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VARIABILITY OF COMPRESSION RING PRESSURE AGAINST THE DEFORMED CYLINDER WALL DURING ENGINE OPERATION

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Abstract

Problems connected with the circumferential distribution of compression ring pressure against the deformed cylinder wall during engine run were presented in this paper. In particular, division of the resultant force pressing the ring against the wall into component connected with ring own elasticity and the one originated from gas pressure was carried out and the influence of each force at individual phases of engine run was evaluated as well. The relationship allowing calculation of the circumferential ring pressure was also established. The paper contains figures and drafts obtained in a course of exemplary calculations of compression ring pressure concerning a running bulldozer engine. These figures show the areas of cylinder wall where so called light slots can appear.

Conclusions resulting from the carried out simulation tests are discussed in Summary. The need for further investigations mentioned in presented study was shown as well.

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

1. Introduction

As soon as new cylinder liner is being installed a considerable deformation can occur caused by an erroneous installation in cylinder block. These deformations can grow to considerable size (of a few dozen micrometers [2]) during engine operation when the liner is being exposed to the influence of changing loads. Examples of such effect are presented in Fig. 1.

The compression ring is being pressed to the cylinder wall with the radial forces that result from material own elasticity and gas pressure acting upon ring surfaces (see Fig. 2). While the elasticity force F_s remains roughly constant during engine run, the other force F_g changes itself. Investigations on the compression ring position in ring groove allow to conclude [1, 4] that the ring is pressed to the groove bottom side on the prevailing section of stroke which means that the acting gas pressure is almost the same as the pressure in combustion chamber (p_a). Only on short sections of ring path the ring loses its contact with the bottom side of groove which causes that the F_g force value depends above all on pressure p_b in space below the ring.

On the other hand the ring is separated from the cylinder wall with the F_f force resulting from the hydrodynamic pressure of oil film. The value of mean pressure is given by the equation [5]

$$p_s = p_a + (p_b - p_a)W_p + \frac{\eta \cdot u \cdot b_f}{h_m^2} W_u + \frac{\eta \cdot v \cdot b_f^2}{h_m^3} W_v, \tag{1}$$

where:

- p_a, p_b – external pressures acting on the ring,
- u, v – ring velocities, axial and radial respectively,
- η – viscosity of lubricating oil,
- h_m – oil film minimum thickness,
- b_f – height of ring face covered with oil film,
- W_i – load capacity coefficients, that allow to consider the effect of external pressures (p index), slide (u) and squeeze (v); their values depend on the profile of ring face.

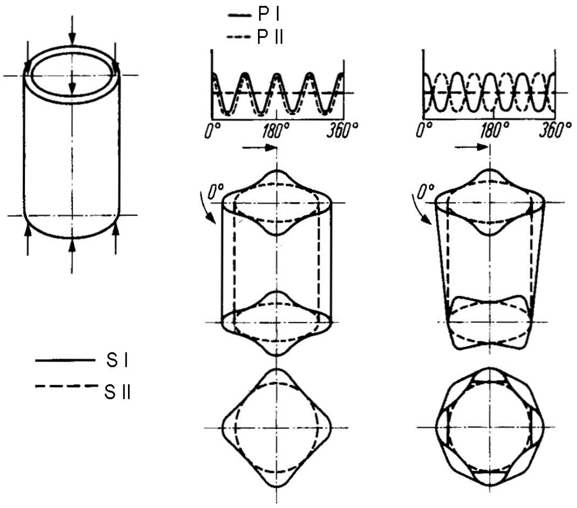

Fig. 1. Deformations of cylinder sinusoidally loaded along its circumference; P I and P II – load of upper and lower part of ring, respectively, S I and S II – form of bore before and after loading [3]

