HEAT RELEASE CHARACTERISTICS IN COMBUSTION CHAMBER OF CI ENGINE

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Abstract

On the basis of the internal heat balance of CI engine with direct fuel injection, the quantity of heat transported during the combustion process from the combustion zone was determined. The real indicator diagrams are the bases for determination of the heat release characteristics during the combustion process and making up of the internal heat balance. These diagrams were worked out for the engine fuelled with diesel oil for three sets of the injection timing and for work of the engine according to the full load engine characteristics, and the load characteristics taken at the engine speed for the maximum engine torque.

Tests with the three different injection timing were carried out. Tests with the load characteristics of the compression-igniting engine for five values of the power were carried out. The pressure in the cylinder of the engine during combustion process was recorded by means of liquid cooled a piezo-electric quartz sensor with a load amplifier. Characteristics of heat release depended on the composition, temperature and the mol quantity of the working charge in the cylinder during the combustion process. Characteristics of the heat release were determined with the use of the novel calculation program. The program was worked out by authors at the assumption that the process of complete combustion was finished at the moment of the opening of the engine exhaust valve. For purpose of simplification calculation of the quantity of heat lost into the cooling system, the radiation, dissociation and incomplete combustion, were assumed to change during the combustion process in a linear way.

Keywords: *internal combustion engines, diesel engines, combustion processes, heat release*

1. Introduction

The indicator diagram, being effect of thermodynamical, thermochemical, hydroaerodinamical and heat transfer process, occurring in the cylinder is a basic quantitative and qualitative source of information on these. It makes possible to determine: indicator coefficients of the engine, performances of heat release during combustion process, the equilibrium constitution of working factor changing as a function of the crank-shaft angle, „the hard work" of the engine expressed with the dp/dα, etc. Character of the indicator diagram depends first of all from: the character and quality of the fuel atomization during injection process, the quantity and aerodynamic proprieties of the air introduced into the cylinder, etc. These values create quality of the flammable mixture. Quality of the conversion of the chemical energy contained in the fuel on the mechanical energy depends on the properly realized process of the combustion. Fuel consumption, pollution and engine noise depends on this conversion. The heat transfer from gases to the combustion chamber walls is relative to conditions existing in their surroundings, which are very diverse. It is the reason, why calculated heat loss depends on the difference between average temperatures of the wall surface and the surrounding gases and on the average heat transfer coefficient α_g [1]. The selection of the reliable value of the heat transfer coefficient is not easy. Its value depends on such factors as: thermal conductivity, viscosity, density, specific heat of gases, their emissivity, gas temperature, geometrical dimensions of combustion chambers, of flame and of the walls. Determination of the heat release curve during combustion process needs knowledge of the exactly recorded of the pressure diagram, temporary values of the cylinder volume, knowledge of the quantity, composition and gas properties and also reliable model of the heat transfer between the gases, and the walls of in the combustion chamber.

2. Research stand and measuring equipment

An object of research was the four-stroke compression ignition engine with the direct fuel injection into combustion chamber in the piston indicated as PERKINS AD3.152 UR. Basic data of the engine are shown in Tab. 1.

Tab. 1. Basic data of the Perkins AD3.152 UR engine [3]

Perkins AD3.152 UR CI Engine		
Parameter	Unit	Value
Cylinder arrangement	-	line
Cylinder number c	-	3
Injection kind	-	Direct
Cylinder work order	-	1 – 2 – 3
Compression rate, ε	-	16,5
Cylinder bore, D	m	$91.44*10^{-3}$
Piston stroke, S	m	$127*10^{-3}$
Engine cubic capacity, V_{ss}	m^3	$2.502*10^{-3}$
Connecting rod length , l_k	m	$(223.80 – 223.85)*10^3$
Engine rated power, N_e	kW	34.6
Engine speed at rated power, n_N	RPM	2250
Maximum engine torque, M_e	Nm	165.4
Engine speed at maximum engine torque, n_M	RPM	1300-1400
Static ignition advance angle, α_{ww}	°CA	17
Engine speed of idle run, n_{bj}	RPM	750±50

Block scheme of high-frequency quantities measuring-system used in investigated engine is presented in Fig. 1.

