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HEAT RELEASE DYNAMICS IN THE PROCESS OF COMBUSTION OF HYDROGEN-CONTAINING FUELS

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With the use of the Vibe formula, the heat release dynamics in the process of combustion of stoichiometric and lean ($\phi = 0.8$) propane-hydrogen-air mixtures with a content of molecular hydrogen from 0 to 14% in a cylindrical chamber has been considered. With the aid of the experimental data obtained for the investigated geometry, the relative characteristic of the heat release, the total combustion time, and the index of the combustion character for two variants of the initiator location (lateral and central firing) have been established. It has been shown that these parameters take into account the main features of combustion and have a quite definite physical meaning associated with the physicochemical properties of the fuel and the flame propagation velocity. The heat release rate and its dependence on the time and the position of the flame front have been investigated.

Keywords: *hydrogen combustion, heat release characteristic, index of combustion character, flame propagation velocity.*

Introduction. The changeover to new kinds of fuel calls for comprehensive studies of the process of their combustion under different conditions. One important direction of such studies in the cylinders of internal combustion engines is the elucidation of the laws of the heat release dynamics of the fuel determining the current thermal and mechanical loads on the combustion chamber walls and their coordination with the operation mechanics of the piston group.

The heat release characteristic (coefficient) x is the ratio of the quantity of current heat released in the process of fuel combustion in the engine cylinder Q as a function of the rotation angle of the crank φ (or the combustion time t) to the quantity of available combustion heat Q_{cyc} of the cyclic fuel dose introduced into the cylinder g_{cyc}:

$$x(\varphi) = \frac{Q(\varphi)}{Q_{\text{cyc}}}, \quad x(t) = \frac{Q(t)}{Q_{\text{cyc}}}. \tag{1}$$

Thus, $x = f(\varphi)$ (or $x = f(t)$) is the portion of released heat depending on the parameter φ (or t), respectively.

The process of combustion is usually accompanied by inevitable losses due to the heat removal into the cooled combustion chamber walls, incomplete fuel combustion, and dissociation of combustion products. Therefore, only part of the heat Q_{used} representing the difference between the total available combustion heat Q_{cyc} and the quantity of various losses in the process of combustion Q_{loss}:

$$dQ_{\text{used}} = dQ_{\text{cyc}} - dQ_{\text{loss}} \tag{2}$$

is supplied to the working substance.

In practice, heat release characteristics are obtained by adequate processing of indicator diagrams which in experimental investigations of the working process of the engine are the basic document [1, 2]. They make it possible to estimate and improve the process of fuel combustion in terms of the efficiency of an engine cycle, as well as give a visual idea about the temporal progress of the process. Thus, the heat release characteristics are a bridge connecting

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fuel combustion as a physicochemical phenomenon with its thermodynamical reflection in the working process of the engine.

Mechanisms of Fuel Burnup. Heat released due to the fuel oxidation is used to deliver work and increase the internal energy of the working substance. In investigations of engine cycles, the heat release is determined by calculation-experimental methods based on the numerical solution of the differential equation of the first law of thermodynamics in a form valid for reversible processes in a thermodynamic system whose center of gravity is static [3–8]. In this case,

$$dQ = dU + pdV + dQ_{\mathrm{w}} + dQ_{\mathrm{dis}} + dQ_{\mathrm{inc.c}}\,, \tag{3}$$

where dU is the change in the internal energy of the gas, pdV is the heat equivalent to the work of the gases; dQ_{w} is the heat loss in the wall; dQ_{dis} is the combustion product dissociation heat loss; $dQ_{\mathrm{inc.c}}$ is the heat loss due to incomplete combustion of the fuel.

The heat released in the process of combustion can be expressed in terms of the lowest heat value H_{u} of the fuel fed into cylinder:

$$Q = \chi H_{\mathrm{u}} g\,(\varphi \cup t)\,, \tag{4}$$

where x is the coefficient of chemical incompleteness of combustion.

