THE COMBUSTION PROCESS AND HEAT RELEASE IN THE GAS ENGINE

Mikhail N. YEROKHIN[1], Otari N. DIDMANIDZE[1], Nikolay ALDOSHIN[1], Ramil T. KHAKIMOV[2]

[1] Russian State Agrarian University - Moscow Timiryazev Agricultural Academy, Russia
[2] St. Petersburg State Agrarian University, Russia

Abstract
The analysis of the material presented in the article showed that the exact mathematical description of the whole gas-dynamic process, the result of which is the effective combustion of the gas-air mixture inside the cylinder, is a difficult task. For this purpose is not appropriate often used option - the creation of a purely empirical regression model, becaus for it would require a very large number of experiments (even in the case of using the method of part factorial experiment); the experience results will be adequate only in the range of changes of parameters in the framework of the conducted experience. Based on the knowledge of modern techniques, it seems appropriate to use the data obtained experimentally as a basis for further recalculation of the parameters taking into account new conditions and physical representations about the working process of gas engines.

Key words: *the process of burning, heat generation, gas engine, flame front, the turbulent velocity.*

INTRODUCTION

The burnout characteristics includes: the duration of the φ1, indicators of the nature of burnout and the proportion of the fuel, burnable in the individual phases of burnout. In the case of diesel engines, there are usually two phases of burnout; in gas and gasoline engines with forced ignition - single-phase heat release (*Vibe i. 1962*),.The purpose of this article is determining the speed of propagation of the frontsevere gas mixture in the combustion chamber, severethrough the angle of rotation of the crankshaft.f

Studies by various authors have confirmed that the angle of the crankshaft rotation (CR) from the beginning of the combustion process to achieve the first maximum rate of heat release in the gas engine under study is practically independent of the operating mode and load characteristics. Accordingly the maximum rate of heat release rate, as indicated above, in the proportion to the heat share released in the first phase of combustion; let us denote it as X_1. According the experimental experience of existing research results and our own tests, the angle φ_1 can be taken equal to 3 – 4° CR, since in this case there is an initial process of laminar motion of the flame front, (*Gainullin F. 1986, Birger I. 1978).* The second phase of heat release is determined according to the presented technique.

MATERIALS AND METHODS

The ratio of burning gas amount to the supplied gas-air mixture gives the additive to the X_1.

$$\tilde{O}_1 = \frac{G_{\tilde{A}\tau i}}{\alpha \cdot G_{\tilde{A}}}, \tag{1}$$

where $G_{\Gamma\tau i}$ – the amount of gas mixture consisting of ethane (propane, butane, inert gases, etc.);
α – air excess factor;
G_{Γ} – amount of natural gas.

Thus, for a large class of chain reactions it is possible to write a differential equation. It will relate the total rate of change in the relative density of the effective centers to its density at a given time:

$$\frac{d\rho}{dt} = a\rho - b\rho^2, \tag{2}$$

where a – proportionality coefficient in 1/sec., or constant of circuit development;
b – the proportionality coefficient, which is an abstract number, or the constant of the circuit break.
For non-chain reactions $a = 0$, $b = 0$, so from the equation (3) will obtain that $\frac{d\rho}{dt} = 0$. Therefore ρ = const and the equation for the proportion of the reacted substance turns into the equation of the monomolecular reaction:

$$x = 1 - e^{-n\rho_0^t} = 1 - e^{-kt}, \tag{3}$$

Ratio $a/b=\rho_s$ is equal to the quasi-stationary density of effective centers.

According to equation (3) of the rate constant of monomolecular reactions

$$k = n\rho_0, \tag{4}$$

For very many chemical reactions this constant depends on the temperature according to Arrhenius law, so can write:

$$n\rho_0 = Ae^{-\frac{E}{RT_0}}, \text{ or } \rho_0 = \rho_{np}e^{-\frac{E}{RT_0}} \tag{5}$$

where E – activation energy of the initiating reaction;

T_0 – absolute temperature at which the heat initiation of the reaction takes place;

R – gas constant;

A constant of co-hit in accordance with the provisions of the gases kinetic theory;

$\rho_{пр}$ – the marginal density of the effective centers under a conditionT = ∞.

According to equation (5), the initial density of the effective centers of hydrocarbon gas fuels in the combustion process can be calculated for a given temperature.

