INVESTIGATION OF HEAT TRANSFER IN TRANSVERSE FLOW OF A HIGH-TEMPERATURE DUST – GAS PAST A CYLINDER

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The results of experimental determinations of the effective coefficient of heat transfer at the front surface (in the region of the forward critical point) of a cylinder with a high-temperature dust–gas flow are presented. A formula is obtained to determine the relative (in comparison with a pure gas) intensity of heat transfer.

The regularities of heat transfer between surfaces and dust–gas flow have been investigated quite thoroughly mainly for the case of longitudinal flow past the surfaces [1]. Data on heat transfer for transverse flow past bodies are very few. There are a few articles [2, 3], in which the process of heat transfer with a spherical surface is investigated. Data on heat transfer of dust–gas flows with a cylindrical surface in the case of transverse flow are practically nonexistent. At the same time heat transfer in such a flow is of great interest for a whole series of devices.

It is known [1, 4] that the presence of solid particles in a gas flow leads to an appreciable intensification of the process of heat transfer with the surface. The heat passes from the flow to the surface due to thermal conductivity, convection, radiation, and also as a result of direct contact of particles with the surface. A mathematical description of such a process is very difficult.

For a quasistationary process and quasistabilized flow of a dust–gas with fine particles having small thermal drag the criterial equation of heat transfer with the surface has the form [1, 5]

$$\mathrm{Nu}_d = \frac{\alpha_d d}{\lambda} = f\left(\mathrm{Re},\ \mathrm{Pr},\ \beta,\ \frac{d}{d_s},\ \frac{T_0}{T_i}\right). \quad (1)$$

The present work is devoted to an experimental investigation of the dependence of heat transfer of the downward dust–gas flow with the front surface of a cylinder on the nondimensional parameters enumerated above. It should be noted that according to [1] for $\beta < \beta_{cr} \simeq 0.03$ Prandtl number Pr for a dust–gas flow differs insignificantly from Pr for a pure gas flow.

An experimental equipment was prepared and set up to study the heat transfer processes; its schematic diagram is shown in Fig. 1. The gas from an air system is blown through a furnace and enters a collector. Particles of aluminosilicate* are poured into the bin and are heated to the gas temperature. When the temperatures of the gas and the particles become equal the shutter opens and the particles enter the collector through a flow gate and are mixed with the gas. The flow rate of the solid phase is controlled by changing the diameter of the flow gate. After the collector the dust–gas flow is directed into a vertical cylindrical channel of 50 mm diameter in order to stabilize the flow. The working channel and the bin are thermally insulated and provided with a sheath of thermal screen intended for eliminating axial temperature gradients. The temperature of the flow and the particles in the bin are measured by a Chromel–Copel thermocouple of 100 μm diameter. Special measurements of the temperatures of the particles and gas, made with the use of a cup thermometer [6], showed that there is practically no difference between the temperatures of the solid and gas components of the flow. The accuracy of maintaining and measurement of temperature is ±2°C.

*Spherical particles of aluminosilicate with mean dimensions $d_s = 60$, 130, 235 μm and density $\rho_s = 1650$ kg/m³ were used as the solid phase.

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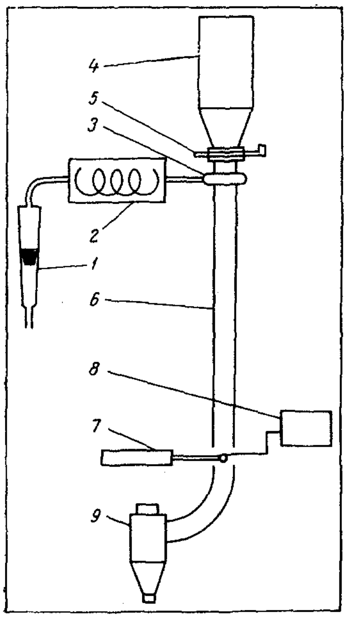

