LITERATURE CITED

1. R. Reed and T. Sherwood, Properties of Gases and Liquids [Russian translation], Khimiya, Leningrad (1971).
2. L. P. Filippov, Investigation of the Thermal Conductivity of Liquids [in Russian], Moscow State University, Moscow (1970).
3. M. P. Vukalovich, S. L. Rivkin, and A. A. Aleksandrov, Tables of Thermophysical Properties of Water and Steam [in Russian], Standartov, Moscow (1969).
4. A. M. Mamedov, "Relationship between a set of thermophysical properties and the compressibility of liquids," Inzh.-Fiz. Zh., 33, No. 1, 91-95 (1977).
5. A. M. Mamedov, "Coefficient of thermal activity of liquids with water as an example," Inzh.-Fiz. Zh., 27, No. 4, 672-678 (1974).
6. A. A. Aleksandrov and D. K. Larin, "Empirical determination of ultrasonic velocity in water within a wide range of temperatures and pressures," Teploenergetika, No. 2, 75-78 (1976).
7. T. S. Akhundov and N. É. Gasanova, "Empirical study of the thermal conductivity of toluene," Neft' Gaz, No. 7, 59-63 (1969).
8. A. M. Mamedov, "On the isobaric heat capacity of liquids," Neft' Gaz, No. 7, 51-55 (1975).
9. A. M. Mamdeov, T. S. Akhundov, and F. G. Abdullaev, "Thermal and calorific properties of aromatic hydrocarbons," Inzh.-Fiz. Zh., 30, No. 4, 705-711 (1976).
10. V. V. Zotov and Yu. A. Neruchev, "Approximate calculation of the thermodynamic properties of toluene and m-xylene on the saturation line," in: Ultrasound and the Physicochemical Properties of Substances, Vol. 3, Kursk (1969), pp. 42-46.
11. Yu. S. Shoitov and N. F. Otpushchennikov, "On the pressure dependence of the speed of sound in liquids," in: Ultrasound and the Physicochemical Properties of Substances, Vol. 4, Kursk (1970), pp. 50-55.

INVESTIGATION OF HEAT EXCHANGE IN THE WATER-SPRAY COOLING OF HIGH-TEMPERATURE METAL SURFACES

L. I. Urbanovich, V. A. Goryainov, V. V. Sevost'yanov, Yu. G. Boev, V. M. Niskovskikh, A. V. Grachev, A. V. Sevost'yanov, and V. S. Gur'ev

UDC 536.242

A method is described and certain results presented of an experimental study of local heat exchange in the forced cooling of metal surfaces heated to high temperatures.

The practice of cooling heated metal surfaces with water sprayed from nozzles is common in several sectors of industry, particularly metallurgy [1].

Analysis of the literature shows that most of the well-known studies have been devoted to the cooling of heated surfaces by single drops [2-4], and little attention has been given to the process of heat exchange occurring when a system of drops strikes a heated surface [1].

The rate of heat exchange in the field of action of a jet on a heated metal surface is generally determined by the method in [5]. The essence of this method is as follows. A steel specimen in the form of a plate [5] or disk [6] of known dimensions containing thermocouples, the junctions of which are located a precisely measured distance from the front surface (the holes for the thermocouples are drilled and the latter inserted through the back surface), is placed in a furnace with an inert atmosphere to be heated to 900-1100°C. The specimen is then transferred to the working chamber, and water is delivered to the front surface at a known rate of flow from a nozzle. The temperature of internal points of the metal is measured as a function of time as the specimen is cooled. Numerical solution of the problem of nonsteady heat conduction using the temperature values obtained for characteristic points in the metal from the experiment make it possible to determine the temperature field in the specimen at any moment of time and the distribution of the heat flux on the sprayed (front) surface. The calculated heat flux and known temperature of the surface are then used to find the heat-transfer coefficient at each moment of time. The foregoing method has the following shortcomings: 1) the laboriousness of the

Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 39, No. 2, pp. 315-322, August, 1980. Original article submitted June 28, 1979.

