DEPENDENCE OF PISTON RING PARAMETERS DETERMINATION ACCURACY ON THE METHOD OF CALCULATION

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Abstract

Mathematical models are commonly used at the stage of piston ring design and during examination of ring collaboration with liner surface. Both analytical and numerical methods are used for development of such models. In the case of models based on analytical methods the range of their application is usually limited only to selected cases while the use of numerical methods allow to widen the range of their applications but their accuracy depends on increment used. The paper presents a procedure of verification of the piston ring mathematical model constructed by the authors using numerical methods. Conformity between the results obtained using this program and the results of analytical calculations concerning the displacement of ring free ends brought about by the acting loads was the aim of the analysis. Exemplary computations were carried out for three compression rings of different geometry. Regarding the conformity of achieved results as satisfactory for practical use probable causes of observed discrepancies were pointed out.

Relations between energy accumulated in curved bar and loading force (according to the Castiliano's theorem) were used in a course of calculations. Exemplary calculations were carried out for compression rings of three types engine, i.e. automotive, bulldozer and marine ones. A probable cause of differences between results of calculations carried out according to various analytical methods have been pointed out assuming their accuracy as satisfactory for practical purposes.

Keywords: *combustion engine, piston ring, ring geometry, ring pressure, simulation*

1. Role of compression ring in operation of piston-cylinder set

In spite of apparently simple construction piston compression rings perform a good number of tasks during engine operation among which tightness of combustion chamber seems to be the most important one. In correctly designed piston-cylinder assembly the ring should adjoin cylinder liner with its entire circumference. However, due to the presence of a number of unfavorable phenomena like liner fitting deformations, cylinder wear etc. the circumferential contact of ring and liner deteriorates, which to limited extension is compensated by ring elasticity and pressure caused by gas forces. Due to the lack of contact, so called light tightness worsens (see Fig. 1) which eventually leads to the increased blow-by resulting in fall of engine power and intensive wear of liner surface. Another damaging result of this phenomenon is an increased oil consumption caused by oil scraping towards combustion chamber (an increase in exhaust toxic compounds is a secondary effect of this phenomenon). Lack of proper contact between ring and liner causes worsening of another vital task of ring, i.e. heat transfer from piston crown to cylinder liner. Because of presented reasons requirements concerning quality of ring to liner contact are very rigorous. For example, slit should not excess 30 μm (and at most expand over 10% of entire ring circumference) on marine engines of large cylinder diameter [4].

A correct design of piston ring and subsequent thorough monitoring of its operation during engine run is being considered very important for limitation of light tightness loss. However, a direct observation of ring operation, especially evaluation of the contact between ring and liner.

This results from unfavorable conditions in the neighborhood of running rings (high temperature and pressure) and presence of so called oil film. Indirect investigations using the measurement of distance between ring and liner or blow-by are highly inaccurate.

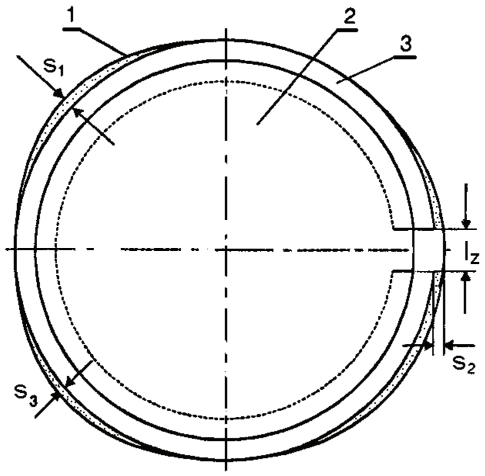

Fig. 1. Scheme of new ring contact with worn liner; 1 – cylinder liner, 2 – piston, 3 – piston ring, l_z – clearance of ring free gap, s_1, s_2, s_3 - maximum gap size [9]

Beside investigations carried out on real objects the model ones are possible as well. A number of more or less complicated mathematical models have been developed that are used for evaluation of ring and liner collaboration [6-8]. Some of them concern a selection of compression rings but unfortunately for a few of them it is difficult to define a precise range of application. Because of that the authors decided to develop a computational model of their own using for its construction relations obtained in a course of theoretical analyses and practical tests as well. Further part of this study presents a basic relations used for development of piston ring mathematical model and eventual examples of its application to verification of this model.

