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THERMAL CALCULATION OF THE FURNACE CHAMBER OF A FIRE-TUBE BOILER WITH A DEAD-END FURNACE

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An engineering procedure for calculating the thermal regime of the furnace chambers of fire-tube boilers with a dead-end furnace is given. The procedure proposed is devoid of many drawbacks that are inherent in the standard method for calculating boiler units and permits more accurate allowance for the special features of combined energy exchange in furnace chambers of boilers that differ significantly in type and size from traditional ones.

Introduction. In recent times much attention has been paid in many developed European countries (Germany, Sweden, Finland, etc.) to the development and production of new fire-tube steam and hot-water boilers with a power of 1 to 4 MW (Fig. 1). They find wide application in locality, factory, and municipal-services heating boiler rooms, replacing cast-iron hot-water boilers. This is due to their lower cost in comparison with the latter, higher efficiency (92–93%), higher degree of automation, and better maintainability. At the same time there is currently no sufficient standard-design base for developing and producing such equipment. Thus, for example, in the standard method for calculating boiler units very approximate allowance is made for the main regularities of heat exchange in the furnace, and for its thermal calculation to be correct a number of empirical corrections are required to be prescribed to determine the Boltzmann number, the emissivity of the furnace, and the nonuniformity of the temperature field in it. However all these corrections are known only for fully developed types of boilers or for boilers of similar types. In addition, in the creation of the standard method of thermal calculation of boiler units no explicit allowance is made for the configuration of the radiant-heat-absorbing surface and of the volume of the furnace chamber. Thus, in designing new power furnaces that differ in type and size from existing ones, in particular, for low-power fire-tube boilers with reversing (dead-end) or straight-line furnaces, thermal calculation by the standard method can yield results that differ significantly from the actual ones. In this connection, the authors propose a refined engineering procedure for calculating the thermal regime of furnace chambers of such boilers that is devoid of many drawbacks inherent in the standard method.

Mathematical Model. A schematic of the furnace chamber is given in Fig. 2. The process of heat exchange in it can be described by the steady-state energy equation [1]

$$\mathrm{div}\,(c_p\,\rho\vec{v}\,T\,(\vec{r}) - \lambda\,\mathrm{grad}\,T\,(\vec{r})) = Q\,(\vec{r}) - \mathrm{div}\,\vec{Q}_{\mathrm{r}}\,(\vec{r})\,. \qquad (1)$$

All the quantities contained in this equation depend on the coordinate $\vec{r}$ and the temperature T. For correct determination of the temperature field in the volume of the furnace chamber from Eq. (1), it is necessary to know the distribution of the velocities $\vec{v}(\vec{r})$ and the heat $Q(\vec{r})$ and radiation $Q_{\mathrm{r}}(\vec{r})$ sources.

The turbulent field of the motion of the mixture of molecular gases can be calculated based on the time-averaged Navier–Stokes equations [2] using a two-parameter k–ε model of turbulence [3]. The system of these equations can be represented formally as the generalized equation

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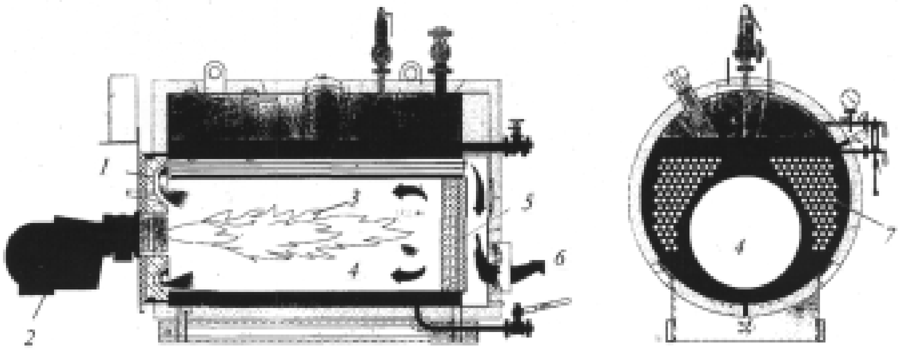

Fig. 1. Overall view of a universal fire-tube boiler with a dead-end (reversing) furnace: 1) front shield; 2) burner; 3) flame; 4) furnace chamber; 5) dead-end shield; 6) flue gases; 7) fire tubes.

