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MATHEMATICAL MODELING OF HEAT TRANSFER IN SCREENED FURNACES OF RADIAL—CYLINDRICAL AND BOX TYPE

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A zonal model of the heat transfer in screened furnaces is proposed. Theoretical dependences for the distribution of the heat fluxes in the combustion chambers of industrial tube furnaces of two types are given.

Increasing the unit power of furnace equipment requires increase in their size and improvement in the sealing of the structure, which complicates experimental investigations. This has led to an increase in the role of mathematical modeling of the heat transfer in furnaces. Zonal methods are currently the most promising for theoretical calculations.

A distinguishing feature of screened furnaces is the complexity and diversity of their construction and also of the methods of organizing the furnace processes, which is extremely difficult to take into account in theoretical models. Simplified schemes are usually used here.

In the present work, an attempt is made to develop a sufficiently universal zonal method of calculation for screened furnaces. To this end, all chambers are divided into two classes. The first consists of chambers of radial—cylindrical type. These are characterized by the presence of cylindrical, conical, annular, and disk surfaces, bounding both the chamber itself and the isolated zones. The second class comprises furnaces of box type, whose volume is bounded by plane surfaces.

The physical parameters of the medium are assumed to be constant within the limits of each zone and to change discontinuously at the zone boundaries.

To universalize the method, the zones are classified in terms of geometric (Tables 1 and 2) and optical (Tables 3 and 4) features. The plus and minus signs in the tables denote the presence or absence of the corresponding initial parameters for zones of different geometric types.

The geometrical zonal model of a furnace may be considered in the form of a set of zones of various geometric forms included in the classification tables, which allows the constructional features of the particular furnaces to be taken into account.

The calculational system of equations used in the mathematical model to find the zonal mean temperatures and heat fluxes takes the form [1]

$$\sum_{i=1}^{N} P_{ij}T_i^4 + \sum_{\beta=1}^{N}\sum_{\gamma=1}^{N} (\delta_\gamma^j - \delta_\beta^j)\,\Omega_{\beta\gamma}T_\beta + C_j = 0,\ j = 1,\ 2,\ \ldots,\ N. \tag{1}$$

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TABLE 1. Initial Parameters for the Calculation of the Generalized Angular Coefficients in Screened Furnaces of Box Type

Geometric type of zone	Zone characteristics	x_0	y_0	z_0	Δx	Δy	Δz	η_1	η_2	z_{η_1}	z_{η_2}	η_E	z_{η_E}	y_E	z_E
1	Volume, $\eta_1 \neq \eta_2 \neq 0$	+	−	+	+	−	+	+	+	+	+	−	−	−	−
2	Volume, $\eta_1 = \eta_2 \neq 0$	+	−	+	+	−	+	+	+	+	+	+	+	−	−
3	Volume, $\eta_1 = 0$, $\eta_2 \neq 0$	+	+	+	+	−	+	+	+	−	+	−	−	−	−
4	Volume, $\eta_1 \neq 0$, $\eta_2 = 0$	+	+	+	+	−	+	+	+	+	−	−	−	−	−
5	Volume, $\eta_1 = \eta_2 = 0$	+	+	+	+	+	+	−	−	−	−	−	−	+	+
	Side wall:														
6	right-hand	+	+	+	+	+	+	−	−	−	−	−	−	−	−
7	left-hand	+	+	+	+	+	+	−	−	−	−	−	−	−	−
	Frontal wall:														
8	front	+	+	+	+	+	+	−	−	−	−	−	−	−	−
9	back	+	+	+	+	+	+	−	−	−	−	−	−	−	−
	Surface:														
10	hearth	+	+	+	+	+	+	−	−	−	−	−	−	−	−
11	arched	+	+	+	+	+	+	−	−	−	−	−	−	−	−
	Slope:														
12	upper	+	−	+	+	−	+	+	+	+	+	−	−	−	−
13	lower	+	−	+	+	−	+	+	+	+	+	−	−	−	−
	Screen:														
14	side	+	+	+	+	+	+	−	−	−	−	−	−	−	−
15	horizontal	+	+	+	+	+	+	−	−	−	−	−	−	−	−
16	tapered	+	−	+	+	−	+	+	+	+	+	−	−	−	−

