NOVEL METHOD APPOINTING OF THE EFFECTIVE HEAT RELEASE COEFFICIENT DURING COMBUSTION PROCESS BASED ON REAL INDICATOR DIAGRAM

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Abstract

An object of the paper is a novel method of effective heat release coefficient appointing during combustion process of the based on the real indicator diagram. The process quality of the heat release during process of combustion is evaluated by value of the effective heat release coefficient.

Accepted assumptions at appointing effective coefficient of heat release refer to acceptable that thermodynamical parameters of the working charge according to points of the real indicator diagram and the computational diagram are such the same, the maximum pressure of real working cycle of the engine is equal to the pressure of the computational cycle, work of the working charge performed in period from point of closing of the inlet valve to point of the beginning of the exhaust process are for both considered working cycles equal, heat values of the carried to the real and computational cycle are such the same.

The method of appointing of the self-ignition delay period is presented in the paper, using to this end intersection point of two graphs of temperature course of the working charge prepared for the first phase of combustion process. Other methods appointing characteristics is suggested in paper basing on the real indicator diagram.

The graphic illustration of the preparing method of characteristics of the relative quantity of the heat release during combustion process is illustrated in paper.

Keywords: *combustion engine, working cycle, thermodynamics, indicator diagram, heat release*

1. Introduction

Most important problems concerning analyses of generalized computational thermodynamical working cycle of the four-stroke combustion engine were prepared as result of the identification of real indicating diagram. The analysis of working cycle of the four-stroke combustion engine was realized from point of view heat efficiency, the average theoretical cycle pressure, as well as of the influence of the organization working cycle on the value of the maximum pressure in the cylinder of a combustion engine.

Improvements and optimizations in the constructive design of engines are made possible only by precise knowledge of fluid mechanics, heat transfer, and combustion processes in the internal combustion engines. These three disciplines have been the subject of a number of investigations over recent years. Modeling engine thermo-fluid characteristics enables new engines to be designed more rapidly, cost effectively, and with improved thermal signatures. Although engine simulations are less expensive and time consuming than prototype construction and testing, they are not without cost. As models become increasingly complex, more inputs are generally required, a faster computer is needed, and the expertise of the model builder must be greater. Therefore, model efficiency must be considered. The thermodynamics which governs the internal combustion process in the combustion chamber, the heat transfer by conduction in the cylinder walls, and the flow as well as the heat transport in the coolant in the water cooling jacket. Owing to the complex physical and chemical processes relevant to turbulent combustion, the determination of the heat transfer from the working gas to combustion-chamber walls is very difficult. In order to obtain the boundary conditions for the FEM analysis at the combustion-chamber side, a simplified gas dynamic cycle calculation is usually carried out.

Main effects of the thermodynamical analysis are:

- calculating of thermodynamical parameters in characteristic points of the working engine cycle, the theoretical efficiency, the average theoretical pressure, the value of the maximum pressure, engine efficiency, theoretical engine power, specific engine fuel consumption,
- calculating of the cylinder filling coefficient,
- calculating of the maximum cycle temperature of working charge while inlet valve opening ,
- calculating the effective coefficient of the heat releasing during combustion process,
- to calculating characteristics of the relative quantity released heat, enabling estimation and the analysis of two phases of combustion processes,
- calculating quantities of heat transferred by the working charge to cylinder walls during combustion process.

2. Heat release coefficient during combustion

The quality of the heat release process during the combustion process is evaluated with the value the effective coefficient of heat release ξ. The value of that coefficient can be determined from the equation of the first principle of thermodynamics for processes occurring in a closed thermodynamic system proceeding from the closing of the inlet valve – the point "d' " to the moment of beginning the exhaust process – the point "w".

The assumptions made during determination of the effective coefficient of heat release ξ:

1. thermodynamic parameters of the working charge corresponding to the points "d'" and "w" (Fig. 1) of the effective indicator diagram and the computational diagram are identical,
2. maximum pressure of the effective engine work cycle is equal to the pressure of the computational cycle in the points "z'" and "z",
3. works of the working charge performed during the period from the moment of closing the inlet valve "d'" to the moment of beginning the exhaust process "w", are identical for both considered work cycles, i.e.:

$$L_{d'-w} = \int_{V_{d'}}^{V_w} p dV , \tag{1}$$

4. values of heat delivered to the effective and computational cycle are identical.

