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ANALYSIS OF HEAT STRESSES OF THE PARTS OF THE CYLINDER–PISTON GROUP WITH HEAT-PROTECTIVE COATINGS IN AN INTERNAL-COMBUSTION ENGINE

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The process of heat exchange in the cylinder of an internal-combustion engine has been investigated and the heat stresses of the working surfaces of the piston and the cylinder head have been analyzed. A method for calculating the heat resistance, the strength, and the thickness of heat-protective coatings has been developed. Results of thermal-cycle testing of a piston with heat-resistant coatings, the properties of which were calculated by the method proposed, are presented.

Introduction. To increase the operating longevity and improve the failure-free performance of an internal-combustion piston engine, it is necessary to provide reliable operation of its most important parts subjected to high heat and mechanical loads. Among these parts is the combustion chamber comprising a cylinder head and a piston. It is known [1–5] that, in a number of cases, it is precisely the heat loads on the combustion-chamber parts that limit the possibilities of further boosting of an internal-combustion engine and improvement of its reliability. This is explained by the fact that an increase in the specific density of the heat flows through the working surfaces of the piston and the cylinder head leads to their heating and premature mechanical failure under the action of high temperatures. The heat resistance of the cylinder head and the piston of an internal-combustion engine can be increased if new, more powerful coolant loops [6–8] or various inserts and coatings, made from materials with a low heat conduction [9], for example, by the method of powder metallurgy [10, 11], are used in its design.

Analysis of works [12–16] shows that the formation of ceramic layers, resistant to heat loads, on the working surfaces of the combustion-chamber parts is a promising method of their heat protection. However, in this case, heat and mechanical stresses arise under temperature changes at the interface between the heat-protective coating and the piston-top surface since their materials have different coefficients of linear and volumetric expansion [17–22]. The values of these stresses determine the optimum thickness of the heat-protective coating. Unfortunately, in [23, 24], devoted to investigating the processes of formation of heat-protective and heat-resistant coatings on the working surfaces of the piston and the cylinder head of an internal-combustion engine, methods that would make it possible to calculate the heat and mechanical stresses of the surfaces of the combustion-chamber parts with account for their heat-stress state and calculate the heat resistance and the strength of the heat-protective coatings obtained are not presented. A method of calculating the optimum thickness of heat-protective coatings is presented in [25]; however, it gives no way of determining, for concrete operating conditions of an internal combustion engine, the value of the heat flow through an area of a heat-absorbing surface and estimating the effect of the thermocycling on its strength, which limits its suitability for determining the optimum thickness of heat-protective and heat-resistant coatings. Therefore, the above-described problem remains pressing, and its solution would make it possible to select heat-protective materials for the combustion chamber of an internal-combustion engine even at the stage of its designing and, therefore, to determine the optimum technological parameters of the process of formation of heat barriers.

The aim of the present work is analysis of the processes of heat exchange in the cylinder of an internal-combustion engine, investigation of the heat stresses of the combustion-chamber parts, and development of a method for calculating the optimum thickness of the heat-protective coating on the heat-absorbing surface of the piston and the cylinder head.

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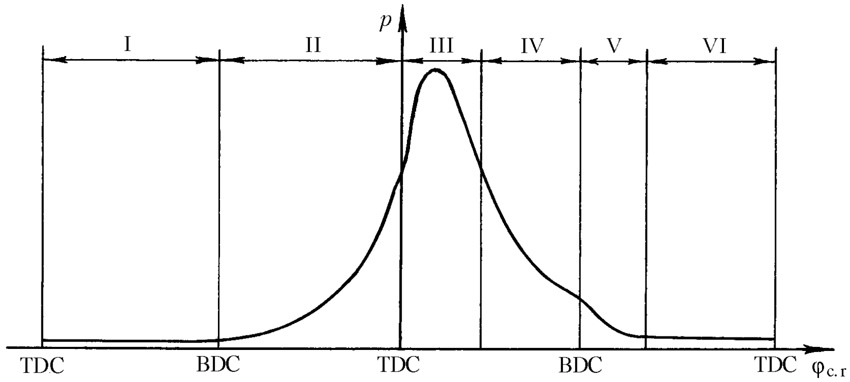

Fig. 1. Phases of the operation process: I) suction; II) compression; III) combustion; IV) expansion; V) release; IV) ejection. TDC is a top dead center, BDC is a bottom dead center.

Analysis of the Heat Exchange in the Combustion Chamber of an Internal-Combustion Engine. The temperature state of the combustion-chamber parts depends on the operation process of an internal-combustion engine determined by the source of heat supplied to the working medium that initially represents a mixture of air with the fuel and then the combustion products. The process of operation of an engine of this type can be divided into six characteristic periods (phases) differing in their heat-exchange conditions (Fig. 1).