TABLE 2. Initial Parameters for the Calculation of Generalized Angular Coefficients in Screened Furnaces of Radial—Cylindrical Type

Geometric type of zone	Zone characteristics	z_0	ρ_0	φ_0	Δz	$\Delta\rho$	$\Delta\varphi$	ρ_E	η_1	η_2	z_{η_1}	z_{η_2}
1	Volume, $\eta_1 \neq \eta_2 \neq 0$	+	−	+	+	−	+	−	+	+	+	+
2	Volume, $\eta_1 = \eta_2 \neq 0$	+	−	+	+	−	+	−	+	+	+	+
3	Volume, $\eta_1 = 0$, $\eta_2 \neq 0$	+	+	+	+	−	+	−	+	+	−	+
4	Volume, $\eta_1 \neq 0$, $\eta_2 = 0$	+	+	+	+	−	+	−	+	+	+	−
5	Volume, $\eta_1 = \eta_2 = 0$	+	+	+	+	+	+	−	−	−	−	−
	Cylinder surface:											
6	internal	+	+	+	+	+	+	−	−	−	−	−
7	external	+	+	+	+	+	+	−	−	−	−	−
	Surface:											
8	arched	+	+	+	+	+	+	−	−	−	−	−
9	hearth	+	+	+	+	+	+	−	−	−	−	−
10	cylindrical screened	+	+	+	+	+	+	−	−	−	−	−
11	Azimuthal plane: $\varphi = \varphi_{min} \geqslant 0$	+	+	+	+	+	+	−	−	−	−	−
12	Azimuthal plane: $\varphi = \varphi_{max} < 2\pi$	+	+	+	+	+	+	−	−	−	−	−
13	Volume, containing a screen	+	−	+	+	−	+	−	+	+	+	+
14	Conical surface	+	+	+	+	+	+	+	−	−	−	−

TABLE 3. Zone Classification with Respect to Optical Features for Screened Furnaces of Box Type

Optical type of zone	1	2	3	4	5	6	7
Zone characteristics	Volume zones, no screen	Absorbing boundary surface	Screen zones	Mirror surface zones	Volume zones with vertical screen	Volume zones with horizontal screen	Volume zones with inclined screen
Numbers of the geometric types of zones appearing in the given optical type	1—5	6—13	6, 7, 14—16	6—9	5	5	2

TABLE 4. Zone Classification with Respect to Optical Features for Screened Furnaces of Cylindrical Type

Optical type of zone	1	2	3	4	5	6	7
Zone characteristics	Volume zones	Cylindrical screen zones	Azimuthal screen zones, $\varphi = \varphi_{min} \geqslant 0$	Azimuthal screen zones, $\varphi = \varphi_{max} < 2\pi$	Azimuthal mirror surface, $\varphi = \varphi_{min} \geqslant 0$	Azimuthal mirror surface, $\varphi = \varphi_{max} < 2\pi$	Absorbing boundary surface
Numbers of the geometric zone types included in the given optical type	1—5, 14	10	11	12	11	12	6—9, 13

