HEAT EXCHANGE OF A GAS-SUSPENSION FLOW IN A SHORT CURVILINEAR CHANNEL

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Results are presented of an experimental investigation of the local heat exchange during turbulent motion of a gas—solid particle suspension in a short curvilinear channel of square cross section.

Short curvilinear channels (bends, branches, elbows) are elements of many power, technological apparatuses and heat exchangers in which two-phase systems of the gas—solid particle type are used as the working body or heat carrier.

At this time a large number of investigations devoted to the question of the convective heat exchange during the flow of a gas with suspended particles in straight pipes are known. These investigations show that the heat exchange for slightly dusted flows (β < 1-2) is independent of the presence of the particles and their concentration, but a reduction in the heat-exchange intensity as compared with the heat exchange of a pure gas is observed in a number of cases [1]. However, the deduction about a low or negative influence of the particles on the heat-exchange intensity of slightly dusted flows cannot possibly be extended to the case of heat exchange with walls causing rotation of the flow. In such cases, inertial particle flows directed to the channel walls occur because of the essential difference between the mass densities of the particles and gas. These particle flows can result in an abrupt growth of the heat-exchange intensity.

Papers devoted to an investigation of the heat exchange of dusted flows in curvilinear channels are not known.

Results of an experimental investigation of the local heat exchange of a gas flow slightly dusted with fine particles to the walls of a short curvilinear square channel under flow cooling conditions are elucidated in the present paper.

The experimental investigation was conducted in a nonstationary heat mode on an apparatus whose diagram is shown in Fig. 1. The apparatus was open to both the gas and solid phase and consisted of an experimental section, a gas line, a delivery, solid-phase metering and collecting systems, stream switching, and cooling and measuring systems.

The experimental section 18 was a short curvilinear channel with a 180° turning angle, 60 × 60-mm-square cross section, and 60 mm bending axis radius. It is formed by four plates, three of which are executed as a single whole, while the fourth is a cover attached to the frame by using bolts. The channel walls were fabricated from stainless steel and were 12 mm thick. In order to prevent overflow of heat between the walls, they were interconnected by thin 1-mm-thick crosspieces. Two-dimensionality of the temperature field in the walls was achieved by heat-insulating their side surfaces, using forge asbestos with an admixture of epoxy glue.

A 1.8-m-long heat-insulated stabilizing pipe was mounted in front of the experimental section.

The local heat-exchange coefficients in sections with central angle φ were determined at the time τ on the concave, convex, and flat surfaces forming the channel from the relationship

$$-\lambda\left(\frac{\partial T}{\partial n}\right)_{n=0,\varphi,\tau}=\alpha_{\varphi,\tau}\left(T_{f,\tau}-T_{w,\varphi,\tau}\right). \tag{1}$$

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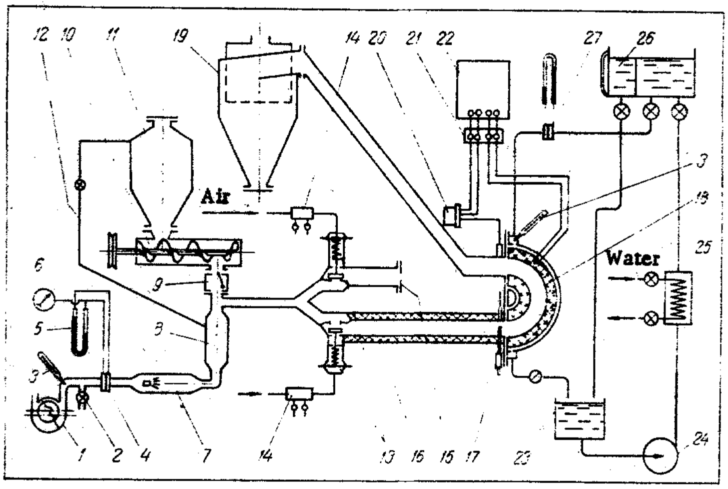

Fig. 1. Diagram of the experimental apparatus: 1) compressor; 2) valve; 3) thermometer; 4) measuring diaphragm; 5) piezometer; 6) manometer; 7) combustion chamber; 8) mixer; 9) valve; 10) spiral feeder; 11) bunker; 12) drainage tube; 13) valves; 14) electrical pneumatic valves; 15) diaphragm; 16) stabilizing heat-insulated pipe; 17) traversing gear with thermocouple; 18) curvilinear channel; 19) dust catcher; 20) pressure sensor; 21) constant temperature chamber; 22) oscillograph; 23) discharge tank; 24) water pump; 25) heat exchanger; 26) pressure tank; 27) measuring diaphragm.

