INVESTIGATION OF HEAT EXCHANGE IN A COUNTERCURRENT APPARATUS WITH FLUIDIZED BED OF INTERMEDIATE HEAT CARRIER

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Heat exchange between two oppositely flowing heat carriers was experimentally investigated. The obtained data agree with the results of theoretical investigation [1].

Theoretical investigation [1] showed that it is possible to construct efficient heat exchangers that can be used, e.g., for utilizing the heat of metal objects after hot plastic deformation (forging or pressing).

For verifying the theoretical model adopted in [1], an experimental investigation was carried out on a laboratory installation (Fig. 1).

The heat exchanger 4 was a box of organic glass 260 mm long, 30 mm wide, and 200 mm high. To reduce the longitudinal thermal conductivity and consequently increase the efficiency of operation of the countercurrent heat exchanger, the cross section and width of the apparatus were selected as the minimally permissible from the point of view of the organization of a satisfactory fluidization regime. The disperse material was several fractions of electrocorundum or magnesite particles with a mean size from 50 to 315 μm. The filling height in the dense bed was 20-60 mm. At a distance of 15 mm from the screen and 7 mm from each other, two nickel pipes were mounted horizontally; they were for hot and cold water and had an outside diameter of 5 mm and an inside diameter of 4.5 mm. Water with room temperature passed through the heat exchanger and then entered the electric heater, where its tempererature was raised by an additional 20-25°C, and then it returned to the heat exchanger; afterwards it flowed into the measuring vessel. The water temperature in the outer and central sections along the heat exchanger was measured by Chromel—Alumel thermocouples, and the temperature gradients between the extreme sections were measured by differential thermocouples. The correctness of measuring the temperature gradients was preliminarily checked with the aid of special experiments involving the cooling of water in the pipes with free convection in air. The temperature gradients calculated from handbook values of α in free convection and found in control experiments did not differ by more than 0.2°C, i.e., the difference was within the measuring error.

At eight points along the fluidized bed the temperature was measured by a mobile thermocouple mounted at an equal distance from the longitudinal walls. In addition to that, for checking the temperature of the bed, fixed thermocouples were mounted along the section of the heat exchanger in the central and outer sections 1 mm from the walls. The maximum difference between the readings of these thermocouples did not exceed 1°C with a longitudinal gradient of 5-6°C. The readings of the mobile thermocouple in the regime of developed fluidization corresponded to the arithmetic mean of the readings of the thermocouples spaced along the walls. The heat transfer coefficients between the water and the fluidized bed were measured by the generally known method in steady-state experiments. The effective thermal conductivity was determined in special experiments without movement of the heat carrier. The measurements were also carried out by a steady-state method analogous to [2].

The experimental results (Fig. 2) showed that for relatively large particles of 315 μm diameter, the height of the bed did not affect the heat transfer coefficient α. With decreasing particle diameter the influence of the height increased. This is apparently due to the experimentally observed worsening of the mixing and the origin of zones where the disperse material settles when the bed of small particles becomes higher. The effect of the height of the bed on its effective thermal conductivity was analogous. Establishing transverse baffles (their number in the experiments did not exceed seven) under our conditions had

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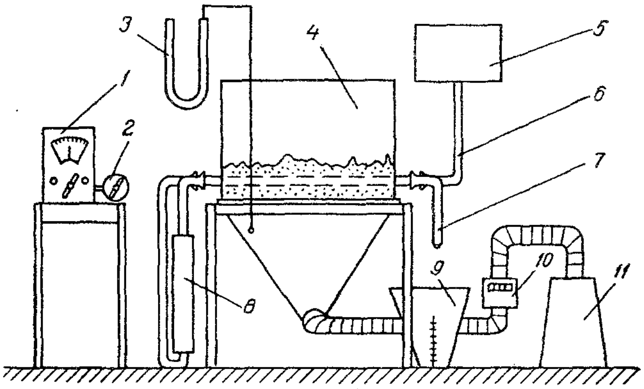

Fig. 1. Diagram of the experimental installation: 1) microvoltmeter; 2) multiple switch; 3) manometer; 4) heat exchanger with fluidized bed; 5) water supply tank; 6) cold-water feed pipe; 7) hot-water discharge pipe; 8) electric water heater; 9) measuring vessel; 10) air-discharge meter; 11) blower.

