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TWO-TEMPERATURE MODEL OF GAS COMBUSTION IN A MODEL BURN DEVICE OF THE CYLINDRICAL FORM

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The two-temperature model of gas combustion in a porous body of cylindrical heat-generator has been offered and numerically analyzed. Heat exchange between solid frame and gas; burning products interaction with heat-exchanger; distinction of diffusion speeds and heat conductivity in the gas phase are considered in the model. The influence of model parameters on characteristics of stationary modes of gas combustion for various heat exchange conditions of porous torch with heat exchanger is investigated. The results of numerical research do not contradict with observable laws that indicates the opportunity of model use for statement and solving the problem of work optimization of the real burning device.

Introduction

The phenomenon of combustion front distribution in porous media at gas filtration attracts the researchers' attention. The academic interest to this group of systems occurred as an answer to the practice requirements including the filtration combustion processes into flow charts of different production. Such large-scale industrial processes as blast-furnace iron smelting, ores burning and agglomeration, catalyst regeneration by the method of carbon burning, oil production by interbedding combustion and others are referred to the objects of filtration combustion.

The process of distributing the region of gas-phase heat-producing reaction in the inert porous medium at filtration supply of gaseous agents to the chemical conversion zone is understood under gas filtration combustion [1]. Such processes represent a kind of heterogeneous combustion as a result of active participation of two phases – solid porous medium and reacting gas – in the mechanism of wave propagation and have the important scientific and practical value. The occurrence of two phases predetermines the multi-parameter of the processes, variation of phase interactions, occurrence of filtration and other effects of heterogeneity. As a result of interaction of various physical processes the multiple stationary and non-stationary thermal combustion modes, different conditions of transformation mode occurrence, combustion waves with specific structure, properties and propagation mechanism are implemented [2, 3].

One of the possible practical applications of filtration combustion refers directly to the development of the environmentally safe porous burners functioning on poor mixtures and providing gas fuel economy; practically full gas combustion in porous body volume and high efficiency.

It is necessary to study possible modes of gas combustion at process parameters variation for the existing burners operation optimization. In the experimental investigation the parameters variation in the wide range of their change is rather difficult. Therefore, mathematical simulation is used for studying combustion modes. The currently known theoretical works on gas filtration combustion are restricted by analytical survey of the experimental data, physical property description and problem statement [4]; or by single private designs on the basis of rather complex models and algorithms [5–7], or by conditions implemented in the laboratory experiment [8].

The model of gas combustion in porous cylindrical burner, the geometry and properties of which correspond to the burner described in [9] in one-dimensional two-temperature approximation is proposed and studied below.

Mathematical statement of the problem

Let us suppose that the burner representing the complete cylinder made of the material with specified density ε, has large sizes: the specified inner R_1 and outer R_2 radii so that the change of gas density ρ_g at thickness of burner working section $R_1 \le r \le R_2$ may be neglected. Fuel gas gets into inner area of cylinder and then it is redistributed by special devices so that the rate of its arrival into porous body V_g along the whole length of the burner (along cylinder) was almost the same. According to Darcy law

$$V_g = -k_f \nabla P,$$

where k_f is the filtration factor; P is the pressure. At specified pressure drop ∇P at gas input into porous body and at output of it at a first approximation gas velocity may be considered constant. The continuity condition according to which and subject to V_g=const we have $\rho_g \sim R_1 \cdot \rho_{g,1}/r$ where $\rho_{g,1}$ is the gas density at the input into porous body, will be fulfilled more strictly in stationary conditions in gas. Gas pressure in pores and its temperature are exactly connected with the state equation

$$P = \rho_g R T_g m_g^{-1},$$

where m_g is the molar mass of agent and reaction product mixture, R is the universal gas constant J/(mole·K).

As a rule, studying the models of filtration combustion one is aimed at determining combustion rate, pressure in gas and phase temperature depending on gas feed rate and conditions of interphasic heat exchange [1]. In this case the combustion rate is determined for infinitely great gas volume at transition to the coordinate system connected with the advancing reaction front.

From practical point of view the stationary combustion modes in a burner unit of finite size is of interest. Such modes are implemented at achievement of stationary operation condition by the burner unit. To determine the combustion rate by traditional method is of no sense in this case.

