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ANALYSIS OF RING PRESSURE DISTRIBUTION ON A DEFORMED CYLINDER FACE

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Abstract

Elastic properties of correctly designed piston compression ring should provide full contact of ring face and the cylinder surface. Actually, because of various phenomena and processes experienced during engine assembly and operation an initially cylindrical liner is being subjected to wear and deformations which eventually affects that contact and cause formation of slots distributed along the cylinder circumference.

Following paper describes the most often met deformations of cylinder and presents an evaluation of their influence on the process of compression ring collaboration with the surface of misshaped cylinder. Mathematical relations that allow to calculate the change of ring cylinder pressure and location of areas where blow-by can occur have been presented as well. The presented analyses were supplemented with charts illustrating changeability of certain quantities characteristic for ring and liner construction, using a marine engine ring as an example. The relations established during investigation will be used for a construction of mathematical model of phenomena accompanying the operation of piston-cylinder assembly elements, in the subject of blow-by in particular.

Keywords: *marine combustion engine, piston ring, ring elastic pressure*

1. Introduction

Guarantee of combustion chamber tightness is considered the most important among other tasks fulfilled by the engine labyrinth sealing. Although during engine run there always happen charge losses, but they could be minor ones when the collaboration of ring and liner is correct one. The mathematical models are constructed in order to properly design and monitor the operation of the labyrinth sealing, but authenticity of results acquired using such models depends on correctness of input data. Values of some of them are difficult to calculate or even to estimate. Geometry of slots along the cylinder circumference that cause the blow-by belong to this group. Information how to draw such leakiness can be found in literature. For example, for determination of the slot area S_{sz} between operating surfaces of ring and liner (assuming circular shape of ring) following expression could be used [3]:

$$S_{sz} = \frac{2 \cdot a \cdot h}{3} + \frac{3 \cdot H^2 \cdot h}{2 \cdot a} - \frac{3 \cdot H \cdot h^2}{2 \cdot a} + \frac{h^3}{2 \cdot a}, \tag{1}$$

where, according to Fig. 1:
a – chord where is no contact,
h – maximum height of slot between ring and cylinder,
H– maximum distance between liner and chord connecting slot ends.

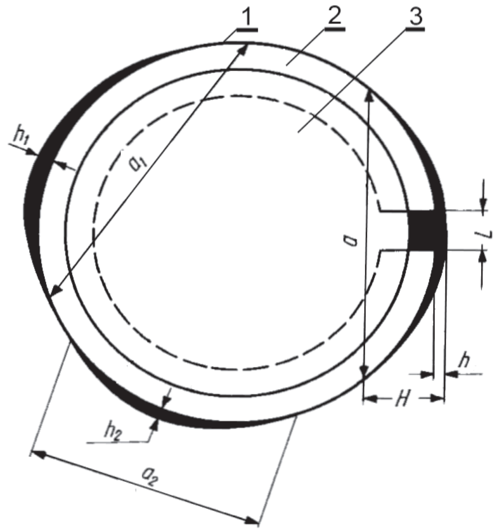


Fig. 1. Schematic of new ring contact with a worn cylinder (visible slots); 1 – cylinder face, 2 – compression ring, 3 – piston [3]

The total area of slots will equal the sum of individual slot areas distributed along the cylinder circumference and the area of ring gap.

The presented way of slot area calculations could be questionable because it does not take into consideration a number of influencing factors that change themselves during engine run. Ring elasticity which extorts its adjustment to a deformed surface of cylinder liner could be classified within this group of factors. This problem was chosen by the Author as the main subject of investigations.

