УДК 621.311.61
DOI: 10.31992/2074-0530-2021-47-1-46-53

METHOD FOR DETERMINING THE TEMPERATURE FIELDS OF THE SPARK PLUG

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This article examines the main factors that determine the thermal performance of a spark plug in the temperature range from 300 to 2500 Kelvin. The optimal value of the temperature of the heat cone was determined. A technique and algorithms for the numerical simulation of the thermal state of a spark plug are presented. These made it possible to calculate the dependence of the thermal conductivity coefficient of ceramic elements of a plug and the specific heat capacity of ceramic insulator on temperature. The calculation of the working cycle in the engine cylinder was carried out. The calculation of the temperature distribution of heat fluxes in the elements of the spark plug design was performed.
The assessment of the thermal characteristics of the spark plug is carried out by the method of numerical modeling of the operating cycle of an internal combustion engine. The calculation of the instantaneous temperature distribution in the body of the spark plug and on its surface is carried out. Calculations of the intensity of heat fluxes between the spark plugs and adjacent parts of the working fluid were carried out. The modeling of the operating cycle for various operating modes of the engine was made. The temperature fields of the spark plugs were determined. An array of initial data for calculating the temperature fields of the spark plug was formed. Dependences of the temperature of the working fluid in the vicinity of the spark plug on the angle of rotation of the crankshaft are determined. The harmonic components of the heat transfer coefficients between the working fluid and the cylinder fire guard (Voshni coefficient) are considered. The harmonic components of the heat flux density are considered. Calculations of the heat field of the spark plug are carried out for various operating modes of the engine, using the finite element method. The calculation of the temperature field of the spark plug by the finite element method was carried out using ANSYS, SolidWorks, Inventor, etc.

Keywords: *numerical simulation, spark plug, thermal state, thermal characteristics, temperature field.*

Cite as: *Yakhutl' D.R., Maleyev R.A., Zuyev S.M., Shmatkov YU.M., Ryabykh YE.A. Method for determining the temperature fields of the spark plug. Izvestiya MGTU «MAMI». 2021. No 1 (47), pp. 46–53 (in Russ.). DOI: 10.31992/2074-0530-2021-47-1-46-53*

Introduction

Heat flow power emitted by elements of its design, such as the thermal cone of an insulator and the central and side electrodes, to the adjacent layer of the fuel–air mixture in the combustion chamber of a gasoline engine at the compression stroke primarily determines the thermal characteristic of a spark plug (SP) [1]. Insufficient flow power results in the formation of carbon deposits. If its power is higher than a certain critical value, then the pre-flame reactions are sharply accelerated in the heated volume of the mixture, and spontaneous combustion occurs, which generates a self-propagating flame, that is, potash ignition (PI) [2].

Study aim

This article aims to investigate the main factors that determine the thermal characteristics of a SP at 300 Kelvin to 2500 Kelvin.

Main part

The minimum volume (ΔVf) of the flame "germ" depends on the air–fuel mixture composition and gas-dynamic conditions in the cylinder. ΔVf can be calculated using the method proposed by B. Lewis and G. von Elbe [3].

t_q is the time required to heat this volume to the temperature (T_q) at which pre-flame reactions are generated, which is calculated

on the basis of the conditions of heat transfer of the heated structural elements of the SP of the air–fuel mixture in the cylinder, and pq is the pressure in the cylinder at t_q [4]. Then, the moment (t_{ign}) of PI occurrence, counted from the moment of closing the intake valve, can be calculated using the following equation:

$$t_{ign} = \Delta t_q + 19.75\left(\frac{ON}{100}\right)^{3.4107} p_q^{-1.7} \exp\left(\frac{3800}{T_q}\right) \quad (1),$$

where ON is the gasoline octane number.

The occurrence of PI in a gasoline-engine cylinder is calculated on the basis of the evident inequality t_{ign}:

$$\frac{180 - \varphi_a - \theta}{n} > t_{ign} \quad (2),$$

where φ_a is the angle of the intake valve closing lag, indicating the degrees of the crankshaft rotation (CR); ϑ is the ignition advance angle, and n is the engine CR speed, min^{-1}.

