Methods for Assessing the Thermodynamic Characteristics of the Combustion Process Using the Indicator Diagram in Spark-Ignition Engines

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Abstract. The paper deals with the evaluation of the thermodynamic characteristics of the combustion process according to the experimental obtained indicator diagram of pressure in spark ignition engines. The proposed method allows to determine the main characteristics of the combustion process (change in internal energy, heat generation characteristic, speed and intensity of combustion processes in the flame front, volume of the burned mixture, volume of unburned mixture) using the experimental indicator diagram of pressure obtained for a spark ignition engine.

1. Introduction
The indicator diagram of the pressure in the engine cylinder carries a significant amount of information about the course of thermal processes in compression, combustion and expansion [1, 2]. The indicator diagram is an integral characteristic of thermal processes. Her detailed analysis allows to determine the following local parameters of the combustion process: the average temperature of the burned and unburned mixtures, the volume of the burned and unburned mixtures, the average propagation velocity of the flame front at each moment of the combustion process, the flame front area, the intensity of the combustion processes in the flame front, the average flame front thickness, heat dissipation characteristic [3]. All these data allow us to better assess the impact of various engine operating parameters on environmental and effective performance. It should also be noted the complexity of the calculations of the combustion process in modern software products with a significant proportion of the dilution of the fuel-air mixture with residual gases that have an unstable chemical composition. Therefore, at present, in addition to the use of modern software products for the calculation of internal combustion engines, experimental studies are also actively used [4]. In such studies, research on engine performance at low loads and idling play a large role. These characteristics significantly affect the performance of the car in a city. In this regard, an increase in reliable data when analyzing the results of experimentally obtained indicator pressure diagrams for engine operation modes with significant dilution with residual gases is an important task of scientific research.

2. Theoretical analysis
The combustion chamber is a thermodynamic system that is limited to the cylinder head, valves, cylinder liner and piston. There are losses of thermal energy through the walls, which can be determined by the equations proposed by Voshni [5]. Since the temperatures in the combustion zone, when operating at idle and low loads, barely exceed 2000 K, there are practically no dissociation effects. Pressure on these modes usually does not exceed 4 – 5 MPa. This allows us to consider the leakage through the piston rings insignificant and neglected in the calculations.

The indicator diagram of pressure from the first law of thermodynamics, you can determine the amount of heat released during combustion in the flame front. The total character of which is a characteristic of heat dissipation and describes the energy release due to combustion (1):

\[ \Delta Q_c = \Delta U - \Delta Q_{\text{walls}} + p \cdot \Delta V. \]
where: \( \Delta Q_C \) – the amount of heat released in the combustion process; \( \Delta U \) – change the internal energy of the system; \( \Delta Q_{\text{walls}} \) – loss of heat energy through the walls; \( p \cdot \Delta V \) – thermodynamic system operation.

In equation (1), heat losses to the cooling system through walls are calculated in accordance with the equation proposed by Voshni [5]. The operation of the thermodynamic system is determined by the indicator diagram and the characteristic motion of the piston for each section of the indicator diagram. The size of the section is adopted so that the pressure change from the beginning to the end of the section can be considered straight, in our case it is 0.2724 degrees of the angle of rotation. Earlier, the authors obtained a method for estimating the internal energy from the parameters of the indicator diagram. The method allows to more reliably determine this parameter under conditions of dilution with residual gases, including during idling and low load [6]

\[
\Delta U = \frac{C_{V_1} + C_{V_2}}{2} \cdot \frac{(V_1 + V_2)}{2} \cdot \left( P_2 - P_1 \right) \cdot \frac{2}{R_m} \cdot \left[ Z_r(p, T_1) + Z_r(p, T_2) \right]
\]

where: \( C_{V_1} \) and \( C_{V_2} \) – average molar heat capacities with a constant volume at the beginning and at the end of the considered section; \( V_1 \) and \( V_2 \) – volumes of the thermodynamic system at the beginning and at the end of the considered section; \( P_1 \) and \( P_2 \) – pressure on the indicator diagram at the beginning and at the end of the section under consideration; \( R_m = 8.314 \text{ J/(mol-K)} \) – the universal gas constant; \( Z_r(p, T_1) \) and \( Z_r(p, T_2) \) – compressibility factors of real gas at the beginning and at the end of the considered section.

