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HEAT CONDUCTION AND HEAT EXCHANGE IN TECHNOLOGICAL PROCESSES

MATHEMATICAL SIMULATION OF CONJUGATE HEAT EXCHANGE IN HEATING FURNACES WITH A MOVING BOTTOM

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A mathematical model of conjugate heat exchange in heating furnaces with a moving bottom (ring furnaces and walking-beam furnaces) has been developed. The model allows one to determine the heating of steel blanks in these furnaces with account for all mechanisms of heat transfer in the high-temperature working region of a furnace, in its lining, and in the steel blanks. On the basis of this model, a more economical regime of operation, as compared to the existing ones, has been calculated for a walking-beam furnace and comparative analysis of the operating characteristics of a ring furnace and a walking-beam furnace has been performed.

The treatment of metal blanks in a heating furnace calls for a large amount of heat energy. Despite the long history of development of the technology for designing and maintaining such furnaces, the regimes of their operation remain not optimum in many cases. At the same time, estimates show that an increase in the fuel utilization factor of only one heating walking-beam furnace by several percent and (or) a decrease in the amount of scale formed on metal blanks treated in it by approximately 1 kg/ton makes it possible to save considerable cash resources (hundreds of thousands of U.S. dollars a year depending on the furnace output). On the one hand, it is very difficult to experimentally determine the optimum operating conditions of a metallurgical furnace because of the complex heat exchange arising in its high-temperature medium between the smoke gases, the furnace lining, and the heated metal and the large number of complex-geometry bodies participating in the heat exchange. On the other hand, such experiments are very expensive. The indicated problem can be solved much more simply by simulation of conjugate heat exchange in a furnace on the basis of a mathematical model of the process being considered. Unfortunately, the existing engineering models [1–4] do not provide the required accuracy of calculations: 5–10^{o}C. Direct numerical simulation of the processes occurring in a large three-dimensional volume of a furnace calls for unwarranted computational resources, which prevents the parametric study of the problem and the determination of optimum operating conditions of the furnace for times acceptable for practice.

In the present work, we propose a model of conjugate heat exchange in heating furnaces, which allows one to calculate the temperature regimes of these furnaces for a comparatively small time and with the required accuracy. On the basis of this model, comparative analysis of the temperature regimes of a ring furnace and a walking-beam furnace has been performed.

Model of the Temperature Regime of a Furnace. The heat exchange between the smoke gases and the heat-absorbing surfaces of the lining and a metal in the working region of a walking-beam furnace is defined by the nonstationary energy-transfer equation [5]

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$$c_{pg}\,\rho_g \frac{\partial T_g(\mathbf{r})}{\partial t} + \mathrm{div}\left[c_{pg}\,\rho_g \vartheta(\mathbf{r})\, T_g(\mathbf{r}) - \lambda_g \,\mathrm{grad}\, T_g(\mathbf{r})\right] = q_c(\mathbf{r}) - \mathrm{div}\,\mathbf{q}_{\mathrm{rad}}(\mathbf{r}) \,. \quad (1)$$

The coefficients of Eq. (1) depend on the coordinate $\mathbf{r}$ and the temperature of the smoke gases T_g. To solve this equation, it is necessary to determine the distribution of velocities $\vartheta(\mathbf{r})$ of gas flows over the furnace volume as well as the volume density of heat (gas combustion and metal oxidation) $q_c(\mathbf{r})$ and radiation div $\mathbf{q}_{\mathrm{rad}}$ sources.

The turbulent velocity field of a gas mixture can be calculated with an acceptable accuracy on the basis of time-averaged Navier–Stokes equations [6] with the use of a two-parameter k–ε model of turbulence [7]. A system of these equations can be represented in the generalized form

$$\mathrm{div}\left(\rho_g \vartheta F(\mathbf{r}) - \Gamma_F \;\mathrm{grad}\, F(\mathbf{r})\right) = S_F(\mathbf{r}) \,. \quad (2)$$

The volume density of radiation heat sources div $\mathbf{q}_{\mathrm{rad}}$ is determined from the radiation-transfer equation. The peculiarities of solving this equation for a furnace medium are described in [8, 9]. Note that the spectral coefficient of scattering of smoke gases in metallurgy furnaces working on natural gas can be assumed to be equal to zero ($\sigma_\lambda = 0$) because these gases can be scattered only by soot particles, the number of which is small in the case where the combustion process is properly organized. In this case, the divergence of the radiant fluxes at each point of the furnace space is determined by the relation

$$\mathrm{div}\,\mathbf{q}_{\mathrm{rad}} = \int_0^\infty \chi_\lambda(\mathbf{r}) \left(4\pi B_\lambda(T_g(\mathbf{r})) - \int_{4\pi} I_\lambda(\mathbf{r}, \mathbf{l})\, d\Omega\right) d\lambda \,. \quad (3)$$

