Keywords: wear; internal combustion engine; resource forecasting; cylinder-piston group; dry ring sealing; graphite paste with liquid glass

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EVALUATION OF THE RATIOS OF THE MAIN INDICATORS OF THE DRY SEALING OF THE CYLINDER-PISTON GROUP OF INTERNAL COMBUSTION ENGINES USING A SOLID LUBRICANT

Summary. This article presents the results of a study of an alternative method for reducing friction losses in the cylinder-piston group of internal combustion engines based on an experimental installation, taking into account changes in the real state of the working surfaces of the mating parts of the piston ring-cylinder pair depending on the operating time. The factors with a progressive effect on the operation of engine friction units as they wear out are studied, and the degree of their influence on wear is estimated. A model of the friction unit of a cylinder-piston group (piston-piston ring pair) of an internal combustion engine based on a solid antifriction material operating without the use of a lubricating fluid is developed and investigated. Comparative results of determining the wear indicators of sealing rings by various methods of wear control are presented. A method for predicting the resource and the real state of the engine is proposed.

1. INTRODUCTION

Changes in the main design parameters of an engine during its operation are largely determined by the actual state of the friction units, primarily the cylinder-piston group. However, existing models of internal combustion engines describing tribological processes do not take into account the dynamics of changes in technical and economic parameters as the friction interfaces wear out [2,7]. The processes occurring in this mechanism are characterized by a high temperature of individual elements, a change in the properties of the lubricating fluid during engine operation, and the appearance of insoluble solid inclusions in the oil. The need to assess the processes—in particular, the nature of the wear of the cylinder-piston group of internal combustion engines occurring during operation – is confirmed by *GOST 27860-88* “Details of rubbing interfaces,” which takes into account changes in the real state of the working surfaces of mating parts and lubricating oil [4].

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2. RELEVANCE

The modern piston engine of internal combustion was formed in the late nineteenth to the early twentieth century. Over time, engines have been improved, and their specific power and efficiency have increased. However, at the same time, the cylinder-piston group has remained the main node for converting the chemical energy of the fuel into the potential energy of the gas, followed by the conversion of pressure into the kinetic energy of the piston movement [1, 2].

This engine assembly, including a cylinder liner, a piston with its seal in the liner using compression rings, and a con-rod or rod connecting the piston to the kinematic mechanism of the engine, has not changed fundamentally [5,6]. Friction losses in the cylinder-piston group are reduced by the placement of the cylinder directly on the engine oil casing, with its rear part open and connected to the casing cavity. Further, the piston space of the cylinder is lubricated and partially cooled by drip oil from the casing, sprayed when the kinematic mechanism is lubricated, or supplied to the rear part of the piston by special nozzles. Excess oil is removed from the liner surface by means of oil-scraper rings [1, 2].

The piston is sealed in the cylinder by a set of compression rings, which have a joint and are pressed against the liner surface due to the elasticity of the rings. When the engine is running, the friction of the compression rings on the liner surface causes both the rings and the liner surface to wear out. As the circular shape of the liner is violated, the gap in the joint between the ends of the rings increases. Gases under pressure break through into the sub-piston volume and enter the groove under the rings, increasing the force with which the rings are pressed into the liner surface [7, 8]. Due to this, unequal wear of the liner in height occurs in accordance with the change in gas pressure during expansion and the stroke of the piston from the upper to the lower dead points [3, 9].

The results of a test conducted on an engine in real conditions and the assessment of its operational characteristics depending on the radial wear of the upper compression ring, taking into account the operating time, allows all the given engine characteristics to be considered depending on the engine operating time [1, 8]. At the same time, all the main operational characteristics after 300 motor hours of operation have an inflection and an increasing gradient of change. In the engine, effects of wear of the elements of the cylinder-piston group occur, by which the main performance characteristics (compression, torque, and effective power) change in the direction of deterioration by at least 10%, with further progression of adverse changes. At the end of the inter-repair period, the specific fuel consumption increases twice, which leads to a corresponding increase in operating consumption and a decrease in the technical and economic indicators of the engine [12, 15].

This is a consequence of the wear of compression rings, sleeves, and pistons themselves. The main function of the piston is to ensure the lossless conversion of gas pressure into piston movement, which is not performed optimally [5, 6]. Lubrication of the sleeve with oil from the crankcase and oil cooling of the hot elements of the cylinder-piston group leads to oil fumes, contamination with combustion products, and decomposition during engine operation with a loss of lubricating properties [2, 14]. The sealing of the piston in the cylinder by means of compression rings, by its design, allows the leakage of gases from the piston volume into the crankcase of the engine [8, 13]. For the same reason, oil from the crankcase enters the over-piston volume, where high-temperature processes are carried out.

