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PHYSICOMATHEMATICAL MODEL OF THE WORKING PROCESSES OF A SLIDE-VALVE COMBUSTION CHAMBER OF CONSTANT VOLUME

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A physicomathematical model of the working processes of a constant-volume combustion chamber, which can be used in promising power plants with a pulsating working process, has been created. A computational experiment the results of which coincide in the main with the data of experimental investigations has been carried out.

Interest in pulse jet engines has increased at present. The main unit determining the perfection of an engine is the combustion chamber. The greatest thermodynamic advantage is offered by combustion at constant volume (V = const); however, the two-valve chambers required for this are structurally complex and they have large overall dimensions and mass which mainly depend on the frequency of working pulsations. The proposed combustion chamber of V = const with a self-driven slide valve is devoid of these drawbacks.

The arrangement of such a combustion chamber as part of a pulse jet engine is shown in Fig. 1. During the operation of the chamber, in rotation of the slide value 2, we have the following successive processes: filling the slide valve with air from inlet 1; injection of fuel by injector 3; ignition by the spark plug 4 and combustion in a closed volume; escape of gases via the outlet device 6 and blowing. Part of the escaping gases pass through nozzle 5 in the slide valve, creating the torque in it (this torque can be employed for driving the slide valve and the units). Using the flame-transferring receiver channel 7, one can organize a pilot flame for decreasing the delay time of ignition and increasing the combustion rate. The seal between the slide valve and the casing is of labyrinth-type. The combustion chamber has been created and is being tested at the Public Corporation Scientific-Association "Saturn" (Rybinsk).

For realization of the potential and optimization of the structural parameters of the slide-valve combustion chamber we created a mathematical model of working processes. Since the working processes in a combustion chamber are analogous to those in fairly well-studied internal combustion engines, we employed the following corresponding accumulated experience [1] under the following assumptions and conditions:

(a) there are no leakages through the seals between the casing and the slide valve; according to the results of calculation investigations and experiments, the pressure loss for chamber volumes of more than 300 cm^3 can be less than 4%;

(b) flow is assumed to be one-dimensional and quasistationary;

(c) an ideal gas (the state is equilibrium) is taken as the working medium;

(d) the hypothesis for layer-by-layer displacement is employed in calculating the blowing;

(e) the equation of heat transfer between a hot gas and the combustion-chamber wall and between the exterior slide-valve wall and cooling air is determined by the formula $\mathrm{Nu} = 0.066\ \mathrm{Re}^{0.67}\ \mathrm{Pr}^{0.4}$ employed in calculations of the heat-transfer coefficient for a rotating sphere [2];

(f) the Reynolds number $\mathrm{Re} = D_{\mathrm{c.ch}} w / \nu$ is calculated from the maximum circular velocity; the heat-transfer coefficient is $\alpha_w = \mathrm{Nu}\ \lambda / D_{\mathrm{c.ch}}$;

(g) the Wiebe equation with coefficients of $c = -6.7$ and $m = 2.1$ (determined at N. É. Bauman Moscow State Technical University in experiments with similar chambers [3]) is employed for calculation of the combustion processes;

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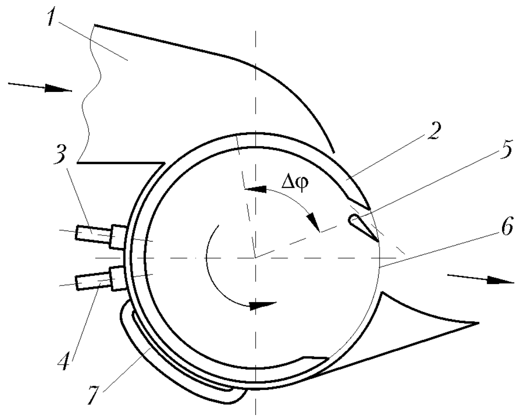


Fig. 1. Schematic diagram of a slide-valve combustion chamber (V = const).