The mathematical statement of stationary problem in cylindrical coordinate system includes the heat conduction equation for gas and solid body and diffusion equation with convective summands and sources of heat and mass owing to chemical reaction:

$$V_g \frac{dT_g}{dr} = \kappa_g \frac{1}{r}\frac{d}{dr}\left(r\frac{dT_g}{dr}\right) - \frac{\alpha}{c_g\rho_g}\frac{1}{\varepsilon}(T_g - T_s) + \frac{Q_0}{c_g\rho_g} k\cdot\eta^n \exp\left(-\frac{E_a}{RT_g}\right); \quad (1)$$

$$\kappa_s \frac{1}{r}\frac{d}{dr}\left(r\frac{dT_s}{dr}\right) + \frac{\alpha}{c_s\rho_s}\frac{1}{1-\varepsilon}(T_g - T_s) = 0; \quad (2)$$

$$V_g \frac{d\eta}{dr} = D\frac{1}{r}\frac{d}{dr}\left(r\frac{d\eta}{dr}\right) - k\cdot\eta^n \exp\left(-\frac{E_a}{RT_g}\right), \quad (3$$

where T_g is the gas mixture temperature, K; T_s is the porous frame temperature, K; t is the time, s; r is the space coordinate, m; κ_g, κ_s are the effective factors of gas and solid body thermal diffusivity respectively m^2/s; α is the coefficient of interphasic heat exchange, W/(K·m^2); c_g, c_s is the specific gas thermal capacity at constant volume and solid frame heat capacity, J/(K·kg); ρ_g, ρ_s are the gas and solid frame densities, kg/m^3; Q_0 is the thermal effect of overall reaction in gas phase, J/m^3; k is the constant of the reaction rate, s^{-1}; n is the reaction order; E_a is the activation energy, J/mole; η is the agent concentration (mass fraction); $(1-\kappa)$ is the conversion degree, V_g is the gas velocity, m/s; D is the diffusion coefficient, m^2/s, which is considered to be different from the coefficient of thermal conductivity κ_g, $D\neq\kappa_g$ in comparison with the known filtration combustion and gas combustion models.

The equation system (1)–(3) is closed by the boundary conditions at inner ($r=R_1$) and outer ($r=R_2$) burner surfaces. Let us use the temperature constancy condition as a boundary condition at inner surface; it equals to cold gas temperature T_0, and conversion degree equal to zero (or agent concentration equal a unit), i.e.

$$r = R_1: \quad T_g = T_s = T_0, \quad \eta = 1; \quad (4)$$

$$r = R_2: T_g = T_{g,1}; \quad \frac{dT_s}{dr} = 0; \quad \eta = \eta_b, \quad (5)$$

where $T_{g,1}$ is the gas temperature at the output of porous body (or the heat exchanger temperature); η_b is the unexpended agent portion.

«Combustion velocity» is determined in the following way. Let us suppose that the continuity condition is fulfilled in the reaction front the position of which $r=r_f$ is unknown beforehand; it is written in the following way

$$\left(\lambda_g \frac{dT_g}{dr}\right)_{r=r_f+0} - \left(\lambda_g \frac{dT_g}{dr}\right)_{r=r_f-0} = Q_0 u_T, \quad (6)$$

where u_T is the «combustion velocity» or burning rate, m/s. It is obvious that gas temperature to the right and to the left from $r=r_f$ is the same.

Then u may be calculated from solution of the problem (1)–(5), where the coordinate of maximum temperature in gas is taken as the reaction front coefficient. If maximum temperature is on the outer surface of the burner then $[\lambda_g(dT_g/dr)]_{\gamma=\gamma_{f+0}}=0$.

More general problem definition given below takes into account the solid frame heat exchange with heat exchanger by radiation by Stefan-Boltzmann law in the condition (5).

Dimensionless parameters

Let us pass on dimentionless variables for carrying out the detailed parametric study allowing determining possible modes of stationary combustion

$$\theta_1 = \frac{T_g - T_0}{T_* - T_0}, \quad \theta_2 = \frac{T_s - T_0}{T_* - T_0}, \quad x = \frac{r}{R_2},$$

where $T_*=T_0+Q_0/c_g\rho_g=T_{g,1}$ is the typical temperature scale (adiabatic temperature of gas combustion). Then the problem (1)-(5) takes the form:

$$w\delta \frac{d\theta_1}{dx} = \frac{1}{x}\frac{d}{dx}\left(x\frac{d\theta_1}{dx}\right) - \frac{\text{Bi}\delta}{\varepsilon}(\theta_1 - \theta_2) + \delta\varphi(\theta_1, \eta); \quad (7)$$

$$\frac{1}{x}\frac{d}{dx}\left(x\frac{d\theta_2}{dx}\right) + \frac{\text{Bi}\,\delta}{K_\lambda}\frac{1}{1-\varepsilon}(\theta_1 - \theta_2) = 0; \quad (8)$$

$$w\delta \frac{d\eta}{dx} = \text{Le}\frac{1}{x}\frac{d}{dx}\left(x\frac{d\eta}{dx}\right) - \delta\varphi(\theta_1, \eta); \quad (9)$$

$$x = x_1: \quad \theta_1 = 0, \quad \theta_2 = 0, \quad \eta = 1; \quad (10)$$

$$x = 1: \quad \theta_1 = 1, \quad \frac{d\theta_2}{dx} = 0, \quad \eta = \eta_e, \quad (11)$$

