Method for evaluating the parameters of the flame front propagation process according to the indicator diagram in spark ignition engines

Natalya Smolenskaya¹, Victor Smolenskii¹, Nikolay Korneev² and Yuri Prus²

¹ Togliatti State University, Togliatti, Belorusskaya st. 14, Russian Federation; ² Gubkin University, Moscow, Leninsky Prospect, 65, Russian Federation

Abstract. The paper presents a methodology for evaluating the parameters of the flame front propagation process using the experimentally obtained indicator pressure diagram in spark ignition engines. The proposed method allows us to determine the main parameters of the flame front propagation process (flame front propagation velocity, flame front heat release rate, flame front thickness, temperature of the burned mixture and temperature of the unburned mixture) from the experimental pressure indicator diagram obtained for a spark ignition engine.

1. Introduction

Research of the working process in spark ignition engines is actively ongoing at the present time [1,2,3]. The main direction of the research is the search for new fuels and ways of organizing the workflow for low-toxic operation of the engine with spark ignition [4,5]. Various approaches are used to study the combustion process, these are laser and optical methods for visualizing the combustion process, this is the use of ionization sensors and other research tools [6,7]. But the most common is the way to evaluate the workflow by changing the pressure in the engine cylinder. This method is well developed for many years of practice in its application. To do this, there is a significant number of diverse measuring tools that allow you to measure the pressure in the internal combustion engine with the necessary accuracy. Processing and analysis of the obtained data is an important task in assessing the conformity of simulation results to the course of the combustion process. Let us consider the possibilities of evaluating the working process and the main characteristics of the propagation of the flame front from the experimentally obtained indicator pressure diagram in engines with spark ignition. The proposed technique allows you to get more information about the combustion process by analyzing the pressure changes in the engine cylinder.

2. Experimental technique

Experimental studies were carried out on a single-cylinder UIT-85 installation (figure 1a). Include information about the geometric parameters of the engine UIT-85: number of cylinders – 1; working volume – 0.652 liter; compression ratio – 7; diameter of the cylinder – 85 mm; piston stroke – 115 mm; length of connecting rod – 266 mm; rotational speed – 900 rpm; ignition – spark plug, at a fixed ignition advance angle of 13 BTDC for a more visual comparison of the results obtained.
The pressure in the engine cylinder was measured with a Kistler pressure sensor. Local conditions of propagation of the flame front was determined by means of sensors of ionization.

3. Results and Discussion
The analysis of thermodynamic processes in the engine cylinder begins with the first law of thermodynamics:

\[ \Delta Q_C = \Delta U + \Delta A - \Delta Q_{\text{walls}}, \]  

where \( \Delta Q_C \) – the change in the amount of heat released during the combustion process; \( \Delta U \) – change in the internal energy of the system; \( \Delta A \) – change in the work of the thermodynamic system; \( \Delta Q_{\text{walls}} \) – change in the amount of heat energy loss through the walls.

![Figure 1. Single-cylinder research unit UIT-85: (a) general view of the installation; (b) combustion chamber with installed sensors.](image)

Quantitatively, the change in work in a thermodynamic system determines how

\[ \Delta A = p \cdot \Delta V. \]  

The amount of thermal energy loss through the walls is determined by the Voshni equation. The magnitude of the change in the internal energy of the system is determined by the method proposed in [8]. The amount of heat that is released in the flame front is determined from the following expression (3):

\[ \Delta Q_C = H_u \cdot \Delta m_{\text{combustion}}, \]  

where \( H_u \) – lower calorific value of fuel; \( \Delta m_{\text{combustion}} \) – mass of burned fuel in the study area.

Therefore, the proportion of the burned mixture in the area is determined from the expression (4):

\[ \Delta \chi = \frac{\Delta m_{\text{combustion}}}{m_{\text{fuel}} - m_{\text{incomplete combustion}}}, \]
where \( m_{\text{fuel}} \) – mass of fuel entering the engine cylinder; \( m_{\text{incomplete combustion}} \) – mass of fuel not involved in the combustion process due to lack of oxygen.

We divide the thermodynamic system in which the combustion process occurs into two zones. The zone with the burned mixture and the zone with the unburned mixture. These zones are separated by a thin flame front, which passes through the middle of the combustion front where combustion occurs.

We present the equations of state for a two-zone model of the combustion process (5, 6):

\[
p \cdot V_{\text{burn}} = m_{\text{burn}} \cdot R_{\text{burn}} \cdot T_{\text{burn}},
\]
\[
p \cdot V_{\text{unburn}} = m_{\text{unburn}} \cdot R_{\text{unburn}} \cdot T_{\text{unburn}},
\]

where \( p \) – pressure in burned and unburned mixtures; \( V_{\text{burn}} \) and \( V_{\text{unburn}} \) – volumes of burned and unburned mixtures; \( m_{\text{burn}} \) and \( m_{\text{unburn}} \) – masses of zones with burned and unburned mixture; \( R_{\text{burn}} \) and \( R_{\text{unburn}} \) – gas constants for burned and unburned mixtures; \( T_{\text{burn}} \) and \( T_{\text{unburn}} \) – temperature of the burned and unburned mixtures.

