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CALCULATION OF HEAT TRANSFER IN HEAT GENERATORS OF LOW POWER

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ABSTRACT

Calculation of heat exchange in heat generators of low power is made; the main problem which arises during the calculating and justification of a certain model of heat exchange for furnace chambers of small volume is exact definition of relative participation of radiation and convective transfer in difficult heat exchange. Comparison of settlement dependences to experimental data has shown that for furnace chambers of small volume the physical model of heat transfer in the field of optical thickness the radiating layer has to be based on the use of a model of heat transfer radiation and averaging of coefficients of absorption Rosseland. The method of calculation of heat exchange in fire chambers with use of physical model of a heat mass transfer is considered. It is received dependence which can be used for testing and constructive calculation of fire chambers of heat generator.

Keywords: heat, boiler, convection, flue gas.

1. INTRODUCTION

Radiation and convective heat exchange between a torch, high temperature products of combustion and walls of furnace cameras of heat generators of a small power has essential difference from similar processes in fire chambers of larger heat generators used (Roddatis*et al.,* 1989). Change of geometrical conditions of the course of processes of heat transfer and other regime parameters of work of fire chambers lead to essential mistakes at application of methods of calculation of heat exchange of large heat generators (Isachenko *et al.,* 1975). The main question arising at justification of this model of heat exchange in fire chambers of small volume is the assessment of a relative contribution of radiation and convective transfer to difficult heat exchange. The solution of this question - in the correct assessment of conditions of radiation heat transfer in fire chambers of the small geometrical sizes by research of optical properties of the radiating and absorbing environment (Mirame*t al.,* 2011).

2. METHOD

The analytical solution of a problem of radiation transfer in the radiating and absorbing gas stream is hampered by the non-linearity of the equations describing heat transfer processes; interaction of radiation with convection; selectivity of radiation of gases; fuel burning processes, etc. In this regard there is a need of experimental works and search of decisions by introduction of a number of assumptions. In order to prevent the integral members in the equation of transfer of energy approximations (Kalchevsky, 2012) optically thin or thick gas layer and their modification, and also model of "grey" gas are used.

When using model of "grey" gas the radiating and absorbing characteristics of the gas environment are assumed independent of the wavelength of radiation and estimated by average coefficient of absorption for all range of radiation. For approximations optically thin and optically a thick gas layer of gas absorption coefficientsare averaged respectively according to Planck and Rosseland.

In approach optically thin layer $\tau \leq 1$ density of a thermal stream can be presented by radiation in the form:

$$q_l = 2K_p / E_o(T), \tag{1}$$

q_l = specific thermal stream of radiation, W/m^2;
K_p = the absorption coefficient average according to Planck, m^{-1};
l = constructional feature, m;
$E_o(T)$ = density of a stream of radiation of absolutely black body, W/m^2.

The absorption coefficient average according to Planck is described by expression:

$$K_p = \frac{\int_{\lambda=0}^{\infty} K_\lambda I_\lambda^o(T) d\lambda}{I^o(T)}, \tag{2}$$

λ = wave length, m;
K_λ = monochromatic coefficient of absorption, m^{-1};
$I_\lambda^o(T)$ = spectral intensity of radiation of absolutely black body, W/(m^2·ster);
$I^o(T)$ = integrated intensity of radiation of absolutely black body, W/(m^2·ster).

In approach optically of a thick gas layer $\tau >> 1$ density of a thermal stream radiation, W/m^2, can be written down in a look:

$$q_l = -\frac{4}{3} \cdot \frac{1}{K_r} \cdot \frac{dE_o(T)}{dl}, \tag{3}$$

K_r = Rosseland average absorption coefficient m^{-1}, which is determined by:

$$\frac{1}{K_r} = \int_{\lambda=0}^{\infty} \frac{1}{K_\lambda} \cdot \frac{dI_\lambda^o(T)}{dI^o(T)} d\lambda. \quad (4)$$

In this case energy transfer by radiation in the environment can be considered as diffusive process in all layer of the radiating gas, but in certain cases diffusive approach is unacceptable near layer borders (Bruykhanov*et al.,* 2014).

