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DETERMINATION OF COMPRESSION RING WALL PRESSURE DISTRIBUTION

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Abstract

On a correctly designed engine piston-cylinder assembly the contact of ring and bore should take place through a layer of oil, called oil film. In order to obtain a continuous oil film a proper lubricating oil should be introduced into the region of node elements collaboration, sliding surfaces should have adequate geometry and parameters of collaboration should be chosen suitably. The ring pressure against the liner is one of important quantities that affect formation of oil film. Selection of ring pressure circumferential distribution is pretty complex and depends on a number of factors and changes along the engine life.

Presented paper discuss the methods of ring pressure distribution along its circumference and indicate problems connected with measurements. Moreover, basic assumptions used for construction of compression ring mathematical model as well as results obtained using that model were presented for full and partial loads. A need for the construction of computational program that could take into consideration evenly worn and distorted bore surface have been validated as well.

Keywords*: marine combustion engine, piston ring, oil film, ring pressure*

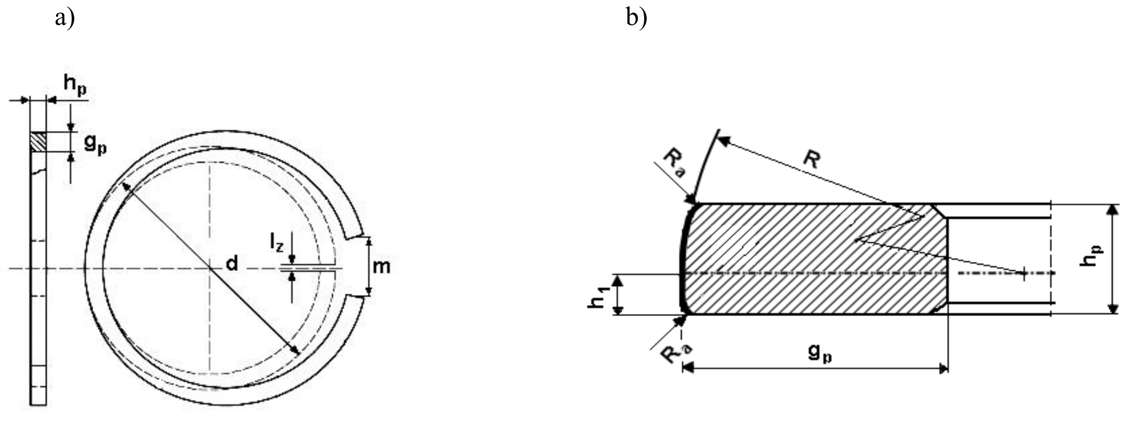
1. Introduction

A required distribution of ring pressure against the bore is being established already at design stage. This pressure should be enough high that ring would precisely contact the bore (without so called light slits where blow-by could occur). Only minor deviations from circular form could occur assuming that possible slits will be filled with lubricating oil. In the case of large diameter engines (marine ones) clearance could reach up to 30 μm, but only along 10% of ring circumference.

For less precise calculations a tangential force F_t is taken for a measure of ring pressure, i.e. force that put to the ring free ends makes them meet to achieve a required clearance l_z (see Fig. 1).

Another requirement that should be met by compression rings is ability to form a continuous oil film. According to the hydrodynamic theory of lubrication such film can occur only when between two moving surfaces there is a gap filled with oil.

Definition of ring momentary position relative to bore requires a knowledge of radial forces loading the ring [7]. The force pressing the ring on bore F_s (resulting from outside pressure p_a and p_b, and ring own elasticity $p(\varphi)$) and F_f force radial component moving ring away (resulting from hydraulic pressure p_f acting in oil film – see Fig. 2) belong to the most important ones.