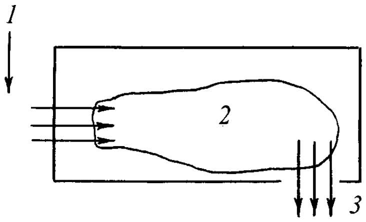

Fig. 2. Schematic of a furnace chamber: 1) fuel mixture; 2) flame; 3) flue gases.

$$\text{div}\,(\rho\vec{v}F(\vec{r}) - D_F\,\text{grad}\,F(\vec{r})) = S_F(\vec{r})\,, \tag{2}$$

where $\vec{v}(\vec{r}) = (v_x, v_y, v_z)$ is the velocity; $F(\vec{r}) = (v_x, v_y, v_z, k, \varepsilon, 1)$ is the generalized variable; D_F and S_F are the diffusion coefficient and the source of F. In a numerical solution of system (2), to calculate the velocity near the solid wall, use is usually made of the method of "wall" functions, which is based on the experimentally established universal character of the velocity and temperature profile in the turbulent boundary layer.

The density of the radiation sources is determined from the solution of the radiation transfer equation, which, given local thermodynamic equilibrium, has the following form in a nonscattering medium characteristic of gas combustion [4]:

$$\vec{l}\cdot\nabla I_\nu(\vec{r},\vec{l}) + (\chi_\nu + \sigma_\nu)\,I_\nu(\vec{r},\vec{l}) = \chi_\nu B_\nu(T) + \\ + \frac{\sigma_\nu}{4\pi}\int\limits_{4\pi} p_\nu(\vec{r},\vec{l},\vec{l}')\,I_\nu(\vec{r},\vec{l}')\,d\Omega'\,. \tag{3}$$

The divergence of the radiant fluxes at each point of the furnace medium is determined from the formula [5]

$$\text{div}\,\vec{Q}_{\text{r}} = \int\limits_0^\infty \chi_\nu(\vec{r})\left(4\pi B_\nu(T(\vec{r})) - \int\limits_{4\pi} I_\nu(\vec{r},\vec{l})\,d\Omega\right)d\nu\,. \tag{4}$$

Each of the problems indicated is rather complicated and time-consuming to solve. Therefore in calculating actual furnace units use is made of various simplifications. At the same time, it is necessary to preserve the contribution of the prevailing mechanisms of energy transfer – convection and radiation. Since the overwhelming role in furnace processes (up to 90%) is played by energy transfer by radiation [6], it is of particular importance to allow most correctly for the contribution of the radiation component of heat exchange.

To simplify the mathematical model, let us integrate the energy transfer equation (1) with respect to the volume:

$$\iiint_V \operatorname{div}\,(c_p\,\rho \vec{v} T - \lambda\,\operatorname{grad}\,T)\,dV = \iiint_V (Q - \operatorname{div}\,\vec{Q}_r)\,dV\,. \tag{5}$$

Using the Gauss divergence theorem and taking into account the continuity conditions $\rho_{en} v_{en} S_{en} = \rho_{ex} v_{ex} S_{ex} = G$ and the adhesion effect on solid surfaces, Eq. (5) can be transformed:

$$Gc_p\,(T_a - T_{ex}) = \iint_S \alpha\,(T - T_w)\,dS + \iiint_V \operatorname{div}\,\vec{Q}_r\,dV = Q_{wc} + Q_{wr}\,. \tag{6}$$

If in this equation the term Q_{wc} is replaced by the coefficient of conservation of heat φ and the radiation flux to the walls is set equal to $Q_{wr} = \varepsilon\sigma_0 S(T_{ef}^4 - T_w^4)$, we obtain the well-known procedure of the F. Dzerzhinskii All-Union Heat Engineering Institute and the G. M. Krzhizhanovskii Institute of Power Engineering (AUHEI–IPE) [6]: $\varphi G c_p(T_a - T_{ex}) = \varepsilon\sigma_0 S(T_{ef}^4 - T_w^4)$. The effective temperature of the furnace medium T_{ef} is determined from the relation $T_{ef} = T_{ex}\left(1 + \sum_{i=1}^{n} \Delta_i\right)$ in terms of empirical corrections Δ_i for the grade of fuel, location of the burners, etc.

