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INVESTIGATION OF HEAT TRANSFER IN THE SPACE ABOVE AN INHOMOGENEOUS FLUIDIZED BED

A. P. Baskakov and S. V. Zvyagin

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Heat transfer from the rarefied to the dense phase of a fluidized bed by feeding a stream of hot air to the rarefied phase (space above the bed) was investigated.

In a number of studies [1, 2] devoted to the flow of horizontal streams (isothermal and nonisothermal) into a fluidized bed the length of the stream was found and the problem of heat transfer from the jet into the fluidized bed was solved. However, for some fluidized bed processes there is a need for a marked increase of the range of the horizontal streams, which is attained upon outflow of the stream into the space above the bed [3].

Investigations of the structure of the above-bed space [4-6] show that there are two zones with a different character of the distribution of the solid phase: a lower ejection (splash) zone and an upper fluidized bed (pneumatic transport) zone. The relationship of the heights of these zones depends on the size of the particles of the fine-grained material and fluidization regime. Some theoretical considerations on the structure of the zones are given in [4]. In the splash zone the average concentration of particles drops steeply with respect to height and the relative fluctuations of the density are high. In the pneumatic transport zone the average concentration of particles is 2-3 orders lower than in the main bed, and the relative fluctuations are small and are determined mainly by the degree of turbulence of the flow.

Investigations of the above-bed space in apparatuses of various sizes by the filtration method [5], cutoff method [6], and also by a capacitance-type transducer and the x-ray method [4] showed that in semilogarithmic coordinates the height distribution of the concentration (at a constant velocity of the gas flow) is described by two straight lines. The first characterizes the marked drop of the concentration of particles with height in the splash zone and corresponds to the region of inertial movement of the particles. The second more sloping line corresponds to a decrease of the concentration of particles with height in the pneumatic transport zone. The point of inflection of the lines of concentration with height corresponds to the boundary between the splash and pneumatic transport zone.

The decrease of the particle concentration in the splash zone is easily explained [4]: the greater the height, the fewer particles thrown up to this height. The causes of the decrease of particle concentration over the height of the pneumatic transport zone are not discussed in the literature. In the case of a gas velocity constant over the cross section and in time there should be no particles at all in this zone, especially particles circulating between this zone and the fluidized bed, since the free-fall velocity of the particles used is greater than the average gas velocity. The presence of particles in this zone [6] and the sufficiently large coefficients of heat transfer between it and the core of the fluidized bed apparently are the consequence of gas circulation in the above-bed space, as a result of which the local and instantaneous gas velocities can differ considerably from the average not only in magnitude but also in sign. Actually, it is difficult to imagine that at a velocity of

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S. M. Kirov Ural Polytechnic Institute. Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 30, No. 1, pp. 14-17, January, 1976. Original article submitted February 12, 1975.

0.2-0.5 m/sec the flow, being continuously disturbed by bubbles breaking on the surface of the fluidized bed (and, moreover, charged with particles making its movement unstable in the gravitational field), has a uniform velocity distribution over the cross section of the chamber. In the given investigation circulation was unconditionally promoted also by the injection of high-velocity streams into this zone.

The intensity of heat transfer by particles into the fluidized bed from the zone of delivery of hot-air streams was investigated in a device measuring 310 × 250 × 750 mm made of chipboards and insulated with 50-mm-thick Porolon. Cold air ($t_{c.a}$ = 25-35°C) was supplied from below through a perforated textolite grating with a 0.7% clear opening covered with a fine metal mesh to eliminate spilling of the particles. White synthetic corundum of a narrow fraction with d = 0.355 mm and wide fraction d_{av} = 0.320 mm was fluidized in the device. The height of the bulk layer H_0 in all experiments was constant and equal to 240 mm. Preheated air ($t_{h.a}$ = 140-150°C) that formed a system of horizontal streams was fed through a 42-mm-diameter pipe with 6- and 4.3-mm-diameter holes into the space above the bed. The discharge velocity of the streams w_s was varied from 25 to 55 m/sec.

To conduct the experiment the device was brought to a steady-state regime and then by means of Chromel—Copel thermocouples with exposed junctions, which were connected to the PP-63 instrument, we measured the temperatures of the fluidized bed, hot and cold air, and mixture at the outlet and, by means of diaphragms, the flow rates of the cold and hot air. To prevent random inflow of air through the top of the device, its height was increased to 2 m by means of a wooden box of equivalent cross section which was covered with a perforated panel.