I. The suction phase is characterized by relatively small changes in the composition of the working medium and in its temperature and pressure; however, the quantity of the working medium and the rate of its motion relative to the cylinder walls change sharply in the process of suction. In this period, a heat flow q is formed due to the convection and is defined by the Newton formula [26]

$$q = q_1 = \xi S \Delta T \tag{1}$$

or

$$q = \frac{d^2 Q}{dSdt}\,. \tag{2}$$

II. In the compression phase, the amount and composition of the working medium remain about the same, the pressure and temperature increase, and the heat flow itself and its direction change. In this case, as in the previous case, $q = q_1$.

III. The phase of intensive fuel combustion is characterized by a varying composition of the working medium, whose amount changes rapidly with temperature and pressure, and the existence of convective and radiative heat exchange. The intensity of heat exchange, which changes with the angle of rotation of the crankshaft, is determined by the character of the fuel-combustion process, in particular by the dynamics of heat release. In this case, the heat flow is due to convective heat exchange (q_1), radiative heat exchange in gases (q_2), and heat of the flame (q_3):

$$q = q_1 + q_2 + q_3\,, \tag{3}$$

where

$$q_2 = \varepsilon_w \psi \left[\varepsilon_g \left(\frac{T_h}{100} \right)^4 - (b_g + b_p) \left(\frac{T_w}{100} \right)^4 \right] S\,; \tag{4}$$

$$q_3 = \varepsilon_{fl} \varepsilon_w \psi \left[\left(\frac{T_{fl}}{100} \right)^4 - \left(\frac{T_w}{100} \right)^4 \right] S\,, \tag{5}$$

TABLE 1. External Heat Balance

Type of engine	Θ_1	Θ_2	Θ_3	Θ_4	Θ_5	Θ_6
Low-speed two-stroke	38–42	9–12	1–4	39–41	2–8	1–3
Moderately forced, moderate-speed four-stroke	37–40	10–17	1–3	36–42	2–5	3–6
Highly forced, moderate-speed four-stroke	39–42	10–13	3–4	33–38	6–8	1–3
Forced high-speed	37–40	11–23	2–6	35–41	4–8	1–4
Forced very-high-speed	35–39	14–30	3–6	30–41	3–7	1–4
Carburetor	23–32	20–35	–	30–55	–	2–3

TABLE 2. Distribution of the Heat Transfer over the Phases of the Working Process, %

Type of engine	Working stroke	Release	Compression
Carburetor	65–70	25–30	1–2
Diesel	80–90	5–15	5–8

in this case, $\psi = 5.7 \cdot 10^{-8}$ W/(m^2·K^4).

IV. At the final stage of the expansion phase, the following constants are established: the amount, composition, and volume of the working medium. Therefore, the total heat flow in this period is also defined by (3).

V. The phase of release of the combustion products is characterized by a varying amount of the constant-composition working heat at a sharply varying pressure and temperature

$$q = q_1 + q_2 \,. \tag{6}$$

VI. The ejection phase is characterized by a varying amount of the constant-composition working medium and relatively small changes in the temperature and pressure. In this period, convective heat exchange becomes dominant and so $q = q_1$.

There are publications [2–4] in which semiempirical analytical dependences for determining ψ, ε_g, ε_{fl}, and ε_w for different types of engines are presented. With them, one can calculate the numerical values of the heat flow in Eqs. (1), (3)–(5) with a certain accuracy.

Analysis of [2–5] shows that the external balance of the heat obtained as a result of the combustion of the fuel delivered to the cylinder of an engine is determined by the features of its operation process, the geometric dimensions of the cylinder, and the design of the engine parts and of the cooling system. With the parameter of relative heat

$$\Theta_i = \frac{Q_i}{Q_f} \cdot 100\% \,,$$

the heat balance will have the form shown in Table 1, where the external heat balance for different types of internal-combustion engines is presented. The distribution of the heat transfer over the phases of the operation process is shown in Table 2.

It follows from Tables 1 and 2 that

1) 1–6% of the heat released as a result of the fuel combustion is expended in heating the piston and 10–35% is expended in heating the cylinder head;

2) a major portion of heat is removed from the working medium through the combustion chamber during a working stroke of the piston.