The above-described mathematical model allows one to simulate the gas dynamics, the radiation transfer, the inner heat sources formed as a result of the combustion of a fuel and the exothermic oxidation of the surface of treated metal blanks, and the temperature field of the medium in a furnace. In this formulation, only the combustion of a fuel is not defined in detail. The heat released as a result of the combustion is defined by mean-volume characteristics. Each of the indicated problems is complex and its solution is time consuming. For example, the length of the tunnel of a ring furnace is equal to 60 m and about 150 steel blanks can be treated in it at a time. Direct numerical simulation of temperature regimes in this system calls for a three-dimensional regular grid with a step of the order of 10 mm and approximately 315 million computational nodes. In this case, it is impossible to calculate even one temperature regime, much less determine the optimum design and regime parameters of the furnace. Therefore, actual furnaces are calculated with the use of different simplifications; however, the contribution of the prevailing mechanisms of energy transfer — convection and radiation — should be retained in this case. Since the energy transfer dominates in the processes occurring in a furnace (its contribution can comprise as much as 90%) [10], it is especially important to determine the contribution of the radiative component of the heat exchange. As was shown in [8, 9], the three-dimensional problem on radiative heat exchange in a furnace can be reduced to a two-dimensional one with an accuracy sufficient for engineering applications. However, in this case, it is impossible to calculate the heat exchange throughout the furnace volume for times reasonable for practice because this calculation calls for a number of computational nodes of the order of 10^6. Because of this, it makes sense to consider one or several blanks that are "pulled" through a furnace operating in a methodical regime. Since the temperature of the medium of a furnace changes insignificantly within the limits of one blank due to the design features (the furnace length is 60 m and the diameter of the blanks is 100–300 mm) and the problem considered is symmetric in space, we can restrict our consideration to two blanks, as shown in Fig. 1 (in this case, the left and right boundaries of the computational region are assumed to be mirror).

The approach proposed involves the organization of an iteration process for verification of the temperatures of the lining of the furnace and of a blank since the immobile roof of the furnace and its side walls are quasistationary, while the mobile furnace bottom and blank are substantially nonstationary (because of the step-type movement of the bottom). The roof and the walls of the furnace have a minimum temperature at the sites of charging of a blank, where the temperature of the smoke gases is minimum. At the same time, here, the bottom of a ring furnace has a high temperature since, during the previous time interval, it was in the soaking zone where the temperature in the cross section of a blank is equalized. Stationary thermophysical conditions are established in a furnace with a bottom moving continuously with a constant velocity after the furnace characteristics reach their operating values. In this case, the itera-

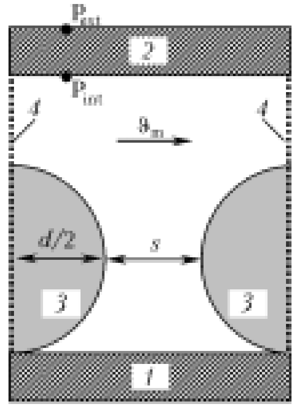

Fig. 1. Computational scheme for determining the temperature regime of a furnace: 1) furnace bottom; 2) furnace roof; 3) steel blanks; 4) mirror boundaries.

tion procedure of refining the temperature of the bottom and the walls can be formulated in the following way: at each point of the furnace the temperature field in the lining of the roof and the walls is calculated at each instant of time until the stationary conditions $T_w(t)$ = const are established. The temperature of the bottom of the furnace at the inlet to the methodical zone (i.e., the initial temperature distribution) is taken to be equal to the temperature at the end of the soaking zone. Then this temperature is calculated with account for the heat-exchange conditions.

As was noted above and was shown in many works (see, e.g., 10, 11]), in the high-temperature region of a furnace, the fraction of the convective energy exchange comprises 10–20% and radiation makes a main contribution to the formation of a temperature field. Because of this, the convective heat exchange of a gas flow with the heat-absorbing surfaces can be estimated to sufficient accuracy with the use of the Newton law

$$\mathbf{q}_{\text{conv}}(\mathbf{r}) = \alpha_g \left[T_g(\mathbf{r}) - T_w(\mathbf{r})\right] . \tag{4}$$

The coefficient α_g of convective heat exchange between the gas and the heat-absorbing surfaces of the furnace is determined according to [12] as

$$\alpha_g = \begin{cases} 0.15\zeta \dfrac{\lambda_g}{D_{\text{eff}}} \text{Pr}^{0.33}\text{Re}^{0.43} , & \text{Re} < 2000 ; \\ 0.023\zeta \dfrac{\lambda_g}{D_{\text{eff}}} \text{Pr}^{0.4}\text{Re}^{0.8} , & \text{Re} > 2000 . \end{cases} \tag{5}$$