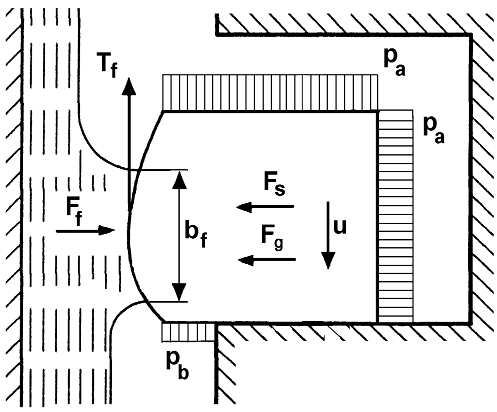

Fig. 2. Sketch of compression ring moving relatively to cylinder wall; p_a, p_b – external pressure, F_s – force of ring own elasticity, F_g – force equivalent to gas forces loading ring, F_f – reaction of oil layer, T_f – friction between ring and oil layer, u – ring speed, b_f – ring axial height covered with oil film

If the value of oil film pressure was not enough to balance the forces pressing the ring against the wall the film rupture would occur and the ring would collaborate with wall in conditions of mixed or even boundary lubrication. In case of incomplete contact of ring face and cylinder wall the circumferential slots can occur which lead to the gas blow-by. This can cause the drop in engine power and an excessive oil consumption (which in turn leads to the increase in exhaust emission). Another task performed by rings, namely the heat conduction from piston to cylinder deteriorates as well.

In his earlier papers author evaluated the ring pressure circumferential distribution against the deformed cylinder wall caused only by the ring own elasticity [7]. This pressure is additionally strengthened by the exhaust on the running engine which can lead to a different collaboration of these parts. Due to that the determination of combined effect of elasticity and gas force on pressure distribution and eventual definition of areas of ring and deformed wall contact was chosen as the goal of described tests.

In order to realize the given aim the model tests were performed in which an extremely unfavorable case was selected, i.e. the lack of continuous oil film over the cylinder wall to eliminate the disturbance factors. Though this assumption is merely of theoretical character (engine operation in such conditions would lead to a quick failure of collaborating parts) the obtained results allow to estimate value of some parameters indeterminable with other methods. One can mention here the forces loading radially the ring indispensable for full contact of ring and arbitrarily deformed cylinder wall.

2. Ring to wall pressure from elastic and gas forces

It was proved in [8] that the ring circumferential pressure distribution $p_m(\varphi)$ over the deformed cylinder wall can be determined according to the formula:

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

where:

- E – Young's modulus,
- I – inertia moment of ring cross section,
- r_m – radius of neutral layer of ring in cylinder,
- h_p – ring axial height,
- z_a – constant deformation of cylinder,
- $z_b(\varphi)$ – deformation of cylinder wall, variable along the cylinder circumference (and its second and fourth derivative),
- K – characteristic coefficient of ring given with the formula

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

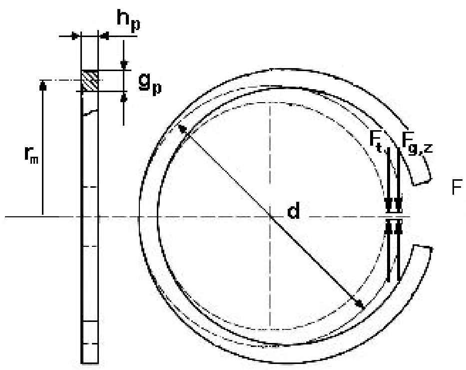

Fig. 3. Sketch of a ring with marked dimensions and position of tangential force F_t and resultant gas force $F_{g,z}$

The value of K coefficient calculated according to formula (3) is just a reflection of ring dimensions and its material properties. However, taking into account the aim of investigation a trial was undertaken to such define the K coefficient that it would describe not only its elasticity but also the effect of exhaust gases. Using the following formula

$$K = \frac{F_t \cdot r_m^2}{E \cdot I}, \tag{4}$$

that is another form of the formula (3) from [6], following relation has been obtained that allow to calculate a modified form of the K coefficient (because of its relation to crankshaft angle α further denoted as $K_z(\alpha)$):

$$K_z(\alpha)=\frac{\left(F_t+F_{g,z}(\alpha)\right)\cdot r_m^2}{E\cdot I} \qquad \text{or} \qquad K_z(\alpha)=K+\frac{p_g(\alpha)\cdot h_p\cdot r_m^3}{E\cdot I}\,, \tag{5}$$

where:

- F_t – tangential force fixed in the plane of neutral axis at ring free gap,
- $F_{g,z}(\alpha)$ – force fixed in the plane of neutral axis at ring free gap, equal to the gas force loading the ring,
- $p_g(\alpha)$ – exhaust gas pressure loading the ring,
- α – crankshaft angle.

