technology of fabricating heat pipes with metal fiber wicks assures reliable fastening and good contact between the wick and the wall because of soldering the fibers together and to the pipe shell. Such a contact is not spoiled during multiple changes in the wall and wick temperatures. If the contact thermal resistivity is neglected, the coefficient of heat elimination in the condensation zone can then be considered dependent on the thickness and the coefficients of thermal conductivity of the wetted wick and the condensate film. The thermal resistivity of the phase transition can exert a noticeable influence on the heat-exchange intensity in the condensation zone in the very low pressure domain.

A maximum radial heat flux density $q_H^{max} \approx 60$ W/cm^2 was also obtained for the heat pipe No. 1 (Fig. 4). The limitation obtained is evidently caused by the boiling crisis in the wick, since the total heat flux corresponding to the value q_H^{max} was below the maximum Q_{max} according to the capillary transport conditions and the quantity q_H^{max} does not change with the change in heat-pipe orientation in the gravitational field ($-5° \leq \varphi \leq +5°$). The quantities q_H^{max} depend on the pressure, the thermophysical properties of the fluid, and the geometry and structural characteristics of the wicks. Thus, heat pipe No. 2, in which the wick thickness is 1.65 times less than in pipe No. 1, in practice, other conditions being equal, operated efficiently for $q_H = 70$ W/cm^2 (Fig. 1, curve 12). The value $q_H^{max} \approx 30$ W/cm^2 is obtained for heat pipe No. 5, which has one-third the mean pore diameter compared to heat pipes Nos. 1 and 2.

NOTATION

L_T, L_C; lengths of the transport and condensation zones; d_{in}, inner diameter of the heat pipe shell F_H/F_C, ratio between the heating and condensation surfaces; δ_W, wick thickness; l_f/d_f, ratio between fiber length and diameter; P, wick porosity; $\bar{t}_W^H$, $\bar{t}_W^C$, mean wall temperatures in the heating and condensation zones; t_{sat}, p_{sat}, saturation temperature and pressure.

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HEAT TRANSFER DUE TO THE INTERACTION BETWEEN CONDENSED PARTICLES AND A WALL

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UDC 536.24

An approximate method of assessing the heat transfer associated with collisions between the solid particles in a flow of gas suspension and the walls of the channel is considered.

During the passage of a two-phase flow through straight tubes and channels the heat transfer between the wall and the condensed particles by direct contact is not very great, since any deposition of the particles on the wall arises mainly from the effect of the pulsational motion of the gas on the particles, and this fails to produce any major flow of particles to the wall. In flows with curved streamlines (curvilinear and swirling flows, flow in nozzles), inertial streams of particles are conveyed to the heat-transfer surface, and the role of contact exchange in the overall heat-transfer process increases.

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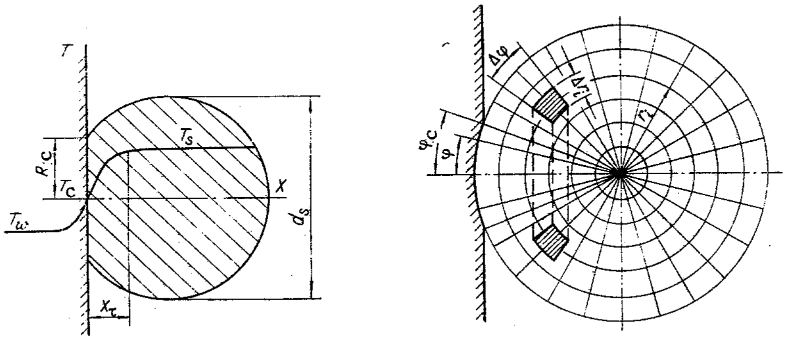

Fig. 1 Fig. 2

Fig. 1. Scheme representing the interaction of a particle with the wall after collision.

Fig. 2. Division of the particle into elementary volumes.

The completeness of the energy transfer between the particles and the wall during the time of contact depends on the state of aggregation of the condensed particles.

If liquid particles arrive at the wall, we may reasonably assume that complete energy transfer takes place between the particles and the wall or condensate film (if formed). The thermal flux density q_c due to heat transfer between the inertially settling particles and the wall or film is given by the equation

$$q_c = c_b g_{in} (T_s - T_c).$$

If the condensed particles are in the solid state, energy transfer between the particles and the wall is incomplete.

