LOCAL HEAT TRANSFER OF A GASEOUS SUSPENSION IN AN ANNULAR CHANNEL

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Using a closed-circuit apparatus, in which the working part is a channel of annular cross section, the coefficients of local heat transfer are measured for flows of pure gas and air containing suspended graphite particles.

The heat-transfer properties of gaseous suspensions are particularly interesting in view of the complexity of the phenomena taking place and the excellent prospects of their practical use. Despite the relatively large number of investigations recently carried out in various countries, many of the problems involved have still not been fully examined, for example, the effect of the geometry of the flow on the heat-transfer coefficients of the suspension in relation to the basic parameters (temperature, rate of flow, concentration of the material). A large proportion of existing data relate to tubes (pipes) of circular cross section, and only in a few cases has the heat transfer associated with the flow of suspensions in annular channels been considered [1-3]. In view of the importance of this configuration in connection with the possible future use of suspensions in the cooling of nuclear reactors, we shall here consider local heat transfer in a channel of this kind. It was emphasized in earlier papers [4, 5, 6, 8] that a knowledge of local heat transfer in the system under consideration should promote a better understanding of the phenomenon and also help in optizing future constructional design.

We accordingly measured the local heat-transfer coefficients for a constant heat flux in channels of annular cross section passing a flow of fine air-suspended graphite particles having a mean diameter of 3 μ.

Experimental Apparatus. In order to measure the heat transfer coefficients we used an installation (Fig. 1) based on a closed circuit, the gas and solid particles being simultaneously driven around this by means of a Roots air blower. The experimental apparatus was no different in principle from that used in [7]. The channel of annular cross section was formed by a seamless tube 30 mm in diameter (with a wall thickness of 1 mm), and 3000 mm long; inside this was a cylinder 16 mm in diameter (or a seamless tube with an internal diameter of 14 mm and an external diameter of 16 mm, closed at both ends). The material of both tubes was stainless steel of the 304 type.

The outer wall of the annular channel (only) was heated by the direct passage of an electric current. Thermocouples were placed on two opposite generators of the outer surface of the external tube, in seventeen heated sections distributed along the channel. The heated tube was thermally insulated from the surrounding medium by scattering sillit powder around the tube in a sheath 200 mm in diameter.

The inner cylinder projected from the heat section by 1400 mm, i.e., by 100 equivalent channel diameters ($d_e = D_1 - D_2$), creating a stabilized section which was regarded as sufficient to produce a fully-developed velocity profile at the entry into the working section 1. The annular cross section also had an adiabatic section 3, 600 mm long. The inner cylinder was suspended on a cross-piece intersecting the adiabatic section along the diameter, and was centered by means of special screws located in the textolite coupling section 9. The lower end of the inner cylinder was made in the form of a cone with an angle of 15° in order to reduce the pressure losses and ensure a more uniform distribution of the particles.

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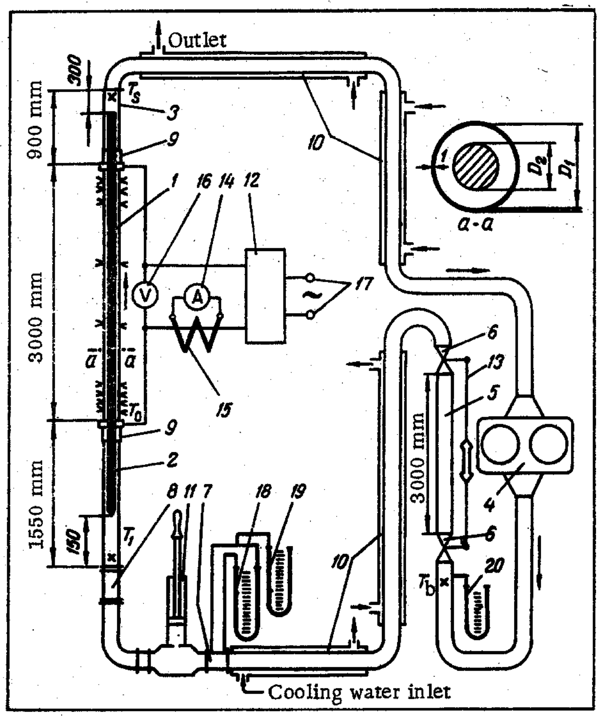