Gas pressure loading the internal and working ring surface depends on the instantaneous pressure over (p_a) and below (p_b) the ring and on the position of ring in groove. For an initially adopted assumption about the lack of continuous oil film over the liner surface a linear drop in pressure along the ring face axial height was assumed. Due to that one can distinguish two cases for which the approximate value of gas pressure loading the ring is:

$$p_g(\alpha)=0.5\cdot\left(p_a(\alpha)-p_b(\alpha)\right), \tag{6a}$$

when the ring leans on the bottom side of groove and

$$p_g(\alpha)=0.5\cdot\left(p_b(\alpha)-p_a(\alpha)\right), \tag{6b}$$

when the ring leans on the upper side of groove.

When the pressure over and below the ring is equal, the resultant pressure is zero, which means that both characteristic parameters are the same, i.e. $K = K_z(\alpha)$.

3. Calculations of compression ring pressure on the deformed cylinder wall

The presented below example concerns a trial of computational definition of compression ring wall pressure distribution against the deformed cylinder of earth moving machine engine (DT 466 International Harvester). Technical data of this ring have been 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]	$112\cdot10^9$
mean pressure p_o	[MPa]	0.178
tangential force F_t	[N]	27.6
stiffness EI	[Nm^2]	2.65
parameter K	[–]	0.0286

According to the primary goal of this study the further presented results concern an evaluation of the effect of gas forces on ring contact with a deformed cylinder wall. However, these deformations were limited to the one case, namely a circumferential line like in Fig. 1 described with the formula

$$z_b = A_4\cos(4\varphi+\delta_4)\,, \tag{7}$$

where A_4 and δ_4 are amplitude and phase shift of the Fourier series fourth harmonic, respectively (see Fig. 4).

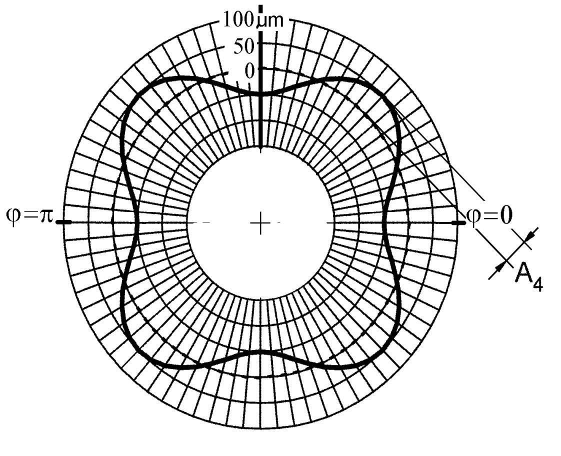

Fig. 4. Course of cylinder circumferential line for $A_4 = 50$ µm and $\delta_4 = \pi$

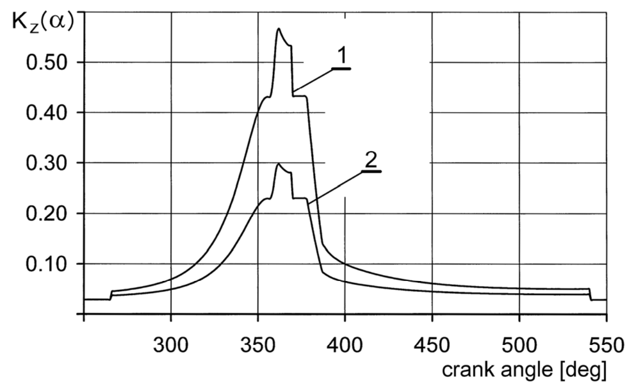

Fig. 5. Course of $K_z(\alpha)$ coefficient value vs. crank angle for full (1) and partial (2) engine load

According to the analysis presented in Chapter 1 it was assumed that the ring is pressed against the bottom side of groove during the stroke of compression and expansion. The ring position during other strokes has been regarded as less important for the analyzed example because of far lower gas pressure (the radial position of ring depends above all on the ring own elasticity).

