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SOME PRINCIPLES OF COMBUSTION OF HOMOGENEOUS FUEL-AIR MIXTURES IN THE CYLINDER OF AN INTERNAL COMBUSTION ENGINE

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An algorithm is presented for the problem of flame propagation rate in combustion of a homogeneous fuel-air mixture in the cylinder of an internal combustion engine. It is assumed that the mixture is not "overturbulized" and that the flame front is spherical. The model used for the phenomenon is based on a turbulent transport mechanism. In the near-wall region the combustion mechanism follows a fine-scale mechanism, but in the core, a large-scale mechanism.

Experiments permitted determination of the character and numerical value of coefficients which consider the effect of turbulence on flame front propagation in the combustion chamber of a ZMZ-4021 engine.

The principles presented can be used as the basis of an algorithm for heat liberation rate in an internal combustion engine with external mixture formation.

Modeling of processes within an internal combustion engine with external mixture formation is hampered at present by lack of a clear physical picture of the heat liberation mechanism in fuel combustion. The situation is no better for engines with internal mixture formation. Heat liberation in the combustion process is a little studied area of internal combustion engine theory.

There have been numerous attempts to solve the problem of heat liberation during fuel burning in the engine cylinder, all of which we may divide into two groups. The first group includes studies of the physics of the combustion process, including chambers of variable volume [1-4]. These studies, while not pretending to create a method for calculating the combustion process in piston engines, do describe the basic principles of fuel combustion.

In the second group we have studies which attempt to formalize the complex combustion process [5, 6]. Usually this formalization is quite specialized and to a great extent empirical.

The principal differences between combustion in engines with internal and external mixture formation will not permit consideration of the unique features of the overall process within the scope of a single journal article. Herein we will only consider some aspects of the process of fuel-air mixture combustion in an engine with external mixture formation.

The majority of experimental studies of such engines has been carried out either with high speed cine photography of flame propagation, or with the aid of ionization sensors.

The studies that have been performed have to a great extent exposed the basic principles of fuel combustion in the variable volume engine chamber [4, 7, 8]. It has been

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established that two fundamentally different combustion mechanisms exist: fine-scale and coarse-scale.

A transition from one mechanism to the other is possible. While the large-scale combustion mechanism is related to the presence in the homogeneous mixture of significant turbulent pulsations, the fine-scale combustion mechanism is explicable by turbulent material transport.

At present no perfected theory of turbulence exists. This complicates construction of a physically justifiable combustion model for the internal combustion engine to a significant degree. However, as studies of boilers, jet engines, etc. show, use of simplified but physically justifiable models for description of both fine- and coarse-scale combustion is possible. The success of such an approach depends to a great degree on the choice of the model itself and the presence of empirical information on the processes.

As Pomerantsev noted [2], well chosen empirical coefficients in the volume (essentially fine-scale) model permits its use over a wide range to describe both large- and fine-scale combustion.

A deep concept underlies this approach – combustion of the homogeneous mixture is determined primarily by turbulent diffusion processes, which have one and the same physical nature, but different transport scales. We will note that the turbulent flame propagation rate (for the volume model) obeys the fundamental rule

$$\frac{U_t}{U_n} = \sqrt{1 + \frac{D_t}{D}}, \tag{1}$$

where U_t, U_n are the turbulent flame propagation rate and the normal rate; D_t, D are the turbulent and molecular diffusion coefficients.

Thus, in essence we may formulate the problem thus: we must find physically justifiable models for description of turbulent transport (D_t) for two different combustion mechanisms in the combustion chamber of the internal combustion engine with forced ignition: fine-scale and coarse-scale.

Why is it that these two different combustion mechanisms exist within the one chamber? Experiments have recorded this fact, but there is also a physical explanation.

Despite the significant turbulence of the charge when the cylinder is filled, in the region near the combustion chamber wall the Karman number (a measure of turbulence) is not large. After the voltage is applied to the spark plug, it is just this near-wall region in which the flame front propagates. Naturally combustion here obeys the fine-scale turbulence law. Estimates show that the extent of the near-wall region does not exceed 1/4H [8], H being the (current) height of the combustion chamber.

In this combustion zone the turbulent diffusion coefficient can be evaluated from the well known expression [9]

$$D_t^1 = \varkappa U_p \delta, \tag{2}$$

where U_p is the piston motion velocity, δ is the current thickness of the near-wall region, and κ is a coefficient which considers the influence of turbulence on transport effects. Its minimum value is $\kappa = 0(0.015\text{-}0.02)$.

Outside the near-wall region the effects of turbulent transport increase. This increase is caused not only directly by the velocity of the piston which initiates the motion, but by shear effects which develop within the flame front itself [7]. Therefore we can write the diffusion coefficient here in the form

$$D_t^2 = \varkappa U^* H, \tag{3}$$

where U* is the characteristic flame front velocity; H is the characteristic turbulence scale, equal to the current dimension (height) of the combustion chamber.