The AUHEI–IPE procedure [6] for calculating the temperature of the escaping flue gases is a component of the standard method for the thermal calculation of boiler units [7]. It is based on a calculational formula that relates the temperature of the gases at the exit from the furnace to the Boltzmann number Bo, the emissivity of the furnace ε, and a certain parameter M that characterizes the nonuniformity of the temperature field: $T_{ex}/T_a = \mathrm{Bo}^{0.6}/(M\varepsilon^{0.6} + \mathrm{Bo}^{0.6})$. This procedure takes very approximate account of the main regularities of heat exchange in the furnace and requires for its use the specification of a number of empirical corrections to determine Bo, ε, and M_0. As has been indicated above, all these corrections are known only for developed types of boilers or for boilers of similar type. In designing new power furnaces that differ significantly from known ones in type and size, calculation by the standard method, because of the absence of corrections that correct serious simplifications of the heat-exchange processes, can yield results that differ significantly from the actual ones.

Determination of the Temperature of the Escaping Flue Gases. Let us consider an idealized furnace in which the feed of the fuel mixture to the volume and the removal of the combustion products occur simultaneously throughout the entire volume. Then the local temperature at any arbitrarily selected point of the volume $T(\vec{r})$ is the local temperature of the escaping flue gases. On condition of subsequent total mixing, which actually does occur at the exit section of the furnace chamber, and for a dead-end furnace – in its volume, too, the temperature of the escaping flue gases is none other than the average value of the local temperatures of the escaping gases in the furnace volume:

$$T_{ex} = \frac{1}{V}\iiint_V T(\vec{r})\,dV\,, \tag{7}$$

and hence Eq. (6) can be represented as

$$\iiint_V (\rho_G c_p T_a - \rho_G c_p T(\vec{r}) - q_c - \operatorname{div}\,\vec{Q}_r(\vec{r}))\,dV = 0\,, \tag{8}$$

where $\rho_G = G/V$ is the volume density of the fuel-mixture flow rate, which, due to the assumptions made, is constant in the furnace volume, and

$$q_c = \frac{1}{V} \iint_S \alpha (T - T_w)\, dS . \qquad (9)$$

To take explicit account of radiation in Eq. (7) we write expression (4) for determination of the divergence of the radiant fluxes in the form

$$\text{div}\, Q_r (\vec{r}) = 4\tilde{\chi}\sigma_0 T^4 (\vec{r}) - \int_0^\infty \chi_\nu (\vec{r}) \int_{4\pi} I_\nu (\vec{r}, \vec{l})\, d\Omega d\nu . \qquad (10)$$

Here $\tilde{\chi} = \pi \int_0^\infty \chi_\nu B_\nu(T) d\nu / (\sigma_0 T^4)$ is the mean-integral coefficient of absorption of the furnace medium. To simplify the representation, we introduce the parameter $U = \int_0^\infty \chi_\nu(\vec{r}) \int_{4\pi} I_\nu(\vec{r}, \vec{l})\, d\Omega d\nu$. With allowance for the above, the equation for determination of local temperatures (7) will acquire the form

$$4\tilde{\chi}\sigma_0 T^4 (\vec{r}) + \rho_G c_p T (\vec{r}) = \rho_G c_p T_a - q_c + U . \qquad (11)$$

It is substantially nonlinear since $\tilde{\chi}$, q_c, and U are strongly dependent on the temperature of the medium and the heat-absorbing surface.