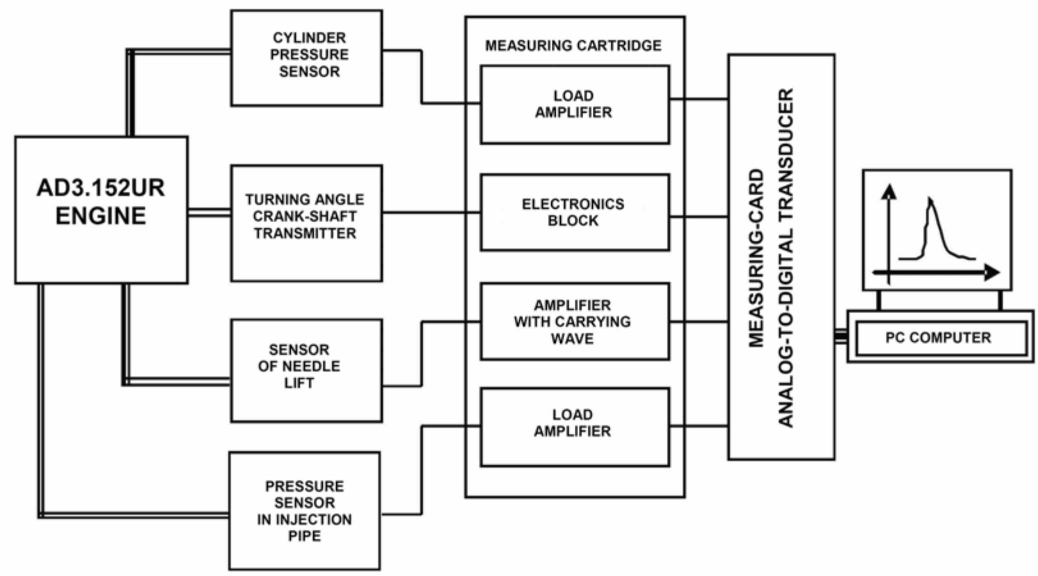

Fig. 1. Block scheme of high-frequency parameters measuring system in compression ignition engine used in the tests

Measurement system presented in Fig. 1 consists of four measuring-lines: the line of the measuring-pressure in the combustion chamber, the line of the measuring-pressure in the injection's installation, the measuring-line lift of the nozzle injector and the decoder of the angle of the crank-shaft.

3. Determination of heat release and heat transfer coefficient

Preparation to the analysis of the experimentally recorded indicator diagram, must take into account its exact location in relation to TDC. Smoothing out of measured pressure is also necessary. Further more, the value exponent of polytropic compression n_1 should be exactly determined. For this purpose the real indicator diagram is introduced in the double logarithm scale $log\,\overline{p} - log\,\overline{V}$ and the linear regression method to appointing equation of straight line illustrating properly processes compression and expansion are used.

The diagram in the double logarithm scale makes possible also to determine beginning and termination of the combustion process, and consequently of the delay period of the self-ignition and the duration of the combustion process.

Characteristics of the heat release is determined based on the real indicator diagram with the use of equation of the first low of thermodynamics and the ideal gas low applied to real gases in the cylinder:

$$\begin{cases} g_c W u dx - \delta Q_{sc} - \delta Q_{nied} - \delta Q_{dys} = dU + pdV, \\ pV = MRT, \end{cases} \tag{1}$$

where:

$$\delta Q_{sc} + \delta Q_{nied} + \delta Q_{dys} = \delta Q_{str} \tag{2}$$

or in a form:

$$\begin{cases} g_c \cdot Wu - Q_{\sum str} = \Delta U + \int\limits_{V_{ps}}^{V_{ow}} pdV, \\ pV = mRT, \end{cases} \tag{3}$$

where:

$Q_{\Sigma str}$ - the total heat losses including heat losses transferred to the walls (that enclosing combustion space) and heat losses related to dissociation and incomplete combustion of fuel.