Below are the results of engine testing and an evaluation of its performance characteristics depending on the radial wear of the upper compression ring. However, the presence of wear depending on the operating time allowed us to consider all the above engine characteristics depending on the operating time of the engine [5, 10]. At the same time, all the main operational characteristics after 300 engine hours of operation have an inflection and an increasing gradient of change. In the engine, the effects of wear of the elements of the cylinder-piston group occur, in which the main operational characteristics, specific and operational fuel consumption, which determine the economic performance of the engine, change in the direction of deterioration by at least 10% with further progression of adverse changes. This is a consequence of the wear of compression rings, liners, and pistons themselves. The main function of the piston is to ensure the lossless transformation of gas pressure into piston movement, which is not performed optimally [3, 11]. Lubrication of the liner with oil from the casing and oil cooling of the hot elements of the cylinder-piston group leads to oil burning, contamination with combustion products, and decomposition during engine operation with a loss of lubricating properties [12, 17]. The sealing of the piston in the cylinder by means of compression rings, by its design, allows the leakage of gases from the piston volume into the casing of the engine [6, 8, 10]. For the same reason, oil from the casing enters the over-piston volume, where high-temperature processes are carried out.

Usually, the inter-repair period of automobile engines is assumed to be equal to 300 motor hours. Due to the wear processes in the piston contact, the compression in the cylinders decreases after the first 300 engine hours of operation, which leads to a progressive decrease in the engine's technical and economic indicators. According to engine tests conducted in real operating conditions, all engine parameters changed significantly (compression, torque, and effective power). At the same time, at the end of the inter-repair period, the specific fuel consumption doubled, which led to a corresponding increase in operating consumption [15, 18].

Therefore, there is an urgent need to develop ways to improve the design of the cylinder-piston group of the internal combustion engine using alternative technical solutions that reduce losses for converting gas pressure into piston movement.

3. PURPOSE AND METHODOLOGY

The tasks were completed on the basis of the development and research of an experimental model of a cylinder-piston group training unit (piston-piston ring pairs) of an internal combustion engine based on a solid antifriction material operating without the use of a lubricating fluid on the principle of dry friction.

This research involved seals in the form of rings with a contact surface based on a solid antifriction material, namely graphite. It is well known that graphite is widely used in engineering as an antifriction material [14, 18]. The use of graphite as a solid lubricant in a movable contact was adopted by analogy with the movable contacts of electric machines, as this situation also requires tight contact between mutually moving elements and minimal wear of the contacting surfaces [14, 19]. The coefficient of friction of graphite and composite materials based on it range from 0.15–0.20.

The operability of such technical solutions has been tested in practice in the S.S. Balandin engine [9, 16]. Previous work in the development and implementation of sealing materials shows that the main criteria for evaluating their performance are the results of field tests.

In the field of improving piston seals, there are a number of ring designs aimed at reducing friction losses and wear of both rings and cylinder liners. There are rings with a conical contact surface of different directions (minute and anti-minute), such as Γ-shaped or triangular rings. However, in an internal combustion engine, a piston seal with conventional rectangular rings remains one of the most conservative design components. Researchers have developed and manufactured an original design of the piston-liner coupling seal of the cylinder-piston group of an internal combustion engine [11, 12].

4. SELECTION OF THE COMPOSITION AND PROPERTIES OF A SOLID ANTIFRICTION MATERIAL

The seal design consisted of two rings made of bronze or steel located in a common annular groove in the piston (Fig. 1). Each of the rings consists of two half-rings with a stepped joint of half-rings in a vertical plane, while the joints of the upper and lower rings are located at 90° relative to each other. The half-rings were unclenched and attached to the cylinder liner mirror by springs. At the same time, the stepped joints slide over each other within certain structural limits, which causes the gas to break through the ring locks into the space under the rings (Fig. 2).

Next, a paste of antifriction material was prepared. Graphite powder was used as the antifriction material. For the preparation of graphite paste, the use of various binders, such as liquid glass and kaolin, was considered. In preliminary experiments, it was found that kaolin does not create a strong structure after drying, while liquid glass can provide the required strength of the material. Three compositions of graphite paste with liquid glass as a binder were prepared and tested, with the following (%/%) ratios of graphite and binder: 25/75, 50/50, and 75/25.

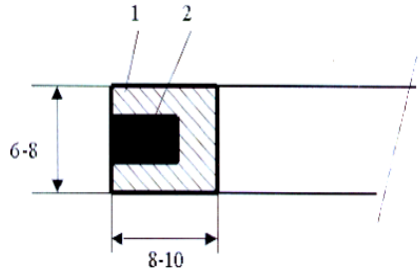

Fig. 1. Cross-section of the ring (dimensions are given in mm): 1 = metal (bronze or steel); 2 = graphite filling

a *b*

Fig. 2. Half-rings for sealing the piston in the cylinder: a) two half-rings for creating a ring of the same level b) a complete set of half-rings for sealing one piston

The effect of temperature on the antifriction material was checked by two methods [14]. When exposed to a temperature of 900 °C for one hour, first on the paste sample, then on the ring, the groove was filled with paste. Then, the ring was dried. After that, the ring with the antifriction filling was also checked for the stability of fixing the paste at the same temperature level.