Determination of the Characteristics of Radiation Transfer. As has already been noted above, the radiation sources can be calculated directly from system of equations (4) and (5). The boundary conditions at point P for the radiation transfer equation that allow for reflection and radiation processes on the radiant-heat-absorbing furnace surfaces, whose optical properties are assumed to be diffusive, are written in the following manner:

$$I (P, \vec{l})\big|_{(\vec{l}, \vec{n})<0} = \varepsilon B (T_w (P)) + \frac{1 - \varepsilon}{\pi} \int_{2\pi} I (P, \vec{l}\,') \cdot (\vec{l}\,' \cdot \vec{n})\, d\Omega' . \qquad (12)$$

A rather large number of various methods of solving radiation transfer equation (3) with boundary conditions (12) are known by now: Monte Carlo method [8], approximation of spherical harmonics [9], method of radiation elements [10], method of characteristics [4, 11], zonal methods [12], and other methods. One recent direction in the procedure for solving the radiation transfer equation is a combination of the discrete-ordinate method [13] with the finite-difference method [14, 15] or the finite-element method [16, 17]. The popularity of this approach for solving the transfer equation is due to the relative simplicity of the computational algorithm and its compatibility with computational schemes for other energy-transfer mechanisms. Nevertheless, at the First International Symposium on Radiation Transfer [18], held in August 1995, at which the current state of research in the field of radiation transfer was discussed, it was noted that a sufficiently reliable and efficient method of calculating the transfer equation remains to be developed despite a number of new approaches to the solution of this problem. Each of the existing methods has its drawbacks and a limited area of application.

In the present work, we propose a new approach to the calculation of radiation-transfer problems that assumes the use of a piecewise-analytical solution of the transfer equation (3) and (12) in its numerical solution. The results of application of this approach to the solution of a number of practical problems [19, 20] have shown that it is free of many drawbacks inherent in other methods and has a number of advantages that make it possible to extend the range of solved practical problems associated with energy transfer by radiation. Ensuring greater accuracy and speed of solution, this method requires significantly reduced computer resources for its implementation.

The basis of the method proposed is a combination of the discrete-ordinate method [13-17] and the ray-tracing method [11, 20], where the radiation intensity is determined along a ray trajectory with account taken of the optical and geometric properties of the medium and the boundary surface by using piecewise-analytical solutions of Eq. (3). The concept of the finite-element method [17, 21] is used for spatial discretization of the calculation region, which makes it possible to describe complex configurations and maintain compatibility with computational schemes for other mechanisms of energy transfer. A detailed description of the proposed method for solution of radiation-transfer problems can be found in [22].

A procedure for calculating spectral coefficients of absorption and scattering of the furnace medium that offers high accuracy with small expenditures of computer resources is presented in detail in [4]. It is expedient to calculate the characteristics of radiation heat exchange within the range of wavelengths λ = 1–6 μm, where the main radiative power of the combustion products is concentrated.

In solution of the transfer equation according to the procedures indicated we determine:

1) the mean-integral coefficient of absorption of the furnace medium

$$\tilde{\chi} = \pi \int_0^\infty \chi_\nu B_\nu (T)\, d\nu / (\sigma_0 T^4) ;$$

2) the parameter

$$U = \int_0^\infty \chi_\nu (\vec{r}) \int_{4\pi} I_\nu (\vec{r}, \vec{l})\, d\Omega d\nu ;$$

3) the local densities of the resultant radiation flux on the furnace-chamber walls

$$q_{\rm wr} (P) = \int_0^\infty \varepsilon \left(\int_{2\pi} I_\nu (P, \vec{l}) \cdot (\vec{l} \cdot \vec{n})\, d\Omega - \pi B_\nu (T_{\rm w} (P)) \right) d\nu ,$$

which are used in subsequent calculations to determine the wall temperature.