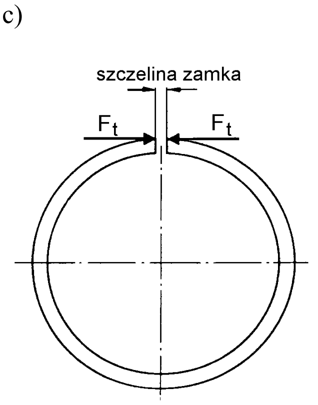


Fig. 1. Sketch of compression ring with marked principal dimensions and points of tangential force F_t application

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Depending on momentary values of such quantities as ring moving speed u, oil viscosity η,

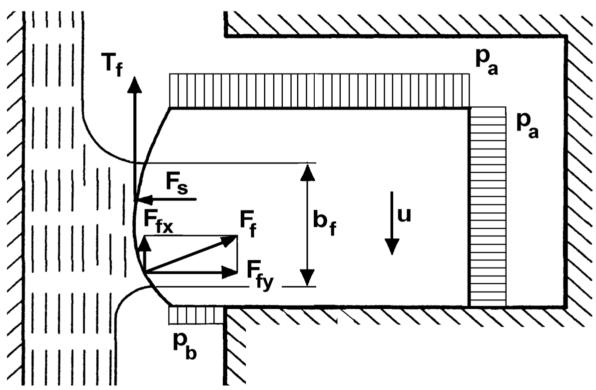


Fig. 2. Sketch of ring moving over bore surface: p_a, p_b – external pressure, F_s – pressing on force, F_f – oil layer reaction force, T_f – ring on oil friction force, u – speed of ring run [6]

ring total pressure against the bore p_c, ring axial width b_f and ring face shape described by the W_u parameter, the minimum oil film thickness h_m changes according to formula (1).

$$h_m = \sqrt{\frac{\eta \cdot u \cdot b_f}{p_c} W_u} \ . \tag{1}$$

Despite the relatively low radial speed u and high total pressure p_c on marine engines, the oil film of thickness securing complete separation of collaborating surfaces emerges during ring travel along the bore. This results above all from big size of these rings in comparison to the rings of other categories of engines. On contemporary high power engines the compression rings have a barrel profile (of curvature radius R up to 1 m) at axial width h_p up to 0.02 m and radial thickness g_p to 0.03 m [2, 8]. As a result, the specific pressure p is not high (for bore larger than 0.5 m it does not exceed 0.1 MPa) in spite of high value of tangential force F_t.

2. Methods of ring to bore pressure measurement

The measurement of tangential force F_t value is performed by special, simple devices (e.g. like that shown in Fig.3). During measurement the tested ring is surrounded by steel band (of 0.08 to 0.1 mm thickness) and tightened till reach an operational clearance l_z. In order to eliminate friction between ring and band the collaborating surfaces are covered with lubricating oil while the reduction in static friction is obtained by the multiple tightening of the band. The deflection of elastic beam 2 means the measure of tangential force F_t.

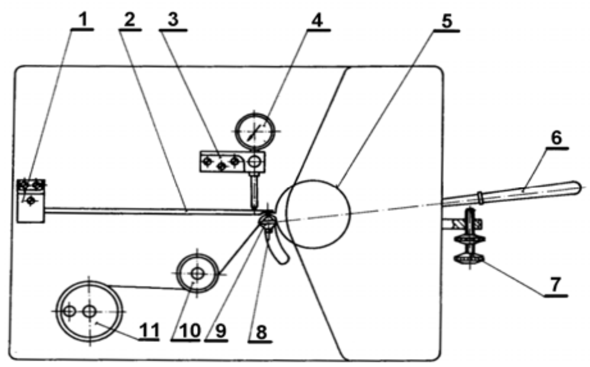


Fig. 3. Schematic of device purposed for ring tangential force measurement: 1 – beam hold, 2 – elastic beam, 3 – sensor hold, 4 – micrometer, 5 – steel band, 6 – lever handle, 7 – limit screw, 8 – screw, 9 – band hold, 10 – rotating drum, 11 – tightening drum [3]

Knowledge of the tangential force F_t value allow to determine the ring elasticity numerically equal to the ring average pressure against bore p_s.

$$p_S = \frac{2 \cdot F_t}{d \cdot h_p} \ . \tag{2}$$

The described method allows only for an estimation of p_s pressure mean value which corresponds to actual pressure $p(\varphi)$only when this pressure in evenly distributed (Fig. 4). For a ring of uneven pressure distribution, the form of ring mounted to the bore would not be perfectly cylindrical despite the correct clearance between ring ends.

