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THROUGHFLOWS OF THE "GAS-SOLIDS" TYPE AS HEAT-TRANSFER AGENTS

V. V. Sapozhnikov and N. I. Syromyatnikov

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Experimentally obtained data are used to demonstrate the advantages of a gas suspension as a heat-transfer agent.

The question of intensifying heat transfer by using heat-transfer agents of the "gas-solids" type is a very topical one. Therefore, the study of the corresponding heat transfer and hydrodynamics in channels of various sections and, in particular, annular channels is of considerable practical interest.

Whereas there have been a number of investigations of the hydrodynamics and heat transfer of gas suspensions in circular channels ([1–5], etc.), the authors are aware of only two papers dealing with heat transfer in annular channels [1, 3], while there is practically no information concerning the corresponding hydrodynamics. Our research relates to the experimental study of the heat transfer and hydrodynamics of an air-graphite flow in annular channels.

The experimental apparatus (Fig. 1) took the form of a closed loop. The experimental section was an annular channel with $d_2/d_1 = 2.63$; 1.75 and 1.47 and a length of 1.5 m. The equivalent diameters were 19.6, 13.6, and 10.3 mm, respectively. We simulated both external heating and heating by an internal source. The experimental method was based on the steady-state regime. The concentration was measured by the cutoff method, checked in relation to the charge and in certain experiments the heat balance of the heat exchanger. The average discrepancy was 7–8%. The graphite-air flow rate was measured by the thermal marker method [10] and checked against the flow rate determined from the balance for the heat exchanger (the discrepancy did not exceed 6–7%). The heat flux was measured from the electrical power consumption, taking into account the losses through the insulation, and checked against the change in the flow enthalpy in the experimental channel.

The experiments were performed on natural graphite ($\gamma = 2170$ kg/m^3) at an average particle size on the order of 10 μ. The particle size was determined by the experimental apparatus.

The method was first tested on pure air, as a result of which it was found that the average heat transfer for air in annular channels at $3000 \leq \mathrm{Re} \leq 30\,000$ (correct to 3–4%) conforms with the formula presented in [7]:

$$\mathrm{Nu}_g = n\,\mathrm{Re}^{0.8}(d_2/d_1)^{0.45}, \tag{1}$$

where n = 0.018 for internal heating and n = 0.0168 for external heating at $d_2/d_1 = 2.63$. At $d_2/d_1 = 1.48$ and 1.75 the difference in heat transfer for the different methods of heating did not exceed 3–4%, which is in agreement with the data of [6]. The hydrodynamic data in this region are perfectly satisfactorily described by the Darcy formula

$$\lambda_{\mathrm{fr}} = 0.316\,\mathrm{Re}^{-0.25}. \tag{2}$$

In the laminar regime $1300 < \mathrm{Re} < 2300$ the experimental heat-transfer data are satisfactorily described by the formula [7]

$$\mathrm{Nu}_g = \left(\frac{wc_p}{\lambda L}\right)^{0.45}\mathrm{Gr}^{0.05}(d_2/d_1)^{0.8} \tag{3}$$

for external heating at $d_2/d_1 = 1.48$ and 1.75, and the hydrodynamics by the formula

$$\lambda_{\mathrm{fr}} = 64/\mathrm{Re}. \tag{4}$$

In relation to the graphite-air suspension, we investigated the effect of the flow velocity and the solids concentration on the convective heat transfer from the heated wall to the suspension, the pressure losses, and the effect of the equivalent diameter and heat flux. As the characteristic dimension we took the hydraulic diameter and as the characteristic temperature the mean flow temperature. The principal parameters were varied over the following ranges: concentration from 0 to 325 kg/kg, flow velocity from 2.0 to 29 m/sec, flow temperature from 53 to 195° C, heat flux from $4.9 \cdot 10^3$ to $20.6 \cdot 10^3$ W/m^2.

