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## *IMPROVING THE DEVELOPMENT OF A PISTON RING BASED ON SOLID LUBRICANT*

*In the article, in relation to the similar conditions of use of internal combustion engines with a crank mechanism and the main indicators of a piston seal having the same design parameters (in particular, cylinder diameter, piston stroke), a comparative assessment of the indicators characterizing their duty cycle and indicator parameters of each of the engines is made (in particular, by the magnitude of the indicator tangential and circumferential force, torque and indicator power). The authors carried out work on the use of sealing in the form of rings with a contact surface based on a solid antifriction material – graphite.*

*It is concluded that, in relation to similar conditions of use, the indicator torques formed during the working cycle of these machines are not equal in magnitude. The indicator torque of the crank mechanism of the engine exceeds the corresponding torque of the internal combustion engine. This conclusion also applies to the ratio of the indicator capacities of the considered thermal machines.*

*Keywords: Internal combustion engine, crank mechanism, working fluid, piston, piston ring, grease.*

**Introduction**

The piston seal in the form of a set of compression rings is one of the main elements of the engine that determines the reliability and efficiency of the engine. The efficiency of piston engines depends on a number of factors. Among them, friction losses, gas leaks during air compression and gas expansion largely determine the technical and economic performance of the engine. The design of compression rings in the form of two or three C-shaped rings made of alloyed cast iron has long been established in the engine industry [1]. At the same time, it is assumed that the sleeve is lubricated with drip oil from the crankcase, and the friction of the compression rings on the surface of the sleeve is liquid with a coefficient of friction equal to 0.07–0.08. However, in practice the situation is different.

Firstly, the oil does not get into the upper part of the sleeve, since it is blocked by the piston.

Secondly, the upper part of the sleeve has an elevated temperature, usually determined by the level of oil resistance and is a maximum of 220–230 °C. The temperature of the piston and cylinder is an important parameter for operational safety and service life. Exhaust gas temperatures vary between 600 to 850 °C for diesel engines, and 800 to 1050 °C for gasoline engines. At this temperature, the oil liquefies and loses its lubricating properties [2].

Thirdly, during the combustion process, when the gas pressure increases sharply, gases penetrate through the joint between the ends of the ring into the cavity between the ring and the piston body and create an additional force pressing the rings against the cylinder liner mirror, much exceeding the force set by the elasticity of the ring. Due to the heat capacity, the piston and other parts in the combustion chamber, it is impossible to accurately determine temperature fluctuations. But it can still be argued that there is a small amplitude of change in the temperature of the piston, albeit a few degrees, depending on the stroke, whether it is the intake or the working stroke. The bottom of the piston is the first to be heated by incandescent gases and absorbs a different amount of heat, depending on the stroke, engine speed and load. The high temperature is primarily discharged through the piston rings to the cylinder walls, and to a lesser extent, by the piston skirt.

**Materials and methods**

As a result, the friction in the cylinder liner, especially in its upper part, increases significantly. In this case, attention should be paid to the form of wear of the working surface of the compression rings and the surfaces of the upper and lower ends. They differ in shape and size in different engine models, and these differences mainly depend on the ratio of the height of the compression rings, the radial thickness and the size of the gap between the upper shelf of the piston groove and the upper end of the compression ring (Figure 1). It is known from the theory of friction and wear that wear is directly related to friction, which clearly confirms the increased friction of compression rings in the upper part of the cylinder liner [2].

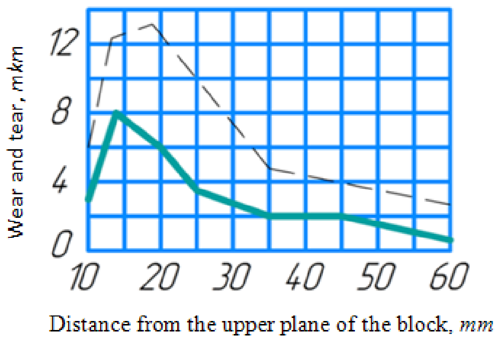


Figure 1 – The dependence of the wear of the cylinder liner on the height of the cylinder liner forming: solid line – average wear in eight directions, dotted line – maximum wear

Analyzing the above graph, if we assume that at a distance of 60 mm from the top of the sleeve, friction corresponds to liquid friction with a coefficient of friction of 0.07–0.08, then within the first 20 mm of the piston stroke, wear is 6–8 times more intense, which can only be with semi-dry or even dry friction of the rings. This means that the coefficient of friction in this part of the cylinder is at least 0.4–0.6.

