COMPRESSION RINGS OF LOW-SPEED HIGH POWER ENGINES

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Abstract

The paper describes the design of compression rings used in low-speed, high power marine engines and it analyses relations between the most important operational parameters. Ring material, dimensions, geometry of sliding surfaces, distribution of circumferential pressure as well as operational conditions (thermal and mechanical loads, way of lubrication) were taken into consideration for the analysis mentioned above. Moreover, the paper points at modifications in ring geometry that have been happening for last years, comparing previous and recent designs. The effect of ring circumferential pressure against liner on the piston-cylinder assembly operation has been considered and the causes of its variability have been pointed out as well. When presenting the methods of measurement and tangential force calculations basic advantages and disadvantages have been presented as well as the evaluation of changes resulting from the ring circumferential wear. The compression ring mathematical model developed by the authors allows for a precise definition of relations between ring geometry and distribution of ring circumferential pressure.

A need for a more accurate method of ring circumferential pressure evaluation has been justified in summary, giving hints necessary for its preparation.

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

1. Introduction

A prime function of a piston ring is to provide a reciprocating seal between the piston and bore limiting the flow of combustion and compression gas into the crankcase. A secondary function, particularly important on highly loaded marine engines, is the heat flow from piston to bore (Fig. 1).

The conditions of ring operation vary continuously and the range of these changes depends on engine type and conditions of its run. Due to the size and power developed by marine engines their piston rings are remarkably loaded this makes that among all elements of piston-cylinder assembly they are the ones most prone to wear. It has been found that there is a high correlation between the technical condition of piston rings and condition of other parts of crank-piston set.

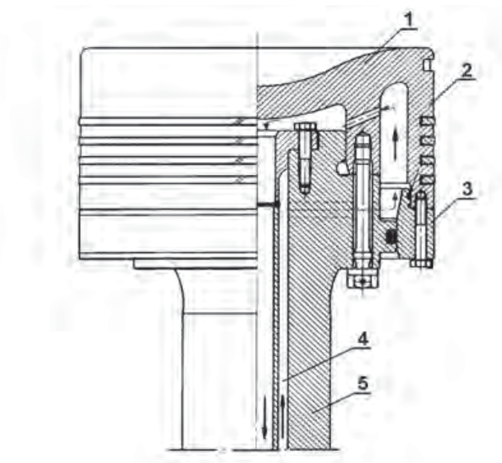

Fig. 1. A sketch of crosshead engine piston; 1 - crown, 2 - land, 3 - skirt, 4 - oil ducts, 5 - piston rod

This correlation is used in diagnostic systems of marine engines, e.g. the SIPWA-TP system developed by the New Sulzer Diesel [7]. A proper selection of material and design (geometry, circumferential pressure, way of lubrication) which guarantees fluid oil film between the collaborating ring face and bore are the most effective ways to extend the ring life. The compression ring material should guarantee high strength to bending and squeezing as well as elasticity that decides about ring pressure against the cylinder liner. Besides the material selection and ring formation other measures are taken in order to expand the ring life. For example the Wärtsilä-Sulzer offers a set of undertakings called “TriboPack” [9] that comprise:

- cylinder liner deep honing,
- use of the “anti-polishing” ring,
- use of standardized profiled ring pack covered with ceramic cover,
- chromed ring grooves,
- multi-level bore lubrication.

From the point of correct ring operation the most important measures are the application of protective covers and the improvement of bore lubrication. Rings are covered with the layer of another metal, most often chromium but also nickel, molybdenum, tin, cadmium or copper. Such prepared layer, of thickness from 10 to 100 μm increases the ring resistance to corrosion and wear within the entire range of loads and temperatures.