Inequality (2) implies that spontaneous combustion of the fuel–air mixture (PI) occurs before the moment of spark formation.

A numerical simulation of the working cycle of the internal combustion engine (ICE) is required to evaluate the thermal characteristic of the SP, accompanied by the calculation of the instantaneous distribution of temperatures T(*t,x,y,z*) in the SP body and on its surface and the intensity of heat fluxes between the SP and adjacent parts of the working body.

Full execution of this program requires significant resources even for modern computing aids. Therefore, for effective modeling of thermal processes, the problem formulation should be simplified.

Thus, we assume that the SP thermal characteristics for the steady-state loaded operating mode of the ICE must be evaluated, and the solution of the numerical simulation problem must be divided into two stages. At stage 1, the working cycle in the engine cylinder is calculated, and at stage 2, the temperature distribution and heat flows in SP structural elements are calculated.

In addition, the numerical simulation of the ICE working cycle can be performed because the temperature of the elements of the cylinder fire protection enclosure (cylinder head, piston fire surface, and cylinder lateral surface) do not depend on the CR angle, whereas the temperature of the working fluid changes by an order of magnitude from T_{min} ≈300 K to T_{max} ≈3000 K.

Therefore, heat propagation is expressed as follows:

$$pc\frac{\partial T}{\partial t} = \mathrm{div}(\lambda \,\mathrm{grad}\, T) \quad (3),$$

where p is the substance density, kg/m^3; c is the heat capacity, and λ is the coefficient of thermal conductivity, W/m • K.

Thermophysical parameters p, c, and λ of metal structural elements are almost independent of temperature T [5]. Therefore, for the elements of the fire protection enclosure of the cylinder, Eq. (3) turns into a linear equation:

$$\frac{\partial T}{\partial t} = \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2}\right) \quad (4),$$

where x= λ /(pc_v) is the thermal diffusivity.

Based on the ICE theory, heat exchange at the boundary between the working fluid and the fire surface of the cylinder occurs in accordance with Newton's law [6]:

$$\lambda\frac{\partial T_\omega}{\partial n} = \alpha_w (T_\omega - T_\infty) \quad (5),$$

where T_ω is the surface temperature; T_∞ is the temperature of the working fluid in the cylinder; $\partial/\partial n$ is the derivative along the normal to the surface, and α_w is the coefficient of heat transfer (thermal emissivity) between the working fluid and fire protection enclosure of the cylinder.

The coefficient α_w is calculated in accordance with the equation proposed by Professor Voshni:

$$\alpha_w = 0.12793 \cdot 10^{-3} D_c^{-0.2} T_\infty^{-0.53} p^{0.8} W^{0.8} \quad (6),$$

where D_c is the cylinder diameter, and p is the pressure measured in bar. The W component in Eq. (6) represents a complex function of the engine geometrical and operating parameters.

In the steady state, the working cycle duration is t = const. Therefore, α_w and the product $\alpha_w T_\infty$

are periodic functions of time, which can be represented as Fourier series:

$$\alpha_w(t) = \overline{\alpha_w} + \\ +\sum_{k=-\infty}^{\infty}[\alpha_{c,k}\,\mathrm{soc}(k\omega t)+\alpha_{c,k}\sin(k\omega t)] \quad (7),$$

$$q_w(t) = \alpha_w T_\infty = \overline{q_w} + \\ +\sum_{k=-\infty}^{\infty}[q_{c,k}\,\mathrm{soc}(k\omega t)+q_{c,k}\sin(k\omega t)] \quad (8),$$

where ω = 2ϖ/t is the angular frequency of the working cycle.