Knowledge of engine constructional data as well as its operational conditions allowed to evaluate on the course of theoretical computations the pressure over and beneath the ring and eventually calculate the value of modified coefficient corresponding to the consecutive positions of piston (only a fraction of these variations were presented in Fig. 5). The calculations of gas forces were carried out for engine full and partial load.

Next, using Eqs. (2) and (7) the dependence (8) was defined that allows the calculation of ring wall pressure at an arbitrary point of cylinder circumference (φ) and for a chosen crankshaft angle (α).

$$p_m(\varphi) = \frac{E \cdot I}{h_p \cdot r_m^4}\left[K_z(\alpha) \cdot r_m - 225 \cdot A_4 \cdot \cos(4\varphi + \delta_4)\right]. \tag{8}$$

Exemplary results presented further were achieved for the fourth harmonic amplitude which equals 10, 20, 50 and 100 µm, respectively.

The lines seen in Fig. 6 and 8 show the places on cylinder wall where the ring pressure falls to zero for a given engine load and amplitude of cylinder wall deformation (because of ring symmetry only its section from 0 to $\pi/2$ was shown). It was assumed that pressure equal or lower than zero means no contact of ring and wall.

As it results from courses presented in Fig. 6 the area of ring face and cylinder wall contact increases along with the increase in ring wall pressure (caused by the increase of gases in combustion chamber). For example, during compression stroke for the cylinder deformation amplitude equal to 10 µm the elastic ring pressure alone does not secure such contact. As soon as the gas pressure reaches the required magnitude which relates to the crank angle 260 degrees the contact is full. For the amplitude A_4 equal to 20 µm the same effect is visible at 300 degrees CA while for 100 µm the full circumferential contact appears close to TDC. During expansion stroke the areas of full and partial contact have different shape because of different angular distribution of gas pressure. Reference [7] provides dependences that allow to calculate the amplitude boundary value of any harmonic which secures a positive ring wall pressure. On that basis the courses were drawn which allow to define the value of amplitudes for the analyzed example.

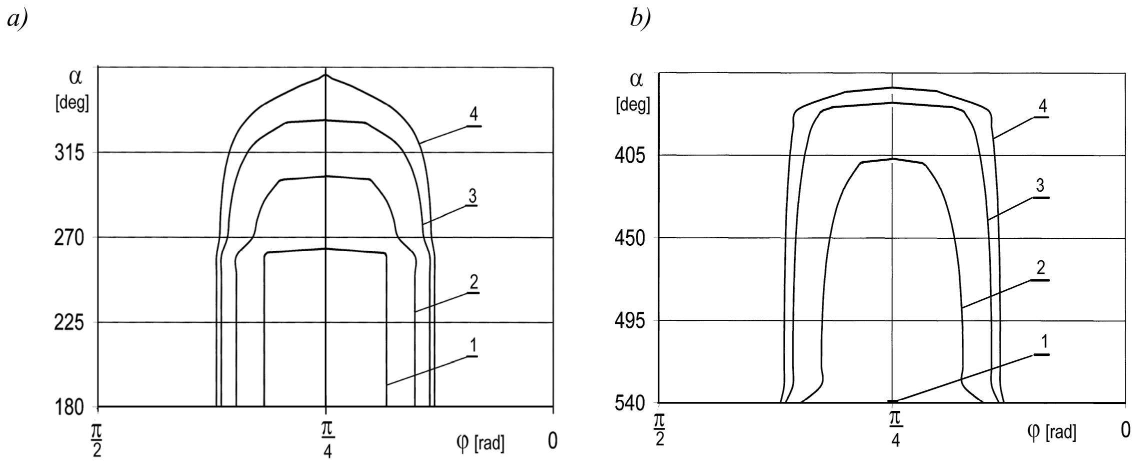

Fig. 6. Course of boundary lines of positive ring wall pressure for ring moving during stroke of compression (a) and expansion (b) for engine full load and selected values of cylinder deformation; A_4: 1 – 10 μm, 2 – 20 μm, 3 – 50 μm and 4 – 100 μm