Considerable modifications have been introduced to the lubrication system of marine engines. Even in the late ninetieth of last century their construction was almost unchanged. Lube oil supplied by the mechanically driven pump was sprayed onto the cylinder bore with special jets. Their number and distribution depended on bore diameter (6 for diameters up to 600 mm and 10 for bigger ones). Electronically controlled lubrication system ALPHA and pulse system PLS have been introduced in recent years [3]. Electronic control used in these systems allow for a precise selection of adequate oil dose and timing depending on engine run conditions and sulphur contents in fuel. A reduction in oil consumption to about 0.6 g per kWh and extend of correct operation of piston-cylinder assembly are the achieved results.

2. Design of marine engine compression rings

Ring dimensions (see Fig. 2), its elastic properties (defined as specific pressure p against the bore and circumferential pressure distribution) as well as material properties described with the Young modulus *E* belong to the most important parameters characteristic for compression ring. In order to evaluate the changes that have been occurred over the last few dozen years in ring design their construction and geometric relations have been analyzed for a selection of compression rings

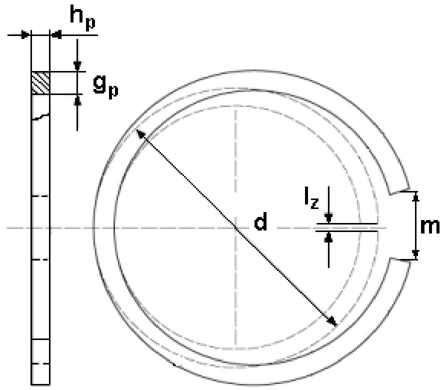

Fig. 2. A sketch of compression ring: free (continuous line) and mounted in bore (dashed line); d – bore diameter, g_p – ring radial thickness, h_p – ring axial height, m – free ring joint, l_z – joint gap

used in to-day marine engines of 480 to 960 mm bore. The statistic analysis on correlation between ring diameter d, its axial height hp and radial thickness gp has been carried out on a sample of ten cases.

For mathematical description of presented relations the exponential function has been applied obtaining the following regression formulas and the correlation coefficient *r*:

$$h_p = 0.773 \cdot d^{0.476}, \text{ for } r = 0.97, \tag{1}$$

$$g_p = 0.0585 \cdot d^{0.908}, \text{ for r} = 0.99. \tag{2}$$

Curves of regression drawn according to results obtained have been presented in Fig. 3 (continuous line 1). Curve's course proves that the ring axial height and radial thickness increase with higher cylinder diameter. In order to compare to-day rings with their predecessors (dashed line 2 in Fig. 3) the curves obtained after statistic analysis of earlier used piston rings geometry have been placed in the same figure (data from [5] were used). This comparison shows that rings used earlier had far higher axial height and radial thickness than those used to-day.

It should be emphasized that the observed changes in ring geometry, axial height in particular, influence ring compression properties as well as parameters of oil film formed under the ring face. Lower the ring, the better contact between the bore and ring and better the tightness of combustion chamber. However, one should remember that lower rings perform lower strength and elasticity, and the oil film has lower thickness.