It follows from the experiments that after calcination, the antifriction material with a graphite content of 25% and a binder content of 75% (b-1) crumbled into separate grains as it did not have sufficient strength. With a ratio of graphite and binder of 50/50% (b-2), after calcination, the material did not scatter but cracked, as it did not have sufficient strength. With a ratio of graphite and binder of 75/25% (b-3), after calcination, the material was a sufficiently strong monolith without cracks and traces of destruction. Similarly, an assessment of the effect of temperature on the sealing rings with an antifriction paste embedded in its groove was carried out (Fig. 3). The rings were placed in a muffle furnace and kept at a temperature of 900 °C for one hour. After cooling, they were checked for the resistance of the antifriction filling of the groove and for the absence of deformation of the rings themselves. As the results of experiments show, the selected composition of the antifriction material in the groove of the ring did not collapse and maintained its strength in the ring. Measurements of the ring dimensions did not show any deviations from their initial size, and their acquisition of a black color indicates the absence of their changes and deformation during heating.

The stepped joint in the vertical plane of one ring did not exclude the hermetic separation of the over-piston and under-piston volume since openings for the passage of gases exist at the joints of the rings. In order to exclude the passage of gases from the piston volume, two rings with 90° joints separated relative to each other were placed in one common groove on the piston. At the same time, a gas-tight barrier was provided between the over-piston and under-piston chambers in the cylinder (i.e., when the rings were tightly pressed against the liner mirror, the gas density of the piston assembly was created).

O-rings have grooves, which must be filled with a solid antifriction material (Fig. 1). For this purpose, two sets of rings were made: one of bronze (Fig. 3) and one of steel.

a *b* *c*

Fig. 3. Checking for the thermal resistance of rings with antifriction and the filling of the groove: a) calcination of rings in a muffle furnace, b) cooling of the rings, and c) checking the durability of the antifriction and filling the groove with no deformation of the rings

5. RESULTS

In modern engine building, there is a tendency to use two compression rings in the cylinder-piston group of internal combustion engines. Considering that the main load from the gas pressure is absorbed by the first compression ring, the second ring accounts for a smaller part of the load. Thus, an oil removal ring with drainage holes in the piston wall is installed below the compression ring to drain the oil removed from the surface of the liner [21, 22].

The following values of the axial reaction and the moment of friction can be taken as the results of the embedding tests: linear wear of the counterbody scallops and the maximum depth of the friction track of the sealing material; the work of the friction forces during the embedding process; and the state of the friction surfaces, determined visually [2, 8, 12].

The cylinder sealing system using a piston with compression rings is shown in Fig. 4. Elastic *C*-shaped rings with a cut by their design did not provide complete sealing of the cylinder. Gases at a pressure of P_1 in the over-piston volume passed through the joint of the upper ring into the cavity between the two compression rings. At the same time, due to the resistance when gases pass through the gap in the first ring, the pressure P_2 in the cavity between the rings was reduced. Further, gases passed through the gap in the second ring into the piston volume with a pressure of P_3. Thus, a set of compression rings, by its design, does not provide complete tightness, as it works as a labyrinth seal, which are widely used in engineering, for example, in turbines [12].

According to the technical conditions, the compression ring must have elasticity in order to create a specific pressure on the liner wall of 0.5 MPa [10, 11]. Gases entering the cavity under the ring multiply the pressure of the ring on the liner wall [8]. This is especially evident when combustion occurs. A ring with a thickness of 3 mm at a length of 10 mm will experience a force on the sleeve wall equal to 15 N. If, as was done above, the combustion pressure is 10 MPa, then this section of the ring from the gas pressure will receive an additional force of 300 N (i.e., a force 20 times greater).

Based on the consideration of the design and operating conditions of the friction contact of the cylinder-piston group of new generation engines, it was established that the piston sealing parts in the cylinder should be made of copper or copper alloy using graphite. This gives the compression rings high thermal conductivity and low friction force, which determines the design features of the sealing unit. Copper and bronze, and especially composite materials with graphite, do not have elasticity. Moreover, composites are brittle. Therefore, rings should have a sufficiently large cross-section to ensure their strength, and their pressing against the surface of the cylinder liner should be carried out on a different basis than *C*-shaped elastic traditional rings made of alloyed cast iron. The solution to this question seems to be the consideration of two variants of rings. The first option is that the rings are made of a composite of copper or bronze with graphite. The second option is that the rings are made of copper or bronze with a graphite belt (Fig. 1). The rings have a cross-section of approximately 8 × 10 or 6 × 8 mm (the size of the radius is 8 ... 10 mm, the thickness of the ring is 6 ... 8 mm).

