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ANALYSIS OF PARAMETERS VARIABILITY OF COMPRESSION RING AND CYLINDER LINER COLLABORATION DURING ENGINE OPERATION

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Abstract

Construction of piston-rings-liner assembly of contemporary engines of low and medium power secures a long time reliable operation. In the case of engines of higher output, e.g. marine and railway ones still occur failures caused by improper collaboration of piston rings and cylinder liner. One can mention variations in collaboration surface and difficulties with supply and proper distribution of lubricating oil over the entire surface of cylinder liner among the most important ones.

Changes in value of parameters characteristic for compression ring and cylinder liner collaboration that occur during engine run have been analyzed in the following study. Attention has been paid to the ability of ring contact with the liner, especially those features that cause the formation of light slots between ring face and liner surface.

Drafts presented in the paper show the results of compression ring pressure distribution and the geometry of light slots carried out for technical data of the earth moving machine engine. Simulation computations take into account both the effect of engine operation cycle phase and selected stages of its run as well.

***Keywords**: IC engine, liner deformation, piston ring, ring elastic pressure*

1. Introduction

During engine operation the compression ring installed in groove is pressed against the wall by radial forces resulting from ring own elasticity and gas force that acts on ring's inner face. When the value of elastic pressure remains approximately constant, the ring wall pressure caused by gas pressure changes itself cyclically. Generally, this phenomenon should be regarded as advantageous one because it leads to the better ring sealing properties during those strokes when the gas pressure in combustion chamber is highest one, i.e. strokes of compression and expansion. This phenomenon gives better ring fit to the deformed cylinder liner preventing or limiting the gas blow-by. However, it should be noted that the ring high wall pressure could negatively affect the engine durability. High pressure could lead to high stress in ring material (which can cause the ring break) and bring about an oil film rupture (ring collaborates with liner under conditions of boundary or even no lubrication).

These problems were analyzed by the Authors in their earlier studies, concerning above all on the analysis of ring wall pressure distribution (using the previous papers concerning this problem as an example, e.g. [1,3]). The mentioned investigations concerned rather simple problems starting from the case of merely ring pressure against ideally circular wall [5] and next - the deformed wall [6,7]. Further the scope of investigations has been widened and besides the ring own elasticity it encompassed the effect of gas forces as well [8]. In present paper the range of investigations has been further broadened including collaboration of compression ring with the worn and deformed cylinder wall taking into considerations also the gas forces.

It should be stressed that the effect of oil layer present on the wall was omitted in this study. Such assumption is necessary for the applied method of computations which allows the determination of certain quantities characteristic for ring and wall collaboration, including forces crucial for full contact as well as location and size of light slots formed on cylinder wall. Taking the oil layer into consideration investigations possible in future will cause the verification of some conclusions, especially those concerning location and shape of slots.

2. Ring pressure against the deformed cylinder wall

It has been proved in [8] that the ring circumferential pressure distribution $p_m(\varphi)$ against the worn and deformed cylinder wall can be determined using the following formula, (symbols as in Fig. 1):

$$p_m(\varphi) = \frac{E \cdot I}{h_p \cdot r_m^4}\left[K_z(\alpha) \cdot r_m - \left(z_a + z_b(\varphi) + 2 \cdot z_b''(\varphi) + z_b^{IV}(\varphi)\right)\right], \tag{1}$$

where:
E –Young's modulus,
I – ring cross-section moment of inertia,
r_m – radius of ring in cylinder neutral layer (see Fig. 1a),
h_p – ring axial height,
z_a – cylinder constant deformation,
$z_b(\varphi)$ – cylinder wall deformation, variable along the cylinder circumference, given by the φ angle (and its second and fourth derivative).