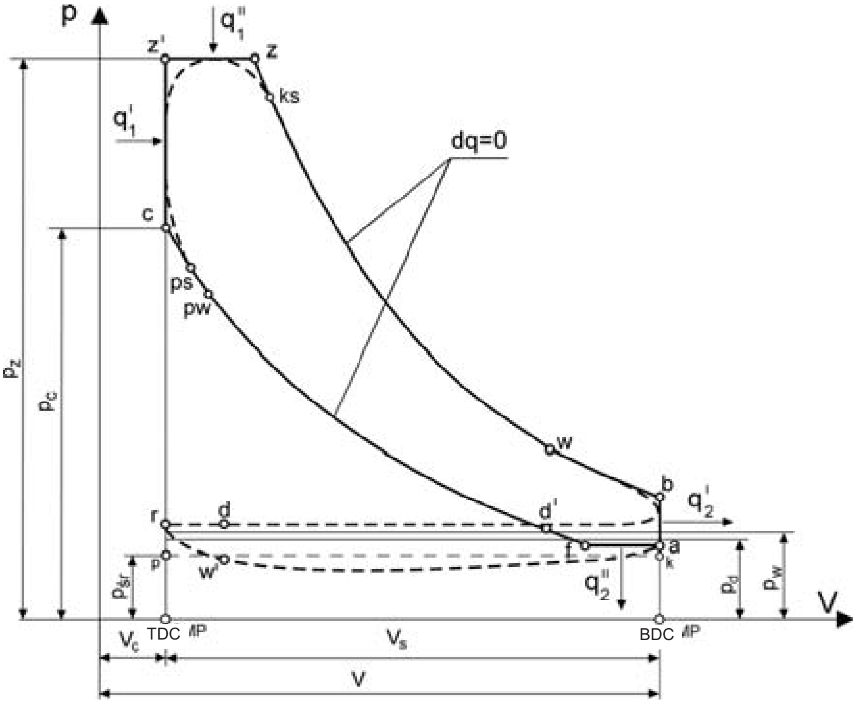

Fig. 1. Thermodynamical engine working cycle

With the above assumptions, the equation of the first principle of thermodynamics for the transformations proceeding from the end of the compression process to the beginning of the exhaust process (the point "w" in Fig. 1) has the form:

$$H_{pal} + Q_{c-z'} + Q_{z'-z} = U_{c-w} + L_{c-w} . \tag{2}$$

Because the transformation z-w proceeds at $\delta q = 0$, then

$$U_{c-w} = U_w - U_c = U_z - U_c , \tag{3}$$

where:

$U_c = \bar{c}_{vc} T_c \lambda M_o g_c (1+\gamma)$ - internal energy of the working charge in the point "c",

$U_z = \beta \bar{c}_{vz} T_z \lambda M_o g_c (1+\gamma)$ - internal energy of the working charge in the point "z".

The work of the working charge L_{c-w} is equal to:

$$L_{c-w} = L_{c-z} + L_{z-w} = p_z (V_z - V_c) + \frac{p_z V_z - p_w V_w}{n_2 - 1} . \tag{4}$$

The enthalpy of the fuel delivered to the cylinder during the engine work cycle is:

$$H_{pal} = c_{pal} g_c T_{pal} , \tag{5}$$

where:

c_{pal} – mass specific heat of fuel.

The heat delivered to the engine work cycle shall be expressed in the form:

$$Q_1 = Q_{c-z'} + Q_{z'-z} = \xi g_c W_u \,. \tag{6}$$

Substituting the above values to (5) and making transformations, we have relations (7) and (8):

$$\xi = \left(\beta \bar{c}_{vz} T_z - \bar{c}_{vc} T_c\right) \frac{\lambda M_o (1+\gamma)}{W_u} + \frac{p_z (V_z - V_c)}{g_c W_u} + \frac{p_z V_z - p_w V_w}{(n_2 - 1) g_c W_u} - \frac{c_{pal} T_{pal}}{W_u} \,. \tag{7}$$

$$\xi = \left(\beta \bar{c}_{vz} T_z - \bar{c}_{vc} T_c\right) \frac{\lambda M_o (1+\gamma)}{W_u} + \frac{p_z V_z}{g_c W_u} \left[\frac{\rho - 1}{\rho} + \frac{\beta \left(\lambda_p - \frac{T_w}{T_c} \right)}{\rho \lambda_p (n_2 - 1)} \right] - \frac{c_{pal} T_{pal}}{W_u} \,. \tag{8}$$