Taking Account of Convective Heat Exchange in the Furnace Chamber. Convective heat sinks $q_{\rm c}$ within the framework of the adopted model are calculated from formula (9). Determination of the coefficient of heat transfer α from the flue gases to the furnace wall is carried out according to the standard method [7]:

$$\alpha = \begin{cases} 0.15\zeta \dfrac{\lambda}{D} \mathrm{Pr}^{0.33} \mathrm{Re}^{0.43} , & \mathrm{Re} < 2000 ; \\ 0.023\zeta \dfrac{\lambda}{D} \mathrm{Pr}^{0.4} \mathrm{Re}^{0.8} , & \mathrm{Re} > 2000 . \end{cases} \quad (13)$$

Here ζ is a correction factor [7] that depends on the ratio of the length (height) of the furnace and its diameter D; $\mathrm{Re} = \tilde{v}D/\nu$ and $\mathrm{Pr} = \nu/a$ are the Reynolds and Prandtl numbers, respectively; ν is the kinematic viscosity of the medium. It is proposed that the velocity of flow of the gas mixture about the furnace-chamber surfaces within the framework of the adopted model be calculated from the formula

$$\tilde{v} = \xi \frac{D}{\tilde{\rho} S_{\rm cs}} , \quad (14)$$

where $\tilde{\rho}$ is the average density of the gases in the furnace chamber. For straight-line furnaces the correction factor is ξ = 1, and for reversing ones, ξ = 2.

The density of the local convective heat flux to the boundary at point P is determined by the expression

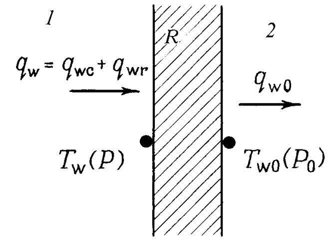

Fig. 3. Calculation of the wall temperature: 1) furnace; 2) heat-transfer agent.

$$q_{wc} = \alpha (P) (T (P) - T_w (P)) , \tag{15}$$

where $T(P)$ and $T_w(P)$ are the temperature of the furnace medium and the wall at point P, respectively.

Determination of the Temperature of the Furnace-Chamber Walls. Under conditions of absence, along the wall, of significant gradients of the flux incident on this wall and low thermal resistance of the wall, which are characteristic of furnace chambers, one can assume in determining the local temperatures along the wall that the total flux q_w incident on the wall at point P from the side of the flame (Fig. 3) is equal to the total flux q_{wc} removed from the opposite side by the heat-transfer agent. Then to determine the temperature at each point of the wall $T_w(P)$ and $T_{wo}(P_o)$ one can use the following system of equations:

$$q_w = \frac{T_w - T_{wo}}{R} = \alpha_o (T_{wo} - T_o) + \varepsilon_o \sigma_0 ((T_{wo})^4 - T_o^4) . \tag{16}$$

For the outside of the working surfaces of the furnace chamber the coefficient of heat transfer from the wall to the boiling water is calculated from nomograms of the standard method [7]. For the exterior surfaces about which flue gases or the ambient air flows (for example, the water wall (furnace shield) and the dead-end) one can use the following formula [23] to calculate the heat transfer coefficient:

$$\alpha_o = 0.6 \frac{\lambda}{H} (\mathrm{Gr\,Pr})^{0.25} , \tag{17}$$

where λ is the thermal-conductivity coefficient of the flowing gas; H is the characteristic dimension of the surface. In determining the Grashof number, one takes the difference between the wall temperature and the ambient temperature as the characteristic temperature drop. Values of the Prandtl number Pr, the kinematic viscosity ν, and the thermal-conductivity coefficient λ as functions of the temperature are also given in [23].

This approach makes it possible to calculate not only the temperature of the wall washed by water (a water-vapor mixture) but also the temperature of the water-wall (shield) surfaces in the furnace.

Procedure for Calculating the Thermal Regime of the Furnace Chamber. To calculate the temperature characteristics of the furnace chamber, discretization of its volume is done in accordance with the finite-element method [21]. Selection of the concept of this method is due to the possibility of description of complex configurations. As a result of the discretization we obtain a certain number of elements N_E and volume N_V and boundary N_B nodes of the subdivision (Fig. 4). At the volume nodes, all sought characteristics of the furnace medium (temperature, optical characteristics, divergence of the radiant fluxes, etc.) are determined. At the boundary nodes, the temperature of the wall inside and outside the furnace and the densities of the radiative and convective fluxes to the wall are calculated.