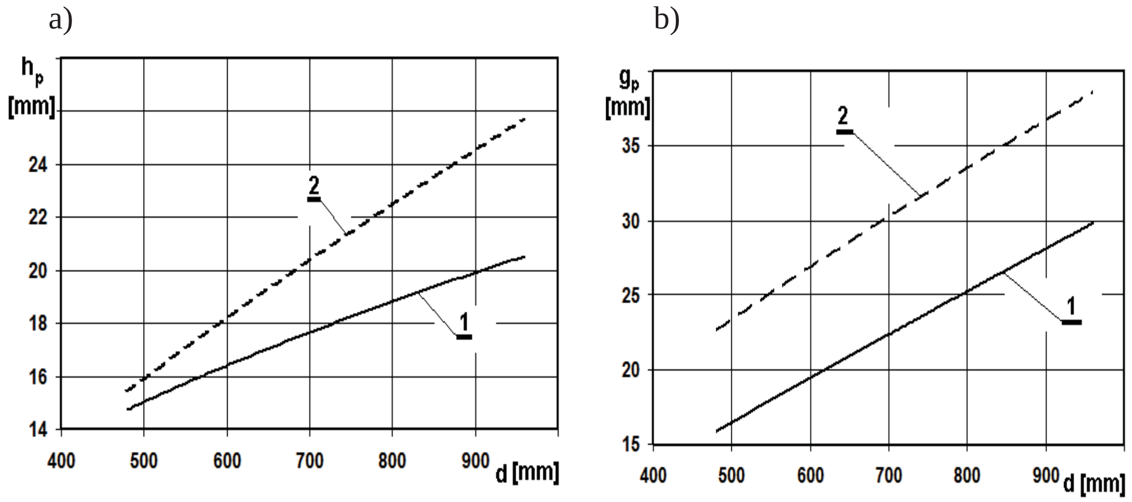

Fig. 3. Relations between ring diameter d and its: a – axial height h_p; b – radial thickness g_p; presented results have been achieved for a group of marine engine piston rings: modern – 1 (continuous line), previous – 2 (dashed line)

Lowering rings number is one of ways leading to reduction in friction losses accompanying ring travel along the bore. Thanks to better and better ring design and applied materials, the number of rings has been diminished on all engines including the marine ones. The carried out analyses showed that the number of piston rings was reduced to 7-8 in the sixties and to 4-5 on contemporary low-speed marine engines, which did not worsen the combustion chamber tightness but distinctly reduced the friction losses of the piston-bore group.

3. Relations between ring geometry and its pressure against bore

The measurement of resilient pressure is one of the mechanical tests that piston rings undergo in a course of condition tests. Indirect evaluation of this pressure is carried out by the determination of tangential force F_t at ring gap or central force F_d located in the middle of ring circumference (see Fig. 4).

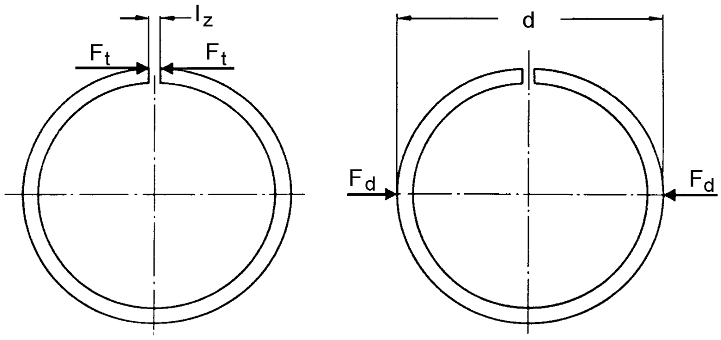

Fig. 4. A sketch of ring with locations of tangential Ft and central Fd force measurements points

Following formulas developed by the Goetze [6] combine both forces and ring material, giving for grey cast iron:

$$F_d = 2.05 \cdot F_t\,, \tag{3a}$$

and for malleable iron:

$$F_d = 2.15 \cdot F_t\,. \tag{3b}$$

Figure 5 presents a sketch of a device dedicated for a measurement of tangential force F_t. The tested ring is surrounded with a metal band (of 0.08 to 0.1 mm thickness) and is tightened till its ends reach the operational gap. Surfaces of ring face and steel band are covered with lubricant while the ring ends are excited to vibrate moderately (with frequency about 45 to 50 Hz) in order to eliminate the effect of friction. An electronic dynamometer is used for reading the tangential force.