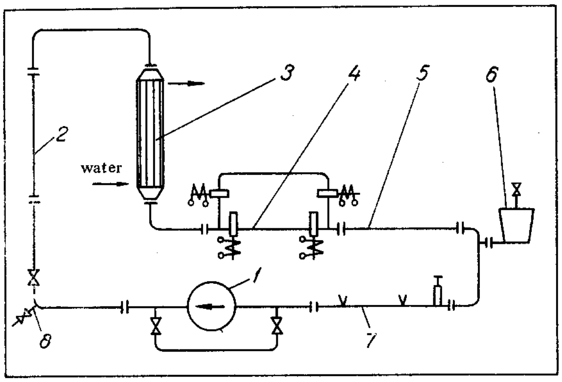

Fig. 1. Diagram of experimental loop: 1) rotary air blower; 2) experimental section; 3) heat exchanger; 4) interceptor; 5) hydrodynamic section; 6) feeder; 7) thermal marker section; 8) outlet valve.

The results of the experiment were correlated in criterial form, as the ratio of the heat-transfer coefficient of the disperse flow to the corresponding quantity for the pure gas. (Fig. 2)

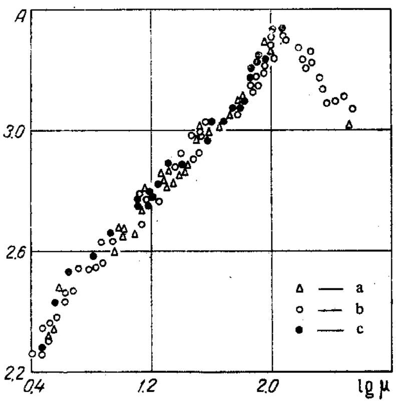

Fig. 2. Generalized concentration dependence of heat transfer for external heating: a) d_e = 10.3 mm b) 13.6; c) 19.6, $A = \lg[(\mathrm{Nu}_f/\mathrm{Nu}_g - 1)\mathrm{Re}^{0.7}(d_2/d_1)^{-0.95}]$.

The correlation of the experimental data gives an empirical relation for calculating the mean heat transfer from the heated wall to the ascending graphite-air flow:

$$\mathrm{Nu}_f/\mathrm{Nu}_g = 1 + c\,\mathrm{Re}^m \mu^n (d_2/d_1)^{0.95}, \tag{5}$$

where for external heating c, m, and n take the following values:

a) for the region $0.6 < \mu < 4$, $4000 < \mathrm{Re} < 26000$

$c = 244;\ m = -0.8;\ n = 0.67;$

b) for $4 < \mu < 45$, $2800 < \mathrm{Re} < 12000$

$c = 112;\ m = -0.7;\ n = 0.6;$

c) for $45 < \mu < 65$, $2000 < \mathrm{Re} < 8300$

$c = 151;\ m = -0.7;\ n = 0.5;$

d) for $60 < \mu < 120$, $1800 < \mathrm{Re} < 7400$

$c = 58$; $m = -0.7$; $n = 0.75$;

e) for $120 < \mu < 325$, $1370 < \mathrm{Re} < 4300$

$c = 0.8 \cdot 10^5$; $m = -0.7$; $n = -0.75$.

In the case of internal heating, the heat transfer is somewhat higher, which is reflected in an increase in the exponent of d_2/d_1 from 0.95 to 1.05. None of the formulas has an error greater than ±9%. With an error of ±12% the experimental data for external heating in the region $3 < \mu < 120$ and $2300 < \mathrm{Re} < 26\,000$ can be represented by a single formula:

$$\frac{\mathrm{Nu_f}}{\mathrm{Nu_g}} = 1 + 107\,\mathrm{Re}^{-0.7}\,\mu^{0.6}(d_2/d_1)^{0.95}. \tag{6}$$