(h) chambers with a volume of 300 to 5000 cm^3 are investigated;

(i) dissociation of the fuel in combustion is not taken into account;

(j) the excess-air coefficient is taken to be 1.3.

1. The process of combustion is described by the differential equation of the first law of thermodynamics for a closed thermodynamic system, which is solved simultaneously with the equation of state of an ideal gas (gas equation):

$$dU_{\mathrm{c.ch}} = XH_{\mathrm{u}}\, g_{\mathrm{c.f}} x - dQ_{\mathrm{w}}\,, \quad P_{\mathrm{c.ch}} V_{\mathrm{c.ch}} = G_{\mathrm{c.ch}} R_{\mathrm{c.ch}} T_{\mathrm{c.ch}}\,, \tag{1}$$

where $x = 1 - \exp\,(c(\tau/\tau_{\Sigma})^{m+1})$.

Heat transfer to the chamber wall is determined from the mathematical model of thermal state [4] and it is no higher than 4% (T_{cool} = 300 K, n = 12,000 rpm) of the heat released in combustion in the presence of a heatproof coating with a working temperature of the surface of 1250°C.

To determine the delay time of ignition we analyzed the results of testing the combustion chamber (V = const) at N. É. Bauman Moscow State Technical University and the empirical formulas for calculation of the delay time of ignition in piston internal combustion engines; the analysis has shown that:

(a) the temperature of the wall surface, which attains 1000°C or more, is determining in a combustion chamber (V = const), while the temperature of the air arriving at the chamber can be nearly atmospheric; the delay time of ignition can be 1.5 to 4.0 msec depending on the dimension of the chamber and it is approximately at the same level as the delay time in an internal combustion engine;

(b) in piston engines with relatively cold cylinder walls, the delay time of ignition is mainly determined by the temperature and pressure of air (the empirical formulas take no account of the wall temperature); accurate determination of the delay time is important in internal combustion engines, since we have combustion in a constantly changing volume.

In accordance with what has been said above and taking into account that combustion in the combustion chamber occurs in a constant volume, we can use a pilot flame; the delay time of ignition in the first approximation has been selected fixed and equal to 1.5 msec.

2. The processes of escape, blowing, and filling in the chamber are described by a system of equations which involves the differential equations of conservation of energy for residual gases, new charge, and their mixture, the differential equation of conservation of mass, and the equation of state of an ideal gas mixture:

$$\begin{gathered} dU_{\mathrm{n.ch}} = i^*_{\mathrm{in}} dG_{\mathrm{in}} - dQ_{\mathrm{w,n.ch}} - L_{\mathrm{frc}}\,, \quad dU_{\mathrm{g}} = i^*_{\mathrm{out}} dG_{\mathrm{out}} - dQ_{\mathrm{w,g}} - L_{\mathrm{frc}}\,, \\ dU_{\mathrm{c.ch}} = dU_{\mathrm{n.ch}} + dU_{\mathrm{g}}\,, \quad dG_{\mathrm{c.ch}} = dG_{\mathrm{in}} - dG_{\mathrm{out}}\,, \quad P_{\mathrm{c.ch}} V_{\mathrm{c.ch}} = G_{\mathrm{c.ch}} R_{\mathrm{c.ch}} T_{\mathrm{c.ch}}\,. \end{gathered} \tag{2}$$

The flow rate of the gases in the inlet and outlet ports are determined from the parameters of the gas in the minimum flow area. The equation of flow rate of the gas has the following form:

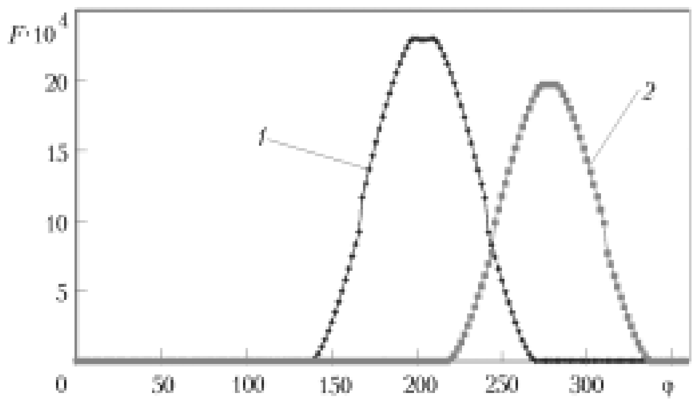


Fig. 2. Change in the flow areas as a function of the angle of rotation of the slide valve ($V_{c.ch}$ = 310 cm^3): 1) outlet port; 2) inlet port. F, cm^2.

for the incoming gases

$$dG_{in} = \mu \sqrt{\frac{2k}{(k+1)\,RT_{in}}}\; P_{in} F_{in} \lambda_{in} \varepsilon\,(\lambda_{in})\; d\tau\;;$$

for the outgoing gases

$$dG_{out} = \mu \sqrt{\frac{2k}{(k+1)\,RT_{c.ch}}}\; P_{c.ch} F_{out} \lambda_{out} \varepsilon\,(\lambda_{out})\; d\tau\;.$$

The dependence of the change in the areas of the flow ports on the angle of rotation of the slide valve for the combustion chamber with $V_{c.ch}$ = 310 cm^3 is presented in Fig. 2.

The system of equations in combustion or in gas exchange was calculated depending on the angle (step) of rotation of the slide valve. As the slide valve was rotated by the next step, we calculated the fluxes of heat and masses and re-calculated the mass, internal energy, concentrations, temperature, and pressure of the gas. The mathematical model created can also be applied to investigation of two-valve combustion chambers, since both types of chambers are thermodynamically and gasdynamically similar. Dissimilar designs will influence mainly the dependences of the change in the flow areas in the valves as a function of the angle of rotation of the camshaft.

Using the mathematical model created, we performed a computational experiment. The main parameters characterizing the perfection of a combustion chamber (V = const) are the maximum pressure and combustion temperature, P_{max} and T_{max} respectively, and the frequency of working pulsations, which is determined by the rotational velocity of the slide valve n. The quantities P_{max} and T_{max} together determine the quality not only of the process of combustion but also of escape, blowing, and filling. Therefore, in the computational experiment, we determined mainly P_{max} and T_{max} as functions of the dimensions (i.e., with proportional change in the combustion-chamber dimensions), the geometric parameters, the pressure difference on the chamber, and the frequency of working pulsations (rotational velocity of the slide valve). We determined the residual-gas coefficient and the coefficient of filling and blowing. In these dependences, we took the instant of the beginning of fuel injection as the reference point. The calculation results are presented in Figs. 3–7.

The analysis of the resultant dependences shows:

1. With increase in the dimensions of the chamber and for constant remaining parameters the characteristics of the working process are deteriorated: the filling coefficient decreases and the amount of the residual gases increases. Consequently, the maximum pressure and combustion temperature in the chamber decrease (see Fig. 3). But the fuel-air mixture is better prepared because of the increase in the amount of residual gases; this can increase the combustion rate. The "step" on the $T_{c.ch} - f(t)$ plot shows that the gases have escaped but blowing accompanied by the drop in the temperature of the gases in the chamber is to occur. As the difference of the pressures (ΔP) at the inlet and outlet of the chamber increases, the amount of residual gases decreases and the maximum pressure and temperature increase.