0022-0841/80/3902-0900 $07.50

experiment, connected with preliminary heating of the test specimen in a furnace and its subsequent transfer to a chamber for forced cooling; 2) the fact that the measurements are made in a nonsteady mode, which leads to errors in the readings of the thermocouples due to the inertia of the latter (this error was not evaluated in [5, 6]); 3) the laboriousness of determining the rate of local heat exchange in the field of action of a jet on a hot surface; 4) the difficulty of reproducing the results.

The authors of [5] investigated the spraying of a surface by fully conical nozzles producing jets in the form of circular cones. The rate of heat exchange under the jet was characterized by the average heat-transfer coefficient for the area sprayed. However, according to [6-8], the density of the spraying (i.e., the volume of water delivered to a unit of the sprayed surface per unit of time) is distributed extremely unevenly over the field of action of the jet, which also means that the cooling rate is unevenly distributed. Consequently, besides the mean values of heat-transfer coefficient, it is also important to find the distribution of local values in the field of action of the jet. The authors of [6] found such local values, but not for the entire field of the spray – only for its central part. It should be noted, however, that it is practically impossible to study flat-spray nozzles by the method used in [5, 6].

Despite the limited applicability of the above method, important results were obtained in [5, 6]: the authors established parameters affecting the rate of heat exchange in the spraying of a highly heated metal surface by a jet of water from a nozzle. These parameters are the spraying density (mentioned earlier), the pressure of the water in front of the nozzle, the size of the water drops, the velocity of the latter, the spraying angle, the temperature of the surface, and the (temperature-dependent) wettability of the surface. It was also established in [6] that the velocity of the drops as they approach the surface has considerably less effect on heat transfer than was previously thought. This conclusion was based on a study of heat exchange in which single drops struck the surface being cooled [9]. Of much greater importance in heat transfer is the number of drops striking the surface per unit of time, i.e., the spraying density.

A more accurate method than that used in [5, 6] was developed by the authors of [10] and is based on measurements made in the steady mode. However, this method does not permit determination of the effect on heat exchange of the angle of attack of the surface by the nozzle jet in the vertical and horizontal planes or the angles of orientation (relative to the horizontal) of the specimen and nozzle in space: all of the experiments in [10] were conducted only for a right angle of attack and vertical location of the specimen and, as in [5-8], the region of water pressures (in front of the nozzle) greater than 1 MPa and region of temperatures (of the sprayed surface) below 700°C were not investigated. Also, the accuracy of the experiments was not evaluated and no analytical relations were derived for calculating heat exchange in nozzle cooling.

Based on the above brief analysis of prior investigations and methods, we can make important conclusions which will be helpful in developing an optimum method for studying heat exchange in nozzle cooling and formulating problems for future investigation: 1) the hydrodynamics of nozzle spraying have a substantial effect on heat exchange in the cooling of heated metal surfaces; heat transfer from the jet must be allied (and studied) with hydrodynamics; 2) the hydrodynamics of nozzle spraying depends on the design of the nozzle and the following parameters: the distance from the nozzle to the surface, the water pressure in front of the nozzle, the angle of attack of the jet relative to the surface being cooled and the angle of orientation of the surface in space, which affect the distribution of spraying density in the field of action of the jet on the surface; 3) heat transfer in nozzle cooling depends on the temperature of the surface being cooled and on the factors noted in part 2); 4) the experimental stand used for studying nozzle cooling should be universal in function: it should permit parallel studies of the hydrodynamics and heat exchange of the given process within a broad range of variation in the temperature of the surface (to 1100-1200°C), water pressure (up to 2.0 MPa), and other parameters noted in part 2).