The surface of a new cylinder liner could be compared to an ideal cylinder but the measurements show that even for a new liner its circumferential line differs from ideal circuit (it is a line resulting from a cut of a cylinder with a plane perpendicular to its axis). Among basic causes of these differences one should mention deformations created during liner assembly into cylinder block or those caused by fixing of cylinder head. Moreover, differentiation of cylinder shape could be observed even within a single engine which is a result of various, often difficult to recognize causes such as different thermal and mechanical loads, faulty operation of injection system, etc. Considerable cylinder deformations could be found along the cylinder generatrix. Their maximal values most often emerge within the area of collaboration of the first compression ring with the cylinder face when piston stays in the TDC. This phenomenon is being explained with a shortage in oil supply and absence of conditions favorable for formation of a continuous oil film. Because this problem was described in detail in earlier studies of the author [4,7], a proposed mathematical description of cylinder will be presented here. It should be noted that the presented studies could be treated as an introduction to further research and it is why several factors relative to engine run have been omitted as for example the oil film over cylinder surface or an effect of gas forces. It is very probable that taking into account influence of these factors the conclusions concerning possibilities of slot formation will change considerably.

2. Mathematical description of cylinder circumferential line

Mathematical notation of cylinder shape should contain a description of its circumferential line course and of a profile line. Deformation of cylinder face, relative to chosen plane of cylinder cross-section can be expressed as the difference between actual value of cylinder radius $r(\varphi)$ and the radius of new cylinder r_o (see Fig. 2):

$$\Delta r(\varphi) = r(\varphi) - r_o \,. \tag{2}$$

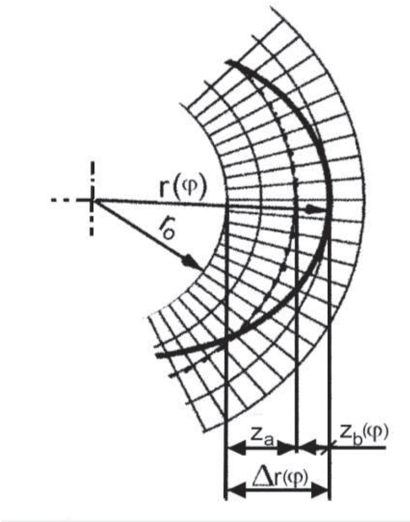


Fig. 2. Course of cylinder circumferential line with explanation of characteristic dimensions

Deformation of the circumferential line $\Delta r(\varphi)$ can be expressed as a sum of following components: z_a and $z_b(\varphi)$:

$$\Delta r(\varphi) = z_a + z_b(\varphi)\,, \tag{3}$$

where the z_a component will be further called the cylinder constant deformation (or even wear of cylinder face) whereas the $z_b(\varphi)$ component will be called cylinder deformation.

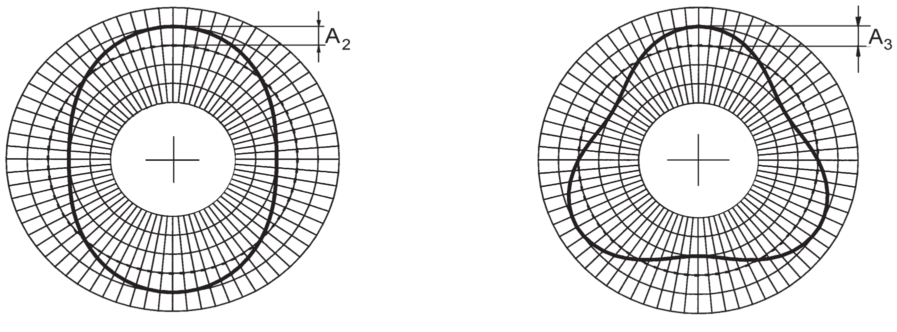
For a mathematic description of cylinder deformation one can use the Fourier harmonic series:

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

where A_h and δ_h are amplitude and phase shift of consecutive harmonics, respectively. Though a high number of harmonics is required for a precise description of circumferential line course, practically only dominant harmonics are taken into account, those of highest amplitudes (research proves that these are second and fourth harmonics [2]). Exemplary courses of the cylinder circumferential line specified with just one harmonic have been presented in Fig. 3.