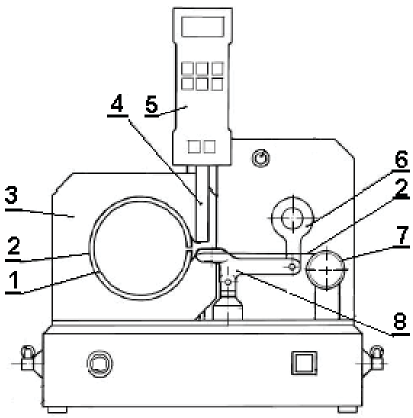

Fig. 5. Test rig for measurements of ring elasticity: 1 – piston ring, 2 – clamping band, 3 – measurement plate, 4 – dynamometer measuring axle, 5 – dynamometer, 6 – vibrator, 7 – reel, 8 – eccentric lever [2]

Knowledge of the tangential force F_t value allows calculating the mean specific pressure of ring against the bore according to following formula:

$$p = \frac{2 \cdot F_t}{d \cdot h_p}\,. \tag{4}$$

Values of specific pressure within the range from 0.04 to 0.8 MPa are assumed as typical ones for large diameter compression rings while the smaller values relate to larger bore (Fig. 6). When selecting pressure it should be taken into consideration that too high ring pressure could cause higher friction losses and higher probability of oil film rupture (between properly matched elements the contact of ring face and bore should take place only through oil film within the major part of piston stroke).

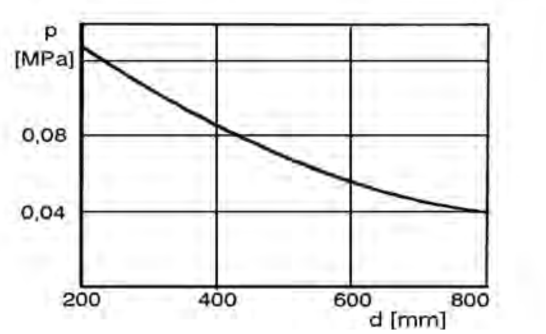

Fig. 6. Change of compression ring mean pressure in low-speed engines [10]

Relations between ring dimensions, material and its elasticity could be determined on analytical way. For example, Goetze – manufacturer of piston rings recommends the use of following formula:

$$F_t = \frac{E \cdot m \cdot h_p}{K \cdot \left(\frac{d}{g_p} - 1 \right)^3}, \tag{5}$$

where K denominates a constant given by the producer of rings equal to 14.14 for smaller rings (modulus of elasticity is E = 100 GPa). However, testing calculations carried out for rings belonging to the group of ten large diameter rings proved that such adoption leads to considerable underestimation of this force (see Fig. 7a, F_{t3} value) in comparison to the value given by the producer (F_{t1} in the same figure). Correction of the K constant to the value of 13.31 (F_{t2}) resulted in decrease of differences between forces (Fig. 7b).

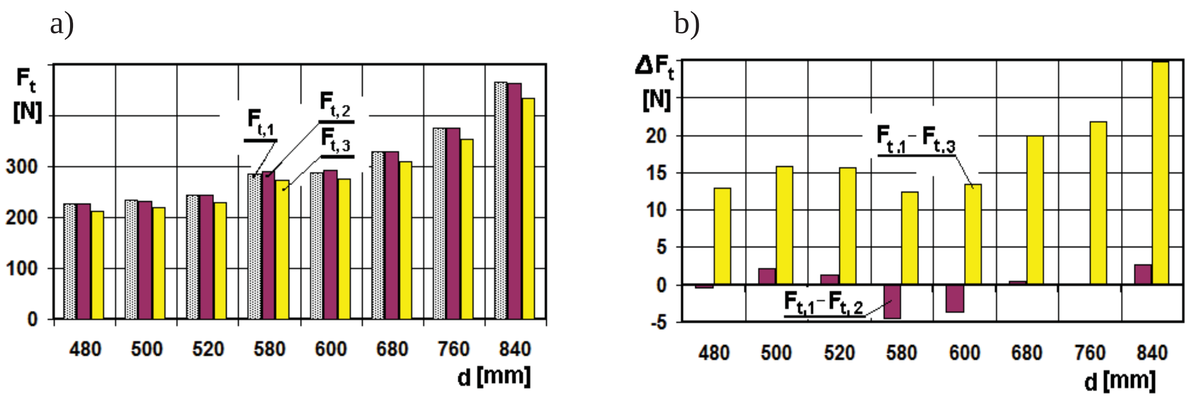

Fig. 7. Comparison of ring tangential force F_t carried out on a test sample of marine engine rings; $F_{t,1}$ *– value given by producer,* $F_{t,2}$ *– value obtained for K = 13.31 and* $F_{t,3}$ *– value obtained for K = 14.14 (a) and arrangement of differences in this force (b)*