Fig. 1

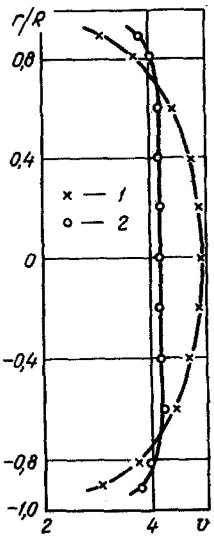

Fig. 2

Fig. 1. Schematic diagram of the experimental equipment: 1) rotameter; 2) furnace; 3) collector; 4) loading bin; 5) flow rate gate; 6) working channel; 7) inlet mechanism; 8) oscillograph; 9) cyclone.

Fig. 2. Velocity profiles of the gas: 1) gas flow in the absence of solid particles; 2) dust–gas flow.

The velocity field of the gas in the dust–gas flow is measured with the use of a combined Pitot–Prandtl tube. The experiments with the dust–gas flow were preceded by measurements of the velocity field for a pure gas flow. The divergence between the average velocities of the gas measured by the rotameter and computed from the readings of the Pitot–Prandtl tube is within 5%. The same agreement between the computed and experimental values of the average velocity of the gas is obtained also for the dust–gas flow. The characteristic profiles of the velocity field of the gas in the case of dust–gas flow and pure gas flow are shown in Fig. 2 for $T_0 = 673°K$ and $v_{av} = 4$ m/sec. It should be pointed out that during the dust–gas flow an equalization of gas velocities occurs over the cross section of the channel; this is perhaps caused by the turbulence generating effect of the particles. The average velocities of the particles, determined with the use of filming, differ appreciably from the velocities of the gas in the entire range of concentrations and sizes of the solid fraction. The flow rate of the particles is determined by the weight method with an accuracy of ±6%. The flow rate volume concentration of the particles in the flow is determined from the expression $\beta = G_S/\rho_S/vS$. This characteristic is related to the flow rate weight concentration (μ) through the following relation: $\beta = \mu\rho/\rho_S$. All the flow parameters are measured in the working section of the channel with steady-state thermal and hydrodynamic regimes (L/D > 60 in the experiment).

An asbestos cement cylinder is used as the working body (samples with 4.8 and 12 mm diameter were used in the experiments) which is introduced into the operating section of the channel by a special spring device. The time of introduction of the sample into the flow is less than 0.1 sec. The basic thermophysical characteristics of asbestos cement were measured by the method of V. S. Vol'kenshtein [7] ($\lambda = 0.52$ W/m · deg, $a = 1.08$ m^2/h). A copper–constantan thermocouple rolled to a thickness of 10-15 μm was fixed to the surface of the cylinder. The estimated error in the measurement of the surface temperature by this thermocouple is less than 4%. The time variation of the temperature on the surface of the cylinder is recorded with the use of an H-105 loop oscillograph.

The method proposed in [8] is used to determine the local heat transfer coefficient in the region of the forward critical point. Solving the problem of heating of an infinite* cylinder [9]

*Cylinders with $(l/d) > 4$ were used in the experiments. The estimated error in the measurement of the surface temperature from computations for an infinite cylinder is less than 2%.

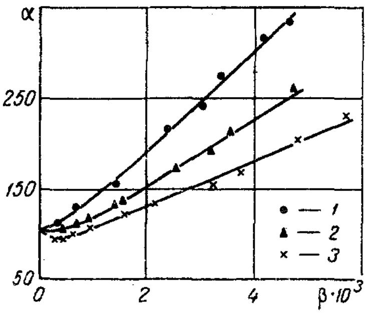

Fig. 3

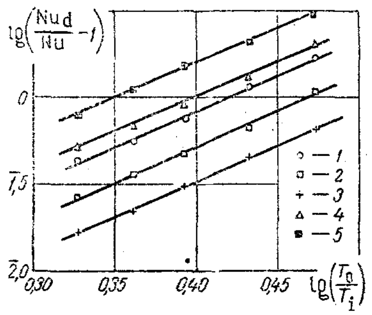

Fig. 4

Fig. 3. Heat transfer coefficient (α, W/m^2·deg) as a function of the flow rate volume concentration (β) and diameter of particles: 1) $d_S = 60$; 2) 130; 3) 235 μm.