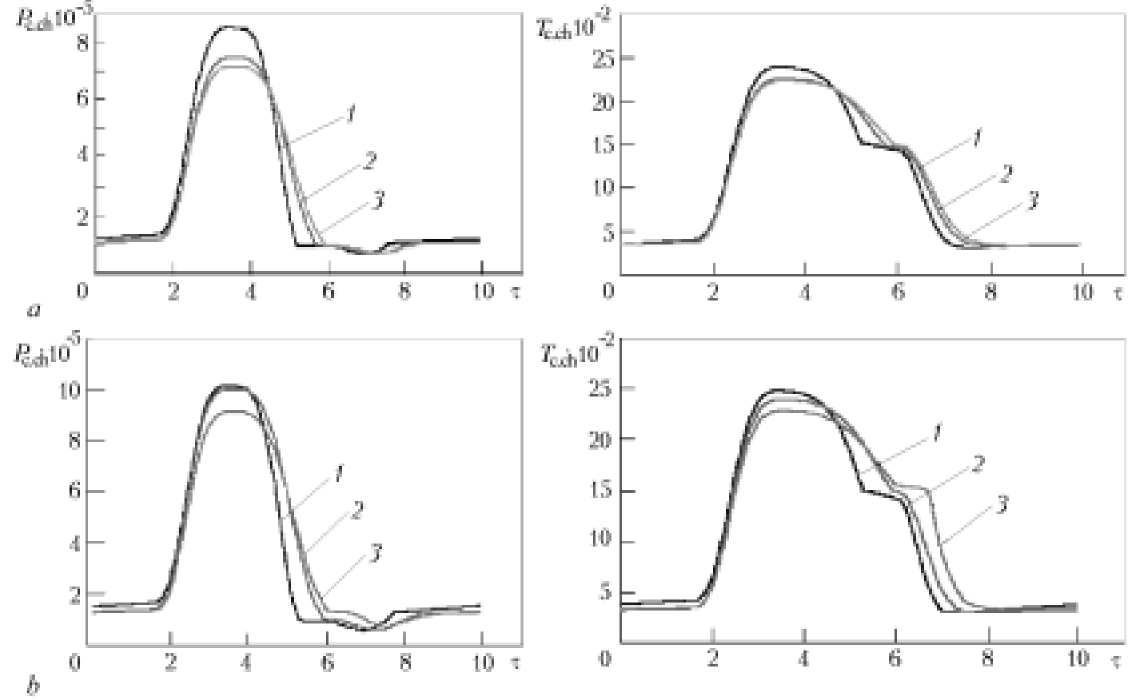
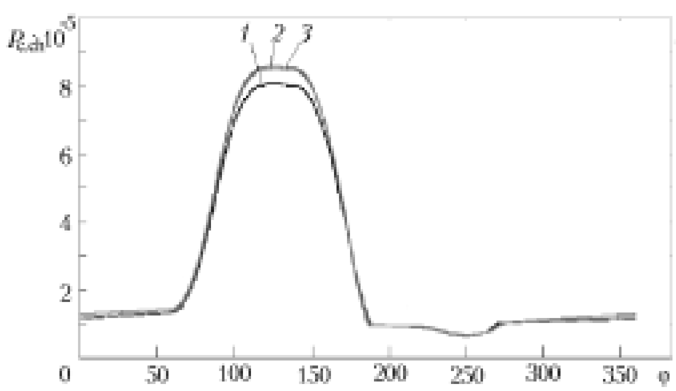


Fig. 3. Influence of the volume on the pressure and temperature in the combustion chamber (n = 6000 rpm, α = 1.3, P_{in} = 101,325 Pa, and T_{in} = 288): a) ΔP = 3039.75 Pa; 1) $V_{c.ch}$ = 310 cm^3, γ = 0.054; η_{fl} = 0.913; η_b = 1.0; 2) 2482 cm^3; 0.175; 0.776; 1.0; 3) 4850 cm^3; 0.203; 0.72; 1.0; b) ΔP = 20,000 Pa; 1) 310 cm^3; 0.0; 0.947; 1.135; 2) 2482 cm^3; 0.052; 0.9; 1.0; 3) 4850 cm^3; 0.157; 0.77; 1.0.