As a result of the change in cylinder shape the ring pressure undergoes adjustment from initially constant ($p(\varphi) = p_z = const.$) to the one changing along the cylinder circumference. Definition of a circumferential variability of ring pressure requires a mathematical specification of cylinder curvature and the course of so called neutral line of a free ring. It is so because after ring installation in cylinder the curvature of its neutral layer changes itself from the one corresponding to the free form $v_p(\varphi)$ to another one corresponding to the curvature of cylinder $v_c(\varphi)$ (assuming full contact of ring and liner).

a) b)



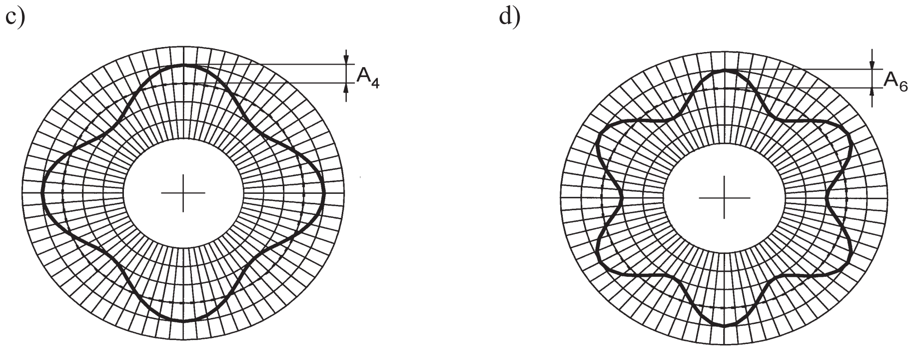


Fig. 3. Course of cylinder circumferential line relative to selected harmonics of the Fourier series: h = 2 (a), h = 3 (b), h = 4 (c),h = 6 (d), for $\delta_h = 0$

The cylinder curvature radius can be written in following form as it has been proved in [6]:

$$\nu_c(\varphi) = \frac{1}{r(\varphi)}\left(1 - \frac{z_b''(\varphi)}{r(\varphi)}\right), \tag{5}$$

while the curvature of free ring neutral layer is given as:

$$\nu_p(\varphi) = \frac{1 - K(1+\cos\varphi)}{r_m}, \tag{6}$$

where K is a ring characteristic parameter (the way of its determination was presented in [4]). The value of bending moment loading the ring is given in the form:

$$M_g(\varphi) = E \cdot I \cdot [\nu_c(\varphi) - \nu_p(\varphi)], \tag{7}$$

where E denominates the modulus of elasticity of a rod (the ring is equate to a rod), while I means an inertia moment of its cross-section. Using the earlier presented relations the formulas have been established that allow calculations of bending moment value $M_g(\varphi)$ and ring pressure $p_m(\varphi)$ at the point of ring neutral layer corresponding to the φ angle (the way of formulation of these equations was given in [6])

$$M_g(\varphi) = \frac{E \cdot I}{r_m^2} \cdot \left[K \cdot r_m \cdot (1+\cos\varphi) - z_a - z_b(\varphi) - z_b''(\varphi)\right], \tag{8}$$

$$p_m(\varphi) = \frac{E \cdot I}{h_p \cdot r_m^4}\left[K \cdot r_m - \left(z_a + z_b(\varphi) + 2z_b''(\varphi) + z_b^{(4)}(\varphi)\right)\right]. \tag{9}$$

Using the presented formulas one could try to determine a changeability of ring pressure against the deformed liner.

3. Determination of ring pressure on deformed liner

As mentioned before the cylinder circumferential line undergoes various changes due to numerous causes. In order to simplify further analyses this study deals only with cases where the cylinder deformation can be described with just one harmonic and the value of even deformation has been assumed as zero, i.e. $z_a = 0$.