The friction of the rings in the cylinder liner leads to a distortion of the cylindrical shape of the sleeve, which leads to a violation of the tightness of the piston contact [5] and a decrease in the quality of the engine (Figure 2).

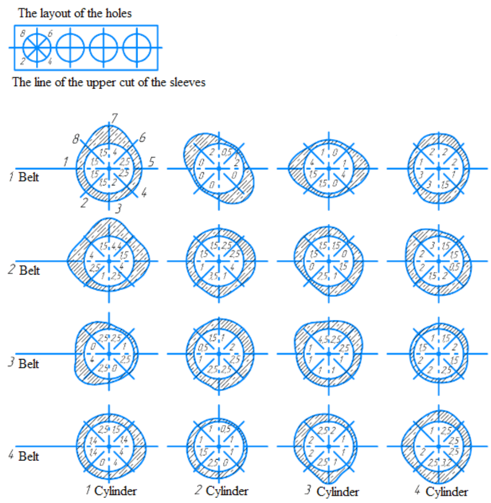


Figure 2 – The plot of the wear distribution along the cylindrical surface in the radial direction along the four belts of the cylinders of the internal combustion engine

Due to the wear processes in the piston contact, compression in the cylinders decreases after the first 500 hours of operation, which leads to a deterioration in the technical and economic indicators of the engine with a tendency to progressive deterioration of all indicators. Usually, the inter-repair period of automotive engines is assumed to be equal to 3000 engine hours. According to engine tests in real operating conditions, all engine parameters have changed significantly (compression, torque, 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.

In the field of improving the piston seal, 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 five-minute), L-shaped, triangular, etc. However, in an internal combustion engine, a piston seal with conventional rectangular rings remains one of the most conservative components in terms of design.

**Results and discussion**

The authors carried out work on the use of sealing in the form of rings with a contact surface based on a solid antifriction material – graphite [2]. Graphite is used in engineering as an antifriction material [9]. The use of graphite as a solid lubricant in a movable contact was adopted by analogy with the movable contacts of electric machines, where it is also required to have a tight contact of mutually moving elements and minimal wear of the contacting surfaces [7,8]. The coefficient of friction of graphite and composite materials based on it is equal to 0.15-0.20.

The seal design consists of two rings made of bronze or steel, located in a common annular groove in the piston. Each of the rings consists of two half-rings having a stepped joint of half-rings in a vertical plane, while the joints of the upper and lower rings are located in 90 degrees relative to each other. The half-rings are unclenched and pressed against the cylinder liner mirror by springs. At the same time, the stepped joints slide over each other within certain design limits, which eliminates the breakthrough of gases through the locks of the rings into the space under the rings (Figures 3-4).

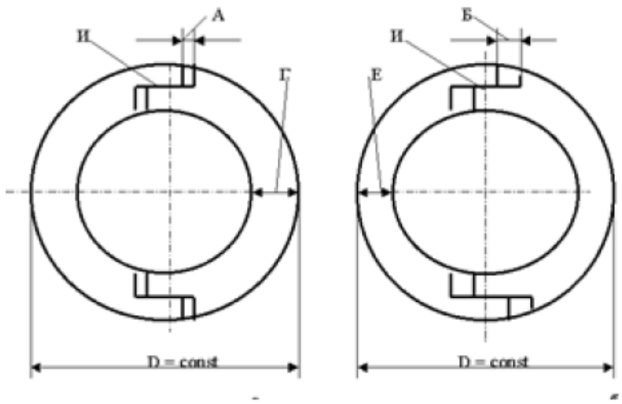


Figure 3 – Design of compression rings with solid lubrication
and stepped joint of half rings

И – the sliding surface of the half-rings docking; А – the mounting gap; Б – the permissible increase in the gap when the half–rings are worn; Г – the width of the new ring; Е – the width of the ring at the point of maximum wear, D = const – the diameter of the sleeve.

