CONVECTIVE HEAT-TRANSFER MECHANISM IN TWO-PHASE FLOWS FOR HIGH CONCENTRATIONS OF FINELY DISPERSED SOLID PARTICLES

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A model is proposed for convective heat transfer and an analytic solution is obtained for the energy equation for a two-phase flow in vertical tubes with high concentrations of finely dispersed solid particles.

Two-phase gas—solid particle media are characterized by a higher intensity of convective heat transfer as compared with single-phase gas heat carriers. However, on the whole, questions of the hydrodynamics and heat exchange in two-phase flows have been studied to a lesser extent and require further theoretical and experimental investigation.

Momentum and heat-transfer processes in two-phase flows are determined, in the main, by singularities in solid particle interaction with the carrying gas. Depending on the relationship between the solid particle relaxation time τ_p that characterizes its inertial properties, and the time scale of the energy carrying turbulent vortices T, the solid particles take part in either just the average motion of the carrying gas ($\tau_p >> T$) or are entrained by the large-scale turbulent vortices ($\tau_p << T$) in addition to the averaged motion. From the viewpoint of obtaining the greatest heat elimination intensity, two-phase flows with finely dispersed particles whose average size does not exceed 10 μm are of special interest.

Finely dispersed solid particles involved in the fluctuating motion of the carrying gas contribute an additional turbulent heat transfer to the flow core. The molecular heat conduction of the two-phase medium increases here as compared with a single-phase, gas, medium. The thermal resistance of the near-wall domain is lowered because of the active turbulizing effect of the particles on the gas viscous sublayer as well as because of the direct heat transfer from the wall by the particles by conductive heat conduction during their multiple collisions with the wall.

At this time there is one theoretical model of convective heat transfer to a two-phase gas—solid particle flow in practice [1, 2]. In conformity with this model, the heat flux from the wall is transferred only to the gas through the viscous sublayer because of molecule heat conduction.

The contribution of the conductive heat conduction at the contact points between the particles and between the particles and the wall is not taken into account. It is moreover considered that the molecular heat conduction of a two-phase medium equals the molecular heat conduction of the gas. On the whole, all this constrains the applicability of the model proposed in [1, 2] to a domain of comparatively moderate volume concentrations $\beta \leq 0.03$. In conformity with this model, the main mechanism for intensification of the convective heat transfer is to diminish the viscous sublayer thickness because of the action of the solid particles.

As the volume of solid particle concentration grows in a flow their turbulizing action on the gas viscous sublayer increases, and the main thermal resistance of the flow is lowered to result, within the framework of the theoretical model being considered [1, 2], in an unconstrained rise in the heat-elimination intensities for high solid particle concentrations. As the flow is saturated by particles, the frequency of their collision with the wall will grow and, therefore, the quantity of heat that they transfer from the wall to the flow will also increase. For significant concentrations the contribution of conductive heat conduction during particle collisions with the wall can become governing in the total thermal flux from the wall to the two-phase medium.

In this paper a model of convective heat transport to two-phase flows with high concentrations of finely dispersed solid particles is proposed, in conformity with which the heat

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transfer directly from the wall will be realized only by the solid particles because of conducting heat conduction during their collisions with the wall. It can here be considered that at the time of contact the finely dispersed particles will instantaneously take on a temperature equal to the wall temperature. After collision with the wall the particle will give up heat to the carrying gas in a relatively thin near-wall layer because of the high intensity of the intercomponent heat transfer. Because of conductive heat conduction during collisions between particles, heat transfer will not be taken into account.

If the time τ_t during which the temperature equilibrium is built up between the gas and the particles is less than the time of the particle mean free path τ_l (prior to collision with adjacent particles), then the contribution of conductive heat conduction to the total heat flux can evidently be neglected during collisions between the particles. It follows from the heat-balance equation for the particles [1]

$$\tau_t = \frac{d_s^2 \rho_s c_s}{6\lambda\, \mathrm{Nu}_s}. \tag{1}$$

The particle mean free path time is determined by the mean spacing between particles l and the velocity v'_s of the transverse fluctuations of the solid particles entrained by the turbulent carrying gas fluctuations $\tau_l = l/v'_s$.