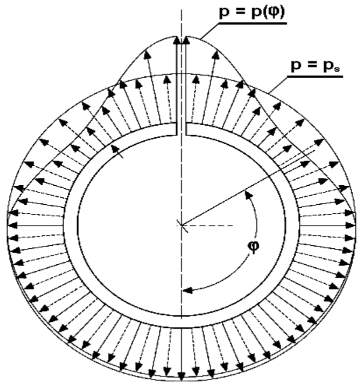


Fig. 4. Uniform ($p = p_s$) and irregular ($p = p(\varphi)$) distribution of ring circumferential pressure [4]

The device presented in Fig. 5 is used for a precise measurement of circumferential pressure distribution.

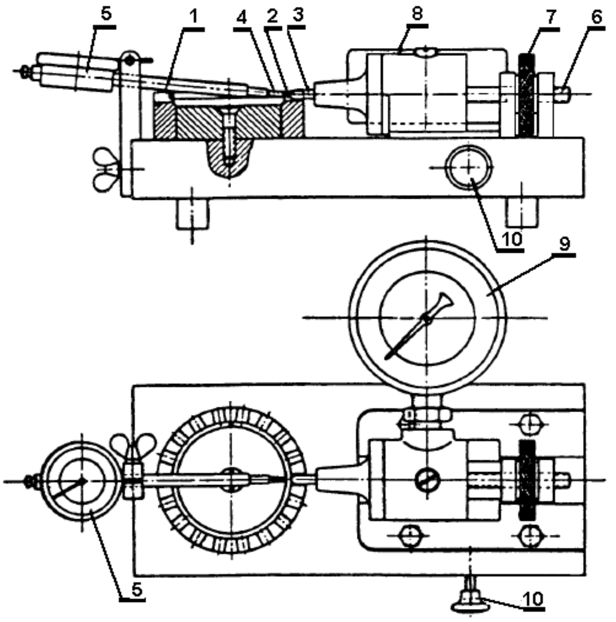


Fig. 5. Construction of device for the measurement of ring contact pressure distribution: 1 – model of cylinder liner, 2 – piston ring tested, 3 – pressing on mandril, 4 – measuring tip of displacement sensor, 5 – dial gauge, 6 – load adjustment screw, 7 – nut, 8 – hydraulic cylinder, 9 – manometer, 10 – release screw [1]

Along the circumference of cylinder liner 1 which encompasses the tested ring 2 there are holes through which mandrels 3 loads radially the ring. A hydraulic arrangement 8 equipped with measuring manometer 9 is used for generation of radial force. When ring loses its contact with a cylinder model (this moment is detected by the dial gauge 5) a value of radial force is being read. Repetition of measurement for consecutive ring positions enables to estimate the distribution of circumferential pressure.

3. Calculations of ring circumferential wall pressure distribution

Author has developed a computer program for analysis of piston rings collaborating with cylinder liner, especially dependences that occur between ring material, its geometry and elasticity. This program contains subroutines that enable to:

- determine the ring free form for a known distribution of pressure against the bore (Fig. 6a),

- determine the distribution of pressure against the bore for a known ring free form (Fig. 6b),
- determine the distribution of pressure and form of ring for a known geometry of bore (case of radial wear - Fig. 6b),
- determine the distribution of pressure and form of ring for a known geometry of bore (case of deformation - Fig. 6b).