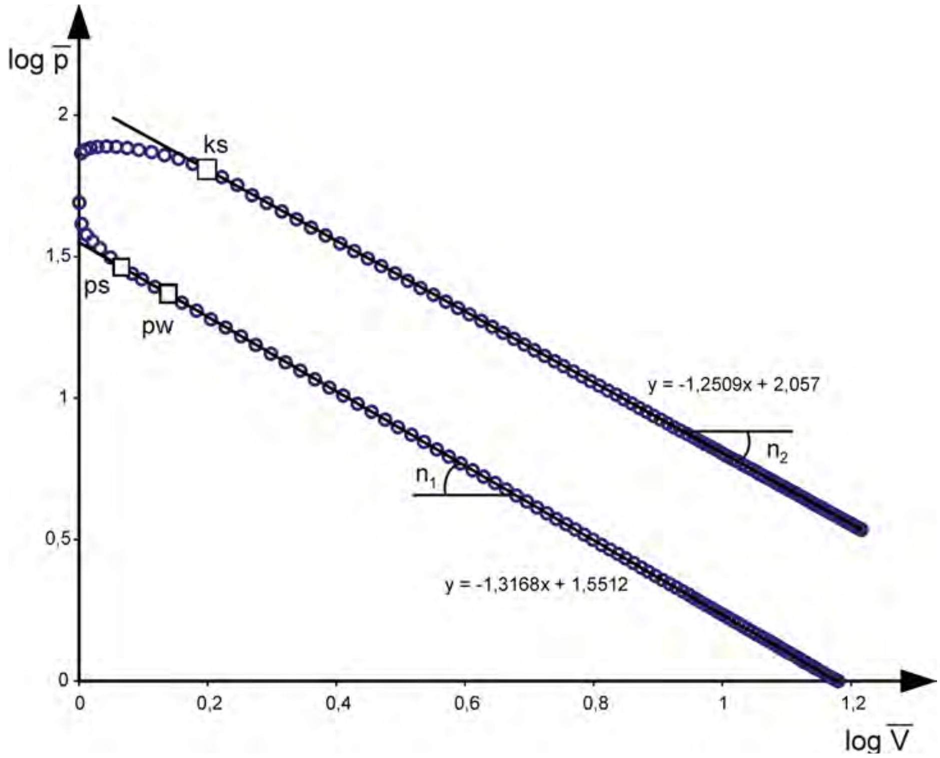

Fig. 2. The indicator diagram of the AD3.152 UR engine working according to the load characteristics at n=1400 [RPM] fed with DF, introduced in the double logarithmic coordinate system, where: n_1 - the exponent of polytropic compression, n_2 - the expression poly-trails of the expansion, p_w - beginning of the fuel injection process, p_s - beginning of the combustion process of the, k_s - end of the combustion process

In view of the fact, that the indicator diagram was determined for the compression ignition engine, characterized with the maximum average temperature of the combustion contained within 1800-2200 K, the calculations of heat losses resulting from dissociation were not taken into account.

The dose of fuel g_c burnt up during one working cycle is determined from the equation:

$$g_c = \frac{G_h}{30 \cdot n \cdot c} \text{ [kg/cycle]}, \tag{4}$$

where:

G_h - fuel consumption per hour, n –engine speed, c - number of cylinders.

Knowing the elementary structure of fuel it is easy to calculate the theoretical quantity of air, M_o, necessary to combustion of 1kg of fuel. Additionally if one knows the excess air coefficient λ, then it is possible to determine moles number of air realizing working cycle of the engine:

$$M1= gc\lambda Mo \text{ [kmol/cycle]}. \tag{5}$$

Dividing the first equation of the system (1) by g_cW_u, one can obtain the equation expressing characteristics of the relative quantity of heat release during combustion process:

$$dx - \delta x_{str} = dx_i\,, \tag{6}$$

where:

$$dx_i = \frac{dU + \mathrm{pdV}}{\mathrm{g_c} \cdot \mathrm{Wu}}, \tag{7}$$

The equation (6) shown in the integral form is:

$$1 - xstr = xi, \tag{8}$$

where x_{str} is relative quantity of the heat lost during combustion process.

In this equation, x_i is an indicator characteristic of heat release of the consumed by internal energy of the working factor and to work accomplishment:

$$x_i = \frac{U_i - U_{ps} + \int_{\alpha_{ps}}^{\alpha} pdV}{g_c Wu}, \tag{9}$$

where:

U_{ps} - an internal energy of the air at the beginning the combustion process, while U_i is current value of the internal energy of the air, calculated from the equation:

$$U_i = M_i \bar{c}_{vi} T_i, \tag{10}$$

where:

Mi - is temporary quantity of kmol of air in the cylinder during the combustion process, determined from the equation:

$$M_i = \beta_x M_{ps}; \quad where: \quad \beta_x = 1 + \frac{\beta_o - 1}{1+\gamma} \cdot x. \tag{11}$$