where $\varphi(\eta,\theta)=\eta^n\cdot\exp((\theta-1)/(\theta+(1-\sigma)/\sigma)/\beta)$; $\delta=R_2^2/(\kappa t_*)$ – is the parameter of Frank-Kamenetskiy (the ratio of outer radius to heat penetration size which is formed for some characteristic time $t_*=k_0^{-1}\exp(1/\beta)$ the time of chemical conversion at $T=T_*$); $w=t_*V/R_2$ is the dimensionless gas speed; $\text{Bi}=\alpha t_*/(c_g\rho_g)$ is the parameter Bio; $\varphi(\eta,\theta)$ is the function of chemical heat generation; $\text{Le}=D/\kappa_g$ is the Lewis number; $\sigma=(T_*-T_0)/T_*$ is the small dimensionless parameter; $x_1=R_1/R_2<1$ is the inner dimensionless radius of cylinder; $K_1=\lambda_s/\lambda_g$; $\beta=RT_*/E_a$ is the parameter characterizing sensitivity of the reaction rate to temperature change, $\sigma=(T_*-T_0)/T_*\equiv\beta\theta_0$, $\theta_0=(T_*-T_0)E/(RT_*^2)$ is the temperature head or Zeldovich number.

Condition (6) now looks like this

$$\left(\frac{d\theta_1}{dx}\right)_{x_{f+0}} - \left(\frac{d\theta_1}{dx}\right)_{x_{f-0}} = \sqrt{\delta}\, w_b, \quad (12)$$

where $w_b=u_T t_*/R_2$, $x_f=r_f/r$.

Stationary problem (8)-(11) is numerically solved by the sweep method by the following algorithm: at first the temperature distribution and conversion degrees (θ_1,η_1) are determined, using «testing» function of heat generation $\varphi(\theta,\eta)$; then the following approximation (θ_2,η_2) is found out, computing the heat generation function by (θ_1,η_1). The algorithm is repeated till the moment when the mean-square deviation of two approximations by temperature and conversion degree is less 1 %.

Gas and solid frame temperature fields, agent concentration, as well as the width of chemical reaction zone (reaction zone coordinate ξ_η was determined by the value of concentration η<0,99) and «combustion rate» w_b depending on model parameters were defined in calculations. Further as the text goes the quotes are omitted.

Supposing that solid frame is made of Al_2O_3+Fe+Cr, and gas is a mixture of 10 % of methane and 90 % of air let us estimate the dimensionless parameters included into model. According to [10, 11], we have: c_s=1250 J/(kg·K); c_g=2600 J/(kg·K); ρ_s=3750 kg/m^3; ρ_g=0,717 kg/m^3, λ_s=8 W/(m·K), λ_g=0,0821 W/(m·K); E_a=103800 J/mole; Q=1,26·10^6 J/m^3; R_1=0,225 m; R_2=0,3 m; D_g=0,2717·10^{-4} m^2/s; k_0=10^{-9} s^{-1}; T_0=300 K; V_g=0,01...2 m/s, α=82.1...10^4 W/(m^2·K). As a result we obtain t_*=5·10^{-3} s; T_*=900 K; K_λ=97,76; σ=0,66; Le=0,6175; β=0,1; Bi=10^{-5}...0,2; δ=4·10^5; w=0,001...0,2.

Combustion behavior in conditions of low thermal exchange of gas and external surface

At numerical calculation of the problem (7)–(11) for a set of parameters characterizing [9], and at low heat exchange between gas and solid frame (Bi<<1) it was determined that within the frame of one model at variation of gas feed rate two different modes of combustion may be realized (Fig. 1). The first one is characterized by practically linear temperature distribution (Fig. 1, *a*) and gas concentration (Fig. 1, *b*) from inner to outer radius of the burner working section. In this case the whole working section may be considered to be the reaction zone. Such mode is not of practical interest. The second mode is observed at w>0,05 and characterized by the reaction zone adjoining the outer surface of the burner (curves 4–7, Fig. 1).

Solid frame is not practically warmed-up due to the inner heat exchange so we do not have to speak about heat exchange with the environment by radiation: solid frame temperature at this set of parameters in the conditions of outer surface (11) does not increase (2...7)·10^{-6}.

Fig. 2, *a*, illustrates the change of the reaction zone boundary coordinate at variation of different parameters at low heat exchange with the frame. In a certain region of changing the gas feeding parameters the boundary position ξ_η changes considerably then the dependence $\xi_\eta(w)$ does not change qualitatively.

Increasing δ the velocity range w where the combustion behavior interesting from practical point of view is realized, is extended. The combustion velocity w_b, determined according to the condition (12) increases at growth of w and the burner radius δ (Fig. 2, *b*).