It will be convenient to rewrite Eq. (1) in a somewhat different form:

$$\frac{U_t}{U_n} = \sqrt{1 + \mathrm{Sc}\,\frac{D_t}{\nu}}, \tag{4}$$

where Sc is the Schmidt number (Sc = ν/D is a measure of the ratio of the kinematic viscosity of the oxidizer to its diffusion coefficient). For gasoline-air mixtures over a wide range of high temperatures Sc = 1.4-1.5. Determining the kinematic viscosity coefficient of the oxidizer presents no difficulties.

Knowing the turbulent flame propagation rate U_t (determining the normal velocity U_n requires no special explanation), we find the coefficient κ from the expression:

a) for the near-wall region

$$\varkappa_n = \frac{[(U_t/U_n)^2 - 1]\,\nu_{0t}}{\mathrm{Sc}\, U_p\, \delta p}\,; \tag{5}$$

b) for the combustion core region

$$\varkappa_c = \frac{[(U_t/U_n)^2 - 1]\,\nu_{0t}}{\mathrm{Sc}\, U^* H p}\,.$$

Here ν_{0t} is the kinematic viscosity coefficient of the oxidizer at a pressure p = 1 bar, p is the current pressure in the engine cylinder.

Thus, solution of the problem of determining characteristics of the turbulence κ requires foremost a knowledge of the turbulent flame propagation rate U_t.

We will demonstrate that the data required for solution of this problem can be obtained from the engine indicator diagram itself.

In fact, if the engine cylinder contains a fuel mass g_f and an air mass M_a in a thoroughly mixed state (homogeneous mixture) located in the combustion region, and if a relative fraction of the fuel x burns up, then the mass of the burned zone will be $(g_f + M_a)x$.

On the other hand, the mass of the burned zone is the product of its volume V_c times its density ρ_c. Then, obviously:

$$(g_f + M_a)\, x = V_c \rho_c. \tag{6}$$

In the first approximation (not considering the Machet effect) the temperature in the burned-up zone is approximately constant $T \approx T_{ad} \approx$ const. Then on the basis of Eq. (6) the value of the burned-up volume of fuel-air mixture in the combustion chamber will be

$$V_c = RT_{ad}(g_f + M_a)\,\frac{x}{p}\,, \tag{7}$$

where R is the gas constant of the combustion products. As is well known, the fraction of burned-out fuel x can be found from the engine indicator diagram p(φ) and a number of other easily determined quantities (g_f, M_a, etc.).

The heat liberation rate, i.e., dx/dφ, can then be found from the expression

$$\frac{dx}{d\varphi} = \frac{Mc_v T}{g_f Q_n}\left(\frac{d\ln p}{d\varphi} + k\,\frac{d\ln V}{d\varphi} + \frac{1}{Mc_v T}\,\frac{dQ_w}{d\varphi}\right), \tag{8}$$

where Mc_vT is the instantaneous loss of internal energy of the working body within the engine cylinder (T is the current temperature and c_p is the isochoric specific heat); $d\ln p/d\varphi = dp/p d\varphi$ is the change in relative pressure with angle of rotation of cranckshaft; $d\ln V/d\varphi = dV/V d\varphi$ is the corresponding change in volume; k is the current value of the adiabatic index; $dQ_w/d\varphi$ is the rate of heat removal from the working body to the cylinder wall.

Having solved Eq. (8) by the Runge-Kutta method, we find the current value of the fraction of burned-out fuel

$$x_i = x_{i-1} + \left(\frac{dx}{d\varphi}\right)\Delta\varphi. \tag{8'}$$

Using Eqs. (8) and (8'), we can uniquely define the volume of burned-out fuel-air mixture with Eq. (7).

TABLE 1. Parameters Characterizing Operation of Engine Studied

Parameter	Notation	Engine speed n, min^{-1}		
		3600	2800	1800
Power (effective), kW	N_e	51,74	40,24	22,18
Excess air coefficient	α	0,914	0,98	1,16
Cyclical flow rate, kg/cycle				
fuel	g_f	$3,84 \cdot 10^{-5}$	$3,87 \cdot 10^{-5}$	$2,92 \cdot 10^{-5}$
air	M_a	$5,48 \cdot 10^{-5}$	$5,89 \cdot 10^{-5}$	$5,52 \cdot 10^{-5}$

To a well known approximation flame propagation in an internal combustion engine cylinder with external mixture formation (carbureted or gas-fueled) can be considered spherical [1, 5]. Then if the spark plug position is known (geometric center of the sphere), it becomes possible to relate the volume of burned out fuel to the current radius of the combustion sphere, i.e., $V_{ci} = V_c(R_i)$. This relationship can be represented in the form of a polynomial

$$V_{ci} = aR_i + bR_i^2 + cR_i^3 + dR_i^4 . \tag{9}$$

Thus, we have all the necessary data for computation of the coefficient κ. Employing the indicator diagram for the given engine operating mode, we use Eq. (8) and define the volume of burned-out fuel-air mixture (7), then use Eq. (9) to find the current radius of the combustion sphere R_i.