In accordance with these recommendations, in the present work we developed an experimental method for the comprehensive study of hydrodynamics and heat exchange in the nozzle cooling of highly heated metal surfaces. The method is to be used to conduct tests on a specially designed stand, a basic diagram of which is shown in Fig. 1. The test specimen 2 is made of a Nichrome alloy in the form of a plate 30 × 10 × 3 mm. The specimen is heated in the working chamber by an alternating current of several hundred amperes supplied to the specimen through step-down transformer 5. The temperature to which the specimen is heated is regulated by changing the voltage in the primary winding of the transformer by means of voltage regulator 6. Water from nozzle 1 is directed onto the front surface of the specimen. In contrast to the method described in [10], only one surface of the specimen is sprayed. Platinum-rhodium–platinum and Chromel–Alumel thermocouples 10 are inserted through the back surface of the plate. The thermocouple junctions are located 0.8-1.0 mm from the surface being sprayed. The Chromel–Alumel thermocouple is connected to an MR-64 regulating millivoltmeter with a scale of 0-1300°C, while the platinum-rhodium–platinum thermocouple is connected to a KSP-4 automatic potentiometer (scale 0-1600°C). Apart from the temperature of the surface, measurements are made of the water pressure (by manometer 7), the water temperature (by thermometer 11), the voltage on the working section of the specimen (by voltmeter 8), and current (by calibrated shunt 4 and ammeter 9). The pumps 3, which deliver the water to the nozzle, can develop pressures up to 2.5 MPa. The dependence of water flow rate on pressure (discharge characteristic) is determined for each of the nozzles by measuring the time required to fill a control vessel for several values of pressure from the range 0-2.5 MPa. The design of the stand allows broad variation in the distance from the nozzle 1 to the specimen

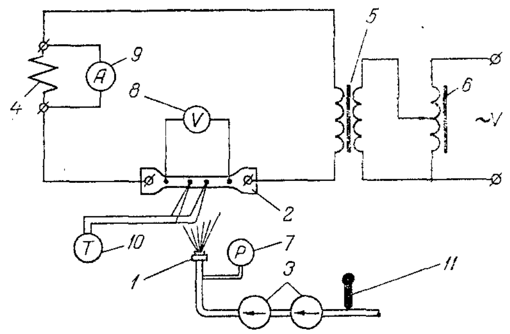

Fig. 1. Diagram of experimental stand.

2) the orientation of the nozzle–specimen system in space relative to the horizontal axis of rotation, and the angles of attack of the specimen surface in the vertical and horizontal planes. The specimen can be positioned at any point in the field of action of the jet in order to measure local heat-transfer coefficients (maximum dimensions of the field 1.8 × 1.0 m).

The distance from the nozzle to the surface being sprayed can be changed from 0.1 to 1.0 m, the angle of rotation of the specimen and nozzle can be changed from 0 to 360°, and the angle of attack can be changed from 90 to 10°. The spraying density in the field of action of the jet is measured using a water receptacle made of organic glass and containing 10 cells with a working cross section 10 × 60 mm. The cells have the form of oblique parallelepipeds. The lateral edges of the parallelepiped are inclined 24° to the horizontal to avoid having the water jet impact directly. The receptacle can be moved along the horizontal, so that the spraying density at any point in the field of action of the jet can be determined. The water from each cell of the receptacle flows into the corresponding cell of a water header along a rubber hose. The cells of the header have the shape of rectangular parallelepipeds and each contains an electronic level gauge. The voltage taken off the gauges is proportional to the spraying density in the sections of the field of the jet being investigated at a given moment of time. These voltages are sent to a K-200 data-servo system for conversion of the signal into digital form and output of the measurements of spraying density by cell in the form of punch tape. After a certain time interval has elapsed, the sytem feeds the signal to an actuator for moving a carriage with the receptacle a specified distance along the horizontal. The water receptacle then makes similar measurements at its new position. After the entire field of the jet has been traversed, the spraying-density measurements on the punch tape are put into a computer, which prints out the test data in the form of a table of "spraying density as a function of jet-field coordinates." Besides automatically measuring the spraying density, the stand determines the opening angle of the nozzle jet (spray) in the vertical and horizontal planes (the location of the boundary positions of the cells of the water receptacle, relative to the vertical and horizontal, at which water ceases to fall into these cells with a given distance from the nozzle to the surface being sprayed) and takes photographs and moving pictures.