Fig. 1. Arrangement of the apparatus: 1) Heat-transfer tube; 2) flow-stabilization section; 3) adiabatic section; 4) Roots air blower; 5) device for measuring the concentration of the suspension; 6) synchronously-acting valves; 7) Venturi tube for measuring the gas flow; 8) Venturi nozzle for redistributing the solid phase; 9) textolite couplings; 10) water-cooled jackets; 11) feed device for solid phase; 12) power transformer; 13) rod; 14) ammeter; 15) current transformer; 16) voltmeter; 17) terminals of supply source; 18)-20) manometers.

For measuring the temperature of the suspensions at the inlet and outlet, thermocouples T_1 and T_s were placed along the axis of the tube in the stabilized section, in front of the inlet into the annular channel and in the adiabatic section beyond the upper end of the annular channel respectively.

The concentration of the suspension was measured by the method of [7], using the trapping device 5; the concentration was determined from the known volume of the tube 5 and the weight of the trapped particles.

Experimental Procedure. The system was calibrated by measuring the local heat-transfer coefficients in a flow of pure air.

The thermal losses through the insulation of the experimental tube were determined by heating the tube, without any circulating flow, to a temperature approximately equal to the average wall temperature during the tests, and measuring the corresponding electrical power.

In order to set the system in motion, an air blower was connected, solid particles were introduced into the circulating flow, and the electrical power was regulated. Readings were taken with the system in a steady state of operation.

The air–graphite suspension was obtained by introducing a specific amount of relatively large graphite particles into the system, the mean diameter of these fell to 3 μ after continuous circulation for some 15 min (maximum 5 μ); the size was determined by means of a microscope. The particle-size spectrum was not specially rechecked – the foregoing dimensions appeared to remain fairly constant after various periods of circulation. The initial reduction in particle size was probably due to their passage through the Roots air blower, or to collisions with the walls.

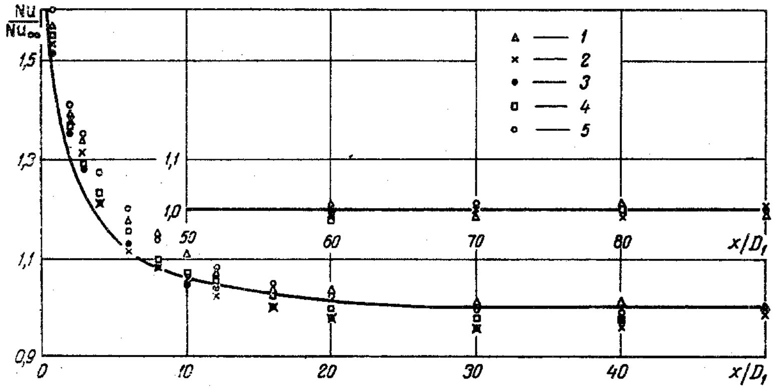

Fig. 2. Local changes in the heat-transfer coefficient for a flow of pure air: 1) Re = 12.6; 2) 27.8; 3) 29.0; 4) 32.0; 5) 32.7; curve = analytical solution [5].

In calculating the wall temperature, the arithmetic mean of the readings of opposite thermocouples was taken, without allowing for the temperature gradient through the wall thickness, which in the present investigation was very slight. The temperature difference between opposite thermocouples in each section never exceeded 1.5°C in the experiments with pure air, or 4°C in those with a gaseous suspension.

Method of Analyzing the Experimental Data. The experimental results were expressed in the form of Nusselt numbers, or the ratios of Nusselt numbers, as functions of the axial distance and the concentration of the solid particles. The local coefficients of heat transfer were calculated from the volume-average temperature of the flow T_b:

$$h = q/(T_w - T_b). \tag{1}$$

The value of the thermal flux q was derived from the amount of electrical power dissipated by the outer tube (allowing for thermal losses through the insulation), and was regarded as uniform along the whole working section. The temperature dependence of the electrical resistance was not taken into account, and this led to certain errors in determining the heat-transfer coefficient at the ends of the tube by means of Eq. (1).