We express the masses of the burned and unburned mixtures through the mass of the working mixture \( m_{\text{work}} \) (mixture of fuel, air, residual gases) and the heat release characteristic:

\[
m_{\text{burn}} = m_{\text{work}} \cdot \chi_i,
\]
\[
m_{\text{unburn}} = m_{\text{work}} \cdot (1 - \chi_i),
\]

where \( \chi_i \) – current value of the heat release characteristic.

As a rule, combustion occurs with a rapid increase in pressure, which leads to uneven temperatures in the areas of the burned and unburned mixtures. As a result, it can be assumed that the expanding zone of the burned mixture adiabatically compresses the zone with the unburned mixture [9]. This assumption is based on the fact that the compression rate of the zone with the unburned mixture is high enough and the amount of heat removed to the wall approximately corresponds to the amount of heat that the zone receives from the flame front.

Then the temperature of the unburned mixture at the time of ignition can be determined from the equation of state (9):

\[
T_{\text{ign}} = \frac{p \cdot V}{m_{\text{work}} \cdot R_e},
\]

where \( T_{\text{ign}} \) – temperature of the working fluid in the engine cylinder at the moment of spark supply; \( p \) – cylinder pressure; \( V \) – volume of unburned mixture; \( m_{\text{work}} \) – working fluid mass in the engine cylinder; \( R_e \) – gas constant for the working fluid in the engine cylinder.

The temperature of the unburned mixture during combustion during adiabatic compression is determined by equation (10):

\[
T_{\text{unburn}} = T_{\text{ign}} \left( \frac{p_2}{p_{\text{ign}}} \right)^{(\gamma-1)/\gamma},
\]
where $T_{\text{unburn.}}$ – temperature in the area with unburned mixture; $P_2$ – pressure in the combustion process at the desired time; $P_{\text{ign}}$ – pressure at the time of sparking; $\gamma$ – adiabatic exponent for a mixture of gases in an unburned mixture.

Suppose that the temperature of the thermodynamic system in the engine cylinder can be determined through the calculated expression for the temperature of the gas mixture without taking into account compressibility (11) [10, 11]. This is possible due to the fact that for temperatures of $1500 – 3500$ K and pressure up to $10$ MPa, the effect of compressibility does not exceed $2\%$. Based on the assumption that the pressure in the burned and unburned part of the combustion chamber, we will equally express the mass of the burned and unburned mixtures through the mass of the working mixture, as is done in equations (7) and (8):

$$T = \frac{m_{\text{burn.}} \cdot C_{\text{burn.}} \cdot T_{\text{burn.}} + m_{\text{unburn.}} \cdot C_{\text{unburn.}} \cdot T_{\text{unburn.}}}{m_{\text{work}} \cdot C_{\text{burn.}} + m_{\text{work}} \cdot C_{\text{unburn.}}},$$  \hspace{1cm} (11)

from here

$$T = \frac{X_i \cdot C_{\text{burn.}} \cdot T_{\text{burn.}} + (1 - X_i) \cdot C_{\text{unburn.}} \cdot T_{\text{unburn.}}}{X_i \cdot C_{\text{burn.}} + (1 - X_i) \cdot C_{\text{unburn.}}}$$

where $C_{\text{burn.}}$ and $C_{\text{unburn.}}$ – average molar heat capacities at a constant volume of burned and unburned mixtures are determined by known thermophysical formulas; $T_{\text{burn.}}$ – temperature in the zone of the burned mixture.

From equation (11) we express the temperature in the zone of the burned mixture (12):

$$T_{\text{burn.}} = \frac{T \cdot (X_i \cdot C_{\text{burn.}} + (1 - X_i) \cdot C_{\text{unburn.}}) - (1 - X_i) \cdot C_{\text{unburn.}} \cdot T_{\text{unburn.}}}{X_i \cdot C_{\text{burn.}}},$$  \hspace{1cm} (12)

Then substituting the masses of the burned and unburned mixtures into the equations (5) and (6) through the mass of the working mixture from equations (7) and (8), we obtain the formulas for determining the volumes of the zones of the burned and unburned mixtures (13) and (14):

$$V_{\text{burn.}} = \frac{m_{\text{work}} \cdot X_i \cdot R_{\text{burn.}} \cdot T_{\text{burn.}}}{P},$$  \hspace{1cm} (13)

$$V_{\text{unburn.}} = \frac{m_{\text{work}} \cdot (1 - X_i) \cdot R_{\text{unburn.}} \cdot T_{\text{unburn.}}}{P}. \hspace{1cm} (14)$$

Let us give an example of analysis of the pressure indicator diagram obtained at a single-cylinder research unit UIT-85 according to the proposed methodology. For example, the operating mode on compressed natural gas was selected with an air excess coefficient of 1.005 and a residual gas coefficient of 0.85. Natural gas is supplied through calibrated nozzles to the intake manifold. The mixture in the engine cylinder is significantly diluted with residual gases, due to the low degree of compression and valve overlap. This allows you to evaluate the performance of the proposed methodology on the single-cylinder unit operating mode simulating the engine at low loads.