The correction coefficient ζ appearing in (5) is of the order of unity [12]. This coefficient depends on the ratio between the length of the furnace and its effective diameter D_{eff}. The latter is determined in the following way for the problem considered:

$$D_{\text{eff}} = \sqrt{\frac{4}{\pi} BH - \frac{ld}{1 + s/d}} . \tag{6}$$

The coefficient α_g depends on the velocity ϑ_g of a gas-mixture flowing over the furnace surfaces (within the framework of the model used, the mean mass rate is scluted as ϑ_g):

$$\vartheta_g = \frac{4G_g}{\pi \tilde{\rho}_g D_{\text{eff}}^2} . \tag{7}$$

Integration of the energy-transfer equation (1) over the volume of the computational region (Fig. 1) with the use of the Ostrogradskii–Gauss theorem and the condition of adhesion of the flow to the walls of the furnace gives balance equations for calculating the temperature of the gas mixture as a function of a coordinate inside the furnace.

Let us divide the space of the furnace into elementary computational regions (see Fig. 1) by the number of blanks found in it at any given time. These regions are numbered beginning with the soaking zone, in which a window for removal of blanks is located. If, in any cross section of the furnace, a temperature inhomogeneity is absent, because of the good mixing of gases, and temperature differences arise only along the furnace length, the above-indicated balance relations lead to the system of equations [4] forming the basis for the zonal method of calculating the temperature regime of a furnace with a moving bottom:

$$c_{pg} T_{gi-1} G_{i-1} - c_{pg} T_{gi} (G_{i-1} + \delta G_i) + \delta G_i Q_{h.c} + G_{oxi} Q_{ox} = Q_{conv} + Q_{rad} . \qquad (8)$$

On condition that the smoke gases are formed in the furnace only due to the fuel supplied through burns, the rate of their flow through the ith zone is determined by the relation

$$G_i = \sum_{k=1}^{i} \delta G_k . \qquad (9)$$

The power of the total convective losses Q_{conv} on the heat-absorbing surfaces of the furnace, involved in (8), is determined as

$$Q_{conv} = \iint_{S_w} \alpha_g (T_g(x) - T_w(x)) \, dS_w . \qquad (10)$$

The third and fourth terms on the left side of (8) represent the total power of the heat released in the volume of the furnace as a result of the gas combustion and the metal oxidation respectively. The specific calorific value of the gas mixture is related to the adiabatic temperature by the relation $Q_{h.c} = c_{pg}(T_{ad} - T_h)$. According to [4], $Q_{ox} \approx K_1 Q_{Fe}$, where $Q_{Fe} \approx 5$ MJ/kg, and $K_1 \approx 0.778$ is a coefficient obtained on the assumption that a scale consists mainly of FeO. The rate of formation of the scale G_{ox} is determined by the dynamics of change in the thickness of the oxidized layer. Based on the Kazantsev relation [13] for the rate of increase in the thickness of the oxidized layer with increase in the temperature of the surface of a steel blank, we obtained the following expression for the rate of formation of a scale:

$$G_{ox} \approx \rho_{ox} S_m \frac{0.25 K_2^2}{\sqrt{\delta_0^2 + 0.5 K_2^2 \tau}} , \qquad (11)$$

where $K_2 = \exp\,[7.25 - 10{,}125/T_m]$ is a scale-formation constant at a metal-surface temperature T_m. Note that expression (11) was obtained on the assumption that air is present in excess and its concentration is not a limiting factor for the growth of an oxide layer.

The method described was developed for an ideal furnace, in which a gas mixture and smoke gases are uniformly supplied to a volume from other zones and are removed from each site of the volume at a time.

System (8) is widely used for calculating the temperature regimes of furnaces and boilers [14–18]. However, as all approximate methods, this approach has a number of limitations. In particular, the temperature in a furnace is assumed to be constant; however, in actuality, the temperature near the heat-absorbing surfaces is lower than the temperature of the other regions of the furnace. This leads to an overestimation of the calculated convective heat removal (4), to a large error in calculating the radiative heat removed from the zone considered, and, as a consequence, to an underestimation of the calculated temperature of the removed smoke gases and an overestimation of the calculated efficiency of the furnace. Consequently, the indicated approach gives an underestimated value of the flow rate of a fuel expended for thermal treatment of 1 ton of a metal as compared to the actual one. Moreover, because of the existence of temperature inhomogeneities in a furnace, in it there can arise local heat flows that can damage the lining of the furnace and cause thermal deformations of a metal blank.

