

# The Mathematical Model of Gas Dynamic Process in the Regulated Heat Transfer at the Inlet

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**Abstract.** The paper presents a mathematical model of heat release in a cylinder of a gas engine based on the previously described differential equations proposed by Professor R.M. Petrichenko. Having studied the method for calculating the heat release through pressure buildup during the gas exchange, a number of shortcomings were identified that made it possible to formulate an improved mathematical model of the gas dynamic process in determining the turbulent propagation velocity of the flame front in the combustion chamber of the gas engine, expressing it through the angle of rotation of the crankshaft, which made it possible to optimize the methane- at the inlet.

## 1 Introduction

The analysis studies have shown that the most common method of determining the heat dissipation in this case is the use of a mathematical model described by R. M. Petrichenko [1,21]. In it are examines the influence of operating parameters one process of gas exchange a gas engine which takes place inside the cylinder. In this case, the proposed model considers the rate of change of pressure in the cylinder of a diesel engine and does not take into account of some features of the engine as in the structural and thermodynamic relations. So it doesn't consider the process of combustion of fuel and air with respect to the increase of the turbulent speed of the flame front in the combustion chamber, expressed by the crankshaft angle.

## 2 Material and methods

For the initial consideration of the issue will use the first law of thermodynamics of variable mass [3-6].

$$i_0 dm = d(mu) + p dV + dQ_w, \quad (1)$$

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where ( $i_0$ ,  $i$  – enthalpy of the working body;  $dm$  – elementary mass of the working body;  $m$  – is the mass of the working body in the cylinder; and  $u$  – specific internal energy of the working body;  $p$  – pressure of the working body in the cylinder;  $dV$  – volume change of the cylinder;  $Q_w$  – heat received (given) the working body in the heat exchange [13,20].

Equation (1) transform given the fact that  $i_0 = c_p T_0$ , and  $u = c_v T$  ( $c_p$  are  $c_v$  respectively the heat capacity at constant pressure and volume;  $T_0$ ,  $T$  then, respectively, the temperature of the working body entering the cylinder and in the cylinder). Is converted the equation, to do this, take the logarithmic derivative of the characteristic equation  $pV = MRT$ . Then we define the formula for the temperature and solve the resulting expression relative to  $dp/p$ ,

$$\frac{dp}{p} = \frac{c_{p0} T_0}{c_v T} \frac{dM}{M} - k \frac{dV}{V} - \frac{dQ_w}{Mc_v T}, \quad (2)$$

Imagine a differential equation for the exhaust stroke (3) and the flow of the combustible mixture to the combustion chamber (4)

$$\frac{dp}{d\phi} = kp \left( \frac{d \ln M}{d\phi} - \frac{d \ln V}{d\phi} \right) - \frac{2}{3} (k-1) \frac{D/2 + H}{DH} \frac{\bar{\alpha}(T - T_w)}{n}, \quad (3)$$

$$\frac{dp}{d\phi} = p \left( \frac{c_{p0} T_0}{c_v T} \frac{d \ln M}{d\phi} - k \frac{d \ln M}{d\phi} \right) - \frac{2}{3} (k-1) \frac{D/2 + H}{DH} \frac{\bar{\alpha}(T - T_w)}{n}, \quad (4)$$

Further according to the method Petrychenko R. M. [1,19] present a differential equation of rate of pressure rise of combustion – expansion (5) and the compression process (6).

$$\frac{dp}{d\phi} = p \left( \frac{g_c Q_H}{Mc_v T} \frac{dx}{d\phi} - k \frac{d \ln V}{d\phi} - \frac{1}{Mc_v T} \frac{dQ_w}{d\phi} \right), \quad (5)$$

$$\frac{dp}{d\phi} = -p \left( k \frac{d \ln V}{d\phi} + \frac{1}{Mc_v T} \frac{dQ_w}{d\phi} \right), \quad (6)$$

In the process of gas exchange parameters as of a methane mixture are changing. In the synthesis process of the working cycle allows to define the work, the medium pressure and temperature and the effective efficiency of the gas engine. For mathematical description of gas dynamic process in the cylinder of a gas engine the methane-air mixture in the cylinder is ready for ignition from the spark plugs. So there is the problem lies in the determination of the average turbulent velocity of the flame front in the combustion chamber [7,8].

### 3 Theory

There is consider the dynamics of the dissipation in the gas engine operating on the Otto cycle as before in two steps: ignition of methane-air mixture and the flame front propagation of the mixture by volume of the combustion chamber. Also we will consider the second part of the dynamics of heat dissipation as it determines the efficiency of the combustion process in the regulation of methane-air mixture at the inlet.