The temperature gradient in the wall was found on the basis of a numerical solution of the problem of a two-dimensional nonstationary temperature field with the temperature dependence of the thermal conductivity and specific heat of its material taken into account. The boundary conditions in the form of time dependences of the temperature on the outline of a longitudinal wall section were determined from the experiment. The principles of the method of determining the heat-exchange coefficients by means of the temperature gradient in the wall of the experimental section in a nonstationary heat mode are described in [2]. Application of this method permits a substantial reduction in the duration of the experiment. The results of the experiments were processed on the M-220 electronic computer. The local heat-exchange coefficients were evaluated at 8 channel sections located every 20° around the bending angle.

The temperature distribution on the channel wall surfaces was measured as a function of time by Nichrome—Constantan thermocouples by using three K-20-22 oscillographs. A total of 53 thermocouples were mounted on the outer (concave), inner (convex), and side (flat) walls. Thermocouple wires 0.2 mm in diameter with heat-resistant enamel insulation were stowed in 0.3 × 0.4-mm grooves and were sealed by using epoxy glue. The exposed ends of a 1-mm-long thermocouple were spot-welded at two 0.2 × 0.1 × 1-mm slots separated by 0.5-1 mm. Therefore, the hot ends of the thermocouple were separated by 0.5-1-mm spacings. All the thermocouples were calibrated under static conditions in combination with the oscillograph galvanometers.

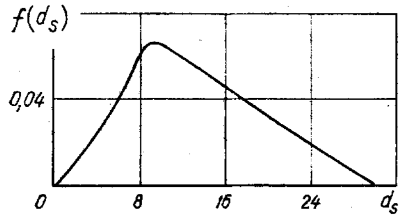

Fig. 2. Distribution density function of the mass of Al_2O_3 particles according to size d_s, μ.

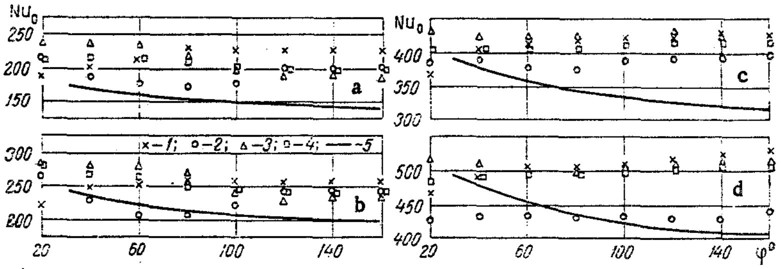

Fig. 3. Dependence of Nu_0 on the angle of rotation φ: a) Re = $5.62 \cdot 10^4$; b) $8.4 \cdot 10^4$; c) $15.05 \cdot 10^4$; d) $20.2 \cdot 10^4$; 1) concave wall; 2) convex; 3) flat; 4) mean values with respect to the perimeter; 5) according to (2).

The external wall surfaces of the experimental channel were water-cooled; the gas channel included the compressor 1 and the combustion chamber 7.

The gas discharge was comprised of the discharges of the air, measured by using the measuring diaphragm 4, and of the fuel, measured by means of the pressure drop in the nozzle.

The solid-phase delivery and collection system consisted of the bunker 11, the spiral-type feeder 10, the valve 9, the mixer 8, and the dust catcher 19. The pressures in the bunker cavities and the mixer were equalized by using the drainage tube 12.