The experimental data on the heater hydrodynamics (Fig. 3) are described by a relation of the type

$$\frac{\Delta P_f}{\Delta P_g} = 1 + k\mu, \tag{7}$$

where the experimental coefficient

$$k = c\,\mathrm{Re}^n\,(d_2/d_1)^{0.145}. \tag{8}$$

The values of c and n were determined in four regions of μ and Re:

a) for $0.6 < \mu < 4$, $4000 < \mathrm{Re} < 26000$

$c = 0.85$; $n = -0.2$ (error ±13%);

b) for $5 < \mu < 45$, $2800 < \mathrm{Re} < 12000$

$c = 1.08$; $n = -0.182$ (error ±9.2%);

c) for $45 < \mu < 120$, $2300 < \mathrm{Re} < 8300$

$c = 1.79$; $n = -0.25$ (error ±9.8%);

d) for $120 < \mu < 325$, $1370 < \mathrm{Re} < 4300$

$c = 0.445$; $n = -0.08$ (error ±12.7%).

The hydrodynamics data can also be represented by a single formula over the entire range of concentration:

$$\Delta P_f = 0.0155\,\gamma_f^{0.82} w_f^{1.82}\,\frac{l}{d_e}\,(d_2/d_1)^{-0.2},\ \mathrm{N/m^2} \tag{9}$$

for $23 \leq \gamma_f w_f \leq 1200$ kg/m²·sec or for the turbulent regime

$$\frac{\Delta P_f}{\Delta P_g} = 0.096\left(\frac{d_e}{\eta}\right)^{0.25}(1+\mu)^{0.82}(d_2/d_1)^{-0.2}(w_g\,\gamma_g)^{0.07} \tag{10}$$

with a mean error of ±11.2%. The hydrodynamic equations were obtained for a stabilized section on the order of 840 mm. In all the heat transfer and hydrodynamic calculations, it was assumed that there was no interphase slip and that the flow-rate concentration was equal to the mass concentration (this is quite permissible for micron-size particles).

An analysis of the results basically confirms previous ideas about the heat transfer of dustladen flows [1, 2] and, in particular, the existence of a critical concentration $\mu = 40-50$ kg/kg and an optimum concentration in the region $\mu = 110-120$ kg/kg. The existence of "critical" concentrations suggests an internal reconstruction of the flow, a change in its aerodynamics and a change in the ratio of the thermal resistances of the boundary layer and the flow core depending on the state of saturation of the flow with solids. At $\mu < \mu_{cr} \cong 45$ kg/kg heat transfer is intensified by the action of the particles on the boundary layer and because the particles participate in radial heat transport, being entrained by the turbulent fluctuations. On transition through the critical concentration (region IV) the flow becomes more restricted, laminarization takes place, the flow velocity falls and consequently the part played by $\mathrm{Nu_g}$ diminishes. In the region $\mu > 120$ (region V) the principal role is played by the thermal resistance of the flow core, which increases with μ, thus reducing the rate of heat transfer. In the region μ = 110–120 kg/kg maximum heat transfer, at which Nu_f/Nu_g reached 16–20, was observed. Our study of the hydrodynamics also showed that there are several different regions of influence of the concentration on the hydrodynamic resistance of the channel and that the concentration is decisive in the pressure loss calculations. The coefficient k as a function of Re, d_2/d_1, etc. actually varied between 0.1 and 0.4 and depended only rather weakly on the velocity. A correlation of the experimental results by analogy with [3] gives satisfactory agreement with a correlation based on the Gasterstädt equation, but the physical significance of the processes is not clearly reflected, which is one of the shortcomings of this method.

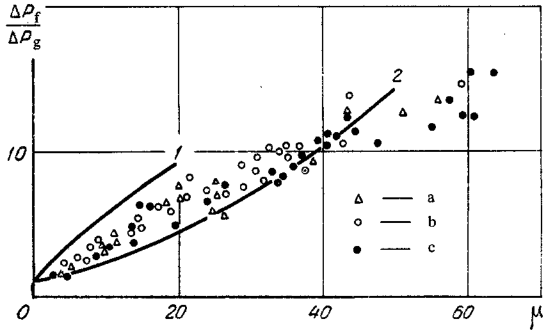

Fig. 3. Pressure drop ratio as a function of concentration (a–c—see Fig. 2): 1) according to Schluderberg's formula for tubes [3]; 2) according to the formula of Hawes et al. for nitrogen at a tube inside diameter of 12.7 mm [4].