Upon integration of the equation of the first law of thermodynamics (3) and its substitution into (1) the expression for the heat release characteristic x takes on the form

$$x = \frac{\Delta U + \int\limits_{V_0}^{V} pdV + Q_{\mathrm{w}} + Q_{\mathrm{dis}} + Q_{\mathrm{inc.c}}}{\chi H_{\mathrm{u}} g_{\mathrm{cyc}}} \tag{5}$$

or

$$x = \frac{\Delta U + \int\limits_{V_0}^{V} pdV}{\chi H_{\mathrm{u}} g_{\mathrm{cyc}}} + x_{\mathrm{w}} + x_{\mathrm{dis}} + x_{\mathrm{inc.c}}\,. \tag{6}$$

Here $\Delta U + \int\limits_{V_0}^{V} pdV = Q_{\mathrm{used}}$, $x_{\mathrm{w}} = \dfrac{Q_{\mathrm{w}}}{\chi H_{\mathrm{u}} g_{\mathrm{cyc}}}$, $x_{\mathrm{dis}} = \dfrac{Q_{\mathrm{dis}}}{\chi H_{\mathrm{u}} g_{\mathrm{cyc}}}$, $x_{\mathrm{inc.c}} = \dfrac{Q_{\mathrm{inc.c}}}{\chi H_{\mathrm{u}} g_{\mathrm{cyc}}}$ are the relative quantities of heat losses into the cylinder walls, dissociation losses of combustion products, as well as the incomplete combustion of the fuel, respectively.

On the other hand, dividing both sides of Eq. (4) by the heat $Q_{\mathrm{cyc}} = \chi H_{\mathrm{u}} g_{\mathrm{cyc}}$ released in the process of combustion in one cycle, we get

$$x\,(\varphi \cup t) = \frac{Q\,(\varphi \cup t)}{Q_{\mathrm{cyc}}} = \frac{\chi H_{\mathrm{u}} g\,(\varphi \cup t)}{\chi H_{\mathrm{u}} g_{\mathrm{cyc}}} = \frac{g\,(\varphi \cup t)}{g_{\mathrm{cyc}}}\,. \tag{7}$$

On the left-hand side of expression (7) the relation representing the mass fraction of burnt fuel is written. Thus, physically, the expression for x characterizes the mechanism of fuel burnup and the features of the current conversion of the fuel chemical energy to heat.

The form of the release characteristic is determined by the laws of fuel combustion in the cylinder and the law of movement of the piston. In a forced-ignition engine, the heat release rate depends on the area and velocity of the flame front; therefore, at the start of combustion, while the front area is small, the heat release rate increases slowly. It reaches the maximum value in the second phase of the process of combustion when the major portion of used heat is supplied to the working substance, approximately in the middle of the combustion process (φ_m/φ_c = 0.55–0.65) [9].

In compression-ignition engines, much less fuel is burnt in the second phase, and the maximum of the heat release rate is located much closer to the start of combustion (at the φ_m/φ_c = 0.1–0.2 [4, 8]) than in forced-ignition engines. This is due to the fact that in diesel engines volume combustion (thermal explosion) occurs almost simultaneously throughout the mixture volume. The reasons for such a wide difference between the heat characteristics are the differences between the principles of mixture formation and combustion.

The efficiency of the working process of an engine is determined by not only completeness but also opportuness (rate) of heat release and, accordingly, the degree of expansion of combustion products.

In modern engines, the maximum power is achieved at a release of 50% of combustion heat in the range of up to 10–12$^{\text{o}}$ of the crank shaft rotation angle after the top dead center (TDC), which corresponds to the position of the pressure maximum at about 15$^{\text{o}}$ after the TDC.

Calculation of the Heat Release Characteristic in the Process of Combustion of Hydrogen-Containing Mixtures. The use of hydrogen-containing mixtures can have a strong effect on the physicochemical processes in the engine cylinder. Therefore, in the mathematical description of the working process of the engine burning partially or completely such fuels it is improper to use heat release characteristics typical of traditional kinds of fuels (gasoline, diesel fuel, etc.).

An appropriate solution may be the use of the well-tested semiempirical relations formally describing the heat release in the process of burning hydrocarbons in view of the revised diagnostic variables for the new types of fuels. Such an approximation can be based on the physical notions about the combustion process and a concrete experimental material for a qualitative estimation of the process.

In calculating the engine cycles, various methods based on the semiempirical relations of heat release characteristics presented, e.g., in [3, 6, 10, 12], etc. are used. In both forced-ignition and compression-ignition engines, the universally recognized dependence for a formal description of the heat release x in the cylinder on the dimensionless time t/t_c (where t_c is the total combustion time) is the semiempirical Vibe equation obtained on the basis of the mechanism of chain reactions and generalization of a large body of experimental material [10]:

$$x = 1 - \exp \ln (1 - x_c) \left(\frac{t}{t_c} \right)^{m+1} . \tag{8}$$

Differentiating Eq. (8) with respect to time, we obtain the relative rate of heat release

$$\frac{dx}{dt} = - C\,(m + 1) \frac{t^m}{t_c^{m+1}} \exp C \left(\frac{t}{t_c} \right)^{m+1} . \tag{9}$$

The Vibe equation includes three diagnostic variables characterizing the law of combustion dynamics: one variable (m) characterizing the combustion dynamics qualitatively, and the other two (t_c and x_c), quantitatively. The correct choice of these indices determines the degree of agreement between the calculated and experimental characteristics of the heat release. To this end, one usually employs experimental data on the heat release in similar engines without account for the physical notions about the character of the flame and its propagation velocity in the engine cylinder. Let us attempt to establish the qualitative and quantitative relations of indices x_c, t_c, and m to the real conditions of the combustion process in a closed vessel — an engine cylinder.

