SOME PECULIARITIES OF THE HEAT TRANSFER DURING THE FLOW OF A GASEOUS SUSPENSION THROUGH A HORIZONTAL DUCT

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Results of an experimental study are shown concerning the local heat transfer during a pneumatic transport of graphite particles (mean size 0.180 mm and 0.065 mm) through a horizontal duct (diameter 18.8 mm).

It is well known [1] that, when solid particles are transported pneumatically through horizontal ducts at an insufficiently high gas velocity, the concentration of particles is much higher in the lower than in the upper portion of the duct. In a heated duct, therefore, large temperature differences may then develop around the wall circumference. For calculating the local heat transfer under such conditions, formulas derived empirically on the basis of a symmetrical distribution of particles as in vertical ducts, for example, become entirely inadequate.

The heat transfer has been analyzed in [2, 3] for a nonuniform cross sectional distribution of particles in a horizontal duct. Here the data obtained in [2] on the variation of heat transfer around the circumference contradict the results obtained in [3].

In this study we were concerned with the local (lengthwise and circumferentially) heat transfer during the flow of an air-graphite mixture through a horizontal duct $d = 18.8$ mm in diameter. The tests were performed at an approximately constant gas velocity $w = 13$ m/sec. The Reynolds number, calculated per total duct section, was $Re = 16,200$. The graphite particles appeared in two fractions: size $d_S = 0.065$ mm and size $d_S = 0.180$ mm. The sieve analysis of these fractions is given in [4]. The mass discharge concentration was varied from 0.9 to 13.2. Visual inspection of the stream pattern in a glass duct revealed a strong bottomward trend in the motion of particles under these conditions.

Test Procedure. The heat transfer was studied at an approximately uniform thermal flux density over the length and over the circumference of the test segment which had been mounted in an apparatus described earlier [4]. The test segment was a tube of grade 1Kh18N9T stainless steel with a 0.36 mm wall thickness. It was heated by alternating electric current fed directly through the wall. The heated segment was 99 bore diameters long, the hooked on isothermal segment for hydrodynamic flow stabilization was 96 bore diameters long. The wall temperature was measured with 29 copper–constantan thermocouples which had been welded directly to the outside surface. These thermocouples were distributed over 11 sections along the heated segment. In each chosen section, except in both end sections, the thermocouples were located on the top and on the bottom generatrix; in six of these sections there were also thermocouples on the lateral generatrices; in one section ($x/d = 92$) there were altogether five thermocouples.

The apparatus operated in the open mode for both gas and solid particles. Solid particles were fed into the gas stream from a tank at the entrance to the test segment. The hot two-phase stream in the test segment was then cooled and fed into a separator, where the solid particles were collected while clean air was exhausted into the atmosphere. The flow rate of the solid phase was determined by weighing the separated particles and the time during which they had been fed into the stream. The flow rate of air was measured with a double diaphragm.

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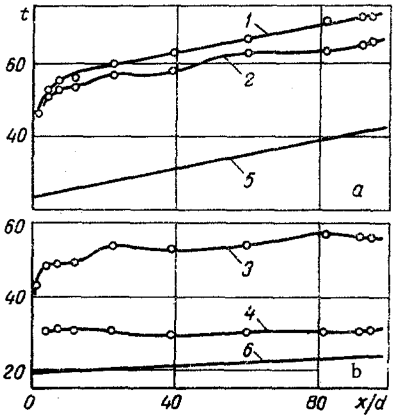


Fig. 1

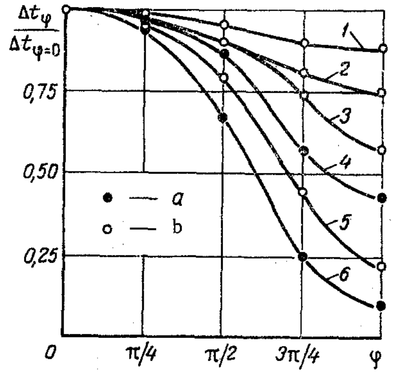


Fig. 2

Fig. 1. Variation of t_w along the top and the bottom generatrix, and variation of t_s along the duct in tests with: d_S = 0.180 mm and K = 1.53 (a), K = 13.2 (b) (t_w at the top generatrix 1, 3; t_w at the bottom generatrix 2, 4; t_s 5, 6). Temperature t (°C).