In this case, 1–2.5% of the heat released as a result of the fuel combustion is removed through an uncooled piston and 2.5–6% of the released heat is removed through a cooled piston. The value of the heat flow formed at a definite instant of time at the piston top can be estimated by the semiempirical formula [5]

$$q_p = b_p v^{0.5} \left(p_m \, g \, \frac{T_{i.e}}{T_0} \right)^{0.88} \left(\frac{D}{\zeta p_{i.e}} \right)^{0.38} \,. \tag{7}$$

TABLE 3. Characteristics of the Final Operation Stage of an Internal-Combustion Engine (ICE)

Final stage of process	p, MPa	T, °C
Diesel ICE		
Suction	0.08–0.09	30–50
Compression	35–40	600–700
Combustion	6.0–9.0	2000–2200
Expansion	0.4–0.5	700–900
Carburetor ICE		
Suction	0.07–0.09	50–80
Compression	0.5–0.9	250–300
Combustion	3.0–3.5	2200–2500
Expansion	0.5–0.6	900–1400

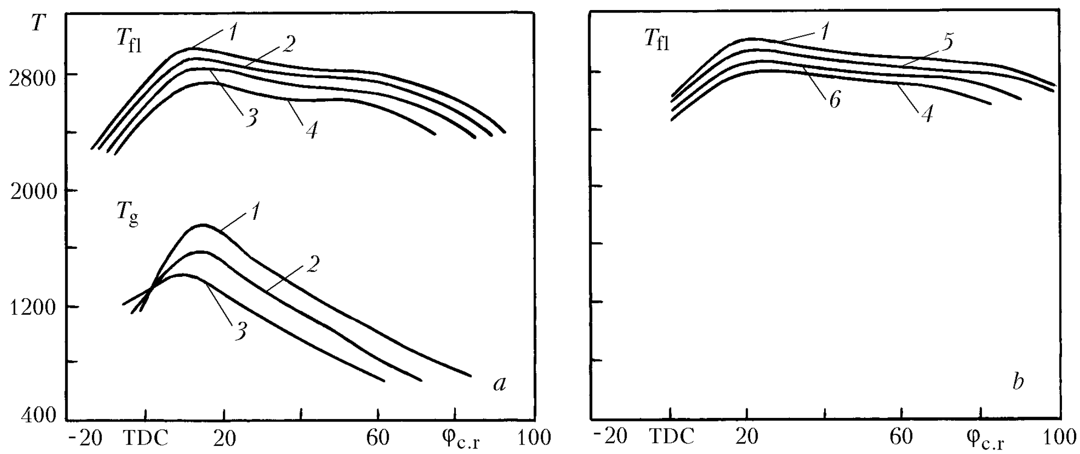

Fig. 2. Change in the temperature of the gas (T_g) and the flame (T_{fl}) versus the load: a) moderate-speed diesel (360 RPM), b) high-speed diesel (1700 RPM); 1) p_r = 100; 2) 80; 3) 60; 4) 40; 5) 84; 6) 56%.

Knowing the value of q_p determined from (7), one can estimate the temperature on the piston surface (T_1) [5]:

$$T_1 = T_0 + 103 a q_p \,. \tag{8}$$

At the center of the top of the uncooled cast-iron piston of a diesel engine a = 1.0, and a = 0.25–0.35 for an uncooled piston made from an alloy having a high thermal diffusivity and for cooled pistons.

Forty–sixty percent of the heat dissipated in the cooling system is removed through the cylinder head. In this case, the removed heat flow can be estimated by the formula [5]

$$q_h = b_h \frac{v^{0.6} p_m g T_{i.e}}{D^{0.5} (\zeta p_{i.e})^{0.4}} \,. \tag{9}$$

Using Eqs. (7), (8), and (9), one can estimate the heat stresses of the surfaces of the piston and the cylinder head with coatings and without them even at the stage of designing the combustion chamber of a new or a boosted internal-combustion engine.

In calculating the heat flows q_p and q_h, it is necessary to take into account the fact that the gas pressure and the temperature of the operation process of the majority of four-stroke engines lie within the ranges presented in Table 3.

It is evident that the heat exchange in the cylinder of an internal-combustion engine is pulsating in character. This explains the variations in the temperature of the flame and the gases (Fig. 2) affecting the heat-absorbing surfaces of the combustion-chamber parts.

The amplitude of the temperature variations in different sites of the surface of the combustion-chamber parts depends on the operating conditions of an internal-combustion engine and therefore can change from cycle to cycle because of their nonidentity. The temperature gradient on the surface of the combustion chamber of low-speed engines can account for 20% of the difference ΔT between the temperatures of the gas and the wall. Its value reaches 50^{o}C.

The temperature difference ΔT decreases markedly with increase in the speed of an engine. As a rule, it does not exceed 10–15^{o}C at a speed of 700–750 cycles/min.

