STUDY OF THE TEMPERATURE FIELD NEAR A HOT PLATE IN A FLUIDIZED BED AND OF THE HEAT TRANSFER BETWEEN THEM

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The results are shown of temperature measurements near a hot plate in various positions in a fluidized bed.

No sufficiently general theory has been developed yet concerning the external transfer in a fluidized bed of fine-grain material, although various models simulating different aspects of the process have been proposed [1-6]. Particularly diverse are the concepts presented by various authors to explain the mechanism of heat transfer between a fluidized bed and rather large hot surfaces such as, for example, the apparatus walls. An attempt will be made here to explain the mechanism of external heat transfer in a fluidized bed on the basis of the experimentally determined temperature field near the hot object.

The tests were performed with an apparatus 170 × 420 mm in the plan section containing a 300 mm high layer of loose material. One end of the apparatus was water cooled. Air at room temperature was used as the fluidizing agent, its flow rate being measured with a twin diaphragm. The gas was distributed through a perforated grid with a 0.185% active cross section. The fluidized bed contained a single-dispersion of grade EN-12 electrocorundum with an average grain size $d = 120\ \mu$ according to GOST (Government Standards) 3647-59 (true density $\rho_M = 3900$ kg/m^3, effective density $\rho_b = 1890$ kg/m^3, thermal conductivity and specific heat of the bed material $\mu_b = 0.202$ W/m · deg and $c_b = 795$ J/kg · deg at 20°C).

Across the apparatus, on the center of a shaft, was placed a flat calorimeter (Fig. 1) which consisted of three identical elements and which partitioned the apparatus into two compartments. Each element was a 5 mm thick copper plate 1 with an electric heater 2 placed underneath, the latter made of size 0.6 mm (diameter) Nichrome wire wound on a porcelain bobbin. At the center of each copper plate was cemented on a Chromel–Copel thermocouple 6 with size 0.5 mm (diameter) electrodes. All the calorimeter components were insulated thermally on the sides and from one another by separators 6, 7 and mounted to a strip 3 on the chassis 4. All chassis details and insulating separators were made of Textolite. Thermal insulation was further ensured by having the calorimeter housing filled with asbestos wool 8. Metallic lugs 5 were attached to the housing, in order to facilitate a rigid mounting on the shaft which rotated the calorimeter housing through positive angles $\varphi > 0°$ (heat emitting surface downward against the air stream) and negative angles $\varphi < 0°$ (heat emitting surface upward). A sheet metal deflector was screwed on to the calorimeter at the bottom, to guide the air from underneath the calorimeter toward the insulated surface and, in this way, to prevent the thickness of the calorimeter from affecting the temperature field around its heat emitting surface. Each heater was energized electrically through a voltage stabilizer and an autotransformer, the current and the voltage being measured with a D-57 0-5 A ammeter of accuracy class 0.1 and a D-523 0-75 V voltmeter of accuracy class 0.5. The temperature of the copper plates was measured by the differential method with a PP-63 potentiometer of accuracy class 0.05 and the cold junction of the thermocouple immersed in the bed core.

A low-(thermal)inertia Chromel–Copel thermocouple with size 0.2 mm (diameter) electrodes was used for searching the temperature field near the calorimeter. This thermocouple was mounted horizontally on a Textolite frame, which in turn was secured in a metallic holder movable in both the vertical and the horizontal direction. The free ends of the thermocouple were connected to copper lead wires and held

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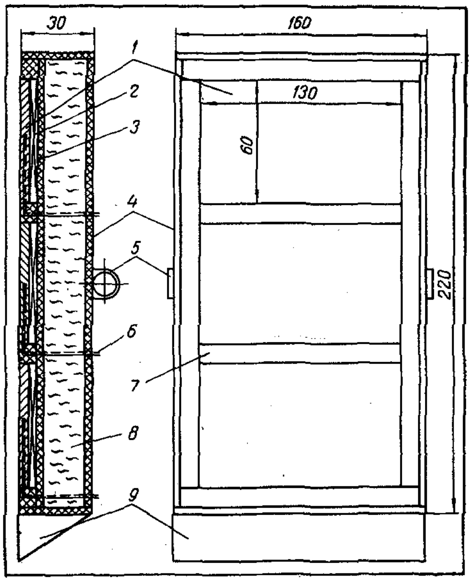