Use of approach of Rosseland for heat exchange calculation with radiation in the cooled furnace chambers proves rightness of its application in the wide range of change of optical thickness of the radiating layer. However for intermediate area of optical thickness of the radiating layer of gases use of both coefficient of the absorption average on Rosseland, and average according to Planck is possible.

Thus, the choice of physical model and method to solve a specific objective should be carried out on the basis on a preliminary estimate of optical thickness the radiating layer, and a averaging the absorption coefficient method to prove analytical and experimental by researches of physical features of process.

The problem was that numerical values of settlement sizes depend on a method of averaging of coefficients of absorption, and geometrical characteristics even of furnace chambers of small volume cover the big range of the linear sizes. It results in need of carrying out parallel calculation of sizes and their subsequent analysis on the basis of experimental data. However, irrespective of a method of averaging of coefficients of absorption, the wide range of practical tasks can't be carried to limit cases of big and small optical thickness of the radiating gas layer (Khavanov,2014).

Skilled installation consisted of the section cylindrical cooled combustion chambers of four diameters of0,18-0,36 mm and the relative length of sections 0,67-2,33 of diameter. Also changed concentration of the radiating components as a part of combustion products in case of a fixed ratio of partial volumes of CO_2 and H_2O. In experiences natural gas and liquid oven fuel of brand A were used. The structure of the radiating environment in volume of a furnace chamber changed in limits: $r_{CO_2} = 0{,}005 - 0{,}11; r_{H_2O} = 0{,}1 - 0{,}192.$

For calculation of coefficients of absorption of products of combustion the coefficients of absorption of carbon dioxide and water vapor average Rosseland and according to Planck received on spectral data (Abu-Romia*et al.,* 1967) for temperatures of 556-2780 K and various pressure were used. Definition of calculated values of coefficient of absorption of products of combustion of K_g according to recommendations about an empirical formula was in addition carried out:

$$K_g = \left(\frac{0{,}78 + 1{,}6 r_{H_2O}}{\sqrt{P_p t}} - 0{,}1\right)\left(1 - 0{,}37 \frac{T_2}{1000}\right)(r_{CO_2} + r_{H_2O}), \quad (5)$$

T_2 = experimentally determined values of temperature of products of combustion at the exit from a fire chamber.

The thermal stream radiation W/m^2, can be estimated for a case optically of a thin layer (τ « 1):

$$q_l = 2K_p / l\sigma_o n^2 T^4, \quad (6)$$

for optically thin layer (τ » 1):

$$q_l = -\frac{16}{3} \cdot \frac{\sigma_o n^2 T^4}{K_r l}, \quad (7)$$

n = index of refraction.

In a furnace chamber the structure of products of combustion and theoretical temperature of burning for fuel of the same structure depend only on coefficient of excess of air. Considering it, calculation of average coefficients of absorption of products of combustion for optically thin and thick layers of "grey" gas is made respectively on:

$$K_r(T_\alpha) = K_r^{CO_2}(T_\alpha) P_{CO_2}(\alpha) + K_r^{H_2O}(T_\alpha) P_{H_2O}(\alpha), \quad (8)$$

$$\overline{K_r}(T_\alpha) = \frac{1}{K_r(T_\alpha)} = \frac{1}{K_r^{CO_2}(T_\alpha) P_{CO_2}(\alpha)} + \frac{1}{K_r^{H_2O}(T_\alpha) P_{H_2O}(\alpha)}, \quad (9)$$

$\overline{K_r}$ = modified Rosseland average absorption coefficient.

Use of "grey" gas model without mutual influence of the radiating and absorptive CO_2 and H_2O properties in mix can lead to an error of 10%.

In calculations with use of average coefficients of absorption for Planck numerical values of optical thickness of a layer of products of combustion in all range of researches are received: for natural gas τ = $K_p l$ = 0,093-0,326; for liquid fuel τ = $K_p l$= 0,11-0,384.