Shown below are some results of a study of local heat exchange obtained on the above-described stand using full-cone nozzles designed by the UZTM (Ural Heavy Machinery Plant). The nozzles were positioned 0.4 m from the heated surface, which was oriented 45° to the horizontal. The angle of attack of the surface by the nozzle jet was 90°.

Figure 2 shows empirical dependences of the heat-transfer coefficient (I) at a surface temperature of 1000°C and spraying density (II), characterizing the change in these quantities along the vertical axis of the field of the jet (a circle), formed by nozzles with a discharge characteristic $V = 0.893p^{0.5}$. The top solid and dashed curves correspond to a pressure of 2.0 MPa, the middle to 1.5, and the bottom to 0.5 MPa. The "pure" effect of pressure on the rate of heat exchange in the field of action of the jet on the heated surface cannot be determined from Fig. 2: an increase in the pressure of the water in front of the nozzle causes the water flow rate to increase in accordance with the above-noted discharge characteristic and, thus, also increases the spraying density (the greater the pressure, the higher the dashed curves in Fig. 2). Comparison of the curves of spraying density and heat-transfer coefficient for identical values of pressure leads to the conclusion that the spraying density has a substantial effect on the heat-transfer coefficient. For example, at 2.0 MPa, the regions of the maximums of spraying density and heat-transfer coefficient coincide and are bounded by the coordinates ±0.12 m, reckoning from the geometric center of the jet. This region is characterized by the presence of a local minimum in the center of the jet, both for the spraying density and for the heat-transfer coefficient. The local "bumps" on the curves of spraying density for the investigated pressures cause similar "bumps" to appear at the same sites in the field of the jet on the curves of the heat-transfer coefficient.

It also follows from Fig. 2 that the curves of spraying density and heat-transfer coefficient "contract" toward the geometric center of the jet field as pressure increases from 0.5 to 2.0 MPa. This is due to a reduction in the opening angle of the jet with an increase in water pressure in front of the nozzle. In analyzing Fig. 2, it is important to note that, as the boundaries of the field are approached (as the spraying density approaches zero), the heat-transfer coefficient becomes equal to the coefficient of heat exchange for the case of radiation from an open surface with a temperature of 1000°C. This fact confirms the reliability of the empirical results obtained here.

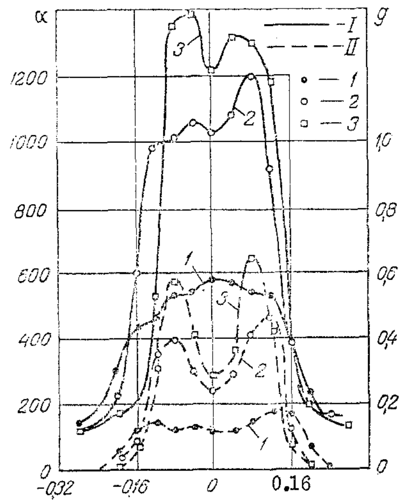

Fig. 2. Dependence of heat-transfer coefficient α, W/(m²·deg K) (I) and spraying density g·10², m³/(m²·sec) (II) on the distance from the center of the jet y, m. Water pressure in front of the nozzle: 1) 0.5; 2) 1.5; 3) 2.0 MPa.

