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HOW THE WEAR OF CYLINDER LINER AFFECTS THE COOPERATION OF PISTON-CYLINDER ASSEMBLY OF IC ENGINE

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Abstract

A ring pack of modern IC engine consists of at least one compression ring that should guarantee secure a necessary tightness of combustion chamber. A circumferential wall pressure of correctly designed compression ring should be sufficient to achieve a full contact between wall and ring, which means light tightness preventing gas blow-by. Lack of tightness is being considered as a principal cause of deterioration of engine performance data including torque instability, and a rapid grow in ring and cylinder wear within the area of blow-by.

During engine design process mathematical models are applied which implement analytical and numerical methods of calculation. Analytical methods that were used earlier presented a limited range of application and lower accuracy in comparison to the numerical methods but they allowed achieving approximate results quicker.

Taking this into consideration the presented study describes the most important relations between ring geometry and its wall pressure, which were obtained using the analytical methods. Presented formulas facilitate the evaluation of cylinder wear effect on the ring circumferential pressure distribution, especially the change in liner diameter. An attempt of light slits evaluation has been performed as well. The study was supplemented with graphs and calculative example.

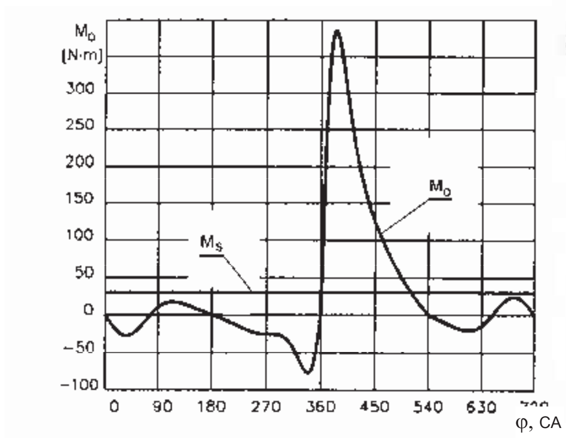
***Keywords**: combustion engine, piston ring, blow-by, torque instability*

1. Introduction

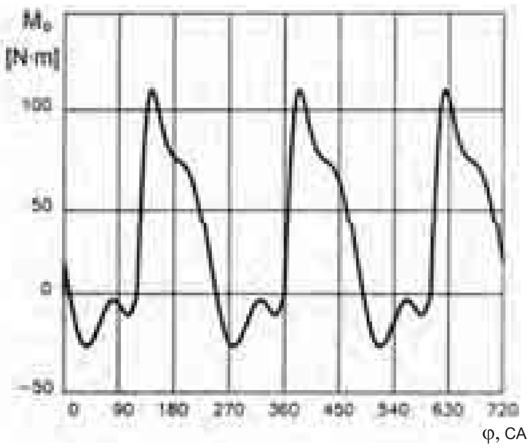
As the measurements of new and operated engine, cylinders show, their geometry differs from the one initially designed, (more on this subject one can find in [3, 7, 8]). Knowledge on actual shape of cylinder sliding surface is essentially important when the correctness of piston-cylinder set parts is to be determined, especially in the case of blow-by prevention. The lack of tightness is regarded as principal cause of engine technical data deterioration. This is the result of engine torque as an outcome of forces present during engine run, the gas force in particular. This force together with other forces acting on the crank mechanism could be transferred to the crank pin where one of its components, namely the tangential force, directed perpendicularly to the crank arm gives the torque of engine. Fig. 1a presents an exemplary course of a 1-cylinder engine torque. On a multi-cylinder engine, the generated torque consists of torques produced by all individual cylinders (as shown in Fig. 1b for a three-cylinder engine). Any random changes in gas force, caused among other by location and size of light slots, give the reduction in gas force value influencing the course of engine torque.

Another harmful effect of the lack of light tightness is an intense wear of cylinder wall and piston rings within the area of blow-by.

Principal reasons of light gap formation are as follows:
– ring movements in a ring groove,
– insufficient ring wall pressure, especially for high wear of ring face,
– radial vibrations of ring.



a)



b)

Fig. 1. Exemplary torque courses of one-cylinder (a) and three-cylinder (b) 4-stroke engines; M_s – mean torque [3]

Determination of ring movement during engine run requires the knowledge of mechanical and thermal loads, and consequently changeability of gas and inertia forces acting on piston and rings.