Fig. 1. Schematic diagram of the calorimeter.

secure inside the bed core, 70 mm away from the ends of the Textolite frame on which the active thermocouple junction had been mounted. In this way, the thermocouple measured the excess temperature (above the temperature of the bed core). The probe had been designed so as to allow a temperature measurement at not less than 0.5 mm from the calorimeter surface. The thermocouple signal was amplified in an F-301 dc instrument amplifier and then recorded on a model NZ26-1 single-channel high-speed automatic instrument. The ranges of the latter were 0-1 mV and 0-2.5 mV within the 2.5 and the 1.5 accuracy classes, respectively. It could record processes at up to a 40 Hz frequency on a 50 mm width chart. From the thermogram we determined the average temperature at a given point (mathematical expectation) as well as the mean absolute deviation and the frequency of temperature fluctuations, the latter estimated as one half the average number of intersections of the thermogram with the mean-temperature line at a given point per unit time. The averaging was based on 100-150 points within 0.2 and 0.5 sec, depending on the process frequency.

The tests were performed, essentially, at a fluidization rate w = 0.2 m/sec, since the temperature data for the bed near the surface measured at various fluidization rates were qualitatively not much different, even though the zone of thermal agitation was becoming wider as the fluidization rate was decreased. The effect of various factors, including the fluidization rate, on the heat transfer between inclined and vertical plates in a fluidized (vibrofluidized) bed has already been analyzed rather thoroughly in [7, 8].

The temperature field was examined with only the center heater of the calorimeter turned on in the vertical position and at attitude angles varying from +15 to −15°. At positive attitude angles φ from 5 to 15° the excess temperature at a distance of 0.5 mm from the surface decreased to 10-20% of the excess temperature at the center heater ϑ (Fig. 2). The frequency of temperature pulsations was 1.5-2.0 Hz. At 1-1.5 mm from the surface the excess temperature decreased almost to zero. The relatively high frequency of temperature pulsations at such calorimeter positions (at the vertical position it was only 0.6-0.8 Hz) is explainable by the frequent turnover of particle packets heated at the calorimeter as a result of the intensive motion of the fluidizing agent accumulating under an inclined plate. The heat-transfer coefficient was maximum at attitude angles of about 10°. The heat transfer became worse at larger angles, on account of the higher porosity under the plate.

As can be seen in Fig. 2, at φ = 10-15° the temperature field around the heat-emitting surface was, under the test conditions, almost symmetrical with respect to the plane perpendicular to this surface and to the plane of the diagram passing through its center. The temperature in the boundary layer was maximum at the center of the calorimeter and decreased toward its edges. A somewhat higher temperature near the upper edge than near the lower edge of the center element of the calorimeter was observed but essentially within the limit of test accuracy.

When φ = 5°, we note a tendency to disturb the symmetry of the temperature field. The increasing asymmetry, especially pronounced at $\varphi \leq 0°$, indicates that at these angles material is moving in one direction downward along the calorimeter surface. At the same time, the zone of thermal agitation with a considerable excess temperature becomes wider as the attitude angle is increased in the negative sense.