When using averaging Rosseland: for natural gas τ = $\overline{K_r} l$ = 0,3–1,14; for liquid fuel τ = $\overline{K_r} l$ = 0,35-1,33.

According to the recommendations: for natural gas τ = $K_g l$= 0,081-0.28; for liquid fuel τ = $K_g l$ = 0,09-0,3.

The received numerical values τ confirm the assumption that irrespective of methods of averaging of coefficients of absorption the layer of the radiating products of combustion can't be carried to an optical and a thin layer. This fact significantly complicates the task and

results in need of additional analysis which can be carried out on the basis of pilot studies.

Considering dependence of a radiant thermal stream on optical properties of the radiating environment and in particular, of the temperature level and concentration of triatomic gases, it was possible by practical consideration to carry out the analysis of influence of coefficient of excess of air α on heat exchange in a furnace chamber.

Influence of coefficient of excess of air on heat exchange by radiation in the furnace camera can be calculated by means of values of average coefficients of absorption and comparison of the received results to the experimental data.

On the basis of the equation (8) calculated by the average obtained approximate dependence of the absorption coefficient according to Planck for products of combustion of natural gas

$$K_p \cong 0{,}5\alpha^{0{,}85}. \tag{10}$$

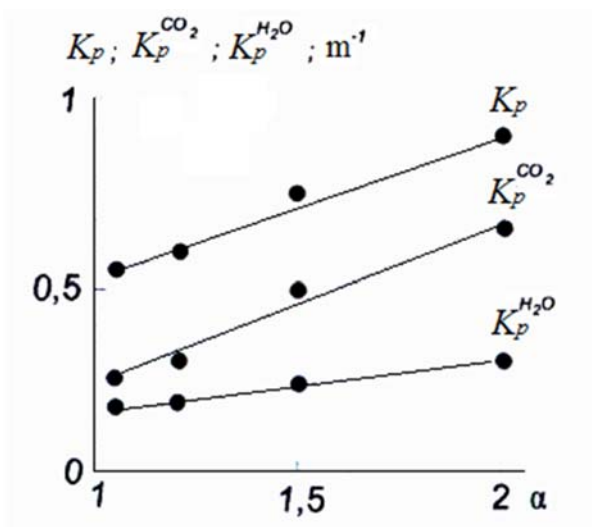

Figure-1. Dependence of the coefficients of absorption of products of combustion average according to Planck on coefficients of excess of air

$$K_p^{CO_2} = \overline{K_p^{CO_2}} \cdot P_{CO_2}; K_p^{H_2O} = \overline{K_p^{H_2O}} \cdot P_{H_2O}; K_p = K_p^{CO_2} + K_p^{H_2O}.$$

Graphic interpretation of dependence of K_g, from coefficient of excess of air (Figure-2).