For simplicity, neglecting the surface curvature, Eqs. (4) and (5) can be rewritten as follows:

$$\frac{\partial T_\omega}{\partial t} = X\frac{\partial^2 T_\omega}{\partial x^2} \quad (9),$$

$$\lambda\frac{\partial T_\omega}{\partial x}|_{x=0} = \alpha_w(t)T_w(t,0) - q_\omega(t) \quad (10).$$

Let L be the conditional wall thickness, and on the surface $x = L$, heat exchange occurs with the cooling system, which has a temperature T_L. Therefore, the following equation is valid:

$$\lambda\frac{\partial T_\omega}{\partial x}|_{x=L} = \alpha_L(t)(T_w(t,L) - T_L), \quad a_L = \mathrm{const} \quad (11).$$

The linearity of Eq. (9) and boundary conditions under Eqs. (10) and (11) enables to write the solution to problem under Eqs. (9) (11) as follows:

$$T(x,t) = \bar{T}(x) + \tilde{T}(x,t) + \tilde{\tilde{T}}(x,t) \quad (12)$$

where $\bar{T}(x)$ is the time-independent (constant) component; $\tilde{T}(x,t)$ is the free pulsating component, and $\tilde{\tilde{T}}(x,t)$ is the forced pulsating component.

The function $\bar{T}(x)$ is the solution to the problem:

$$\begin{cases} \dfrac{d^2T(x)}{dx^2} = 0 \\ \lambda\dfrac{d\bar{T}(x)}{dx}|_{x=0} = \bar{\alpha}_w\bar{T}(0) - q_w \\ \lambda\dfrac{d\bar{T}(x)}{dx}|_{x=L} = \alpha_L(\mathrm{\bar{T}(L)} - \mathrm{T}_L \end{cases} \quad (13).$$

This solution is expressed as follows:

$$\bar{T}(x) = \\ = \frac{\lambda(\bar{\alpha}_w T_w + \bar{\alpha}_L T_L) + L\bar{\alpha}_w\bar{\alpha}_L T_\omega - x\bar{\alpha}_w\bar{\alpha}_L(T_\omega - T_L)}{\lambda(\bar{\alpha}_w + \bar{\alpha}_L) + L\bar{\alpha}_w\bar{\alpha}_L} \quad (14).$$

Given that $T_w > T_L$, it is a decreasing linear function of the x coordinate, that is, the distance from the cylinder fire surface.

The pulsating components $\tilde{T}(x,t)$ and $\tilde{\tilde{T}}(x,t)$ have the form of Fourier series:

$$\tilde{T}(x,t) = \\ = \sum_{n=1}^{\infty}\left[\tilde{T}_{c,n}(x)\cos(n\omega t) + \tilde{T}_{c,n}(\mathrm{x})\sin(n\omega t)\right] \quad (15),$$

$$\tilde{\tilde{T}}(x,t) = \\ = \sum_{n=1}^{\infty}\left[\tilde{\tilde{T}}_{c,n}(x)\cos(n\omega t) + \tilde{\tilde{\mathrm{T}}}_{c,n}(x)\sin(n\omega t)\right] \quad (16),$$

which no longer contain permanent terms. By separating the variables to determine the coefficients of series under Eq. (15), we obtain the following system of equations:

$$\begin{cases} pc_v n\omega\tilde{T}_{c,n}(x) = \lambda\dfrac{d^2\tilde{T}_{c,n}(x)}{dx^2} \\ \lambda\dfrac{d\tilde{T}_{c,n}(\mathrm{x})}{dx}|_{x=0} = \alpha_w\tilde{T}_{c,n}(0) + \alpha_{c,n}\tilde{T}(0) \\ -\lambda\dfrac{d\tilde{T}_{c,n}(x)}{dx}|_{x=L} = \alpha_L\tilde{T}_{c,n}(\mathrm{L}) \\ -\mathrm{pc}_v\, n\omega\tilde{T}_{c,n}(x) = \lambda\dfrac{d^2\tilde{T}_{c,n}(x)}{dx^2} \\ \lambda\dfrac{d\tilde{T}_{s,n}(x)}{dx}|_{x=0} = \bar{\alpha}_w\tilde{T}_{s,n}(0) + \alpha_{s,n}\bar{T}(0) \\ -\lambda\dfrac{d\tilde{T}_{s,n}(x)}{dx}|_{x=L} = \alpha_L\tilde{T}_{s,n}(L) \\ n = 1,2,3,\ldots \end{cases} \quad (17)$$