Fig. 2. Variation of the temperature excess ($\Delta t = t_W - t_S$) around the duct circumference at x/d = 92 ($q \approx 1{,}800$ W/m^2) and K = 0.86 (1), 1.53 (2), 2.6 (3), 2.3 (4), 13.2 (5), 12.5 (6): (a) d_S = 0.065 mm, (b) d_S = 0.180 mm. Top generatrix $\varphi = 0$, bottom generatrix $\varphi = \pi$.

The hydraulic resistance and the heat transfer with clean air were measured in a preliminary test where the Reynolds number ranged from 10,000 to 50,000. The hydraulic resistance data agreed with the Blasius formula within ± 2%. In the heat transfer measurements the thermocouple readings were almost identical around any one section, and the values of the heat transfer coefficients beyond the thermal-stabilization segment agreed within ±5% with the formula in [5]. In comparing our test data with those in [5], a correction was made for the thermal influence factor T_W/T_S not exceeding 1.2 in either tests with clean air or in tests with the air–graphite mixture.

The heat transfer coefficient was calculated according to the formula

$$\alpha = \frac{q}{t_W - t_s}. \tag{1}$$

The thermal flux density was determined from measurements of the electric current and of the electrical resistance. Heat losses were determined in a prior test without an air stream through the duct.

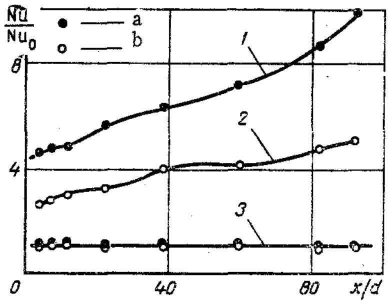
The theoretical stream temperature at a given section was calculated according to the formula

$$t_s = t_1 + \frac{q\pi dx}{Gc + G_S c_S}. \tag{2}$$

Test Results. Measurements of local wall temperatures have shown that the cooling rate varies considerably around the duct circumference during a bottomward motion of particles. The wall temperature is higher at the top than at the bottom generatrix, while this temperature difference within any one section increases with the concentration of particles. It has been found in most tests that the longitudinal variation of the wall temperature is wavelike. A typical curve of wall temperature as a function of the length coordinate is shown in Fig. 1, based on the tests with 0.180 mm particles. Analogous curves have been obtained also from the tests with 0.065 mm particles.

The appreciable difference between temperatures at the top and the bottom generatrix is explainable by the much higher concentration of solid particles near the bottom of the duct. The wavelike longitudinal variation of the wall temperature is, apparently, related to the sinusoidal trajectory of particle motion [1], which produced different local concentrations along the bottom generatrix of the test duct.

The circumferential variation of the wall temperature is shown in Fig. 2, based on the tests with 0.180 mm and 0.065 mm particles. It follows from these graphs that, under almost the same conditions, the temperature varied around the circumference over a wider range in the case of 0.065 mm particles than in the case of 0.180 mm particles.



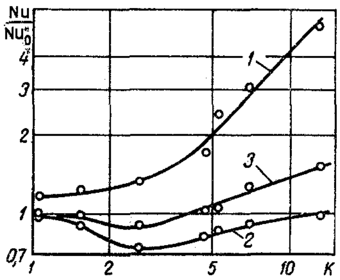


Fig. 3 Fig. 4

Fig 3. $Nu/Nu_0 = f(x/d)$: (a) $d_S = 0.065$ mm and $K = 12.5$, (b) $d_S = 0.180$ mm and $K = 13.6$ (bottom generatrix 1, 2; top generatrix 3).

Fig. 4. $Nu/Nu_0 = f(K)$ at $x/d = 92$ and for $d_S = 0.180$ mm (bottom generatrix 1; top generatrix 2; mean-over-the-circumference heat transfer 3).

The heat transfer along both the top and the bottom generatrix within the test segment is shown in Fig. 3. Here Nu_0 and Nu denote the Nusselt numbers for clean air and for the mixture at a given section.

The effect of concentration on the ratio Nu/Nu_0 is shown in Fig. 4, based on the tests with 0.180 mm particles.