The temporary value of the molar specific heat of air during the combustion process is determined from the equation:

$$\bar{c}_{vi} = a_i + b_i \cdot T_i, \tag{12}$$

where:

$$a_i = a_{spr}(1-x) + xa_\gamma, \; b_i = b_{spr}(1-x) + xb_\gamma. \tag{13}$$

Values of coefficients of specific heats of the air during the compression process are calculated from the equation:

$$a_{spr} = \frac{a_\lambda + \gamma a_\gamma}{1+\gamma} \quad and \quad b_{spr} = \frac{b_\lambda + \gamma b_\gamma}{1+\gamma}. \tag{14}$$

The value of the specific heat ratio during the combustion process is calculated from the equation:

$$k_i = \frac{\bar{R}}{c_{vi}} + 1. \tag{15}$$

The graphical illustration of the method used to build characteristics of heat release and heat loss during the combustion process is presented in Fig. 3.

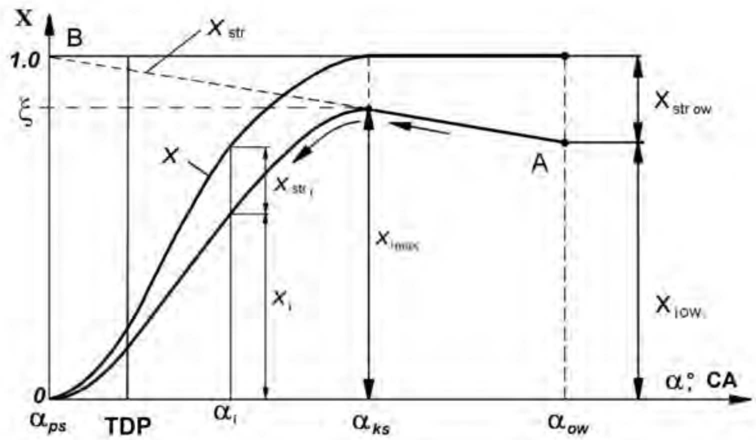

Fig. 3. Characteristics of heat release ratio and heat loss ratio during the combustion process: x, x_i and x_{str}, where: α_i - current value of the angle of the crank-shaft, α_{ps} - beginning of the combustion process, α_{ks} - end of the combustion process , α_{ow} - beginning of the opening of the exhaust valve

The maximum value of x_{imax} is simultaneously a value of the coefficient of the effective utilization of heat during the combustion process ξ. Calculations could be made following to papers [1, 2].

The quantity of the heat lost by the air during the combustion process in interval of the crank-shaft rotation for $\alpha \in \langle \alpha_{ps}, \alpha_{ow} \rangle$ angle is calculated according to the equation:

$$Q_{str} = g_c Wu - \left(\left(U_{ow} - U_{ps} \right) + \int_{V_{ps}}^{V_{ow}} pdV \right). \tag{16}$$

The relative quantity of the lost heat is determined by the equation:

$$x_{str} = 1 - \frac{U_{ow} - U_{ps} + \int_{V_{ps}}^{V_{ow}} pdV}{g_c Wu}. \tag{17}$$

The total heat losses of including: those transferred to walls, losses of incomplete combustion and dissociation and other are determined from the equation:

$$Q_{str} = g_c Wu \left(1 - x_i \right). \tag{18}$$

Subtracting from the value equal for one, we receive the value of the relative quantity indicator emitted heat, the relative quantity of the heat lost during the combustion process, point A (Fig. 2). Assuming linear equation of heat losses from the angle of the crank-shaft, by point A and point B corresponding, we bring on straight line (Fig. 2). This right line characterises relative losses of heat during the combustions process which are determined with the equation:

$$x_{str} = \frac{x_{str\,ow}}{\alpha_{ow} - \alpha_{ps}} \left(\alpha_i - \alpha_{ps} \right), \tag{19}$$

where α_i is variable value of the of the crank-shaft angle.

Preparing the characteristics of the relative quantity of the heat releasing during the combustion process is accomplished according to the method described in [1].