In studying the heat release dynamics, of great importance is the value of the ratio t_m/t_c, where t_m is the maximum time of the (relative) rate of heat release. Therefore, it is necessary, in the first place, to evaluate the factors influencing the maximum value of the rate of heat release. The determining factor is the flame velocity in the cylinder, which in the general case with account for the turbulent character of its propagation is related to the rate of heat release in the forced-ignition engine by the relation [1]

$$\frac{dx}{dt} = \frac{u_t \gamma}{V} , \tag{10}$$

where γ is the coefficient taking into account the width of the flame front and the completeness of combustion.

At the very start of motion of the front from the ignition source the flame velocity is close to the rate of laminar combustion (with account for the expansion of combustion products). Then the flame propagation velocity gradually increases. The laws to which the flame propagation obeys differ significantly for individual stages of the process. In the initial phase where the scales of turbulent pulsations l are smaller than the width of the laminar combustion zone δ_{lam} ($l < \delta_{\mathrm{lam}}$), the combustion process obeys mainly the laws of small-scale turbulent combustion whose rate $u_{\mathrm{t.s}}$ should increase proportionally to the ratio of the width of the turbulent small-scale flame $\delta_{\mathrm{t.s}}$ to that of the laminar flame δ_{lam} [11]:

$$\frac{u_{\mathrm{t.s}}}{u_{\mathrm{n}}} = \frac{\delta_{\mathrm{t.s}}}{\delta_{\mathrm{lam}}}, \tag{11}$$

hence

$$u_{\mathrm{t.s}} = u_{\mathrm{n}} \frac{\delta_{\mathrm{t.s}}}{\delta_{\mathrm{lam}}}. \tag{12}$$

The flame velocity in the main phase of combustion, where $l \gg \delta_{\mathrm{lam}}$, obeys the laws of large-scale turbulent combustion and is practically fully determined by the rate of large turbulent pulsations. In this case, the turbulent flame velocity is related to the normal flame velocity by the relation

$$\frac{u_{\mathrm{t}}}{u_{\mathrm{n}}} = \frac{S_{\mathrm{t}}}{S_{\mathrm{lam}}} = \overline{S}, \tag{13}$$

where $\overline{S}$ is the relative surface of the turbulent front of the flame, hence

$$u_{\mathrm{t}} = u_{\mathrm{n}}\overline{S}. \tag{14}$$

Since the main combustion phase is characterized by steady large-scale turbulence of approximately constant size, in expression (10) the coefficient γ taking into account the flame front width varies insignificantly. From this it follows that at V = const the heat release rate reaches the largest value at the maximum rate of turbulent combustion u_{t}. The values of the rates u_{t} will be determined by the normal flame velocity characterized by the rates of chemical reactions and heat and mass transfer in the narrow oxidation zone and by the intensity of the turbulent transfer causing a change in the width δ_{t} and surface S_{t} of the turbulent flame front. The increase in the mass rate of turbulent combustion is largely determined by the increase in the flame front surface S_{t}.

Increasing rapidly with time, the turbulent flame front surface S_{t} and, accordingly, the rate of turbulent combustion reach their maximum values at a certain distance from the ignition sources. Apparently, when the flame front reaches the position of the maxima of the above parameters the rate of heat release also becomes maximal. This fact was noted in [12]. Then, to determine the parameters of the Vibe formula, it suffices to find the maximum position of the rate of heat release t_{m} whose value is easy to calculate knowing the quantity of heat released in the period from the onset of combustion to the point t_{m} or the values of the parameters t_{c} and m. These parameters are related to each other by the relation [10]

$$\frac{t_{\mathrm{m}}}{t_{\mathrm{c}}} = \left[\frac{m}{-\ln(1 - x_{\mathrm{c}})(m + 1)}\right]^{\frac{1}{m+1}}. \tag{15}$$