DISCUSSION OF RESULTS

The surprising discovery in this study was that the heat transfer rates at the top and at the bottom generatrix differed more in the tests with fine particles than in the tests with coarse particles. A similar observation was made by the authors of [3], who experimented with glass particles $d_S = 0.030$ mm and $d_S = 0.200$ mm in diameter. This marked effect of particle size on the circumferential heat transfer distribution was, apparently, due to the greater nonuniformity of fine particle distribution over a duct section during the tests. It has been pointed out in [3] that this marked effect of the size of solid particles contradicts the well known hypothesis according to which the vertical and the horizontal distribution of a suspension density in a turbulent stream is determined by the ratio v_{sus}/v_* [6], i.e., according to which a decrease in the particle size results in a more heightwise uniform concentration. This hypothesis remains valid, however, when gravitational downward sedimentation is compensated by a turbulent upward lift, i.e., when the solid particles follow the turbulent pulsations of the carrier medium. The latter condition can be ascertained, for example, by comparing the orders of magnitude of the characteristic time scales of particle motion τ_S and of turbulent carrier pulsation τ. Estimating them at $\tau_S \sim \rho_S d_S^2/\rho\nu$ [7] and $\tau \sim d/w$ respectively, we have

$$\frac{\tau_S}{\tau} \sim \frac{\rho_S}{\rho}\left(\frac{d_S}{d}\right)^2 \mathrm{Re}. \qquad (3)$$

Obviously, particles will follow the turbulent gas pulsations only as long as $\tau_S \ll \tau$. Under our test conditions, close to those stipulated in [3], $\tau_S/\tau \sim 10^2$ for the 0.065 mm particles and $\tau_S/\tau \sim 10^3$ for the 0.180 mm particles, which indicates the absence of a noticeable effect of turbulent stirring on the cross sectional distribution of both the fine and the coarse particles. According to these estimates, then, there is no contradiction between our results and the effect of turbulent stirring on the cross sectional distribution of particles.

Among the possible factors causing an accumulation of finer particles in the bottom part of a duct is the clotting phenomenon [8]. The gist of its mechanism is that, under certain conditions, the drag coefficient decreases sharply when particles come very close together. At that time the initial gas velocity is too low to hold such particles in suspension and thus they drop down. Since there are more fine than coarse particles per unit volume at the same mass concentration, hence this mechanism of particle precipitation may turn out to be more effective just in the case of fine particles. Another factor here is, apparently, the size of particles d_S relative to the thickness of the viscous sublayer δ. When ratio d_S/δ is large, the lifting force can more easily eject the particles from the boundary zone at the duct wall. We note the agreement here with the well known fact [9] that particles lying on the bottom are blown off when the stream velocity becomes such as to make $\delta < d_S$. In our tests $d_S/\delta = 0.27$ for the 0.065 mm particles and $d_S/\delta = 0.75$ for the 0.180 mm particles, which indicated more favorable conditions for an accumulation of fine particles at the bottom of the duct.

We note, in conclusion, that these factors are not the only ones which would explain our test results. The distribution of particles is, apparently, also affected in some way by the frequency of their collisions with the duct wall. The significance of this effect increases with the gas velocity and is most strongly felt in ducts with a small diameter. This, in particular, explains why the results of the earlier study [4] with the same particles in 8.16 mm horizontal ducts yielded differences between the heat transfer at the bottom and at the top generatrix which did not exceed the limits of test accuracy.

NOTATION

w	is the gas velocity;
v_{sus}	is the soaring velocity;
v_*	is the dynamic velocity;
d, d_S	is the diameter of duct and mean diameter of solid particles respectively;
α	is the heat transfer coefficient;
q	is the thermal flux density;
t_W, t_S, t_1	is the temperature of wall, stream, and at the entrance to the stream respectively;
G, G_S	is the mass flow rate of gas and of particles respectively;
c, c_S	is the specific heat of gas and of particles respectively;
$K = G_S/G$	is the mass concentration;
x	is the longitudinal coordinate;
ν	is the kinematic viscosity;
λ	is the thermal conductivity of gas;
ρ, ρ_S	is the density of gas and of particles respectively;
τ, τ_S	is the characteristic time scales for the gas and for the particles respectively;
$Re = wd/\nu$	is the Reynolds number;
$Nu = \alpha d/\lambda$	is the Nusselt number;
Nu_0	is the Nusselt number for the gas.

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