AN EXPERIMENTAL STUDY OF HEAT TRANSFER TO A STREAM OF GASEOUS SUSPENSION DURING PHYSICOCHEMICAL CHANGES

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An experimental apparatus is described and test results are shown pertaining to the heat transfer to a gaseous stream with a suspended chemically inert or reacting solid phase.

There are data available in the technical literature pertaining to the heat transfer to a stream of a chemically inert gaseous suspension and, as is well known, the formulas derived by various authors on the basis of their test data differ [1]. This disagreement between the test data obtained by various authors can be explained by a lack of precise and reliable concepts concerning the mechanism of heat transfer, with a consequent incomplete consideration of all the factors affecting the process.

Thermal processes in industrial applications with gaseous suspensions are often accompanied by chemical changes. The particles of finely comminuted solid materials decompose then when heated and the mass of the solid phase decreases, while the composition and the amount of the gaseous phase also change correspondingly. The heat effect of a chemical decomposition reaction and the gas generation in the solid phase both cause a redistribution of mass and energy in the stream, thus additionally affecting the net heat transfer.

In this article the authors present the results of an experimental study concerning the heat transfer to a gaseous suspension with an inert or a decomposing solid phase flowing through a circular horizontal pipe. As an inert material we used white marble, as a reacting material we used sodium bicarbonate (baking soda). These materials have similar densities, $\rho = 2653$ kg/m^3 and 2200 kg/m^3 respectively. The tests were performed with a single narrow-band size fraction of particles: 0.11-0.22 mm. Air served as the carrier medium.

The tests were performed with an apparatus shown schematically in Fig. 1.

Solid particles were fed steadily from bin 1 through an orifice in an interchangeable stopper into the air stream and were then accelerated to full speed within the 0.8 m long ($x/d = 61.5$) pipe 2 serving as the stabilization zone and made partly of glass for visual tracking of the stream.

The basic component of the measuring equipment was a grade 1Kh18N9T stainless steel pipe 3, 16/13 mm in diameter and 3 m long. This pipe was heated segmentally in model SUOL-0,15.2/12 MR electric ovens. Each of the three segments was 1 m long. The ovens were energized from a model RNO-250-10 voltage regulator through a model LATR autotransformer, making it possible to adjust the temperature distribution in the pipe wall along each segment and the entire test zone. The heat transfer was studied at a constant wall temperature. In order to account for heat leakage into the ambient medium, we had a water jacket 5 wrapped around the test pipe segments. The water inlet and outlet temperatures were measured with mercury thermometers, accurately within 0.1°C, while the water flow rate was measured with a rotameter 8. The water jacket was covered on top with a layer of thermal insulation, which almost completely eliminated any stray heat losses.

The air flow rate during this experiment was measured with a Pitot tube 18. The feed rate of solid material was determined from the loss of weight of bin 1 while the charge was ejected into the carrier stream. Chromel–Alumel thermocouples were installed at the centers of the respective pipe sections,

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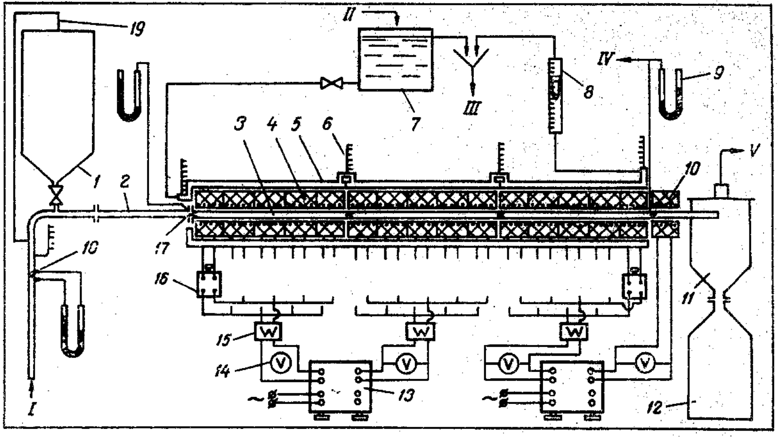