An analysis of the temperature fields suggests the following concepts about the mechanism of heat transfer between rather large surfaces and the surrounding fluidized bed. At small positive angles of inclination the heat from the hot surface is transmitted to packets of particles which are heating up around it and which are circulating intensively driven by the movement of gas bubbles, the latter effectively covering the inclined surface. The rate at which the packets or clusters of particles stir is so high that the temperature of the gas ascending along the surface remains almost constant and, therefore, the heat-transfer rate remains equally high along the height of the calorimeter. The same situation prevails when all three heaters are turned on. At that time, a complete turnover of packets heated up at the surface and new cold ones is not taking place, apparently, even then. The hot particles move not only deep into the bed but also along the surface, while, by virtue of the stochastic nature of such a motion, the time-average

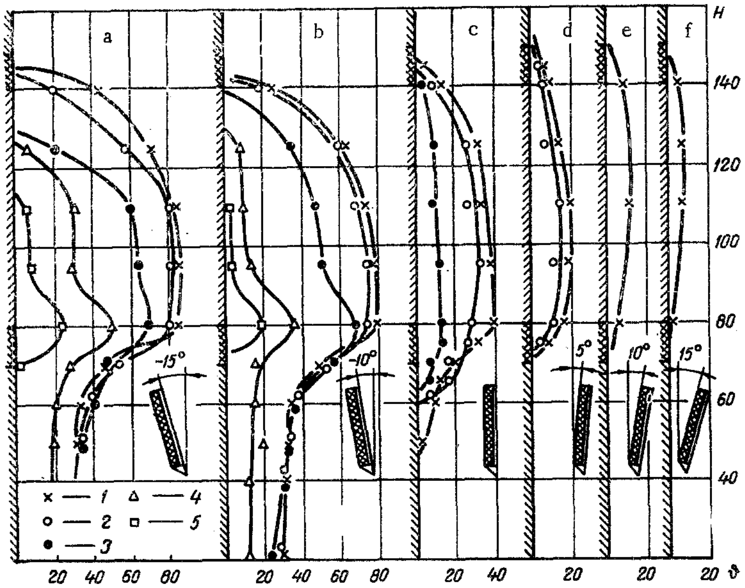

Fig. 2. Temperature field near the calorimeter, with the center heater turned on: 120 μ corundum, w = 0.2 m/sec. Axis of abscissas: excess temperature (above the temperature of the bed core), as % of the excess temperature of the center plate ϑ, °C. Axis of ordinates: distance from the lower edge of the calorimeter H, mm. Distance from the calorimeter surface: 1) 0.5 ; 2) 1.5; 3) 2.5; 4) 3.5; 5) 4.5 mm. Attitude angle (from the vertical): a) 15; b) 10; c) 0; d) 5; e) 10; f) 15°.

temperature at the calorimeter edges is lower than at the center. This temperature difference should become greater, of course, as the height of the calorimeter is increased. For this reason, even under such favorable conditions, the heat-transfer coefficient should decrease as the height of the heat-emitting surface is increased. This agrees with the test data (Table 1) [9].

When $\varphi \leq 0°$, then at the heat-emitting surface, if it is sufficiently large, there appears a unique kind of boundary layer. Particles are not any more thrown off the surface by the stream of gas bubbles, but simply move along it while intermixing in the process. On this random motion is superposed a regular downward motion due to forces of gravity, since fine grains at the surface are shielded from the air stream and, therefore, the material becomes less porous here than it is in the bed core. As the attitude angle is increased, the motion of particles is less intensive in the direction perpendicular to the heat-emitting surface and, consequently, the effective thermal conductivity of the fluidized bed decreases at the surface. At the same time, the velocity of bed material descending along the surface decreases. All this results in a thicker boundary layer and a lower heat-transfer rate (Fig. 3). Naturally, the vertical height of the heat emitting should be more influential at such angles than at $\varphi > 0°$ (see Table 1).

The concepts suggested here have also been proven valid by the temperature behavior in the case of two unheated plates placed, respectively, above and below the heated one (see Table 1). When $\varphi = -5$ to $-15°$, the descending particles which had been heated up by the center plate did in turn heat up the lower plate.) For this reason, its temperature was much higher than the temperature of the upper plate, the latter having been heated up only by particles approaching it infrequently (as a result of pulsations) and by heat reaching it through the insulation. As hot particles flowed around the lower cold plate, the temperature distribution across the boundary layer became extremal, since, upon contact with the cold plate, heat was carried away from the layer of descending particles not only to the bed core but also toward that plate. This can be verified in Fig. 2a, b, c.