For the characteristic of the intensity of heat transfer by splashes from the rarefied into the dense phase of the fluidized bed we used the conditional heat-transfer coefficient α, which is determined by the formula

$$\alpha = \frac{Q}{F\,(t_{\mathrm{mix}} - t_{\mathrm{f.b}})}.$$

The flow of heat being returned to the bed by the particles that during splashing flew up to the stream supply zone is determined on the basis of the heat balance in the device: $Q = Q_{f.b} - Q_{c.a} + Q_l$ (Fig. 1a). In each operating regime of the device we determined three or four values of α, which were then averaged. The scatter of the values of α did not exceed 10% of the average. We thus obtained the experimental functional dependences $\alpha = f(w, w_s, H)$.

The intensity of heat transfer is determined by the quantity of particles circulating between the stream supply zone and fluidized bed. It is completely natural that the heat-transfer coefficient α decreases markedly with increase of the height of delivery of the streams, changing analogously to the change of particle concentration in the above-bed space [4-6].

We see from Fig. 1a that when the streams are delivered at a height of $(1-1.2)H_0$ the value of α remains practically constant. The concentration of particles in this zone is so great and the heat transfer from it is accordingly so intense that the difference of temperatures ($t_{mix} - t_{f.b}$) is small (fractions of a degree), which reduces the accuracy of determining α. When H < 60 mm the error in determining α can reach 40% even when measuring the different ($t_{mix} - t_{f.b}$) by a differential thermocouple. It also follows from Fig. 1a that in the splash zone the heat-transfer coefficient can be approximated in semilogarithmic coordinates by a straight line:

$$\alpha = \exp(aH + b)\ [\mathrm{kW/m^2 \cdot {}^\circ K}].$$

When d_{av} = 320 μm and w = 0.4 m/sec, a = −25 1/m, and b = 3.4.

In the pneumatic transport zone where the particle concentration depends more weakly on the height and is smaller in absolute value than in the splash zone, α depends more weakly on H. For this zone a = −6 1/m and b = 0.34 for the same d_{av} and w.

We also see from Fig. 1a that in the case of fluidization of the narrow fraction the heat-transfer coefficient in the splash zone is slightly less than for the wide fraction with practically the same average diameter of the particles (with respect to the weight composition of particles >385 μ, 10.6%; 300-385 μ, 77.2%; 250-300 μ, 7.2%; 200-250 μ, 4%; and <200 μ, 1%).

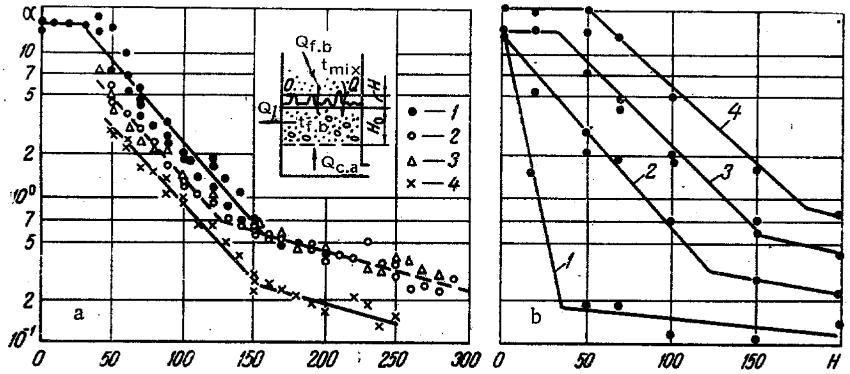

Fig. 1. Heat-transfer coefficient α ($kW/m^2 \cdot °K$) vs height of supply of streams H (mm) above the level of the bulk bed and diagram of heat fluxes in device: a) For H_0 = 240 mm, w = 0.4 m/sec. 1) d_{av} = 0.320 mm, w_s = 51 m/sec; 2) d = 0.355 mm, w_s = 51 m/sec; 3) 0.355 and w_s = 55; 4) 0.355 and 25.5. b) For H_0 = 240 mm, w_s = 51 m/sec, d_{av} = 0.320 mm. 1) w = 0.2 m/sec; 2) 0.3; 3) 0.4; 4) 0.5.

This is explained by the presence in the wide fraction of a certain amount of fines circulating in the splash zone and intensifying heat transfer in it.

In both zones the heat-transfer coefficient increases with an increase of w_s for constant hole diameters and temperature $t_{h.a}$ (Fig. 1a). A dependence of α on the diameter of the holes of the nozzles (upon a change of the diameter from 4.2 to 6 mm and constant flow rate of hot air) was not found. The decrease of α upon a decrease of the discharge velocity of the streams is related with a decrease of the striking range of the stream, since the stream energy is dissipated more rapidly, the smaller w_s.