For complete entrainment of the solid particles by large-scale energy carrying turbulent vortices of the carrying gas (the particle and gas velocity fluctuations are equal), the following relationship can be obtained from the condition $\tau_t \ll \tau_l$ for the limit value of the volume particle concentration at which the influence of conductive heat conduction can still be neglected during collisions between particles:

$$\beta_{\mathrm{lim}} \approx (240\pi)^3 \left(\mathrm{Nu_s} \frac{\lambda}{\eta c_s} \frac{\rho}{\rho_s} \frac{D}{d_s} \frac{\ln \mathrm{Re} - 0.9}{\mathrm{Re}} \right)^3. \tag{2}$$

The relationship obtained in [3] connecting the amplitude of the gas velocity fluctuation with the mean value, with respect to the section, of the averaged gas velocity, and also the formula for the spacing between particles presented in [4] are used in deriving (2). Estimates by means of (2) show that for two-phase flows with $\mathrm{Re} \le 10^4$ ($\rho/\rho_s = 5 \cdot 10^{-4}$, $D/d_s = 10^3$, carrying gas, air) the limit value of the concentration exceeds about 0.6.

Therefore, for $\beta < \beta_{\mathrm{lim}}$ the contribution of the conductive heat conduction can be neglected for collisions between particles.

Heat propagation in the case of an axisymmetric hydrodynamically and thermally stabilized two-phase flow is described by the energy equation [1, 2]:

$$\beta \rho_s c_s u_s \frac{\partial t_s}{\partial x} + (1-\beta)\rho c_p u \frac{\partial t}{\partial x} = \lambda_s \beta \left(\frac{\partial^2 t_s}{\partial r^2} + \frac{1}{r}\frac{\partial t_s}{\partial r} \right) + \lambda (1-\beta) \left(\frac{\partial^2 t}{\partial r^2} + \frac{1}{r}\frac{\partial t}{\partial r} \right). \tag{3}$$

Neglecting the velocity and temperature slip of the solid particles relative to the carrying gas and using the thermal balance equation to determine the temperature change of the two-phase flow along the length, we convert the energy equation (3) under the condition q_w = const to the form

$$2 q_w \frac{u}{U} \frac{r}{R} = \frac{\partial}{\partial r} \left[(\lambda_{\mathrm{ef}} + \lambda_a^* + \lambda_s^* + \lambda^*) r \frac{\partial t_f}{\partial r} \right]. \tag{4}$$

A general expression for the Nusselt number for the flow of a two-phase medium in a circular tube

$$\mathrm{Nu}_f^{-1} = \frac{\lambda (t_w - t_f)}{q_w D} = 2 \int_0^1 \frac{\lambda \left(\int_0^\gamma \frac{u}{U} \gamma d\gamma \right)^2}{(\lambda_{\mathrm{ef}} + \lambda_a^* + \lambda_s^* + \lambda^*)\gamma} d\gamma \tag{5}$$

can be obtained by the method of Lion for a single-phase fluid from (4). Taking into account the mixing action of the solid particles $u/U = 1$ [2], expression (5) goes over into the following:

$$\mathrm{Nu}_f^{-1} = \frac{1}{2}\int_0^1 \frac{\lambda\gamma^3}{\lambda_{ef} + \lambda_a^* + \lambda_s^* + \lambda^*}\,d\gamma. \tag{6}$$

In conformity with the model proposed for heat transfer in a two-phase flow with high volume concentrations of finely dispersed solid particles, the whole flow domain is provisionally partitioned into two zones, a thin near-wall layer and the flow core. In the near-wall layer the heat from the wall to the two-phase medium is transferred mainly because of conductive heat conduction under short-term contacts between the particles and the wall. Here, after collision with the wall, the particle gives the heat obtained off to the carrying gas in the near-wall layer. Heat transfer is realized in the flow core because of molecular heat conduction (λ_{ef}) and by turbulent fluctuations of the particles and gas ($\lambda^*_s + \lambda^*$). Then it follows from relationship (6)

$$\mathrm{Nu}_f^{-1} = \frac{1}{2}\int_0^{\gamma_1} \frac{\lambda\gamma^3}{\lambda_{ef} + \lambda_s^* + \lambda^*}\,d\gamma + \frac{1}{2}\int_{\gamma_1}^1 \frac{\lambda}{\lambda_a^*}\,d\gamma, \tag{7}$$

where $\gamma_1 = r_1/R$; $\delta_a = R - r_1$ is the near-wall layer thickness.