The amount of heat transferred to the wall by the particle on collision may be determined from the temperature gradient at the point of contact, if allowance is made for the transient temperature field in the particle.

An exact solution of the problem regarding the transient two-dimensional temperature field of the particle during the brief contact is very difficult. Any estimate of the contact heat transfer between the particle and the wall based on the assumption of one-dimensional heat conduction in the particle [1] may lead to serious errors.

In this paper we shall propose an approximate method of estimating heat transfer during the collision of solid particles with a wall; it will be based on a one-dimensional model of the process, with the introduction of corrections allowing for the difference between the one-dimensional temperature field in the particle and the two-dimensional field calculated by a numerical method.

The boundary conditions for calculating the temperature field in a particle of spherical shape (Fig. 1) are determined after considering the brevity of its contact with the wall and on the assumption that during its contact with the wall the particle has no heat transfer with the gas phase. During the brief contact between the bodies a temperature T_c is established on their contiguous surfaces, remaining constant during the whole contact period [2]. If before contact the wall temperature is T_w and the particle temperature T_s, the temperature at the contact surface T_c will be given by

$$\frac{T_s - T_c}{T_c - T_w} = \frac{\sqrt{\lambda_w c_w \rho_w}}{\sqrt{\lambda_b c_b \rho_b}}. \tag{1}$$

For the one-dimensional model of the process the instantaneous flow of heat from the particle to the wall Q'_τ and the amount of heat Q'_c transferred by the particle during the period of contact τ_c are [2]

$$Q'_\tau = R_c^2 \sqrt{\pi \lambda_b c_b \rho_b} \; \frac{1}{\sqrt{\tau}} (T_s - T_c), \tag{2}$$

$$Q'_c = \int_0^{\tau_c} Q'_\tau d\tau = 2R_c^2 \sqrt{\pi \lambda_b c_b \rho_b} \sqrt{\tau_c} (T_s - T_c). \tag{3}$$

The radius of the plane contact surface R_c and the period of contact τ_c in Eqs. (2) and (3) may be determined by using the theory of elastic impact [1] developed by H. Hertz and set out in [3]. According to this theory

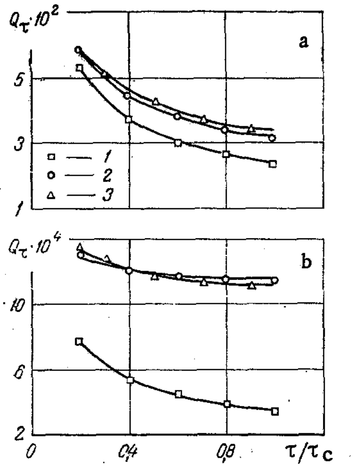

Fig. 3

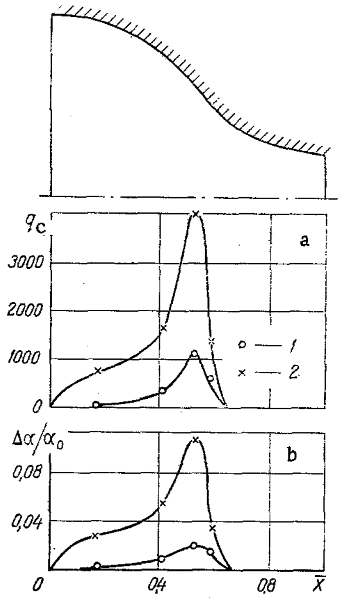

Fig. 4

Fig. 3. Instantaneous heat flows from aluminum oxide particles to the wall: a) d_s = 11.5 μ; b) d_s = 1.75μ; 1) one-dimensional model; 2) numerical method; 3) using K_T, Q_T, W.