With these assumptions it can be proved that relations describing changes in bending moment and ring pressure take on the following form (similar dependences were given in [1]):

$$M_g(\varphi) = \frac{E \cdot I}{r_m^2}\left[K \cdot r_m \cdot (1+\cos\varphi) + A_h \cdot (h^2-1)\cdot \cos(h\varphi + \delta_h)\right], \tag{10}$$

$$p_m(\varphi) = \frac{E \cdot I}{h \cdot r_m^4}\left[K \cdot r_m - A_h \cdot (h^2-1)^2 \cdot \cos(h\varphi + \delta_h)\right]. \tag{11}$$

Depending on the size of deformation amplitude A_h the ring pressure will change. When the amplitude does not exceed a certain value (called critical) the ring will contact along the entire circumference to the deformed cylinder (Fig. 4).

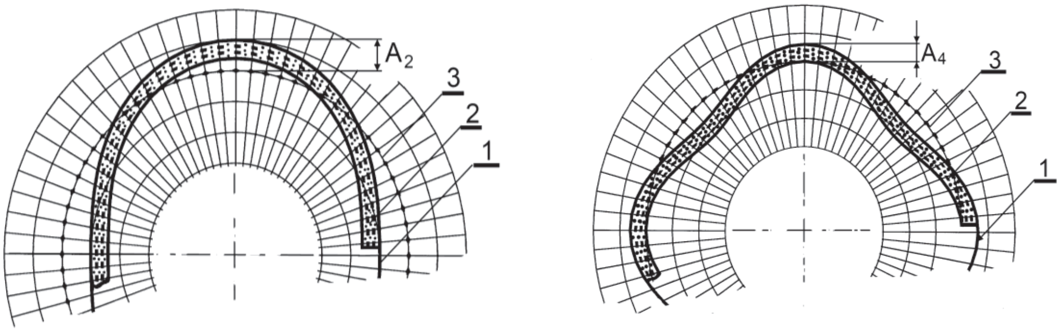


Fig. 4. A sketch of ring section touching the face of deformed liner: 1 – actual circumferential line of a deformed liner, 2 – ring section, 3 – circumferential line of a new ring (there is no proportion between size of elements and deformation in this figure)

Until after exceeding the critical value of amplitude A_{kr} areas where the pressure resulting from ring own elasticity is insufficient to push the ring against the liner (i.e. areas where ring pressure is naught or negative) will appear. An assumption has been adopted that due to this phenomenon slots between ring and liner will occur where blow by can take place.

According to (11) the $p_m(\varphi)$ pressure is equal to zero when following condition is fulfilled:

$$\cos(h \cdot \varphi) = \frac{K \cdot r_m}{A_h \cdot (h^2-1)^2} . \tag{12}$$

Taking into consideration that the value of cos(x) does not exceed 1 critical values of deformation amplitude for selected harmonics could be determined by the transformation of above formula:

$$A_{kr,h} = \frac{K \cdot r_m}{(h^2-1)^2} \tag{13}$$

Which means that the critical amplitude values of individual harmonics are:

$$A_{kr,2} = \frac{K \cdot r_m}{9}; \qquad A_{kr,3} = \frac{K \cdot r_m}{64}; \qquad A_{kr,4} = \frac{K \cdot r_m}{225}; \qquad \text{and so on.}$$

Higher the number of harmonic used for description of cylinder deformation lower the value of critical amplitude. For example, for the harmonic $h = 2$ the critical amplitude is about 25 times higher than the amplitude for harmonic $h = 4$.

In turn, the minimum value of the φ angle that define the slot limits can be calculated using following formula

$$\varphi_o = \frac{arc\cos\left(\dfrac{K \cdot r_m}{A_{h\cdot}(h^2-1)^2}\right) - \delta_h}{h} = \frac{arc\cos\left(\dfrac{A_{kr,h}}{A_{h\cdot}}\right) - \delta_h}{h}. \qquad (14)$$

This formula can be used only when the condition of $A_h \geq A_{kr,h}$ is fulfilled.