The density of the heat flux due to the conductive heat conduction during particle interaction with the wall is determined as follows:

$$q_a = \rho_s c_s \frac{\pi d_s^3}{6}[t_w - t_f(\delta_a)]\,n, \tag{8}$$

where n is the frequency of particle collision with unit surface of the tube and $t_f(\delta_a)$ is the temperature of the two-phase medium at the boundary of the near-wall layer.

Assuming that only particles in an annular layer of thickness δ_a (particles from the flow core do not penetrate into the near-wall layer because of the small mean free path in a flow with high concentrations), we obtain for the collision frequency

$$n = \frac{1}{\pi DL}\,\frac{\pi DL\delta_a}{\pi d_s^3/6}\,\bar{\beta}\,\frac{v_s'}{2\delta_a} = \frac{3\bar{\beta}v_s'}{\pi d_s^3}. \tag{9}$$

The distribution of the particle volume concentration over the tube section is assumed uniform and the mean value of the volume concentration $\bar{\beta}$ with respect to a section is taken in (9).

Using the analog of the heat-conduction coefficient λ^*_a in the general case, an expression can be written for the heat flux density q_a

$$q_a = -\lambda_a^*\frac{[t_w - t_g(\delta_a)]}{\delta_a}. \tag{10}$$

Then taking account of (9), we obtain a relationship from (8) and (10) for the analog of the heat-conduction coefficient λ^*_a characterizing the conductive heat transfer by particles from the wall:

$$\lambda_a^* = \frac{1}{2}\bar{\beta}\rho_s c_s v_s'\delta_a. \tag{11}$$

Particle interaction with the wall introduces a corresponding contribution even to the total hydraulic drag during the flow of a two-phase medium. As flow saturation by particles increases, the component of the total pressure loss due to this process grows and for definite concentrations becomes governing [1]. In conformity with [5] these pressure losses are associated with the energy expenditures by the carrying gas in restoring the initial longitudinal velocity of the particles after the collisions and are determined by the relationship

$$\Delta p_s = n\pi DL\rho_s\frac{\pi d_s^3}{6}U_s k_{sp}\frac{4}{\pi D^2}. \tag{12}$$

Here the particle longitudinal velocity directly ahead of impact is taken equal to the mean velocity U_s over the section.

Taking account of (9) for the collision frequency, there follows from (12)

$$\Delta p_s = 2\frac{L}{D}\bar{\beta}\rho_s v_s' k_{sp} U_s. \tag{13}$$

Having determined the transverse particle fluctuation velocity v_a from (13), we obtain an expression for the analog of the heat-conduction coefficient characterizing the heat transfer in the near-wall domain

$$\lambda_a^* = \frac{1}{4} \frac{D}{L} \frac{\Delta p_s \delta_a c_s}{k_{sp} U_s}. \tag{14}$$

Turbulent heat transfer in the flow core because of the fluctuating carrying gas motion and the entrained solid particles is described by the appropriate analogs of the heat-conduction coefficients

$$\lambda^* = (1 - \overline{\beta}) \rho c_p \overline{v' t'} / (dt_f/dr), \tag{15}$$

$$\lambda_s^* = \overline{\beta} \rho_s c_s \overline{v_s' t_s'} / (dt_f/dr). \tag{16}$$

Under conditions of complete agreement between the fluctuating velocities v' and v'_s and the temperature t' and t'_s of the gas and particles (completely noninert particles), an expression can be obtained for the sum of the heat-conduction coefficient analogs governing the intensity of the turbulent heat transfer to the flow core

$$\lambda^* + \lambda_s^* = [(1 - \overline{\beta}) \rho c_p + \overline{\beta} \rho_s c_s] \varepsilon_{qf}, \tag{17}$$

where $\varepsilon_{qf} = \overline{v't'}/(dt_f/dr) = \overline{v'_s t'_s}/(dt_f/dr)$ is the coefficient of turbulent heat transfer.

