the change in their character (Fig. 1).

Detailed information on the mechanisms causing the cylinder liner failures during engine run one could find in [4]. The result is the change in cylinder diameter and surface deformations caused, for example by uneven thermal load.

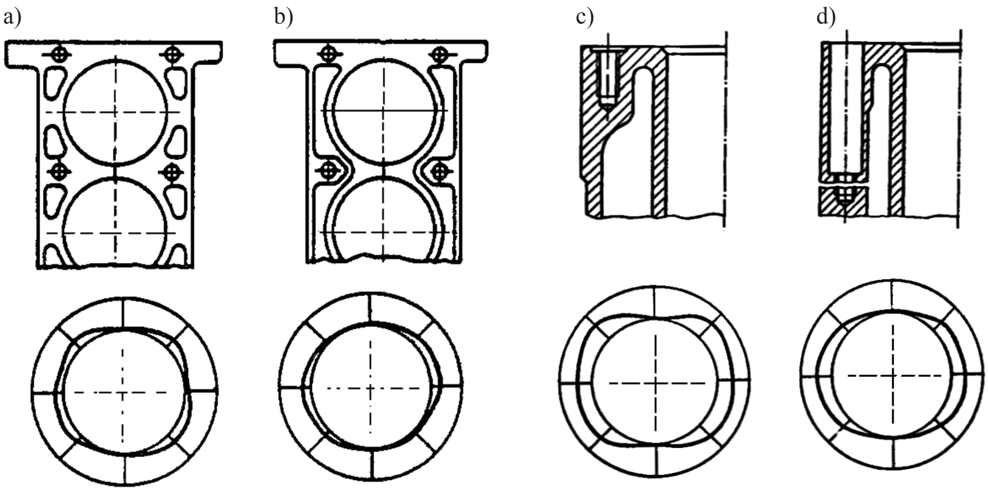

Fig. 1. Exemplary changes in cylinder circumferential line caused by: different construction of cylinder block (a and b) and different installation of head bolts (c and d) [3]

Rys. 1. Przykładowe zmiany linii obwodowej cylindra spowodowane: odmiennym wykonaniem bloku cylindrowego (a i b) oraz odmiennym sposobem kotwiczenia śrub mocujących głowicę (c i d) [3]

The amplitude of cylinder surface deformations, variable along its circumference, most often is far smaller than the wear caused by the collaboration of piston skirt and rings with the cylinder wall (Fig. 2).

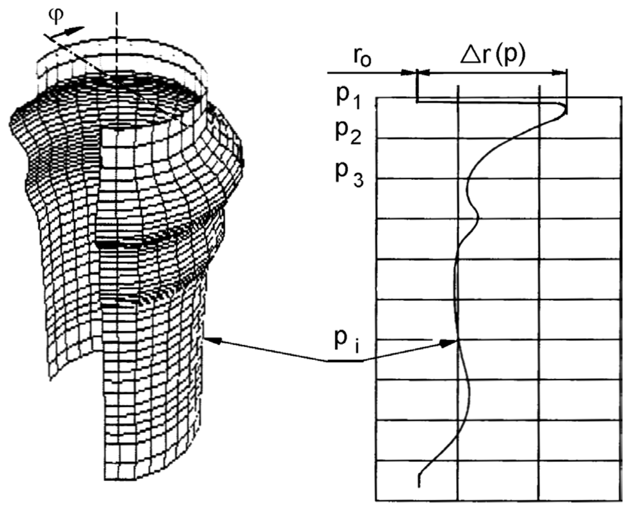

Fig. 2. Exemplary course of cylinder generatrix of an operated engine

Rys. 2. Przykładowy przebieg tworzącej eksploatowanego cylindra silnika spalinowego

A typical course of cylinder generatrix (this line is a cross section of cylinder surface with a plane laying in its symmetry axis, Fig. 2) shows that maximum wear takes place in upper part of liner where the top compression ring contacts with liner surface (the region of TDC).