In the present work, we propose a modified approach to determination of the temperature of the smoke gases in the ith zone of a furnace, which allows one to take into account the temperature inhomogeneity in this zone. Note that the approach proposed was used to advantage for calculating the temperature regimes of boilers [11]. As in the case of the zonal method, we will consider an ideal furnace, in which a gas mixture and smoke gases are supplied uniformly into the volume of the zone considered from other zones and are removed simultaneously from each site of this volume. In this case, the local temperature at each point of this volume $T(\mathbf{r})$ is the local temperature of the smoke gases removed from this point. On condition that the smoke gases are then completely mixed (this condition is fulfilled with a high degree of accuracy for the output cross section of the zone), the temperature of the removed smoke gases represents their local temperature averaged over the volume of the zone considered:

$$\tilde{T}_{gi} = \frac{1}{V} \iiint_V T_{gi}(\mathbf{r})\, dV . \tag{12}$$

Assuming, as in the zonal method, that the smoke gases are well mixed, we will consider the following mean-volume quantities: $q_{\text{conv}} = Q_{\text{conv}}/V$, $R = G/V$, $\delta R = \delta G/V$, $R_{\text{ox}} = G_{\text{ox}}/V$, and $U = \int_0^\infty \chi_\lambda(\mathbf{r}) \int_{4\pi} I_\lambda(\mathbf{r}, \mathbf{l}) d\Omega d\lambda$.

To estimate the influence of the radiation on the temperature field, we will rewrite expression for the divergence of radiant fluxes (3) in the form

$$\text{div}\, \mathbf{q}_{\text{rad}}(\mathbf{r}) = 4\tilde{\chi}\sigma_0 T^4(\mathbf{r}) - \int_0^\infty \chi_\lambda(\mathbf{r}) \int_{4\pi} I_\lambda(\mathbf{r}, \mathbf{l})\, d\Omega d\lambda . \tag{13}$$

The mean-integral absorption coefficient $\tilde{\chi}$ of the furnace medium at a temperature T in (13) is determined from the relation [9]

$$\tilde{\chi} = \pi \int_0^\infty \chi_\lambda B_\lambda(T)\, d\lambda / (\sigma_0 T^4) . \tag{14}$$

In this case, the energy equation (1) for the furnace zone considered will take the following form at any instant of time:

$$\iiint_V \left[c_{pg}(R + \delta R)\, T_{\text{g}} - c_{pg} R T_{\text{g,in}} + q_{\text{conv}} \right] dV = \iiint_V \left[\delta R c_{pg}(T_{\text{ad}} - T_{\text{h}}) + R_{\text{ox}} Q_{\text{ox}} + U - 4\tilde{\chi}\sigma_0 T_{\text{g}}^4(x) \right] dV . \tag{15}$$

From (15) we obtain an equation for determining the local temperatures inside the indicated zone in explicit form:

$$4\tilde{\chi}\sigma_0 T_{\text{g}}^4(x) + c_{pg}(R + \delta R)\, T_{\text{g}}(x) = c_{pg} R T_{\text{g,in}} + \delta R c_{pg}(T_{\text{ad}} - T_{\text{h}}) + R_{\text{ox}} Q_{\text{ox}} + U - q_{\text{conv}} . \tag{16}$$

This equation is nonlinear for each instant of time of heating of a blank (or, interchangeably, for the coordinate $x = \vartheta_{\text{m}}\tau$ inside the furnace) since $\tilde{\chi}$, q_{conv}, and U are strongly dependent on the temperature of the furnace medium and the heat-absorbing surfaces. In this case, one should take account of the fact that the roof and walls of the furnace are under stationary conditions at each instant of time and their internal and external temperatures can be determined using simple expressions for equality of heat flows [11], according to which the total flow $q_{\text{w,int}}$ incident on the furnace side at the point P_{int} is equal to the total flow $q_{\text{w,ext}}$ removed from the exterior of the furnace to the environment $q_{\text{w,int}} = q_{\text{w,ext}} = q_{\text{w}}$. Then the temperature at each point of the wall $T_{\text{w}}(\text{P}_{\text{int}})$ and $T_{\text{w}}(\text{P}_{\text{ext}})$ (see Fig. 1) can be determined from the equation

$$q_{\text{w}} = \frac{T_{\text{w,int}} - T_{\text{w,ext}}}{R_{\text{t}}} = \alpha_{\text{ext}}(T_{\text{w,ext}} - T_0) + \varepsilon_{\text{ext}}\sigma_0 (T_{\text{w,ext}}^4 - T_0^4) , \tag{17}$$

where $\alpha_{\text{ext}} = 0.6\frac{\lambda_{\text{g}}}{H}(\text{Gr Pr})^{0.25}$ [11].