At the maximum rate of heat release in view of formula (15) the Vibe equation for any kind of fuel takes on the form

$$x_{\mathrm{m}} = 1 - \exp\left[\ln(1 - x_{\mathrm{c}})\left(\frac{t_{\mathrm{m}}}{t_{\mathrm{c}}}\right)^{m+1}\right] = 1 - \exp\left(-\frac{m}{m + 1}\right). \tag{16}$$

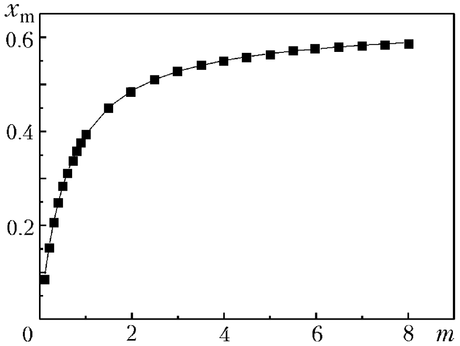

Fig. 1. Heat release characteristic as a function of the index of the character of combustion *m*.

According to formula (16), the heat release characteristic x_m (at the moment the maximum rate of heat release is attained) is a function of only the value of *m*, the form of whose dependence is shown in Fig. 1. The index of the combustion character *m* increases with increasing fraction of burnt fuel x_m corresponding to the maximum rate of heat release. In the range $x_m < 0.4$ (which corresponds to $m < 1$), with increasing parameter x_m the index *m* increases slowly. Beginning with $x_m = 0.5$, with increasing *m* the parameter x_m increases slowly.

Thus, the fraction of the fuel x_m that has reacted by the moment the maximum rate of heat release was attained is determined in the case of using the semiempirical expression (8) proposed in [10] and the assumptions made only by the value of the index of the combustion character *m* depending mainly on the physicochemical factors (rates of chemical reactions and diffusion mass transfer), which is confirmed by the investigations of [13–15].

Based on the analysis of the process of heat release in the engine cylinder, let us attempt to establish the laws of heat release dynamics in the process of combustion of various alternative motor fuels in a model chamber representing a closed cylindrical vessel of radius 40 mm and height 32 mm with two variants of location of the ignition initiator (lateral and central firing).

Let us estimate quantitatively the parameters of the Vibe equation t_m, x_c, and *m*. To find the time t_m corresponding to the maximum rate of heat release, it is necessary to know the above rate which, with account for the features of combustion in a closed volume, can be determined from the expression

$$\frac{dQ}{dt} = u_n F \left(\frac{p}{p_0}\right)^{\frac{1}{k}} h \, . \tag{17}$$

Analysis of formula (17) shows that the rate of heat release reaches its maximum at the completing stage of combustion. The parameters entering into it were determined from the following considerations.

In the process of combustion in a closed volume, combustion products compress the fresh mixture. The linear velocity of flame propagation *u* in the vessel is composed of two components:

$$u = u_n + u_{exp} \, , \tag{18}$$

where u_{exp} is the velocity of motion of the flame front due to the expansion of combustion products.

As the flame front moves through the chamber, the change in the above components of the linear velocity has the opposite character: u_n continuously increases and u_{exp} decreases. By the end of the process the flame velocity is essentially determined by the value of the normal velocity of flame propagation u_n since it markedly increases while the velocity u_{exp}, because of the decrease in the density ratio between the fresh mixture and combustion products, becomes so much lower than u_n that it can be neglected. Thus, at the final stage of combustion $u_n \approx u$.

The values of the linear velocity *u* and pressure *p* were determined experimentally with the use of the method of flame propagation photorecording [16]. The surface area of the flame was calculated on the basis of experimental data with the aid of mathematical analysis [17].