Also, the amount of heat released in the flame front is determined from the following expression (3):

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

where: \( H_u \) – net caloric value of the fuel; \( \Delta m_{\text{combustion}} \) – mass of burnt fuel in the area under study.

Therefore, the proportion of burnt 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: \( m_{\text{fuel}} \) – the mass of fuel entering the engine cylinder; \( m_{\text{incomplete combustion}} \) – mass of fuel not participating in the combustion process due to lack of oxygen when working on rich fuel-air mixtures.

Thus, we can determine the heat release characteristic from the expression (5):

\[
\chi = \sum \Delta \chi,
\]

As a result, we have all the necessary data regarding the thermodynamic system in which combustion takes place. The thermodynamic system, where the combustion process takes place, can be divided into several zones. The dual-zone approach is often used to simulate combustion [7,8]. The combustion chamber is divided into a hot zone with a burnt fuel-air mixture and a cold zone with an unburned fuel-air mixture. These zones are separated by a thin flame front (middle line passing through the middle of a real flame front) in which combustion and heat release occurs (Figure 1). It is also assumed that inside the combustion chamber the pressure propagates at the speed of sound, and the rate of combustion is significantly lower than it. Therefore, the pressure in the burned and unburned mixtures is considered the same [9].

We present the equations for the two-zone model of the combustion process. The equations of state for the two-zone combustion process model (6, 7):

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

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

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

![Figure 1. Two-zone simulation of the combustion chamber system](image)

Express the mass of the burned and unburned mixtures through the mass of the working mixture (mixture of fuel, air, residual gases) and the characteristics of heat generation:

$$m_{\text{burn}} = m_{\text{work}} \cdot \chi_i,$$

$$m_{\text{unburn}} = m_{\text{work}} \cdot (1 - \chi_i).$$

where: $\chi_i$ – current value of heat dissipation characteristic.

Express from equations (6) and (7) the temperature of the burned and unburned mixtures:

$$T_{\text{unburn}} = \frac{p \cdot V_{\text{unburn}}}{m_{\text{unburn}} \cdot R_{\text{unburn}}} = \frac{p \cdot V_{\text{unburn}}}{m_{\text{work}} \cdot (1 - \chi_i) \cdot R_{\text{unburn}}};$$

$$T_{\text{burn}} = \frac{p \cdot V_{\text{burn}}}{m_{\text{burn}} \cdot R_{\text{burn}}} = \frac{p \cdot V_{\text{burn}}}{m_{\text{work}} \cdot \chi_i \cdot R_{\text{burn}}}.\,$$

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. Since it is already assumed that the pressure in the burned and unburned part of the combustion chamber is the same. Express the mass of burned and unburned mixtures through the mass of the working mixture as it is done in equations (8) and (9):

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

where: $C_{\text{burn}}$ and $C_{\text{unburn}}$ – average molar heat capacity with a constant volume of burned and unburned mixtures.
The average molar heat capacity at a constant volume of the burned and unburned mixtures is determined by the well-known thermophysical formulas.

Express from equation (12) the temperature of the burned and unburned mixtures, taking into account expressions (10) and (11):

$$T_{\text{burn}} = \frac{T \cdot (\chi_i \cdot C_{\text{burn}} + (1 - \chi_i) \cdot C_{\text{unburn}}) - C_{\text{unburn}} \cdot m_{\text{work}} \cdot R_{\text{unburn}}}{m_{\text{work}} \cdot \frac{p \cdot V_{\text{unburn}}}{R_{\text{unburn}}}}$$; \hspace{1cm} (13)

$$T_{\text{unburn}} = \frac{T \cdot (\chi_i \cdot C_{\text{burn}} + (1 - \chi_i) \cdot C_{\text{unburn}}) - C_{\text{burn}} \cdot m_{\text{work}} \cdot R_{\text{burn}}}{m_{\text{work}} \cdot \frac{p \cdot V_{\text{burn}}}{R_{\text{burn}}}}.$$. \hspace{1cm} (14)

The volume of the thermodynamic system in the engine cylinder at the current time is determined by the expression (15):

$$V_i = V_{\text{comb. cham.}} + \frac{\pi \cdot D^2}{4} S_i,$$ \hspace{1cm} (15)

where: $V_{\text{comb. cham.}}$ – volume of the combustion chamber; $D$ – cylinder diameter; $S_i$ – piston stroke relative to the top dead center.