The local volume-average temperature of the flow was determined by linear interpolation of the temperature of the suspension measured at the inlet and outlet of the working section.

In the experiments with suspensions, the gas flow was determined by reference to the thermal balance of the working section, on the assumption that no slip existed between the phases ($r = G_p/G_g$):

$$G_g = Q/\left[(c_g + rc_p)(T_s - T_1)\right], \tag{2}$$

where Q is the total amount of heat transferred from the tube to the suspension. The only shortcoming of this method lies in the fact that the flows obtained in different experiments assume random values; however, in the present investigation the Reynolds numbers were varied as little as possible.

The Reynolds number was calculated from the rate of flow of the gas phase and the equivalent diameter $d_e = D_1 - D_2$:

$$\mathrm{Re} = 4G_g/\pi d_e \mu. \tag{3}$$

The Nusselt number was found from the thermal conductivity of the gas and the equivalent diameter:

$$\mathrm{Nu} = hd_e/k_g. \tag{4}$$

For the experiments with pure air, in which the rate of flow was determined by means of a Venturi tube, the local volume-average temperature was found more precisely, by reference to the thermal balance of the section between the inlet and the cross section X:

$$T_{bg} = T_1 + \pi D_1 qX/G_g c_g. \tag{5}$$

The thermodynamic properties of the two phases were calculated at a temperature intermediate between those existing at the inlet and outlet of the experimental tube.

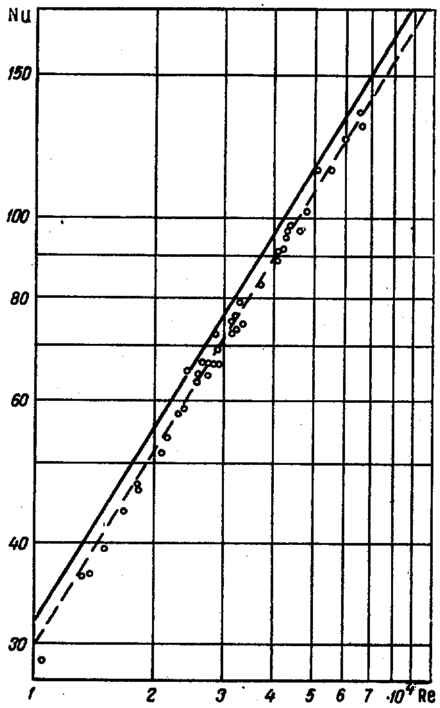

Fig. 3

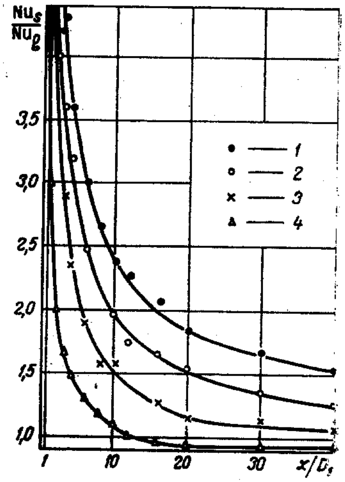

Fig. 4

Fig. 3. Heat-transfer coefficients in relation to the mode of flow: points represent experimental data, the broken line constitutes the graphical form of [7], the continuous line represents the McAdams correlation.

Fig. 4. Effect of the solid phase on heat transfer as a function of the distance from the inlet into the heat exchanger for various concentrations (r) of the solid phase: 1) r = 25.2, Re = 15.6; 2) respectively 22.3 and 20.4; 3) 13.8 and 31.5; 4) 2.5 and 15.0.

An estimation of the accuracy of the experiments aimed at determining the heat-transfer coefficient between the wall and the flow of gaseous suspension gives a maximum error of ±12% and a mean error of ±6%.

Results of the Experiments. The results obtained from the experiments may be divided into two groups: the fundamental group, including the experimental data relating to the suspensions, and the auxiliary group, including the results of the calibrating experiments with pure air.