The process of burning the working mixture consisting of natural gas and air in the main phase is expressed by an angle φ_2, it indicates the flow of the process from the beginning of combustion to the maximum value of the heat release rate in the combustion chamber of the engine. In the formulation of the problem in the process of modeling the heat release process, we believe that this angle shows, ceteris paribus, the determination of the flame front velocityand the distance traveled by the frontal boundary of the flame from the ignition point to the point of the released heat near the walls of the combustion chamber. It follows that the change in the velocity of the turbulent flame propagation in the combustion chamber is directly related to the component composition of the gas-air mixture (*Khakimov R. et al., 2018).*

As a result, the angle φ_2 when the maximum heat release rate in the second combustion phase is reached linearly depends on the turbulent velocity of the flame front propagation in the combustion chamber, according to the data (*Petrichenko R., Rusinova R., 1983*). Based on the above studies on the increasing to the maximum speed of the flame front propagation, we try to determine and construct a type of dependence for the angle of maximum heat release in the main phase of the burn-up process. (*Alumbauch R.L., Keeton I.R, 1984*). Therefore, the expression will have the form that was described (*Khakimov R.T., Didmanidze O.T., 2017):*

$$\bar{u}_T \approx (P)^{a_1}(T)^{a_2}(U_H)^{a_3}(\alpha)^{a_4}(n)^{a_5}(V/n)^{a_6}(q_Z)^{a_{11}} \ , \tag{6}$$

where T - the average temperature of the flame front during its propagation in the combustion chamber; α - air excess factor; U_H - normal speed during the combustion process; P - average combustion pressure during its rise in the combustion chamber; n - the rotational speed of a gas engine a crankshaft; V - volume of gas-air mixture entering the cylinder; q_z – total specific heat of combustion used according Vibe i. (1962), $q_z = {Q_z}/{G_{mc}G}$; G_{mc} – the amount of fuel supplied to the engine cylinder in one cycle; G – weight of working fluid per 1 kg of fuel.

Given the fact that $u_H(P,T,\alpha,n)$, add new value q_z so the dependence will take the following form:

$$\bar{u}_T \approx (P)^{a_1}(T)^{a_2}\left((P)^{a_7}(T)^{a_8}(\alpha)^{a_9}(n)^{a_{10}}\right)^{a_4}(n)^{a_5}(V/n)^{a_6}(q_Z)^{a_{11}} \ , \tag{7}$$

Having studied the above, it is necessary to determine the remaining power values in the form of the following coefficients a_1 a_2, a_7, a_8. In this case, we use the ideas about the effect of P, T on the turbulent velocity of the flame in the combustion chamber. Materials on the effect of T on the velocity u_T of a turbulent flame, degree values of the coefficients sum a_2+(*Kagan L., et al., 2010*). The theory about the relationship between the turbulent flame velocity and P can be represented by the values of the following coefficients equal to $a_1 = 0,302$ и $a_7 = -0,305$. (*Austin J.M., Pintgen F., Shepherd J.E – 2005*). To simplify the determination of coefficients a_{11}, a_9, a_6 и a_5, set the value of the coefficient equal to $a_3 = 1$. Further, in order to simplify the expression, let's combine some power values with the same terms, then the calculated dependence will look like this:

$$\bar{u}_T \approx (P)^0 (T)^{1,6} (\alpha)^{a_9} (n)^{a_5} (V/n)^{a_6} (q_z)^{a_7} = (T)^{1,6} (\alpha)^{a_9} (n)^{a_5} (V/n)^{a_6} (q_z)^{a_{11}} , \tag{8}$$

The calculated dependence of the angle φ2 characterizing the second phase of the turbulent burning

velocity of flame propagation inside the engine cylinder can be represented by the following formula:

$$\varphi_2 = \frac{l \cdot in}{\bar{u}_T} , \tag{9}$$

where l - distance from the ignition point to the point of reaching the cylinder walls; n - the frequency of the crankshaft rotation; i – the number of the engine cylinders.

It follows that the dependence on determining the angle φ_2 of rotation of the crankshaft at the maximum speed of the turbulent flame, taking into account the dependence for u_m will be as follows:

$$\varphi_2 \approx l \cdot n \cdot (T)^{-1,65} (\alpha)^{-a_8} (n)^{a_5 - 1} (V/n)^{-a_6} (q_z)^{a_{11}} , \tag{10}$$

Coefficient a_8 and a_5 values aren't defined we will look for the values of the degree at the excess air ratio and speed, denote them t>1 и b_2.

$$\varphi_2 \approx (T)^{-1,65} (\alpha)^{b_1} (n)^{b_2} (V/n)^{b_3} (q_z)^{a_{11}} , \tag{11}$$

RESULTS AND DISCUSSION

To determine the degree coefficients, we use the experimental data (*Belyaev A.I., Afanasyev 2016*). As a result, the set values of the coefficients confirm the findings *(Kagan L., et al., 2010*), about the absence of the relationship between the crankshaft speed and the achievement angle of the maximum value of the turbulent flame velocity in the combustion chamber, expressed by the second combustion phase φ_2; (*Ciccarelli G, 2011*). On the basis of the dependence. (*Oran E.S., Gamezo V.N 2005*) degree value for the coefficient b_3 set equal to $b_3=1$. Next, to determine the degree value of the coefficient b_1, we turn to the experimental data presented in figures 1 and 2. They clearly shows the parabolic nature of the dependence of the normal (u_T) combustion of the gas-air mixture on the coefficient (α) with an extremum corresponding to the composition of the mixture in the stoichiometric ratio.