TABLE 1. Investigated Combustible Mixtures

Mixture number	Initial pressure, MPa	Initial fuel	Chemical composition of the mixture	Relation (15)
Lateral firing, ϕ = 1				
1	0.108	Propane	$C_3H_8 + 5O_2 + 18.81N_2$	$\frac{t_m}{t_c} = [\frac{m}{2.996(m+1)}]^{\frac{1}{m+1}}$
2	0.491			
3	0.108	Propane + 4% of hydrogen (in air)	$C_3H_8 + 1.1H_2 + 5.54O_2 + 20.85N_2$	$\frac{t_m}{t_c} = [\frac{m}{3.539(m+1)}]^{\frac{1}{m+1}}$
4	0.491			
5	0.108	Propane conversion products	$CO + 1.33H_2 + 1.17O_2 + 6.27N_2$	$\frac{t_m}{t_c} = [\frac{m}{3.606(m+1)}]^{\frac{1}{m+1}}$
6	0.491			
Lateral firing, ϕ = 0.8				
7	0.098	Propane + 2% of hydrogen (in air)	$0.8C_3H_8 + 0.51H_2 + 5.26O_2 + 19.76N_2$	$\frac{t_m}{t_c} = [\frac{m}{3.311(m+1)}]^{\frac{1}{m+1}}$
8	0.196			
9	0.491			
10	0.491	Propane conversion products	$0.8(CO + 1.33H_2 + 1.88\ N_2) + 1.17O_2 + 4.39N_2$	$\frac{t_m}{t_c} = [\frac{m}{3.606(m+1)}]^{\frac{1}{m+1}}$
Central firing, ϕ = 0.8				
11	0.098	Propane + 2% of hydrogen (in air)	$0.8C_3H_8 + 0.51H_2 + 5.26O_2 + 19.76N_2$	$\frac{t_m}{t_c} = [\frac{m}{3.311(m+1)}]^{\frac{1}{m+1}}$
12	0.196			
13	0.491			
14	0.098	Propane conversion products	$0.8(CO + 1.33H_2 + 1.88\ N_2) + 1.17O_2 + 4.39N_2$	$\frac{t_m}{t_c} = [\frac{m}{3.606(m+1)}]^{\frac{1}{m+1}}$
15	0.196			
16	0.491			

TABLE 2. Values of Parameters in the Vibe Equation for the Investigated Combustible Mixtures

Combustible mixture	P_0, MPa															
	0.098				0.108				0.196				0.491			
	m	t_m, msec	R_m, mm	t_c, msec	m	t_m, msec	R_m, mm	t_c, msec	m	t_m, msec	R_m, mm	t_c, msec	m	t_m, msec	R_m, mm	t_c, msec
1, 2	–	–	–	–	2.38	25.7	65.3	39.4	–	–	–	–	2.90	33.4	65.3	47.8
3, 4	–	–	–	–	3.63	21.1	66.8	29.2	–	–	–	–	4.08	27.9	66.4	37.3
5, 6	–	–	–	–	4.27	12.8	64.6	17.0	–	–	–	–	3.41	12.3	64.4	17.4
7–9	2.56	33.3	64.2	51.1	–	–	–	–	2.69	37.9	64.7	57.1	2.09	43.3	63.8	72.4
10	–	–	–	–	–	–	–	–	–	–	–	–	3.55	17.9	64.4	25.0
11–13	2.68	21.1	35.8	31.8	–	–	–	–	3.01	25.6	36.7	37.1	3.73	34.6	38.1	46.9
14–16	8.72	15.1	38.5	17.4	–	–	–	–	6.31	14.0	38.4	17.0	4.02	13.0	37.3	17.6

According to Eq. (1), the relative quantity of heat released at the end of combustion should be equal to one, i.e., $x_c = 1$. In real engines, however, this condition is not fulfilled since combustion continues, though insignificantly, in the process of expansion. Therefore, we take the x_c values with account for this fact. For hydrocarbon fuels, a good agreement with the real heat release characteristic is obtained at $x_c = 0.950$ [7, 9]. Thus, for propane modeling a hydrocarbon fuel the coefficient $C = \ln(1 - x_c) = \ln(1 - 0.95) = -2.996$. For hydrogen, with account for the features of its combustion, let us assume $x_c = 0.999$, then $C = \ln(1 - x_c) = \ln(1 - 0.999) = -6.908$. The x_c values for the other investigated fuels were determined proportionally to the concentration of molecular hydrogen in the fuel composition. Then formula (15) for the investigated mixtures, independent of the variant of ignition initiator location, acquires the form shown in the last column of Table 1.

For the hydrogen-air mixture of stoichiometric composition with 1% of methane

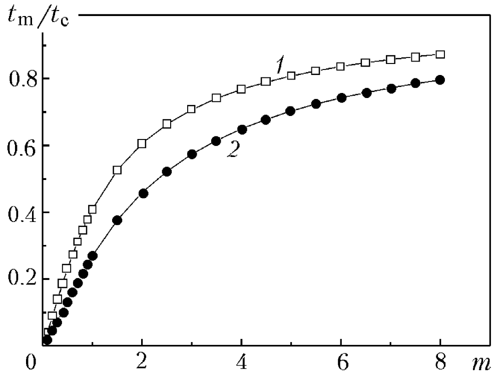

Fig. 2. Coordinate t_m/t_c of the maximum rate of heat release as a function of the index of the character of combustion m: 1) for the propane-air mixture; 2) for the hydrogen-air mixture.