2. Dependences used for design of compression rings

For years specialists of piston-cylinder assembly have been trying to establish dependences between ring shape and forces loading the ring, especially on definition of so called ring free shape. First studies on this subject were published in the first half of XXth century. These studies presented trials on relations between selected quantities while the range of their applications was limited to some cases of their variability. For instance, a constant circumferential load or ring radial thickness were assumed constant – more information one can find in [1, 2].

Basic characteristic parameters of compression ring are as follows: external ring diameter d (equal to the liner diameter), radial thickness g_p, axial height h_p, distance between ends of ring free shape m and ring gap when in cylinder l_z (Fig. 2a).

On the other hand tangential and radial force, F_t and Q respectively (Fig. 2b) and circumferential pressure p (Fig. 2c) as well which are related to the modulus of elasticity E are used for evaluation of ring elastic properties. Values of Ft and Q forces are defined as those

which make ring ends close with the gap of lz.

The scheme of ring fitted into liner and pressing continuously on its surface (as shown in Fig. 2c) was used for calculation of relations between ring geometry and actual loads (Fig. 3 shows only a half of ring for simplicity). Results of analyses and calculations presented further are related to so called ring neutral axis distant from the center of clamped ring of r_m (perfect ring circularity is assumed).

a) *b)* *c)*

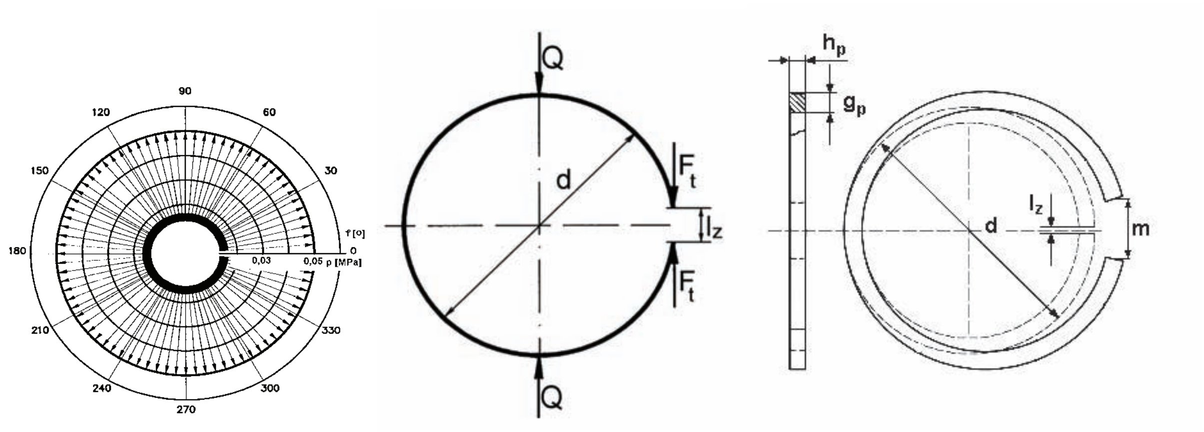

Fig. 2. Sketch of a compression ring: free and tighten form (a), loaded with force F_t or Q (b) and exemplary picture of uniform wall pressure distribution (c)

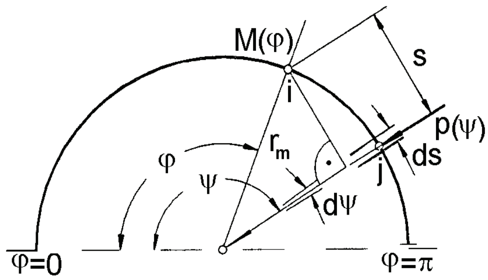

Fig. 3. Sketch of a loaded ring for derivation according to [1]

According to the presented scheme dM(φ) differential of ring bending moment at point „i" equals:

$$dM(\varphi) = p(\psi) \cdot h_p \cdot s \cdot ds \,, \tag{1}$$

where:

p(ψ) - local value of ring to wall specific pressure at point „j",

h_p - ring axial height,

s - distance between point „i" and the radius of force $P(\psi) = p(\psi) \cdot h_p \cdot ds$ operation,

ds - ring section within angle dψ.