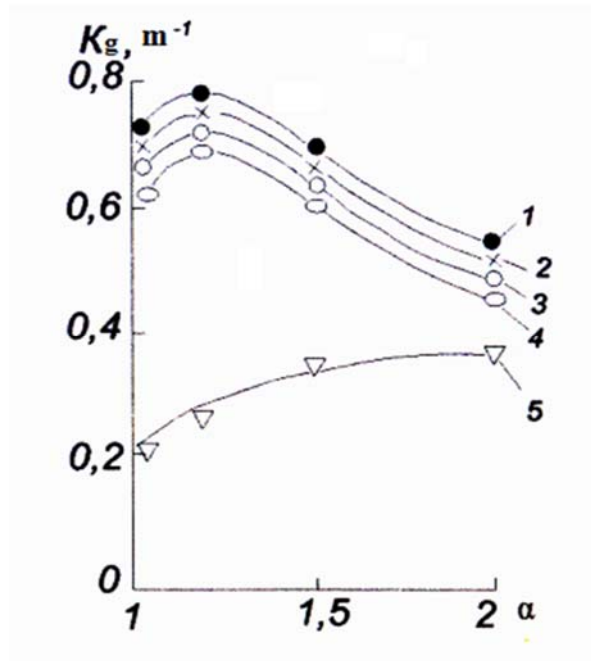

Figure-2. Calculated values of coefficients of absorption of products of combustion depending on coefficient of excess of air on experimental temperature at the exit from a fire chamber $T_g'' = f(q_v)$ at 1 - q_v =0,762·10³ kW/m³; 2 - q_v =0,922·10³ kW/m³; 3 - q_v =1,462·10³ kW/m³; 4 - q_v = 1,724·10³ kW/m³; and at the theoretical temperature of burning $5 - T_T \neq f(q_v)$

Similar calculations for the average absorption coefficient of the Rosseland have been carried out for products of combustion of natural gas and fuel oil. Approximate dependence on coefficient of excess of air has an appearance:

$$\overline{K_r} = 1 / K_r = \overline{K_r^o} \frac{1}{\alpha}, \tag{11}$$

$\overline{K_r^o}$ = Rosseland average modified absorption coefficient of dilution of the combustion products (α = 1).

Comparison of the calculated dependence (10) and (6) leads to the conclusion that by averaging the absorption coefficients for the Planck density of the heat flux is proportional to the excess air ratio:

$$q_l \sim \alpha^m, \tag{12}$$

comparison of dependences (11) and (7) by averaging the absorption coefficients of the Rosseland leads to addiction:

$$q_l \sim (1/\alpha)^n, \tag{13}$$

The experimental investigation of heat transfer in the combustion chamber, conducted according to the purpose of obtaining the integral transfer of the excess air ratio at constant geometrical (d_c = const; l/d_c =const), the hydrodynamic ($Re_н$ =const) conditions (Aronov1964) and under identical conditionsof burning fuel led to the equation

$$K_T = (I_a - I_T^{''})/(I_a - I_o) = A(\frac{1}{a})^{0,47}, \tag{14}$$

$I_a; I_T^{''}; I_o$: full enthalpy of products of combustion respectively at a theoretical temperature of burning and distempers at the exit from a fire chamber and a wall of a fire chamber, kJ/kg.

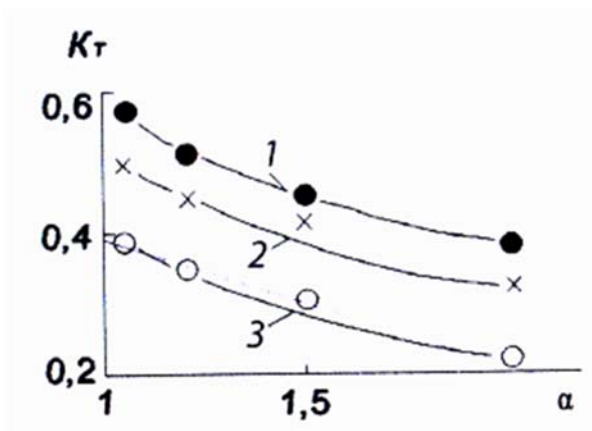

Figure-3. Dependence of number of integrated heat transfer on coefficient of excess of air at combustion of natural gas in a fire chamber d_c = 240 mm at: 1 -l/d_c = 2,33; $Re_н$ = 101 ± 15%; 2 – l/d_c = 2,33; $Re_н$ = 148 ± 10%; 3 – l/d_c = 1,33; $Re_н$ = 145 ± 12%.

The received expression confirms justice of dependence (13). Processing of an additional series of experiences in fire chambers of various sizes with use of the coefficients of absorption average Rosseland (11) lets summarize the received dependence (14) for the cooled furnace chambers of various diameter:

$$K_T = A_1(\overline{K_r} d_c)^{0,47} = A_1(\frac{\overline{K_r^o} d_c}{\alpha})^{0,47}, \tag{15}$$

$\overline{K_r^o}$ = 3,34 m^{-1} – for products of combustion of natural gas at α = 1; K_r^o = 3,92 m^{-1}- for products of combustion of fuel oil at α = 1; K_r^o = 3,89 m^{-1}- for products of combustion of TPB of brand A at α= 1.