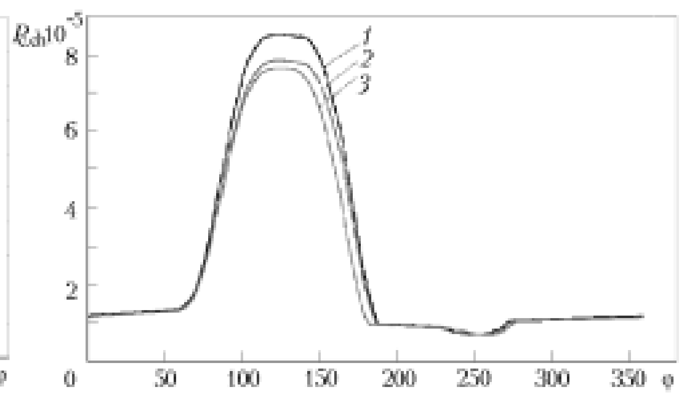


Fig. 4. Influence of the change in size of the flow sections on the pressure in the combustion chamber ($V_{c.ch}$ = 310 cm^3, n = 6000 rpm, α = 1.3, ΔP = 3039.75 Pa, P_{in} = 101,325 Pa, T_{in} = 288 K): 1) D = 0.050 m; γ = 108; η_{fl} = 0.847; η_b = 1.0; 2) 0.054 m; 0.054; 0.913; 1.0; 3) 0.057 m; 0.0057; 0.95; 1.0.

Fig. 5. Influence of the mutual position of the inlet and outlet ports on the pressure in the combustion chamber ($V_{c.ch}$ = 310 cm^3, n = 6000 rpm, α = 1.3, ΔP = 3039.75 Pa, P_{in} = 101,325 Pa, T_{in} = 288 K): 1) $\Delta\varphi$ = 74^o; γ = 0.054; η_{fl} = 0.913; η_b = 1.0; 2) 81^o; 0.131; 0.815; 1.0; 3) 88^o; 0.179; 0.75; 1.0.

2. As the port diameters are changed, we can note the following fact: an increase in them improves blowing and, conversely, a decrease in them makes it worse (see Fig. 4).

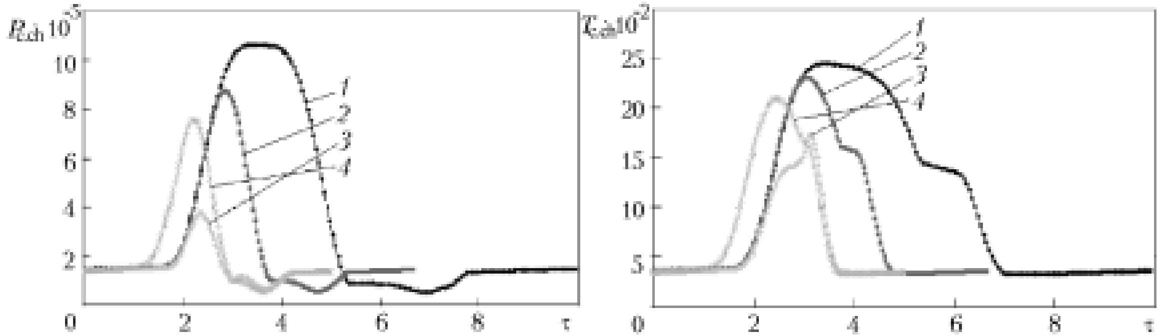


Fig. 6. Pressure and temperature in the combustion chamber vs. time for different rotational velocities of the slide valve ($V_{c.ch}$ = 310 cm^3, α = 1.3, ΔP = 30,000 Pa, P_{in} = 101,325 Pa, T_{in} = 288 K): 1) n = 6000 rpm; γ = 0.0; η_{fl} = 0.919; η_b = 1.22; 2) 9000; 0.0; 0.919; 1.057; 3) 12,000; 0.046; 0.881; 1; 4) 12,000 (τ_d = 1 msec); 0.032; 0.91; 1.0.