After necessary transformations an equation has been obtained that allows to carry out calculations of bending moment M(φ) at the point defined by angle φ

$$M(\varphi) = r_m^2 \cdot h_p \int_{\psi=\pi}^{\varphi} p(\psi)\sin(\psi - \varphi)d\psi \,. \tag{2}$$

Ring circumferential load resulting from its installation in liner often differs from the even one. This could be a result of designer's intentional effort or cylinder deformation. Because the variability of circumferential load makes difficult or even prevents the determination of relevant analytical relations the assumption of load permanence should be adopted, i.e. $p(\psi) = p_o = const$ which makes that the Eq. (2) takes the following form

$$M(\varphi) = p_o \cdot r_m^2 \cdot h_p (1 + \cos\varphi) \,. \tag{3}$$

One of the ring characteristic features is so called tangential force F_t acting in the vicinity of ring free gap which causes the ring tightening up to the moment when the gap reaches the value of l_z (equal to the gap of ring fitted into cylinder liner). For evaluation of ring elastic properties and especially the tangential force measurement as well as displacement of ring free ends caused by this force special devices are used presented for instance in [8] (one of them is presented in Fig. 4).

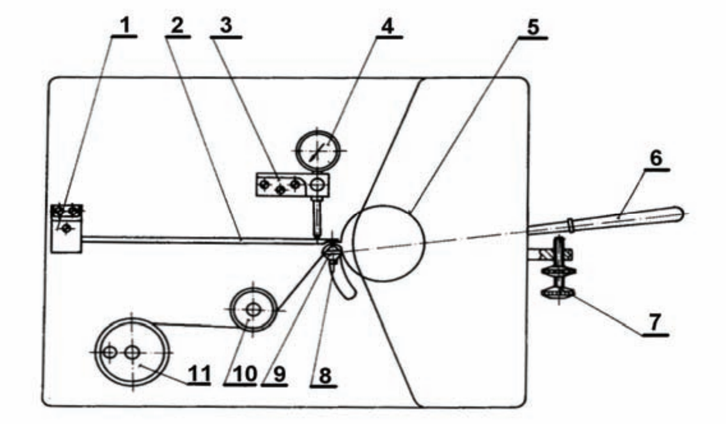

Fig. 4. Schematic of device purposed for ring tangential force measurement: 1 – beam hold, 2 – elastic beam, 3 – sensor hold, 4 – micrometer, 5 – steel band, 6 – lever handle, 7 – limit screw, 8 – screw, 9 – band hold, 10 – rotating drum, 11 – tightening drum [1]

Using a sketch shown in Fig. 3 the dependency was established allowing to define the value of bending moment for arbitrary angle φ as:

$$M(\varphi) = F_t \cdot r_m \cdot h_p (1 + \cos\varphi) \cdot \tag{4}$$

Comparing values of moments defined by equations (3) and (4) one can find a relation between circumferential load (corresponding to the constant wall pressure of ring p_o) and the tangential force F_t:

$$p_o = \frac{F_t}{r_m \cdot h_p} . \tag{5}$$

Presented relation is precise only in a case when the tangential force does not cause a ring deformation off a perfect circle (actually such phenomenon occurs).

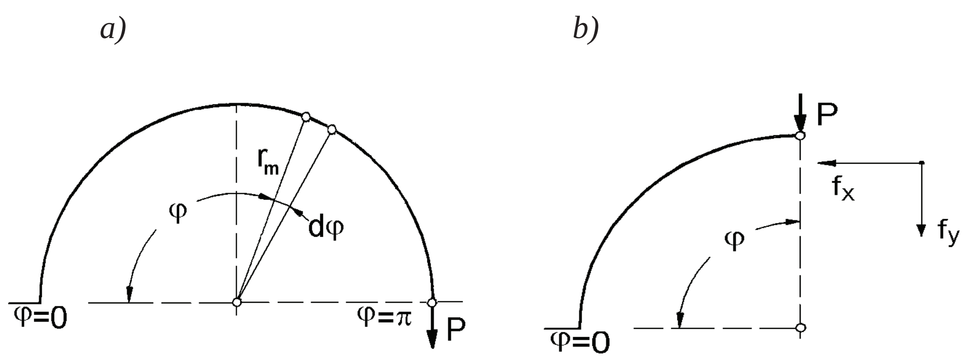

Fig. 5. Sketch of ring loaded by force P at the end (a) and in the middle of circumference (b)