Quantity $K_z(\alpha)$, later called a ring characteristic dynamic coefficient, can be described with the following formula [4]

$$K_z(\alpha) = K + \frac{p_g(\alpha) \cdot h_p \cdot r_m^3}{E \cdot I}, \tag{2}$$

where:
$p_g(\alpha)$ – pressure that presses on the ring to cylinder wall (its value depends on pressure over and below the ring); this pressure can be expressed as the function of crank angle α,
K – a ring characteristic constructional coefficient given by

$$K = \frac{p_m \cdot h_p \cdot r_m^3}{E \cdot I}. \tag{3}$$

Values of constructional and dynamic ring characteristic coefficients are equal, i.e. $K_z(\alpha)=K$ when there is no gas pressure and the ring wall pressure $p_m(\varphi) = p_m$ results exclusively from ring own elasticity and does not depend on crank angle

a) *b)* *c)*

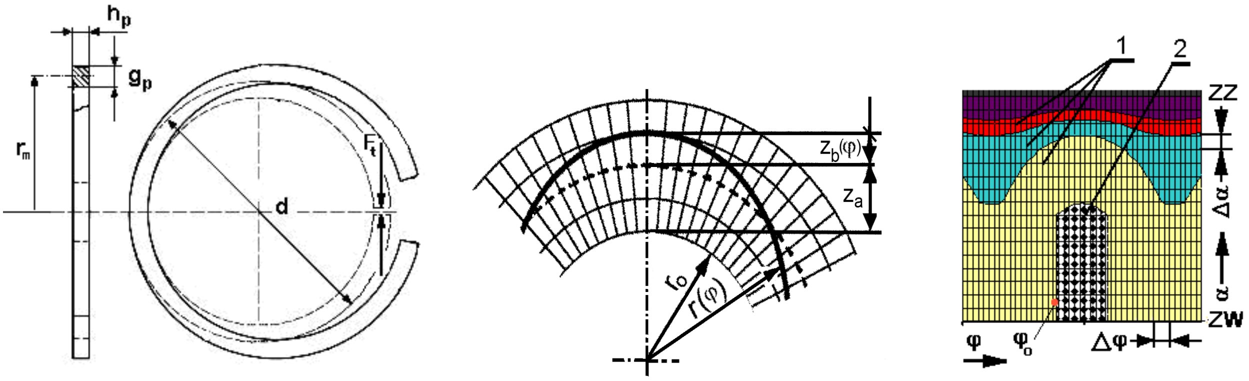


Fig. 1. Draft of ring and liner with symbols used in the paper: 1 – areas of similar ring wall pressure, 2 – areas of no pressure (light slot)

The deformed cylinder (or more precisely – course of its circumferential line) has been expressed as a sum of Fourier series harmonics

$$z_b(\varphi) = \sum_{h=1}^{n} A_h \cos(h\varphi + \delta_h)\,, \tag{4}$$

where the quantities of A_h and δ_h are amplitude and phase shift of the series successive harmonics h, respectively. It should be mentioned that the presented way of description of cylinder wall geometry assumes the invariable deformation along the entire cylinder height, which does not take into account the real shape of the profile (measurements on worn cylinders show a significant wear in the area of ring TDC).

Besides the cylinder deformation itself an even wear of its surface z_a occurs during long-time engine run, which also affects the ring circumferential wall pressure (Eq. 1).

Using dependencies (1) and (4), following equation has been obtained after necessary transformations:

$$p_m(\varphi) = \frac{E \cdot I}{h_p \cdot r_m^4}\left[K_z(\alpha)\cdot r_m - \left(z_a + \sum_{h=1}^{n}\left(h^2-1\right)^2 A_h \cos\left(h\cdot\varphi + \delta_h\right)\right)\right]. \tag{5}$$

Analysis of circumferential wall pressure distribution allows a definition of excessive pressure areas on cylinder wall as well as those where pressure is equal to zero and light slots can occur. A trial of definition of such areas extreme points by the angle φ_o defining position of points on cylinder circumferential line (as in Fig. 1c) was carried out in [8]. For a mathematic description of ring circumferential line with just one harmonic of Fourier series an angle lowest value coresponding to the beginning of slot is given by

$$\varphi_o(\alpha) = \frac{arc\cos\left(\dfrac{K_z(\alpha)\cdot r_m - z_a}{A_{h\cdot}(h^2-1)^2}\right) - \delta_h}{h}\,, \tag{6}$$

For more complex cylinder deformations (when a higher number of harmonics is needed for description of circumferential line) such a trial leads to more complicated dependencies and definition of demanded angles would be extremely difficult. A possible solution that leads to an approximate result is an expansion of a cosine function of Eq. 6 in a power series, according to [2]:

$$\cos x = 1 - \frac{1}{2!}x^2 + \ldots + \frac{(-1)^n}{(2\cdot n)!}x^{2n}\quad ,\qquad |x| < \infty\,. \tag{7}$$