Comparison of the calculated dependences with the experimental data shows that in the test environment to use as the absorption coefficient of the combustion medium, averaged Plank or on the recommendations (Figure-3) of the calculated, leads to a contradiction between the calculated and experimental data.

In this regard for furnace chambers of small volume the physical model of process of heat transfer in the studied area of optical thickness of a layer of products of combustion of hydrocarbonic fuel has to be based on use of diffusive model of heat transfer by radiation and averaging of coefficients of absorption Rosseland.

Thus, the furnace environment in heat generators of low power can't be considered as optically thin. The gas stream actively interacts with radiation and has significant effect on radiation transfer.

Experimentally reasonable engineering technique of thermal calculation creates the heat generator design which is optimum balanced on thermal processes, significantly reduces amount of works at an operational development stage to design indicators. For constructive and testing calculation of heat generators of low power using thermal methods of calculation. The method is based on empirical dependences obtained during the test of power boilers, or more powerful heat sources, for example, iron sectional boilers.

Differences in geometrical and regime parameters of operation of heat generators of low power influence physical conditions of course of processes and, therefore, for fire chambers of small volume the physical model of heat transfer also has to undergo changes. So, as a result of reduction of the geometrical sizes of a fire chamber, even at preservation by constants of structure and temperature of the furnace environment, the optical thickness of the radiating and absorbing layer of products of combustion decreases and, as a result, heat transfer by radiation decreases, the relative contribution of convection to difficult heat exchange increases that demands the corresponding reflection in physical model of processes of heat transfer (Bell*et al.,* 1972).

Change of the geometrical sizes also affects regime parameters of operation of heat generators of low power. For example, the gas water heater of AGV-120 which is in operation a long time equipped with an ejector torch of incomplete preliminary mixture has the thermal tension of furnace volume of q_v twice surpassing recommended for powerful power boilers and by 1, 9 times - for pig-iron, and the thermal tension of surfaces of heating in $q_н$ fire chamber - approximately the same. The main reason is a geometrical factor of small fire chambers for which the volume relation to a surface of heating Is minimum. So, for a cylindrical fire chamber the volume relation to a surface of heating is expressed by diameter d and length l to the following:

$$\frac{V}{H}=\frac{\pi d^2 l}{4\pi dl}=\frac{1}{4}d,$$

for prismatic *aal*

$$\frac{V}{H}=\frac{aal}{4al}=\frac{1}{4}a,$$

In practice of design of heat generators of low power there is some positive experience of the authors developed method of calculating heat transfer based on the model of the stationary radiation and convective transfer of energy a turbulent stream of the radiating and absorbing "grey" gas for axisymmetric channels. The generalizing dependence received by S. N. Shorin (Shorin*et al.,* 1967):

$$St=\frac{1}{1+A\mathrm{Re}_н\cdot 0{,}115/l_e}, \tag{16}$$

St = Stanton's criterion;
$\mathrm{Re}_н$ = Reynolds criterion;
l_e = the defining size.

The method of calculation of heat exchange in fire chambers of heat generators of low power is based on use of the physical model of processes of a heat mass transfer (Hawes 1976)offered by S. N. Shorin in more general problem definition for cases of combustion of gaseous, liquid and solid fuel in furnace chambers of various geometrical sizes under changed conditions of burning and input of fuel-air mix. The used physical model of process considers heat transfer from a stream of the radiating combustion products from fire chamber volume to the walls through the interface which is formed in wall area. Transfer of heat from fire chamber volume in an interface is carried out by the radiation and turbulent diffusion, assuming that molecular transfer of heat from volume to an interface can be neglected. Heat is transmitted through an interface by the radiation and the molecular heat conductivity on heat perceiving surfaces which can make a considerable share in a total thermal stream.