Using symbolic mathematics software packages such as Wolfram Mathematica or Maple, an analytical solution to the system of equations (17) can be obtained. Without writing out the solution because of its cumbersomeness,

we present the main result of analysis. The functions $\tilde{T}_{c,n}(x)$ and $\widetilde{T}_{s,n}(x)$ decrease exponentially in the direction from the fire protection enclosure of the cylinder because they are proportional to the following equation:

$$A(x) = \exp\left(-x\frac{\sqrt{n\omega\rho c_v}}{2\lambda} \right) \qquad (18).$$

The Fourier coefficients of expansion under Eq. (16) satisfy the system of equations:

$$\left\{ \begin{array}{l} pc_v n\omega \tilde{T}_{c,n}(x) = \lambda \dfrac{d^2\tilde{T}_{c,n}(x)}{dx^2} \\ \lambda \dfrac{d\tilde{T}_{c,n}(x)}{dx}|_{x=0} = -q_{c,n} \\ -\lambda \dfrac{d\tilde{T}_{c,n}(x)}{dx}|_{x=L} = 0 \\ -pc_v n\omega \tilde{T}_{c,n}(x) = \lambda \dfrac{d^2\tilde{\tilde{T}}_{s,n}(x)}{dx^2} \\ \lambda \dfrac{d\tilde{T}_{s,n}(x)}{dx}|_{x=0} = -q_{c,n} \\ -\lambda \dfrac{d\tilde{T}_{s,n}(x)}{dx}|_{x=L} = 0 \\ n = 1,2,3,... \end{array} \right\} \qquad (19).$$

The solution of the system of equations (19), performed by using Wolfram Mathematica package of symbolic mathematics, also exhibits an exponential decrease based on the law described by Eq. (18).

The abovementioned analysis shows that when heat propagates in the elements of the fire protection enclosure of the cylinder (cylinder head, piston, and cylinder liner), a thermal "skin effect" is observed, and temperature pulsations are practically concentrated in a thin layer of the protection enclosure material [7]. Based on the work data, the penetration depth of temperature pulsations is 0.5 mm for aluminum elements of the cylinder structure, 0.3 mm for steel elements of the structure, and 0.15 mm for ceramic SP parts. Therefore, when calculating thermal processes in the ICE combustion chamber, with sufficient accuracy for practice, the temperature of the surfaces of the cylinder fire protection enclosure at steady-state operation mode of the ICE does not depend on the rotation angle of the crankshaft, that is, it is taken constant. However, the calculation of this temperature requires the solution of the thermal conductivity equations (3) or (4) the pulsations of the working fluid temperature in the ICE cylinder and operating modes of the cooling system.

In contrast to the calculation of the temperature field of metal elements, the ICE cylinder is also designed, where the thermophysical parameters of the material can be considered constant. When calculating the temperature field of a SP, the dependence of the heat capacity and thermal conductivity coefficient of its ceramic parts on temperature T can be observed [8]. Thus, the dependence of the thermal conductivity and specific heat capacity of corundum ceramics on temperature is presented as follows:

$$\lambda(T) = \frac{1.063\cdot 10^4}{T} + 0.420 - 8.08\cdot 10^{-3}T + 4.35\cdot 10^{-6}T^2 \qquad (20),$$

$$C(T) = \frac{0.345\cdot 10^5}{T} + 1.123 + 0.126\cdot 10^{-3}T \qquad (21).$$

Figures 1 and 2 present the graphs of these functions.

A characteristic aspect of the $\gamma(T)$ function is the minimum at a temperature of T~1500 K. The presence of this minimum ensures the temperature stability of the heat cone of the SP with fluctuations in the working fluid temperature T_∞.

Further discussion of the properties of the SP temperature field will be discussed below. Thus, mathematical modeling of the ICE working cycle should be performed.

When developing an algorithm for the numerical simulation of thermal processes in an ICE cylinder, attention should be paid to two of these processes, namely, compression and combustion expansion (Fig. 2), which are characterized by the highest rates of change in temperature, pressure, and thermophysical parameters of the working fluid [9]. Experience indicates that gas exchange processes have little effect on the thermal characteristics of the SP, and they can be neglected when simulating the temperature field of the SP [10].