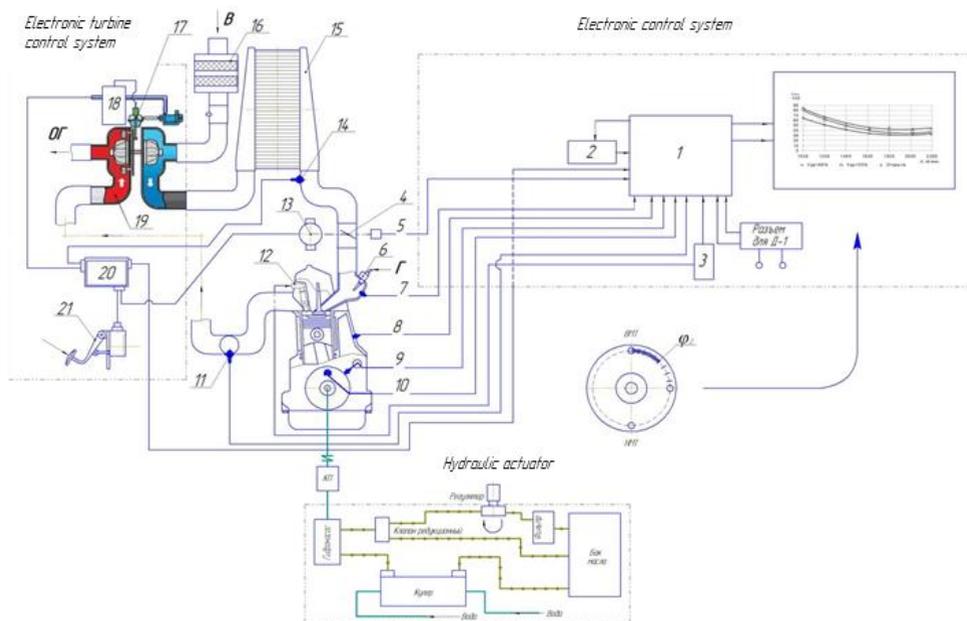
We use the B.P. Pugachev formula for the mathematical description of the combustion process in the gas engine [19]:

$$\frac{dx}{d\phi} = \frac{X_1}{\phi_1} \left(\frac{\phi}{\phi_1}\right) \exp\left[-0.5\left(\frac{\phi}{\phi_1}\right)^2\right] + \frac{X_2(k_2-1)}{\phi_2} \left(\frac{\phi}{\phi_2}\right)^{k_2-1} \cdot \exp\left(-\frac{k_2-1}{k_2}\left[\frac{\phi}{\phi_2}\right]^2\right), \quad (7)$$

Where  $X_1$  и  $X_2$  - this fraction of heat released in two phases beginning of ignition, efficient combustion; – the current value of the crankshaft angle; the angles of the combustion process from the start until maximize the speed of propagation of the flame front in the combustion chamber [9,10].

The degree of conformity of calculated and experimental characteristics of heat generation depends on a correct choice of these parameters. In this case, it is desirable to use different approaches to determine the thermal performance in the test engine. This process is carried out with limited consideration of physical ideas concerning the nature of flame propagation in the combustion chamber. Using these approaches, you can establish a quality relationship of indicators with the actual conditions of the combustion process [15,16].

For general idea consider the diagram electronic control gas engine with spark ignition of methane-air mixture (figure 1).



1 - electronic management block for gas engine (Corvette M10.3); 2 – Potentiometer; 3 - ignition Coil; 4 - Throttle; 5 - the Sensor of angular position of the throttle; 6 - Gas dispenser; 7 - temperature Sensor at the inlet; 8 - coolant temperature Sensor OJ; 9 - angular rate Sensor and position KV; 10 - Optoelectronic sensor for the regulation of the inlet; 11 - Oxygen sensor (lambda sensor); 12 - the spark plug; 13 - the Electric throttle valve; 14 - the boost pressure Sensor; 15 - Cooler; 16 - Air filter; 17 - Sensor back pressure; 18 - Servo adjustment of the boost pressure; 19 - Turbocharger; 20 - the CDI control Unit (N3/9); 21 - Electronic pedal [14,22].

**Fig. 1.** Control scheme for the stand when tested GE supercharged [4]

The regulation of methane-air mixture takes place without the participation of the  $\lambda$ -probe as the optimization of the selection of the ratio of methane and air is carried out by determining the speed of turbulent propagation of the flame front in expressing it through the angle of rotation of the crankshaft. Definition angle of rotation crankshaft determine due to a special device consisting of optical sensors, which in turn determine the position of VMT and NMT of the crankshaft. Then it defines the increase of speed of the piston through a

special calibrated holes in increments of  $10^\circ \frac{1}{4}$  part of the disk with intermediate cross-cutting notches with the division of  $2^\circ$ . Is created the best combustion of methane-air mixture the combustion with optimum advance angle, and this affects the speed crankshaft [11,12].

Accordingly, the crankshaft angle from the start of ignition of the mixture by an electric discharge of the spark plug prior to the formation of maximum speed of heat dissipation practically does not depend on the operating mode of the internal combustion engine. So the first phase ( $\varphi_1$ ) may be accepted in the amount of  $3-4^\circ$  of crankshaft rotation.