Heat transfer is intensified considerably upon an increase of the velocity of the fluidizing agent (Fig. 1b) in the region of developed fluidization ($1.6 < W < 4$). Upon an increase of w from 0.2 to 0.5 m/sec, α increased by an order and more (depending on H). This agrees qualitatively with the results obtained on investigating [7] the supply of heat to the above-bed space by radiation. The maximum value of α obtained on delivering streams into the dense phase of the fluidized bed remains approximately constant at a level of 20 $kW/m^2 \cdot °K$ regardless of the rate of fluidization. As the streams are raised above the bed the heat-transfer coefficient drops more steeply, the smaller w, which corresponds to a more pronounced drop of the concentration. When w = 0.2 m/sec, α decreases to a value of the order of 100-200 $W/m^2 \cdot °K$ already when the streams are delivered at a height of 50 mm from the level of the bulk bed.

For all velocities the dependence α(w) as usual is described by exponential formulas, the slope of the lines (Fig. 1b) for the pneumatic transport zone being practically independent of the velocity upon its change from 0.2 to 0.5 m/sec.

Heat transfer by particles from the rarefied to the dense phase of a fluidized bed is used when organizing two-stage gas combustion, when heating a fluidized bed by submerged burners [8], and in other energy and chemical processes.

NOTATION

$t_{c.a}$, $t_{h.a}$, t_{mix}, $t_{f.b}$, temperature of cold air, hot air, mixture of hot and cold air at outlet from the device, in the core of the fluidized bed; d, d_{av}, average diameter of particles of narrow and wide fractions; H_0, H, height of bulk bed and of hot-air nozzles above the level of the bed; w, w_s, rate of fluidization, discharge velocity of streams; W, fluidization number; α, conditional heat-transfer coefficient; Q, $Q_{c.a}$, $Q_{f.b}$, Q_l, fluxes of heat: being returned to the bed by particles from the stream supply zone, being introduced into the bed with the fluidizing agent, being carried away from the surface of the fluidized bed with the fluidizing agent (heated to the temperature of the bed core), and being lost through the insulation; F, cross-sectional area of chamber in plan; a, b, empirical coefficients.

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HEAT TRANSFER OF A CONICAL TUBE WITH INTERNAL COOLING BY A SWIRLING AIR JET

A. V. Sudarev, V. A. Maev,
and M. V. Goryacheva

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The results of an experimental investigation of local heat transfer during flow of turbulent semibounded jets in a conical tube beyond an annular swirler installed at the inlet are presented.

In calculations of complex heat transfer in furnaces, in the nozzle apparatus of turbomachines, in ducts of variable cross section, etc., the problem arises of determining the temperature level of conical elements cooled on the inside by an annular swirling air jet. The few investigations of heat transfer and aerodynamics under conditions of a swirling motion of the coolant are devoted mainly to problems of the swirling flow in nozzles and diffusers [1-6].

To obtain the necessary mathematical dependences for the local heat transfer of a swirling jet stream in a conical duct, investigations were carried out on the experimental device shown in Fig. 1a. The main element of the test bench is a conical measuring section 1 with cylindrical entry and exit pipes 2 and 3. An interchangeable vaned swirler 4 with fairing 5 is installed in the entry pipe. The air flows through the swirler along the inside surface in the form of an annular swirling jet. The design of the test bench permitted installing cones with a different apex angle 2ψ (Table 1) and swirlers with a different angle of twist φ_0 and relative area of the hub β (Table 2). The swirlers were made in the form of annular cascades with straight vanes and constant angle of twist along the radius, as was used in the designs of the front devices of gas-turbine combustion chambers. The local heat-transfer coefficients were measured by means of built-in miniature electrical alpha-calorimeters 6 which were mounted on insulation boards 7 installed in longitudinal grooves (along four generatrices) of the cone flush with its inside surface. To prevent disturbances in the wall boundary layer owing to possible roughness at the site of installing the boards and calorimeters, their inside surface was machined jointly with boring of the conical part of the measuring section. The design of the calorimeter is shown in greater detail in Fig. 1b. The calorimeter consists of a copper housing 8 made in one piece with a bottom and a copper cover 9, which form a closed unit, inside which are located a thermocouple assembly 10 (copper—Copel) soldered to the bottom and an electrical heater 11. The thermocouple assembly and the heat losses of the calorimeter were calibrated preliminarily (Fig. 1c) and showed sufficient stability for the technology of manufacture and assembly of the calorimeters that was used.

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