Fig. 1. Schematic diagram of the test apparatus: I) air from a fan developing a high pressure head; II) water from a water conduit; III) water drain; IV) gas sampling for analysis; V) air exhaust into the atmosphere; 1) discharge bin; 2) glass pipe; 3) test zone; 4) electric oven; 5) water jacket; 6) mercury thermometer; 7) pressure tank; 8) rotameter; 9) U-tube manometer; 10) standby electric oven; 11) cyclone chamber; 12) receptacle; 13) pressure regulator; 14) voltmeter; 15) wattmeter; 16) autotransformer; 17) thermocouple for measuring the stream temperature; 18) Pitot tube; 19) connecting tube.

for measuring the temperature of the stream at the entrace to and the exit from the test zone as well as between the pipe segments, with the thermocouple junctions shielded against radiation from the wall by cylindrical stainless steel tubes coaxially inside the pipe 5.5 mm in diameter and 12 mm long each. The temperature of the pipe wall was measured with Chromel–Alumel thermocouples spaced at fifteen points 0.2 m apart along the a lateral generatrix, beginning at 0.1 m from the entrance to the test zone. These junctions were pressed with special grommets into 0.6 mm deep holes which had been drilled into the pipe wall.

The emf of the thermocouples was measured with a model PP-63 potentiometer. The power to the segmental ovens was measured with model D539/1 class 0.5 wattmeters. The maximum power actually supplied to the test apparatus throughout the experiment was 5.5 kW. The power losses in the autotransformer were determined beforehand under idle conditions without air flow through the pipe.

Measurements were made under steady thermal and dynamic conditions.

Experiment with an Inert Solid Phase. The wall temperature in the hot zone was varied from 83 to 670°C, with the difference between maximum and minimum temperature along the entire pipe not exceeding 40°C during any one test. The temperature of the air stream at the entrance to the test zone was varied from 15 to 42°C. The Reynolds number of the gas stream ($\mathrm{Re}_{0,\mathrm{d}} = Gd/\mu_0$), referred to entrance conditions, varied from 4500 to 16,300 and this corresponded to an air velocity from 5.25 to 18.7 m/sec. Measurements were made at discharge concentrations ranging from 0 to 8.87 kg of solid per 1 kg of air. Tests with the gaseous suspension were preceded by tests with pure air. After a steady state had been reached, we measured all quantities necessary for calculating the heat transfer coefficient in pure air, whereupon solid particles were fed into the stream. It could be seen through the glass wall along the stabilization zone that, at low air velocities the concentration of solid particles was somewhat higher in the bottom part than in the top part of the pipe. Over the entire range of air velocities, however, the transport process appeared stable and no sedimentation of particles on the bottom of the pipe occurred. Special measurements indicated almost the same temperature around the pipe circumference, and this agreed with the data in [2].

The mean heat transfer coefficient was defined in two ways:

$$\alpha = \frac{Q}{F(t_x - t_0)} \ln \frac{t_\mathrm{W} - t_0}{t_\mathrm{W} - t_x} \tag{1}$$

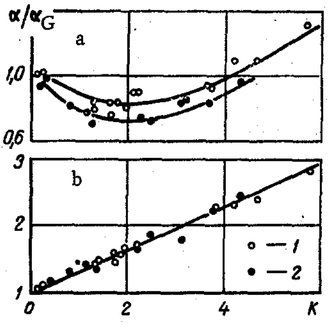

Fig. 2

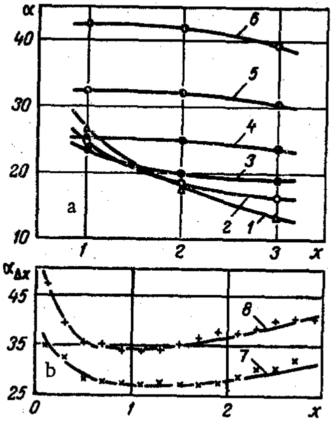

Fig. 3

Fig. 2. Relative heat transfer rate as a function of the discharge concentration, at x = 3 m and $t_W = 300$°C: (a) heat transfer coefficient according to formula (1), (b) heat transfer coefficient according to formula (2); 1) $\mathrm{Re}_{0,d} = 6300$; 2) $\mathrm{Re}_{0,d} = 12{,}000$.