Fig. 4. Dependence of relative intensification of heat transfer on the temperature factor: 1) $\beta = 1.6 \cdot 10^{-3}$, $d_S = 60\ \mu$m; 2) $\beta = 1.6 \cdot 10^{-3}$, $d_S = 130\ \mu$m; 3) $\beta = 1.6 \cdot 10^{-3}$, $d_S = 235\ \mu$m; 4) $\beta = 2.4 \cdot 10^{-3}$, $d_S = 130\ \mu$m; 5) $\beta = 3.8 \cdot 10^{-3}$, $d_S = 130\ \mu$m.

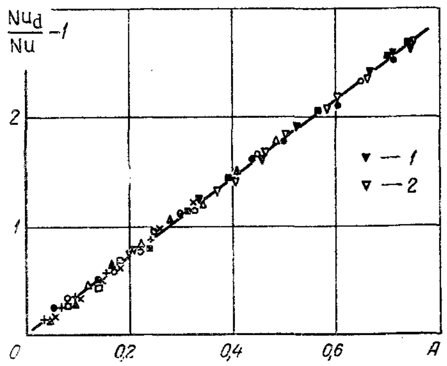

Fig. 5. Generalized dependence of relative intensification of heat transfer ($A = \beta^{1.38}(T_0/T_i)^{3.8} \times (d/d_S)^{0.8}$): 1) Re = 60; 2) Re = 570; for remaining notations see Figs. 2 and 3.

$$\frac{\partial T}{\partial t} = a\left(\frac{\partial^2 T}{\partial r^2} + \frac{1}{r}\frac{\partial T}{\partial r}\right) \quad (t > 0;\ 0 \leqslant r \leqslant R) \tag{2}$$

with the initial and boundary conditions

$$T(r,\ 0) = T_i;\quad -\lambda\frac{\partial T(R,\ t)}{\partial r} + \alpha\left[T_0 - T(R,\ t)\right] = 0;\quad \frac{\partial T(0,\ t)}{\partial r} = 0,$$

we obtain the function T(R, t):

$$\frac{T(R,\ t) - T_i}{T_0 - T_i} = 1 - \sum_{n=1}^{\infty} A_n J_0(\mu_n)\exp(-\mu_n^2 \mathrm{Fo}), \tag{3}$$

where

$$\frac{J_0(\mu_n)}{J_1(\mu_n)} = \frac{1}{\mathrm{Bi}}\mu_n;\quad A_n = \frac{2\mathrm{Bi}}{J_0(\mu_n)(\mu_n^2 + \mathrm{Bi}^2)}.$$

The heat transfer coefficient is found from the solution of Eqs. (3) using experimentally determined temperature of the surface. The nonstationarity of the heat transfer, related to the used procedure of experiment, imposes certain restrictions on the computation of the heat-transfer coefficient from formula (3). However, from the approximate solution [10] of equations of nonstationary heat transfer and our experiments the time for the steady-state regime to get established (in terms of α) is less than 0.3 sec; this permits us to use the solution of Eqs. (3) for the computation of the heat-transfer coefficient with adequate accuracy.

In order to check the suitability of the used procedure experiments with a pure gas were carried out before the investigation of heat transfer of the dust–gas flow. The analysis of these experimental results showed that the heat transfer between air flow and the cylinder is well approximated by the criterial equation [11]

$$\mathrm{Nu} = \frac{\alpha d}{\lambda} = 1.14\,\mathrm{Re}^{0.5}\,\mathrm{Pr}^{0.371}. \tag{4}$$

These results were used also for the calculation of the relative intensification of heat transfer by the dust–gas.