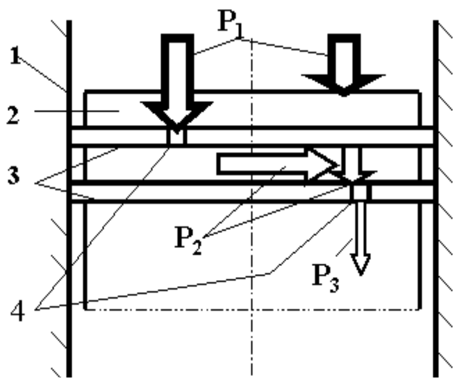

Fig. 4. The scheme of sealing the cylinder with a piston with compression rings. 1 = the surface of the cylinder liner; 2 = the piston body; 3 = compression rings; 4 = joints of compression rings. P_1 = the pressure of gases above the piston; P_2 = the pressure of gases in the cavity between the compression rings; P_3 = the pressure in the piston volume and the casing of the engine

The high level of operational characteristics of the developed ring made of the marked material was ensured by their tight pressing against the sliding surface of the sleeve and their separability. In this case, the stepped joint of the half-rings is formed in a vertical plane. This eliminates the penetration of gas under pressure into the cavity under the ring and the creation of uncalculated clamping forces. The rings are pressed against the surface of the sleeve with the help of springs, which are located on the back of the piston and act on the rings through the rods, which prevents the springs from heating, which weakens their tension.

During the operation of the engine, the surface of the rings wears out at a certain intensity in any case. Thus, the springs will constantly select the gap formed between the cylindrical surface of the liner and the adjacent surface of the rings [10, 11]. In this case, the radial gap between the ends of the half-rings at the joints will increase. As a result, the half-rings will compensate for the amount of wear on their outer surface due to their elastic properties. However, the fit density of the half-rings on the cylindrical surface will not be subject to wear, and the half-rings will move apart by the amount of wear on their outer surface. Therefore, the density of the cylindrical joint will remain the same. This will ensure a constant and unchangeable gas density of the piston throughout the entire service life of the cylinder-piston group.

Experiments were conducted to determine the compression on the stand when the engine was cold-scrolled. Compression was determined by a standard method using a compressor at a speed of 900 min^{-1} engine shaft rotation. In the experiments, compression was obtained in the traditional version of the cylinder 0.40 ...0.45 MPa and in the dry seal version 0.35 ... 0.40 MPa. The amount of compression depends on the degree of compression and the condition of the cylinder-piston group. It should be noted that an engine with an estimated compression ratio of 5 was used as an experimental engine. In the experimental version, in order to avoid the impact of the piston on the cylinder head, an increased gap was set between the bottom of the piston and the upper wall of the cylinder head, which reduced the geometric compression ratio. Therefore, a value of 0.35...0.40 MPa indicates sufficient compression in the cylinder.

A comparison of the measurement results on a standard engine and an experimental engine shows that the dry seal provides the necessary compression at the level of the engine's nominal value and that the friction of the piston shift with the experimental pistons in the cylinder liner is 58% from a standard seal with sleeve lubrication.

Wear assessment experiments were carried out only with a dry seal, taking into account the recommendations of *GOST 27860-88* [23], which contains recommended methods for measuring the wear of parts. The main experiments were carried out in hot mode (i.e., with the actual operation of the engine using recommendations for testing engines) [2, 20]. The results of the experiments were compared with data presented in the literature [24]. All problems with the wear of the cylinder-piston group were considered from the standpoint of the behavior of compression rings with developed seals. Therefore, a study on the wear of the cylinder liner was not carried out. The studies involved assessing the wear of the rings by changing their mass, radial size, and chord length [11, 20].

A pilot plant based on a serial engine has been created with the fixation of wear indicators and resources of the proposed sealing rings. The research facility is a test bench based on the *UD-2M* with samples of cylinders with a piston diameter of 72 mm. For testing, the engine was upgraded to a variant of the piston-sleeve pair in dry friction mode (Fig. 5).

Fig. 5. The reconstructed engine used to study the dry sealing of the piston in the cylinder

The seal design consists of two rings made of bronze or steel located in a common annular groove in the piston (Fig. 6). Each of the rings consists of two half-rings with a stepped joint of half-rings along a vertical plane, while the joints of the upper and lower rings are located 90° relative to each other. The half-rings were unclenched and pressed against the cylinder liner mirror by springs. At the same time, the stepped joints slid over each other within certain design limits to eliminate the breakthrough of gases through the locks of the rings into the space under the rings (Fig. 7).

Based on the analysis of the operation of the internal combustion engine, the technical solution was justified, and the design of a dry piston seal in a cylinder with self-sealing compression rings without the use of liquid engine lubrication was developed. In the course of comparative studies to characterize the process, the half-rings were marked, and the values characterizing their wear were periodically measured. Measurements were made after 100 engine hours of operation of the experimental engine.

The wear control of the sealing rings was carried out by three methods. The first method is the determination of wear by the weight of the rings and the loss of mass due to wear. Wear assessment by this method was carried out by weighing half-rings on analytical scales and fixing the weight after a certain time of sealing operation (Table 1).