Fig. 3. Heat transfer coefficients ($\mathrm{W/m^2 \cdot deg}$) as functions of the pipe length x (m): (a) mean heat transfer coefficient α and (b) local heat transfer coefficient $\alpha_{\Delta x}$, for 1) K = 0 (pure air); 2) K = 0.384; 3) K = 1.1; 4) K = 2.11; 5) K = 3.7; 6) K = 5.75; in (b) 7) $\mathrm{Re}_{0,d} = 7360$ and K = 1.97 and $t_W = 335$°C; 8) $\mathrm{Re}_{0,d} = 8900$ and K = 3.8 and $t_W = 385$°C.

and

$$\alpha = \frac{Q}{F\,(t_{\mathrm{W}} - t_0)}, \tag{2}$$

with the thermal flux Q along the test zone from 0 to x (x = 1, 2, 3 m) equal to the difference between the electric power supplied for heating and the heat carried away by the cooling water in the jacket through the distance x, and with F = πdx denoting the inside surface area of the pipe.

The temperature of the air stream t_x at section x was calculated according to the same formula as in [3]:

$$t_x = \frac{Q + t_0\,(Gc_0 + G_{\mathrm{S}}c_{\mathrm{S},0})}{Gc_x + G_{\mathrm{S}}\,c_{\mathrm{S},x}}. \tag{3}$$

In the tests with pure air (K = 0) the temperature of the stream according to formula (3) was generally 1-3°C higher than indicated by the thermocouples, evidently because the latter read local values within a given pipe section. In the tests with a gaseous suspension, on the other hand, the calculated temperature was lower than measured, the difference reaching 40°C at sections behind the first and the second one. At the last section of the test zone both the calculated and the measured values came closer together. Such a divergence of data indicated an appreciable temperature unbalance between the components of the stream.

The data pertaining to pure air throughout the test range of parameter variation fitted, within 3%, the well known formula

$$\mathrm{Nu}_{G,d} = 0.018\,\mathrm{Re}_{G,d}^{0.8}\,(T_G/T_{\mathrm{W}})^{0.5}. \tag{4}$$

at section x = 1 m (x/d = 77). At sections x = 2 m (x/d = 154) and x = 3 m (x/d = 231) the actual heat transfer rate was 12-20% lower than calculated according to this formula, which could be explained by a laminarization of the stream during heating [4].

The effect of the discharge concentration on the heat transfer rate at section x = 3 m is shown in Fig. 2 for various values of the Reynolds number and at approximately the same 300°C wall temperature. It is quite evident here that this effect appears different, depending on the method by which the heat transfer coefficient is determined. The effect of the Reynolds number (i.e., the effect of the air velocity) is shown in Fig. 2a and is seen here to degenerate into no effect when the Reynolds number becomes high: no distinct segregation of test points is noted at $Re_{0,d} > 12{,}000$. These results agree with the results in [3], where the relative heat transfer rate also did not depend on the Reynolds number within the range of turbulent air flow. With the heat transfer coefficient determined according to formula (2), the relation $\alpha / \alpha_G = f(K)$ was linear over the entire range of concentrations (Fig. 2b) and no segregation of test points with respect to the Reynolds number took place.

Obviously, then, formula (2) is preferable to formula (1). The temperatures of the wall and the stream at the pipe entrance in formula (2) are always known, while formula (1) is based on temperature t_x according to formula (3) and t_x does not quite characterize the complex process of heat transfer in a gaseous suspension where the temperature unbalance between the phases is appreciable, so that formula (1) is hardly valid here.