The foregoing allows us to propose the following algorithm for determining the local temperatures within the zone allocated for one blank in a furnace:

1. The following parameters are assumed to be known at any instant of time t: the temperature of the smoke gases inflowing into the furnace $T_{g,in}$ and the density of their flow rate R, the adiabatic temperature of combustion T_{ad}, the temperature of the preliminary heating T_h and the flow-rate density of the gas mixture δR, the temperatures of the surface of a metal $T_m(x)$ and of the furnace bottom $T_{f.b}(x)$, and the thermal resistance R_t of the roof and walls of the furnace.

2. It is assumed that the local temperatures in the furnace region considered are constant ($T_g(x)$ = const) and the temperature of the outside surface of the furnace walls does not exceed the normative temperature, equal to 55^{o}C ($T_{w,ext} = T_{f,ext} = 328$ K).

3. The internal temperature of the furnace roof $T_{f.r,int}$ is determined in the first approximation from (17).

4. On the basis of the known temperature fields of the furnace, the metal, and the furnace bottom and walls, the following parameters are determined:

a) the density of convective heat flows q_{conv};

b) the mean-integral absorption coefficient of the furnace medium $\tilde{\chi}$;

c) the density of the radiant fluxes incident on the heat-absorbing surfaces and the value of U from the two-dimensional energy-transfer equation [8, 9];

d) the density of the scale formed R_{ox};

e) the internal and external temperatures (17) of the furnace roof $T_{f.r,int}$ and $T_{f.r,ext}$.

5. A new distribution of local gas temperatures $T_g(x)$ is determined by solving the nonlinear equation (16) for each point of the furnace zone considered.

6. The new local-temperature distribution at each point of the furnace medium is compared with that obtained earlier. If the maximum difference between them exceeds any definite value ΔT, the calculation is repeated beginning with item 4.

Note that, when the work of a ring furnace is simulated, the nonstationary heat-conduction equation is solved for its bottom. If the work of a walking-beam furnace is simulated, the temperature of its bottom is determined under quasi-stationary conditions, which allows one to determine this temperature from the balance of heat flows.

Comparison of Theoretical and Experimental Data. To verify the above-described mathematical model, we compared the results of calculations performed on its basis with the results of a natural experiment conducted on a 320 mill of the Belarusian Metallurgical Works. In the process of experimental investigations, blanks of St3 steel of cross section 125 × 125 mm were heated in the running-fit regime. Before a blank was charged into a furnace, six chromel-alumel thermocouples were caulked in holes drilled preliminary at the center of the cross section, on the upper and lower faces, and at the cones of the blank. In the process of heating, the following parameters were fixed in addition to the indications of the thermocouples: the efficiency of the furnace, the rates of the gas and air flows in zones of the furnace, the total gas and air flow rates, the temperature of the preliminary heated air supplied for combustion, the temperature of the gases at the output of the furnace, and the temperature and volume of the cooling water. This allowed us to determine the energy and technological operating characteristics of the heater. After the process was completed, scale samples were taken from the surface of blanks. Comparison of the calculated and experimental temperatures determined for different points of the cross section of a blank has shown that the error of calculations performed by the method proposed does not exceed 5%. The error in determining the loss of the metal did not exceed 1.5–3%. Thus, the mathematical model developed can be used for determining new economical regimes of heating of blanks in furnaces with a mechanized bottom.

It was established that, at an output of 100–120 tons/h, it makes sense to redistribute the thermal power between the weld zones (zones II and III) by displacement of the temperature maximum to the second weld zone. This makes it possible to fairly effectively influence the process of scale formation because the temperature of the surface of a metal reaches 850–900^{o}C at a later time; at higher temperatures, a scale begins to actively form. Calculations have shown that, when the output increases to 125–140 tons/h, a large increase in the flow rate of the gas burnt in the second weld zone and a decrease in the flow rate of the gas burnt in the first weld zone can cause a deformation of a blank in the process of running lift. This can be prevented in the case where a metal with an increased heat content (with an initial temperature of 100–130^{o}C) is charged into a furnace, which is possible in the case where continu-