Figure 3 shows the dependence of the heat-transfer coefficient in the geometric center of the field on the temperature of the surface at a water pressure of 0.5-2.0 MPa in front of full-cone nozzles with a discharge characteristic $V = 0.893p^{0.5}$. The distance from the surface to the nozzle, 0.4 m, was calculated from the following formula:

$$\alpha = [N - N_c F_c] (F_c + F_0)] | [F_0 (t - t_w)]. \qquad (1)$$

This formula takes into account that heat is removed by the dispersed medium only through the front surface of the specimen, with heat exchange occurring by radiation and convection with the environment on the other surfaces. An evaluation of the error in the empirical determination of the heat-transfer coefficient made by analyzing the errors of all of the measured quantities and those entering into Eq. (1) shows that the relative error is no greater than 12%. The maximum error in the measurement of spraying density is 10%. It follows from Fig. 3 that there is a similar law of change in the heat-transfer coefficient for all pressures: there is a region of sharp increase in the heat-transfer coefficient in Fig. 3, this region terminating in the temperature interval from 100 to 330-470°C. The upper limit of this interval increases with an increase in water pressure in front of the nozzle, i.e., an increase in water pressure (and, thus, the level of spraying density) expands the range of heat exchange (with respect to temperature) in which the heat-transfer coefficient reaches values characteristic for nucleate boiling of water in a large volume. It is apparent from Fig. 3 that the width of the aforementioned range increases 42.5% with a change in pressure from 0.5 to 2.0 MPa.

The dashed lines in Fig. 3 represent the region of transition from the region of maximum values of heat-transfer coefficient to the region of sheet boiling on the surface. It is unstable for conducting measurements. Located above 700°C is the region of minimum values of heat-transfer coefficient. It is interesting to note that with sheet boiling (made of nonwetting of the metal surface) the heat-transfer coefficient is nearly independent of the temperature of the surface or, strictly speaking, the precision of the experiment (with a maximum error of 12%) is inadequate to reveal such a relationship. These empirical results are in complete accord with the conclusions reached in [10], the authors of which studied heat transfer in the surface temperature range 700-1000°C but in a narrower interval of water pressures (no greater than 1.0 MPa). At the same time, the substantial effect of pressure (spraying density) on the rate of heat exchange can be seen from the curves in Fig. 3 in the region of sheet boiling. Thus, at a surface temperature of 900°C, an increase in water pressure from 0.5 to 2.0 MPa leads to an increase in the heat-transfer coefficient from 530 to 1300 W/(m²·deg K), i.e., by a factor of 2.5.

Figure 4 shows the presence of a direct proportional relationship between the heat-transfer coefficient in the center of the field at a surface temperature of 1000°C and the water pressure in front of the nozzles. The solid lines correspond to nozzles with a discharge characteristic $V = 1.673p^{0.5}$, while the dashed lines correspond to nozzles with a discharge characteristic $V = 0.893p^{0.5}$. The top solid line and dashed straight line were obtained for a distance of 0.2 m between the nozzle and surface, the middle solid and dashed lines for 0.4, and the bottom solid and dashed lines for 0.6 m.

In order to make practical use of the test data for heat-engineering calculations, we statistically analyzed the empirical measurements of spraying density and heat-transfer coefficient in the region of sheet boiling and obtained

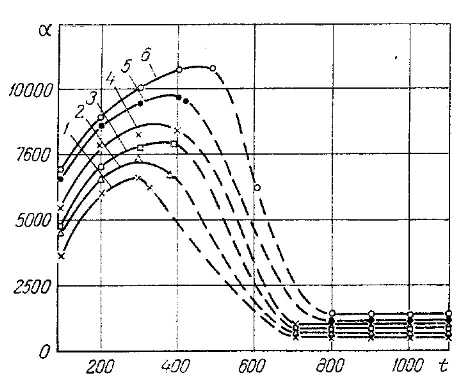

Fig. 3

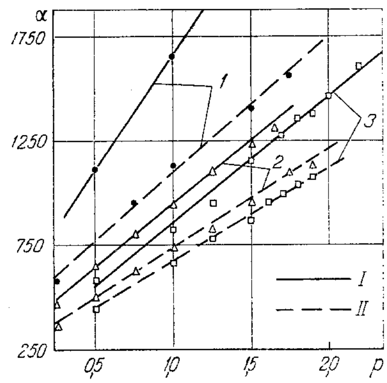

Fig. 4

Fig. 3. Dependence of heat-transfer coefficient in the center of the nozzle jet. Water pressure in front of the nozzle: 1) 0.5 MPa; 2) 0.75; 3) 1.0; 4) 1.5; 5) 1.75; 6) 2.0. t, °C.

