Radiative and convective heat exchange in technological furnaces of petrochemical industry

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Abstract. Complex heat transfer in radiant chambers of technological furnaces of the petrochemical industry is considered. The object of research is a vertically located flat channel, approximately simulating a radiant chamber. The walls of the channel are formed by a refractory lining and a tubular screen with a known temperature. The chamber volume is filled with selectively emitting, absorbing and weakly scattering media of gaseous fuel combustion products. The research method is based on the numerical solution of two-dimensional integro-differential equations of radiation gas dynamics, closed by a two-parameter model of turbulent convection. The influence of the modes of organization of the combustion process, the radiation properties of the combustion products and refractory lining on the distribution of heat fluxes and the characteristics of the total heat transfer are considered.

Tubular furnaces are widely used in the oil refining and petrochemical industries. Most of the used tubular furnaces are radiant-convection. In radiant chambers there are a number of complex interconnected physical and chemical processes. Heat to tubular screen is mostly transmitted by radiation from flames, hot surfaces of fire-resistant lining and partially convection from fuel combustion products. The convective component of heat exchange at temperatures characteristic of radiant chambers of tubular furnaces is small. However, in the volume of the furnace there is an intense mixing of hot streams of combustion products with relatively colder streams. These processes, along with others, have a strong impact on the distribution of temperature in the volume of the furnace, hence on the overall heat exchange as a whole.

The mathematical model of the processes taking place in the radiant chamber is based on the numerical solution of the system of two-dimensional integrative-differential equations of radiation gas dynamics, a closed two-parametric model of turbulent convection [1, 2].

The equation of the transfer of radiant energy in a rectangular area filled with radiating, absorbing and isotropy scattering medium has the form:

$$\mu \frac{\partial I_{\lambda \Omega s}}{\partial x} + \xi \frac{\partial I_{\lambda \Omega s}}{\partial y} = \alpha_{\lambda} I_{\lambda b \Omega} - (\alpha_{\lambda} + \beta) I_{\lambda \Omega s} + \frac{\beta}{4\pi} \int I_{\lambda \Omega s} d\Omega_s.$$  (1)

In here $\mu, \xi$ – guide cosines, $I_{\lambda \Omega s}$ – spectral radiation intensity in the direction of the vector $\hat{s}$, $I_{\lambda b \Omega}$ – spectral radiation intensity of an absolutely black body, $\alpha_{\lambda}$ – spectral absorption coefficient, $\beta$ – the scattering factor of the environment.
The boundary condition to the equation (1) for diffuse radiation and reflection from the walls has a view:

\[ I_{\lambda_s} = \varepsilon I_{b\lambda} + \frac{r}{\pi} \int_{(s')\hat{n} \leq 0} I_{\lambda_s}(s') \cos(s') d\Omega_s, \]  

(2)

where is \( \varepsilon, r \) – integral degree of blackness and reflectivity of the boundary surface.

The distribution of temperature in the calculated area is determined by solving the energy equation:

\[ cp\rho u \frac{\partial T}{\partial x} + cp\rho v \frac{\partial T}{\partial y} = \nabla \left[ (\lambda_m + \lambda_T) \nabla T \right] + Q - \nabla \cdot \vec{q}_p. \]  

(3)

In here \( u, v \) – components of the vector of the velocity of combustion products, \( cp, \rho \) – heat capacity and density of combustion products, \( \lambda_m, \lambda_T \) – molecular and turbulent thermal conductivity ratios, \( Q \) – volume density of thermal power sources, \( \vec{q}_p \) – vector of density of the resulting flow of radiant energy. Divergence of radiant heat flows is determined by the formula:

\[ \nabla \cdot \vec{q}_p = \frac{1}{4\pi} \left( 4\pi I_{b\lambda} - \int_{4\pi} I_{\lambda_s} d\Omega_s \right) d\lambda. \]  

(4)

The object of the study is a vertically located flat channel height \( L \) and width \( H \), approximate modeling the radiant chamber (figure 1). The walls of the channel are formed by a refractory lining and a tubular screen with a known temperature. The chamber volume is filled with selectively emitting, absorbing and weakly scattering media of gaseous fuel combustion products.