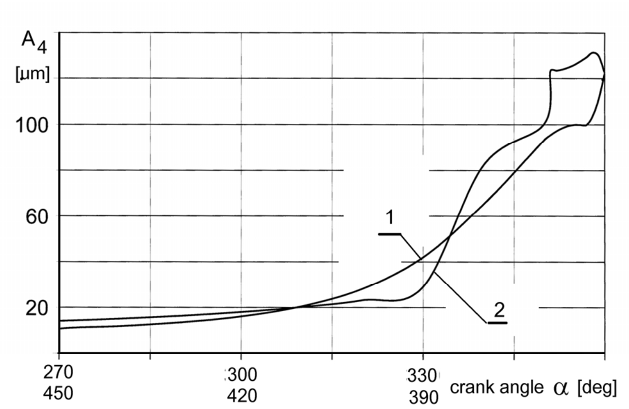

Fig. 7. Fourth harmonic amplitude value vs. crank angle for engine full load at: 1 – compression stroke, 2 –expansion stroke

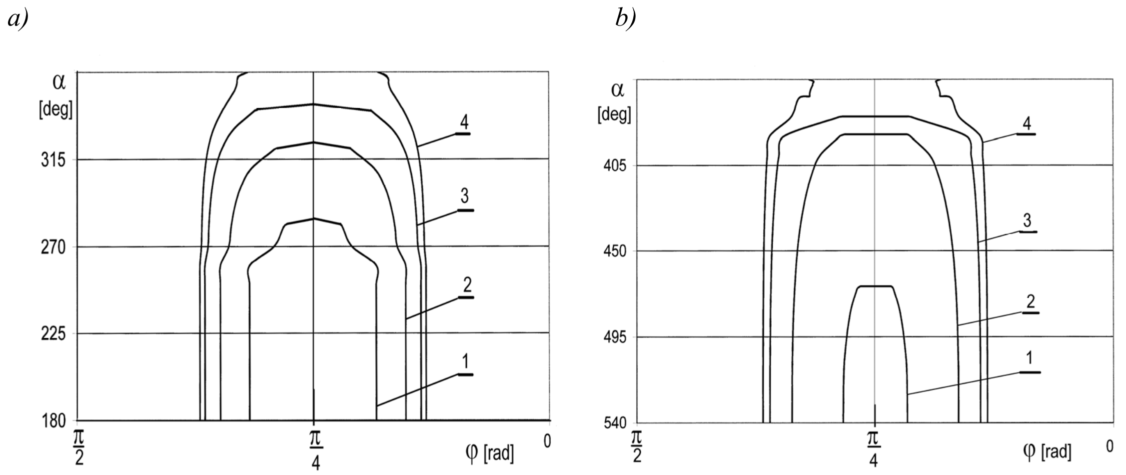

Fig. 8. Course of boundary lines of positive ring wall pressure for ring moving during stroke of compression (a) and expansion (b) for an engine partial load and selected values of cylinder deformation; A_4: 1 – 10 μm, 2 – 20 μm, 3 – 50 μm and 4 – 100 μm

Along with the decrease in engine load and reduction of gas forces the ring wall pressure also decreases which one can observe in the course of ring boundary pressure (Fig. 8). For example, for the amplitude of 100 μm and engine full load the ring touches the wall with its entire circumference close to the TDC. For the same amplitude of cylinder deformation and engine reduced load the light slot within a considerable angle of cylinder circumference emerges at the same area.

4. Summary and conclusions

Taking into consideration the results of theoretical studies and results of exemplary calculations following conclusions can be drawn:

- the ring wall pressure caused by gas forces exceeds several times the pressure from ring own elasticity at strokes of compression and expansion,
- the ring elastic pressure ensures the full wall contact only for minor deformations of cylinder wall (less than 10 μm for the analyzed example),
- for major deformation of cylinder wall ring pressure depends mainly on gas forces during strokes of compression and expansion close to the TDC in particular.

It is worth mentioning that the presented studies concern the case of no lubrication. When the presence of lubricating oil is taken into account, the minor slots are filled with oil and gas blow-by would not happen.

Another problem that should be the subject of eventual tests but was completely ignored in presented study is the analysis of stress in ring material during ring collaboration with deformed (worn) cylinder wall. Such tests should define the border values of cylinder deformations which would not lead to the ring failure.

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