3. Characteristic of heat release during the combustion process

Determination of the characteristic of the relative quantity of the heat emitted during the combustion process, taking into account: the changing quantity of moles of the working charge during combustion and the change of its specific heat depending on the changing composition and temperature and the quantity of heat exchanged with the walls limiting the combustion space, utilising the first principle of thermodynamics, has been described in [1] in a fairly detailed manner. A method for determining the self-ignition delay period has also been presented there, utilising the point of intersection of two diagrams of the temperature course of the working charge, made for the initial phase of the combustion process.

The present paper is to propose a somewhat different methodology for determination of the above mentioned characteristic, also on the basis of the effective indicator diagram. In that method the calculations begin from the point $\alpha = \alpha_{ow} = 540 - \alpha_w{}^\circ$ of crankshaft rotation and they are conducted in the direction of decreasing value of the engine crank angle, with the computational step $\Delta\alpha$. It is assumed that the end of complete and perfect combustion occurred before the beginning of the exhaust process (the beginning of opening the outlet valve) of the exhaust gas, i.e. in the angle range $\alpha \le 540 - \alpha_w$, where α – angle of advance of opening the outlet valve, counted in relation to the piston BDC, corresponding to α = 540° of crankshaft rotation. At the moment of ending the combustion process, till the beginning of the exhaust gas exhausting process, the value of relative quantity of the emitted heat $x = x_i + x_{sc}$ is equal to one. As follows from that assumption, the calculations do not take into account the quantity of heat consumed for the processes of dislocation of the combustion products, which seems to be justified for compression ignition engines in which the maximum mean combustion temperature does not exceed 2500 K

Having the effective indicator diagram p(α) and knowing the basic dimensions of the engine (S – piston stroke, D – piston diameter, ε – compression ratio) and the rotational velocity of the crankshaft n, as well as the thermodynamic parameters in the characteristic points of the computational indicator diagram (Fig. 1) "c" and "w" and the values ξ, g_c and W_u, we can utilise for calculations the equation of the first principle of thermodynamics in the form:

$$\xi g_c W_u - Q_{sc} = U_w - U_c + \int_{V_c}^{V_w} p\,dV \,. \tag{9}$$

Dividing equation (9) by $Q_1 = \xi g_c W_u$ we obtain the equation:

$$1 - x_{sc} = x_i \,, \tag{10}$$

where:

$x_{sc} = \frac{Q_{sc}}{\xi g_c W_u}$ - relative quantity of the heat exchanged with the cylinder walls, while

$x_i = \frac{U_w - U_c + \int_{V_c}^{V_w} pdV}{\xi g_c W_u}$ - relative indicated quantity of the emitted heat going for the exchange of the internal energy of the working charge and for the performing of absolute work by it.

Utilising equation (9) and knowing the temperature of the working charge in the points "c" and "w" (Fig. 1) and the values $p(\alpha)$ (the effective indicator diagram), we can determine both the total and the relative quantity of the heat transferred by the working charge to the cylinder walls during the transformations proceeding between the beginning of the combustion process and the beginning of the exhaust gas exhausting process:

$$Q_{sc} = \xi \cdot g_c \cdot W_u - (U_w - U_c + \int_{V_{ps}}^{V_w} pdV), \tag{11}$$

$$x_{sc} = 1 - x_i = 1 - \frac{U_w - U_c + \int_{V_{pc}}^{V_w} pdV}{\xi \cdot g_c \cdot W_u}, \tag{12}$$

where:

$$U_c = \lambda \cdot M_o \cdot g_c \cdot \bar{c}_{vc} \cdot T_c \cdot (1+\gamma), \tag{13}$$

$$U_w = \beta \cdot \lambda \cdot M_o \cdot g_c \cdot \bar{c}_{vw} T_w \cdot (1+\gamma), \tag{14}$$

$$\int_{V_{pc}}^{V_w} pdV = L_{pc-c} + L_{c-w} = \int_{V_{ps}}^{V_c} pdV + p_z(V_z - V_c) + \frac{p_z V_z - p_w V_w}{n_2 - 1}. \tag{15}$$

Graphic illustration for the method of making the characteristic of the relative quantity of the heat emitted during the combustion process is presented in Fig. 2.