Heat Transfer in Pure Air. The calibrating experiments with pure air flows were carried out for Reynolds number of 10,000 to 60,000. Since the validity of the results obtained with the experimental apparatus employed was verified earlier [7], the results of our present experiments on heat transfer in pure air may be of independent interest.

The heat-transfer coefficient is shown as a function of distance along the axis in Fig. 2, together with Sparrow's theoretical curve [5]. The agreement between the experimental points and the theoretical curve is excellent, although the curve was actually obtained for a tube of circular cross section. In the thermal inlet section, the experimental points lie slightly higher than the theoretical curve, while the outlet section (ending at the point in which the heat-transfer coefficient reaches 105% of its asymptotic value), roughly 13 diameters long, may be calculated in the same way as in [5]. A similar result was also obtained in [4] and [7]. In the asymptotic region the maximum scatter of the experimental points equals ±4%.

In Fig. 3 the experimental values of the asymptotic heat-transfer coefficients (in the form of Nusselt numbers) are shown in relation to the Reynolds number, in accordance with the Dittus–Boelter equation

$$\mathrm{Nu} = 0.023\,\mathrm{Re}^{0.8}\mathrm{Pr}^{0.4}, \tag{6}$$

which gives the average values of the heat-transfer coefficients. The experimental value lie approximately 6% below the criterial values owing to the influence of the thermal inlet section. In actual fact, if we

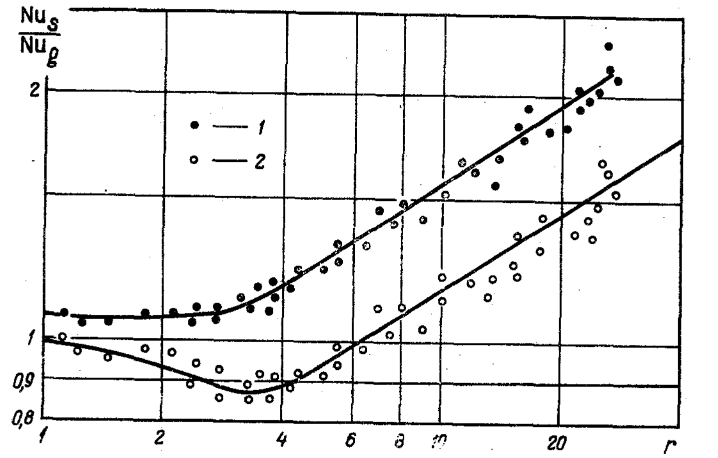

Fig. 5. Relative influence of the solid particles on heat transfer in relation to the concentration of the suspension, for two distances from the inlet into the heat-exchanger: 1) $x/D_1 = 10$; 2) $x/D_1 = 30$ diameters. $r = G_p/G_g$.

integrate the curve of Fig. 3 graphically, we obtain a coefficient of 1.056 for transferring from the asymptotic to the mean values. Thus in the case of an annular cross section Eq. (6) gives the correct mean values for the heat-transfer coefficients in tubes with a length equal to the length of the tubes used in the present investigation (100 diameters). For shorter tubes Eq. (6) may be refined by using the curve of Fig. 2.

The experimental asymptotic values of the heat-transfer coefficients may be generalized by the equation

$$\mathrm{Nu} = 0.0187\,\mathrm{Re}^{0.8} \tag{7}$$

after taking Pr = 0.7.

Heat Transfer in Experiments with an Air—Graphite Suspension. In the experiments with the gaseous suspension, the concentration of the solid phase was varied from 0 to 27 kg of graphite per kg of air. The Reynolds number was varied roughly between 18,000 and 25,000, while the corresponding velocities of the gas phase were 19-28 m/sec. The temperature of the suspension at the inlet into the experimental section (according to the readings of thermocouple T_1, Fig. 1) fluctuated from 50 to 70°C, while the temperature difference between the inlet and the outlet was about 10°C. The maximum temperature of the tube wall (at the highest measuring point, 90 diameters from the inlet) was maintained at 150°C by constantly regulating the heat flux, which in the present investigation varied over the range $4.2 \cdot 10^3$–$8.4 \cdot 10^3$ W/m^2.

All the experiments were carried out with the tube in a vertical position.