For gas-air mixtures of methane and oxygen, the stoichiometric composition is equal to the ratio of 17.2%, therefore, the flame propagation velocity of these mixtures will correspond to the maximum limit of 45-55%. This ratio is explained by the weighting of molecules, while the concentration boundaries of ignition are significantly narrowed, hence the curves of the flame propagation velocity in the percentage ratio are also reduced.

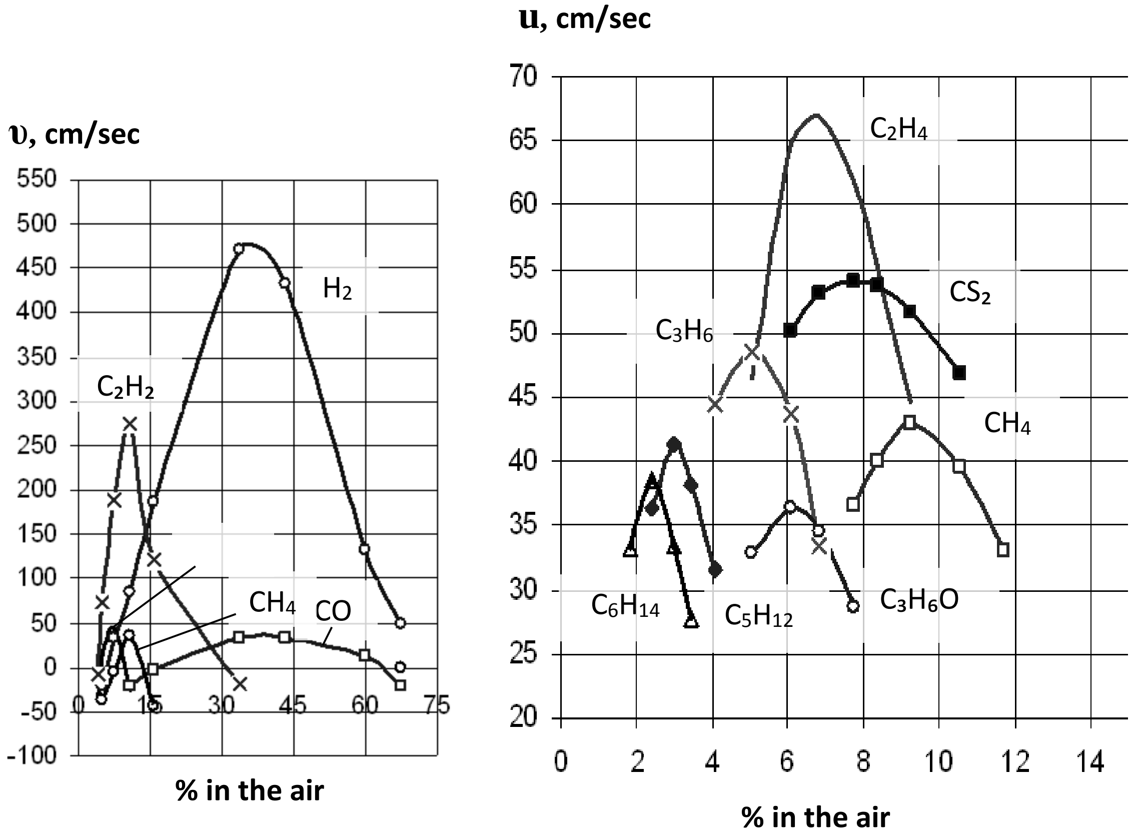

Fig.1 The dependence of the flame spreading speed and the normal speed of the mixture