Distribution of oil film thickness on cylinder wall plays a significant role when position of the piston ring is being considered. Investigation on this subject is pretty complicated and the results of analytical research are hardly ever confirmed by the results of test stand investigation. The basic reason of these discrepancies is the high number of factors relative to the operation of the piston-cylinder assembly that should be taken into consideration in such model [10].

Presented here considerations concern only the latter of earlier mentioned causes of combustion chamber tightness lack, i.e. insufficient ring-to-wall pressure. They include among other the trial of description of the relations between the change in ring wall pressure and deformations of cylinder wall, however the influence of gas force as well as the effect of oil film on tightness of combustion chamber have been omitted.

2. Description of cylinder wall geometry

Determination of the effect of cylinder wall shape on the distribution of the ring circumferential wall pressure requires a correct mathematical specification of this shape. Natural course of cylinder sliding surface or its failure brings about a change in the cylinder radius $r(\varphi)$ along the cylinder circumference different for various values of the φ angle. The cylinder radius can be written as a resultant of three elements (Fig. 2):

$$r(\varphi) = r_o + z_a + z_b(\varphi)\,, \tag{1}$$

where:

r_o - radius of a new ring,

z_a - wear, assumed as constant along the entire cylinder circumference,

$z_b(\varphi)$ - wear variable along the cylinder circumference.

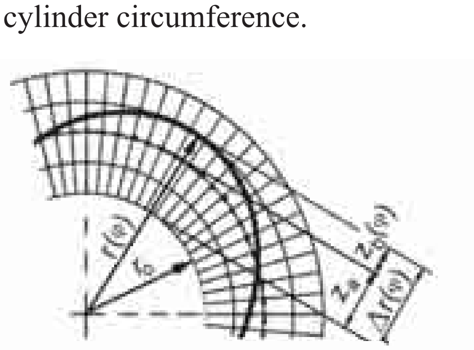


Fig. 2. Course of cylinder circumference line with applicable denomination of characteristic quantities

An initial cylinder radius r_o remains unchanged within the area outside the ring travel (or changes regularly as a result of deformations) while on the other hand it becomes the highest around the TDC. A few basic cases of cylinder wall wear could be distinguished after analysis of formula (1) components. The least complex one takes place when the cylinder is neither worn nor deformed ($r(\varphi) = r_o$ = const) while the most complex case – for the wear changing itself along the cylinder circumference (i.e. $z_a \neq 0$, $z_b(\varphi) \neq 0$).

As a result of cylinder wear and deformation the assumed initially ring wall pressure changes from the one assumed in this study as constant one $p(\varphi) = p_z$ to the one changing itself irregularly along the cylinder circumference. A trial on mathematical specification of ring pressure variations will be discussed in the further part of this paper.

3. Determination of ring pressure against worn cylinder wall

A basic relation between the change of beam curvature Δv (the piston ring can be treated as a bended beam for simplicity) and bending moment M_g imposing this change has the following form [5]:

$$\Delta v = -\frac{M_g(\varphi)}{E \cdot I}, \tag{2}$$

where E is the beam the tensile modulus and I is the beam cross section's moment of inertia. After ring fitting into the groove and eventual piston installation in cylinder the curvature of ring neutral plane changes itself from the one corresponding to the ring free form ($v_p(\varphi)$) to the curvature of new cylinder ($v_{c,p}$):

$$v_p(\varphi) - v_{c,p} = -\frac{M_g(\varphi)}{E \cdot I}. \tag{3}$$

If the ring contacts with the cylinder wall with its entire circumference, the curvature of neutral plane (of r_m radius) is given by the formula $v_{c,p} = 1/r_m$ and then:

$$v_p(\varphi) = \frac{1}{r_m} - \frac{M_g(\varphi)}{E \cdot I}. \tag{4}$$

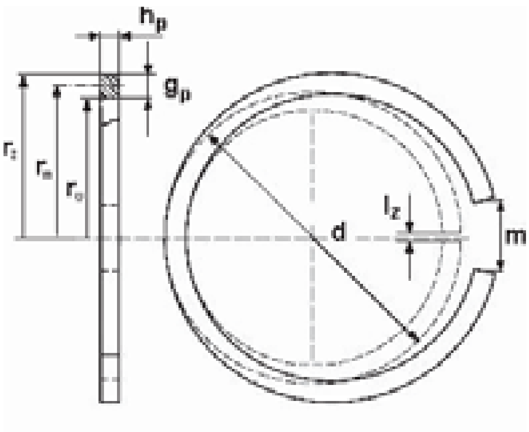


Fig. 3. Sketch of the compression ring

Performing appropriate operations one can prove that bending moment loading the ring put into a new cylinder is given by the following dependency:

$$M(\varphi) = \frac{K \cdot E \cdot I}{r_m}(1 + \cos\varphi), \tag{5}$$

where K is a characteristic parameter of the ring. Different forms of the formula helpful in calculations of this parameter one can find in [7]. One of the most often used formulas also applied to the presented calculations is of following form:

$$K = \frac{p_m \cdot h_p \cdot r_m^3}{E \cdot I}, \tag{6}$$

where p_m denominates a constant ring-to-wall pressure relative to the ring neutral plane. It should be noted here that a uniform value of ring wall pressure was applied to the presented considerations (it is a simplifying assumption because the compression ring pressure is usually increased in the vicinity of gap on modern combustion engines).

Using the relations given earlier one can prove that the free ring curvature of constant wall pressure when put into a new cylinder is given by the equation:

$$\nu_p(\varphi) = \frac{1 - K(1 + \cos\varphi)}{r_m}, \tag{7}$$

or

$$\nu_p(\varphi) = \frac{2}{d - g_p}\left[1 - K(1 + \cos\varphi)\right], \tag{8}$$

when the ring curvature is expressed, using ring outer dimensions (defined as in Fig. 3).

The ring relative curvature defined as $\nu_{wz} = \nu_p(\varphi)/\nu_{c,p}$ diminishes along with the distance from the ring gap and its changes are higher for higher values of the K parameter (Fig. 4).

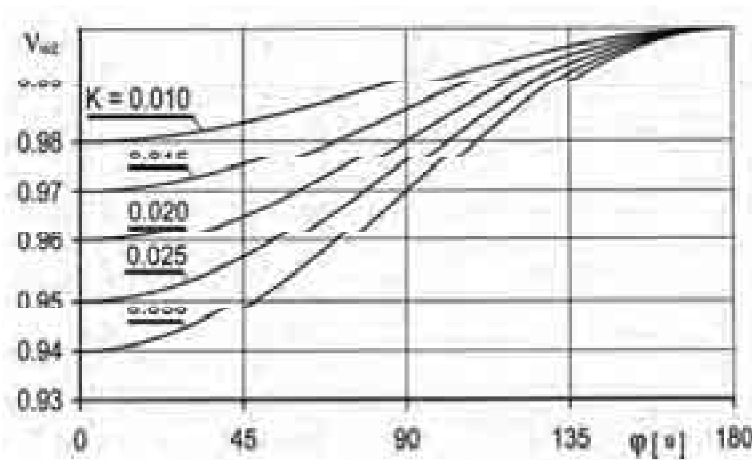


Fig. 4. Ring relative curvature variations draft for selected values of K parameter

After ring, installation into already worn cylinder the curvature of ring neutral plane changes itself from the one relative to the ring free form $\nu_p(\varphi)$ to the form corresponding to the cylinder actual curvature (ν_c). The cylinder curvature can be defined as [9]:

$$\nu_c(\varphi) = \frac{1}{r(\varphi)}\left(1 - \frac{z_b''(\varphi)}{r(\varphi)}\right). \tag{9}$$

The circumferential distribution of bending moment acting on the ring proceeds depending on cylinder's condition, in particular on factors characteristic for cylinder wear, i.e. z_a and $z_b(\varphi)$. Starting from the formula (3) following relation has been obtained:

$$M_g(\varphi) = E \cdot I \cdot [\nu_c(\varphi) - \nu_p(\varphi)], \tag{10}$$

which could be converted into:

$$M_g(\varphi) = \frac{E \cdot I}{r_m^2} \cdot \left[K \cdot r_m \cdot (1 + \cos\varphi) - z_a - z_b(\varphi) - z_b''(\varphi) \right]\# \tag{11a}$$

or

$$M_g(\varphi) = \frac{4 \cdot E \cdot I}{(d - g_p)^2} \cdot \left[\frac{K \cdot (d - g_p)}{2} \cdot (1 + \cos\varphi) - z_a - z_b(\varphi) - z_b''(\varphi) \right], \tag{11b}$$

after appropriate transformations.