Since the system of equations for determining the thermal regime of the furnace chamber (3) and (7)-(17) is substantially nonlinear, it is solved by means of successive refinement of its constituent parameters by the iteration method. Here the computational algorithm includes the following stages:

1. At the initial instant we set $T_i = T_{en}$ for all volume nodes $i = 1 \ldots N_V$ and $T_{w,j} = T_o$ for all boundary nodes $j = 1 \ldots N_B$.

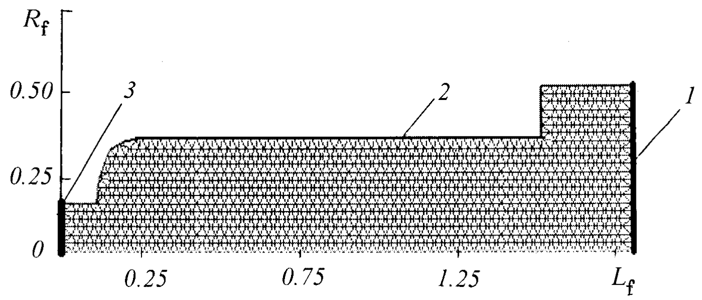

Fig. 4. Computational grid for determining the thermal regime of a furnace chamber: 1) front shield about which the ambient air flows; 2) walls of the furnace chamber washed by water; 3) dead-end shield about which flue gases flow.

2. In accordance with (14) the average velocity of motion of the furnace medium $\tilde{v}$ is determined, starting from which the coefficient of convective heat transfer α and the density of the convective flux (15) to the wall q_{wc} at the boundary nodes of the calculation region are calculated. Integrating q_{wc} over the surface we calculate the volume density of the convective heat sinks Q_{wc} in the volume of the furnace chamber (9).

3. In the spectral range λ = 1–6 μm, the optical characteristics of the furnace medium are determined for each node of the calculation region as functions of the temperature at this node. Next we calculate the mean-integral absorption coefficient of the furnace medium $\tilde{\chi}_i$ over the spectral range for each node of the calculation region. The system of radiative transfer equations (3) and (12) is solved at each point of the spectrum, and integral values of U_i for all nodes i = 1 ... N_V and integral densities of the radiation flux q_{wrj} to the wall for all boundary nodes j = 1 ... N_B are calculated.

4. Solving Eq. (11) we determine new values of the temperature of the furnace medium T_i at each node of the calculation region I = 1 ... N_V.

5. From the system of equations (16) we calculate the temperature of the walls of the furnace chamber $T_{w,j}$ and $T_{wo,j}$ at each boundary node of the calculation region j = 1 ... N_B.

6. We determine the error of the coincidence of the temperature values in the neighboring iterations s and $s+1$:

$$\delta = \max \begin{cases} |T_i^{s+1} - T_i^s| , & i = 1 \ldots N_V , \\ |T_{w,j}^{s+1} - T_{w,j}^s| , & j = 1 \ldots N_B . \end{cases} \tag{18}$$

7. If $\delta > \delta_0$ (δ_0 is the assigned accuracy of calculation; usually $\delta_0 \approx 0.1^{o}$C is assigned) the calculation is repeated starting from point 2. Otherwise, the calculation is considered to be complete.

Results of Thermal Calculation of the Furnace Chamber of a Natural-Gas Fire-Tube Boiler with an Efficiency of 1 ton/h (KP-1.0-0.6G). The indicated boiler has a cylindrical shape with a reversing furnace chamber whose dimensions are given in Fig. 4. The fuel mixture (natural gas with air, the flow rate is G = 0.323 kg/sec, the adiabatic combustion temperature is $T_a = 1945^{o}$C) is fed through a burner located on the axis of the cylinder on the front shield (the boundary AB, Fig. 5), which is made of heat-insulating material and has thermal resistance R = 0.54 m$^{2}\cdot^{o}$C/W. Then a jet of gases strikes the dead-end shield (the boundary JK, Fig. 5), also made of fire-resistant material with thermal resistance R = 0.02 m$^{2}\cdot^{o}$C/W, and, coming back, washes the heat-absorbing surface of the furnace (the boundary BCDEFJ, Fig. 5). The wall about which the heat-transfer agent flows is made of steel and has thermal resistance R = 0.0002 m$^{2}\cdot^{o}$C/W.