The equation of energy transfer by radiation is solved in the approximation of the method discrete ordinate by approximating the corresponding equations on an uneven finite difference grid. The gas-dynamic part of the problem is solved numerically on the finite difference grid of the "chess" type according to the known algorithm SIMPLER [3].

![Figure 1. Object of study. Indirect directional (a) and evenly distributed (b) heat exchange modes.](image)

The temperature of the refractory lining and the heat flows on it are determined from the heat balance equation:

\[ \frac{\lambda_w}{d} (T_W - T_0) = \left( \frac{\lambda_m}{d} \frac{\partial T}{\partial x} \right)_w - (\vec{q}_p \vec{n}), \]  

(5)

where is \( \lambda_w, d \) – thermal conductivity factor and lining thickness, \( \vec{n} \) – vector of a single internal normal to the surface of the lining.

Near a hard border formed the so-called turbulent boundary layer, within which there are quite large values of temperature gradient. Therefore, for numerical approximation of the equation (5) the method of "wall functions" is used [4]. The equation (5) is presented in the form of:

\[ \frac{\lambda_w}{d} (T_W - T_0) = \frac{\lambda_m}{d} \frac{x_p^+ (T_1 - T_W)}{F_p x_p} - (\vec{q}_p \vec{n}), \]  

(6)

where is \( T_1 \) the temperature of combustion products at the first nodal point from the wall, \( x_p^+ \) – dimensionless distance.
The shear stress on the wall is calculated by the formula:

\[ \tau_w = \frac{2C_{\mu}^{1/4} k_1 V_1}{\ln(E x_p^+)} \quad (8) \]

In here, \( \mu \) – coefficient of dynamic viscosity of combustion products, Carman constant \( \chi = 0.41 \); factors \( E = 8.8; C_{\mu} = 0.09 \). The longitudinal velocity \( V_1 \) and kinetic energy of turbulent pulsations \( k_1 \) in the first nodal point from the wall is determined by solving the corresponding equations of gas dynamics and turbulence model. \( F_p \) – empirical function:

\[ F_p = Pr_T \chi^{-1} \ln(E x_p^+) + 9.24 Pr_T \left( \frac{Pr_T}{Pr} - 1 \right) \frac{1}{4} \]

where \( Pr \) – Prandtl number, \( Pr_T \) – turbulent Prandtl number. Formulas (6), (7), (8) are valid for

\[ 30 \leq x_p^+ \leq 400 \]

The equation (6) can be presented as:

\[ T_w = \frac{R_w T_1 - R_c (R_w \tilde{q}_p \vec{n} - T_0)}{R_c + R_w} \quad (9) \]

Through \( R_w \) and \( R_c \) are indicated the thermal resistance of the lining and the boundary layer, respectively:

\[ R_w = \frac{d}{\lambda_w} \quad ; \quad R_c = \frac{x_p^+ F_p}{\lambda_m x_p^+} \]

The resulting flow of radiant energy on the canal wall is determined by the formula:

\[ (\tilde{q}_p \vec{n}) = E_w - E_{abs.} = E_w - \varepsilon E_f \quad (10) \]

The surface density of the wall’s own radiation \( E_w \) is calculated in the "grey" approximation, the density of the incident radiation flux is calculated by summing the spectral emission bands of the combustion products within the hemisphere:

\[ E_f = \sum_{k=1}^{N_s} \Delta \lambda_k \sum_{m=1}^{N_{\mu}} I_m \lambda_k \mu_m \Omega_m \quad (11) \]

The density of the lining’s own radiation depends on its temperature \( T_w \), therefore, the expression (9) can be formally presented as a linear function:

\[ T_w = \frac{R_w T_1 + R_c (T_0 + \varepsilon R_w E_f)}{R_w + R_c} - \frac{R_w R_c E_w T_w}{(R_w + R_c) T_w^*} \quad (12) \]

where \( T_w^* \) – the lining temperature calculated on the previous iteration step. From here we find a formula for calculating the temperature of the lining:

\[ T_w = \frac{R_w T_1 T_w^* + R_c T_w^* (T_0 + \varepsilon R_w E_f)}{T_w^* (R_w + R_c) + R_w R_c E_w} \quad (13) \]