When $\varphi > 0°$, on the contrary, the ascending stream of gas and particles caused the temperature of the upper plate to rise slightly above the temperature of the lower plate, but this effect was almost unappreciable. When $\varphi < -15°$, along the surface descended an almost solid layer of fine grains thicker than the thermal boundary layer. One may assume, to the first approximation, that the thermal conductivity of

TABLE 1. Test Results Pertaining to the Thermal Interaction between Calorimeter Components

Conditions	Temperature drop between bed and plate at w = 0.2 m/sec	Angle of calorimeter inclination from vertical 15°	10°	5°	0°	−5°	−10°	−15°
Center element of the calorimeter turned on	Upper plate ϑ', °C	2	2	1,5	1,5	1,5	1,5	1,5
	Center plate ϑ, °C	37	35,5	36	39	45	48,5	51,5
	Lower plate ϑ'', °C	0,5	0,5	1	4	14	18	20
	Heat transfer coefficient α, $W/m^2 \cdot °C$	485	505	497	460	358	306	267
All three calorimeter elements turned on	Heat transfer coefficient f for the center element α, $W/m^2 \cdot °C$	448	459	453	385	268	235	189

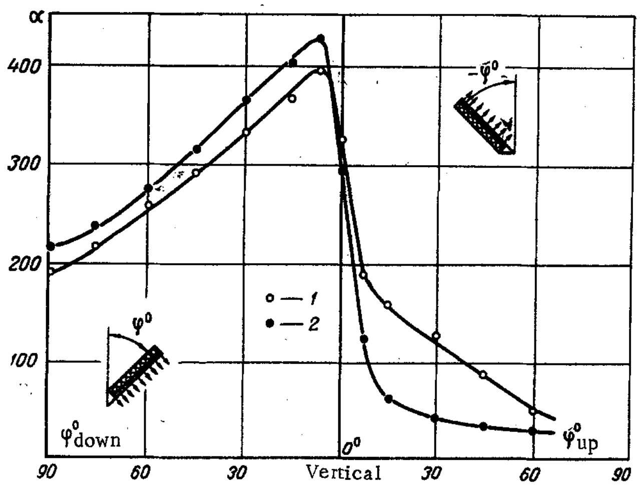

Fig. 3. Mean-over-the-height heat-transfer coefficient as a function of the angle by which the calorimeter is inclined from the vertical: size 120 μ corundum, w = 0.15 m/sec. Axis of abscissas: inclination angle of the calorimeter from the vertical φ^0. Axis of ordinates: heat-transfer coefficient. 1) Data based on the standard method; 2) data based on the regular mode of the first kind; α, $W/m^2 \cdot deg$.

the thermal boundary layer would be the same as that of the fine-grain layer and, neglecting any heat leakage along the thermal boundary layer on account of its thinness, one can determine the coefficient of heat transfer from the surface at $\varphi = -15$ to $-60°$ (at $\varphi < -60°$ the boundary layer may already turn into a stationary cap of fine grains), treating this as the problem of a semiinfinite mass with the thermophysical properties of a stationary layer heated by a plate at a constant temperature [10]:

$$\alpha_y = \sqrt{\frac{\lambda_H c_H \rho_H}{\pi \tau}} = \sqrt{\frac{\lambda_H c_H \rho_H V}{\pi y}}.$$

The contact resistance between the plate and the first row of particles pressed against it may in this case be disregarded, since the time of contact between the boundary layer and the surface is sufficiently long.