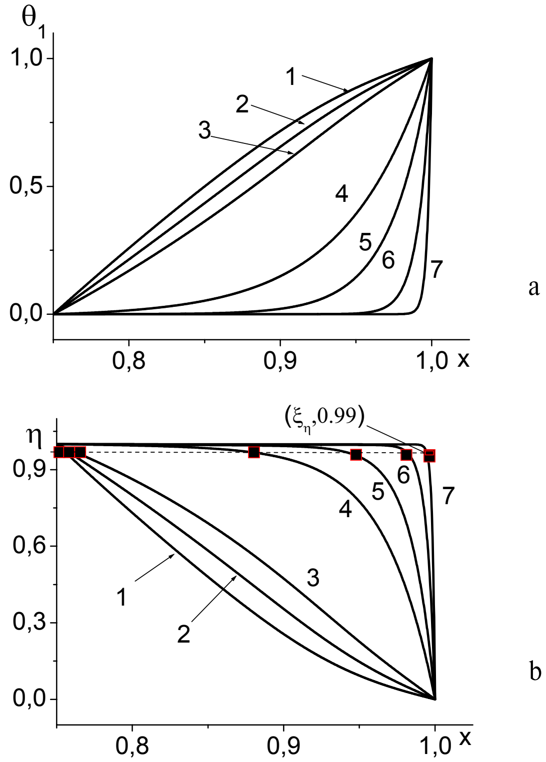

Fig. 1. *Distribution of: a) gas temperature; b) agent concentration along the burner radius at different gas feed rates and* x_1*=0,75;* β*=0,1;* K_λ*=97,56;* ε*=0,5;* σ*=0,66;* η_b*=0; Le=0,6175;* δ*=300; Bi=3·10*$^{-5}$*. Gas feed rate w: 1) 3,75·10*$^{-4}$*; 2) 0,005; 3) 0,01; 4) 0,05; 5) 0,1; 6) 0,3; 7) 1*

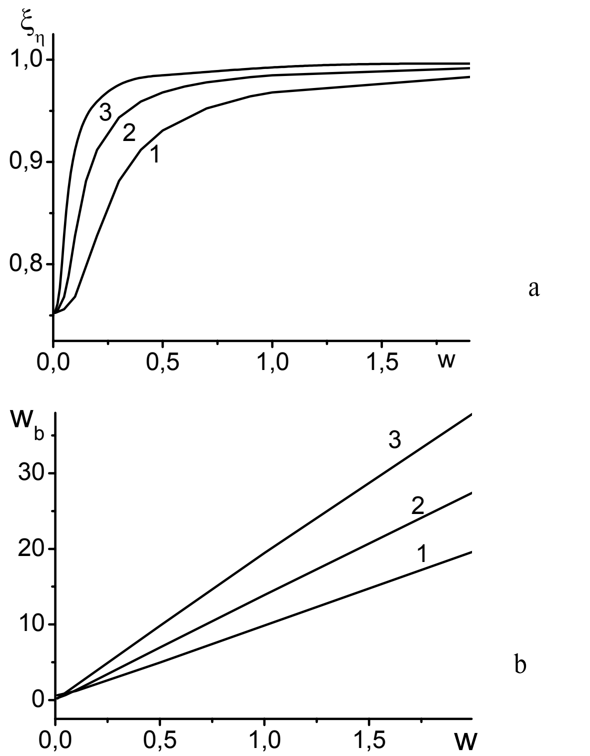

Fig. 2. *Dependence of: a) the reaction zone coordinate* ξ_η*; b) combustion rate* w_b *on gas feeding rate w at* x_1*=0,75;* β*=0,1;* K_λ*=97,56;* ε*=0,5;* σ*=0,66;* η_b*=0; Le=0,6175; Bi=3·10*$^{-5}$*. 1 –* δ*=100; 2 –* δ*=200; 3 –* δ*=400*

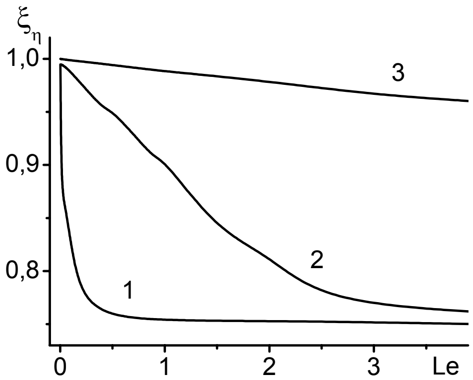

Fig. 3. *Dependence of the reaction zone coordinate ξ_η on Lewis number Le at different gas feed rates w: 1) 0,01; 2) 0,1; 3) 1; x_1=0,75. β=0,1; K_λ=97,56; ε=0,5; σ=0,66; η_b=0; Le=0,6175; Bi=$3\cdot10^{-5}$; δ=400*

The frame warming up at change of Bi in the range of 0,001...0,1 for the set of parameters characterizing the burner [9], in the whole, within the frames of the model (7)–(11) turns out to be insignificant.