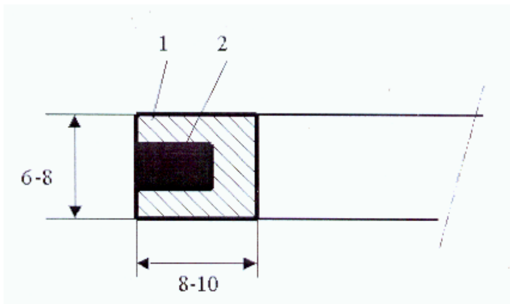


Figure 4 – Cross section of the ring: 1 – metal (bronze or steel);
2 – graphite filling (figures – dimensions in mm.)

The seal design was tested on an experimental engine based on the UD-2M engine with cylinders with a diameter of 72 mm [3]. The engine has been upgraded for cylinder operation conditions without lubrication in dry seal mode. As a result of experiments, it was found that the seal provides compression at the level of the engine's nominal value, the friction of the piston shift with the experienced pistons in the cylinder liner is 58 % of that for a standard seal with sleeve lubrication. Experiments to assess the intensity of wear with wear measurement using three methods (direct measurement of dimensions at the point of maximum wear, measurement of changes in ring mass and measurement of chord changes on the cylindrical surface of the ring, the method according to STST [4]) showed the operability of the piston seal, its reliability, as well as a resource commensurate with the resource of a typical seal, but, unlike the typical variant, without a noticeable decrease in quality indicators throughout the entire period of operation [8].

At the same time, an analysis of the operation of the pilot seal showed that not all potential opportunities for improving the quality of the piston seal were achieved [3,9]. To ensure the tight fit of the cylindrical surface of the rings to the cylinder mirror, the rings must be movable and constantly pressed by springs to the contacting surface of the sleeve. Taking into account the temperature deformations, the rings in the annular groove of the piston were installed with a gap. When the piston moves up or down, the rings are pressed against the upper or lower end surface of the annular groove due to friction. When a cyclic dose of fuel is burned, an increased gas pressure occurs in the over-piston volume of the cylinder [3]. Since the piston body has a certain gap, and the rings are pressed against the cylinder mirror by springs, the gas pressure acts on the upper end surface of the ring and presses the package of rings down within the gap. In this case, gases enter the gap above the upper ring into the annular cavity, creating an additional force of pressing the rings against the cylinder liner mirror. Accordingly, there is an additional force for the movement of the piston, as well as increased wear of the rings.

To eliminate this effect, a new ring design was developed. They are also made of two half-rings, have stepped joints of half-rings, two rings are located in a common annular groove in the piston. The difference lies in the shape of the cross sections of the rings.

The upper ring has a rectangular cross-section. To press it against the cylinder liner mirror, one or two C-shaped springs are used with joints placed in diametrically opposite places [7,9]. This ensures that the semi-rings are pressed against the contact surface of the sleeve without distortions.

The lower ring has a smaller diameter of the inner hole, and there is an angular bevel in its lower part. A steel C-shaped spring is placed in the corner groove under the ring. Acting on the angular surface, the spring simultaneously presses the lower half-rings to the cylinder liner mirror, and also creates a force that presses the entire set of rings to the upper end surface of the annular groove. Due to this, the breakthrough of gases into the annular cavity is excluded.

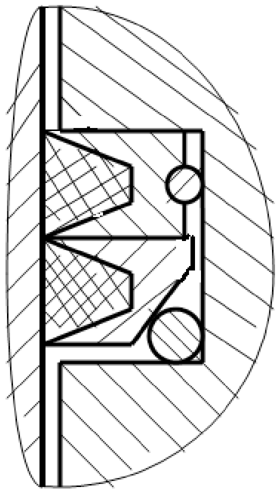

Figure 5 – Cross section of the package of half-rings with by pressing the rings to the upper end the surface of the annular groove by means of an annular spring and an oblique cut of the edge of the lower ring

The design of the ring allows you to calculate the main characteristics of such a piston seal. Thus, the pressing force of the upper ring can be determined by analogy with the brushes of electric machines. A specific pressure of 0.2 kg/cm$^2$ is recommended for them [9].

So, if the rings are designed for a cylinder with a diameter of 72 mm, and the height of the ring is assumed to be 6 mm, the area of pressing the ring against the cylinder mirror is approximately 1/3 on each side. At the accepted dimensions, the area of the force contact is about 23 cm$^2$. At a specific pressure of 0.2 kg/cm$^2$, the spring force for the upper ring should be equal to 0.46 kg. This is easy to adjust by placing the piston with the rings on a flat surface and loading the ring with a load at the top until the ring is completely drowned in the piston groove.