Piston ring elastic pressure against the bore and its circumferential distribution should be selected appropriately to engine construction and its operational parameters. Too low ring pressure could lead to excessive blow-by to crank case while too high pressure could bring about increased friction losses and higher probability of oil film rupture [1, 8].

Piston ring pressure against the bore is a resultant of two forces, namely the force generated by ring own elasticity and the force generated by gas pressure acting on ring inner surface. First of these forces should be large enough to adjoin the ring to bore along the entire circumference without so called light slot, which could lead to gas blow-by. In order to carry out the test of correct contact the ring should be placed in the least worn region of cylinder liner or in a special device made with H7 dimensional accuracy class and 0.009 tolerance of roundness. Depending on ring manufacturer a light slot is allowed if it happens at most at two positions distant at least 1/12 of the circumference (30 deg) from each of ring ends. The width of light slots should not exceed 1/8 of the circumference, i.e. 45 deg while for the chrome plated faces it should not exceed 1/12 of the circumference, i.e. 30 deg [4]. In the case of marine engines producers permit maximum value of a light slot smaller than 30 µm over a 10% section of the circumference.

Equilibrium of ring put into bore requires the sum of both forces and moments equal to zero. Fig. 8a presents an example of a balanced ring with continuous distribution of pressure directed toward the ring centre. However, ring deformations of various types (thermal, mechanical and due to wear ones) occur during engine run leading to the drop of pressure around the joint. To prevent the outcomes of this disadvantageous phenomenon (higher blow-by, for example) increased ring elasticity in this region is foreseen already at the stage of ring design (Fig. 8b). On the other hand, on the two stroke engines where ring meets the scavenging ports the elasticity of ring close to its ends is purposely lowered in order to avoid the ring break off (see Fig. 8c – such load distribution is unfavourable because of high probability of blow-by).