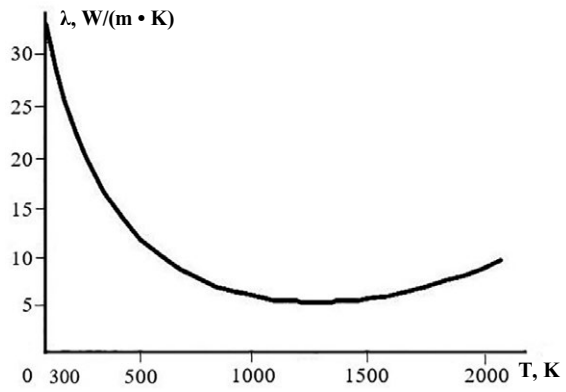

Fig. 1. Dependence of the thermal conductivity of the insulator ceramic λ on the temperature T

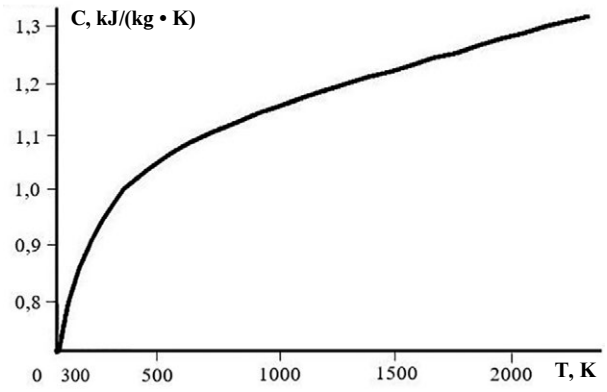

Fig. 2. Dependence of the specific heat capacity of the insulator ceramic C on temperature T

Compression is based on the energy balance equation in the ICE cylinder, which is written as follows:

$$M_a C_{vu} \frac{dT_u}{d\varphi^o} + p\frac{dV}{d\varphi^o} + \frac{1}{6n}\alpha_w \sum_{i=1}^{3} F_i (T_u - T_{w,i}) = 0 \quad (22),$$

where φ°, deg., is the rotation angle of the crankshaft; the index "*u*" ("unburned") refers to the unburned mixture; $T_u = T_u(\varphi°)$, K, is the instantaneous temperature value of the unburned (fresh) fuel–air mixture; $p = p(\varphi°)$, MPa, is the instantaneous value of pressure in the cylinder; $V = V(\varphi°)$, m^3, is the instantaneous value of the cylinder volume; n, min^{-1}, is the CR frequency; M_a, kmol, is the amount of the working fluid in the cylinder; $C_{Vu} = C_{Vu}(T_u)$, kJ/(kmol × K), is the heat capacity of the working fluid at constant volume; $T_{w,i}$, K, is the average temperature of the *i*-th section of the cylinder fire protection enclosure (block head, cylinder liner, and piston fire surface); F_i, m^2, is the area of the *i*-th section of the cylinder fire protection enclosure.

The instantaneous values of the cylinder volume and area of its parts are considered as given functions φ°, and the heat capacity C_{Vu} is a given function of temperature, M_a = const. The dependence of the heat transfer coefficient α_w on the process parameters is described using Eq. (6). Thus, Eq. (22) comprises two unknown functions, namely, *p* and T_u. The differential form of the Clapeyron–Mendeleev equation of state is used to complete the definition of the problem:

$$\frac{dT_u}{d\varphi^o} = \frac{1}{M_a R_\mu}\left(V\frac{dp}{d\varphi^o} + p\frac{dV}{d\varphi^o}\right) \quad (23),$$

where R_μ is the universal gas constant.

The simplest model of combustion (Fig. 3) in gasoline engines divides the working fluid division into three zones:

- the zone of the unburned part of the charge containing M_u kilomoles of a mixture, consisting of air, residual gases, and fuel vapors; occupying volume V_u; and containing T_u;
- the area of the burned part of the charge containing M_b kilomoles of combustion products and T_b and occupying volume V_b;
- the zone of the flame front, characterized by the adiabatic combustion temperature T_{fad} and volume Δf as the depth of the flame zone and S_f as its area.
-

The instantaneous value of gas mixture pressure $p(\varphi°)$ in all zones is equal.