The temperature variations on the surface of the parts, especially, of high-speed internal-combustion engines, are small, as compared to the difference between the temperatures of the gas and the wall, and comprise 1–2.5% of ΔT. Their ultimate values for the combustion-chamber parts made from different materials [6–8] are as follows: 250–350^{o}C for aluminum alloys, 400–450^{o}C for cast iron, 450–500^{o}C for structural steels, and 600–650^{o}C for high-temperature steels. Heating the material of the engine parts to temperatures exceeding the indicated ultimate temperatures leads to changes in the structure of the material and in its physicomechanical properties, with the result that the reliability of the parts sharply decreases. Therefore, the maximum temperature of pistons with a steel head rarely reaches the ultimate values of 350–400^{o}C and the temperature of pistons of aluminum alloys is usually lower than 300–350^{o}C.

The temperature of the surface of the engine parts that are in contact with the oil must not exceed the values at which a varnish and a carbon deposit begin to form rapidly. The investigations have shown that the amount of deposits on the surfaces of the parts of the cylinder–piston group depends insignificantly on the quality of the fuel and oil used. The varnish formation was detected at 180–200^{o}C for oils free of additions and at 240–260^{o}C for oils with additions. Under real conditions, the varnish formation begins, as a result of the action of aggressive hot gases and the formation of a high-temperature oil film on the cylinder mirror, at a lower temperature that markedly decreases as the oil ages.

The foregoing and the experience in using engines of different types suggest that the ultimate temperature of the piston top positioned above the first packing ring is 180–200^{o}C; for high-speed engines, on the service life of which very strict requirements are not imposed, this temperature is equal to 220–240^{o}C in the case where oils with additions are used. For modern, forced marine engines having an increased service life, the indicated temperature does not exceed 150^{o}C. For modern, reliable engines, the temperature on the piston-top side cooled by the oil is 200–220^{o}C.

The temperature distribution over the surface of the cylinder-cover flame bottom, especially of four-stroke engines is very nonuniform and varies within the following limits: 105–245^{o}C at the seat of the discharge valve and 170–380^{o}C at its injector, 205–230^{o}C at the periphery, and 265–390^{o}C at the discharge-valve strap.

The foregoing allows the following conclusions:

1. In deciding on the material of heat-resistant and heat-protective coatings, it is necessary to take into account the fact that the temperature at different sites of the surface of the combustion-chamber parts subjected to heat stresses may differ from the mean temperature by no more than $\pm 100^o$C.

2. Depending on the operating conditions of an internal-combustion engine, which are primarily determined by the load and the speed, the temperature of the working surfaces of the combustion-chamber parts can change by 10–15^{o}C during a cycle.

3. In deciding on the material of the coating, it is necessary to estimate the heat flows through the cylinder head and the piston top with account for the external heat balance and the distribution of the heat transfer over the phases of the process.

4. The thickness of heat-resistant and heat-protective coatings should be determined with account for the temperature state of the surface of the combustion-chamber parts. The temperature of this surface under the coating should not exceed the following values: 250–350^{o}C for aluminum alloys, 400–450^{o}C for cast iron, 450–500^o for steels, and 600–650^{o}C for high-temperature steels.

Method for Calculating the Heat Resistance and the Strength of Heat-Protective Coatings. Heat transfer occurring in the material of a coating due to its heat conduction is defined by the Fourier law [19]

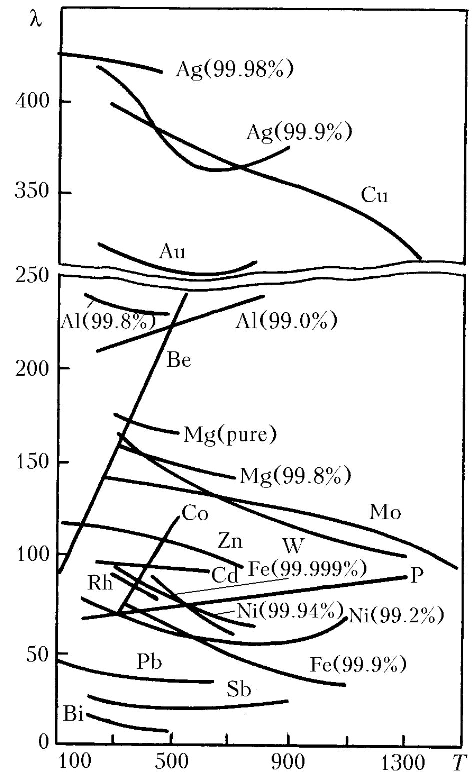

Fig. 3. Change in the heat conduction of metals and their alloys versus the temperature.

$$q_{\mathrm{h.a.s}} = -\lambda \,\mathrm{grad}\,(T)\,. \tag{10}$$

In this case, the heat conduction of coatings having a polycrystalline structure is due to the kinetic energy transfer caused by thermal motion of free electrons and interactions between the vibrating lattice sites. If it is assumed that the transfer of kinetic energy by the electron gas (characteristic of current-conducting coatings) plays a dominant role in the heat exchange, λ may be estimated as

$$\lambda = \frac{\pi^2}{3}\,\frac{k^2 n_{\mathrm{e}} l}{m_{\mathrm{e}} v}\,T\,. \tag{11}$$