A knowledge on actual geometry of the cylinder surface is of principal importance when the correctness of piston-cylinder assembly is to be determined, especially the problem of the combustion chamber tightness. Even minor deflections might lead to the loss of so called light tightness, i.e. spaces between the ring face and the cylinder surface emerge which cause the blow-by if not filled with lubricating oil. An analysis of the effect of cylinder geometry on distribution of circumferential ring pressure is necessary for the trial to point out the location of light slots and their shape. The construction of a model that could allow for such an analysis should be preceded by a mathematical description of changes in cylinder liner geometry as a result of engine operation.

2. Description of engine cylinder geometry

Determination of the effect of cylinder shape on the distribution of circumferential ring wall pressure requires a mathematical description of its geometry. Natural course of cylinder wear or the one caused by a failure make the value of cylinder radius difference for different cross sections p_i (distributed evenly along the cylinder generatrix – Fig. 2). The initial cylinder radius r_o remains unchanged outside the piston travel (or changes itself regularly as the result of deformations). In most cases the wear of the cylinder surface Δr is highest at the area of piston TDC. This phenomenon could be explained by the presence of high temperature (which affects the continuity of oil film) and high ring pressure (caused by exhaust). Precise measurements of used cylinder liners prove that the wear has more complicated form which could not be explained fully by the hydrodynamic theory of lubrication. Differentiation of wear that could be observed between individual cylinders of the same engine could be caused by errors committed during engine assembly and adjustment (common turbocharger, uneven supply of lube oil etc. [9]). The wear of cylinder is far lower at the BDC where the conditions of liner lubrication are far more favorable.

Identification of pressure distribution changeability requires the definition of changes in the value of cylinder radius and curvature depending on location of the cross section p_i for which this increment is being defined (Fig. 2). In mathematical models functions of the *sin*(*x*) and e^x type are used for the description of cylinder generatrix course [3].

Polar coordinates are used for the description of cylinder circumference course (relative to the chosen plane p_i) giving the value of radius $r(\varphi)$ for specific values of the angle φ (denominations as in Fig. 3b):

$$r(\varphi) = r_o + \Delta r(\varphi) \tag{1}$$

where:

r_o – radius of a new cylinder,
$\Delta r(\varphi)$ – change in cylinder radius along circumference.

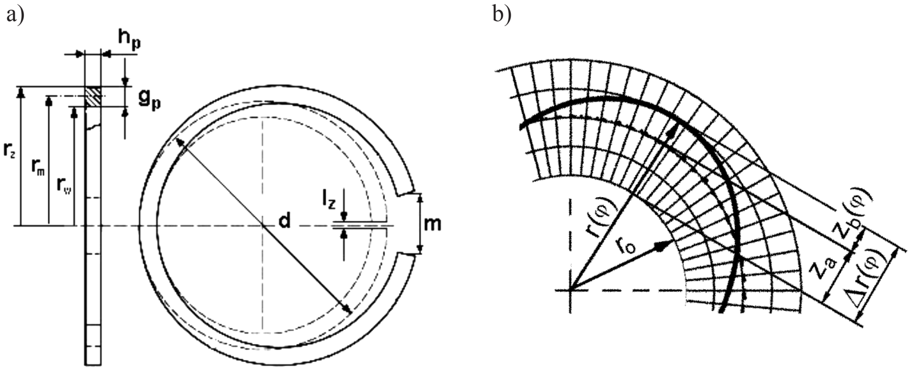

Fig. 3. Sketch of a compression ring (a) and course of cylinder circumferential line (b); r_z, r_m, r_w – radii of a ring installed into cylinder, respectively of outer surface, neutral layer and inner surface; l_z, m – ring gap: compressed and free

Rys. 3. Szkic tłokowego pierścienia uszczelniającego (a) oraz przebieg linii obwodowej cylindra (b); r_z, r_m, r_w – promienie pierścienia osadzonego w cylindrze, odpowiednio powierzchni zewnętrznej, warstwy obojętnej i wewnętrznej, l_z, m – odległość pomiędzy końcami pierścienia w stanie ściśniętym i swobodnym