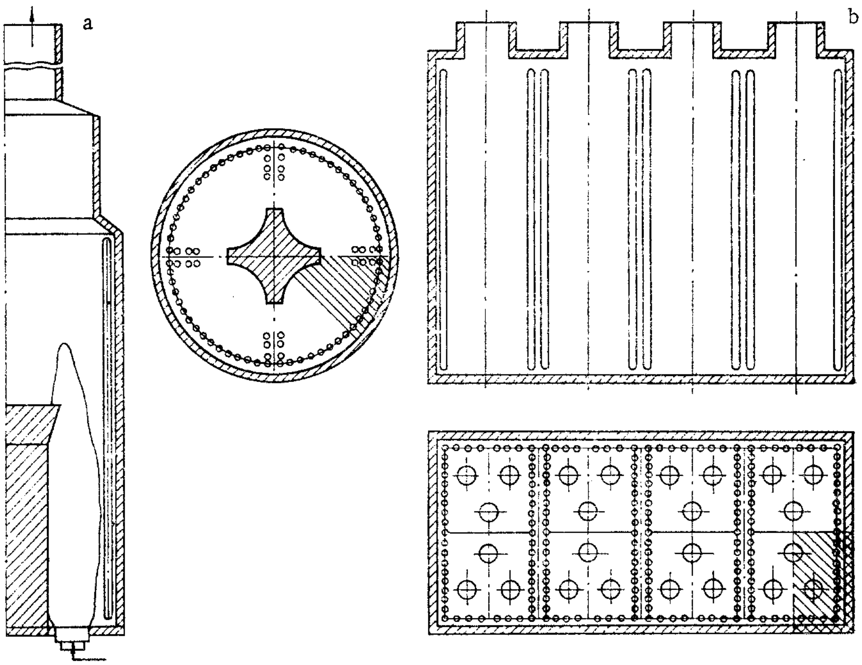

Fig. 1 TsD4 630/15 (a) and VS4 1400/13 (b) tube furnaces.

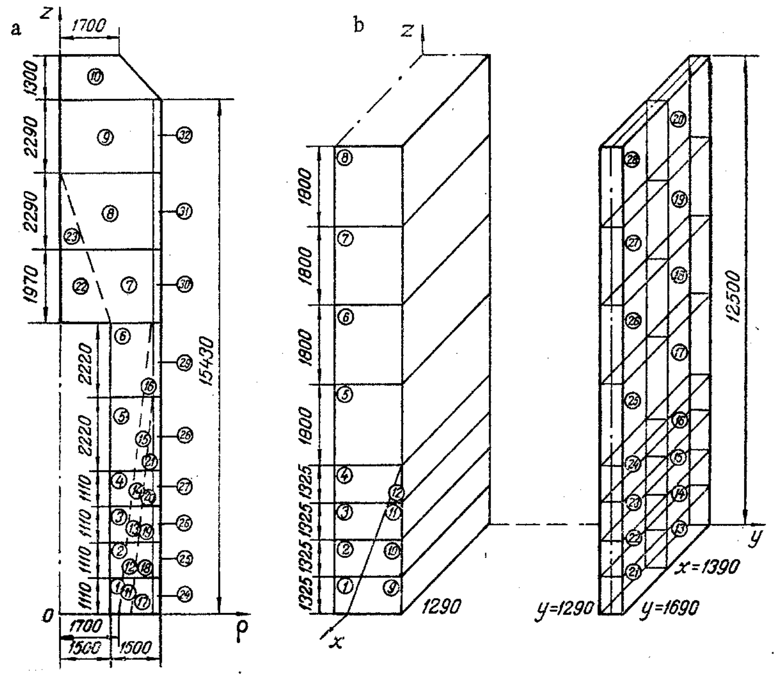

Fig. 2. Geometric zonal models of the TsD4 630/15 (a) and VS4 1400/13 (b) tube furnaces. The circled figures give the volume-zone numbers. The dimensions are given in mm.

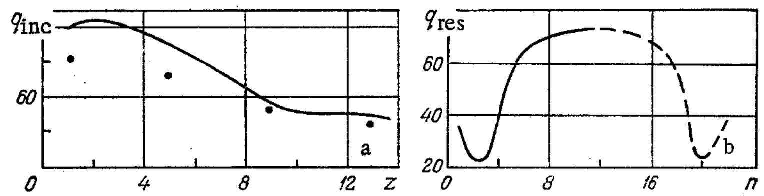

Fig. 3. Distribution of the incident (a) and resulting (b) specific heat fluxes (kW/m^2) over the heating surface in the TsD4 630/15 tube furnace. The points correspond to experimental values; z is the distance from the hearth, m; n is the number of tubes in the section.