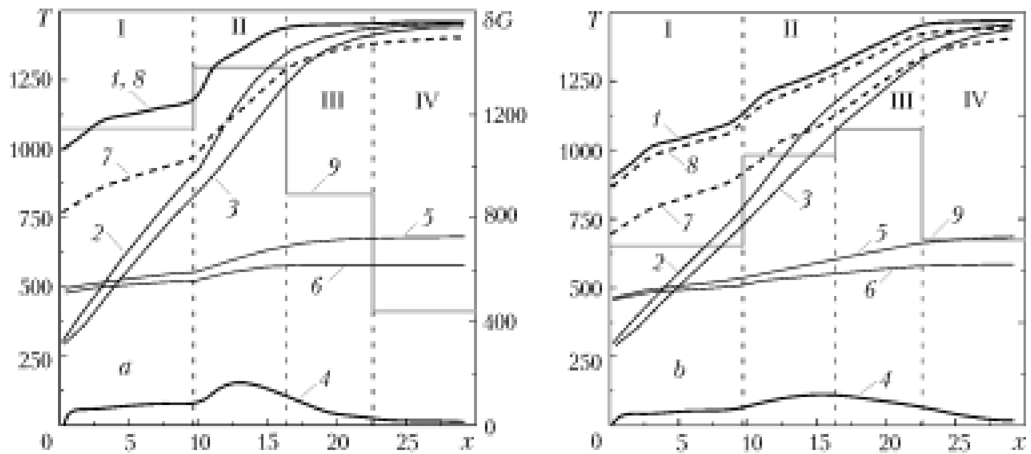

Fig. 2. Results of calculating the temperature regime of a walking-beam furnace by a production regime map (a) and the new regime of operation of this furnace (b): 1) temperature of the smoke gases; 2, 3) maximum and minimum temperatures of a metal blank; 4) maximum temperature difference in a metal blank; 5, 6) temperature of the outside surface of the bottom and roof of the furnace; 7, 8) temperature of the inside surface of the bottom and roof of the furnace; 9) flow rate of the gas burnt for heating of blanks in furnace zones [I–IV) furnace zones; vertical dashed lines) boundaries of the zones]. *T*, K; δ*G*, m^3/h; *x*, m.

TABLE 1. Difference between the Calculated Operating Parameters of the Heating Furnace of a 320 BMZ Rolling Mill and the Operating Parameters Used at Present

Diameter of a rolled blank, mm	Output of a furnace, tons/h	Decrease in the burning loss of metal		Decrease in the fuel flow rate, kg of eq.f/ton
		kg/ton	%	
10	100—110	1.7	26	1.6
12	100—110	1.7	26	1.6
14	118—120	0.9	11	1.8
25	130—140	0.6	8.5	1.8

ously casted blanks, which still did not lose their heat, are transported from a store house for a short time or immediately after the casting.

The new temperature regimes of heating in the furnace of a 320 RUP BMZ mill can be used most effectively in the rolling of small profiles (of diameter 10 and 12 mm) at an output of 100–110 tons/h (see Table 1). In this case, the velocity of movement of a metal in a furnace is relatively small and it is heated gradually with a small temperature gradient in its cross section, which makes it possible to organize a forced regime in the second weld zone. Moreover, a decrease in the temperature of the removed gases and the redistribution of the thermal power and the gas-flow rate between the weld zones makes it possible to decrease the flow rate of the fuel and the scale formation.

Figure 2 shows the data of calculation of the existing and new regimes of operation of a walking-beam furnace. The redistribution of volumes of the burned gas between the furnace zones made it possible to improve the following parameters: at a furnace output of 116 tons/h (blanks of 40Kh steel of square cross section 125 × 125 mm) the gas-flow rate was decreased from 3808 to 3607 m^3/h, the scale formation was decreased from 7.4 to 6.0 kg/ton, and the temperature of the removed smoke gases was decreased from 730 to 630^{o}C. As a result, the specific flow rate was decreased from 25.1 to 23.5 kg/ton. At the indicated output of a furnace and the existing prices for the natural gas and metals, only one furnace operating in the new regime makes it possible to economize 500,000 dollars a year.