Displacement of individual points of ring could be calculated analytically. The ring should be treated as bent bar of sufficiently large radius of neutral axis r_m (relatively to its radial thickness g_p). Vertical displacement of the ring free end f_y (Fig. 5) is defined as the derivative of bar potential energy V relative to the force P (according to the Castigliano's theorem [3]):

$$f_y = \frac{\partial V}{\partial P} . \tag{6}$$

The increase of potential energy dV caused by bending moment M(φ) along the increase of angle dφ equals (other forces and moments are omitted as negligible ones)

$$dV = \frac{M^2(\varphi) \cdot r_m \cdot d\varphi}{2 \cdot E \cdot I}. \tag{7}$$

It could be proved that the potential energy accumulated in ring within the section defined by the angle ($0 - \varphi_1$) equals:

$$V = \frac{P^2 \cdot r_m^2}{2 \cdot E \cdot I} \int_0^{\varphi_1} (1 + \cos\varphi)^2 d\varphi, \tag{8}$$

and total displacement of ring at the place where the force acts (ring free end):

$$f_y = \frac{P \cdot r_m^3}{E \cdot I} \int_0^{\pi} (1 + \cos\varphi)^2 d\varphi. \tag{9}$$

Similarly the horizontal displacements f_x of ring points are being determined. Changing the range of ring section subjected to load up to the ring half (Fig. 5b) dependences allowing to calculate the displacement resulting from radial force Q (ring part on the right hand side of the force Q is unloaded and does not deform) could be defined. The developed equations are summarized in Tab. 1. The last row shows how many times the displacement caused by the force acting at the ring free gap is higher than the displacement relative to force of the same magnitude but applied at the middle of circumference.

Tab. 1. Displacements of ring points caused by the P force

Point of force location* (Fig. 2)	Displacement along the x axis	Displacement along the y axis	Resultant displacement f
Middle of ring circumference	$f_x = \frac{Q \cdot r_m^3}{2 \cdot E \cdot I}$	$f_y = \frac{\pi \cdot Q \cdot r_m^3}{4 \cdot E \cdot I}$	$f = \frac{Q \cdot r_m^3}{2 \cdot E \cdot I}\sqrt{1+\left(\frac{\pi}{2}\right)^2}$
Ring free gap	$f_x = \frac{2 \cdot F_t \cdot r_m^3}{E \cdot I}$	$f_y = \frac{3 \cdot \pi \cdot F_t \cdot r_m^3}{2 \cdot E \cdot I}$	$f = \frac{2 \cdot F_t \cdot r_m^3}{E \cdot I}\sqrt{1+\left(\frac{3\pi}{4}\right.}$
Relation of displacements for equal forces Q and F_t	4	6	≈ 5.5

*Usually, the P force at ring free gap is called tangential force and is designated F_t, while the P force at the middle of ring is called radial force and designated Q.

3. Application of analytical relations to verification of compression ring mathematical model

As mentioned earlier, the most significant studies on ring shape determination were published already in the first half of last century. Because of obvious reasons the analytical dependences presented in these studies were developed using far going limitations, which brought about the ring calculations to the conditions far different from those of ring actual operation. Among other the assumptions were as follows: the ring is located in perfectly circular liner and with its entire circumference contacts the cylinder, and the wall pressure is evenly distributed along the full circumference (as shown in Fig. 2c). The simplifications were introduced because variability of some parameters of ring geometry and material were too difficult to describe with analytical functions. Fortunately, numerical methods are free of such restrictions. Nowadays, the piston rings can be designed with arbitral accuracy according to the mathematical models presented in literature, also the Polish one. One of the methods concerning the distribution of ring elastic wall pressure presented Iskra [1]. This method was used for the development of mathematical model applied by the Authors to their computations described in [6, 7]. However, the useful application of any new model should be preceded by the verification which mean the comparison of obtained results with results achieved using other methods. Following study presents results of model calculations and their comparison with results got using analytical methods concerning piston compression rings.

Technical data of analyzed rings was necessary to begin the comparative calculations. These data, summarized in Table 2 are taken from manufacturers' catalogues [5] or were measured directly. Their dimensions and objects of application were different, i.e. automotive engine 170A.000, bulldozer DTI-817C and marine engine L48/60CR. Beside different diameter other characteristic features are different including tangential force which allowed to verify correctness of model applied for various ring design. The unknown value of the radial force Q which cause the same displacement along the y axis as the tangential force F_t was determined according to the formula Q = 2,639 F_t (this formula, presented in [4] for example will be verified in another study).