Assuming the turbulent heat-transfer coefficient distribution uniform in the flow core with the condition $\delta_a << R$ taken into account, we obtain for the Nusselt number as a result of integrating (7):

$$\mathrm{Nu}_f^{-1} = \frac{1}{8} \frac{\lambda}{\lambda_{ef} + [(1 - \overline{\beta}) \rho c_p + \overline{\beta} \rho_s c_s] \varepsilon_{qf}} + 4 \frac{k_{sp} \lambda U_s L}{D^2 \Delta p_s c_s}. \tag{18}$$

Therefore, to determine the heat-elimination intensity by means of dependence (18) it is necessary to have data on the turbulent heat-transfer coefficients in the flow core ε_{qf} and the pressure losses because of particle interaction with the wall Δp_s. The quantities mentioned, as well as the values of the heat-elimination coefficients to the dropping two-phase air flow with boron carbide B_4C particles in tubes with D = 10.56 and 14.4 mm were obtained experimentally. The mean particle size determined by the sedimentation weight method was 5 μm.

In conformity with the additivity principle, the total pressure losses during two-phase medium flow are represented in the form of an algebraic sum of the separate components [1]. In particular, for an isothermal stabilized dropping flow, the total pressure losses are determined by three components

$$\Delta p_{to} = \Delta p + \Delta p_s - \Delta p_b. \tag{19}$$

The Δp_s was calculated from the total measured pressure losses Δp_{to} obtained in the mass flow rate concentration range μ = 183-411, which corresponded to a change in the mean particle volume concentration over the section within $\overline{\beta}$ = 0.084-0.171. The pressure losses due to viscous friction of the carrying gas in the two-phase flow can differ substantially from the corresponding values for a single-phase fluid flow for identical Reynolds numbers [6]. As a whole, this is associated with the influence of particles on the averaged and fluctuation characteristics of the carrying gas. Consequently, the quantity Δp cannot be computed by the known empirical dependences for a single-phase fluid in the general case. However, as the solid particle concentration grows the contribution of the quantity Δp to the more general hydraulic drag is lowered. As is shown in [7], for μ > 40 ($\rho_s/\rho \approx 2 \cdot 10^3$), Δp is not more than 10% of the total pressure losses. Therefore, the quantity Δp can be neglected in the domain of sufficiently high particle concentrations.

The turbulent heat-transfer coefficients ε_{qf} were determined from experimentally obtained distributions of the average temperature of the two-phase flow over the section (measurements were limited to the domain $0 \le r/R \le 0.83$). The thermal flux from the wall to the two-phase medium was produced because of electrical heating during direct transmission of a low-voltage alternating electric current over a stainless steel tube wall. Investigations of the temperature distribution over the flow section were by using Chromel—Alumel microthermocouples in a stainless steel, 1-mm-diameter capillary which was displaced over the tube

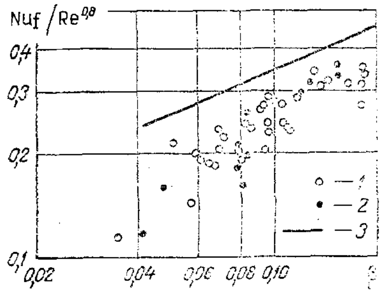

Fig. 1. Comparison of the theoretical solution and experimental data on stabilized heat elimination to a two-phase flow: 1) D = 14.4 mm; 2) 10.56; 3) theoretical solution. Along the horizontal $\bar{\beta}$.

radius by using a plotter unit, a micrometer screw. To diminish the perturbations induced in the flow, as well as the measurement error because of the heat outflow from the thermocouple electrodes and the capillary, the thermocouple was directed opposite to the flow parallel to the channel axis. The wall temperature was also measured by a Chromel—Alumel thermocouple soldered to the outer surface by a capacitor discharge.

Under the assumption of agreement between the fluctuating velocity and temperature components of the particles and the gas, the total heat flux density is represented as follows:

$$q_f(r) = -\{\lambda_{ef} + [(1-\bar{\beta})\rho c_p + \bar{\beta}\rho_s c_s]\,\varepsilon_{qf}\}\,dt_f/dr. \tag{20}$$

Values of the turbulent heat-transfer coefficients ε_{qf} were determined directly from (20), here the heat flux density distribution $q_f(r)$ was considered linear. To determine the values of the derivatives dt_f/dr experimentally, the obtained temperature profiles were approximated by second-order polynomials whose coefficients were calculated by least squares on an electronic computer. The effective heat-conduction coefficient of the two-phase medium λ_{ef} was computed by means of the known Maxwell formula

$$\lambda_{ef}/\lambda = \frac{2\lambda + \lambda_s - 2\bar{\beta}(\lambda - \lambda_s)}{2\lambda + \lambda_s + 2\bar{\beta}(\lambda - \lambda_s)}. \tag{21}$$