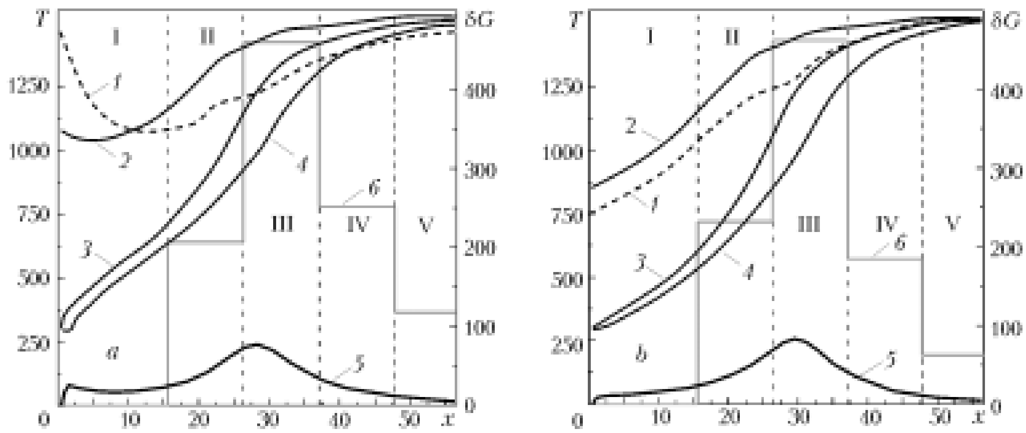

Fig. 3. Results of calculating the temperature regime of a ring furnace (a) and a walking-beam furnace identical in characteristics and sizes to the ring furnace (b): 1) temperature of the inside surface of the furnace bottom; 2) temperature of the smoke gases; 3, 4) maximum and minimum temperatures of a metal blank respectively; 5) maximum temperature drop difference in a metal blank; 6) flow rate of the gas burnt for heating of blanks in furnace zones [I–V) furnace zones; vertical dashed lines) boundaries of the zones]. T, K; δG, m^3/h; x, m.

Comparative Analysis of the Operating Regimes of a Ring Furnace and a Walking-Beam Furnace. Of great practical interest is comparison of the operating regimes of a walking-beam furnace and a ring furnace having analogous characteristics. A mathematical simulation carried out on the basis of the model developed allowed us to perform such a comparison. We considered a standard ring furnace with a tunnel of length 56.6 m. The internal space of the furnace was divided into zones (I–V) of length 15.3 m, 10.85 m, 10.85 m, 10.85 m, and 8.74 m respectively. The width of the furnace tunnel is equal to 3.5 m and its height is 1.57 m. The furnace bottom of thickness 300 mm is made from chamotte. The roof and walls consist of two layers: a mineral wool layer of thickness 50 mm and a chamotte layer of thickness 200 mm. The walking-beam furnace considered had the same sizes and thicknesses of the walls as the above-described furnace and a significantly different tunnel: the tunnel of the ring furnace had the form of a ring, and the tunnel of the walking-beam furnace was elongated along a straight line. Calculations were performed for a furnace output of 33.4 tons/h. The results of calculation of the temperature regimes of these furnaces are presented in Fig. 3. It was established that the heating of steel blanks in the walking-beam furnace is more economical. Practically equal amounts of a scale are formed in both furnaces: 6.8 kg of a scale per ton of metal in the ring furnace and 7.0 kg/ton in the walking-beam furnace. At the same time, they differ substantially in the gas-flow rate. Calculations have shown that the gas-flow rate is equal to 1035.4 m^3/h in the ring furnace and 939.9 m^3/h in the walking-beam furnace with analogous parameters. Such an operating parameter of a furnace as the gas-utilization coefficient is equal to 23.4 kg/ton of a metal in the first case and 21.2 kg/ton in the second case. The relative economy of the gas in the walking-beam furnace is more than 9% as compared to the ring furnace. The continuous character of the work furnaces of this type makes it possible to economize large resources for decreasing the gas-flow rate. Note that in the walking-beam furnace the temperature of the removed smoke gases (577^{o}C) is substantially lower than that of the ring furnace (803^{o}C). This also points to the fact that the walking-beam furnace offers an advantage in energy over the ring furnace. However, it does not always happen that the possibility of economical heating determines the advisability of using one furnace or another. For example, a ring furnace is more preferential for use for a number of factors, among which are the cost of engineering of a furnace and the possibility of its placement clear of the technological axis, which makes it possible to decrease the total length of a production shop.

CONCLUSIONS

1. A mathematical model of conjugate heat exchange has been developed. This model allows one to determine the heating of steel blanks in ring furnaces and walking-beam furnaces. The model accounts for, in different approxi-

mations, all mechanisms of heat transfer in the high-temperature operating space of a furnace and in its lining and steel blanks. A peculiarity of the model proposed, by which it differs from the existing analogs, is that it provides a much higher accuracy of calculating the thermal-radiation transfer.

2. On the basis of the model proposed, we have developed a computer program that allows one to calculate one temperature regime for 30 min on a personal computer having a medium output with a processor operating at a clock frequency of 1 GHz.

3. Comparison of the data of simulation and the data of regime-adjustment testing has shown that the model proposed defines, with an accuracy sufficient for practice, the temperature regimes of operation of a walking-beam furnace with an output of 100–130 tons of blanks in an hour.

4. An economical regime of operation of a walking-beam furnace has been calculated and comparative analysis of the operating characteristics of a ring furnace and a walking-beam furnace has been performed. It has been established that the heating of steel blanks in a walking-beam furnace is more economical.