Values of the $\varphi_{o,i}$ angle (where i are the consecutive natural numbers) that determine the slot limits (areas where is no ring pressure) can be determined using a general formula:

$$\varphi_{o,i} = \frac{\pi \cdot (i+1)}{h} - \varphi_o \qquad \text{– for odd values of } i,$$

$$\varphi_{o,i} = \frac{\pi \cdot i}{h} + \varphi_o \qquad \text{– for even values of } i.$$

Presented below exemplary calculations were performed for a compression ring of marine engine (such ring was earlier a subject of investigation reported in [4]). Basic technical data are summarized in Table 1.

Table 1

Technical data of exemplary IC engine compression rings

Quantity		Ring (marine engine)
cylinder diameter d	[m]	0.480
ring neutral radius r_m	[m]	0.232
axial height h_p	[m]	0.015
radial thickness g_p	[m]	0.016
gap clearance m	[mm]	49.0
Young modulus E	[Pa]	$105 \cdot 10^9$
mean pressure p_o	[MPa]	0.063
tangential force F_t	[N]	219
stiffness EI	[Nm^2]	537.6
parameter K	[–]	0.0220

The effect of cylinder deformation size on a course of bending moment variations and ring pressure was evaluated in a course of calculations. Two cases of cylinder deformations were analyzed, namely when the cylinder circumferential line was described with harmonics of the 2nd and 4th order. The obtained results for selected values of cylinder deformation amplitude have been presented on graphs in Figs. 5 and 6.

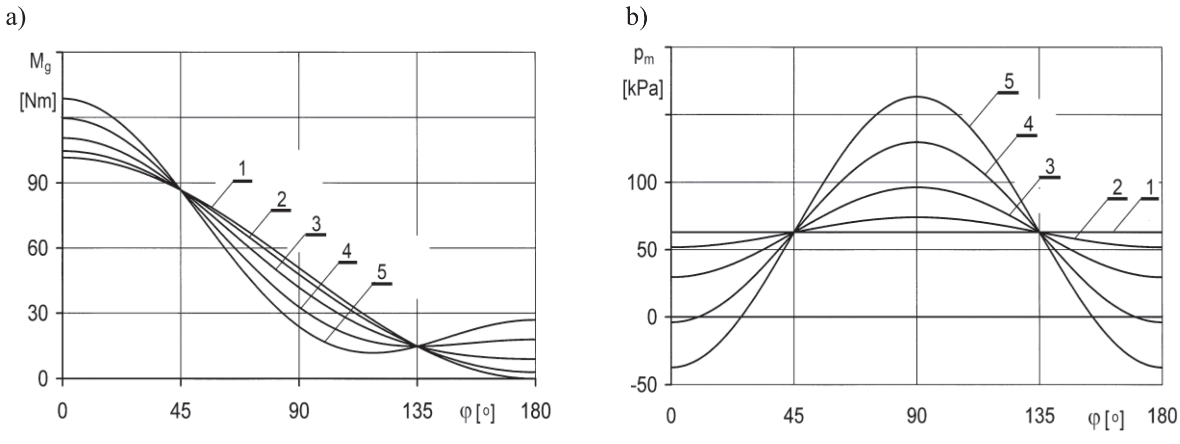


Fig. 5. Courses of bending moment Mg (a) and ring pressure p_m (b) defined along its circumference for selected values of deformation amplitudes A_2: 1 – 0 μm, 2 – 100 μm, 3 – 300 μm, 4 – 600 μm, 5 – 900 μm