The knowledge of the course of the $M_g(\varphi)$ moment pressing the ring against cylinder wall allows to define the areas where its value is equal or lesser than zero (which means no ring wall pressure). Due to that, it is possible to define hypothetic light slots between surfaces of ring and cylinder wall. The knowledge of moment course also allows determining a ring circumferential wall pressure $p(\varphi)$. When the general form of formula [5]:

$$p(\varphi) = \frac{1}{h \ \cdot r^2} \left(M_g + \frac{d^2 M_g}{d\varphi^2} \right), \tag{12}$$

is being related to the ring neutral plane and the earlier presented relations are used the following formula has been obtained:

$$p_m(\varphi) = \frac{E \cdot I}{h_p \cdot r_m^4} \left[K \cdot r_m - \left(z_a + z_b + z_b'' + z_b'''' \right) \right], \tag{13a}$$

or

$$p_m(\varphi) = \frac{16 \cdot E \cdot I}{h_p \cdot (d - g_p)^4} \left[\frac{K \cdot (d - g_p)}{2} - \left(z_a + z_b + z_b'' + z_b'''' \right) \right]. \tag{13b}$$

A trial of defining the ring wall pressure inside the worn and deformed cylinder could be carried out on the basis of earlier obtained dependencies. Because of the initial assumptions, the results should be treated as approximate ones.

4. Distribution of ring pressure against the evenly worn cylinder wall

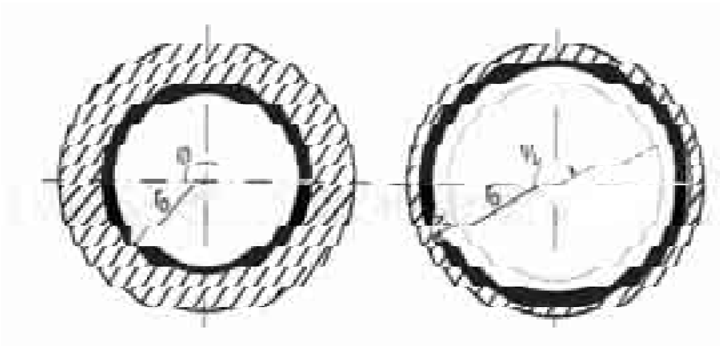
The wear of cylinder surface increases in the course of engine operation and its value is the highest near the TDC while its distribution is most often uneven along the cylinder circumference [4]. Because of that is extremely difficult to evaluate the effect of wear on ring wall pressure. Taking into account that the amplitude of surface unevenness is far lower than the mean wear a constant circumferential value of wear z_a could be used for an approximate evaluation of this effect. In this case, the ring wall pressure relative to the neutral plane becomes:

$$p_m = \frac{E \cdot I}{h_p \cdot r_m^4} \left[K \cdot r_m - z_a \right], \tag{14a}$$

on the other hand, introducing the concept of relative cylinder wear as $Z_{wz} = z_a / r_m$ to the form:

$$p_m = \frac{E \cdot I}{h_p \cdot r_m^3} \left[K - Z_{wz} \right]. \tag{14b}$$

Along with the increase of wear z_a the contact line of its face with cylinder wall decreases (assuming the full ring contact at a left hand side of cylinder, see Fig. 5b, the light slot comes along at the vicinity of ring gap and will be bigger along with the increase of wear). Knowing the course of moment pressing the ring against the wall one can calculate the critical value of the φ_k angle defining the slot limits. However, it should be noted that the assumption on ring contact could be not fulfilled since the slot might be present at other location as the ring is being moved under the acting forces.



a) b)

Fig. 5. Sketch of ring installed in new (a) and worn (b) cylinder

The value of critical angle φ_k, defining the place where ring face loses its contact with the cylinder wall is given by the formula:

$$\varphi_k = arc\cos\left(\frac{Z_{wz}}{K} - 1\right). \tag{15}$$

As it results from this formula, for a new cylinder ($Z_{wz} = 0$), the φ_k angle is equal to π, therefore marks an area of ring gap (it has been assumed $l_z = 0$ in presented calculations).