The experimental data were generalized after simply allowing for the influence of the concentration of the solid phase and the longitudinal coordinate on the heat-transfer coefficient, but neglecting the influence of the Reynolds number on the $\mathrm{Nu_s/Nu_g}$ ratio, since, for the narrow range of Reynolds-number measurements embraced, this influence was commensurate with experimental error [1, 9, 7].

The changes taking place in the heat-transfer coefficients on passing along the axis of the tube are shown in Fig. 4, in the form of the ratio of the local Nusselt number of the suspension to the local Nusselt number of the pure gas. Apart from the fact that the increase in the local Nusselt number is associated with the increase in the concentration r, it may also be partly due (see Figs. 2 and 4) to the elongation of the inlet section, which for the gaseous suspension extends through a distance of more than 13 diameters (this latter figure being characteristic of a flow of pure gas). On analyzing the experimental data relating to distances from the inlet greater than those illustrated in Fig. 4, we found a tendency for the heat-transfer coefficients to fall continuously with increasing distance, so that (for higher concentrations) we were unable to determine the initial thermal section precisely. This continuous fall in $\mathrm{Nu_s}$ with increasing distance from the inlet (up to the maximum value) may be explained by the fact that, in determining the heat-transfer coefficients, any changes in the electrical resistance of the heat-exchange tube with temperature were neglected; thus in Eq. (1) the value of the thermal flux q was taken as being smaller than the real value. In actual fact the values of h calculated for the hotter end of the tube from Eq. (1) are smaller than the real values, and the error increases on increasing the distance from the inlet. In order to eliminate this error by allowing for changes in the electrical resistance with temperature we should have to abandon the assumption that the heat flux in the tube was constant. This assumption is reasonably justifiable for the lower values of heat flux. In order to preserve a unified treatment of all the results, we therefore made no corrections to the heat flux values, but used an average value for the whole tube; this gave a correct result for at any rate the middle section of the tube.

Figure 5 illustrates the Nusselt number as a function of r for distances of 30 and 10 diameters. For a distance of 30 diameters and over, the minimum heat-transfer coefficients occurred for a solid-phase concentration of between 3 and 4 kg of graphite/kg of air; this figure is comparable with those of [4, 6, 7]. For smaller axial distances no heat-transfer minimum could be distinguished. Thus we see from Fig. 5 that for an axial distance of 10 diameters there is no such minimum.

The experimental results of the present investigation may be generalized for concentrations of over 4 kg of graphite/kg of air and axial distances shorter than 50 diameters by the following empirical equation:

$$\mathrm{Nu}_s/\mathrm{Nu}_g = 1.32\,(x/D_1)^{-0.257}\,(1 + c_p/c_g r)^{0.360} \tag{8}$$

which is valid to an accuracy of $\pm 14\%$.

Discussion of Results. In the experiments with pure air, the changes taking place in the heat-transfer coefficient with distance along the axis, also the length of the "five-percent" initial section (expressed in diameters of the outer tube D_1), are practically identical for the annular channel and the tube of circular cross section (diameter D_1). This means that, under the conditions of our experiments, the thermal boundary layer developed in the heat-exchange tube independently of the presence of the concentric inner cylinder. These observations show that, for the channels under consideration, the heat-transfer coefficients may be generalized (with due allowance for the mode of flow) by the Dittus–Boelter equation for tubes of annular cross section, where the Nusselt and Reynolds numbers are determined from the equivalent diameter $d_e = D_1 - D_2$.