The expression (13) is non-linear, so the calculation of the lining temperature is done by method successive approximations.
The calculations were performed for the parameters characteristic of tubular furnaces: the temperature of gases at the entrance to the canal $1500 \text{ K}$; tubular screens temperature $1100 \text{ K}$; thermal resistance of lining $R_w = 0.5 \frac{m^2\text{K}}{W}$. The nine-lane model of the radiation spectrum of fuel-combustion products was used, taking into account the bands 1.5; 2.7; 6.3; 10 μm radiation spectrum $\text{H}_2\text{O}$ and 2.7; 4.3; 15 μm $\text{CO}_2$. The radiation transfer equation was solved in the $S_4$-approximation of the discrete ordinate method.

The influence of radiant heat transfer on the convective heat transfer coefficient on the heating surface is considered (figure 2). Large values of the convective heat transfer coefficient at the initial section of the channel are due to the high temperature and high velocity of the incoming gas stream. A slight increase in convective heat transfer at the outlet of the channel occurs due to temperature equalization across the channel.

Radiant heat exchange has a significant effect on the convective heat transfer coefficient. When the partial pressure of emitting gases $P_W$ of water vapor $\text{H}_2\text{O}$ and $P_C$ carbon dioxide $\text{CO}_2$ increases by a factor of three, the convective heat transfer coefficient decreases by an average of 20%. This can be explained by the more intense cooling of the gas flow in the channel due to greater radiant heat transfer.

![Figure 2. Convective heat transfer coefficient.](image1)

![Figure 3. Thermal resistance of the boundary layer on the tubular screens.](image2)

The effect of radiant heat transfer to the thermal resistance of the boundary layer on the heating surface was also investigated (figure 3). No significant effect of radiant heat transfer to the thermal resistance of the boundary layer has been detected, at least this effect does not go beyond the error of the calculation method. As a rule, the convective heat transfer factor is determined experimentally by the average flow characteristics. Therefore, in thermal calculations of radiant chambers, when the transfer of heat by radiation has a significant impact on the current processes, it is more acceptable to determine convective thermal flows through thermal resistance of the border layer.

The results of the calculations show that radiant heat transmission in the “windows” of transparency of the radiation spectrum have a significant impact on the formation of thermal flows. In the indirect directed mode of heat exchange (IDM) the share of radiant heat transmission in the “windows” of transparency in the total thermal balance of the radiation chamber is 45%, with evenly distributed mode (EDM) - 36%. The reason is that transparency “windows” are relatively wider than spectral absorption bands of emitting gases. The second reason is that the transparency “windows” are located in the spectrum region, where the intensity of radiation of an absolutely black body at the temperatures typical of tubular furnaces is of maximum value.
Table 1. The proportion of radiant heat transfer in the intervals of the spectrum, %.

| Mode | Spectrum wavelength, µm | Transparency windows, µm |
|------|-------------------------|--------------------------|
|      | 1.5 | 2.7 | 4.3 | 6.3 – 10 | 15 | 0 – 1.5 | 1.5 – 2.7 | 2.7 – 4.3 |
| IDM  | 6   | 22  | 7   | 16       | 4  | 9      | 25         | 11         |
| EDM  | 7   | 26  | 10  | 16       | 5  | 8      | 20         | 8          |

According to table 1, the largest amount of radiant heating surface energy is transmitted in the spectral band of 2.7 µm and within the "window" of transparency between the spectral bands of 1.5 µm and 2.7 µm.

With an indirect directional mode of heat exchange in the volume of the radiation chamber there are two rather large areas of the recycling current, which are absent with an evenly distributed mode of heat exchange. These areas, having a lower temperature, shield the heating surface from both the radiating surface of the lining and from hot combustion products. Therefore, an evenly distributed mode of heat exchange provides greater heat perception of the heating surface.