The change in cylinder radius could be expressed as a sum of two components, i.e. the wear z_a (constant along the cylinder circumference) and deformation z_b (variable depending on the angle φ):

$$\Delta r(\varphi) = z_a + z_b(\varphi) \tag{2}$$

The Fourier series is often used for a mathematical description of the course of cylinder circumference. The series can be written in two forms [2]. Expression $y(\varphi)$ is a sum of components characterized by amplitudes a_i and b_i for selected values of the angle φ_i (a_o is an average value) in the orthogonal form:

$$y(\phi) = a_0 + a_1\cos(\varphi_1) + a_2\cos(\varphi_2) + a_3\cos(\varphi_3) + \ldots + b_1\sin(\varphi_1) + b_2\sin(\varphi_2) + b_3\sin(\varphi_3) \quad (3)$$

On the other hand, for the amplitude form given as:

$$y(\varphi) = c_0 + c_1\sin(\varphi + \beta_1) + c_2\sin(2\varphi + \beta_2) + c_3\sin(3\varphi + \beta_3) + \ldots \quad (4)$$

the expression $y(\varphi)$ is a sum of sinusoids characterized by amplitudes c_i and angles $\varphi + \beta_i$ where β_i are the angles of phase shift (c_o is an average value).

Using designations from the description of cylinder geometry to the formula (4) following equation has been obtained:

$$r(\varphi) = r_o + z_a + \sum_{h=1}^{n} A_h \sin(h\varphi + \beta_h) \quad (5a)$$

where r_o is an initial value of cylinder radius (new one), z_a is an accepted cylinder constant average wear, A_h – amplitudes of consecutive harmonics h of the Fourier series and n is a number of harmonics taken into account to the description of cylinder circumferential line. Some references [1] instead of the formula (5a) use the following form:

$$r(\varphi) = r_o + z_a + \sum_{h=1}^{n} A_h \cos(h\varphi + \delta_h) \quad (5b)$$

where instead of phase shift β_h another value has been introduced, namely δ_h (both values are interrelated by the following relationship $\delta_h = \pi/2 - \beta_h$).

Following cases relative to the selected states of cylinder surface could be defined when analyzing possible courses of the cylinder circumferential line (Fig. 4):

a) a new cylinder:

$$r(\varphi) = r_o = \text{const}. \quad (6)$$

b) new cylinder, its circumferential line underwent a deformation, e.g. during assembly but the mean value of cylinder radius remains unchanged:

$$r(\varphi) = r_o + z_b(\varphi) \quad (7)$$

c) the cylinder is evenly worn, i.e. its radius exceeds the radius of new cylinder by the value of z_a:

$$r(\varphi) = r_o + z_a \quad (8)$$

d) the cylinder is worn and deformed; that is the most general case relative to the situation when the course of cylinder circumferential line differs from a circle and the mean radius is longer than r_o:

$$r(\varphi) = r_o + z_a + z_b(\varphi) \quad (9)$$

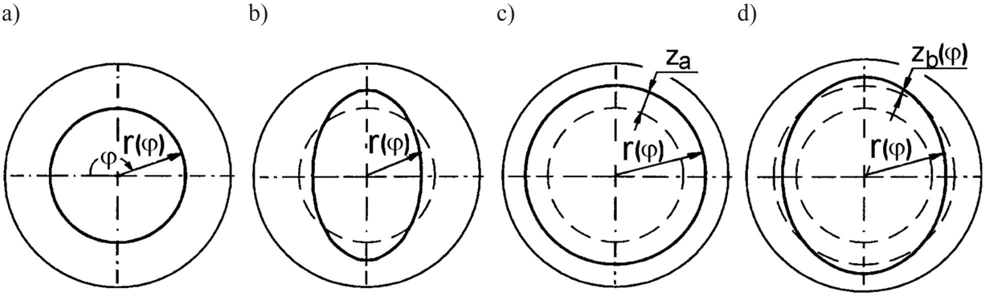

Fig. 4. Sketches of cylinder circumferential line course for selected cases of deformation and wear: a) new cylinder, b) new deformed cylinder, c) cylinder evenly worn, d) cylinder worn and deformed

Rys. 4. Szkice przebiegu linii obwodowej cylindra dla wybranych przypadków jego deformacji i zużycia: a) cylinder nowy, b) cylinder nowy zdeformowany, c) cylinder równomiernie zużyty, d) cylinder zużyty i zdeformowany