Fig. 4. Thermal flux density (a) and increment in heat-transfer coefficient (b) due to contact heat transfer between the particles and the wall: 1) aluminum oxide particles; 2) graphite; q_c, W/m^2.

$$R_c = 0.76 R_s [5\pi^2 W_{sn}^2 \rho_b (K_w + K_b)]^{1/5}; \tag{4}$$

$$\tau_c = \frac{1.69 R_s}{W_{sn}} [5\pi^2 W_{sn} \rho_b (K_w + K_b)]^{2/5}, \tag{5}$$

where K_w, K_b are the elastic parameters of the wall and particle,

$$K_w = \frac{1 - \nu_w^2}{\pi E_w}; \quad K_b = \frac{1 - \nu_b^2}{\pi E_b}.$$

Using Eqs. (2)-(5) we may estimate the contact heat transfer between the particle and the wall, regarding heat conduction in the particle as one-dimensional in a direction normal to the contact surface.

In actual fact the temperature field in the particle is two-dimensional and axisymmetrical relative to the normal to the contact surface passing through the center of the particle. If the temperature field is considered in spherical coordinates, the variation in the particle temperature takes place in the radial r and circumferential φ directions (Fig. 2).

The equation of the two-dimensional transient temperature field in the particle expressed in spherical coordinates is

$$\frac{\partial T}{\partial \tau} = a_b \left[\frac{\partial^2 T}{\partial r^2} + \frac{2}{r} \frac{\partial T}{\partial r} + \frac{1}{r^2 \sin\varphi} \frac{\partial}{\partial \varphi} \left(\sin\varphi \frac{\partial T}{\partial \varphi} \right) \right]. \tag{6}$$

The boundary conditions are

$$\tau = 0, \quad T = T_s; \quad 0 < \tau \leqslant \tau_c, \quad 0 \leqslant \varphi \leqslant \varphi_c, \quad r = R_s, \quad T = T_c;$$
$$\varphi_c < \varphi \leqslant \pi, \quad r = R_s, \quad \frac{\partial T}{\partial r} = 0. \tag{7}$$

Here φ is the angle reckoned from the normal n to the contact surface passing through the center of the particle.

The problem of calculating the two-dimensional transient temperature field in the particle and estimating its contact heat transfer to the wall was solved numerically, subject to boundary conditions (7), after allowing for the changes taking place in the thermal conductivity and specific heat of the particle with temperature. The splitting of the particle into elementary volumes is illustrated in Fig. 2. The computing equations for determining the temperature at the nodal points were obtained on the basis of the elementary-balance method [4]. In order to ensure the necessary accuracy of the calculations while maintaining the shortest possible computer time, the particle was divided into elementary volumes with a nonuniform radial step; the number of divisions in the direction normal to the surface and the circumferential direction was chosen after making some initial test calculations.

Figure 3 illustrates the results of our calculations of the instantaneous heat flows from aluminum oxide particles (diameter 11.5 and 1.75 μ) to the wall, based on the one-dimensional modal, together with the results of a numerical calculation (T_s = 600°K, T_w = 310°K). We see from the curves that the use of the one-dimensional model for determining Q_τ leads to underestimated results. This applies especially to the finer particles.

In order to secure a quantitative estimation of the difference between the actual particle/wall heat transfer estimated numerically and that calculated on the basis of the one-dimensional model, and also to determine the corresponding corrections, we carried out a series of calculations regarding the temperature state of the particles on interacting with the wall. In order to establish the most general laws applicable, the calculations were executed for various kinds of particles: steel, aluminum oxide, and graphite with sizes of d_s = 1.75; 5.5; 11.5; 29.5; 50 μ at T_s = 500-1200°K for a wall temperature T_w = 310°K. Analysis of the computed data showed that the instantaneous thermal fluxes from the particle to the wall Q_τ were approximately

$$Q_\tau = (1 + K_\tau) Q'_\tau, \tag{8}$$

where K_τ is a coefficient allowing for the increment in the instantaneous flow of heat from the particle to the wall by comparison with that calculated on the one-dimensional model.

When analyzing the calculated data the values of K_τ at specified instants of time were determined from

$$K_\tau = \frac{Q_\tau - Q'_\tau}{Q'_\tau}.$$

It was found that K_τ depended on the nature of the particle material and the particle size. This relationship was approximated by the following equation:

$$K_\tau = 7.88 \left(\frac{X_\tau}{R_s}\right)^{1.23}, \tag{9}$$

where X_τ is the distance from the contact surface to the cross section in which at the instant of time τ the temperature differs from the temperature of the particles T_s by 3% (Fig. 1), $X_\tau = \sqrt{9.47 a_b \tau}$.