The program has been developed according to assumptions given and explained in [1].The essence of the method lies in substitution of real ring with its computational model, consisting of a number of rigid rectilinear sections connected one to another by joints. It is assumed that bending of substitute ring is possible only in joints and substitutional rigidity should correspond to the real rigidity dispersed along the ring entire circumference (see Fig. 7).

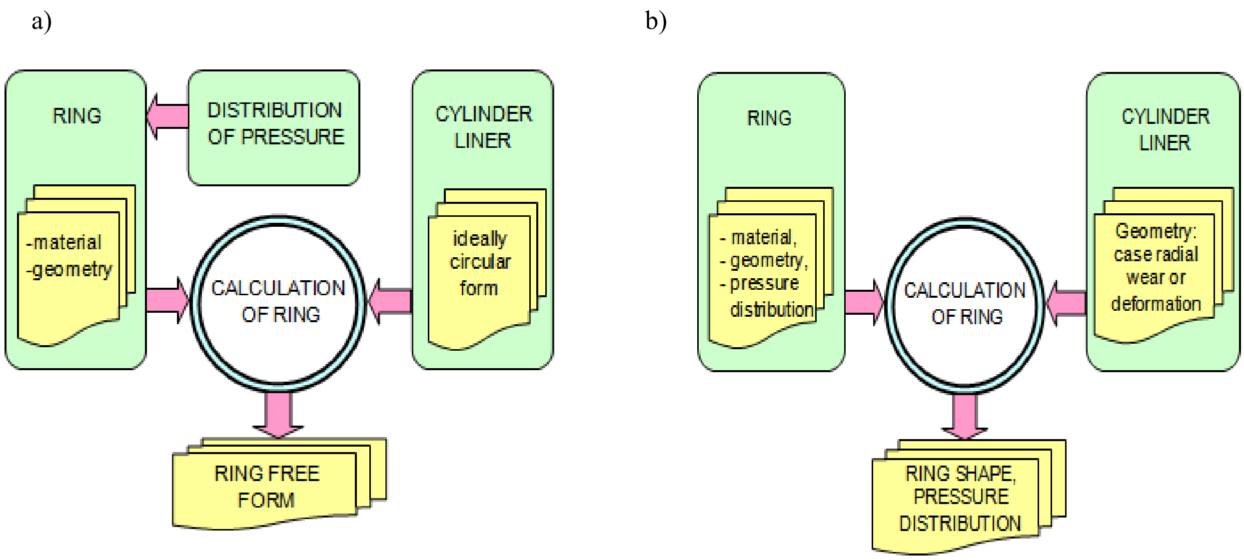


Fig. 6. Flow chart of computer program: a – definition of ring free shape, b – definition of pressure form and distribution in deformed bore

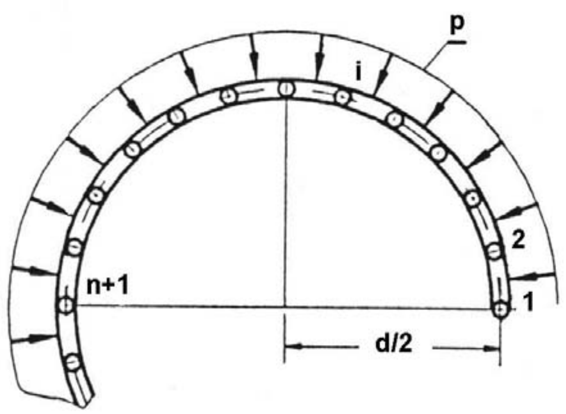


Fig. 7. Schematic diagram of piston ring substitutional model [1]

Beside ring geometry also the value of ring mean pressure p_s and pressure at the vicinity of gap p_m are used as program input data. Following formula has been used for mathematic description of this distribution at an arbitrary point of ring circumference (given by φ angle):