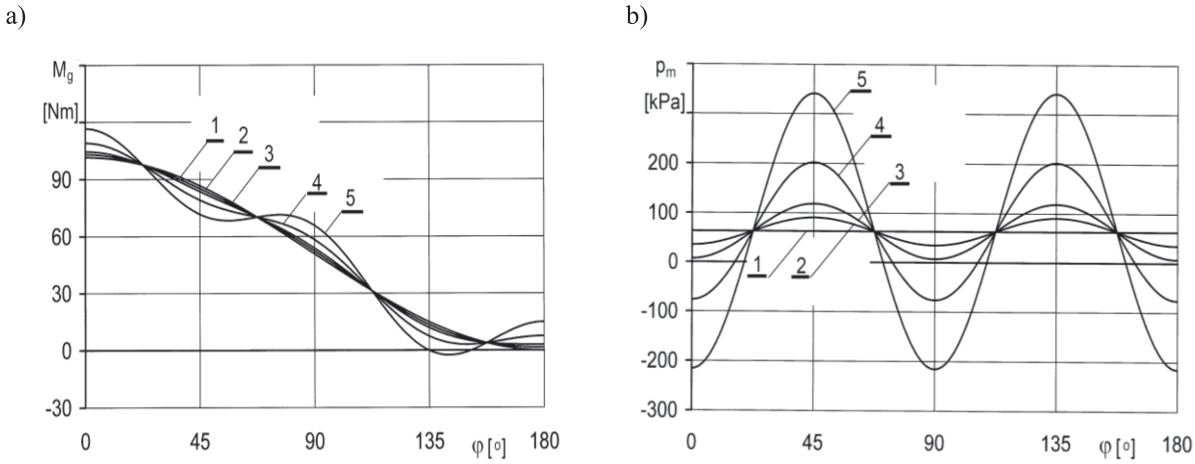


Fig. 6. Courses of bending moment Mg (a) and ring pressure p_m (b) defined along its circumference for selected values of deformation amplitudes A_4: 1 – 0 μm, 2 – 10 μm, 3 – 20 μm, 4 – 50 μm, 5 – 100 μm

The curves course shows that within regions where the increase in amplitude leads to lesser cylinder diameter the ring pressure increases while for bigger diameter this pressure decreases. There is a critical value of deformation amplitude for which the ring is not pushed against the liner by its own elasticity though it still touches the cylinder face (value of these amplitudes calculated according to Eq. (13) are summarized in Table 2.

Table 2

Values of $\varphi_{o,i}$ angle for selected values of 2[nd] and 4[th] harmonics

	h =2 $A_{kr,2}$ = 567 μm		h =4 $A_{kr,4}$ = 22,7 μm				
	$\varphi_{o,i}$ [rad]/[°]		$\varphi_{o,i}$ [rad]/[°]				
A_i [μm]	i = 0	i =1	A_i [μm]	i = 0	i = 1	i = 2	i = 3
300	–	–	20	–	–	–	–
600	0.166 / 9.50	2.97 / 170,5	50	0.275 / 15.7	1.29 / 74.3	1.85 / 105.7	2.86 / 164.3
900	0.445 / 25.4	2.69 / 154.5	100	0.335 / 19.2	1.23 / 70.8	1.91 / 109.2	2.81 / 160.8

An increase of the amplitude value over its critical value leads to an increase in area of ring contact absence described with angles $\varphi_{0,i}$ (14). These values of angle (only for a section of ring) are presented in Table 2 and in Fig. 7 (only selected ones).

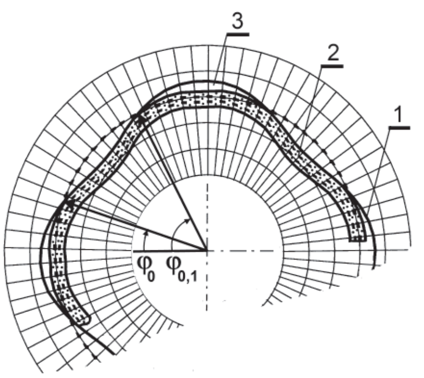


Fig. 7. Sketch of ring section in deformed cylinder: 1 – actual circumferential line of deformed cylinder, 2 – section of ring, 3 – slot hypothetic location ; $\varphi_{0,i}$ – angles defining boundary points of ring to deformed cylinder face contact

Within the areas of ring-liner contact so called light slots appear. Their geometry depends on the course of ring neutral line. Definition of course of that line, consequently the area of slots, is a complicated problem which will be discussed in further papers.

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