Assuming that the zones are separated by a thin flame front (middle line passing through the middle of the real flame front), we can express the volume of the thermodynamic system through the volumes of burned (16) and unburned (17) mixtures:

$$V_{\text{burn}} = V_i - V_{\text{unburn}},$$ \hspace{1cm} (16)

$$V_{\text{unburn}} = V_i - V_{\text{burn}}.$$. \hspace{1cm} (17)

Let us express from the equations (6) and (7) the volumes of the burned and unburned mixtures taking into account (8) and (9) respectively:

$$V_{\text{burn}} = m_{\text{work}} \cdot \frac{\chi_i \cdot R_{\text{burn}} \cdot T_{\text{burn}}}{p},$$ \hspace{1cm} (18)

$$V_{\text{unburn}} = m_{\text{work}} \cdot \frac{(1 - \chi_i) \cdot R_{\text{unburn}} \cdot T_{\text{unburn}}}{p}.$$. \hspace{1cm} (19)

Consider the solution of expressions (18) and (19) separately. Let us substitute into expression (18) the formula (13) in which we express the volume through expression (16):

$$V_{\text{burn}} = m_{\text{work}} \cdot \frac{\chi_i \cdot R_{\text{burn}} \cdot T_{\text{burn}}}{p} \cdot \frac{T \cdot (\chi_i \cdot C_{\text{burn}} + (1 - \chi_i) \cdot C_{\text{unburn}}) - C_{\text{unburn}} \cdot m_{\text{work}} \cdot R_{\text{unburn}}}{m_{\text{work}} \cdot \frac{p \cdot [V_i - V_{\text{burn}}]}{R_{\text{burn}} \cdot C_{\text{burn}} - R_{\text{unburn}} \cdot C_{\text{unburn}}}}.$$. \hspace{1cm} (20)

In formula (20) one unknown quantity is the volume of the burned mixture. Simplify the formula (20) and expressing from it the volume of the burnt mixture, we obtain the calculation formula (21):

$$V_{\text{burn}} = m_{\text{work}} \cdot \frac{R_{\text{burn}} \cdot R_{\text{unburn}} \cdot T \cdot (\chi_i \cdot C_{\text{burn}} + (1 - \chi_i) \cdot C_{\text{unburn}}) - p \cdot R_{\text{burn}} \cdot C_{\text{burn}} \cdot V_i}{p \cdot (R_{\text{unburn}} \cdot C_{\text{burn}} - R_{\text{burn}} \cdot C_{\text{unburn}})},$$ \hspace{1cm} (21)

Let’s carry out similar actions with the formula (19) to determine the volume of the unburned mixture and get the calculation formula (22)

$$V_{\text{unburn}} = m_{\text{work}} \cdot \frac{R_{\text{unburn}} \cdot R_{\text{burn}} \cdot T \cdot (\chi_i \cdot C_{\text{burn}} + (1 - \chi_i) \cdot C_{\text{unburn}}) - p \cdot R_{\text{unburn}} \cdot C_{\text{burn}} \cdot V_i}{p \cdot (R_{\text{burn}} \cdot C_{\text{unburn}} - R_{\text{unburn}} \cdot C_{\text{burn}})},$$ \hspace{1cm} (22)

Determining by the formula (21) or (22) the volume of the burned or unburned mixture, we can determine all the remaining volumes by the formulas (16) and (17). And using formulas (10) and (11), we can calculate the temperature of the zone of the burned mixture and the zone of the unburned mixture.
By formulas (8) and (9) we know the mass of the zone of the burned mixture and the mass of the zone of the unburned mixture.

3. Experimental technique

Experimental studies were carried out on a single-cylinder UIT-85 installation. Include information about the geometric parameters of the engine UIT-85: number of cylinders – 1; working volume – 0.652 l; 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.

4. Results and Discussion

Let us give an example of the analysis of the pressure indicator diagram obtained on the UIT-85 single-cylinder research unit using the proposed method. As an example, the mode of operation of the plant on compressed natural gas is chosen. Natural gas is fed through a calibrated nozzle into the intake manifold. The mixture in the engine cylinder is significantly diluted with residual gases, due to the low compression ratio and valve overlap. This allows you to evaluate the performance of the proposed method on the mode of operation of a single-cylinder unit simulating engine operation at low loads.