The mass of the half-ring after 300 motor hours of sealing operation was 98.695% of the initial mass. The mass of the lost metal or the wear of the sealing element was 1.305%.

The second method, recommended by *GOST* [23], is a direct measurement of the radial size of the ring in the middle of the half-ring arc, subject to maximum wear where the most intense wear is possible. The measurements were taken with a micrometer at certain intervals (Table 2).

According to the results of the direct measurement of the radial size of the ring, wear for 300 motor hours was estimated at 0.175%. This value of wear in relative units was also confirmed by the assessment of wear in relation to the average initial radial size, which was 0.17%.

The results of the wear assessment according to previously proposed methods [11, 24], in which the change in the length of the chord is measured, are shown in Table 3.

In accordance with the above methodology for applying the wear assessment by changing the length of the chord, we could determine the coefficient of the multiplicity of the wear value with respect to the change in the length of the chord for specific experimental conditions with a ring diameter of 72 mm and an initial chord length within 28 mm (Table 3). Since the multiplicity coefficient changes its value depending on the amount of wear, we determined it within the range of the probable magnitude of the change in the radial size determined by direct measurement of the radial size with a micrometer, i.e., in the range of 0.01–0.02 mm (Table 4).

Table 1

Evaluation of ring wear by ring weight change

Ring		Half-ring weight									
		Initial		After 100 m/hour		After 200 m/hour			After 300 m/hour		
Marking		g	%	g	%	g	%	Average %	g	%	Average %
Pair of one half0-rings	0	42.00	100	42.00	100	41.00	97.62	98.76	40.60	96.67	98.24
	5	42.16	100	42.16		42.12	99.11		42.08	99.81	
Pair of 2 half-rings	2	40.00	100	40.00	100	39.90	99.75	98.80	39.86	99.65	99.15
	6	40.00	100	40.00		39.94	99.85		39.86	99.65	
Average performance		Average wear by weight for 300 m/hour: 1,305 %									98.695

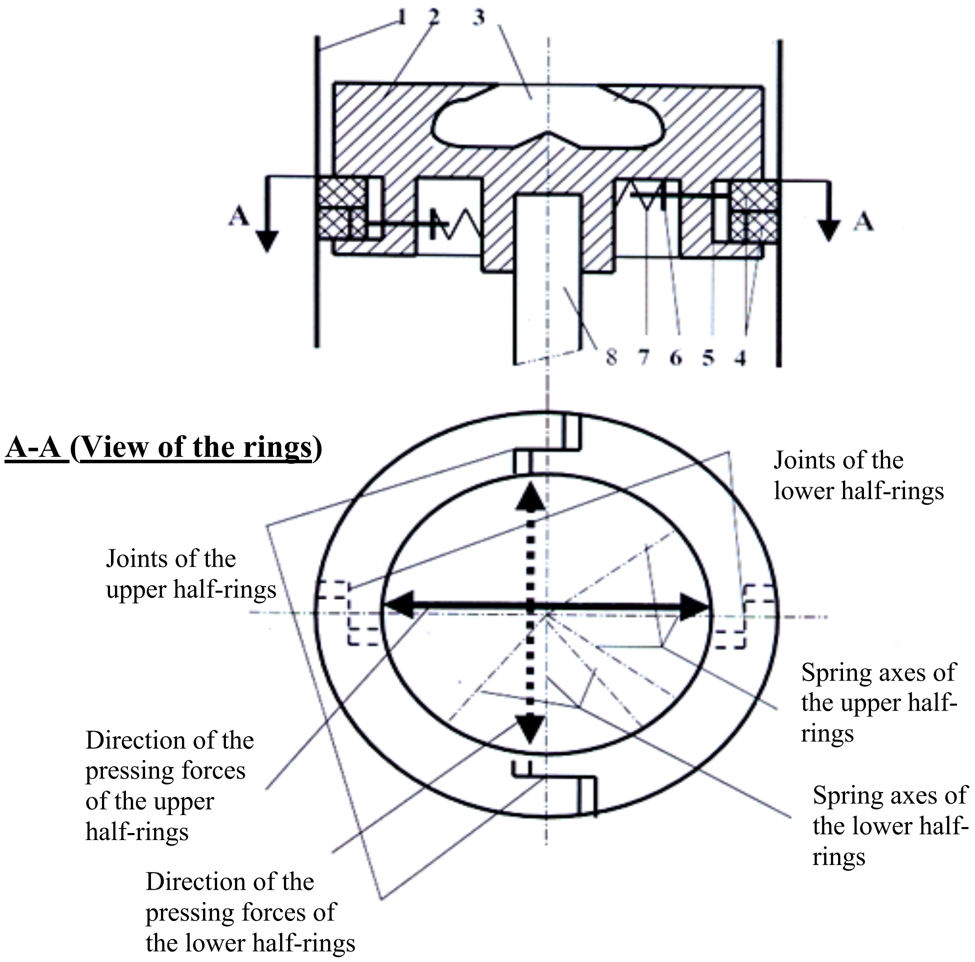

Fig. 6. The piston of the internal combustion engine: 1 = wall of cylinder liner; 2 = piston body; 3 = combustion chamber in the piston; 4 = compression rings made of an antifriction composite material, two rings in a common groove, each consisting of two half-rings; 5 = a groove in the core of the piston for placing rings; 6 = pins for pressing rings with stops; 7 = springs for pressing the rings to the cylinder mirror; 8 = piston rod