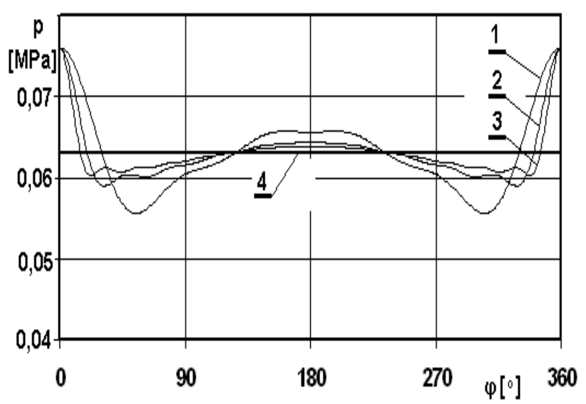
$$p(\varphi) = p_0 + p_1 \cdot \cos\varphi + p_2 \cdot \cos 2\varphi + \ldots + p_n \cdot \cos((lh) \cdot \varphi), \tag{3}$$

where lh is a number of harmonics taken into consideration.

Taking into account conditions of ring balance in bore one can define the formulas enabling computations of elastic wall pressure at any angle φ. In particular, the value of pressure at ring gap (i.e. at point 1, for $\varphi = 0$) should be equal to preliminarily assumed pressure p_m (maximum p_{max} or minimum p_{min}, depending on circumferential distribution assumed at the beginning). Due to the limited volume of this study the method of pressure distribution definition will not be presented here (it might be found in [1]).

The number of harmonics lh taken into consideration for calculations of wall pressure affects its distribution along the circumference (for given p_s and p_m). Larger their number taken into account in computations of loads, the quicker fall (or rise) in pressure as driving away from ring gap. Fig. 8 (in Cartesian coordinates) and Fig. 9 (in polar coordinates) show changes in the ring elastic pressure wall distribution depending on number of harmonics (computations were carried out for marine engine compression ring of 0.48 m in diameter, where the mean pressure is 0.063 MPa) and pressure in the vicinity of gap higher or lower from the mean value by 20%. The course of curves shows that lower number of harmonics gives higher deviation from the mean value not only at the gap vicinity but also along the entire circumference. It seems that distribution obtained for a greater number of harmonics is more advantageous for a correct ring operation (however, a ring section of higher of lower pressure becomes shorter). Because of that 10 harmonics have been taken into account in further studies presented in the paper.

a)



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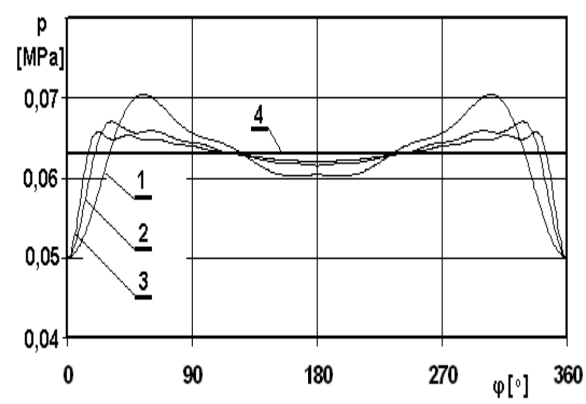

Fig. 8. Changes in circumferential pressure distribution relative to the number of harmonics taken into account for a ring of increased (a) and decreased (b) load in ring gap vicinity and for 5 harmonics (1), 10 harmonics (2), 15 harmonics (3); the pressure mean value p_s = 0,063 MPa, p_{max} = 0,076 MPa, p_{min} = 0,050 MPa

a) b)