The methodology of making the characteristics of the relative quantity of the heat emitted during the combustion process is realised in two stages. In the first stage we assume that $Q_{sc} = 0$ and we determine the characteristic x, beginning the calculations from the point of the beginning of the outlet valve opening, for which, according to the assumptions made, x = 1 and the working charge is the exhaust gas, the quantity of which is equal to $M_2 = \beta \cdot \lambda \cdot M_o \cdot g_c$, and the exponent of the adiabatic curve $\kappa = \frac{\bar{c}_{pspl}}{\bar{c}_{vspl}}$.

The first stage of making the characteristic of the heat release is realised for the crank angle counting from α=540^o of crankshaft rotation – α_w to the piston TDC position, i.e. for α=360^o of crankshaft rotation, with the step $\Delta\alpha$. The value of change of the relative quantity of the emitted heat in the computational range of the crank angle in this computational process is:

$$\Delta x_j = \frac{1}{(\kappa-1)\cdot\xi\cdot g_c\cdot W_u}\left[\kappa\frac{p_j+p_{j-1}}{2}(V_{j-1}-V_j)+\frac{V_j+V_{j-1}}{2}(p_j-p_{j-1})\right]. \qquad (16)$$

Note: in this range is: $V_j > V_{j-1}$ and $p_j < p_{j-1}$, therefore $\Delta x_j \le 0$.
The value:

$$x_{j-1} = x_j + \Delta x_j\ . \qquad (17)$$

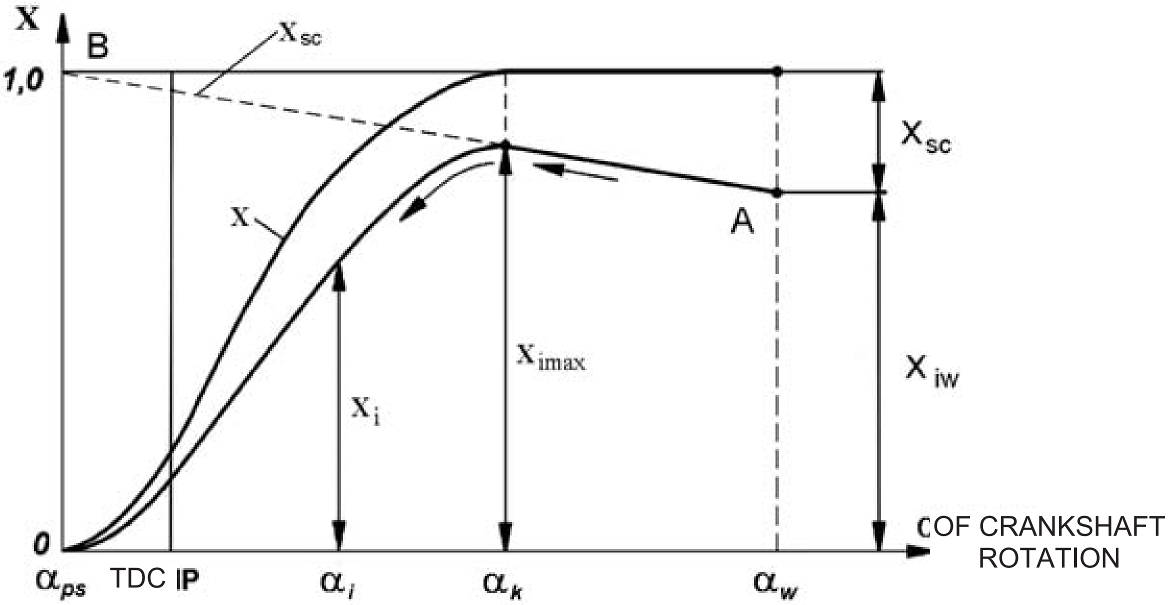

Fig. 2. The graphic illustration for determining characteristics x, xi and xsc: α_{ps} - the beginning of the combustion process, αi – current value of the crank angle, α_k -the end of the combustion process, α_w - the beginning of the engine outlet valve opening

After performing the calculations for the angle $360 \le \alpha < 540 - \alpha_w$, the second stage calculations are performed, i.e. for the angle $360 - \alpha_{pw} \le \alpha \le 360^o$ of crankshaft rotation.