Using only initial elements of the series (and assuming that $\delta_h = 0$) a p_m pressure has been defined as

$$p_m(\varphi) = \frac{E \cdot I}{h_p \cdot r_m^4}\left[K_z(\alpha)\cdot r_m - \left(z_a + \sum_{h=1}^{n}\left(h^2-1\right)^2 A_h - 0.5\varphi^2 \sum_{h=1}^{n} h^2\left(h^2-1\right)^2 A_h\right)\right], \tag{8}$$

as well as the minimum value of φ angle corresponding to the beginning of slot on cylinder circumferential line

$$\varphi_o(\alpha) = \sqrt{\frac{2\left[\sum_{h=1}^{n}\left(h^2-1\right)^2 A_h - K_z(\alpha)\cdot r_m + z_a\right]}{\sum_{h=1}^{n} h^2\left(h^2-1\right)^2 A_h}}\,. \tag{9}$$

Estimation of the relative difference between boundary angles calculated according to (6) and (9) for geometry of typical cylinder shows that it remains on the level of 5%.

Application of presented equation is limited only to the cases for which ring does not contact with liner surface with its entire circumference (or at least touches it without any pressure). This corresponds to the situation when the numerator of Eq. 9 has positive value. For a simple case when the cylinder deformation is described by a single harmonic the amplitude of deformation is given as

$$A_h = \frac{K_z(\alpha)\cdot r_m - z_a}{(h^2-1)^2}\,. \tag{10}$$

3. Determination of compression ring aganst the deformed cylinder liner – computational example

A presented further example concerns a trial of computational determination of compression ring wall pressure distribution against cylinder surface of increasing wear along with engine mileage.

Technical data and course of wear assumed in presented example concerns an engine of earth moving machine (International Harvester DT 466). Most important data of compression ring are presented in Table 1.

Tab. 1. *Technical data of exemplary IC engine compression rings*

Quantity		Ring
Cylinder diameter d	[m]	0.109
Ring neutral radius r_m	[m]	0.0522
Axial height h_p	[m]	0.003
Radial thickness g_p	[m]	0.0046
Young modulus E	[Pa]	$112 \cdot 10^9$
Mean pressure p_o	[MPa]	0.178
Tangential force F_t	[N]	27.6
Stiffness EI	[Nm^2]	2.65
Parameter K	[–]	0.0286

The course of cylinder circumferential line varies itself along with engine mileage and therefore requires a higher number of harmonics. In order to ease the analysis of results obtained the period of engine run has been divided in four stages corresponding with suitable sets of data z_a and z_b, (see Table 2).

Tab. 2. *Cases of cylinder deformation*

Case No	Wear z_a	Cylinder deformation z_b (A_h amplitude of the h harmonic)	Stages of cylinder life

	μm	μm	
1 (Fig. 4a)	0	-	A new cylinder, not worn and not deformed
2 (Fig. 4b)	0	$A_4 = 10$	Liner installation; cylinder deformation described by a single harmonic of 4th order
3 (Fig. 4c)	50	$A_2 = 30, A_4 = 15$	Initial stage of operation; worn and deformed cylinder
4 (Fig. 4d)	200	$A_2 = 100, A_4 = 25, A_6 = 5$	Further stage of operation; worn and deformed cylinder

Values of cylinder liner deformation and wear given in Table 2 were selected on the basis of measurements carried out on a real engine and the selection was aimed at achievement of unambiguous conclusions from the analyzed problem.

According to Eq. 5 the knowledge of the $K_z(\alpha)$ coefficient is needed for determination of ring wall circumferential pressure. Values of this coefficient (determined in [9]) corresponding to the engine full load operation (Fig. 2) are used in presented example.