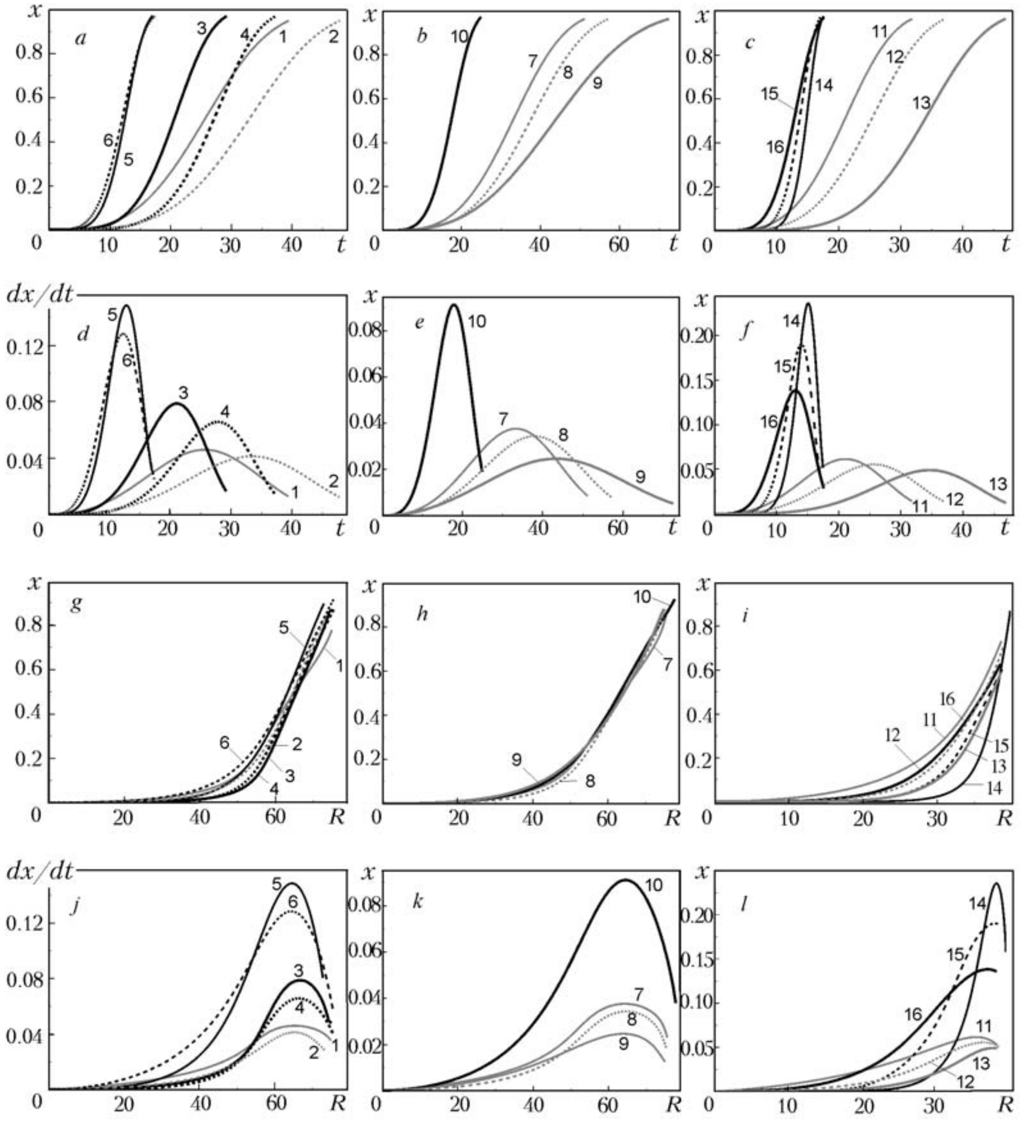

Fig. 3. Heat release characteristic x and relative rate of heat release dx/dt as a function of time t and the flame radius R for the investigated combustible mixtures (1–16, mixture numbers). t, msec; R, mm.