The influence of Lewis number Le on combustions modes is of interest. At the selected set of parameters (x_1=0,75; β=0,1; K_λ>1; ε=0,5; σ=0,66; η_b=0; Le=0,6175; Bi=0,001...1,0) the ratio of the diffusion coefficient to the factor of thermal conductivity affects slightly the temperature profile form but it affects considerably the agent concentration distribution and therefore, the position of the reaction zone coordinate ξ_η. Fig. 3 illustrates this: the reaction zone coordinate tends to the burner outer radius at increase of gas feed rate.

Possible modes of the reaction agent conversion

At model parameter variation the other modes of conversion of the reaction products which may be of practical interest were found out. The values of Lewis number rather different (both upwards and downwards) from real values were purposely selected for illustration. In spite of the fact that the form of heat generation function (Fig. 4, *a*, dotted lines) changes considerably at Le change because of the different character of the reaction zone (Fig. 4, *c*), the temperature distribution in solid frame and its numerical value (continuous curves Fig. 4, *b*) do not practically differ from each other. At Le increase at gas feed rate w=0,1 the maximum heat generation in the reaction decreases, almost the whole working section of the burner becomes the reaction zone, therefore, the heat used for the solid frame warming up decreases as well. Owing to the high thermal conductivity of the frame it temperature equalizes and the frame becomes the source of heat for gas in the center of the burner that is seen from the comparison of continuous and dotted curves in Fig. 4.

At low rate of gas feeding the marked maximum of heat generating function is in the medium part of the burner (Fig. 4, right), that results in occurrence of heat generation maximum at Le<1 in the region distant from the outer surface x=1. The maximum temperature exceeds considerably the adiabatic temperature $\theta_{g,1}$=1 at x=1. Increasing Le owing to diffusive admixture the reaction zone broadens, the maximum in gas temperature disappears (dotted curves 1 and 2 in Fig. 4, *b*, right). Solid frame is the «heat source» for gas near the outer surface owing to its high heat conductivity, K_λ>>1.

The influence of the number Le on the combustion rate w_b, determined by the equation (12) turns out to be different. For example, at high rate of gas feeding w=0,1 the change of Le by two orders (from 0,01 to 10) resulted in change of the rate w_b from 6,82 to 8,83 (Fig. 4, *b*, left). But at w=0,03 the combustion rate increased considerably (Fig. 4, *b*, right).

The calculations showed that the decrease of solid frame heat conductivity at consistency of all the rest parameters results in the reaction zone shift to the outer surface of the burner both at w=0,1, and at w=0,03 (it is not shown in the Figures). A narrow reaction zone (with a complex structure) much more less than the burner working section requires the use of another algorithm with assignment of the region of considerable temperature change and agent concentration.

Combustion behavior at high external heat exchange of gas with surface

In a real burner device the outer surface of solid frame is warmed up to high temperatures and starts irradiating that supports heat exchange with heat exchanger. The outer surface warming up ($r=R_2$) may be connected with good heat exchange inside the porous burner device and with heat exchange with gas (combustion products), which is turbulized at output from the working body. The introduced parameters show low interphasic heat exchange. The condition of the external heat exchange in the mathematical model may be expressed by the boundary condition of the form

$$-\lambda_s \frac{dT_s}{dr} = \alpha_e (T_s - T_{g,1}) - \sigma\varepsilon_0 (T_s^4 - T_t^4), \qquad (16)$$

where $T_{g,1}$, as before the hot gas temperature at the output from the burner device; α_e is the coefficient of the external heat exchange (α_e>>α), depending on the character of gas flow, the external flow rate; σ is the Stefan-Boltzmann constant; ε_0 is the dark index; T_t is the temperature of the heat exchanger. In the dimensionless variables at x=1 we have

$$-K_\lambda \frac{d\theta_2}{dx} = \mathrm{Nu}(\theta_2 - 1) -$$
$$-B_2((\theta_2 - \frac{1-\sigma}{\sigma})^4 - (\theta_\mathrm{e} + \frac{1-\sigma}{\sigma})^4) = q_\mathrm{conv} - q_\mathrm{rad},$$

where Nu is the $\alpha_e R_2/\lambda_g$ Nusselt number, $\theta_e=(T_t-T_0)/(T_*-T_0)$, $B_2=\sigma\varepsilon R_2(T_*-T_0)^3/\lambda_g$, $q_\mathrm{conv}=\mathrm{Nu}(\theta_2-1)$ is thee convective current, $q_\mathrm{rad}=B_2((\theta_2-(1-\sigma)/\sigma)^4-(\theta_e+(1-\sigma)/\sigma)^4)$ is the heat radiation flux of the frame.

Let us estimate the parameters Nu=2,25...220,5, B_2=38,25, θ_e=0,14 (heat exchanger temperature T_t is taken equal 373 K) for the ceramic frame of Al_2O_3+Fe+Cr and gas – mixture of methane and air.