If the piston body has a gap in the cylinder liner of 0.25 mm (0.125 mm per side), the end surface of the ring in the gap perceives the pressure of gases at the time of the combustion phase, the area of such a surface is about 0.26 cm$^2$. At a gas pressure of 70 kg /cm$^2$ at the moment of combustion, the force squeezing the package of rings from the upper edge of the annular groove is 19 kg. Since the force of 19 kg is directed upwards, and in order to press the half-rings to the sleeve mirror, a horizontal force of about 0.5 kg is required, the angle of the bevel of the edge of the lower half-rings is determined [3.5]. At the given values, it is approximately 10 o with respect to the horizontal surface when using a spring with greater elasticity.

**Conclusions**

The considered design of the piston seal is applicable both for traditional engines with cylinder lubrication with drip oil, and for new engine designs in which the oil sump is isolated from hot cylinders operating with the use of solid lubricant.

Thus, the rings during engine operation have a tight contact with the cylinder liner mirror, and are also pressed against the upper end surface of the annular groove in the piston, which creates a sealed contact and eliminates the appearance of additional forces pressing the ring to the sleeve surface, providing minimal friction and low wear of the rings and the sleeve.

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## ҚАТТЫ МАЙЛАУ НЕГІЗІНДЕ ПОРШЕНЬДІК САҚИНАНЫ ДАМЫТУДЫ ЖЕТІЛДІРУ

*Мақалада іштен жану қозғалтқышын КШМ-мен қолданудың ұқсас шарттарына және бірдей құрылымдық параметрлері бар поршеньді тығыздаудың негізгі көрсеткіштеріне (атап айтқанда, цилиндрдің диаметрі, поршень жүрісі) қатысты, олардың жұмыс циклін және қозғалтқыштардың әрқайсысының индикаторлық параметрлерін сипаттайтын көрсеткіштерді салыстырмалы бағалау жасалады (атап айтқанда, тангенциалды және айналмалы күштердің индикаторлық шамасы, айналу моменті және индикаторлық қуат). Авторлар қатты үйкеліске қарсы материал-графит негізінде жанасу беті бар сақиналар түріндегі тығыздағышты пайдалану бойынша жұмыс жүргізді.*

*Осы машиналардың жұмыс циклі кезінде қалыптасқан ұқсас пайдалану жағдайларына қатысты индикаторлық моменттер шамасы бойынша тең емес деген қорытынды жасалады. Қозғалтқыштың КШМ индикаторлық айналу моменті іштен жану қозғалтқышының тиісті моментінен асады. Бұл қорытынды қарастырылып отырған жылу машиналарының индикаторлық қуаттылығының арақатынасына да қатысты.*

*Кілтті сөздер: Іштен жану қозғалтқышы; иінді-шатунды механизм, жұмыс сұйықтығы, піспек, піспек сақинасы, майлау.*

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## СОВЕРШЕНСТВОВАНИЕ РАЗРАБОТКИ ПОРШНЕВОГО КОЛЬЦА НА ОСНОВЕ ТВЕРДОЙ СМАЗКИ

*В статье применительно к сходственным условиям использования двигателей внутреннего сгорания с кривошипно-шатунным механизмом и основных показателей поршневого уплотнения, обладающих одинаковыми конструктивными параметрами (в частности, диаметром цилиндра, ходом поршня), делается сравнительная оценка показателей, характеризующих их рабочий цикл и индикаторные параметры каждого из двигателей (в частности, по величине индикаторных тангенциального и окружного усилий, крутящего момента и индикаторной мощности). Авторами проведена работа по использованию уплотнения в виде колец с контактной поверхностью на основе твердого антифрикционного материала – графита.*

*Делается заключение, что, применительно к сходственным условиям использования, формируемые в течении рабочего цикла этих машин индикаторные крутящие моменты не равны по величине. Индикаторный крутящий момент кривошипно-шатунного механизма двигателя превышает соответствующий момент двигателя внутреннего сгорания. Этот вывод распространяется и на соотношение индикаторных мощностей рассматриваемых тепловых машин.*

*Ключевые слова: Двигатель внутреннего сгорания, кривошипно-шатунный механизм, рабочая жидкость, поршень, поршневое кольцо, смазка.*

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