Tab. 2. Technical data of exemplary compression rings of combustion engines

Quantity		Ring No 1 (automotive engine)	Ring No 2 (engine of a bulldozer)	Ring No 3 (marine engine)
Ring neutral radius r_m	[m]	0.0382	0.0655	0.232
Axial height h_p	[m]	0.0014	0.003	0.015
Radial thickness g_p	[m]	0.0034	0.005	0.016
Modulus of elasticity E	[Pa]	$115 \cdot 10^9$	$112 \cdot 10^9$	$105 \cdot 10^9$
Tangential force F_t	[N]	9.60	18.6	219
Radial force Q	[N]	25.3	49.1	578

Conformity of calculations of ring point displacements under load of forces F_t and Q calculated with the use of data presented in Tab. 1 and own mathematical program was analyzed during verification studies. Calculations according to mathematical model were carried out for 720 sections of ring circumference.

As it can be seen from the data presented in Tab. 3 considerable differences occur between displacements fx and fy calculated using formulas from the Tab. 1 and authors' program, nevertheless their specific value does not exceed 5%.

Tab. 3. Results summary of ring points displacement calculation carried out with analytical (A) and numerical (N) methods for load brought about by F_t or Q force

	Ring No 1			Ring No 2			Ring No 3		
Displacement	Method A	Method N	δ [%]	Method A	Method N	δ [%]	Method A	Method N	δ [%]
f_x [mm] (F_t)	2.03	1.96	3.45	2.98	2.84	4.69	10.2	9.71	4.80
f_y [mm] (F_t)	4.78	4.92	2.93	7.04	7.20	2.27	24.0	24.5	2.08
f_x [mm] (Q)	1.34	1.40	4.47	1.97	2.05	4.06	6.71	6.99	4.17
f_y [mm] (Q)	2.10	2.11	0.47	3.09	3.13	1.29	10.5	10.6	0.10

Among basic reasons of the noticed discrepancies one should point out that analytical calculations were carried out for deformed ring, i.e. of the shape not circular. This conclusion is corroborated by the courses presented in Fig. 5 obtained during model investigation carried out for ring number 3. This course shows that for the analyzed ring distance of points r_i from ring center is constant only for evenly distributed load (line 1). The greatest variation of this distance occurs for the point number 1 (ring free end) when ring is loaded with the radial force Q.

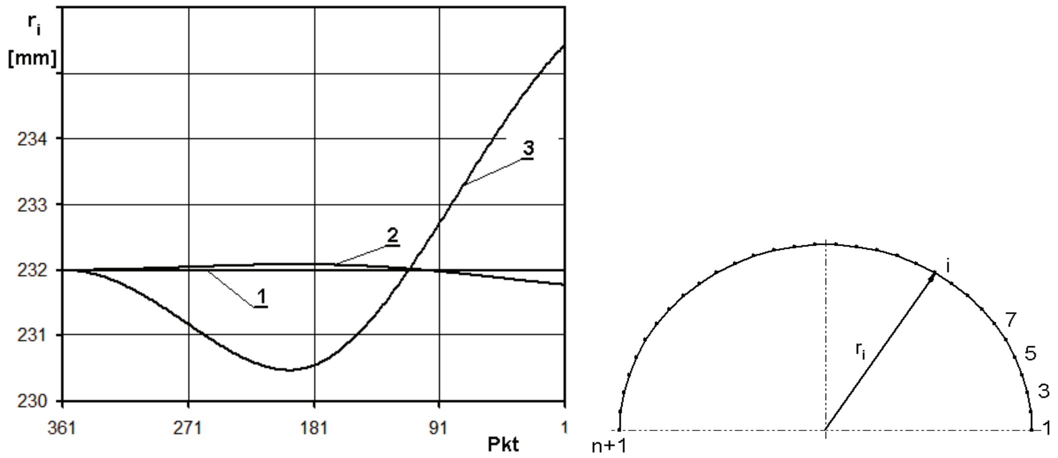

Fig. 5. Variations of calculation point distances r_i situated on circumference of loaded ring: 1 – evenly, 2 – by the force Ft , by the force Q; results for the ring No 3 from Table 2 (n=360)

Summarizing one should mention that for the group of quite different rings differences between results of analytical and model investigations do not exceed 5% (differences could be higher for rings of other construction or different load distribution). Taking this into account it was acknowledge that accordance of the results is quite satisfactory for next step of verification that will consist in comparison of accuracy of ring free form definition. Results will be presented in the next paper.

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