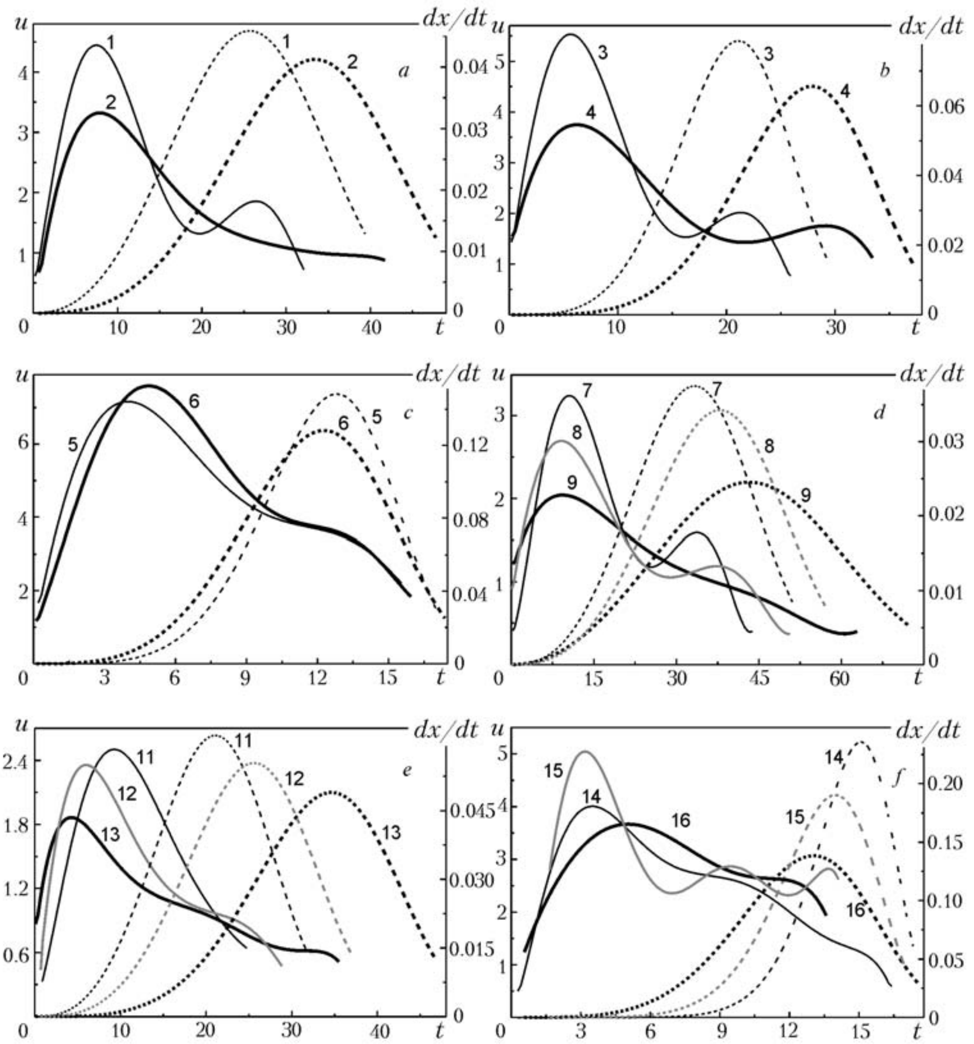

Fig. 4. Curves of change in the flame velocity *u* (solid lines) and in the relative rate of heat release *dx/dt* (dashed lines) depending on the time *t* for the investigated combustible mixtures (1–9, 11–16, mixture numbers). *u*, m/sec; *t*, msec.

$$\frac{t_{\mathrm{m}}}{t_{\mathrm{c}}}=\left[\frac{m}{6.908\,(m+1)}\right]^{\frac{1}{m+1}}. \tag{19}$$

Substituting the t_{m} and t_{c} values and solving the obtained equations by the root separation method, we obtain the index values of the combustion character *m*. The values of the parameters of the Vibe equation x_{c}, t_{m}, t_{c}, and *m* for the investigated mixtures are given in Table 2. As an example, we give the Vibe equation for mixture 1:

$$x=1-\exp\left[-2.996\left(\frac{t}{39.4}\right)^{3.38}\right], \tag{20}$$

$$\frac{dx}{dt}=\frac{t^{2.38}}{24\,594}\exp\left[-2.996\left(\frac{t}{39.4}\right)^{3.38}\right]. \tag{21}$$

The left-hand side of the formulas from Table 1 and expression (19) gives the quantity t_m/t_c depending, in the physical sense, on the propagation velocity and surface of the flame front, and the right-hand side — a function of the parameter *m* alone. The dependences of the dimensionless time t_m/t_c on the parameter *m* for the propane-air and hydrogen-air mixtures are shown in Fig. 2. In the range of *m* values of interest to us (see Table 2) these functions for the investigated mixtures are close to linear ones. Note that at one and the same value of the parameter *m* propane will have a larger value of the coordinate of the maximum rate of heat release t_m/t_c than hydrogen, i.e., the maximum of this rate in propane is farther from the start of the combustion process.