The results of our calculations of Q_τ using the coefficient K_τ are shown in Fig. 3. Clearly, by using the correcting factor K_τ we may determine the instantaneous heat flows from the particle to the wall quite accurately, allowing for the two-dimensional nature of the temperature field. This also applies to the amount of heat Q_c transferred from the particle to the wall during the whole period of contact. Thus, the improved value of Q_c may be determined as follows:

$$Q_c = \int_0^{\tau_c} (1 + K_\tau) Q'_\tau d\tau. \tag{10}$$

Integrating Eq. (10) and allowing for (2) and (9), we obtain

$$Q_c = 2R_c^2 \sqrt{\pi \lambda_b c_b \rho_b} \sqrt{\tau_c} (T_s - T_c) \left[1 + 12.34 \left(\frac{a_b \tau_c}{R_s^2}\right)^{0.615}\right]. \tag{11}$$

After using Eq. (11) to determine the amount of heat Q_c transferred to the wall by one particle, we may find the thermal flux density q_c and the increment in the heat-transfer coefficient $\Delta\alpha_{sc}$ due to the contact heat transfer of the set of particles to the wall:

$$q_c = \frac{6 g_{in}}{\pi d_s^3 \rho_b} Q_c; \quad \Delta\alpha_{sc} = \frac{q_c}{T_c^* - T_w}.$$

If the condensate comprises many particle sizes, q_c is first determined for each fraction, and then the total characteristics of contact heat transfer associated with the passage of the two-phase flow are found:

$$q_c = \sum_1^m q_{ci}; \quad \Delta\alpha_{sc} = \frac{\sum_1^m q_{ci}}{T_g^* - T_w}.$$

Thus, in order to estimate the contact heat transfer between the particles and the wall during the passage of a two-phase flow we need to know the type of particle and wall material, the size of the particles, the density of the inertial mass flow of particles to the wall g_{in}, the component of particle velocity normal to the wall W_{sn}, and the particle and wall temperatures before the collision T_s and T_w.

Of the foregoing quantities, the type of particle and wall material, the wall temperature, and the particle size are included in the boundary conditions of the problem. The quantities g_{in}, W_{sn}, T_s are determined by calculating the flow of a two-phase system of specified parameters in the channel.

Using the method indicated, we estimated the contact heat transfer during the passage of a two-phase flow in the subcritical part of a nozzle for cases in which the flow contained aluminum and graphite particles d_s = 1-32 μ in size. The profile of the nozzle is shown in Fig. 4. The diameter of the inlet section is 106.8 mm and that of the critical section, 25 mm. The calculation was carried out for the following conditions at the entrance into the nozzle: pressure 3.1 bars, temperature 570°K. The particle flow concentration was β = 0.5.

The results of the calculations are shown in Fig. 4. We see that the thermal flux density q_c due to heat transfer between the particles and the wall increases on passing along the nozzle; in the cross section $\bar{x}$ = 0.53 it reaches a maximum value and then declines. The maximum q_c corresponds to the maximum density of the inertial mass flow of particles to the wall. The relative increment in the heat-transfer coefficient due to contact heat transfer $\Delta\alpha_{sc}/\alpha_0$ (α_0 is the heat-transfer coefficient from the gas phase) varies in an analogous manner. When the flow contains aluminum oxide particles this increment is 3%; for graphite it is 11%. This difference in heat-transfer intensification is due to the fact that the thermal diffusivity a_b of graphite is much greater than that of aluminum oxide. Thus, when there is a relatively high fallout of solid particles of condensed material with a relatively high thermal diffusivity on the wall the contact heat transfer between the particles and the wall may be very considerable and must be taken into account when calculating the heat transfer of two-phase flows.

The foregoing method is suitable for calculating the contact heat transfer of particles with a wall of any configuration if the particle diameter is small compared with the radius of curvature of the wall at the point of contact, the particle wall collision is elastic, and $X_T < 2R_s$.