It follows from dependence (11) that the heat conduction of the materials of such coatings may vary in a wide range. For metals and alloys, λ can change from 2 to 450 W/(m·K) in the temperature range 73–1273 K (Fig. 3). This change is predominantly determined by the concentration of impurities. For example, in the temperature range 300–700 K, λ changes by $\pm$(5–13)% for steels and by $\pm$(10–15)% for light alloys having a high heat conduction; therefore, the temperature dependence of the heat conductivity coefficient of the material of a coating can be ignored in preliminary estimation of its heat properties.

For coatings obtained by the powder-metallurgy method, it is necessary to take into account that [27]

$$\lambda = \lambda_0\,(1 - 1.5\Pi)\,. \tag{12}$$

If it is assumed that a heat-protective coating represents a heat barrier at which the temperature drops depending on the difference between the temperatures of the heat-absorbing (T_{11}) and the heat-releasing surfaces (T_{12}),

$$q_{\mathrm{h.a.s}} = G\,(T_{12} - T_{11}) = \frac{T_{12} - T_{11}}{R}\,, \tag{13}$$

where, for a plane surface,

$$R = \frac{h}{\lambda S}\,, \tag{14}$$

and, for a cylindrical surface,

$$R = \frac{\ln [d_1/d_2]}{2\pi\lambda L}\,. \tag{15}$$

It follows from (13) that the temperature change is related to R and $q_{\text{h.a.s}}$ by the relation

$$\Delta T = R q_{\text{h.a.s}}\,. \tag{16}$$

It is evident from (16) that the larger the heat flow through the heat-protective coating, the larger the difference between the temperatures of the heat-absorbing and the heat-releasing surfaces.

In calculating the heat resistance for a heat-protective coating with the use of (13)–(16), it is necessary to take into account that T_{12} must not exceed the ultimate temperature of the material of the piston and the cylinder head. In this case, the heat flow in (13) can be determined with the use of (7) and (9) or (3)–(6). The durability of such a heat barrier is determined by its strength properties.

The above analysis of the heat exchange in the combustion-chamber parts shows that in the material of the heat-protective coatings of the piston and the cylinder head there arises a complex stress that is due to the action of both the mechanical and the heat loads causing a three-dimensional or a uniaxial compression of the material or its extension.

The mechanical stress arising in dense and isotropic materials of coatings is defined as

$$\sigma = \frac{E_0}{3\,(1 - 2\mu)}\,\frac{\Delta V}{V} \tag{17}$$

in the case of three-dimensional compression (extension) and

$$\sigma = E_0\,\frac{\Delta h}{h} \tag{18}$$

in the case of uniaxial compression (extension).

It should be noted that σ comprises mechanical stresses caused by an external action on the material $\sigma_{\text{e.a}}$, the temperature difference inside the material $\sigma_{\text{t.d}}$, and the interaction between the media at the coating–base interface $\sigma_{\text{b.c}}$:

$$\sigma = \sigma_{\text{e.a}} + \sigma_{\text{t.d}} + \sigma_{\text{b.c}}\,, \tag{19}$$

which are calculated as

$$\sigma_{\text{e.a}} = E_{\text{c.m}}\,(\Delta h_{\text{c.m}}/h)\,, \tag{20}$$

$$\sigma_{\text{t.d}} = \alpha_{\text{c.m}} E_{\text{c.m}} \Delta T\,, \tag{21}$$

$$\sigma_{\text{b.c}} = (\alpha_{\text{b.m}} E_{\text{b.m}} - \alpha_{\text{c.m}} E_{\text{b.m}})\,\Delta T\,, \tag{22}$$

$$E_{\text{b.m}} = E_0\,(1 - K_{\text{s.s}}\Pi)\,. \tag{23}$$

TABLE 4. Dependences for Calculating the Ultimate Strength of Porous Materials [27]

Dependence	Values of constants	Author
$\sigma_{ult} = \sigma_{d.m}(1-\Pi)^j$	$j = 3–6$	M. Yu. Bal'shin
$\sigma_{ult} = \sigma_{d.m} \exp(-K_{s.s}\Pi)$	$K_{s.s} = 4–7$	E. Ryshkevich
$\sigma_{ult} = \sigma_{d.m}(1-\Pi^2) \exp(-K_{s.s}\Pi)$		N. I. Shcherban'
$\sigma_{ult} = \sigma_{d.m}(1-1.5\Pi)/(1-1.5K_{s.s}\Pi)$	$K_{s.s} = 2–2.25$	V. T. Troshchenko and A. Ya. Krasovskii
$\sigma_{ult} = \sigma_{d.m}(1-\Pi)/(-K_{s.s}\Pi)$		R. Kheines