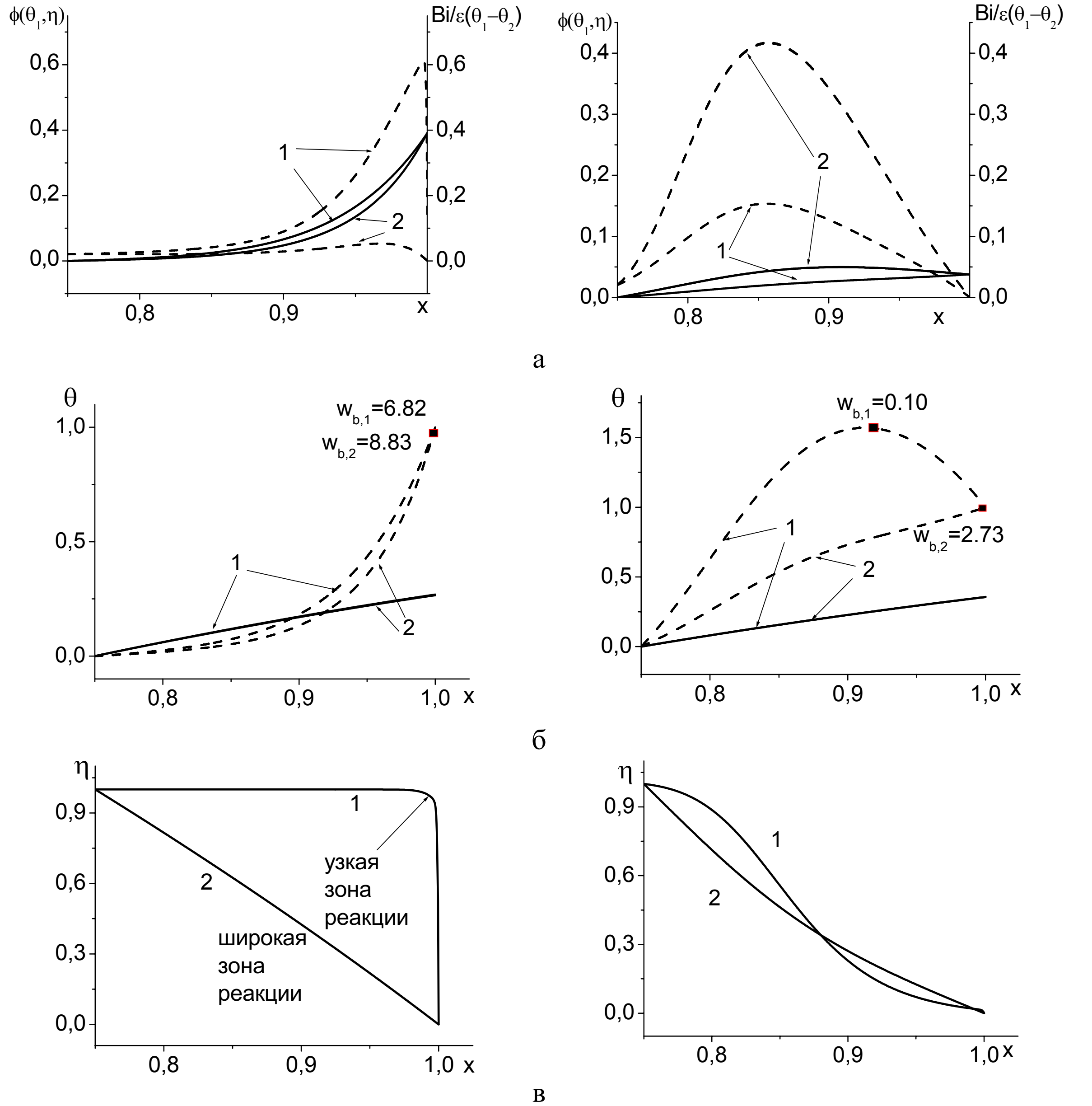

Fig. 4. *Distribution along the burner radius: a) heat generation due to the chemical reaction (dotted curve) and heat loss for the frame warming up (continuous); b) gas temperature (dotted curve) and the frame (continuous); c) agent concentration at different values of gas feeding rate and Lewis number at* x_1*=0,75;* β*=0,5;* K_λ*=97,56;* ε*=0,5;* σ*=0,66;* η_b*=0; Bi=0,2;* δ*=200; 1 – Le=0,01; 2 – Le=10; w=0,1 (lefta); w=0,03 (right)*