Comparison with the heat-transfer data of other authors (Fig. 4) showed fairly good agreement with the data of [3]. It was less easy to compare our data with those of [1] for external flow since the particles employed were larger (140 μ or more), but it should be noted that our data lie above those of [1], which reflects the greater intensifying role of the fine particles and the fact that the experiments in [1] were concerned with cooling in an annular channel, whereby graphite deposits may be formed on the cold wall at low temperatures.

The hydrodynamics (Fig. 3) data are in fairly good agreement with those of [3–5] for finely dispersed flows in circular tubes.

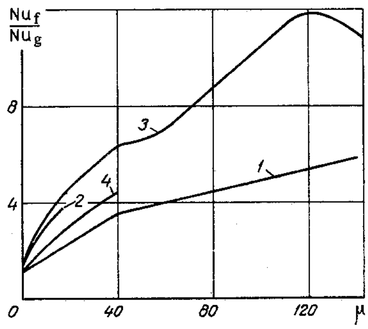

Fig. 4. Comparison of heat-transfer data with the data of other authors (Re = $7 \cdot 10^{-3}$): 1) Bakhtiozin's and Gorbis' formulas [1] for particles measuring 140 and 165 μ and d_e = 20 mm; 2) from Schluderberg's formula for nitrogen at d_2/d_1 = 1.32 [3]; 3) our data for d_e = = 19.6 mm; 4) according to [4].

As an example, the relations obtained made it possible to carry out thermal calculations by the simplified method of [9–11] for the steam generator and reactor of a two-loop atomic power plant with a helium-graphite heat-transfer agent. In the calculations it was assumed, in accordance with [5], that the effect of the properties of the gas phase is taken into account by introducing the ratio c_t/c_{pg}. For comparison we made similar calculations for pure helium assuming the same thermal capacities, the same design and the same structural materials. For example, at an electrical capacity of 200 mW for a steam generator of the "tube-in-tube" type the gain in heating surface with a helium-graphite suspension (at P = 20 bar) was by a factor of 4.5, while the pumping losses were smaller by a factor of 5 than for a steam generator with helium at 60 bars. A similar comparison for heterogeneous thermal-neutron reactors showed that the core dimensions can also be reduced by a factor of 5, and the pumping losses by a factor of 6. The efficiency of the plant increases by 3–4%. In both cases, the comparison was carried out for variants selected from calculations of the optimum concentration, diameter, and velocity. Thus, for example, for the steam generator with a suspension, the values of these quantities were: μ = 100 kg/kg, d_e = 39 mm, and w_f = 10 m/sec, while for the reactor at the same concentration the core velocity was 6 m/sec and the equivalent diameter 11.5 mm at a channel length of 4.8 m and t_f^{av} = 525° C.

In conclusion, we note that the calculated data correspond to the results of the calculations given in [1, 3]. Moreover, as our experiments and the experiments in [2–5] (with micron-size graphite particles) showed, there is almost no abrasive wear. The suspension is easily displaced, regulated and bypassed.

NOTATION

Re, Nu_g, ΔP_g, λ_{fr} are the criteria, pressure losses, and friction factor for pure gas; Nu_f, ΔP_f are the Nusselt number and friction pressure losses for suspension flow; γ_f and w_f are the specific weight and flow velocity of the suspension; d_2, d_1, and d_e are the outside, inside, and equivalent channel diameters; μ is the mass concentration, kg/kg; η is the dynamic viscosity.

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Kirov Polytechnic Institute,
Sverdlovsk