The efficiency of heat exchange in the "windows" of spectrum transparency can be characterized by a k-factor equal to the heat flow to the heating surface in the "windows" of transparency to the total thermal flow. The dependence of this factor on the degree of blackness of the lining is presented in figure 4. Spectrum transparency windows are most effectively used at high blackness values of the radiating lining. With the increase in the degree of blackness of the lining from the value of 0.6 to 0.8 heat flow to the surface of the heating in the "windows" transparency increases by 22% in both modes of heat exchange. The spectrum radiation reflected from the lining in most part is "striped", having in the absorption bands of combustion products the highest intensity, and in the "windows" of transparency between them - the lowest intensity. The small intensity of reflected radiation in the "windows" is explained by the scattering and reflection of radiation from the surface of heating and lining. At the same time, the spectrum of own radiation of lining is solid and for this radiation the spectrum intervals between the absorption bands are transparent.

Experimental studies carried out in the work [5] show that the degree of blackness of refractory strongly depends on the wavelength of radiation (table 2). If you take the average temperature in the combustion chamber equal to 1300K, the maximum radiation intensity of the absolutely black body corresponds to the wavelength of 2.23 µm. In an practice almost interesting spectrum range  =1÷5 µm there is a decline in the degree of blackness of refractory’s more than twice, which negatively affects the heat perception of the heating surface.

The results of the calculations are presented in table 3. In general, refractory lining of fireclay bricks provides higher values of the heat flux to the heating surface. The difference in total heat transfer is approximately 5%.

An analysis of the spectral distribution of radiant fluxes to the heating surface shows that the thermal perception of the heating surface within the absorption bands of the combustion medium in
Table 2. Average degree of blackness of refractory on the spectral band.

| Refractory | Radiation bands, μm | Transparency windows, μm |
|------------|---------------------|--------------------------|
|            | 1.5     | 2.7     | 4.3     | 6.3+10 | 15     | 0÷1.5   | 1.5÷2.7  | 2.7÷4.3  |
| Fireclay   | 0.35    | 0.23    | 0.46    | 0.96   | 0.80   | 0.34    | 0.31     | 0.29     |
| Clay       | 0.34    | 0.20    | 0.27    | 0.91   | 0.80   | 0.22    | 0.20     | 0.24     |

Table 3. Spectral distribution of heat flow density to screen surface, kW/m².

| Refractory | Radiation bands, μm | Transparency windows, μm |
|------------|---------------------|--------------------------|
|            | 1.5     | 2.7     | 4.3     | 6.3+10 | 15     | 0÷1.5   | 1.5÷2.7  | 2.7÷4.3  |
| Fireclay   | 1.28    | 11.04   | 6.88    | 11.07  | 3.18   | 1.17    | 4.34     | 3.06     |
| Clay       | 1.27    | 11.03   | 6.88    | 10.99  | 3.19   | 0.82    | 3.12     | 2.67     |
| "Grey" model | ε = 0.6 | 1.51    | 11.11   | 6.88   | 10.50  | 3.05    | 1.93     | 7.55     | 5.50     |
|            | ε = 0.8 | 1.70    | 11.15   | 6.88   | 10.82  | 3.19    | 2.51     | 9.77     | 7.07     |
|            | ε = 1.0 | 1.88    | 11.19   | 6.88   | 11.14  | 3.32    | 3.09     | 11.98    | 8.64     |

The framework of the “gray” model for refractory lining and the spectral model is approximately the same. This can be explained by the fact that the combustion medium in the absorption bands has a sufficiently high optical density and shields the refractory lining from the heating surface, thereby reducing the influence of the radiation properties of the lining on the value of heat fluxes. As follows from table 3, in the transparency windows, the results of calculations according to the “grey” model are several times higher than the values of heat fluxes calculated using the spectral model of the radiation of the refractory lining. Calculating radiant heat exchange in transparency windows without taking into account the selectivity of the radiation properties of refractory lining leads to overestimated values of the heat fluxes to the heating surface.

References

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