If the *x*-th part of the charge has burned out by a certain moment of time *t*, then M_u is determined using the following equation:

$$M_u = (1 - x) M_\alpha R_\mu T_u \quad (24).$$

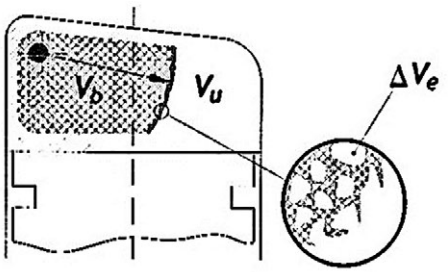

Fig. 3. Diagram of the combustion process in a gasoline internal combustion engine: V_b – area of combustion products; V_u – area of fresh mixture; ΔV_e – burning out globule of the fresh mixture

The equation of state of the gas mixture, which forms the unburned part of the charge, is expressed as follows:

$$pV_u = (1-x)\,\mathrm{M}_\alpha\, R_\mu T_u \qquad (25).$$

Performing logarithmic differentiation with regard to the rotation angle (φ°) of the crankshaft, this equation can be presented as follows:

$$p^{-1}\frac{dp}{d\varphi^o} + V_u^{-1}\frac{dV_u}{d\varphi^o} = T_u^{-1}\frac{dT_u}{d\varphi^o} - (1-x)^{-1}\frac{dx}{d\varphi^o} \qquad (26).$$

Similarly, the number of kilomoles M_b of combustion products is determined using the following ratio:

$$xM_b = x\mu_0 M_\alpha \qquad (27),$$

where μ_0 is the coefficient of molecular change in the working fluid during combustion.

The equation of the gas mixture state in the zone of the burned part of the charge is presented as follows:

$$pV_b = xM_b R_\mu T_b \qquad (28).$$

The differential form of this equation is as follows:

$$p^{-1}\frac{dp}{d\varphi^o} + \frac{dV_u}{d\varphi^o}V_b^{-1}\frac{dV_b}{d\varphi^o} = x^{-1}\frac{dx}{d\varphi^o}T_b^{-1}\frac{dT_b}{d\varphi^o} + M_b^{-1}\frac{dM_b}{d\varphi^o} \qquad (29).$$

In the presence of residual gases in the unburned mixture, the coefficient of the molecular change of the working fluid during combustion can be written as follows:

$$\mu_0 = \frac{M_b}{M_a} = \left\{\begin{array}{l} \dfrac{0,21(1-\alpha)L_0 + g_H/4 - 1/m_F + \gamma_r}{(\alpha L_0 + 1\dfrac{1}{m_F}(1+\gamma_r)}, \text{ if } \alpha < 1 \\ \dfrac{\alpha L_0 + \dfrac{g_H}{4+\gamma_r}}{(\alpha L_0 + \dfrac{1}{m_F})(1+\gamma_r)}, \text{ if } \alpha \ge 1 \end{array}\right\} \qquad (30),$$

where m_F is the molar mass of the fuel, kmol.

Eq. (30) uses the notations of L_0 as the amount of air required for complete combustion of 1 kg of liquid fuel, kmol/kg; α as the excess air coefficient; g_H as the amount of hydrogen in 1 kg of fuel; m_F as the molar mass of the fuel; and γ as the coefficient of residual gases.

For gasoline of standard composition, g_H = 0.145 kg, m_F = 114 kg/kmol, and the balance equation of gasoline is C8H8.