For the experiments with suspensions, the results have the same general character as for the experiments in the circular tube, namely: an increase in the length and significance of the initial thermal section, and an increase in the absolute values of the heat-transfer coefficients with increasing concentration of the material, and also the existence of a minimum in the wall heat-transfer characteristic for a concentration of between 3 and 4 kg of graphite/kg of air. However, our results differed sharply from the measurements of [7], which were carried out in a circular tube with the same graphite suspension, in that, in the present investigation, $\mathrm{Nu}_s/\mathrm{Nu}_g$ increased much less rapidly with increasing concentration of the material r than in the tube of circular cross section. A qualitative explanation may be given for this fact if we examine the heat-transfer coefficients h instead of the Nusselt numbers. In fact the heat-transfer coefficients for pure air in an annular tube are $D_1/(D_1 - D_2)$ times greater than the coefficients for a circular tube of the same diameter D_1 and the same mode of flow (same Re number); this means that the presence of the inner cylinder in the heat-exchange tube produces a change in the shape of the temperature distribution curve across the cross section, the corresponding temperature gradients becoming sharper. In this case, when a suspension is used as the heat-transfer medium instead of air, the influence of the solid particles is considerable for both the annular and the circular cross sections; it causes a smoothing of the temperature profile in the center of the cross section, so that the presence of the inner cylinder has hardly any influence on the temperature gradients around the tube wall. This assertion is supported by the values of the heat-transfer coefficients obtained in the present investigation and in [7], which are approximately equal for the same material concentrations and Reynolds numbers. We may thus conclude that the ratio h_s/h_g (or $\mathrm{Nu}_s/\mathrm{Nu}_g$) obtained for the tube of annular cross section should be reduced by $D_1/(D_1 - D_2)$ times by comparison with the round tube for the same concentration r; this formally brings the results obtained in the present investigation and those of [7] into a state of agreement.

Another characteristic of our present results lies in the fact that the 5% thermal inlet section cannot be determined for higher concentrations. We find that the local heat-transfer coefficients fall continuously on approaching the end of a tube 90 diameters long. This is mainly due to the temperature coefficient of electrical resistance of the tube wall material, and thus to the change in the heat flux along the tube. Hence the real situation differs from that relating to a constant heat flux, especially for high thermal loads.

There is yet another reason for the restriction of the thermal flux to low values, namely, the difficulty of cooling the suspension after it has passed through the heated section and the Roots air blower.

For the same reason the concentration of the solid phase was limited to r = 27 kg of graphite/kg of air, since for large concentrations the temperature of the suspension at the outlet from the air blower increased to values too great for the cooling capacity of the water jackets 10 in Fig. 1, and the temperature T_1 at the inlet became too high. (For this reason the working values of the temperature T_1 were also taken quite high, roughly equal to 70-80°C.)

In the present investigation we discovered yet another effect associated with the cooling of the suspension under examination (graphite particles with a mean size of 3 μ), which we attempted to carry out by means of a tubular heat-exchanger specially made for this purpose. The heat-exchanger consisted of a pile of copper tubes with an internal diameter of 8 mm, through which the suspension passed, while cooling water flowed through a jacket around the tubes. The number of tubes (20) was chosen so as to make their total cross-sectional area equal to the cross section in the other parts of the closed circuit, and so as to preserve the same rate of flow of the suspension. Despite all the measures taken to ensure favorable conditions for the entry of the suspension into the heat-exchanger, the graphite particles settled on the walls of the cooled tubes and blocked them completely. The settling of the solid particles was due to the negative temperature gradients on the cooled surface. This was confirmed in the following way. When no water was fed into the heat-exchanger jacket, the suspension circulated (uncooled) until an equilibrium temperature was reached in the heat-exchanger (inner tube). In this case no settling was encountered; the flow parameters in the closed circuit remained constant. However, if cold water were suddenly passed into the jacket, settling commenced, and after a short while the solid particles had completely blocked the tubes. This settling of particles on the cooled surface, due to the radiometric effect, creates great difficulties in the practical use of suspensions containing very fine particles.

NOTATION

T_1 is the temperature at the inlet;
T_s is the temperature at the outlet;
T_0 is the temperature of the tube walls;
T_b is the local volume temperature;
G is the mass flow rate;
q is the heat flow (thermal flux);
r is the concentration of solid particles;
h is the local heat-transfer coefficient;
c_p is the specific heat of the solid particles;
c_g is the specific heat of the gas at constant pressure;
Re is the Reynolds number;
Nu is the Nusselt number;
D_1 is the internal diameter of outer tube;
D_2 is the external diameter of inner tube;
d_e is the equivalent diameter;
μ is the viscosity;
k is the thermal conductivity;
x is the distance along the tube.

Subscripts

s denotes the suspension;
p denotes the particles;
g denotes the gas;
∞ denotes the asymptotic value.

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