The calculations indicate that the angle of approach of the maximum rate of heat release in the phase ( $\varphi_2$ ) depends on the turbulent velocity of the flame front. We will present the dependence of the rotation angle of the crankshaft in the second phase of combustion and the maximum rate of propagation of the flame front in the combustion chamber based on the characteristics of the combustion process of methane and air in a gas engine [2]:

$$\bar{u}_T \approx (P)^{a_1} (T)^{a_2} (U_H)^{a_3} (\alpha)^{a_4} (n)^{a_5} (V/n)^{a_6}, \quad (8)$$

where  $P$  - is the average pressure during the propagation of combustion in the gas mixture,  $T$  - is the average temperature during the propagation of combustion in the gas mixture, the  $U_H$  - the normal rate of combustion,  $\alpha$  - coefficient of excess air in the gas mixture,  $n$  - is the frequency of rotation of the crankshaft of the engine,  $V$  - is the volume entering the cylinder gas mixture.

Given the findings of reference [17,18], we'll take the following - all the value of the degree's coefficients ( $a_4, a_0$ ) depending on the physical phenomena is either equal to zero or are constant values  $a_1=0,3$   $a_2=1,6$ ,  $a_7=-0,3$ ,  $a_8=1,6$ .

To facilitate will combine the degree with the same members, then the dependence becomes:

$$\bar{u}_T \approx (P)^0 (T)^{1,6} (\alpha)^{a_0} (n)^{a_5} (V/n)^{a_6} = (T)^{1,6} (\alpha)^{a_0} (n)^{a_5} (V/n)^{a_6}, \quad (9)$$

where  $l$  - is the distance from the ignition source to the combustor wall.

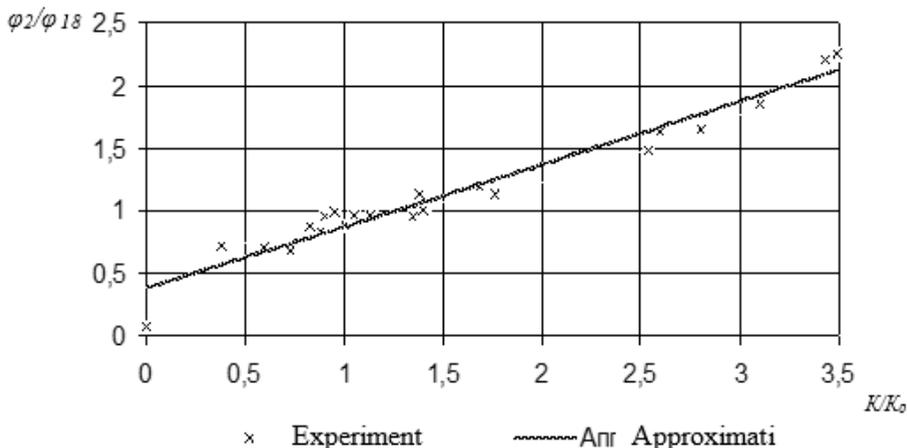
Knowing the indicators of the gas mixture and engine parameters in the source mode ( $V_0, T_0, \alpha_0, n_0$ ) and those in the mode, you can associate the turbulent velocity in these two modes [2]. Obviously:

$$\frac{\phi_2}{\phi_{18}} \approx \frac{(T)^{-1,6} (\alpha)(V/n)}{(T_0)^{-1,6} (\alpha_0)(V_0/n_0)} = \frac{K}{K_0}, \quad (10)$$

## 4 Results

This ratio gives an accurate representation of the correspondence between theoretical calculation indices with the actual obtained in the test gas of the engine. The precise control of methane input and air due to the choice of the optimum approximating a line within the second phase of combustion (Figure 2), provides a balanced ratio of power and economy gas engine [2].

To determine the dependence of the relative angle of the burnout phase of combustion from  $K/K_0$  will show to figure 2.



**Fig. 2.** The dependence of the relative angle of the burnout phase of combustion from the relative complex, [2].

Therefore, when calculating the value of we can use the linear relationship:

$$\phi_2 = \left( 0,3150 \left( \frac{T_0}{T} \right)^{1,6} \left( \frac{\alpha}{\alpha_0} \right) \left( \frac{V}{V_0} \right) \left( \frac{n_0}{n} \right) + 0,6378 \right) + \phi_{18}, \quad (11)$$

where  $n_0, n$  - is the number of revolutions of the engine at work in source and design modes;

$\alpha_0, \alpha$  - coefficient of excess air when operating in source and design modes;

$T_0, T$  - is averaged over the period of combustion temperatures, when you work in source and design modes;

$V_0, V$  - volume of gas-air mixture entering the cylinder when you work in source and design modes.

## 5 Discussion

The obtained results testify to the validity of the developed physically based model, where the key indicators in the regulation of the gas-air mixture are:  $T, \alpha, V, n$ . The optimum angle of rotation of the crankshaft of a gas engine at which the maximum speed of propagation of the flame front in the combustion chamber is reached is  $\phi_2 = 15^\circ \dots 18^\circ$ .

## 6 Conclusion

The analysis of the parameters determining the combustion in the cylinder of a gas engine.

Developed a set of upgraded models and calculation methodology of working cycle of a gas engine to take into account the peculiarities of physical processes in the cylinder, caused by combustion of the gas mixture.

We propose a rational method of regulation of gas engines – KamAZ.

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