The value Δx_j in that range is also calculated according to formula (16), however their values are greater than zero. Therefore the value x_{j-1} for that range is calculated from the formula:

$$x_{j-1} = x_j - \Delta x_j\ . \qquad (18)$$

We realise the calculations in this stage until the moment when $x_{j-1} \approx 0$. The angle α for which $x_{j-1} = 0$, is assumed as the angle of the beginning of the combustion process. Utilising the calculation results we make the characteristic x, see Fig. 2. Then according to formula (12) we calculate the value x_{sc}, which we mark as shown in Fig. 2, obtaining the point A. Assuming the linear dependence of the relative quantity of heat x_{sc} on the crank angle, through the point B corresponding to the beginning of the combustion and through the point A corresponding to the beginning of the outlet valve opening we draw a straight line, which represents the characteristic of the relative quantity of the heat exchanged between the working charge and the walls limiting the combustion space in the cylinder. Subtracting the value x_{sc} from the value x, we obtain the characteristic of the relative indicator quantity of the heat emitted during the combustion process.

Differentiating the characteristic x or x_i in relation to the crank angle, we obtain the characteristic of the velocity of the adequate relative quantity of the heat emitted during the combustion process. Examples of characteristic diagrams of the relative quantities of the heat emitted in T359 engine are presented in Fig. 3. Knowing the angle corresponding to the beginning of the process of injecting the fuel into the cylinder, and the angle corresponding to the beginning of the combustion process, we can determine the angle of delay of the self-ignition in a compression ignition engine:

$$\Theta_{os} = \alpha_{ps} - \alpha_{pw} \,. \tag{19}$$

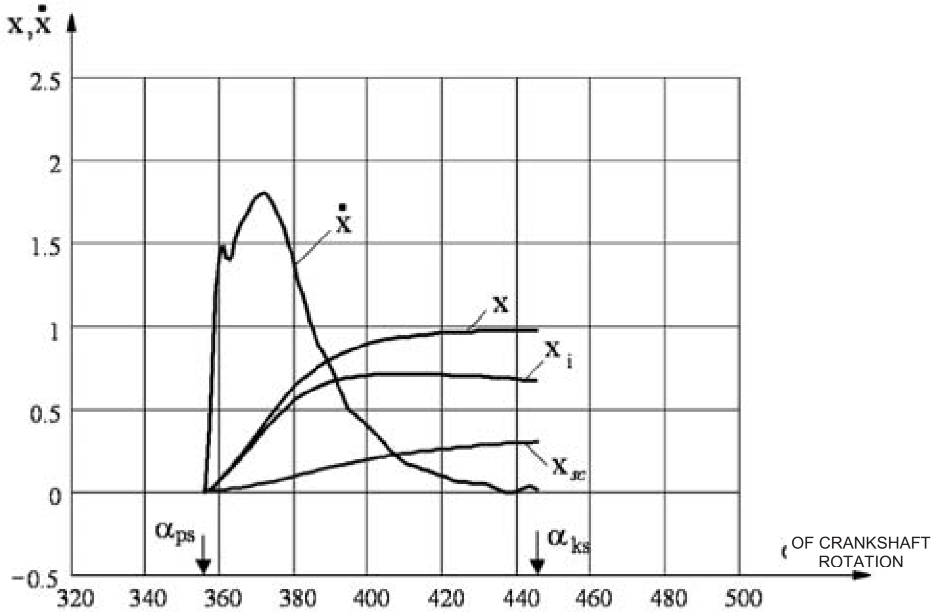

Fig. 3. Relative quantities of the emitted heat during the combustion process and the release velocity of the emitted heat relative quantity in T359 engine [3], working at: n=1900 min^{-1}, g_c=7,16·10^{-5}kg/cycle, p_{pw}=18,5 MPa and α_{ww}=18° of crankshaft rotation before TDC

Presented below are effective indicator diagrams (developed in function of the angle α and closed in p-V coordinates) of AD3.152 engine powered with EDZ fuel and working at n = 1400 RPM and M_e = 168 Nm.