From Table 2 it is seen that the coordinate t_m/t_c of the maximum rate of heat release has the largest value for mixture 14 and the minimal value for mixture 9. In the case of central firing, the value of the dimensionless coordinate t_m/t_c of the above-mentioned maximum for all investigated variants is somewhat higher than in the case of lateral firing.

Figure 3a–f shows the functions $x = f(t)$ and $dx/dt = f(t)$, and Fig. 3g–l gives their dependences on the flame front radius *R* for the investigated mixtures. Analysis of the curves obtained in view formula (15) shows that the smaller the index *m,* the steeper the heat release characteristic and the closer the maximum position of the rate of heat release to the moment of ignition and its nucleation site. Mixture 9 has the minimum, and mixture 14 the maximum, rate of heat release. Comparing the combustion of the investigated kinds of combustible mixtures at a different location of the ignition source, it can be noted that in the case of central firing the value of the maximum rate of heat release is larger than in the case of lateral firing by a factor of 1.6–2 in mixtures of propane with 2% of H_2 (depending on p_0) and by a factor of 1.5 in mixtures of conversion products and is observed closer to the moment of completion of the process, e.g., the coordinates t_m/t_c of the maximum rate of heat release in mixture 9 at lateral firing and in mixture 13 at central firing are equal to 0.60 and 0.74, respectively.

Comparison of the dynamics of the change in the flame propagation velocity *u* and in the relative rate of heat release *dx/dt* (Fig. 4) shows that the heat release maxima are delayed significantly from the maxima of the flame velocities in both time and space (i.e., they are observed at a larger distance from the initiation source). This is mainly due to the fact that in the region of maximum velocities of flame propagation the adiabatic compression of the fresh mixture is not yet significant. Therefore, first, the normal rate of combustion here is lower and markedly increases in a constant-volume chamber upon receding from the ignition source [18] and, second, the volume heat content of the fresh mixture here is much lower than in the mixture multiply compressed by combustion products at the final stage of combustion.

CONCLUSIONS

1. The heat release characteristic gives an idea about the temporal progress of the process and the losses due to the incompleteness of combustion, the heat transfer into the cylinder walls, and the dissociation of combustion products, and permits estimating and improving the process of combustion in terms of the thermodynamic efficiency of the engine cycle.

2. It has been shown that the mechanism of fuel burnup can be described by empirical equations taking into account the qualitative characteristics and features of the combustion process for each kind of fuel, e.g., by the semiempirical Vibe equation.

3. On the basis of the analysis of the relation of the heat release characteristic to the mixture composition and the process of flame propagation in the case of combustion in a closed vessel, we have determined the parameters x_c, t_c, and *m* entering into the semiempirical Vibe equation for air mixtures of propane, propane with 2% of H_2, propane with 4% of H_2, and propane conversion products. They take into account the features of the process of fuel combustion and are associated with the physicochemical properties of the fuel and the flame front velocity.

4. It has been shown that the smaller the index of the combustion character *m*, the steeper the heat release characteristic and the closer the location of the maximum of the rate of heat release to the moment of ignition at its site.

5. The results of the investigation make it possible to establish the dynamics of heat release and calculate the parameters determining the mechanism of fuel burnup: the fraction of burnt fuel and the absolute and relative rates of heat release.

6. The proposed method for calculating the process of heat release is fairly universal and can be used in modeling cycles of engines burning alternative kinds of fuel.

NOTATION

F, surface area of the flame front; *g*, quantity of burnt fuel; *H*, fuel combustion heat; *h*, mixture combustion heat; *k*, adiabatic index; *l*, scale of turbulent pulsations; *m*, index of the combustion character; *p*, pressure; *Q*, current combustion heat; *R*, radius of the flame front; *S*, surface area of the flame front; *t*, time; *U*, internal energy; *u*, flame propagation velocity; *V*, current volume of the combustion chamber; *x*, heat release characteristic (coefficient); χ, coefficient of chemical incompleteness of combustion; ϕ, excess fuel coefficient (stoichiometric coefficient); φ, rotation angle of the crank; γ, coefficient taking into account the flame front width and completeness of combustion. Subscripts: 0, initial; u, bottom; w, wall; c, parameter at the end of combustion; dis, dissociation; used, used; lam, laminar; m, maximum; n, normal; inc.c, incomplete combustion; loss, losses; exp, parameter of expansion of combustion products; t, turbulent; t.s, turbulent small-scale; cyc, parameter of the cyclic dose.

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