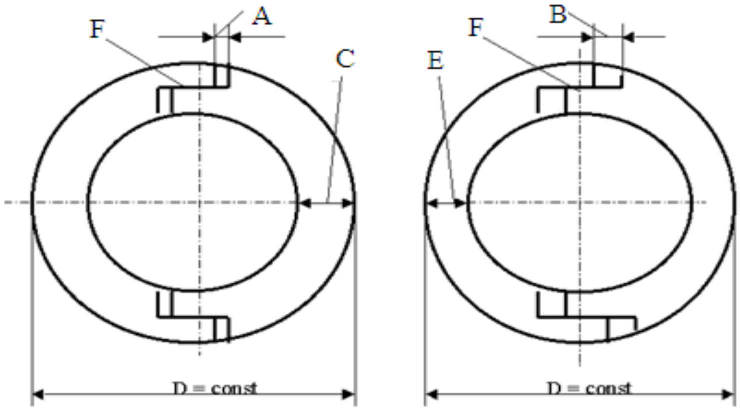

Fig. 7. Construction of compression rings with solid lubrication and a stepped joint of half rings: F = a sliding surface of half-rings joints; A = mounting gap; C = permissible increase in the gap during the wear of the half-rings; E = the width of the new ring; D = the width of the ring in the place of maximum wear; D=const = diameter of the liner

Considering that the radial size according to the direct measurement for 300 motor hours of operation was 0.016 mm and that the coefficient of the multiplicity of the conversion of the arc length changed into the true value of radial wear, we will take the average between the possible values of radial wear 0.010 and 0.020.

The value of the coefficient of the multiplicity of radial wear size to the change in the length of the chord for the specified conditions was within the range of 5.6–9.4.

Table 2

Ring wear measurements based on direct measurements of radial size

Ring		Radial size										Change in length
Marking		Initial size		After 100 m/hour		After 200 m/hour			After 300 m/hour			
		mm	%	mm	%	mm	%	aver.,%	mm	%	*aver., %*	*mm*
Pair of 1 semi-rings	0	9.400	100	9.400	100	9.390	99.89	98.930	9.380	96.67	99.865	0.020
	5	9.330	100	9.330		9.328	99.97		9.325	99.81		0.005
Pair of 2 semi-rings	2	9.200	100	9.200	100	9.190	99.89	99.935	9.180	99.65	99.780	0.020
	6	9.310	100	9.310		9.309	99.98		9.290	99.65		0.020
Average indicator		9.310		9.310		9.304			9.294		99.825	0.016
		The average wear based on measurements of the radial size for 300 hours is 0.175%										

Table 3

Evaluation of ring wear by chord length changes for 300 hours of sealing operation

Ring		Chord length, mm										Length change
		Initial length		After 100 m/hour		After 200 m/hour			After 300 m/hour			
Marking		mm	%	mm	%	mm	%	Average %	mm	%	Average%	mm
Pair of 1 half-rings	0	26.80	100	26.80	100	26.05	97.20	97.975	25.05	93.47	96,055	1.75
	5	28.10	100	28.10		27.75	98.75		27.55	98.04		0.55
Pair of 2 half-rings	2	28.00	100	28.00	100	27.61	98.61	98.535	27.26	97.36	97,140	0.74
	6	26.00	100	26.00		25.60	98.46		25.20	96.92		0.80
Average performance		27.23		27.23		26.75			26.27		96.597	0.96

The results obtained are a sufficient basis for a predictive assessment of the effectiveness of using the recommended dry seal design in real engines [11, 20]. Copper and copper alloys can be considered suitable structural materials in mechanical engineering if their properties mostly meet the requirements of the mechanisms in which they can be used [18]. In this case, bronze is the optimal option when designing a dry seal.

It should be noted that the last two methods determined the radial wear of the ring in linear units (mm). However, in essence, the results characterize wear in volumetric units since they were obtained experimentally at a certain ring height.

The obtained results of the evaluation of the work according to the first method (wear in weight units) were based on the scientific basis of the study of the properties of materials in the conditions of their operation in the cylinder-piston group, including the properties of composite materials for the creation of antifriction elements in mechanical engineering.