The processes of heat exchange and combustion in the furnace chamber can be considered to be axisymmetric with a high degree of accuracy. In this connection it is sufficient to consider only half the radial cross section of the furnace chamber (see Fig. 4). Results of calculating the temperature of the escaping flue gases, the temperature field in the furnace chamber, and the distribution of the heat fluxes over its interior surface are given in Figs. 5 and 6. The results of the numerical calculations showed that the total heat flux on

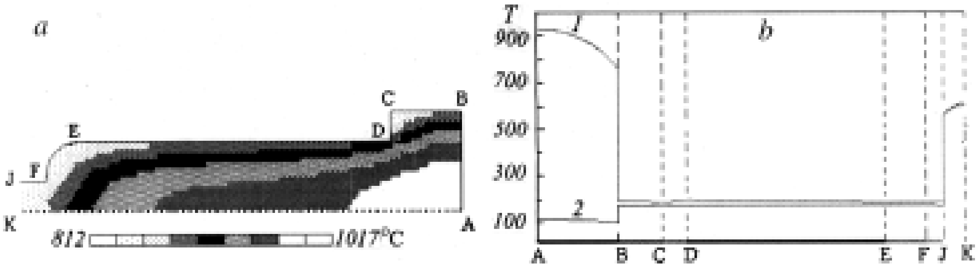

Fig. 5. Thermal regime of the furnace chamber of a KP-1.9-0.6G boiler: temperature distribution inside the furnace (a) and on its interior (1) and exterior (2) surface (b). T, ^oC.

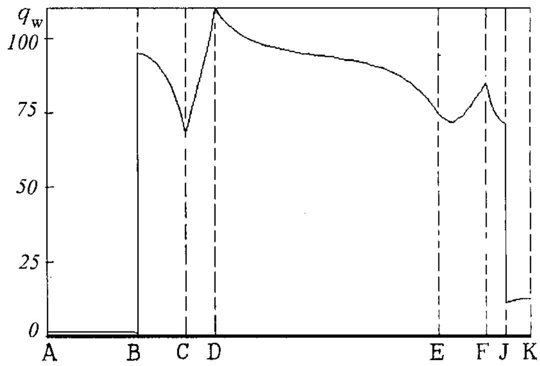

Fig. 6. Density distribution of the resultant heat flux on the heat-absorbing surfaces of the furnace chamber of a KP-1.0-0.6G boiler. q_w, kW/m^2.

the wall of the furnace chamber is Q_w = 425.9 kW. The main contribution is accounted for by the radiation component (82%). The share of the convective component is 18%.

As a result of the calculations done it was established that the temperature of the escaping flue gases was 929^{o}C. We note that, according to the standard method [7], its value is 1108^{o}C. As regards experimental measurements, they showed that the temperature of the escaping flue gases equaled 910^{o}C.

A comparison of the presented results demonstrates that for this type of boiler the proposed procedure is more correct. The calculation shows that, despite the high velocities of motion of the furnace medium in the dead-end furnace and the high degree of turbulization of the flow, the prevailing mechanism of energy transfer is radiation, which accounts for over 80% of the total heat removal in the furnace chamber. The distribution of the density of the resultant heat flux on the heat-absorbing surfaces of the furnace chamber given in Fig. 6 shows that the most loaded portion of the heat-absorbing surface is located near the entry of the flue gases into the fire tubes (the boundary CD) because of which this portion of the boundary is, as was confirmed in the process of boiler operation, the most accident-prone.