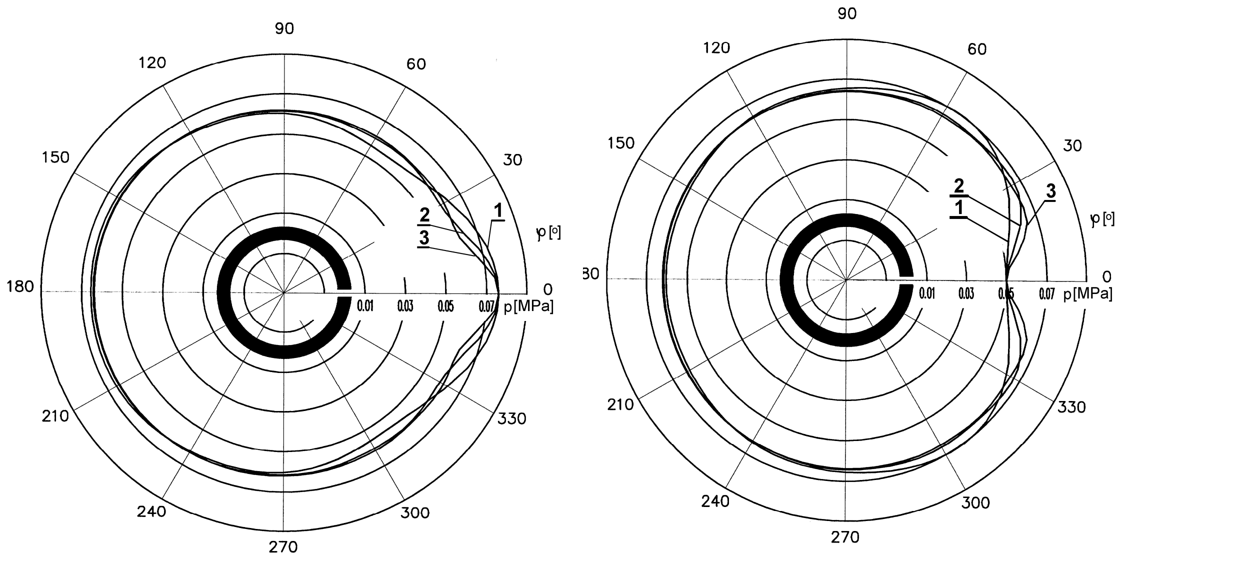

Fig. 9. Changes in circumferential pressure distribution relative to the number of harmonics taken into account for a ring of increased (a) and decreased (b) load in ring gap vicinity and for 5 harmonics (1), 10 harmonics (2), 15 harmonics (3); ring data as in Fig. 8

As mentioned above the ring of not uniform contact pressure around the bore adopts the form different from the ideal circle after being surrounded by the steel belt (as in Fig. 5) and tightened. Computations carried out using the earlier presented program revealed that for the ring presented in Figs. 8 and 9 deviations are about 150 μm close to the gap, which means less than 1% relatively to ring radius. Moreover, the precise calculations show that the value of tangential force changes itself as well. For example, for a ring presented in Fig. 10a the tangential force rises from 216 to 220 N, while for a ring in Fig. 10b – from 216 to 227 N.

a) b)