It follows from (19)–(23) that the value of σ depends not only on the physicomechanical properties of the coating material but also on *S*, *h*, and *T*. These quantities should be selected such that σ is always smaller than the ultimate stress of the coating material (σ_{ult}) at which it does not yet experience a plastic deformation. To correct for the above-indicated quantities, we will introduce a comparative evaluation of the stress state of a surface, comprising three criteria:

1) the relative temperature of the heat-absorbing surface in characteristic sites (zones)

$$\Omega_r = T_{h.a.s}/T_{ult}\ ; \tag{24}$$

2) the relative mechanical stress of the coating material

$$\Lambda_{r.m.s} = \sigma/\sigma_{ult}\ ; \tag{25}$$

3) the relative strength of adhesion of the coating with the base.

If there is an adhesion bond between the base and the coating

$$\Lambda_{r.a.s} = \frac{\alpha_{b.m}E_{b.m} - \alpha_{c.m}E_{c.m}}{F} S\Delta T\ , \tag{26}$$

where *F* is determined by the ratio between the specific adhesion work *A* and the range of action of interatomic attracting forces

$$F = A/r\ . \tag{27}$$

In this case, *A* is calculated by the Young equation [27]:

$$A = U\,[1 + \cos(\varphi)]\ . \tag{28}$$

For high-melting metallic coatings applied by the method of powder metallurgy, it is necessary to take into account their porosity due to which their ultimate strength differs from the strength of dense materials. However, there are a number of dependences (Table 4) with which σ_{ult} can be calculated with a certain accuracy for powder materials too.

The ultimate strength of the material of protective ceramic coatings can be determined in the following way [28]:

$$\sigma_{ult} = \sigma_{d.m}\,(1 - \Phi\Pi)\ . \tag{29}$$

For a round pore $\Phi = 1.5$, for a pore in the form of a cube with an arbitrary orientation $\Phi = 1–3$, and for a cylindrical pore $\Phi = 1–1.27$.

It is evident that in deciding on the material of protective coatings on the basis of analysis of their stress state determined by dependences (17)–(23), it is necessary to select $\sigma_{e.a}$, $\sigma_{t.d}$, and $\sigma_{b.c}$ such that Ω_r, $\Lambda_{r.m.s}$, and $\Lambda_{r.c.s}$ is always smaller than unity. In this case, it is necessary to take into account the fact that the mechanical stresses arising

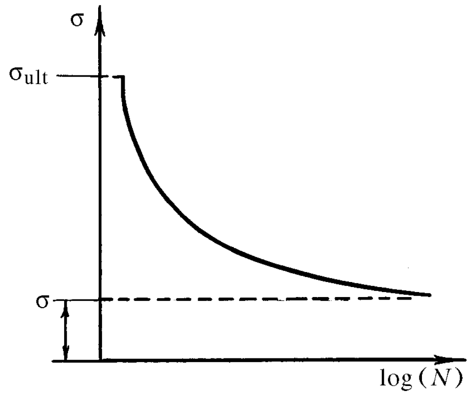

Fig. 4. Experimental curve of cyclic loads of coatings.

in the material of a coating are varying in character. Their cyclicity depends on the rate of rotation of the crankshaft of an engine. However, this parameter was not taken into account in earlier works despite the fact that it undoubtedly influences the values of the parameters calculated. Because of this, we attempted to estimate Ω_r, $\Lambda_{r.m.s}$, and $\Lambda_{r.a.s}$ more exactly.

It is known from [29–31] that in a symmetric cycle of varying loads the safety factor is determined by the value of the endurance strength of an engine part and is defined as

$$\Lambda_{s.m} = \chi\sigma_{e.lim}/\sigma_{max} , \tag{30}$$

where

$$\chi = \chi_{s.f}\chi_{s.r}/\chi_{s.c} . \tag{31}$$

The value of $\sigma_{e.lim}$ is determined by the curve (Fig. 4) constructed based on the experimental data.

In an asymmetric cycle of varying loads, the safety factor is determined by the analytical ratio

$$\Lambda_{s.m} = \frac{\sigma_{e.lim}}{\dfrac{\sigma_{e.lim}}{\sigma_{ult}}\sigma_1 + \dfrac{\sigma_2}{\chi}} , \tag{32}$$

where

$$\sigma_1 = 0.5\,(\sigma_{max} + \sigma_{min}) ; \tag{33}$$

$$\sigma_2 = 0.5\,(\sigma_{max} - \sigma_{min}) . \tag{34}$$

In deciding on the endurance limit of a heat-protective coating, it is necessary to take into account the fact that $\Lambda_{s.m}$ should be equal to the safety factor of the piston or the cylinder head of an internal-combustion engine.