As the result of a different course of cylinder circumferential line than the one assumed initially, the ring wall pressure p_o would change itself as well. An increase of cylinder radius by a constant value (case 4c) will cause only a decrease in ring wall pressure, approximately proportional to the increase. In other cases, when the change in course of circumferential line is an irregular one, changes in ring wall pressure are difficult to establish. An additional problem that should be taken into consideration when analyzing the variability of ring pressure is a changeability of cylinder liner along the cylinder generatrix (Fig. 2b). Definition of the pressure variability requires a knowledge of parameters $z_a(p)$ and $z_b(\varphi, p)$.

A basic relation between the change in bar curvature Δρ and loading deflecting moment M_g has a form [6]:

$$\Delta\rho = -\frac{M_g(\varphi)}{E \cdot I} \tag{10}$$

where the product of modulus of elasticity E and the moment of inertia of bar cross section I is so called bar rigidity. Behavior of a typical piston ring could be approximated with such bar. After the ring installation in ring groove and insertion of complete piston into cylinder liner, the curvature of ring neutral layer changes from the one of the free form (ρ_p) into another fitted to the cylinder curvature (ρ_c):

$$\rho_p - \rho_c = -\frac{M_g(\varphi)}{E \cdot I} \tag{11}$$

It can be proved (it will be the subject of another paper) that the ring circumferential pressure against the wall of deformed surface is given by the formula (denomination as in Figs 3 and 4):

$$p(\varphi) = \frac{16 \cdot E \cdot I}{h_p \cdot d \cdot (d - g_p)^3}\left[\frac{K \cdot (d - g_p)}{2} - \left(z_a + z_b + z_b'' + z_b''''\right)\right] \tag{12}$$

where K is a characteristic parameter of ring (described in details in [9]).

The dependence (12) shows that circumferential distribution of ring pressure $p(\varphi)$ depends not only on ring's characteristic features but also on condition of cylinder wall, on wall wear parameters in particular.

3. Description of cylinder surface curvature

A knowledge of cylinder radius $r(\varphi)$ at a certain point of circumference line (given by the φ angle) allows to define the value of cylinder curvature ρ_c at this point [2]:

$$\rho_c(\varphi) = \frac{r^2(\varphi) + 2(r'(\varphi))^2 - r(\varphi)r''(\varphi)}{[r^2(\varphi) + (r'(\varphi))^2]^{3/2}} \tag{13}$$

Because for cylinder liners used on typical combustion engines following relations take place:

$$\left(\frac{r'(\varphi)}{r(\varphi)}\right)^2 << 1 \text{ and, } \left(\frac{r''(\varphi)}{r(\varphi)}\right) > \left(\frac{r'(\varphi)}{r(\varphi)}\right)^2$$

the equation (13) takes simpler form

$$\rho_c(\varphi) = \frac{1}{r(\varphi)}\left(1 - \frac{r''(\varphi)}{r(\varphi)}\right) \tag{14}$$

Remembering that $r(\varphi) = r_o + z_a + z_b(\varphi)$ and performing necessary transformations we obtain:

$$\rho_c(\varphi) = \frac{1}{r(\varphi)}\left(1 - \frac{z_b''(\varphi)}{r(\varphi)}\right) \tag{15}$$

or

$$\rho_c(\varphi) = \frac{1}{r_o + z_a + z_b(\varphi)}\left(1 - \frac{z_b''(\varphi)}{r_o + z_a + z_b(\varphi)}\right) \tag{16}$$

Formula (9) can be more simplified if we take into consideration that $r_o > z_a$and $r_o >> z_b(\varphi)$. Finally:

$$\rho_c(\varphi) = \frac{1}{r_o}\left(1 - \frac{z_b''(\varphi)}{r_o}\right) \tag{17}$$

In literature one can find a statement that though the course of cylinder circumferential line could be expressed as a sum of infinite number of Fourier series harmonics, most often only some of them of highest amplitudes (namely second and fourth) are used. In order to corroborate this observation an examination of cylinder liners from six cylinder diesel driving an earth mover for 11 thousand hours was carried out. Measurements with profilograph were performed at 16 points distributed evenly along liner circumference at piston TDC. Already an initial analysis (available in other papers) showed that the value of individual liner wear and its circumferential distribution alike differ considerably one from another. In Fig. 5d the liner radial wear meant as a difference between values of actual radius and the one corresponding to the new liner has been marked for one of the liners.