It must be emphasized that under these conditions the coefficient of heat transfer from the calorimeter at a constant temperature, should be higher than the coefficient of heat transfer from an object cooled (or heated) in the fluidized bed. In order to prove this, let us consider the cooling of an infinitely large plate in a medium at a constant temperature under boundary conditions of the fourth kind. If the origin of coordinates is placed at the center of the plate, then the temperature distributions in the plate (subscript 1) and in the medium (subscript 2) are determined from the following equations:

$$\theta_1 = \frac{T_1(x, \tau) - T_c}{T_0 - T_c} = 1 - \frac{1}{1 + k_\varepsilon} \sum_{n=1}^{\infty} (-h)^{n-1} \left\{ \operatorname{erfc} \frac{(2n-1)R - x}{2\sqrt{a_1 \tau}} + \operatorname{erfc} \frac{(2n-1)R + x}{2\sqrt{a_1 \tau}} \right\};$$

$$\theta_2 = \frac{T_2(x, \tau) - T_c}{T_0 - T_c} = \frac{k_\varepsilon}{1+k_\varepsilon} \operatorname{erfc} \frac{x-R}{2\sqrt{a_2\tau}} - \frac{k_\varepsilon(1+h)}{1+k_\varepsilon} \sum_{n=1}^{\infty} (-h)^{n-1} \operatorname{erfc} \frac{x - R + 2nR\sqrt{\frac{a_2}{a_1}}}{2\sqrt{a_2\tau}}.$$

At small values of τ one may omit all but the first terms in the series. For a thermally thin body with the same temperature across its thickness, one may let $R \to 0$ and a_1 be rather large. The condition that $R/\sqrt{a_1\tau} \to 0$ holds then at any instant of time $\tau > 0$ and, therefore, $\exp(-R^2/a_1\tau) \to 1$ as well as $\operatorname{erfc}(R/\sqrt{a_1\tau}) \to 1$, the expression for the instantaneous coefficient of heat transfer from the cooled plate α' becoming

$$\alpha' = -\frac{\lambda_2}{\theta_1(R, \tau)} \left(\frac{\partial \theta_2}{\partial x}\right)_{x=R} \approx \frac{k_\varepsilon}{1+k_\varepsilon} \sqrt{\frac{\lambda_2 c_2 \rho_2}{\pi\tau}} = K\alpha.$$

Coefficient $K > 1$ for any positive $k_\varepsilon < \infty$. This result has been checked experimentally: the rate of heat transfer from the calorimeter at a constant temperature with $\varphi < 0°$ was, indeed, higher than from a cooled α-calorimeter of the same dimensions whose heat-transfer coefficient had been determined in the regular mode of the first kind [8] (Fig. 3). The slight differences between the results at $\varphi > 0°$ remain within the limits of experimental error. Therefore, when calculating the rate of cooling or heating an object with large inclined portions in a fluidized bed, one must consider that, with all other conditions the same, the rate of heat transfer between the bed and the object is lower than between the bed and a calorimeter at a constant temperature. The heat transfer is considerably reduced, furthermore, by any asperities and unevenness along the inclined surface which impede the downward motion of the boundary layer. Thus, at $\varphi = -30°$, a size 0.2 mm (diameter) wire placed horizontally 1 mm away from the surface at the center of the calorimeter reduced the coefficient of heat transfer from the center element to one half. At angles from the vertical smaller than 15° the presence of a temperature measuring probe had no effect on the heat transfer, i.e., had almost no effect on the hydrodynamic conditions around the plate in the fluidized bed.

NOTATION

ρ	is the density;
λ	is the thermal conductivity;
c	is the specific heat;
y	is the distance from the upper edge of the heat-emitting surface;
τ	is the time through which material remains at the surface;
V	is the velocity of the descending boundary layer;
α_y	is the local heat-transfer coefficient;
θ	is the dimensionless temperature;
T_c	is the temperature of medium far from the plate;
T_0	is the initial temperature of plate;
R	is the half-thickness of plate;
$k_\varepsilon = \sqrt{\lambda_1 c_1 \rho_1 / \lambda_2 c_2 \rho_2}$	is the critical number characterizing the thermal activity of a plate;
$h = (1 - k_\varepsilon)/(1 + k_\varepsilon)$	is the dimensionless quantity;
x	is the space coordinate in the direction perpendicular to a plate;
a	is the thermal diffusivity;
α	is the instantaneous heat-transfer coefficient;
φ^0	is the inclination angle of the calorimeter from the vertical.

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