The aforesaid allows the conclusion that the heat resistance of the heat-protective coating of the piston and the cylinder head of an internal-combustion engine can be calculated based on dependences (10)–(16); in this case, it is necessary to take into account the stress state of their surfaces which is determined on the basis of analysis of the heat exchange in the cylinder of the engine with the use of dependences (17)–(34).

Determination of the Thickness of Heat-Protective and Heat-Resistant Coatings of a Piston. With the use of the above-described method of calculating the thickness of a heat-resistant coating we have determined the optimum thickness of such a coating for a 245–1004021 piston (Fig. 5) of a high-speed MMZ internal-combustion engine No. 01171.01938 developing a maximum power at v = 30 m/sec and $p_{i.e}$ = 10 MPa. The material of the piston and the cylinder heat was an AL9 silumin. The dimensions of the combustion chamber corresponded to the engine of an MTZ-80 tractor.

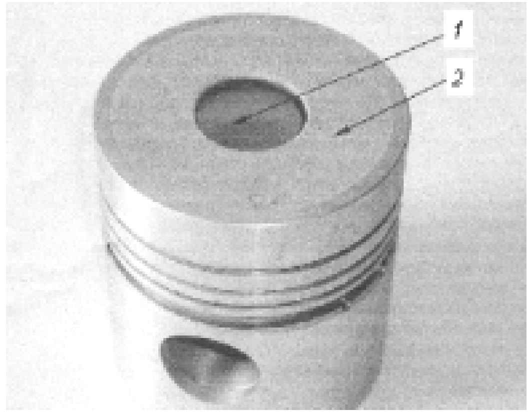

Fig. 5. Piston with a heat-protective and a heat-resistant AMDO coating on its combustion-chamber walls (1) and on the top (2).

As the material of the coating (heat-protective barrier) we used a ceramics based on alumina oxide with additions of silicon oxide for which $\alpha_{c.m} = 7.5 \cdot 10^{-6}$ K^{-1} and $\lambda_{c.m} = 60$ W/(m·K). In this case, the ultimate mean temperature of the material under the coating was not higher than 250^{o}C.

The calculation done with (8), (9), (12)–(17), (21)–(23), and (26)–(34) has shown that a piston on the heat-absorbing surface of which an oxide-ceramics coating of thickness 40 μm is applied should stand not less than 5000 thermal cycles at a temperature varied from 700 to 300 K, a piston with the same coating of thickness 60 μm should stand not less than 3800 thermal cycles under the same conditions, and a piston without a coating should stand 1800 thermal cycles at the same temperature variations.

Pistons with protective oxide-ceramics coatings formed by microarc anodizing were tested. The tests were carried out at the Minsk Engine Works. To determine the heat resistance of the heat-resistant coatings obtained by microarc anodizing, we heated the heat-absorbing surface of a piston by a high-frequency current to 700 K and then cooled it by a water shower to 300 K. The results of the tests correspond to the calculation data.

It follows from the foregoing that the calculation method proposed makes it possible to calculate the optimum thickness of heat-protective coatings on the combustion-chamber parts on the basis of analysis of their heat stress.

CONCLUSION

On the basis of analysis of the heat-exchange in the cylinder of an internal-combustion engine and investigation of the heat stress of the combustion-chamber parts, we have developed a new method for calculating the optimum thickness of a heat-protective coating on the heat-absorbing surface of the piston and the cylinder with account for the varying stresses arising in the combustion chamber in the process of operation of the engine.