Table 4

Determination of the coefficient of the multiplicity of the size of the radial wear to change the length of the chord

Determined size	Formula for calculating	Numerical values for radial wear (Δh mm)		
		0.0	0.010	0.020
R*	R* = R - Δh	36.000	35.990	35.980
cos α*	Cos α* =OK / R*	0.9167	0.9169	0.9172
α*	Bytables	23° 34’	23° 32’	23° 28’
tg α*	Bytables	0.4358	0.4350	0.4330
L/2	L/2 = OKTg α*	14.383	14.355	14.289
L, L*	L = (L / 2) 2	28.766	28.710	28.578
ΔL	ΔL = L- L*	0.0	0.056	0.188
Divisiblefactor Δh	K = ΔL / Δh	-	5.6	9.4

Note: Initial data: ring diameter 72 mm, constant values calculated: R = 36 mm, h = 3 mm, OK = (R - h) = (36 - 3) = 33 mm)

6. CONCLUSIONS

The developed original design of the piston-sleeve coupling seal of the cylinder-piston group of an internal combustion engine has sealing bronze rings with grooves, which must be filled with a solid antifriction material at a ratio of graphite and binder glass of 75/25%. After being heated at a temperature of 900 °C for one hour, the antifriction material in the groove of the ring was a sufficiently strong monolith without cracks or destruction, and the dimensions of the ring did not show any deviations from the initial size.

The rings acquired a black color, which indicates the absence of changes and deformation during heating. The bronze ring with a radius of 8...10 mm and a thickness of 6 ...8 mm with a dry seal had a high level of operational characteristics, which is ensured by their tight pressing against the sliding surface of the sleeve and separability. At the same time, the stepped joint of the semicircles in the vertical plane eliminated the penetration of gas under pressure into the cavity under the ring and the creation of uncalculated clamping forces. The rings were pressed against the surface of the sleeve with the help of springs, which were located on the back of the piston and acted on the rings through the rods, which prevented the springs from being heated and their tension from being weakened.

A comparison of the measurement results on a standard engine and an experimental engine showed that the dry seal provided the necessary compression at the level of the engine’s nominal value of 0.35 ...0.40 MPa. The friction of the piston shift with the experimental pistons in the cylinder liner was 58% of that of a standard seal with sleeve lubrication. An evaluation of the operation of compression O-rings with detachable locks and antifriction belts using methods compared to changes in weight, radial size, and chord length showed that the proposed option had an advantage in terms of wear and provided tightness in the cylinder with some wear to the surface of the rings.

References

1. Chao, Cheng. *Power Cylinder System for Internal Combustion Engines*. 2017. 178 p. DOI: 10.5772/intechopen. 69762. Available at: https://www.intechopen.com/books/improvement-trends-for-internal-combustion-engines/power-cylinder-system-for-internalcombustion-engines.
2. Райков, И.Я. *Испытания двигателей внутреннего сгорания.* Москва: Высшая школа. 1975. 169 с. [In Russian: Raikov, I.Ya. *Tests of internal combustion engines*. Moscow: High School].
3. Bazhenov, Yu. & Kirillov, A. & Bazhenov, M. Examination of engine cylinder-piston group damages. In: *IOP Conf. Series: Materials Science and Engineering.* 2020. Vol. 896. Paper No. 012100. DOI: 10.1088/1757-899X/896/1/012100.
4. ГОСТ 27860-88. *Детали трущихся сопряжений. Методы измерения износа*. М. Государственный комитет СССР по управлению качеством продукции и стандартам (утвержден в Казахстане). Введен 1990.01.01. Москва: Государственный комитет СССР по стандартам, 1989. 30 с. [In Russian: *Details of rubbing interfaces. Methods of wear measurement.* Moscow: USSR State Committee for Product Quality Management and Standards (approved in Kazakhstan). Moscow: USSR State Committee on Standards].
5. Söderfjäll, M. *Friction in Piston Ring - Cylinder Liner Contacts*. Luleå: University of Technology. 2017. 223 p. ISBN: 978-91-7583-824-3. Available at: https://www.diva-portal.org/smash/get/diva 2:1076831/ FULLTEXT01.pdf.
6. Anglin, D.L. *Internal combustion engine*. 2018. DOI: 10.1036/1097-8542.349500.
7. Halderman, J.D. *Automotive Engines: Theory and Servicing.* 9th ed. Pearson. 2018.
8. Momani, W. Influence of internal combustion engine parameters on gas leakage through the piston rings area. *Modern Mechanical Engineering*. 2017. Vol. 7(1). No. 27. P. 27-33. DOI: 10.4236/mme. 2017.71003.
9. Sharoglazov, B.A & Kuanyshev, M.K. & Podsedov, K.D. Calculation-analytical estimation of the working cycle parameters of the crankless and crank engines under similar loading conditions. In: ICIE 2018: *Proceedings of the 4th International Conference on Industrial Engineering. Lecture Notes in Mechanical Engineering*. 2019. P. 109-118.
10. Заренбин, В.Г. & Колесникова, Т.Н. К расчету течения газов через кольцевые уплотнения ДВС с учетом динамики поршневых колец. *ДВС. Всеукраинский научно-технический журнал*. 2017. No. 1. P. 27-33. DOI: 10.20998/0419-8719.2017.1.06. [In Russian: Zarenbin, V.G. To the calculation of the flow of gases through the annular seals of the internal combustion engine, taking into account the dynamics of piston rings. *Internal combustion engines. All-Ukrainian Scientific and Technical Journal*].
11. Serdecki, W. Determination of compression ring wall pressure distribution. *Journal of POLISH CIMAC. Energetic aspects*. 2010. Vol. 5. No. 1. P. 171-178.
12. Serdecki, W. & Krzymień, P. Dependence of piston ring papametrs determination accuracy on the method of calculation. *Journal of KONES, Powertrain and Transport*. 2011. Vol. 18. No. 4. P. 435-441.
13. Собачкин, А.В. & et al. Исследование жаростойкости покрытия Ti-Al-Nb на модельных образцах поршневого алюминиевого сплава. *Ползуновский вестник*. 2020. No. 3. P. 88-92. DOI: 10.25712/ASTU.2072-8921.2020.03.016. [In Russian: Sobachkin, A.V. & et al. Investigation of the heat resistance of Ti-Al-Nb coating on model samples of piston aluminum alloy. *Polzunovsky Bulletin*].
14. Собачкин, А.В. & et al. Получение композиционных материалов системы Ti-Al-Nb методом высокотемпературного синтеза. *Эволюция дефектных структур в конденсированных средах: сборник тезисов XVI Международной школы-семинара*. 2020. P. 159-160. [In Russian: Sobachkin A.V. & et al. Obtaining composite materials of the Ti-Al-Nb