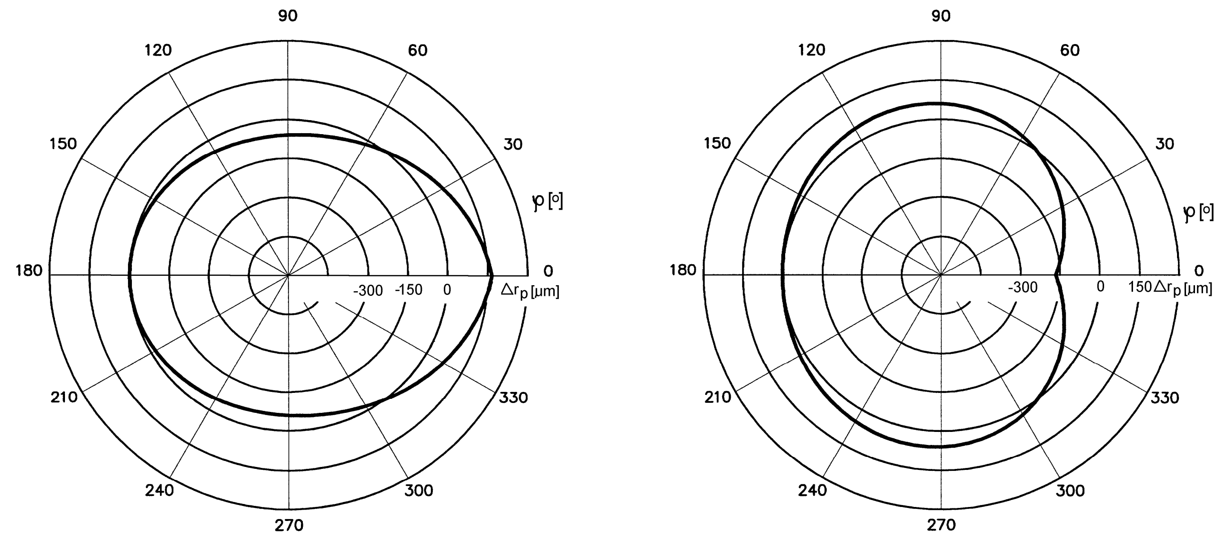

Fig. 10. Ring shape after its encircling by the steel band of device presented in Fig. 5

It seems possible to use the measurement of ring loaded this way as a base for determination of ring pressure distribution and its change with ring life.

The evaluation of piston-cylinder assembly elements wear is a significant task connected with the operation of this set. Along the engine mileage the cylinder radius grows and the ring face wears which means that the ring expands and its contact wall pressure drops. Fig. 11 illustrates the ring shape for selected values of its wall pressure. Line 1 shows the ring put into bore of ideally circular form, while the line 4 – ring free shape. The intermediate lines illustrate the ring free shape for partial loads. The ring pressure drop is accompanied by the increase in gap between ring ends (the gap was not presented in Fig. 11 in order to make draw simple).

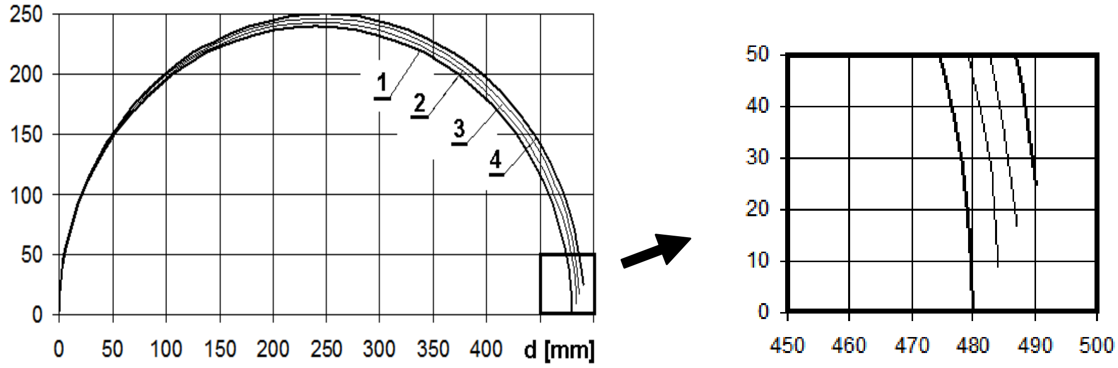

Fig. 11. Shape of fully loaded (1), free (2) and partly loaded (3) ring: p_s = 0,040 MPa (2) and 0,020 MPa (3); d = 0.48 m

Its worth mentioning that uniform contact pressure around the bore on a real engine would happen only if the ring was put into the bore of shape exactly the same as the shape of ring (therefore different than ideal circle – Fig. 11). In the case of combustion engine liners the real typical wear of liner has different form. Taking this into consideration when evaluating the distribution of ring pressure against the worn bore (and ring as well) the computational program has been developed and another subroutines have been added (Fig. 6b). Results of computations obtained using that version of program will be presented in another paper due to their complexity.

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