Having a three-dimensional model of a combustion chamber with a movable piston (figure 2a), one can obtain a characteristic of the propagation velocity of the flame front. To do this, it is necessary to build, for the required moment of the combustion process, a zone of the burned mixture spherically...
distributed from the spark plug, the volume of which corresponds to the volume calculated by equation (13). The characteristic of the propagation velocity of the flame front is shown in figure 2b, which shows the change in velocity between the studied areas and shows two points obtained from experimental data. These points correspond to the appearance of a signal on the ionization sensors. The first ionization sensor is installed 7 mm from the spark plug electrode. The second ionization sensor is installed 80 mm from the spark plug electrode.

![Figure 2. Analysis of changes in the velocity of propagation of the flame front: (a) three-dimensional model of the combustion chamber, cylinder and piston UIT-85 for determining the parameters of the combustion process; (b) characteristics of changes in the velocity of propagation of the flame front and changes in the volume occupied by the flame front.](image)

To determine the width of the flame front, at each moment of the combustion process (17), we determine the density of the unburned mixture zone $\rho_{\text{unburn.}}$ from the known volume (16) and the mass of this zone (15). Figure 2b shows the change in volume in which the combustion process occurs, for each section in 0.2724 degree steps of the rotation angle. Also, according to the three-dimensional model of the combustion chamber with a movable piston and the area of the burned mixture spherically distributed from the spark plug, we determine the area of the flame front $\Delta S_{\text{comb.}}$. Then the combustion zone will be the volume in which the fraction $\Delta \chi$ of the mass component is burned $\Delta m_{\text{comb.}}$. The general equations for the calculation are given in (15), (16) and (17):

$$\Delta m_{\text{comb.}} = \Delta \chi \cdot m_{\text{work}},$$ (15)

$$\Delta V_{\text{comb.}} = \frac{\Delta m_{\text{comb.}}}{\rho_{\text{unburn.}}},$$ (16)

$$\Delta \delta_{\text{comb.}} = \frac{\Delta V_{\text{comb.}}}{\Delta S_{\text{comb.}}}.$$ (17)

Figure 3 shows graphs of the obtained thermodynamic characteristics of the combustion process according to the indicator diagram in UIT-85. The obtained characteristics of changes in the volume of burned and unburned mixtures (figure 3a) are explained by the shape of the UIT-85 combustion
chamber. Figure 3b shows that already at 352 degrees crankshaft angle a stable flame front is formed with an approximately constant thickness, corresponding to the combustion conditions at low turbulence. An analysis of the shape of the pressure indicator diagram (figure 3a) with flame front propagation characteristics (figure 3b) shows the reliability of the calculation.

The presented results make it possible to simplify the analysis of thermodynamic parameters of the combustion process. Including operating modes having increased emissions of toxic components. This may allow a new look at the possibilities of optimizing the workflow to create modern low-toxic engines.

4. Conclusion

The possibility of evaluating the main parameters of the flame front propagation process using the experimentally obtained pressure indicator diagram in spark ignition engines is shown.

The proposed method allows us to determine the main parameters of the flame front propagation process using the experimentally obtained indicator pressure diagram in spark ignition engines, including when the engine is idling and low loads. In these modes, there is a significant amount of residual gases that significantly complicate the modeling process for these conditions, which requires the mandatory use of bench experimental studies.

References

[1] Li Y, Jia M, Chang Y, Kokjohn S L and Reitz R D 2016 *Applied Energy* **180** 849-58
[2] Zhao Z, Wang S, Zhang S and Zhang F 2016 *Energy* **102** 650-9
[3] Sohret Y, Gürbüz H and Akçay I H 2019 *Energy* **175** 410-22
[4] Siti Sabariah M et al 2019 *IOP Conf. Ser.: Mater. Sci. Eng.* **469** 012076
[5] Sezer İ and Bilgin A 2014 *Thermophysics and Heat Transfer* **28**(2) 347–55
[6] Irimescu A et al 2014 *Energy Conversion and Management* **87** 914–27
[7] Yuedong Chao et al 2019 *Energy Procedia* **158** 2098–105
[8] Smolenskaya N M et al 2018 *IOP Conf. Ser.: Earth Environ. Sci.* **121** 052009
[9] Duarte J et al 2016 *TECCIENCIA* **11**(20) 57-65
[10] Mamalis S et al 2014 *International J of Engine Research* **15**(6) 641–53
[11] Barjaneh A and Sayyaadi H 2015 *Energy Conversion and Management* **105** 607–16