During combustion, when turning the crankshaft through an angle dφ°, the number of kilomoles of aerated concrete mixture in the zone of the unburned part of the charge changes by value $d[(1-x(\varphi^\circ)M_a] = -M_a dx(\varphi^\circ)$, thereby changing the enthalpy of this zone by $-M_a H_u(T_u)dx(\varphi^\circ)$. Therefore, the power balance equation in this zone is as follows:

$$-H_u(T_u)M_a\frac{dx}{d\varphi^o} = \frac{d}{d\varphi^o}[M_a(1-x)U_u(T_u)] + p\frac{dV_u}{d\varphi^o} + \frac{dQ_{w,u}}{d\varphi^o} = 0 \qquad ,$$

where $\delta Q_{w,u}$ is the elementary amount of heat lost (received) by the zone from the fire protection enclosure of the cylinder.

After performing some elementary calculations, we obtain the following differential equations:

$$M_a[(1-x(\varphi^o))C_{v,u}(T_u(\varphi^o))\frac{dT_u(\varphi^o)}{d\varphi^o} + R_\mu T_u(\varphi^o)\frac{dx(\varphi^o)}{d\varphi^o}] + pT_u(\varphi^o)\frac{dV_u(\varphi^o)}{d\varphi^o} + \frac{\delta Q_{w,u}(T_u,\varphi^o)}{d\varphi^o} = 0 \qquad (31).$$

An elementary portion $M_a dx(\varphi^\circ)$ of the unburned mixture, which have entered the flame zone, burns adiabatically, turning into $M_b dx(\varphi^\circ)$ kilomoles of combustion products. Its combustion time (δt) satisfies the condition $\delta t \ll (6n)^{-1}\, d\varphi^\circ$; therefore, combustion can be considered as instantaneous. The adiabatic flame temperature (T_{fl}^{ad}) can be determined from the enthalpy balance condition by using the following equation:

$$C_{v,u}(T_u)\frac{dT_u}{d\varphi^o} - \mu_0 C_{p,b}(T_{fl}^{ad})\frac{dT_{fl}^{ad}}{d\varphi^o} = 0 \qquad (32).$$

The power balance equation in the zone of combustion products is written as follows:

$$H_b(T_{fl}^{ad})\frac{d}{d\varphi^o}[M_b x(\varphi^o)] =$$

$$= \frac{d}{d\varphi^o}[M_b x(\varphi^o)U_b(T_b)] + p\frac{dV_b(\varphi^o)}{d\varphi^o} + \frac{\delta Q_{w,b}}{d\varphi^o} \quad (33).$$

In classical combustion schemes for a gasoline–air mixture, when a stoichiometric mixture burns, only H_2O and CO_2 molecules are obtained (namely, complete combustion). Moreover, during the combustion of nonstoichiometric fuel–air mixtures, the coefficient of molecular change does not depend on time (i.e., angle φ°) but rather on the excess air coefficient α. In this case, after the necessary transformations, Eq. (33) can be rewritten as follows:

$$M_b x C_{v,b}(T_b)\frac{dT_b}{d\varphi^o} + p\frac{dV_b}{d\varphi^o} +$$
$$+M_b[U_b(T_b) - H_b(T_{fl}^{ad})]\frac{dx}{d\varphi^o} + \frac{\delta W_{w,b}}{d\varphi^o} = 0 \quad (34).$$

In solving the abovementioned system of equations, the equation of the burnout of the air–fuel mixture proposed by prof. Vibe is used, which is written as follows:

$$\frac{dx}{d\varphi^o} = 6.908\frac{m+1}{\varphi^o z}\left(\frac{\varphi^o + \vartheta^o}{\varphi^o z}\right)^m (1-x) \quad (35),$$

where $\varphi_z°$ is the duration of combustion, measured in degrees of CR; ϑ^0 is the ignition advance angle, CR degrees, and m is the combustion indicator.

The heat transfer $\frac{\delta Q_{w,u}}{d\varphi^o} = \frac{\delta Q_{w,b}}{\varphi^o}$ between the zones of the working fluid and elements of the fire protection enclosure of the cylinder is described using the following equations:

$$\frac{\delta Q_{w,u}}{d\varphi^o} = \alpha_w \sum_{i=1}^{3} F_{i,u}(T_u - T_{w,i}) \quad (36),$$

$$\frac{\delta Q_{w,b}}{d\varphi^o} = \alpha_w \sum_{i=1}^{3} F_{i,b}(T_u - T_{w,i}) \quad (37).$$

In these equations, the heat transfer coefficient α_w is calculated using the Voshni equation (6), and the function $F_{i,u}(\varphi°)$ and $F_{i,b}(\varphi°)$ denotes the areas of the *i*-th element of the cylinder fire protection enclosure, washed by the fresh mixture and combustion products, respectively.