Figure 2 shows the graphs of the obtained thermodynamic characteristics of the combustion process according to the indicator diagram in the UIT-85. For example, the mode of operation on compressed natural gas is selected with an excess air ratio of 1.005 and a residual gas ratio of 0.85 [3]. The results obtained in Figure 2 show the applicability of the proposed method for analyzing the thermodynamic characteristics of the combustion process according to the experimentally obtained indicator diagram of pressure in spark ignition engines. The obtained characteristics of changes in the volume of burned and unburned mixtures are explained by the shape of the UIT-85 combustion chamber. In Figure 2a it can be seen that the active increase in the volume of the burnt mixture starts at 370 degrees BTDC. And at 395 degrees BTDC the activity of combustion and the growth of the volume of the burned mixture is reduced. This is confirmed by the above parameters of the characteristics of heat generation and changes in the volume of burned and unburned mixtures in Figure 2b. An analysis of the shape of the pressure indicator chart with the characteristics of changes in the volume and mass of the burned mixture (Figure 2a and 2b) shows the accuracy of the above calculation.

**Figure 2.** Graphics thermodynamic characteristics of the combustion process depending on the angle of rotation of the crankshaft: (a) indicator pressure, volume of the combustion chamber, volumes of burned and unburned mixtures; (b) the characteristic of heat generation is given, and the change is the mass of the zone of the burnt and unburned mixtures

Having obtained the necessary data on the volume change during the combustion process, one can proceed to the analysis of the change in the velocity of propagation of the flame front. To do this, we construct a three-dimensional model of the thermodynamic system under consideration, which is a combustion chamber with a moving piston. Using the 3D model, it is possible to determine, for each moment of time under consideration, the position of the spherical flame front (Figure 3a). The
propagation parameters of the spherical flame front, taking into account the position and volume of the burned mixture, were determined in increments of 5 degrees BTDC. The results are shown in Figure 3b. Figure 3b shows the characteristics of the average speed of propagation of a spherical flame front from the moment of the spark discharge to the corresponding position of the flame front to the angle. The speed was considered from the change in the radius of the spherical flame front. Also shown in Figure 3b is the change in speed between the test sites, and two points are shown, obtained from experimental data. These points correspond to the appearance of a signal on the ionization sensors. The first ionization sensor is mounted 7 mm from the spark plug electrode. The second ionization sensor is installed 80 mm from the spark plug electrode.

![Figure 3a](image1.png)
![Figure 3b](image2.png)

**Figure 3.** Analysis of the change in the speed of propagation of the flame front: (a) a three-dimensional model of the combustion chamber, cylinder and piston UIT-85 to determine the parameters of the combustion process; (b) Characteristics of the change in the flame propagation velocity of the flame front.

The results obtained correspond to the general picture of the combustion process and are fairly well correlated with the experimental data obtained for the initial combustion phase (first ionization sensor) and the beginning of the final combustion phase (second ionization sensor).

Therefore, assuming that on average the flame front extends from the spark plug spherically, you can determine the exact coordinates of its front border. This will determine the speed of propagation of the flame front. You can also determine the flame front area, from where, knowing the current mass burning rate and the average density of the unburned mixture, you can determine the average flame front thickness at a given time [10, 11, 12].

The presented results make it possible to simplify the process of analyzing the thermodynamic parameters of the combustion process, on operating conditions with increased emissions of toxic components. This may allow a fresh look at the possibilities of optimizing the workflow for the creation of modern low-toxic engines.

5. **Conclusion**

1. The possibility of evaluating the main characteristics of the combustion process is shown by analyzing the indicator diagram of pressure in the cylinder of a spark-ignited engine.

2. The proposed method allows to determine the main thermodynamic characteristics of the combustion process according to experimentally obtained indicator diagram of pressure, including when the engine is running at idle 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. The proposed method allows to expand the understanding of the
conditions for the formation of toxic substances in spark ignition internal combustion engines. This is necessary to identify opportunities to improve the environmental performance of the engine at low loads and idling.

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