NOTATION

A, work, J; a, dimensionless coefficient characterizing the material and design of the piston; b_h, dimensionless coefficient characterizing the heat flow through the head ($b_h = 147$ for a four-stroke internal-combustion engine (ICE) and $b_h = 262$ for a two-stroke ICE); b_g, dimensionless coefficient characterizing a heated gas flow; b_p, dimensionless coefficient characterizing the heat flow through the piston ($b_p = 1.0$ for a four-stroke ICE and $b_p = 1.78$ for a two-stroke ICE); D, diameter of the piston, 10^{-2} m; d_1 and d_2, outside and inside diameters of the cylinder, m; E_0, modulus of elasticity of the material (Young's modulus), Pa; $E_{c.m}$, modulus of elasticity of the coating material, Pa; $E_{b.m}$, modulus of elasticity of the base material, Pa; F, adherence of a coating with the surface of the base, H; G, heat conduction of the piston, W/(m·K); g, specific flow rate of the fuel, m^3/sec; h, thickness of a heat-protective layer, m; $K_{s.s}$, dimensionless coefficient characterizing the structural state of a porous body; k, Boltzmann constant ($k = 1.38 \cdot 10^{-23}$ J/K); L, height of the cylinder, m; l, mean-free path, m; m_e, electron mass ($9.1 \cdot 10^{-31}$ kg); N, number of loading cycles; n_0, electron concentration, m^{-3}; p, pressure of gases, Pa; $p_{i.e}$, pressure of the gas mixture upstream of the inlet elements of the engine, Pa; p_r, relative pressure, %; p_m, mean effective pressure of the gas mixture, Pa; Q, heat, J; dQ, change in the heat, J; Q_i, amount of heat given to a thermodynamic-system element as a result of the fuel combustion, J; Q_f, amount of heat released as a result of the fuel combustion, J; q, heat flow, W; q_1, convective heat flow, W; q_2, radiative heat transfer at temperatures lower than the flame temperature, W; q_3, heat given by the flame, W; q_h, heat transfer by the cylinder head, W; q_p, heat flow through the heat-absorbing surface of the piston, W; $q_{h.a.s}$, heat flow through a heat-absorbing surface, W; R, heat resistance of a coating, W/K; r, range of action of interatomic forces, m; S, area of a surface, m^2; T, temperature, K; T_0, temperature under normal conditions (T_0 = 293 K); T_1, temperature of the piston surface after the fuel combustion, K; T_{11}, temperature of a heat-absorbing surface, K; T_{12}, temperature of a heat-releasing surface, K; $T_{i.e}$, temperature of the gas mixture upstream of the inlet elements of the engine, K; T_h, temperature of the cylinder head, K; T_g, temperature of the gas mixture in the combustion chamber, K; T_{ult}, ultimate temperature of a heat-protective coating, K; T_{fl}, flame temperature, K; T_w, temperature of the combustion-chamber walls, K; $T_{h.a.s}$, temperature in a given site of the heat-absorbing surface, K; t, time, sec; U, free energy, J; V, volume, m^3; v, mean velocity of the piston, m/sec; $\alpha_{c.m}$, coefficient of linear expansion of the coating material, K^{-1}; $\alpha_{b.m}$, coefficient of linear expansion of the base material, K^{-1}; Δh, linear deformation, m; ΔT, temperature change, K; ΔV, volume change, m^3; ε_w, effective blackness of the wall; ε_g, blackness of the gas; ε_{fl}, blackness of the flame; ζ, space factor; ξ, heat-transfer coefficient, W/(m^2·K^4); Θ_i, relative heat released in the thermodynamic system; $\Lambda_{s.m}$, safety margin; $\Lambda_{r.m.s}$, relative mechanical stress of the coating material; $\Lambda_{r.a.s}$, relative adhesion strength; λ, heat-conductivity coefficient of a material, W/(m·K); λ_0, heat-conductivity coefficient of a dense material, W/(m·K); $\lambda_{c.m}$, heat-conductivity coefficient of the coating material, W/(m·K); μ, Poisson ratio; Π, porosity; σ, mechanical stress, Pa; σ_{max}, maximum mechanical stress, Pa; σ_{min}, minimum mechanical stress, Pa; $\sigma_{e.a}$, mechanical stress caused by an external action, Pa; σ_{ult}, ultimate mechanical stress, Pa; $\sigma_{d.m}$, ultimate mechanical stress of a dense material, Pa; $\sigma_{b.c}$, mechanical stress arising at the base-coating interface, Pa; $\sigma_{e.l}$, endurance-limit mechanical stress, Pa; $\sigma_{t.d}$, mechanical stress caused by a temperature difference, Pa; Φ, coefficient dependent on the pore shape; φ, wetting angle, rad; $\varphi_{c.r}$, angle of rotation of the connecting rod, deg; χ, surface-state coefficient; $\chi_{s.f}$, roughness scale factor; $\chi_{s.r}$, surface-roughness factor; $\chi_{s.c}$, effective strength reduction factor; ψ, blackbody-radiation constant; Ω_r, relative temperature. Subscripts: 0, initial state; 1–6, order of enumeration; 11, heat-absorbing surface; 12, heat-releasing surface; e, electron; i, summation index; j, exponent; max, maximum value; min, minimum value; e.a, external action; i.e, inlet elements; h, cylinder head; g, gas; ult, ultimate value; s.m, safety margin; d.m, dense material; s.c, stress concentration; s.f, scale factor; b.m, material of the base; c.m, material of the coating; r.m.s, relative mechanical strength; b.c., base-coating; r.a.s, relative adhesion strength; r, relative; p, piston; e.lim, endurance limit; fl, flame; t.d, temperature difference; m, mean; s.s, structural state; w, wall; h.a.s, heat-absorbing surface; c.r., connecting rod; s.r, surface roughness.

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