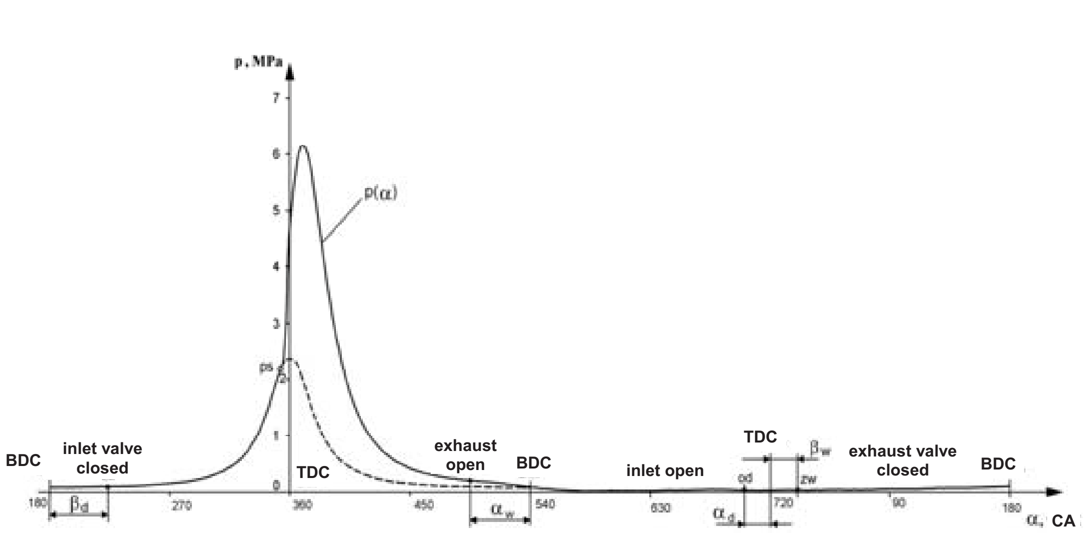

Fig. 4. The effective developed indicator diagram for AD3.152 EDZ-powered engine, working at n = 1400 RPM and M_e = 168 Nm

4. Summary

The problems of thermodynamics of the work cycles of four-stroke piston engines presented in the paper, and the methods of analysis and calculations of work cycles of those engines presented therein, indicate the advisability and necessity of using the proposed generalised engine work cycle for theoretical calculations and analysis of effective work cycles. That permits the determination of the design parameters and the parameters essentially influencing the course and organization of the work cycle (the combustion process) that are the most rational in terms of operational features of the engine, as well as makes it possible to evaluate the usefulness of applying mineral-based fuels, or other, e.g. plant-based fuels, for powering engines.

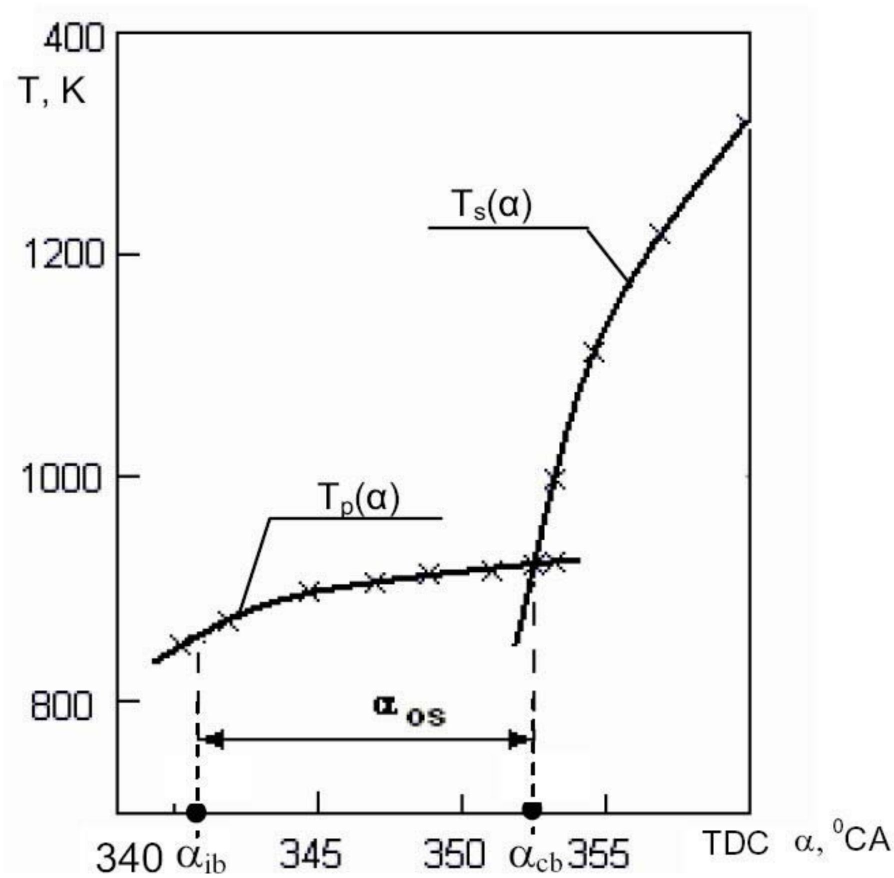

Fig. 5. Graphic presentation of the method for the determination of ignition delay angle α_{pw} – fuel injection beginning, $\alpha_p s$ – combustion beginning