The value of the internal energy of the working factor at the beginning the combustion process of the in the „ps" point and in the moment beginning of the opening of the exhaust-valve „ow" (Fig. 2)" is counted from the below equations:

$$U_{ps} = M_1 \bar{c}_{V_{ps}} T_{ps}, \tag{20}$$

$$U_{ow} = \beta M_1 \bar{c}_{V_{spl}} T_{ow}. \tag{21}$$

Absolute value of heat losses of is constituted from the equation:

$$Q_{str} = g_c Wu\, x_{str\,ow} = g_c Wu \left(1 - x_{iow} \right). \tag{22}$$

The value of the coefficient of heat transfer can be appointed according to many empirical equations, e.g.: Eichelberg, Woschni, Hardenberg and other. In this paper, the value of the coefficient of heat transfer during the combustion process is marked basing on the Newton equation:

$$Q_{SC}(\alpha_i) = \alpha_g \cdot F_{SC}(\alpha_i) \cdot \left(T(\alpha_i) - T_{SC}(\alpha_i) \right) \cdot \Delta t, \tag{23}$$

where:

$\Delta t = \dfrac{\alpha_{i-1} - \alpha_i}{6 \cdot n}$,

α_g - heat transfer coefficient,

F_{SC} - surface area of heat exchanging,

T, T_{SC} - temperature of the working factor and surface heat exchanging, respectively,
α - rotational angle of the crank-shaft,
n - rotational speed of the crank-shaft.

Tab. 2. Values of the heat transfer coefficient during the combustion process and other characteristic values of parameters describing the heat exchange in the cylinder of the AD3.152 UR engine working according to the external speed characteristics at n=1400RPM, fed with diesel oil

No.	α ^{O}CA	x_{sc}	Q_{sc} J	F_i (α) m^2	T_i K	T_{sc} K	ΔT K	$\alpha_g(\alpha)$ W/m^2K
1	352.97	0.000	1.34	0.0180	732.68	571.36	161.32	991.29
2	358.59	0.006	10.69	0.0179	1360.00	571.54	788.46	1757.23
3	364.22	0.011	21.38	0.0179	1773.00	571.49	1201.51	588.83
4	369.84	0.017	32.06	0.0182	2052.00	571.21	1480.79	491.84
5	375.47	0.022	42.75	0.0187	2207.00	570.71	1636.29	442.74
6	381.09	0.028	53.44	0.0194	2243.00	570.05	1672.95	419.51
7	386.72	0.033	64.13	0.0204	2215.00	569.27	1645.73	407.29
8	392.34	0.039	74.82	0.0215	2141.00	568.42	1572.58	401.63
9	397.97	0.044	85.51	0.0228	2051.00	567.55	1483.45	397.17
10	403.59	0.050	96.19	0.0243	1957.00	566.69	1390.31	395.23
11	409.22	0.055	106.88	0.0259	1860.00	565.85	1294.15	394.02
12	414.84	0.061	117.57	0.0275	1772.00	565.07	1206.93	390.68
13	420.47	0.066	128.26	0.0293	1699.00	564.35	1134.65	388.45
14	426.09	0.072	138.95	0.0312	1623.00	563.69	1059.31	383.60
15	431.72	0.077	149.64	0.0330	1558.00	563.09	994.91	379.34
16	437.34	0.083	160.32	0.0349	1493.00	562.56	930.44	376.91
17	442.97	0.088	171.01	0.0368	1458.00	562.08	895.92	371.73
18	448.59	0.094	181.70	0.0386	1398.00	561.66	836.34	370.19
19	454.22	0.099	192.39	0.0404	1351.00	561.28	789.72	371.43
20	459.84	0.105	203.08	0.0421	1320.00	560.95	759.05	367.30
21	465.47	0.110	213.77	0.0437	1295.00	560.67	734.33	361.24
22	471.09	0.116	224.45	0.0452	1256.00	560.41	695.59	361.90
23	476.72	0.121	235.14	0.0466	1239.00	560.19	678.81	360.01
24	482.34	0.127	245.83	0.0479	1216.00	560	656	353.54
25	487.97	0.132	256.52	0.0491	1217.00	559.83	657.17	351.75
26	493.59	0.138	267.21	0.0502	1172.00	559.69	612.31	351.22

After conversion of the equation (23), formula on the calculation of the value of the heat transfer coefficient is received in form:

$$\alpha_g(\alpha_i) = \frac{Q_{SC}(\alpha_i)}{F_{sc}(\alpha_i)\cdot[T(\alpha_i) - T_{sc}(\alpha_i)]}\cdot\frac{6\cdot n}{\alpha_i - \alpha_{i-1}}. \tag{24}$$