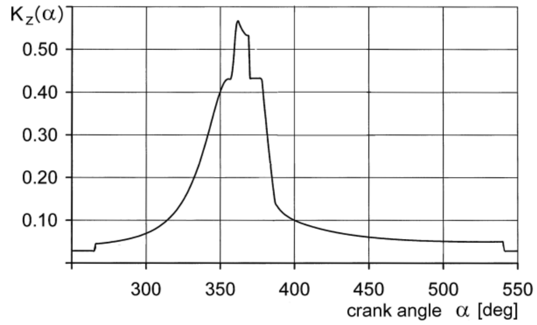


Fig. 2. Course of $K_z(\alpha)$ coefficient value vs. α angle at engine full load

Cylinder surface profiles corresponding to the cases collected in Table 2 are shown in Fig. 3 (for better clarity the z_a wear is not present). Assumption of phase shift makes that amplitudes of all harmonics taken into account sum up for $\varphi = 90^{\circ}$ (0.5π rad) angle.

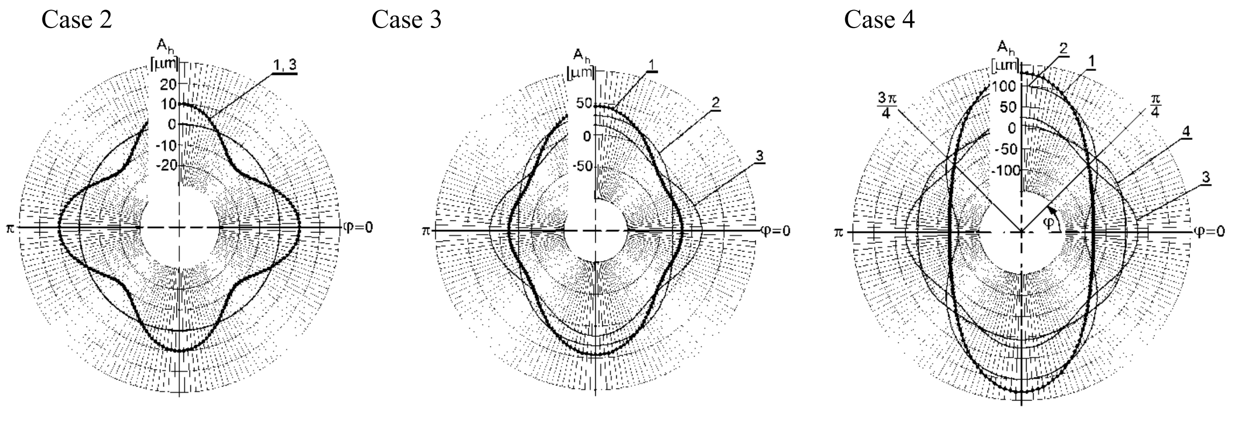
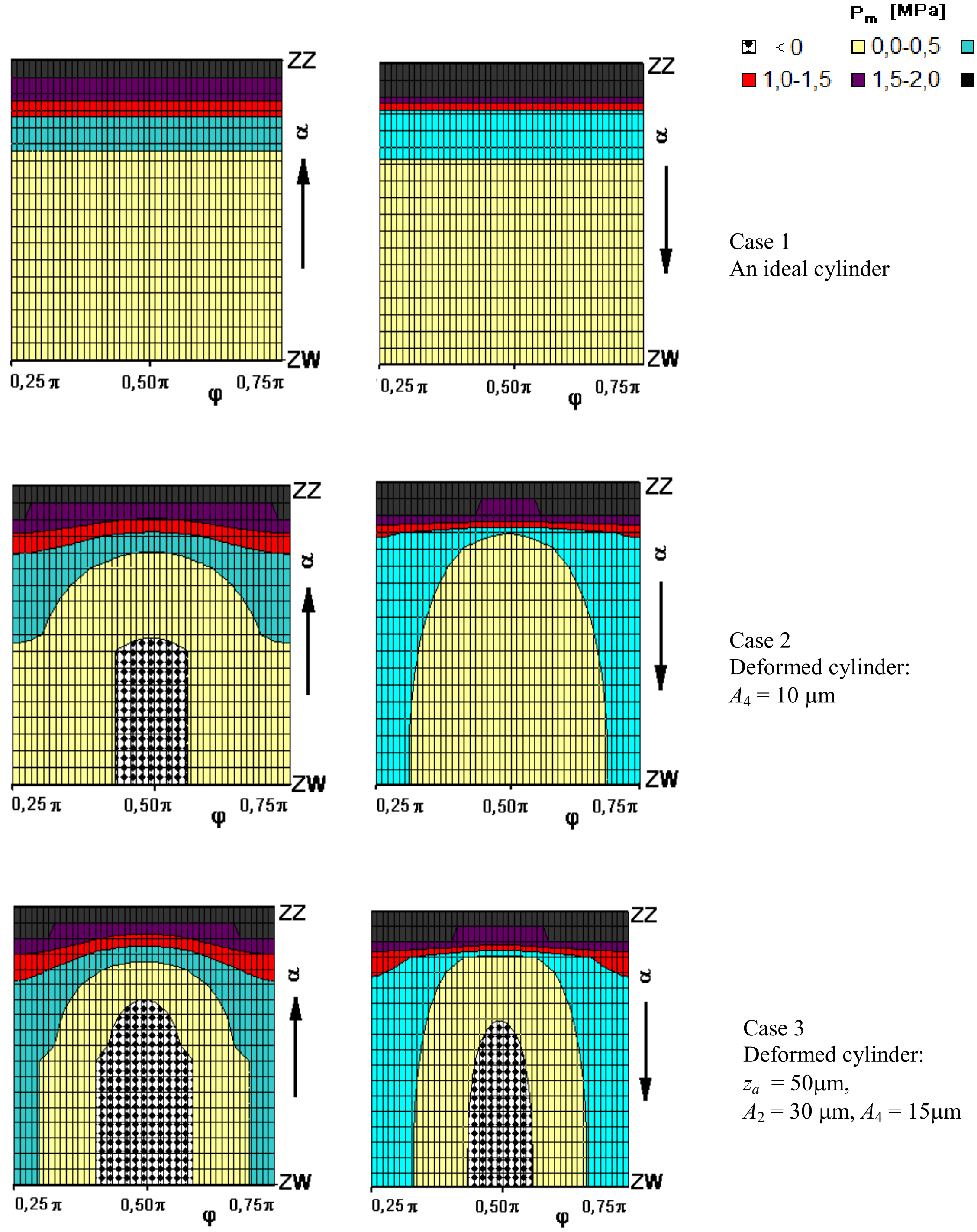
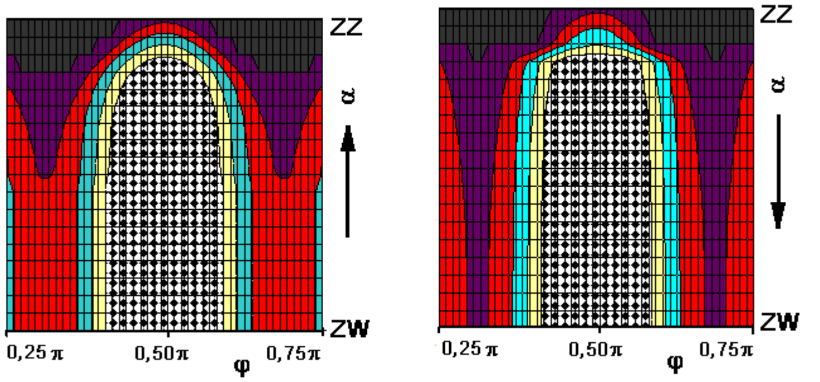