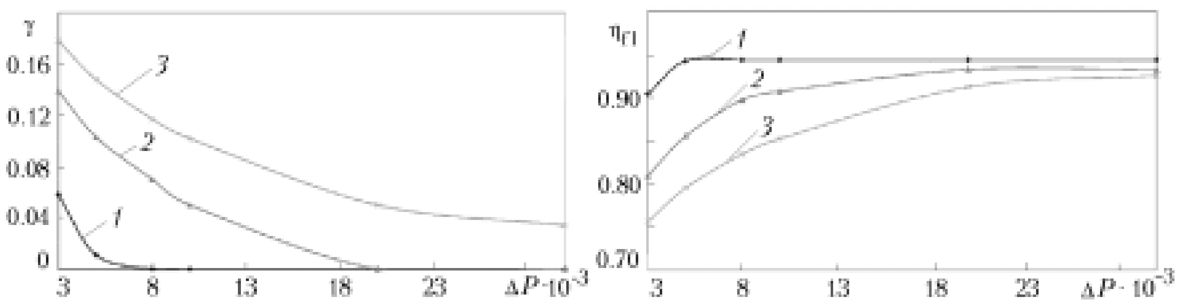


Fig. 7. Influence of the pressure difference and the rotational velocity on the amount of residual gases and the filling of the combustion chamber: 1) n = 6000; 2) 9000; 3) 12,000 rpm.

3. With increase in the parameter $\Delta\varphi$ characterizing the mutual position of the inlet and outlet ports, the parameters of the working process become worse because of the growth in the amount of residual gases. Therefore, it is expedient to design the chamber in such a manner that the value of $\Delta\varphi$ is minimum (see Fig. 5).

4. A strong influence on the processes in the combustion chamber is exerted by the rotational velocity of the slide valve. Figure 6 plots the pressure and temperature as functions of time for different rotational velocities. As the velocity increases, the pressure and temperature in the chamber drop because of the growth in the amount of residual gases. To increase the frequency of working pulsations we must increase the pressure difference on the chamber. Thus, at a pressure difference ΔP = 30,000 Pa, a frequency of working pulsations of 200 Hz is possible for a chamber with a volume of 310 cm^3, i.e., twice as high as in value-type chambers; however, here the time for combustion can be insufficient to obtain the calculated pressure values, and special measures must be taken to reduce the ignition delay and to increase the combustion rate. Thus, a reduction of 0.5 msec in the ignition delay sharply increases the combustion pressure (Fig. 6). This 0.5 msec can also be obtained upon the (leading) transfer of fuel injection to the process of filling (after blowing), when the slide-valve port is not yet covered with the casing of the chamber.

5. Figure 7 plots the changes in the coefficients of residual gases and filling as functions of the pressure difference on the chamber for different rotational velocities of the slide valve. It is clear that the increase in the difference decreases the amount of residual gases and improves the degree of filling of the chamber, whereas the amount of residual gases grows with increase in the velocity and the degree of filling becomes lower. Therefore, one must increase the pressure difference on the chamber with increase in the rotational velocity for maintaining an acceptable level of residual gases.

Thus, typically optimization of a combustion chamber (V = const) involves selection of its parameters with specified conditions (dimensions of the chamber, pressure difference on it, and others) for realization of the maximum pressures and frequencies of working pulsations. To do this combustion must stop at the instant of opening of the outlet port, while the gases must cease to escape at the instant of beginning of blowing.

We carried out field tests of the combustion chamber (V = const) of capacity 300 cm^3 simultaneously with the computational experiment. The maximum frequency of working pulsations was 100 Hz (it was limited by the capabilities of the available combustion equipment).

During the tests, we: a) have obtained startup at the pressure difference on the chamber ΔP = 2000 Pa; b) have ensured normal operation of the chamber throughout the range of working-pulsation frequencies; c) have obtained the calculated values of the combustion pressures for $\alpha \le 1.7$.

The combustion chamber is under operational development at present with the aim of obtaining design parameters in the limiting regimes (α = 1.3).

CONCLUSIONS

1. A mathematical model of the working processes of a slide-valve combustion chamber of V = const, which enables one to design the optimum structure of the chamber with prescribed initial data, has been developed. The mathematical model can also be applied to investigations of two-valve combustion chambers.