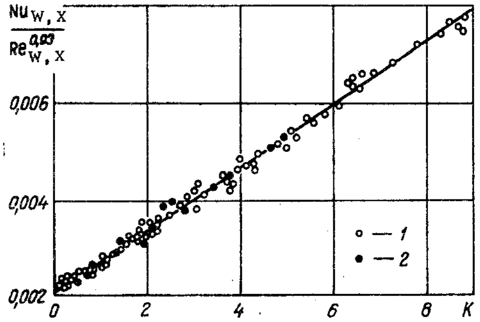

Fig. 4. Test data on the heat **transfer** to a chemically inert and to a chemically reacting stream: 1) inert stream; 2) stream with decomposing solid particles.

Typical variations of the heat transfer coefficient along the pipe, according to formula (2) are shown in Fig. 3 for various concentrations at a Reynolds number $Re_{0,d} = 8100$ and a wall temperature $t_w = 300°C$. It is quite evident here that, depending on the pipe length, at low concentrations the rate of heat transfer here can be higher or lower than to a pure air stream. In all the tests with a gaseous suspension the relative heat transfer rate increased with the pipe length.

In order to evaluate the variation in the local heat transfer along the pipe, from some of the test data we calculated the local heat transfer coefficient for segments $\Delta x = 0.2$ m long according to the formula

$$\alpha_{\Delta x} = \left| \frac{q}{t_w - t_x} \right|_{\Delta x}.$$

Here the local thermal flux density q and the temperature difference $(t_w - t_x)$ were determined from temperature curves which had been plotted on the basis of test data. The results of such calculations for two test runs are shown in Fig. 3b. It is quite evident that behind the thermal starting zone $(x/d > 70)$ $\alpha_{\Delta x}$ increased with the pipe length.

This trend of the heat transfer rate according to the test results could be explained as follows. Within the first pipe segment there existed two zones: a thermal boundary layer with a higher air viscosity at the wall and a relatively cold mainstream. According to our analysis, solid particles can be expelled from the boundary layer, where the viscosity is higher, into the mainstream region of a much higher velocity. The mainstream becomes then less turbulent and more stable, which results in a longer thermal starting zone. As a consequence, the heat transfer rate is now low and, at low concentrations of particles, was lower at x = 1 m than in the case of a pure air stream (Fig. 3a). Behind the thermal starting zone the carrier stream becomes more and more homogeneous and the solid particles from the mainstream gradually distribute themselves over the entire pipe section. They penetrate into the viscous sublayer at the wall, decreasing its thickness and its specific heat referred to volume, and participate in the radial heat transfer. At higher air temperatures the stream velocity around the particles and the rate of internal heat transfer in the stream both increase too, bring the temperatures of both components closer together. All these factors cause the heat transfer rate to increase along the pipe.

At higher concentrations the flow of solid particles in the mainstream becomes more constrained and, therefore, the particles penetrate fast into the region of higher air viscosity. As a result, the thermal resistance is lower now and the heat transfer rate increases with increasing concentration. At K = 1.3 the heat transfer rate at x = 1 m in our tests was minimum. Evidently, this concentration level was the critical one under our test conditions and at concentrations higher than that particles began to be expelled from the "oversaturated" mainstream.

The generalization of our test data for section x = 3 m is shown in Fig. 4. As the graph indicates, the test points fit about a straight line which represents the equation

$$Nu_{w,x} = 0.00205\ Re_{w,x}^{0.93}(1 + 0.32\,K). \tag{5}$$

The heat transfer coefficient for (5) is determined according to formula (2). The maximum error of Eq. (5) is 4%.

Experiment with a Decomposing Solid Phase. The test procedure with sodium bicarbonate was the same as with marble. The operating parameters were varied over the following ranges: $Re_{0,d} = 4500$-11,200, $t_W = 380$-635°C, K = 0.5-4.9 kg/kg, and $t_0 = 17$-23°C.

The decomposition (calcination) usually begins at 160°C and is an endothermal reaction. Its equation is

$$2NaHCO_3(\text{solid}) = Na_2CO_3\,(\text{solid}) + CO_2\,(\text{gas}) + H_2O\,(\text{gas}) - 129\ \text{kJ}. \tag{6}$$

In our tests the decomposition began along the first pipe segment and proceeded to the end of the pipe. After readings had been taken at the exit section of the test zone, the gas was sampled for analysis. On a model GKhP-3M gas analyzer we determined the CO_2 content in these samples. On the basis of this analysis and stoichiometric ratios in Eq. (6), we then calculated the quantitative composition of the reaction products and the accompanying heat effect.