Conclusion. Although the proposed procedure for calculating the thermal regime of furnace chambers gives a somewhat distorted picture of the temperature field inside the chamber due to the simplifications adopted, it has a number of advantages over the standard method. The mathematical model of the furnace is closed (does not require the specification of correction factors) and makes it possible, taking into account the configuration of the furnace volume and selective properties of the furnace medium and the heat-absorbing surface, to determine with a high degree of reliability the convective and radiative components of the heat exchange, the qualitative structure of the temperature field in the furnace chamber, and the local thermal loads on the heat-absorbing surfaces of the furnace and to investigate the effects of the optical and thermophysical properties of the furnace medium and the walls on the level of the latter's environmental contamination. Taking all this into account, one can recommend that the procedure proposed be used as an instrument to determine the optimum structural and regime parameters of furnace chambers of low-power fire-tube power boilers.

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NOTATION

V, volume of the furnace chamber, m^3; D, diameter of the furnace chamber, m; S, area of the interior surface of the furnace chamber, m^2; S_{cs}, cross-sectional area of the furnace chamber, m^2; R_f and L_f, radius and length of the furnace chamber, respectively, m; λ, c_p, and ρ, coefficients of thermal conductivity and heat capacity and density of the medium, respectively; v, velocity of motion of the medium, m/sec; T, temperature, ^oC; $\vec{r}$, radius vector, m; μ, radiation wavelength, μm; $I_\nu(\vec{r}, \vec{l})$, spectral intensity of radiation in the direction $\vec{l}$ at the point of the medium $\vec{r}$; $B_\nu(T)$, spectral intensity of black-body radiation at the temperature T; $p_\nu(\vec{r}, \vec{l}, \vec{l}')$, scattering indicatrix of radiation in its interaction with the volume element of the medium at the point $\vec{r}$; χ_ν and σ_ν, spectral absorption and scattering coefficients of the medium, respectively; T_{en}, ρ_{en}, and v_{en}, temperature, density, and rate of feed of the fuel mixture to the entrance cross section S_{en} of the furnace chamber; T_{ex}, ρ_{ex}, and v_{ex}, temperature, density, and velocity of outflow of flue gases from the exit cross section S_{ex} of the furnace chamber; G, flow rate of the fuel mixture, kg/sec; T_w, temperature of the interior surface of the furnace chamber, ^oC; T_{wo}, temperature of the exterior surface of the furnace chamber, ^oC; T_o, temperature of the heat-transfer agent that flows about the exterior surface of the furnace chamber, ^oC; T_a, adiabatic temperature in combustion of the fuel, ^oC; T_{ef}, effective temperature of the furnace medium in the procedure of the F. Dzerzhinskii All-Union Heat Engineering Institute and the G. M. Krzhizhanovskii Institute of Power Engineering, ^oC; Q, volumetric heat release due to combustion, W/m^3; Q_{wc} and Q_{wr}, total heat fluxes on the furnace surface produced by convection and radiation, W; q_c, mean-integral volume density of the heat sinks due to convection, W/m^2; q_{wr}, local density of the radiation flux to the furnace-chamber surface, W/m^2; q_{wc}, local density of the convective flux to the furnace-chamber surface, W/m^2; q_w, local density of the resultant heat flux to the furnace-chamber surface, W/m^2; α, coefficient of convective heat transfer from the medium flux to the furnace walls, $W/(m^2 \cdot {}^oC)$; α_o, coefficient of convective heat transfer from the exterior surface of the chamber to the ambient medium, $W/(m^2 \cdot {}^oC)$; ε, emissivity of the interior surface of the furnace chamber; ε_o, emissivity of the exterior surface of the furnace chamber; $\sigma_0 = 5.668 \cdot 10^{-8}$, Stefan–Boltzmann constant, $W/(m^2 \cdot K^4)$; P, point on the interior surface of the furnace chamber; P_o, point on the exterior surface of the furnace chamber; R, thermal resistance, $m^2\ {}^oC/W$; H, characteristic dimension, m; Bo, Re, Gr, and Pr, Boltzmann, Reynolds, Grashof, and Prandtl numbers, respectively. Subscripts: en, entrance cross section; ex, exit cross section; cs, cross section; c, convective component; r, radiative component; w, surface; o, external medium; i and j, indices of the computational grid; f, furnace chamber.

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