NOTATION

$B_\lambda(T)$, spectral intensity of the blackbody radiation; B and H, width and height of the internal space, m; c_{pg}, heat capacity of the furnace medium, J/(kg·K); $D_{\rm eff}$, effective diameter of a furnace; d and l, diameter and length of a metal blank; $F(\mathbf{r})$, generalized variable; $G_{\rm g}$ and $\tilde{\rho}_{\rm g}$, flow rate and average density of the smoke gases in the furnace cross section considered; $G_{\rm ox}$, rate of formation of a scale, kg/sec; G, flow rate of the smoke gases passing through a control volume, kg/sec; δG, additional flow rate of the smoke gases that are due to the gas mixture supplied through the control volume, kg/sec; Gr, Grashof number; $I_\lambda(\mathbf{r}, \mathbf{l})$, spectral intensity of radiation at the point $\mathbf{r}$ in the direction $\mathbf{l}$; K_2, scale-formation constant at a metal-surface temperature $T_{\rm m}$; P, point on the inner or outer surface of the furnace wall; Pr, Prandtl number; $Q_{\rm conv}$ and $Q_{\rm rad}$, total density of the convective and radiation losses on the heat-absorbing surfaces of a furnace; $Q_{\rm h.c}$, specific heat of a gas mixture, J/kg; $Q_{\rm ox}$, specific calorific value of the scale formation; $Q_{\rm Fe}$, specific calorific value of the iron oxidation; $q_{\rm c}(\mathbf{r})$ and div $q_{\rm rad}(\mathbf{r})$, volume density of heat (gas combustion and metal oxidation) and radiation sources, W/(m^3·sec); $q_{\rm w,int}$, total flow incident on the inner wall of a furnace at the point P; $q_{\rm w,ext}$, total flow removed from the outer surface of a furnace to the environment; $q_{\rm conv}$, mean-volume density of convective heat flows on the heat-absorbing surfaces, W/m^3; R and δR, volume density of the flow rate of the smoke gases and gas mixture flowing through the zone considered, kg/(m^3·sec); $R_{\rm ox}$, volume density of scale formation, kg/(m^3·sec); $R_{\rm t}$, thermal resistance of the roof and walls of a furnace; $\mathbf{r}$, radius-vector of a point; $\mathrm{Re} = \vartheta_{\rm g} D_{\rm eff}/\eta$, Reynolds number; S_F, source of the quantity F; s, distance between blanks, m; $S_{\rm m}$, area of the surface of a metal blank, m^2; $S_{\rm w}$, area of the furnace-wall surface, m^2; t, time, sec; T, temperature; $T_{\rm h}$, temperature of a gas mixture before it is supplied to a furnace for combustion; $T_{\rm ad}$, adiabatic temperature of combustion of a gas mixture; $\tilde{T}_{{\rm g}i}$, average local temperature of the removed gases in the volume of the ith zone; T_0, ambient temperature; $T_{\rm g,in}$, temperature of the smoke gases inflowing to the volume; V, volume of a furnace zone, m^3; x, coordinate inside a furnace, m; $\alpha_{\rm g}$, coefficient of convective heat exchange between the gas and the heat-absorbing surfaces of a furnace; $\alpha_{\rm ext}$, heat-transfer coefficient of the outer surface of a furnace; Γ_F, diffusion coefficient of the scalar quantity F; δ_0, initial thickness of the oxidized layer before a metal is placed in the furnace, m; $\varepsilon_{\rm ext}$, blackness of the outer surface of the furnace walls; $\vartheta_{\rm m}$, linear velocity of movement of a blank, m/sec; $\vartheta(\mathbf{r})$, velocity of a gas at the point $\mathbf{r}$; $\vartheta_{\rm g}$, velocity of a gas mixture flowing over the surface of the furnace chamber, m/sec; σ_λ, spectral coefficient of scattering of smoke gases; $\sigma_0 = 5.68 \cdot 10^{-8}$ W/(m^2·K^4), Stefan–Boltzmann constant; η, kinematic viscosity of smoke gases; ζ, correction coefficient; $\rho_{\rm ox}$, density of a scale, kg/m^3; $\rho_{\rm g}$, density of the furnace medium, kg/m^3; $\lambda_{\rm g}$, effective (with account for turbulent pulsations) coefficient of heat conduction of the furnace medium, W/(m·K); λ, wavelength of electromagnetic radiation, nm; τ, time of treatment of a metal in a furnace, sec; $\tilde{\chi}$, mean-integral absorption coefficient of the furnace medium; χ_λ, spectral absorption coefficient; Ω, solid angle. Subscripts: ad, adiabatic; conv, convective; c, combustion; eff, effective; ext, exterior; f, furnace; f.b, furnace bottom; f.r, furnace roof; g, smoke gases of the furnace medium; h.c, combustion heat; h, preliminary heating; i, ordinal number of a furnace zone; in, inflow to the volume; int, interior; m, metal; ox, scale formation; rad, thermal radiation; t, thermal; w, wall.

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