On account of the complexity of the given system, the generalized angular coefficients ψ_{ij} appearing as parameters in the expression for the radiational-transfer coefficients P_{ij} are found by means of the Monte Carlo method [2]. Surinov's resolvent system of equations is used to find the resolving angular coefficients. No account is taken of scattering of the radiant energy in the furnace medium.

The realization of the Monte Carlo method is based on the derivation of dependences allowing the emitting points to be uniformly distributed in the zones appearing in the classification tables and also allowing the radiation vector field to be modeled. By combining the two numbers denoting the geometric and optical type of the zones, the beam behavior may be influenced in accordance with the physical laws of emission and adsorption.

A tubular screen is regarded as a perforated surface partially retaining the incident radiation. The probability of beam incidence on the tube is determined as a function of the tube diameter, the tube spacing, and the projection of the angle of incidence on the plane perpendicular to the tube axis.

The nongray emitting properties of the gas medium are taken into account using the Hottel quasigray model [3, 4]. Each single test in the calculation of the angular coefficients according to a nongray model constitutes a history of a sheaf of rays, the energy of each one being absorbed by a "gray gas." Simultaneous tracing of several beams in this way permits a considerable economy of machine time in comparison with calculations in each band of the spectrum. Comparative calculations show that the time required for a calculation by the quasigray model is only 20% more than the time required for the calculation by the gray model.

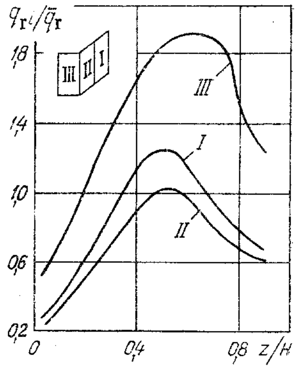

Fig. 4. Distribution of the dimensionless radiational specific heat fluxes over the sections of the heating surface of the VS4 1400/13 tube furnace; H is the tube length (H = 13 m); $\bar{q}_p$ = 25.85 kW/m².

Using the given method, mathematical modeling of the heat transfer in TsD 630/16 VS 1400/13 commercial tube furnaces is undertaken. Diagrams of these furnaces are shown in Fig. 1.

The TsD4 630/15 furnace is of a radial—cylindrical type. A scatterer—distributor runs along the axis of the radiation chamber, in the form of a pyramid with concave faces. The faces form supporting walls for the flames of burners in the bottom of the furnace. The scatterer—distributor divides the radiation chamber into four independent heat-transfer zones with autonomous regulation of the radiant-coil heat stress.

The VS4 1400/13 furnace is of box type. The radiant chamber of the furnace is divided into four independent sections by means of producing tubes; in each section, six burners are installed in the hearth.

To simplify the calculations, and also reduce the machine time and the extent of detailing of the model, the investigation of the heat transfer is limited to the regions of the screened furnaces shaded in Fig. 1. The remainder of the furnace volume may be regarded as the sum of the calculated sections, in view of the symmetry. Radiation from the remaining volume is taken into account by introducing mirror zones into the model, positioning them in the symmetry planes of the sections.

The working space of TsD-type furnaces (Fig. 2a) is divided into 103 zones, of which 32 are volume zones, 17 are stacking zones, six are mirror zones, 45 are surface-heating zones, and three are hypothetical adiabatic, absolutely black, surface zones closing the holes for the introduction of fuel, the exhaust of flue gases, and the upper part of the scatterer—distributor. Inclined conical surfaces separate the flow parts of the oily (zones 1-6) and gas (zones 11-16) flames. The boundary layer of gases (zones 24-32) at the supporting wall is also distinguished in the model. Zones 17-23 represent inverse flue-gas current.