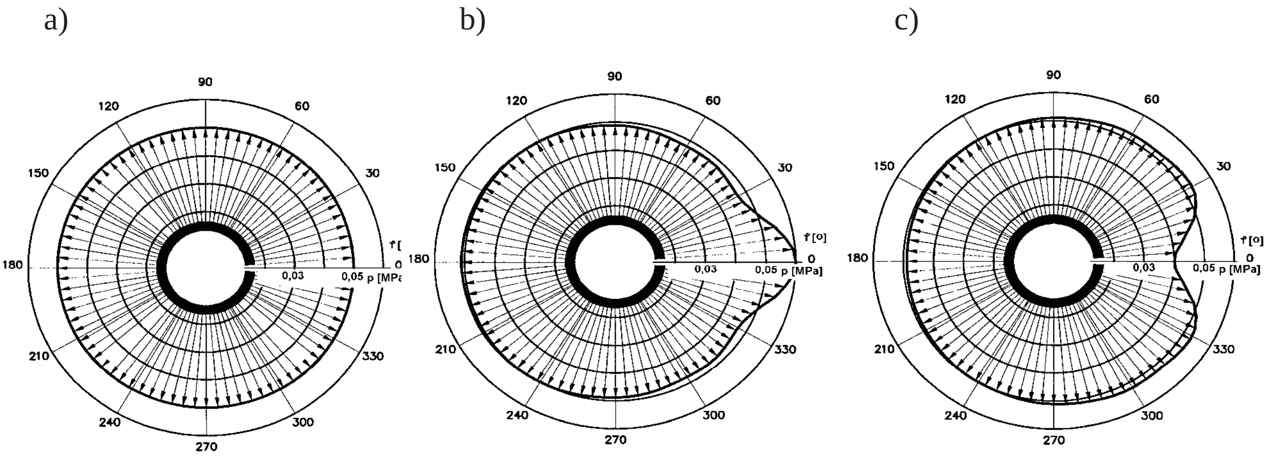

Fig. 8. Exemplary ring to bore pressure distribution; a – even pressure, b – pressure higher in the vicinity of joint, c – pressure lowered in the vicinity of joint; mean pressure p = 0.05 MPa

Distribution of ring pressure against the bore presented in Fig. 8 relate to new, unworn elements of kinematic pair “compression ring – bore”. It should be noted that tangential force measurement with the device presented in Fig. 5 does not allow for determination the distribution if this distribution is not an even one.

4. Effect of ring pressure variations on oil film distribution over bore surface

Ring to bore pressure resulting from its own elasticity has a decisive effect on correct contact of ring to bore. This contact also affects the distribution of oil film thickness, which derives from the equation (this equation does not take into account other effects but the wedge effect).

$$h_m = \sqrt{\frac{\eta \cdot u \cdot b_f}{p} W_u} \,. \tag{6}$$

As it outcomes from the equation (6) for fixed values of ring axial speed u, axial height b_f, oil viscosity η and shape of run surface W_u, the variations of oil film thickness will be counter proportional to the square root of change in ring pressure p. During ring operation the pressure distribution (and eventual contact to the bore) will diminish as the result of wear, for example. In order to evaluate the drop in pressure caused by the change in ring radial thickness g_p one could use the relation obtained from the transformation of equation (5):

$$F_t = \frac{E \cdot h_p \cdot \left(m + 2 \cdot \pi \cdot g_p \cdot \frac{\Delta g_p}{g_p} \right)}{K \cdot \left(\frac{d}{g_p \left(1 - \frac{\Delta g_p}{g_p} \right)} - 1 \right)^3} \; . \tag{7}$$

Expression $\Delta g_p/g_p$ appearing in the equation is a relative change in ring radial thickness. Fig. 9 presents the estimated changes in radial force following the change in ring radial thickness calculated according to Eq. (6). As it outcomes from the course presented in this figure the drop in pressure is not big (from 63 kPa to 54 kPa) and it would not affect the changes in oil film thickness to the great extend because of other forces that act upon the ring at the same time.

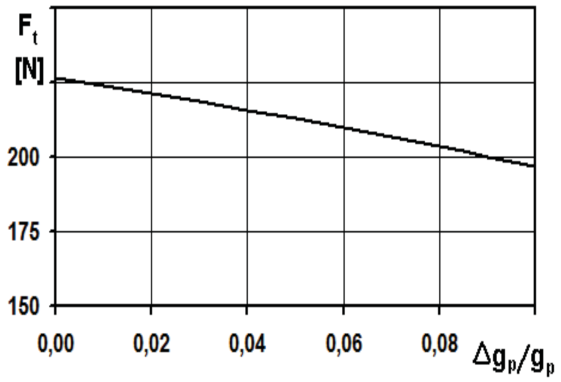

Fig. 9. The effect of relative drop in ring radial thickness $\Delta g_p/g_p$ on value of tangential force Ft for ring diameter d =480 mm

Summarizing the presented study one should conclude that the results of circumferential pressure measurements (using for instance the device presented in Fig. 5) allow only for an approximate evaluation of its value but the circumferential distribution is still unavailable. Due to that the actual pressure distribution should be measured using far more sophisticated devices or evaluated on the way of computer calculations. Taking it into consideration the Authors developed a mathematical model of compression ring (later transformed into computational procedure) which allows for a precise definition of relations between ring geometry and the circumferential pressure distribution. The description of the program and exemplary results will be presented in another study.

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