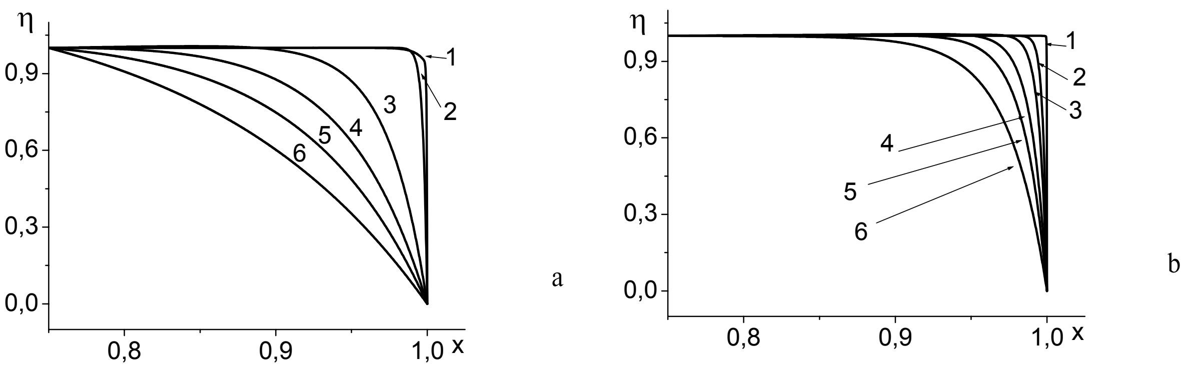

Fig. 5. *Dependence of agent concentration on spatial coordinate at different values of Lewis number: 1 – 0,01; 2 – 0,5; 3 – 1; 4 – 2; 5 – 3; 6 – 5;* x_1*=0,75;* β*=0,17;* K_λ*=50;* ε*=0,5;* σ*=0,66;* η_b*=0; Bi=*$3\cdot10^{-5}$*;* δ*=400* θ_e*=0,14; Nu=87,5;* B_2*=36,25; a) w=0,1; б) w=0,5*

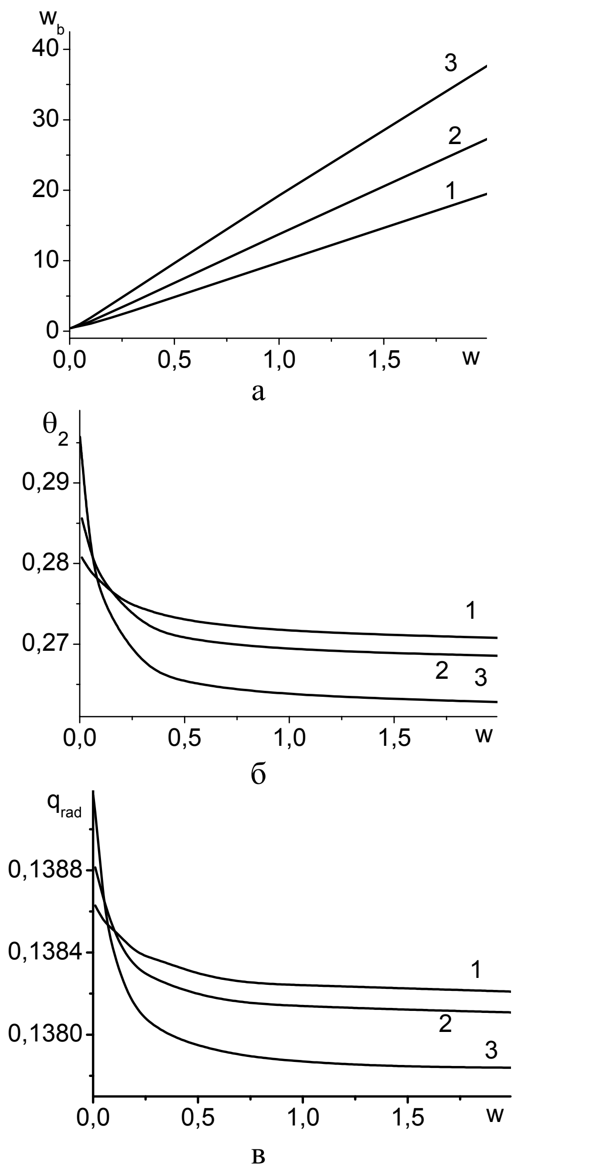

Fig. 6. *Dependence of: a) combustion rate; b) frame temperature at x=1; c) thermal irradiation on gas velocity and burner dimension at:* $\beta=0{,}1$; $x_1=0{,}75$; $\beta=0{,}17$; $K_\lambda=50$; $\varepsilon=0{,}5$; $\sigma=0{,}66$; $\eta_b=0$; $Bi=0{,}15$; $\theta_e=0{,}14$; $Nu=87{,}5$; $B_2=36{,}25$; $Le=0{,}6175$. 1 – $\delta=100$; 2 – $\delta=200$; 3 – $\delta=400$

As it is seen from Fig 5, when changing the number Le the curves qualitative distribution does not change. Comparing with the previous one, increasing Le the reaction zone spreads considerably that affects obviously the heat generation function. But a good heat exchange on the boundary $x=1$ (at the selected parameter values) and high frame heat conductivity result in «insensitivity» of gas temperature and frame to the Lewis number change even at low filtration rates in comparison with that what described above.