2. The results of the computational experiment have shown a high level of characteristics of the chamber: a) the frequency of working pulsations can attain 200 Hz ($V_{c.ch}$ = 310 cm^3, ΔP = 30,000 Pa); b) normal operation is ensured when ΔP = 3000 Pa, which is particularly important for startup.

3. Preliminary results of the tests of the experimental combustion chamber have confirmed in the main the reliability of the mathematical model; thus, startup at ΔP = 2000 Pa and the calculated values of pressure in the intermediate operating regimes have been obtained.

NOTATION

Nu, dimensionless Nusselt number; Re, Reynolds number calculated from the maximum circular rotational velocity of the slide valve; Pr, Prandtl number; $D_{c.ch}$, diameter of the combustion chamber, m; w, velocity of motion of the gases, m/sec; ν, kinematic viscosity, m^2/sec; α_w, heat-transfer coefficient, W/(m^2·K); λ, thermal-conductivity coefficient, W/(m·K); U, internal energy of the working medium, J; X, heat-utilization factor; H_u, calorific power of the fuel, J/kg; $g_{c.f}$, cyclic fuel supply, kg; x, function of fuel burnout; Q_w, heat transferred to the combustion-chamber walls, J; $P_{c.ch}$, pressure in the combustion chamber, Pa; $V_{c.ch}$, volume of the combustion chamber, m^3; $T_{c.ch}$, temperature in the combustion chamber, K; $G_{c.ch}$, mass of gases in the combustion chamber, kg; $R_{c.ch}$, gas constant of gases in the combustion chamber; c, burnout constant; m, index of the process of combustion; τ, running time, sec; τ_Σ, combustion time of the fuel-air mixture, sec; T_{cool}, coolant temperature, K; n, rotational velocity of the combustion-chamber slide valve, rpm; $U_{n.ch}$, internal energy of the new charge of air in the combustion chamber, J; i^*_{in}, enthalpy of the retarded flow in the minimum cross section of the inlet port, J; dG_{in}, amount of gases passed through the inlet port, kg; $Q_{w,n.ch}$, quantity of heat transferred from the new charge to the combustion-chamber wall, J; L_{frc}, work of forcing through of hot gases by the new charge, J; U_g, internal energy of hot gases in the combustion chamber, J; i^*_{out}, enthalpy of the retarded flow in the minimum cross section of the outlet port, J; dG_{out}, amount of gases passed through the outlet port, kg; $Q_{w,g}$, quantity of heat transferred from hot gases to the combustion-chamber wall, J; $U_{c.ch}$, internal energy of the gases in the combustion chamber, J; μ, flow-rate coefficient; k, adiabatic exponent; P_{in}, pressure at the inlet to the combustion chamber, Pa; T_{in}, temperature at the inlet to the combustion chamber, K; F_{in}, running area of the inlet port, m^2; λ_{in}, reduced velocity at the inlet to the chamber; F_{out}, running area of the outlet port, m^2; λ_{out}, reduced velocity at the outlet from the chamber; P_{max}, maximum combustion pressure, Pa; T_{max}, maximum combustion temperature, K; D, diameter of the port in the slide valve, m; α, excess-air coefficient; $\Delta P = P_{in} - P_{out}$, pressure difference, Pa; γ, residual-gas coefficient; η_{fl}, filling coefficient; η_b, blowing coefficient; $\Delta\varphi = \Delta\varphi_{in} - \varphi_{out}$, angle between the axes of the inlet and outlet ports, degree; φ, angle of rotation of the slide valve, degree; τ_d, delay time of ignition of the fuel-air mixture, sec. Subscripts: c.ch, combustion chamber; c.f, cyclic fuel supply; u, low; w, wall; cool, coolant; n.ch, new charge; g, hot gases; in, inlet; out, outlet; Σ, sum; d, delay; max, maximum; fl, filling; b, blowing; frc, forcing through.

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