Research results and their discussion

Based on the results of numerical modeling of the operating cycle of the considered ICE, the following values are calculated:

1) average indicator pressure of the cycle p_i;
2) approximation using a trigonometric polynomial of degree *N*, that is, the working fluid temperature in the vicinity of the SP:

$$T_\infty(\varphi^o) = \overline{T}_\infty + \sum_{n=1}^{N}\left[\alpha_n \cos\left(n\varphi^o\right) + b_n \sin(n\varphi^o)\right] \quad (38);$$

3) approximation of the heat flux using a trigonometric polynomial of degree *N*:

$$q_w(\varphi^{o0}) = \overline{q}_w + \sum[cn\cos(n\varphi^o) + dn\sin(n\varphi^o)] \quad (39).$$

The coefficients of Eqs. (38) and (39) are written in the form of tables suitable for subsequent use in programs for the numerical simulation of the SP field.

Conclusion

This study investigates the main factors determining the thermal characteristics of the SP and describes the methodology and algorithms for the numerical simulation of the thermal state of the SP. Numerical simulation of the ICE working cycle has been performed. The article substantiates the need to study the thermal diffusivity of the SP ceramic elements to obtain information on the properties of the temperature field of the SP for analysis and calculation. The advantages and disadvantages of the proposed algorithm for calculating the thermal characteristics of a SP are presented, and safe and optimal operating modes of SPs have been developed.

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МЕТОДИКА ОПРЕДЕЛЕНИЯ ТЕМПЕРАТУРНЫХ ПОЛЕЙ СВЕЧИ ЗАЖИГАНИЯ

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В данной статье исследованы основные факторы, определяющие тепловую характеристику свечи зажигания в диапазоне температур от 300 до 2500 градусов Кельвина. Определено оптимальное значение температуры теплового конуса. Представлена методика и алгоритмы численного моделирования теплового состояния свечи зажигания, которые позволили выполнить расчеты зависимости коэффициента теплопроводности керамических элементов свечи и удельной теплоемкости керамики изолятора от температуры. Проведен расчет рабочего цикла в цилиндре двигателя. Выполнен расчет распределения температуры тепловых потоков в элементах конструкции свечи зажигания.
Проведена оценка тепловой характеристики свечи зажигания методом численного моделирования рабочего цикла двигателя внутреннего сгорания. Проведен расчет мгновенного распределения температур в теле свечи зажигания и на ее поверхности. Проведены расчеты интенсивности тепловых потоков между свечей зажигания и прилегающих к ней частей рабочего тела.
Проведено моделирование рабочего цикла для различных режимов работы двигателя. Определены температурные поля свечей зажигания. Сформирован массив исходных данных для расчета температурных полей свечи зажигания. Определены зависимости температуры рабочего тела в окрестностях свечи зажигания от угла поворота коленчатого вала. Рассмотрены гармонические составляющие коэффициентов теплопередачи между рабочим телом и огневым ограждением цилиндра (коэффициент Вошни). Рассмотрены гармонические составляющие плотности теплового потока. Проведены расчеты теплового поля свечи для различных режимов работы двигателя с использованием метода конечных элементов. Расчет температурного поля свечи методом конечных элементов производился с использованием программ ANSYS, Solid Works, Inventor и др.

Ключевые слова: *численное моделирование, свеча зажигания, тепловое состояние, тепловые характеристики, температурное поле.*

Для цитирования: *Яхутль Д.Р., Малеев Р.А., Зуев С.М., Шматков Ю.М., Рябых Е.А. Методика определения температурных полей свечи зажигания // Известия МГТУ «МАМИ». 2021. № 1 (47). С. 46–53. DOI: 10.31992/2074-0530-2021-47-1-46-53*