If in the given operating regime the crankshaft rotation rate is n, the time required for its rotation through an angle $\Delta\varphi$, which corresponds to a change in radius of the combustion sphere ΔR_i, comprises $\Delta t = \Delta\varphi/6n$. The turbulent flame velocity is then

$$U_{\text{T}i} = \frac{\Delta R_i}{\Delta t} .$$

We will now analyze some aspects of the results obtained.

The method developed above was tested on a ZMZ-4021 engine from the Zvolzhsk motor works. The engine was a four-cylinder, four-stroke engine, with specifications D = 92 mm, S = 92 mm, $\lambda = R/L = 0.2738$.

Tests were performed at full throttle; the engine indicator diagram was recorded on an oscilloscope using piezoquartz sensors, while fuel and air expenditure, rotation speed, etc. were also measured. Table 1 shows the parameters of the regimes tested.

Figure 1 shows the indicator diagram of the engine corresponding to the third regime (N_e = 51.74 kW, n = 3600 min^{-1}).

The combustion chamber was approximated by a polynomial

$$V_{ci} = 94{,}970R_i + 16{,}048R_i^2 + 0{,}592R_i^3 - 7{,}999R_i^4 \ (\text{cm}^3)$$

(valid for $R_i > 10$ mm). At small values of combustion sphere radius, i.e., for combustion in the near-wall region, where $\delta < R_i = 10$ mm, the relationship

$$V_{ci} = \frac{2}{3} R_i^3$$

is valid.

The extent of the near-wall region is a variable quantity, dependent on the engine operating regime (load, speed). Based on [8], we can say that the thickness of the near-wall region $\delta \sim D/Re^{0.25}$ (where D is diameter cylinder and Re is Reynolds number defined from the mean piston velocity and characteristic dimension equal to the cylinder diameter).

Transforming the relationship thus obtained and considering that for the majority of engines with external mixture formation $S \approx D$ (where S is the piston stroke), we obtain

$$\delta = D^{0,5} \nu_{0i}^{0,25} p^{-0,25} n^{-0,25} , \tag{10}$$

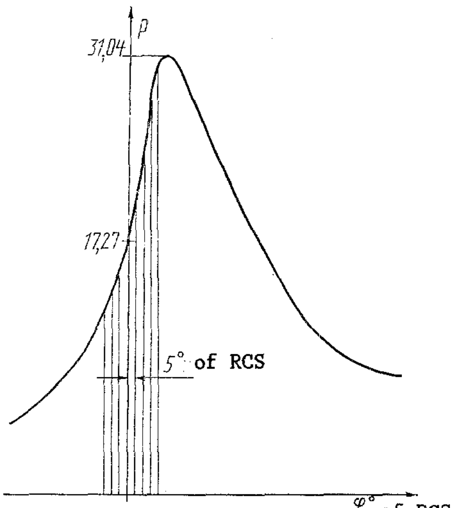

Fig. 1

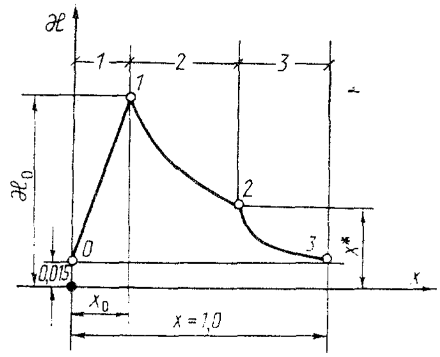

Fig. 2

Fig. 1. Indicator diagram of carbureted engine. p, bar.

Fig. 2. Change in coefficient κ which considers effect of turbulence on flame propagation rate in engine combustion chamber: 1) near-wall region near spark plug; 2) core region (coarse-scale turbulence); 3) near-wall region (area of fine-scale burn-up).

where ν_{0t} is the kinematic viscosity coefficient of the working body at the current temperature in the cylinder and atmospheric pressure of the body; n is the crankshaft rotation speed.

Thus, the thickness of the near-wall region, where combustion begins after mixture ignition, is a variable quantity, which changes during development of the process. The thickness of the near-wall region decreases with increase in load and crankshaft speed.

The flame front exiting the near-wall region has a velocity which obeys large-scale turbulence laws [1].

Processing of the experimental material with Eq. (5) showed that for fuel combustion in the core there exists a clearly expressed dependence between the coefficient considering the effect of turbulence κ and the fraction of burned-out fuel x:

$$\varkappa = \frac{\text{const}}{x}.$$

Consequently, the character of the change in the coefficient κ during fuel burn-up in the cylinder can be represented in the following manner (Fig. 2):

a) in the near-wall region, where the fraction of burned-up fuel changes from x = 0 to $x = x_0$, the function κ changes from $\kappa \approx 0.015$ to $\kappa_0 = \text{const}/x_0$;

b) in the main heat liberation region (burn-up in the core)

$$\varkappa = \text{const}/x;$$

c) when the flame front reaches the cylinder wall opposite the spark plug, the change in κ is opposite to the initial one, i.e., κ changes from κ^* to $\kappa = 0.015$.

The above considerations can serve as a basis for calculating the heat liberation rate in burn-up of a fuel-air mixture in the cylinder of an engine with external mixture formation.

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