Regimes of filtrational gas combustion in a cylindrical annular burner with radiation heat transfer

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Abstract. A problem of premixed filtrational gas combustion in a cylindrical annular burner has been solved in the frame of one-dimensional diffusion-thermal model taking into account internal radiative heat transfer in porous burner and radiative heat flux emitted by internal layers of porous material. Two different stationary combustion regimes are detected under the same values of gas flow rate. It was shown that stationary flame radius depends on the initial ignition conditions. Solid and gas temperature characteristics depending on flame radius and gas flow rate are studied. Estimation of the burner radiation efficiency is carried out. Significant difference between temperature characteristics of gas and solid phases and burner radiation efficiency for different combustion regimes are shown.

1. Introduction

Burning premixed gases based on filtrational combustion in porous solid matrices is a relevant area of research, since it contributes to the creation of compact and efficient radiative burners for small energy with low emissions of NOx and CO [1]. A feature of such devices is the possibility of burning lean combustible mixtures, as well as the effective conversion of heat from gas combustion into an infrared heat flux from a porous burner. Recent experimental studies [1, 2] show that radiation heat transfer inside a porous body affects significantly the characteristics of the burner and the combustion wave stabilization.

In paper [3], a mathematical model of filtrational gas combustion in an inert porous media with allowance for radiation heat transfer was proposed. This model uses the same approximation as the classical two-temperature diffusion-thermal model [4] and it implies gas incompressibility, small pressure changes and constancy of the thermophysical coefficients. In paper [5], the results of numerical simulations of filtrational gas combustion in a cylindrical porous body were obtained within the framework of two models. The both models took into account radiation from burner outer surface, and radiation heat transfer inside the porous burner was ignored in one model, whereas another model included the radiation heat transfer effects and radiation into the environment from the inner layers of the porous body. The comparison between two models demonstrated importance of the effects related to infrared emission of the internal volumes of the porous body into the external environment. At the same time, consideration was limited to the assumption that the flame is inside the porous matrix. In this paper, gas combustion is considered both inside the porous body and in areas outside the porous matrix on the basis of the model [5].
2. Mathematical model

The scheme of filtrational gas combustion in an annular cylindrical burner is shown in Fig. 1. There are three regions: the region 1 \([r_0, r_{p0}]\) refers to the internal chamber, the region 2 \([r_{p0}, r_{p1}]\) refers to the porous matrix, and the region 3 \([r_{p1}, r_1]\) is outside the porous burner. The model includes nondimensional equations which describe heat transfer in the gas (1) and in the solid (4), transport of fuel (2) and air (3) and the equation for the density of radiation (5):

\[\frac{\partial \Theta}{\partial t} + V(r) \frac{\partial \Theta}{\partial r} = \frac{1}{r} \frac{\partial}{\partial r} \left( r \frac{\partial \Theta}{\partial r} \right) - \Omega_g (T - \Theta) + W (T, C_{ax}, C_g) \quad \{r_0 \leq r \leq r_1\} \quad (1)\]

\[\frac{\partial c_{ax}}{\partial t} + V(r) \frac{\partial c_{ax}}{\partial r} = \frac{1}{r} \frac{\partial}{\partial r} \left( r \frac{\partial c_{ax}}{\partial r} \right) - v W (T, C_{ax}, C_g) \quad \{r_0 \leq r \leq r_1\} \quad (2)\]

\[\frac{\partial c_f}{\partial t} + V(r) \frac{\partial c_f}{\partial r} = \frac{1}{r} \frac{\partial}{\partial r} \left( r \frac{\partial c_f}{\partial r} \right) - W (T, C_{ax}, C_g) \quad \{r_0 \leq r \leq r_1\} \quad (3)\]

\[\frac{d \phi}{d t} = k \frac{d}{r} \left( r \frac{d \phi}{d r} \right) + \Omega_s (T - \Theta) + \Omega_r (U_r - 2 \Theta^4) \quad \{r_{p0} \leq r \leq r_{p1}\} \quad (4)\]

\[\int_{r_{p0}}^{r_{p1}} \left( r \frac{d U_r}{d r} \right) = \mu^2 (U_r - 2 \Theta^4) \quad \{r_{p0} \leq r \leq r_{p1}\} \quad (5)\]

The temperatures of the gas \(T\) and the solid \(\Theta\) are scaled by the flame adiabatic temperature \(T_b\) corresponding to the given mixture and depending on the dimensionless radius \(r\), which is measured in the units of heat thickness \(l_{th}\). Premixed methane-air mixture with equivalence ratio \(\Phi = 0.7\) \((U_b=20\, \text{cm/sec}, \, T_b=1700\, \text{K})\) is considered. \(\sigma_0\) is a non-dimensional initial temperature in units of \(T_b\). Mass fraction \(\Sigma\) is scaled by one of the fresh mixtures. Indexes \(f\) and \(ox\) correspond to the fuel and oxidizer. \(U_r\) is related to the volumetric radiation energy and measured in units of \(\sigma_0 T_b^4\), where \(\sigma_0\) is a Stefan-Boltzmann constant. Gas velocity \(V\) is measured in units of laminar burning velocity \(U_b\) and has the following form:

\[V(r) = \frac{V_0}{r_0} \quad \frac{d}{dr} \left( r \frac{d}{dr} \right) \]