The working space of the VS-type furnace (Fig. 2b) is divided into 65 zones, of which 28 are volume zones, nine are surface stacking zones, 24 are surface heating zone, two are hypothetical, absolutely black, surface zones closing the holes of the introduction of fuel and the exhaust of flue gases, and two are mirror zones.

The fuel oil burns in the first six zones. Regions of inverse flue-gas current (zones 9-12) and gas boundary layer (zones 13-28) are also distinguished in the model.

The heat load of the sections considered was 4225 kW for the TsD-type furnace and 6381 kW for the VS-type furnace.

The distribution of the incident and resulting heat fluxes with respect to the heating surfaces is shown in Figs. 3 and 4. In both furnaces, significant nonuniformity is observed in their distribution. Thus, for the furnace of TsD type the nonuniformity coefficient (the ratio of the maximum specific heat flux to the mean) is 2.07, while for the VS-type furnace it is 1.82. The central tubes of the supporting wall are the most stressed in the TsD-type furnace (Fig. 3b), which is explained by the maximum angle of visibility of the high-temperature regions of the flame, the combustion products, and the scatterer—distributor for these tubes.

Comparison of the results of experiment and calculation (Fig. 3a) shows that the incident radiational heat fluxes are in satisfactory agreement. The maximum discrepancy in these fluxes corresponds to the lower section of the tubes and is of the order of 20%.

On the whole, the results of the investigation show that the proposed mathematical model of heat transfer is of sufficeint accuracy for engineering calculations, and may be used to determine the temperature fields and resulting heat fluxes in commercial furnaces.

NOTATION

x_0, y_0, z_0, ρ_0, φ_0, minimal abscissa, ordinate, z coordinate, radius vector, and polar angle of zone, m, rad; Δx, Δy, Δx, $\Delta \rho$, $\Delta \varphi$, dimensions of the zones along the coordinates, m, rad; y_E, z_E, ρ_E, corresponding coordinates of the surface passing through the tube axes, m; η_1, η_2, angles between the z axis and the surfaces lying close to and far from the z axis, rad; η_E, angle between the z axis and the surface passing through the tube axes, rad; z_{η_1}, z_{η_2}, $z_{\eta E}$, z coordinates of the points of intersection of the corresponding surfaces with the z axis, m; P_{ij}, radiation-transfer coefficients, W/K^4; Ω_{ij}, coefficients of convective—turbulent transfer, W/K; T_i, T_β, temperatures of zones i and β; K; $\delta^j{}_\gamma$, $\delta^j{}_\beta$, Kronecker deltas; c_j, free term of the j-th equation; N, total number of volume and surface zones.

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NUMERICAL CALCULATION OF THE GENERALIZED ANGULAR EMISSION COEFFICIENTS IN TWO-DIMENSIONAL SYSTEMS

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UDC 536.3

We describe a method of calculating the generalized angular emission coefficients by expressing them in the form of finite series.

The calculation of the generalized angular emission coefficients is the most important step in applying zonal methods to the calculation of heat exchange in various radiating systems. The calculation of the generalized angular emission coefficients by direct integration is possible only in special cases [1]. The resulting formulas are complex and not very convenient in engineering applications. This deficiency is also present in the approximate method developed in [2, 3]. An exception is the approximate method of [4,5], in which relatively simple analytical expressions can be obtained with the help of theorems on the mean. In this method, the generalized angular emission coefficient is written as a product of a geometrical angular coefficient and a certain average transmissivity of the medium. The latter is considered as a purely geometrical characteristic of a radiating layer, and this leads to significant errors.

In the present paper, the generalized angular emission coefficients are obtained for a two-dimensional system as finite series, where each term is a product of a geometrical angular coefficient and the transmissivity of the medium. The resulting algorithm for the numerical computation of the generalized angular emission coefficients can be used to determine the coefficients for two-dimensional systems of complicated geometry. In this approach, the amount of calculation is about 100-1000 times less than in the Monte Carlo method [5] which is usually used in these problems.

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