In conditions of good external heat exchange Nu>>1 the temperature of the solid frame inner surface becomes high that results in heat exchange with heat exchanger by irradiation. The temperature of solid frame outer surface in these conditions depends practically on all parameters of the model, for example, decreases at increase of gas feed rate (Fig. 6, *b*). In the last case the decrease of θ_2 ($x=1$) is connected with heat removal by heat conductivity deep into the burner. Gas temperature at the output of the burner is specified ($\theta_1=1$, $x=1$). Increasing the gas feed rate, as it was before, the range of considerable change of gas temperature and the reaction zone get narrow (it is not shown in Figures).

Increasing the burner radius and gas feed rate the combustion rate w_b grows (Fig. 6, *a*), that is connected with the increase of temperature gradient in the reaction zone; it results, in its turn in decrease of radiation flow (Fig. 6, *a*), that coordinates in the whole with the known concepts. The constancy of Nusselt number Nu at gas rate variation is not quite correct from the physical point of view – the coefficient of external heat exchange is the function of many values characterizing this technology and depending both on w, and δ. Within the frames of this model the processes at the output of the burner device are not considered therefore, it is not still possible to take it into account.

Conclusion

The two-temperature model of gas combustion in porous body of cylindrical heat generator is proposed and numerically analyzed. The influence of the model parameters on characteristics of stationary modes of gas combustion at variation of heat exchange conditions of porous burner with heat exchanger is studied in details.

The regularities conforming qualitatively to the known ideas were determined on the basis of the stated model that indicates the model validity to the experimental data. It was numerically shown that the increase of the burner outer radius and gas feed rate results in the reaction zone shift to the outer surface of the burner and drop of combustion rate for any set of thermaphysical parameters; change of the number Le influences greatly the reaction zone width and at decrease of activation energy resulting in increase of the parameter β, and reduction of the frame heat conductivity it influences the temperature of gas and frame; the increase of gas feed rate at constancy of internal and external heat exchange factors results in decrease of the temperature of frame and heat flux by irradiation. Unfortunately, these phenomena are not analyzed in many theoretical works.

As the detailed study is complicated at experimental investigations the proposed model may be used for obtaining the preliminary estimations when studying the functioning of the real burner devices and set of the problem of their optimization.

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IGNITION OF POROUS HIGH-ENERGY SUBSTANCES BY LIGHT RADIATION

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The possibility of calculation of high-energy solid fuel ignition processes within the limits of the porous reacting body model has been shown. Using the given model of ignition, it is possible to consider the dependence of ignition time on pressure which is ascertained experimentally while within the limits of the classic solid-phase ignition theory the ignition time does not depend on the initial and the external pressure.

As the experimental investigations show [1–3], the process of ignition of solid fuels (SF) is accompanied by various physical phenomena, in particular: SF gasification with heat release; the gasification products motion in pores; homogeneous chemical reactions in the condensed phase. At the same time the existing «solid-phase» (heterogeneous) and gaseous ignition theories are to some extent limiting and do not consider the whole variety of physical phenomena connected with SF ignition. For example in [4, 5] the solid-phase ignition model is used. Applying it the minimum sizes of heated bodies capable of igniting solid fuel are determined. In this work the above listed processes are taken into account in the development of solid phase ignition model within the frames of the model of porous reactive medium [6]

The reactive medium is supposed to be one-temperature; in the condensed phase the only effective homogeneous reaction of the form $v_1M_1 \to v_2M_2 + v_3M_3$, where v_1M_1 is the mass of the initial condensed substance (SF) occurs; v_2M_2, v_3M_3 is the mass of condensed and gaseous products of SF combustion reaction. The flux equal q_e falls from the external radiation source to the fuel surface. Gaseous products motion in pores and heat-mass exchange of fuel with the environment are taken into account. Let us study the ignition mechanism and determine the ignition time.

The equation system [6] describing the concerned process is written down in dimensionless form

$$\frac{\partial \varphi_1}{\partial t} = -\gamma_1 \varphi_1 \exp\frac{\theta}{1+\beta\theta},$$

$$\frac{\partial(\rho\varphi_3)}{\partial t} + \frac{\partial(\rho\varphi_3 u)}{\partial x} = \gamma_3 \varphi_1 \exp\frac{\theta}{1+\beta\theta}, \qquad (1)$$

$$c_{ps}\frac{\partial\theta}{\partial t} + \rho\varphi_3 c_p u\frac{\partial\theta}{\partial x} = \frac{\partial}{\partial x}\left(\lambda_s\frac{\partial\theta}{\partial x}\right) + \varphi_1 \exp\frac{\theta}{1+\beta\theta}, \quad (2)$$

$$u = -g\,\pi_u\frac{\partial p}{\partial x},\quad p = \frac{\rho(1+\beta\theta)}{\pi_p},\quad g = \frac{\varphi_3^3(1-\varphi_3)^{-2}}{\sqrt{1+\beta\theta}}. \quad (3)$$

This system was solved at the following boundary conditions