In single tests the decomposition reaction was ran to 90% completion, and the weight increment of the gaseous phase ran to 146%. The heat effect of the reaction was 15-51% of the heat supplied to the original gaseous suspension.

The mean heat transfer coefficient at section x = 3 m was determined according to the formula

$$\alpha = c_{s,w} \frac{Q}{F\,(i''_{s,w} - i'_{s,0})}, \tag{7}$$

with the enthalphy $i''_{s,w}$ including the heat effect of the reaction, in accordance with the formula $i''_{s,w} = c''_{s,w} t_w + Q_{chem}/(G + G_S)$, and with the enthalpy $i'_{s,0} = c'_{s,0} t_0$. The specific heat of sodium bicarbonate and of calcinated soda was assumed independent of the temperature.

From these test results, as from the earlier tests with marble, we determined the Nusselt and the Reynolds number in terms of the gas parameters. An evaluation of these data indicated that it did not matter whether the specific heat $c_{s,w}$ of the stream components in formula (7) referred to the pipe inlet or outlet, as long as the thermal conductivity and the viscosity of the gas in these similarity criteria referred to the same condition. For practical reasons, it is preferable to determine $c_{s,w}$, λ_s, and u_w under inlet conditions, since otherwise certain difficulties may arise in calculating the thermal conductivity and the viscosity of a gas mixture.

An evaluation of the test results is shown in Fig. 4. Evidently, the test points fit closely about the straight line (5). The value which Eq. (5) yields for the heat transfer coefficient is off by not more than 6%.

An analysis of our test data has shown that Eq. (5) remains valid even with the gas flow rate for the Reynolds number taken as the average between entrance rate and the exit rate, and with the concentration of the solid phase defined as the ratio of its average flow rate and that of the gas. The values of $c_{s,w}$, λ_w, and μ_w were also calculated on the average basis. Thus, according to our method of test data evaluation, a redistribution of phases along the stream does not affect the heat transfer, since a drop in the concentration of the solid phase is compensated by a rise in the gas velocity and an increase in total turbulence of the stream due to the gas generated during the decomposition of solid particles. It may be assumed that the trends which the heat transfer follows while the solid particles decompose are, according to our measurements, independent of the mode and the rate of the reaction. The occurrence of secondary reactions in the gaseous phase can in many cases [5] have an additional effect on the resulting heat transfer.

Our study has shown that the heat transfer to a chemically inert gaseous suspension or to one where a chemical reaction occurs can be described by the same relation, the only difference being that the heat transfer coefficient is defined according to formula (1) in the first case and according to formula (2) in the second case.

NOTATION

G	is the mass flow rate of gas;
G_S	is the mass flow rate of solid particles;
α	is the heat transfer coefficient;
d	is the inside diameter of the pipe;
x	is the length coordinate;
λ	is the thermal conductivity of the gas;
μ	is the viscosity of the gas;

t_0	is the stream temperature at the pipe entrance;
t_x	is the stream temperature at section x along the pipe;
$T_w(t_w)$	is the wall temperature;
T_G	is the mean temperature of pure gas;
c	is the specific heat of the gas;
c_S	is the specific heat of the solid phase;
c_s	is the overall specific heat of the stream;
K	is the discharge concentration;
Q_{chem}	is the power of the heat effect in the reaction;
i	is the enthalpy.

Subscripts

w	denotes to the value of a physical property at wall temperature;
0	denotes the value of a flow parameter at the entrance temperature;
G	denotes pure gas;
d	denotes the characteristic dimension (pipe diameter);
x	denotes the characteristic dimension (pipe length);
s	denotes the value of a stream parameter calculated by the rule of additivity as applied to stream components (phases);
'	denotes flow conditions at the pipe entrance;
"	denotes flow conditions at the pipe exit.

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