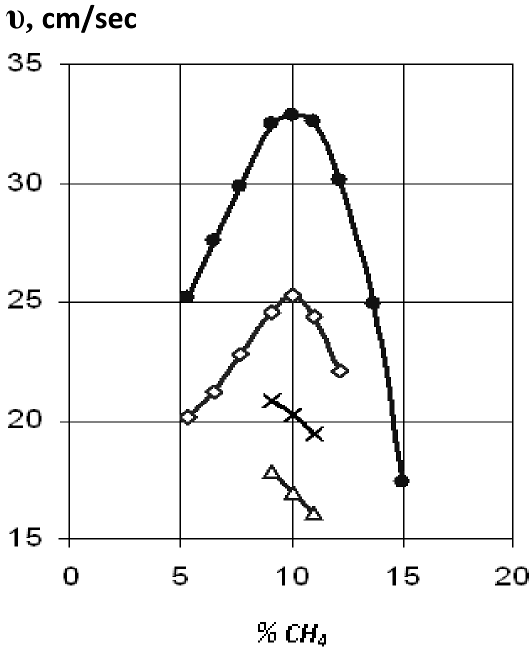

Fig. 2 The normal velocity of the methane-air mixture flame from the pressure according *(Petrichenko R., Rusinov R., 1983), Novichlov M. (2004).*

1 – velocity of propagation of methane-air mixture at pressure Pz = 5,0…5,4 MPa;
2 – velocity of propagation of methane-air mixture at pressure Pz = 4,5…4,8 MPa;
3 – velocity of propagation of methane-air mixture at pressure Pz = 4,0…4,3 MPa;
4 – velocity of propagation of methane-air mixture at pressure Pz = 3,4…3,8 MPa.

CONCLUSIONS

1. On the basis of theoretical studies of the main parameters of in-cylinder processes influence on the rate of heat release and propagation of the flame front in the second phase of combustion and taking into account the applicability of hydrocarbon fuels of methane series, a physically justified model of the processes was formulated.
2. A mathematical model and an algorithm for calculating the operating cycle of the gas engine, allowing to take into account the peculiarities of the physical processes in the cylinder due to the combustion of the gas-air mixture was developed.

REFERENCES

1. Vibe I. I. New about the working cycle of the engine (combustion rate and engine duty cycle). – M. : Mashgiz, 1962. – 271 p.
2. Gainullin F.G. Natural gas as a motor fuel for transport. - M .: Nedra, 1986. - 327 p.
3. Birger I.A. Technical diagnostics. - M.: Mashinostroenie, 1978. - 240 p.
4. Khakimov R. T., Didmanidze O. N., Afanasiev A. S. Studies of heat dissipation of gas engines // Notes of the Mining Institute. – 2018. – Vol. 229. – P. 50-55.
5. Petrichenko P. M., Rusinov R. V. heat Exchange in the fuel torch // engine building. – 1983. No. I. – P. 9-12.
6. Alumbauch R.L., Keeton I.R., Нитт E.F. Experimental polyurethane roof systems // J. Cellular Plastics. – 1984. – № 4. – P. 257-273.
7. Khakimov R.T., Didmanidze O.T Improving the supply system gas engine to improve energy efficiency // Transportation Research Procedia. – 2017. – P. 183-191.
8. Kagan L., Valiev D., Liberman M., Gamezo V., Oran E., Sivashinsky G. Effects of hydraulic resistance and heat losses on deflagration-to-detonation transition // In Deflagrative and detonative combustion (eds G.D. Roy & S.M. Frolov). – 2010. – pp. 157 – 168.
9. Austin J.M., Pintgen F., Shepherd J.E. Reaction zones in highly unstable detonations // Proc.Combust.Inst. – 2005. – vol. 30. – pp. 1849-1857.
10. Belyaev A.I., Afanasyev, A.S., Efficiency of vehicle operation / International journal of economics and financial issues, № 2, т 6, 2016. P. 24 - 30.
11. Ciccarelli G. Transition in the propagation mechanism during flame acceleration in porous media / G. Ciccarelli, C. Johansen, M. Parravani // Proc. Combust. Inst. 33. – 2011. – pp. 2273 –2278.
12. Oran E.S., Gamezo V.N. Flame acceleration and detonation transition in narrow tubes // Paper presented at 20 th Int. Colloq / On the Dynamics of Explosions and Reactive Systems (ICDERS), McGill University, Montreal, 31 July–5 August – 2005.
13. Yerokhin M. N., Didmanidze O. N., Mityagin G. E. Problems and prospects of import substitution in road transport // In the collection: Information technologies and innovations in transport materials of the 2nd International scientific and practical conference. – Orel state University. I. S. Turgeneva, 2016. – P. 88-96.

Corresponding author:
Prof. Otari Didmanidze, Head of Department of Automobile transport, Russian State Agrarian University-Moscow agricultural academy named K.A. Timiryazev, Russia,Timiryazevskaya str.49,Moscow, Russia, 127550, cxm.msau@yandex.ru