4. Calculation results

Calculations of the value of the coefficient heat transfer, α_g, are brought on for the turning angle of the crank-shaft within the range from the beginning of the combustion process to end of the combustion process, with the computational step Δα = 1.4OCA. Calculated values of α_g as well other essential values describing process of heat transfer such as: share of heat, xsc and the quantity of heat, Qsc transferred to sides the combustion chamber, the change of area surface exchanging heat Fi (α), the temperature of the working factor Ti, the temperature of sides the combustion chamber Tsc, and the temperature difference ΔT between the working factor, and sides the combustion chamber are introduced in the Tab. 2.

In the Tab. 2 values of the coefficient of taking over the heat αg(α) and other calculated parameters with the step Δα=5.62OCA are also introduced. Calculations brought over basing on appointed characteristics of heat release prepared for the AD3.152UR engine working according to the external speed characteristics at n=1400RPM and at feeding it diesel oil.

The plot of the change of the value of the heat transfer coefficient α_g during combustion process of the in the AD3.152UR engine operating according to the external speed characteristics , at the rotational speed according to the maximum engine torque and at engine fed with diesel oil are presented in Fig. 4.

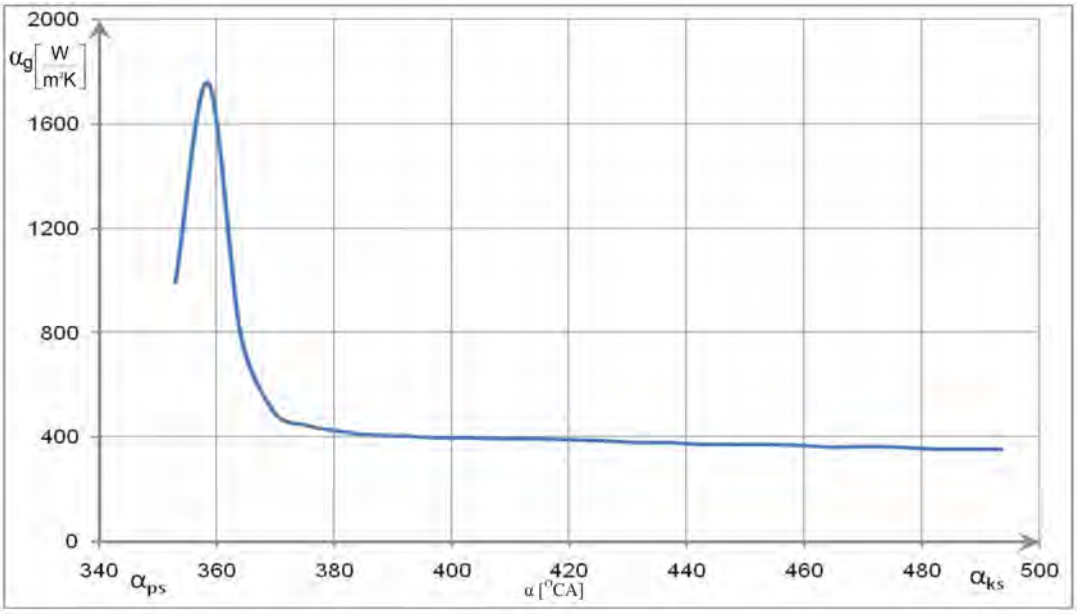

Fig. 4. The course of the change of the heat transfer coefficient α_g during combustion process at the AD3.152UR engine operating according to the external speed characteristics and at the rotational speed according to the maximum torque at n=1400RPM and at the engine fed with diesel oil

5. Conclusions

Most of all difficulties in the calculation of the heat transferred to walls of the combustion chamber causes correct and reliable choice of the heat transfer coefficient. As showed research presented in the paper [10], the error in qualifying of average values of wall temperatures and their surface has an influence on accuracy of calculations of heat losses to walls of the combustion chamber, and consequently on calculated in the paper a value of the heat transfer coefficient. The proposed in the paper method makes possible to mark in simple way the value of the heat transfer coefficient during the combustion process basing on knowledge of the real engine indicator diagram, without necessaries of using to this propose of empirical equations.

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