Experiments on the heat transfer between the dust–gas flow and a fixed cylinder were carried out for the following ranges of variation of the controlling factors:

$$\mathrm{Re} = 60 - 600,\ \beta = (3 - 60)\cdot 10^{-4},\ T_0/T_i = 2 - 3,\ d/d_S = 17 - 200.$$

Each point on the graphs corresponds to the average value of experimental data from 3-4 identical experiments. The maximum deviation from the average values does not exceed 5%.

The dependence of the local coefficient of heat transfer in the region of the forward critical point of the cylinder on the concentration of solid phase and the diameter of the particles is shown in Fig. 3 for $T_0 = 673°K$ and $v = 2$ m/sec. The form of the curves remains unchanged on varying the flow parameters. An increase of the concentration or a decrease in the particle diameter leads to an appreciable intensification of the heat transfer. It should be noted that in the range of small concentrations ($\beta < 10^{-3}$, which corresponds to $\mu < 3.5$) for large particles ($d_S = 235\ \mu m$) an anomalous decrease of the heat-transfer coefficient is observed. An analogous phenomenon was also observed in [12, 13] for the case of longitudinal flow. A qualitative explanation of this effect is attempted in [1]. In criterial analysis this range of concentrations was excluded. The following relations were established on the basis of the experimental data:

$$\left(\frac{Nu_d}{Nu} - 1\right) \sim \beta^{1.38},$$

$$\left(\frac{Nu_d}{Nu} - 1\right) \sim (d/d_S)^{0.8}.$$

Here $\frac{Nu_d}{Nu} - 1$ is the relative intensity of heat transfer of the dust–gas flow in comparison with pure gas. Nusselt number for the pure gas is computed from formula (4).

The effect of the temperature of the flow on heat transfer was investigated for $T_0 = 573$–$873°K$ (Fig. 4). The dependence of the relative intensification of heat transfer on the temperature factor is given by the relation

$$\left(\frac{Nu_d}{Nu} - 1\right) \sim (T_0/T_i)^{3.8}$$

in the entire range of variation of the parameters. In the case of longitudinal flow [13, 14] the effect of the temperature factor is explained as due to radiation. This facilitates dividing the effective heat transfer coefficient into two components, radiative and convectional; the latter is independent of the temperature. However, similar computations carried out by us showed that in our case the dependence on the temperature factor is of a more complicated nature and cannot be attributed to radiation alone. This indicates a significant role of the contact heat transfer due to the collisions of the particles with the sample in the case of transverse flow.

The dependence of the relative intensification of heat transfer on Reynolds number was investigated in a wide range of variation of the basic parameters. It was found that $(Nu_d/Nu-1)$ is practically independent of Re.

A combined analysis of all the experimental data on heat transfer is given in Fig. 5. The following dependence describing the heat transfer between the cylinder and the dust–gas flow is obtained by the method of least squares with a probable error of ±10%:

$$\frac{Nu_d}{Nu} - 1 = 3.6\beta^{1.38} (d/d_S)^{0.8} (T_0/T_i)^{3.8}. \tag{5}$$

This dependence is valid in the ranges $10^{-3} < \beta < 6 \cdot 10^{-3}$, $17 < d/d_S < 200$, $2 < T_0/T_i < 3$, $60 < Re < 600$.

The present results were used in the investigation of the regularities of ignition of condensed material by high-temperature dust–gas flow.

NOTATION

D is the diameter of the channel;
S is the area of the transverse cross section of the channel;
d is the diameter of the cylinder;
d_S is the diameter of solid particles;
T_0 is the temperature of the flow;
T_i is the initial temperature of the cylinder;
G_S is the weight flow rate of solid component, kg/sec;
β is the flow rate volume concentration, $m^3 \cdot sec/m^3 \cdot sec$;

μ is the flow rate weight concentration;
α is the heat transfer coefficient;
ρ is the density of gas at T_0;
λ, a are the thermal conductivity and thermal diffusivity, respectively;
Re is the Reynolds number;
Nu is the Nusselt number;
Pr is the Prandtl number.

Subscripts

d is the quantity pertaining to dust–gas flow.

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