system by high-temperature synthesis. *Collection of abstracts of the XVI International School-seminar*].

15. Maryin, D. & et al. Results of engine tests of an experimental gasoline internal combustion engine. *International Scientific-Practical Conference "Agriculture and Food Security: Technology, Innovation, Markets, Human Resources*. *BIO Web Conf.* 2020. Vol. 17. P. 1-5. https://doi.org/10.1051/bioconf/20201700078
16. Синицын, В.А. & et al. Оценка теплового состояния деталей цилиндро-поршневой группы двигателей внутреннего сгорания с помощью критериев. *Алтайский государственный технический университет им. И.И. Ползунова. Барнаул, Ползуновский альманах*. 2018. No. 3. P. 10-16. [In Russian: Sinitsyn, V.A. et al. Assessment of the thermal condition of parts of the cylinder piston group of internal combustion engines using criteria. *Barnaul, Polzunovsky almanac*].
17. Guardiola, C. & Pla, B. & Bares, P. & Barbier, A. An analysis of the in-cylinder pressure resonance excitation in internal combustion engines. *Appl Energy.* 2018. Vol. 228. P. 1272-1279.
18. Kuanyshev, M. et al. The improvement of friction bearing manufacturing technology by using copper alloy JTP. *International Jurnal of Advnced Manufacturing Technology of plasticity*. 2017. Tom: 88. Release: 1-4. P. 317-324. DOI. 10.100. Available at: https://link.springer.com/article/10.1007/s00170-016-8758-2.
19. Valiev, M. & Kosimov, K. Diagnosing the technical condition of the diesel cylinder-piston group. *E3sWeb. Conf.* 2021. Vol. 264. Paper No. 04061. DOI: 10.1051/e3sconf/202126404061.
20. Arshad, W. & Hanif, M.A. & Bhutta, M.U. & Mufti, R.A. & Shah, S.R. & Abdullah, M.U. & Najeeb, M.H. Technique developed to study camshaft and tappet wear on real production engine. Ind. Lubr. *Tribol*. 2017. Vol. 69. P. 174–181. DOI: https://doi.org/10.1108/ILT-06-2016-0135.
21. Bazhenov, Y. & Bazhenov, M. Research of operational dependability of automotive engines. *Dependability*. 2018. Vol. 18(4). No. 22. DOI: 10.21683/1729-2646-2018-18-4-22-27.
22. Heywood, J.B. *Fundamentals of the Internal Combustion Engine*. McGraw Hill Education. 2018. Available at: https://www.pdfstall.online/2018/12/internal-combustion-engine-fundamentals.html.
23. ГОСТ 30480. *Обеспечение износостойкости изделий. Методы испытаний на износостойкость. Общие требования*. Введен 1998.07.01. Минск. Межгосударственный совет по стандартизации, метрологии и сертификации. Издательство стандартов. 1998. 16 p. [In Russian: *Ensuring the wearresistance of products. Methods of testing for wear resistance.* General requirements. Minsk. Interstate Council for Standardization, Metrology and Certification].
24. Farafonov, D.P. & Migunov, V.P. & Saraev, A.A. & Leschev. N.E. Abradability and erosion resistance of seals in turbine engine air-gas channel. *Proceedings of VIAM*. 2018. No. 8(68). P. 70-80. Federal State Unitary Enterprise «All-Russian Scientific Research Institute of Aviation Materials» State Research Center of the Russian Federation. DOI: 10.18577/2307-6046-2018-0-8-70-80.

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