Fig. 3. The course of cylinder circumferential line (line 1) expressed as the sum of respectively: second(2), fourth (3) and sixth (4) harmonic of Fourier series determined for data collected in Table 2 (the z_a wear was not taken into account)

After determination of the cylinder circumferential line for each of analyzed cases corresponding to successive stages of engine operation the ring pressure distribution against the deformed cylinder wall was computed with particular attention paid to the areas where this pressure falls down to zero (Fig. 1c shows the way to identify this area).

As the result of carried out calculations the ring pressure distribution along the cylinder circumferential line was determined (using the angle increment $\Delta\varphi = 2°$) repeating calculations for consecutive positions of ring between the dead centers for stroke of compression and expansion (with the angle increment Δa = 10° CA). Charts presented in Fig. 4 cover the area of highest probability of light slot presence, i.e. the one for φ angle between 45 and 135° (0,25π and 0,75π rad, as indicated in Fig. 3c).





Case 4
Deformed cylinder:
z_a = 200 μm,
A_2 = 100 μm, A_4 = 25μm,
A_6 = 5μm

Fig. 4. Analyzed cases of ring wall pressure distribution

4. Summary and conclusions

The analysis of pressure distribution for selected stages of engine operation shows that differentiation of ring pressure increases along with the increase of wear z_a and deformation z_b of cylinder surface. For a new cylinder this distribution is even and changing itself under influence of gas forces (Fig. 4, case 1) undergoes the substantial differentiation even for minor deformation of its surface. Already for the 4th order harmonic of only 10 μm amplitude (case 2) an area of zero ring pressure appears during compression stroke and gas blow-by can occur. This area disappears during expansion stroke thanks to the gas pressure far higher than the ring elastic pressure. Further deformation increase results in rise of slot circumferential area, simultaneously encompassing higher and higher cylinder surface (beginning of this process is gradually situated close to TDC). The pressure distribution and increase in slot area is particularly unfavorable influenced by such cylinder deformation which requires consideration of Fourier series higher harmonics for its mathematical description (Case 4). This conclusion has been confirmed by the results of supplementary investigations (see Fig. 5) concerning the ring pressure against the deformed wall described by a single harmonic of the same amplitude (5 μm) but higher order. The investigation prove that the increase in harmonic number substantially affects the stress gradient value and size of slot area. However, it can happen for a higher number of harmonics that the achieved resultant could be lower than the one obtained for a single harmonic because of a mutual shift among the harmonics of different order.

Carrying out the additional research (no results in this paper) showed that even high values of cylinder wear z_a (of order of few hundred micrometers) affects the pressure distribution to far lower extent than the small deformations z_b (of order of several micrometers).

a) *b)* *c)*

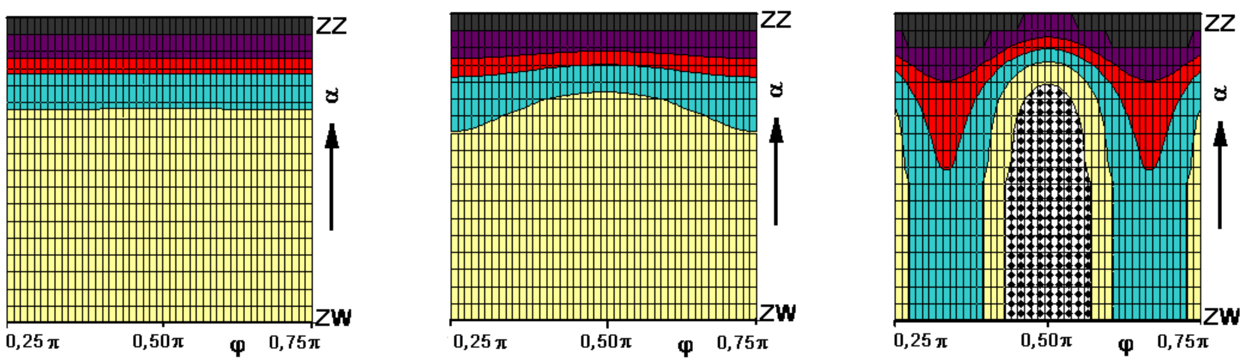


Fig. 5. Distribution of ring pressure against deformed wall described by a single harmonic of 5 μm amplitude obtained at compression stroke: (a) second order harmonic, (b) fourth order one, (c) sixth order one

Generally, on the basis of detailed conclusions and observations one can conclude that:

- ring elastic pressure provide its full contact with the cylinder wall only for minor deformations,
- ring wall pressure depends above all on gas pressure in strokes of compression and expansion, particularly close to the TDC,
- cylinder deformations that require description using harmonics of higher order cause higher gradients of ring wall pressure and favors the formation of light slots to higher degree than harmonics of lower order,
- more the course of circumferential line is close to the circle, lower is the risk of slots formation and blow-by,
- higher engine mileage affects its wear and deformations, and finally the increase of slots geometry and gas blow-by.

The presented investigations were carried out assuming the lack of oil layer on cylinder wall which should be stressed once again. Taking the oil film into consideration would result in complete or partial filling the slots with oil. This eventually will cause the change in pressure distribution and will lead to smaller area or even to complete disappearance of light slots.

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