Fig. 4. Dependence of heat-transfer coefficient in the center of the jet on water pressure (MPa) in front of the nozzle for nozzles with characteristics $V = 1.673p^{0.5}$ (I) and $V = 0.893p^{0.5}$ (II), respectively. The distance from the surface to the nozzle: 1) 0.2; 2) 0.4; 3) 0.6 m; p, MPa.

regression equations for nozzles of each type. For example, the following formulas were found for full-cone nozzles with a discharge characteristic $V = 1.673p^{0.5}$:

$$g = 0.446 \cdot 10^{-2} p^{0.5} \exp\left[-7.73\,(x/l)^2 - 6.31\,(y/l)^2\right], \tag{2}$$

$$\alpha = 132 + 6.24p + 197767g - 336.67pg. \tag{3}$$

Equations (2) and (3) are applicable for a distance of 0.4 m between the nozzle and surface. They can be used to calculate local heat exchange at any point in the field of the jet if the water pressure in front of the nozzle varies within the range 0.1-2.0 MPa and the surface temperature is equal to 700-1100°C. The multiple correlation coefficient for Eqs. (2) and (3) is about 0.9, which indicates the existence of a close functional relationship between the quantities in question. Equations (2) and (3) are also valuable in that they can be used to reveal the effect of individual parameters (such as pressure or spraying density alone) on the rate of local heat exchange, as cannot be done by experiment.

NOTATION

l, distance from nozzle to heated surface; x, y, horizontal and vertical coordinates in the field of the jet, read from the geometric center of the jet; V, volumetric flow rate of water through the nozzle; p, excess pressure of the water in front of the jet; F_0, area of sprayed surface of specimen; F_c, area of unsprayed surface of working part of specimen; t, temperature of sprayed surface of specimen; t_w, temperature of the water entering the nozzle; N, N_c, power required to heat the working part of the specimen to a given temperature with and without the spraying of water onto the surface, respectively; g, local spraying density; α, local heat-transfer coefficient.

LITERATURE CITED

1. A. Diner, "Survey of the literature on heat transfer in jet cooling," Chern. Met., No. 4, 26-29 (1976).
2. J. C. Leidenfrost, De aquae communis nonnullis qualitatibus tractatus, Duisberg, 1756.
3. L. H. J. Wachters, "An experimental study of spray cooling," Chem. Eng. Sci., 21, 1233-1239 (1966).
4. C. O. Pederson, "Heat transfer characteristics of water droplets impinging upon a heated surface," Int. J. Heat Mass Transfer, 13, 396-401 (1970).
5. E. A. Mizikar, "Spray cooling investigation for continuous casting of billets and blooms," Iron Steel Eng., 47, No. 6, 53-60 (1970).
6. C. J. Boogendoorn and R. Hond, "Leidenfrost temperature and heat transfer coefficients for water sprays impinging on a shot surface," Proceed. of Fifth Int. Heat Transfer Conference, Tokyo (1974).
7. V. N. Kulichkov, "Design and study of a secondary cooling system for shaped radial ingots," Author's Abstract of Candidate's Dissertation, Institute of Casting Problems, Academy of Sciences of the Ukrainian SSR, Kiev (1977), p. 23.
8. H. Junk, "Untersuchungen an fünf Warmbandstrassen die Auskühlung des Walzgutes auf dem Auslaufrollgang," Neue Huette, 17, 13-18 (1972).