Results of computing the Nusselt number by means of (18) with the experimentally obtained values of Δp_s and ε_{qf} taken into account are presented in the figure as compared with experimental data on the stabilized heat elimination, represented in the form of a dependence of the quantity $Nu_f/Re^{0.8}$ on the volume concentration β (the proportionality $Nu_f \sim Re^{0.8}$ was determined for the whole range Re = 2300-11,000 investigated). The loss coefficient of the longitudinal velocity for finely dispersed solid particle interaction with a wall was here taken equal to one $k_{sp} = 1$ (the longitudinal velocity of the particle drops to zero during interaction with the wall). It follows from the figure that the results of computing the Nusselt number in conformity with the proposed model of heat transfer in a two-phase flow with finely dispersed particles are qualitatively and quantitatively in satisfactory agreement with the experimental results in the bulk concentration domain $\bar{\beta} \geq 0.05$.

NOTATION

β, solid particle volume concentration; ρ, ρ_s, gas and solid particle densities; c_p, c_s, gas specific heat at constant pressure and the solid particle specific heat; u, u_s, gas and particle longitudinal velocities; U, U_s, gas and solid particle mean velocities over the section; λ, λ_s, λ_{ef}, gas, solid particle, and two-phase medium heat-conduction coefficients; η, dynamic coefficient of gas viscosity; λ^*_a, λ^*, λ^*_s, analogs of the heat-conduction coefficients taking account of the conductive heat transfer by the particles during interaction with the wall, turbulent heat transfer in the gas, and by particles during fluctuating motion, respectively; q_w, t_w, heat flux density at the wall and the wall temperature; t, t_s, t_f, gas, solid particle, and two-phase medium temperatures; t', t'_s, fluctuating gas and particle temperature components; v', v'_s, transverse fluctuating gas and particle velocity components; ε_{qf}, turbulent heat-transfer coefficient; d_s, mean solid particle size; x, r, cylindrical coordinates; R, D, L, tube radius, diameter, and length; $\gamma = r/R$, relative radial coordinate; Δp_{to}, Δp, Δp_s, Δp_b, total pressure losses because of viscous friction of the gas, because of of solid particle interaction with the wall, because of the effect of mass forces; Nu_f, Nusselt number for the two-phase flow; Nu_s, Nusselt number for intercomponent heat transfer; and Re, Reynolds number.

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EXPERIMENTAL STUDY OF CONVECTIVE HEAT TRANSFER IN BEDS OF SPHERES OF DIFFERENT THICKNESS

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Results are presented from an experimental study of convective heat and mass transfer in spherical beds with the number of layers N = 1, 2, 3, 5, and 13 in the Reynolds number range Re_e = 60-1900.

An analysis of the literature data on convective heat and mass transfer in spherical beds showed the following.

1. Nearly all experiments have studied spherical beds with a fairly large number of layers, when the mean heat- and mass-transfer coefficients are independent of bed thickness. At the same time, high-temperature heat exchangers [1] and selective catalytic reactors [2] developed in recent years require that experiments be performed in beds with the number of layers N = 2-5.

2. The distribution of the heat- and mass-transfer coefficients along beds with N > 10 has an inlet section 2-3 sphere layers long, a section of stabilized transfer, and an outlet section 1-2 layers long. Meanwhile, the heat- and mass-transfer coefficients on the stable section are higher than on the inlet and outlet sections [3-5].

The goal of this study is to empirically examine convective transfer in beds in which the inlet and outlet phenomena affect the mean heat- and mass-transfer coefficients.

We used an ion-exchange method in our experiments. A mass-transfer method was chosen as the method of study due to the relative simplicity of its realization and the absence of heat transfer by conduction through the bed. The analogy established between heat- and mass-transfer processes in spherical beds under certain conditions [6-8] makes it possible to use the mass-transfer data to study heat transfer as well. The analogy applies only to the convective components of heat and mass transfer.

Figure 1 shows a diagram of the experimental unit. Solutions of the salts Na_2CO_3 or $UO_2(NO_3)_2$ in distilled water were pumped through a monodisperse bed of grains of ion-exchange

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