Harmonic analysis of the acquired data allowed to calculate the amplitudes of individual harmonics, angle of phase shift and constant component (Table 1). Harmonics 2 and 4 are characterized by the highest amplitude. Their courses (after addition to the mean wear) are presented in Figs 5c and 5d. Comparison of the sum of these harmonics with the distribution of measuring points (Fig. 5a) shows that the calculations were carried out correctly (this is confirmed by detailed calculations – not presented here; difference between specific wear, measured and calculated does not exceed 2%).

Table 1

Results of harmonic analysis of cylinder wall wear distribution

No of harmonic	Amplitude A_h	Phase shift δ_h
	µm	rad
0	22.13	–
1	0.22	2.575
2	3.40	–2.310
3	0.17	–2.100
4	0.50	–1.240
5	0.10	–0.420
6	0.04	–0.400

If the course of cylinder circumferential line was described only with one harmonic (an approximate description) the Eq. (5b) can be written in following form:

$$r(\varphi) = r_o + A_o + A_h \cos(h\varphi + \delta_h) \tag{18}$$

Depending on the value of wear z_a and amplitude A_h taken into consideration to the calculations of h harmonic, the ring wall pressure will change itself. For high amplitude of harmonic areas could appear along the cylinder circumference where the ring loses its contact with wall (pressure lower than naught).

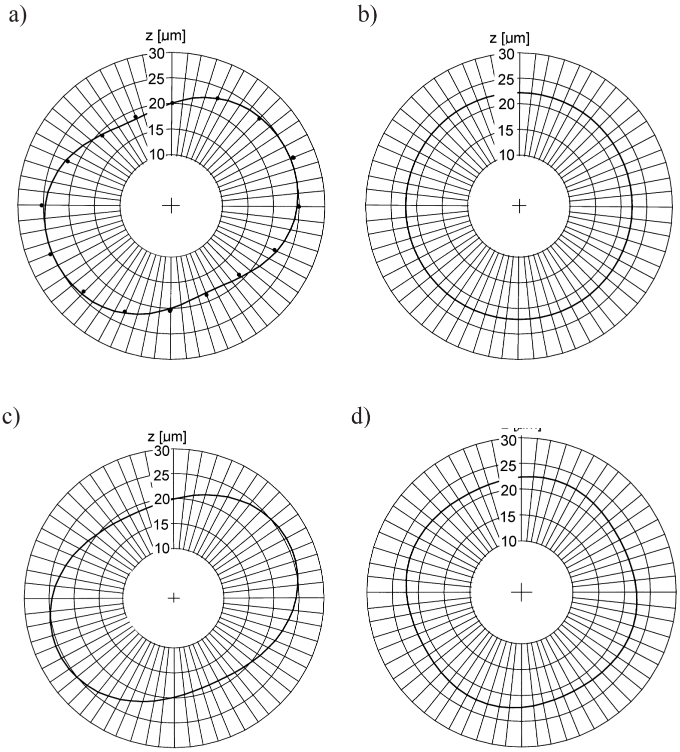

Fig. 5. Exemplary results of cylinder wall measurements (points) and their approximation with a sum of harmonics of the 2nd and 4th order (a), and the value of wear: mean one (b), the one described with 2nd harmonic (c) and the one described with 4th harmonic (d)

Rys. 5. Przykładowe wyniki pomiaru zużycia gładzi cylindra (punkty) oraz ich przybliżenie sumą harmonicznej 2 i 4 rzędu (a), a także przebiegi: zużycia średniego (b), opisanego 2 harmoniką (c) i 4 harmoniką (d)

Problems relative to the definition of bending moment and distribution of ring pressure in case of the worn cylinder will be analyzed in future papers.

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