NOTATION

d_s, particle diameter; m, number of fractions in the condensate when this comprises particles of many sizes; q_c, $\Delta\alpha_{sc}$, thermal flux density and increment in heat-transfer intensity due to contact heat transfer between the particles and the walls; q_{in}, density of inertial mass flow of particles to the wall; T_s, T_w, T_c, particle and wall temperature before the collision and common temperature at the point of contact; λ_b, c_b, ρ_b, a_b, λ_w, c_w, ρ_w, thermal conductivity, specific heat, density, and thermal diffusivity of particle and wall materials, respectively, at $T = (T_s + T_c)/2$; Q_T, instantaneous flow of heat from the particle to the wall during interaction; Q_c', Q_c, amount of heat transferred from the particle to the wall during the whole period of contact determined on the basis of the one-dimensional model of the process and the numerical method, respectively; K_T, coefficient allowing for the increment in the instantaneous flow of heat from the particle to the wall after allowing for the two-dimensional nature of the temperature field in the particle; R_c, τ_c, radius and time of contact of the particle with the wall; W_{sn}, component of particle velocity normal to the wall before the collision; ν_b, E_b, ν_w, E_w, Poisson coefficients and elastic moduli of the particle and wall materials; β, flow flow concentration of the particles (numerically equal to the ratio of the rates of flow of the solid and gaseous phases).

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CHARACTERISTICS OF RADIATIVE HEAT TRANSFER IN MULTIZONE SYSTEMS TAKING ACCOUNT OF ISOTROPIC SCATTERING

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UDC 536.3

A procedure for calculating heat-transfer characteristics in absorbing and isotropically scattering media is developed and tested by applying it to internally fired furnaces.

Optimization of the thermal performance of industrial heating units imposes increasingly rigid requirements on the accuracy and refinement of heat-transfer calculations. Taking account of radiation scattering by solid particles suspended in streams of dusty gases in working chambers of internally fired furnaces and fireboxes may significantly affect the calculated results. Therefore, it is of practical interest, particularly since detailed information has appeared [1-3] on the radiation characteristics of dusty streams and luminous flames, to present a solution within the framework of the zonal method of the heat-transfer problem, taking account of radiation scattering in complex three-dimensional systems filled with a radiating and absorbing medium.

The available literature ([4-10] et al.) contains reports on the effect of scattering on radiative transfer for such relatively simple radiating systems as a slab, an isothermal medium with uniform radiation characteristics, black walls, etc. In [11] a procedure was developed and heat-transfer calculations were performed for a steel-making furnace, taking account of radiation scattering by the Monte Carlo method. This solution was based on the determination of photon mean free paths by using random numbers. Generalized angular coefficients of radiation between zones ψ_{ij} were calculated taking account of scattering in a dusty gas medium [11]. In the calculation of the reduced resolving radiation coefficients f_{ij} the reradiation of energy by surface zones was determined from the linear radiation equations [12-14]. The temperatures of volume and surface zones were calculated by solving the system of nonlinear algebraic heat-transfer and heat-balance equations of the zones [15, 16].

We have developed a procedure for determining the generalized angular and resolving radiation coefficients and have solved a zonal heat-transfer problem in the working chamber of a steel-making furnace, taking account of isotropic scattering of radiation by solving the system of linear algebraic equations of radiative heat transfer. The use of an isotropic scattering indicatrix for real media is justified to a certain degree by the fact that under certain conditions, in particular for relatively large dust particles, the scattered part of the radiation flux can be assumed isotropic, and the diffracted part extremely elongated forward, that is, coincident with the transmitted radiation [17, 18].

The essence of the proposed method consists in the following. First the generalized angular radiation coefficients ψ_{ij}^{att} are found by the Monte Carlo method [15], or by some other method if the radiating systems are simple, by replacing the absorption coefficient α of the volume zones by the attenuation coefficient $K = \alpha + \beta$, where β takes account of isotropic scattering only; the diffracted radiation is taken into account in the transmitted radiation. The values of the coefficients ψ_{ij}^{att} obtained in this way differ from the generalized angular coefficients ψ_{ij}, which take account of scattering by the method used in [11], since the part of the energy scattered into volume zone j is included in the value of the coefficient ψ_{ij}^{att}. An analysis of the radiative transfer equation in an absorbing and scattering medium [19]

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