The following methods may be numbered among the most valuable of the methods proposed in the present paper:

- the methodology for determination of thermodynamic parameters in characteristic points of the generalised engine work cycle, its theoretical efficiency, mean theoretical cycle pressure, the value of maximum cycle pressure, and the methods of influencing the values of the indicated quantities, and the methodology for calculations of such basic engine operation indicators as: efficiency, indicated power, unitary consumption of fuel, etc.
- the method for determining the cylinder filling ratio in a more exact manner,
- the methodology for determining the maximum cycle temperature and the working charge temperature at the moment of opening the outlet valve,
- the methodology for determination of the effective coefficient of the heat release during the combustion process,

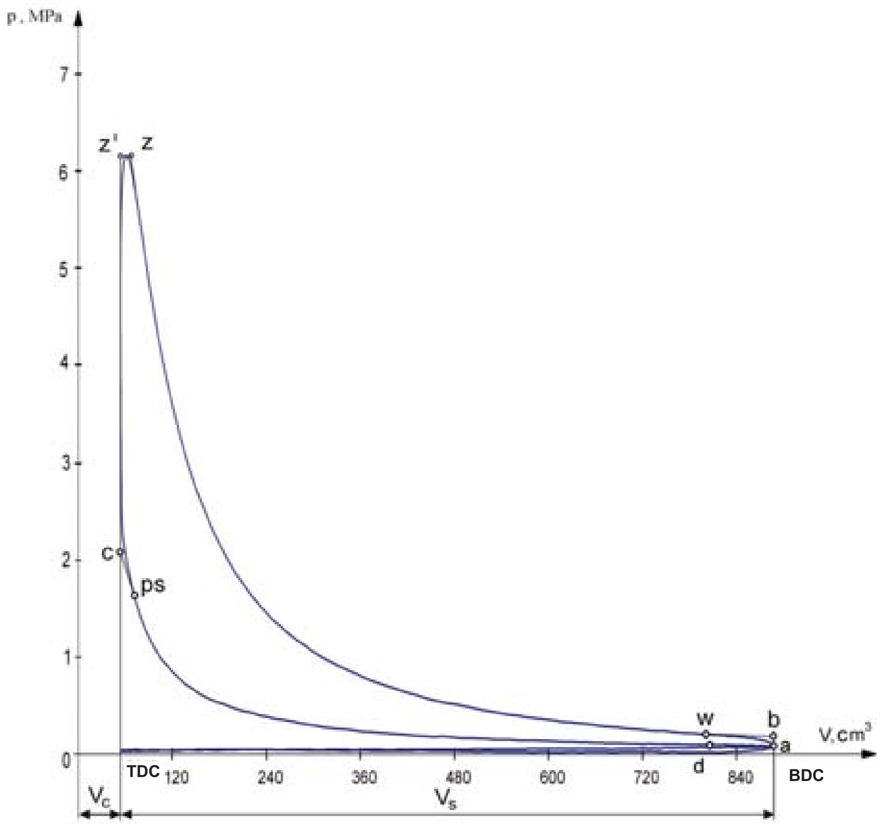

Fig. 5. The effective closed indicator diagram for AD3.152 EDZ-powered engine, working at n=1400 RPM and M_e = 168 Nm

- the way for determining the parameters of the beginning and the end of the combustion process, the self-ignition delay angle, and making the characteristic of the relative quantity of the emitted heat, permitting the evaluation and analysis of the two stages of the combustion process, i.e. the combustion that proceeds according to the kinetic mechanism and the diffusive combustion mechanism,
- the way for determining the quantity of heat transferred by the working charge to the cylinder walls during the combustion process, utilising the effective indicator diagram and the parameters of the identified generalised engine work cycle.

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