9. M. Cumo and R. Pitimada, "Investigation of cooling of high temperature surfaces," Symposium Proceedings on Two-Phase Flow Dynamics, Eindhoven (1967), pp. 1325-1328.
10. H. Müller and R. Jeschar, "Untersuchung des Wärmeübergangs an einer simulierten Sekundärkühlzone beim Stranggiessverfahren," Arch. Eisenhüttenwesen, 44, No. 8, 589-594 (1973).

MASS TRANSFER OF A SINGLE BUBBLE IN A MINIMALLY FLUIDIZED GRANULAR BED

V. A. Borodulya, Yu. A. Buevich, and V. I. Dikalenko

UDC 66.096.5

The transfer of a gas admixture through the boundary of a cloud of closed circulation of gas is examined with a view to both molecular and convective dispersion.

Mass transfer of gas bubbles with a dense phase of the fluidized bed largely determines the effectiveness of operation of catalytic chemical reactors and other industrial equipment with granular material. Therefore, in addition to the accumulation of experimental data, simple models of mass transfer were also suggested; some of them were explained in [1-10]. However, the problem of the theoretical determination of the corresponding mass-transfer coefficient and its dependence on the physical and regime parameters is far from being satisfactorily solved; this is undoubtedly due to the variety of phenomena of differing physical nature affecting the process of mass transfer.

The numerous difficulties marking the problem of exchange of a single bubble with a single-phase liquid are in our case compounded by the fact that the dense phase of the fluidized bed is gas-permeable. This leads to the appearance of supplementary convective flows and complicates the purely hydrodynamic part of the problem: before the problem of mass transfer itself is solved, it is necessary to construct an acceptable model of the motion of both phases in the vicinity of the bubble; this in itself is a very nontrivial problem [1, 11, 12]. Two fundamentally different regimes of bubble motion are possible: with a cloud of closed circulation of gas and without it; the nature of the mass transfer will then also be fundamentally different.

Furthermore, in the gas stream permeating the dense phase, the compressibility of the process of molecular diffusion of the admixture is important, i.e., a magnitude of the type of the known coefficient of sinuousness has to be introduced. moreover, additional convective dispersion appears due to the mixing of elementary jets in the intersected pore space of a moving porous body formed by the moving particles which macroscopically can be described as a diffusion-type random process (see, e.g., [13, 14]). This dispersion is already substantial for beds with particles of $\sim 10^{-2}$-cm diameter; as a result, the effective diffusion coefficient in the vicinity of the bubble is nonuniform, being dependent on the particle size and the local porosity of the dense phase and on the relative gas velocity.

The supplementary transfer of the gaseous admixture in the general case is effected by particles that absorb or adsorb it, and in this process particles participate that belong to the dense phase as well as those that come through the bubble [5, 6, 15]. Moreover, adsorption of the admixture by the particles, as well as chemical reactions with its participation, obviously affect the convective diffusion of the admixture in the gas phase.

Finally, serious difficulties are also posed by the necessity of expressing the nonsteadiness of the process of mass transfer. In addition to non-steady-state effects connected with the establishment of the steady-state regime and ceasing to be substantial after a certain time interval since the beginning of the process has passed, exceeding the value $2R/U$, where R is the order of magnitude of the bubble radius or of the cloud of closed circulation around it, there appear effects that have no analog in the mass transfer of a bubble in a single-phase liquid. Firstly, a real bubble in a fluidized bed changes its volume in accordance with its lift, and this gives rise to a radial gas stream affecting the mass transfer [9, 10]. Secondly, the random pulsations of the bubble obviously lead not only to some nonsteadiness of the hydrodynamic fields but also to the "detachment" of parts of the cloud together with the gas contained in them; this is bound to intensify the mass transfer [4].

Lykov Institute of Heat and Mass Transfer, Academy of Sciences of the BSSR, Minsk. Institute of Problems of Mechanics, Academy of Sciences of the USSR, Moscow. Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 39, No. 2, pp. 323-333, August, 1980. Original article submitted July 24, 1979.

0022-0841/80/3902-0905 $07.50