The following boundary conditions are used for the system of equations (1) - (5):

\[r = r_0: \quad T = \sigma_0, \quad C_f = 1, \quad C_{ax} = Y_{ax}/Y_f; \quad r = r_1: \quad dT/dr = dC_f/dr = dC_{ax}/dr = 0; \quad r = r_{p0}: \quad d\Theta/dr = dU_r/dr = 0; \quad r = r_{p1}: \quad d\Theta/dr = -(\Omega_r/k) \cdot (l_{out}^+ - \sigma_0^2) \quad U_r = l_{out}^+ + \sigma_0^2 \quad (6)\]

Integral radiative heat flux \(l_{out}^+\) from the burner towards the external environment has the form, as in the paper [5]:

\[l_{out}^+ = \mu l_{th} f_{p0} r_{p1}^4 \Theta^4(r) \exp \left( -\mu l_{th} (r_{p1} - r) \right) \quad dr\]

Initial distribution of the gas temperature has the following form:

\[T(r) = \sigma_0 + (1 - \sigma_0) \cdot \exp(-0.1 \cdot (r_{f0} - r)^2)\]

where \(r_{f0}\) corresponds to the radius of the heated area. Two cases of ignition
area location are considered: a) in the internal chamber of the burner \( \{r_0 < r_f < r_{p0} \} \); b) in the area surrounding the burner \( \{r_{p1} < r_f < r_1 \} \).

3. Results and discussions

Numerical modeling of the problem (1) - (6) shows existence of two different solutions. Dependence of a stationary radius of the flame front on the flow rate is shown in Fig. 2. It is shown that there are two stationary solutions under the same flow rates \( Q < 100 \) l/min. Dashed and solid lines in Fig. 2 correspond to the solutions, obtained with initial condition \( a \) and \( b \), respectively. The flame is stabilized in the internal chamber of the burner or at the outlet surface of the burner, it depends on the initial position of the heated area. Existence of these two combustion regimes coincides with “internal” and “external” combustion regimes described in experimental paper [7]. In numerical modeling “internal” regime is observed under the low flow rates \( Q < 50 \) l/min in case of initial condition \( a \), when the flame is stabilized in the internal chamber of the burner (region 1 in Fig. 1). In case of initial condition \( b \) the external combustion regime is set, when the flame is stabilized at the external surface of the porous matrix \( (r_f \approx r_{p1}) \). Increase of the flow rate over 50 l/min leads in flame stabilization inside the porous matrix for the initial condition \( a \), but for the initial condition \( b \) the stationary flame front occurs outside the burner (region 3 in Fig. 1). This solution corresponds to the free cylindrical flame without heat exchange between the gas and the solid phases.

Dependencies of solid temperatures on flow rate and radius were calculated. Dependencies of the burner outer surface temperature \( \theta(r_{p1}) \) on the flow rate for internal (solid line) and external (dashed line) combustion regimes are shown Fig. 3. It is shown that for internal combustion regime, a monotonous increase in the temperature is observed. For external combustion regime, \( \theta(r_{p1}) \) reaches maximum value at the gas flow rate \( Q \approx 35 \) l/min. Values of the temperature for internal regime exceed the one for external case up to two times. These results of numerical modeling are in good agreement with experimental results from the paper [7].
Dependencies of the burner radiation efficiency on the flow rate are shown in Fig. 4. Burner efficiency is defined as follows: \( \eta = \frac{W_{\text{rad}}}{W_{\text{ch}}} \), \( W_{\text{ch}} = S(p_0)\rho_0 c_p T_b (1 - \sigma_0) \), \( W_{\text{rad}} = S(p_1)I^+_\text{out} \), \( S \) is surface area. Monotonous decrease in the burner radiation efficiency is observed for internal (solid line) and external (dashed line) combustion regimes. It was experimentally shown in papers [1,7] that efficiency decreases for both regimes with an increase in the gas flow rate. Experimental results for internal combustion regime from the paper [1] are shown in Fig. 4 (squares). It was shown numerically that efficiency for internal regime significantly exceeds the one for the external regime at the same velocities of gas filtration.

Conclusions
A model of filtrational gas combustion in an annular cylindrical porous burner, which takes into account radiation heat transfer inside the porous matrix and the output of thermal radiation from the inner layers of the porous body to the external environment, is proposed. A numerical experiment shows the existence of two different stationary regimes at the same mixture flow rates. Depending on the initial ignition conditions, the flame can stabilize in the inner chamber of the burner, on the outer surface of the porous burner or inside the porous matrix. It was shown that the internal combustion regime is characterized by maximal temperatures of the solid matrix in comparison with external regime. The temperature of the burner outer surface increases monotonously with an increase in gas flow rate for internal combustion regime. For the external regime, the existence of an optimal gas flow rate is observed, at which the temperature reaches a maximum value. It is shown that the radiative efficiency of the burner decreases monotonously with increasing gas flow rate for both regimes. For the internal combustion regime, the efficiency is significantly higher (up to two times) than for the external one. These results are in qualitative agreement with experimental studies [1,2,7].

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