Along with the increase in cylinder wear the value of φ_k angle decreases, which means the increase in the value of light slot area S_{sz}. Assuming that the form of ring free section can be described with a circle of radius the same as cylinder radius at the φ_k angle and ignoring the ring gap area, an approximate formula could define the area of light slot:

$$S_{sz} = z_a \cdot (r_o + z_a) \cdot \left[\sin\varphi_k + (\pi - \varphi_k) \cdot \cos\varphi_k\right]. \tag{16}$$

Shortage of detailed information on the form of ring free section causes that the dependence (16), particularly for higher wear, z_a is of the approximate character. An analysis carried out using numerical methods shows that the actual area of the light slot would be higher than the one calculated according to the given equation.

Exemplary calculations have been carried out for compression rings of an automotive and marine engines running in evenly worn cylinders. Relations between absolute and relative wear, and critical value of angle and slot area were established for the data presented in Tab. 1. The achieved results are illustrated in Fig. 6.

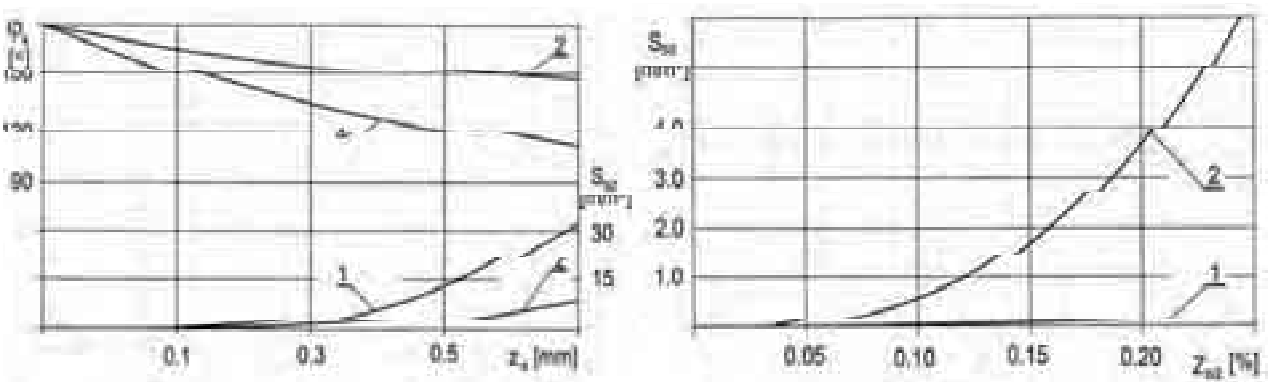
Tab. 1. Technical data of exemplary IC engine compression rings

Quantity		Ring 1 (automotive engine)	Ring 3 (marine engine)
cylinder diameter d	[m]	0.08	0.480
ring neutral radius r_m	[m]	0.0382	0.232
axial height h_p	[m]	0.0014	0.015
radial thickness g_p	[m]	0.0034	0.016
Young modulus E	[Pa]	$115 \cdot 10^9$	$105 \cdot 10^9$
mean pressure p_m	[MPa]	0.180	0.063
parameter K	[–]	0.0266	0.0220

5. Conclusions

Analyses presented in this paper include among other comparison of analytical and numerical calculations. The achieved results allow concluding that there is high conformity of the angle φ_k value (relative differences do not exceed 1%). On the other hand, differences between the areas of light slot are far bigger. Presumably it outcomes from the lack of proper analytical model of ring free section that is not pressed against the cylinder wall. However, regardless on applied method one can note that the light slot area increases for higher cylinder wear.

The case of change in ring wall pressure presented in this paper applies to a constant wear of cylinder wall measured along its circumference. The tests mentioned earlier show that this wear is not even (because of different reasons) and need to be thoroughly investigated. A trial on evaluation of the effect of uneven cylinder wall wear on distribution of ring wall pressure and chances for presence of light slots (case of $z_b(\varphi) \neq 0$) will be carried out in further papers.



a) *b)*

Fig. 6. Variations in φk angle and area of Ssz slot vs. absolute za and relative Zwz value of cylinder wear for an automotive (1) and marine (2) engine

Acknowledgements

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