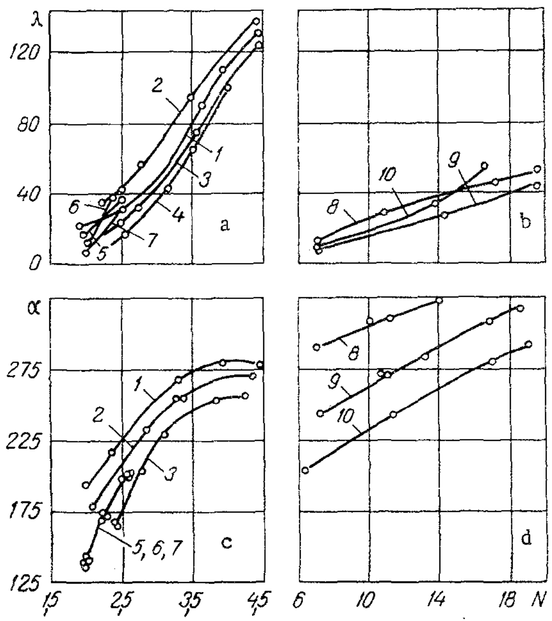

Fig. 2. Experimental values of the effective longitudinal thermal conductivity λ and heat-transfer coefficient α (a and c, magnesite): 1, 2, 3, 4) d = 180 μm, H = 20, 30, 40, 60 mm, respectively; 5, 6, 7) d = 315 μm, H = 20, 30, 40 mm, respectively (b and d, electrocorundum): d = 50 μm; 8, 9, 10) H = 20, 35, 50 mm, respectively.

practically no effect on the distribution of the temperature fields, and that means that it did not affect the magnitude of λ either. It may be assumed that with increasing dimensions of the installation, the influence of transverse baffles on the effective thermal conductivity will increase.

On the basis of the results of preliminary measurements of α, λ, G, the dimensionless parameters γ, β, δ were calculated. In calculating δ, we took into account the heat losses with the fluidizing air and through the walls. The magnitude of these losses was checked by compiling the thermal balance of the heat exchanger. The maximally possible errors in determining these values in different experiments lay within the limits: $\Delta\gamma$ = 15–34%, $\Delta\delta$ = 20–44%, $\Delta\beta$ = 53–72%. It should be pointed out that the true errors were smaller than the maximum ones. Moreover, an analysis of the results of the calculations, part of which were

TABLE 1. Comparison of Theoretical and Experimental Data

d, μm	Fluidization No. N	$\vartheta_3(0)$			$\vartheta_2(0)$			$\vartheta_2(1)$			$\vartheta_1(1)$		
		theoretical	experimental	discrepancy, %	theoretical	experimental	discrepancy, %	theoretical	experimental	discrepancy, %	theoretical	experimental	discrepancy, %
50	17	0,602	0,627	4,2	0,245	0,216	11,8	0,39	0,37	5,1	0,188	0,18	4,1
180	2,6	0,562	0,54	3,9	0,197	0,18	8,6	0,292	0,32	9,6	0,144	0,137	4,7
180	2,9	0,516	0,532	3,1	0,191	0,18	5,8	0,325	0,326	0,3	0,171	0,16	6,4
180	3,1	0,52	0,545	4,8	0,225	0,2	11,1	0,329	0,332	0,9	0,187	0,167	10,7
280	2	0,762	0,768	0,8	0,191	0,2	4,7	0,238	0,25	5,0	0,065	0,053	18

presented in [1], showed that the errors of determining the temperatures of the heat carriers at the outlet from the heat exchanger $\vartheta_1(1)$ and $\vartheta_3(0)$ are much smaller than the corresponding errors of the parameters γ, β, and δ. The temperatures were affected most by a change in the value of δ, but even in this case a deviation of $\Delta\delta = 20\%$ leads to a change in $\vartheta_1(1)$ of no more than 8%, and of $\vartheta_3(0)$ by approximately 7%.

A comparison of the experimental and theoretical data (Table 1) showed that the discrepancy between them lies within the limits of possible experimental errors; this testifies to the correctness of the adopted theoretical model.

NOTATION

c, heat capacity; d, mean particle diameter in the fluidized bed; F, area of the flow surface of heat carrier (cold or hot) in contact with the fluidized bed; G, discharge; H, height of disperse bed in the dense state; L, length of the heat exchanger; N, fluidization number; t, temperature; V, volume taken up by the fluidized bed; x, dimensionless coordinate along the heat exchanger read off from the inlet of cold heat carrier (x = 0) to the inlet of hot heat carrier (x = 1); Q, dimensionless heat losses through the heat exchanger walls; α, heat transfer coefficient between the heat carrier and the disperse medium, γ_i, β_i, δ,* dimensionless coefficients; $\vartheta_i(x)$,† dimensionless temperature; λ, effective thermal conductivity of the fluidized bed. Subscripts: i = 1, 2, 3) cold heat carrier, fluidized bed, and hot heat carrier, respectively; b) fluidizing agent.

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* $\gamma_i = \frac{\alpha_i \cdot F}{G_i \cdot c_i}$, $\beta_i = \frac{\alpha_i F L^2}{\lambda V}$, $\delta = \frac{G_b \cdot c_b \cdot L^2}{\lambda V} + Q$.

† $\vartheta_i(x) = \frac{t_i(x) - t_1(0)}{t_3(1) - t_1(0)}$.