For a special case of layered combustion of solid fuel in heat generators of the decentralized heat supply heterogeneous burning in a layer can be presented as an infrared torch or a set of gas torches. Heat exchange within a thick layer of the burning fuel with a wall of a furnace chamber has to be considered separately.

The analysis of the offered physical model provides the generalized equation of similarity of heat transfer at combustion of gaseous, liquid and solid fuel:

$$K_T=f(\mathrm{Re}_н;Bu;\alpha;l/d_e;\sigma), \tag{17}$$

K_T = number of integrated heat transfer;
$\mathrm{Re}_н$ = the conditional number of Reynolds counted on the speed of the gas stream carried to heat perceiving fire chamber surface;
Bu = Buger number;
A = air-fuel ratio;
l/d_e = ratio of length and equivalent diameter of a fire chamber;
σ = the parameter considering burning conditions, a way of burning and type of the furnace device.

3. DISCUSSION AND RESULTS

In developing the method used material experimental studies of heat exchange in furnace chambers and their models with the regime and geometrical parameters close to characteristic of fire chambers of heat generators of the decentralized heating furnaces.

The method of calculation of heat exchange is based on use received from (17) dependences of a look:

$$K_T=\frac{1}{1+A\frac{1}{\xi}\sigma\mathrm{Re}_н^{0,55}Bu^{-0,86}(l/d_e)^{-0,75}}, \tag{18}$$

A = numerical coefficient;
σ = the coefficient considering a way of combustion of fuel;
ξ = the same, design and condition of surfaces of heating in a fire chamber.

The number of integrated heat transfer of K_T estimates overall performance of a fire chamber in general and it makes the physical sense similar to Stanton's number, i.e., shows a ratio of the thermal stream apprehended by heating surfaces in a fire chamber and combustion, extremely possible on condition of cooling of products, up to the temperature of T_w of the heat perceiving surface:

$$K_T=\frac{H_l(\sigma\omega_н)c_p(T)(T_a-T_T^{''})}{H_l(\sigma\omega_н)c_p(T)(T_a-T_w)}=\frac{I_a-I_T^{''}}{I_a-I_w}, \tag{19}$$

H_l = surface of heating of a fire chamber, m^2;
$(\sigma\omega_н)$ = the mass speed of products of combustion carried to a surface of heating of a fire chamber, m/s;
$c_p(T)$ = specific average mass, isobaric thermal capacity, kJ/(kg·K);
T_a = theoretical temperature of burning, K;
$T_T^{''}$ = temperature of products of combustion at the exit from a fire chamber, K;
$I_a;I_T^{''};I_w$ = enthalpy of products of combustion respectively at theoretical to the burning temperature,

temperature at the exit from a fire chamber and a heat perceiving surface, kJ/kg .

The conditional number of $\mathrm{Re}_н$ reflects dynamics of transfer of masses from a gas stream on a heat perceiving surface and is unambiguously connected with the most important regime characteristic of work of a fire chamber - the thermal tension of volume of q_V.

The conditional number of $\mathrm{Re}_н$ is connected with Re number calculated in size of speed of a gas stream in fire chamber section, a ratio:

$$\mathrm{Re} = 4l / d_e \mathrm{Re}_н , \tag{20}$$

l = fire chamber length on the products of combustion flow course, m;

d_e = the defining size, m.

Buger's number characterizing conditions of radiation heat exchange taking into account the optical density of a stream of the radiating combustion products is determined by:

$$Bu = \overline{K^r} d_e , \tag{21}$$

$\overline{K^r}$ = Rosseland average absorption rate of combustion products, m^{-1};

4. CONCLUSIONS

The geometrical characteristic of a fire chamber is defined by the criteria l / d_e complex. As the defining temperature in the equation (18) the theoretical temperature of burning fuel at calculated values of coefficient of excess of air for this type of fire chambers is used. The equation (18) can be used for testing and constructive calculation of a fire chamber of a heat generator for sizes in the range of measurement of $K_т$= 0,15-0,67; $\mathrm{Re}_н$ = 55- 400; Bu = 0,25-1,1.

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