The discharge of the solid phase was determined in the following manner. The dust charge needed to conduct one experiment was loaded into the bunker. The time at the onset of dust feeding and the time at its completion were determined by the change in flow pressure and temperature determined on the oscillograms; during delivery of the dust, visible aperiodic fluctuations were observed on the pressure and temperature curves. Nonuniformity of the dust delivery to the working band of the discharges was 4-5%.

The stream pressures at the entrance to the curvilinear channel and at its exit were measured by using 20 sensors and standard manometers. The stream temperature at the channel entrance was measured by an open Chromel—Copel thermocouple mounted on the traversing gear 17.

The flow switching system is intended to accomplish a sudden start-up of the stream in the experimental section and comprises the valves 13, which operated in synchronization and were controlled by using the electrical pneumatic valves 14. By using the valves 13 the flow was directed either through the stabilizing heat-insulated pipe 16 in the experimental section or through the adjusted mainline in the atmosphere. The hydraulic resistance of the adjusted mainline was set equal to the resistance of the fundamental mainline by using the detachable diaphragm 15. This assured equality of the discharges during operation in the fundamental and adjusted mainlines.

The experimental section was disconnected from the stabilizing pipe before the start of the experiment, and a channel of equivalent geometric characteristics was mounted in its place. The feed pipes of the fundamental mainline were heated by hot gas prior to the start-up of the stationary thermal state, after which the hot gas was switched to the adjusted mainline, and the experimental section was mounted in place. The recording apparatus was switched on, the hot stream was switched into the fundamental mainline by using the valves, valve 9 was simultaneously opened, and the dust catcher was switched on. The duration of each experiment was 60-70 sec.

The relative limiting error in determining the heat-exchange coefficients of a gas-suspension flow was 22%.

The investigation was conducted in such a way that the relative influence of the solid particles on the heat exchange could be clarified by comparing the heat-exchange coefficients measured in the tests for the two-phase stream and for the pure gas.

The carrying medium was a mixture of the combustion products of kerosene and air. Because the combustion of the kerosene occurred under conditions of large values of the excess air coefficient (the combustion temperature was comparatively low), the physical properties of the mixture were close to the properties of air.

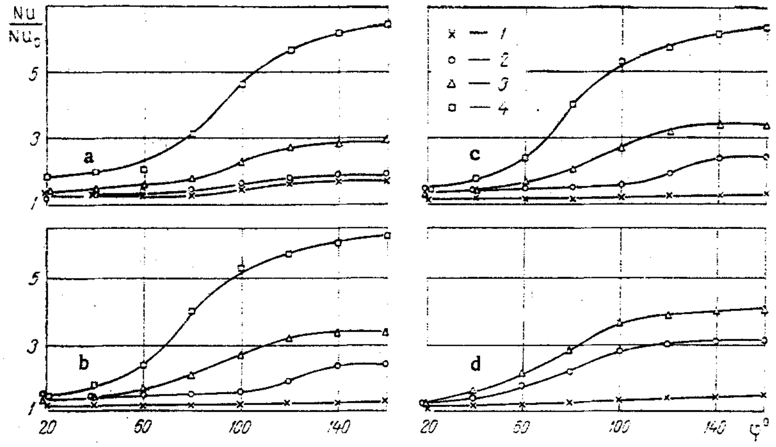

Fig. 4. Dependence of Nu/Nu_0 on a concave wall on the angle of rotation: a: Re = $5.62 \cdot 10^4$; 1) β = 0.285; 2) 0.295; 3) 0.528; 4) 0.966; b: Re = $8.4 \cdot 10^4$; 1) β = 0.177; 2) 0.312; 3) 0.484; 4) 1.026; c: Re = $15.05 \cdot 10^4$; 1) β = 0.141; 2) 0.304; 3) 0.605; 4) 0.856; d: Re = $20.2 \cdot 10^4$; 1) β = 0.147; 2) 0.396; 3) 0.53.

The stream was dusted with aluminum oxide powder with a maximum particle size of 30μ. The distribution density function of the solid-phase mass according to particle size is presented in Fig. 2.

The tests were conducted under the following conditions: T_f = 440-520°K; T_w = 310-370°K; p = 1 bar; w = 30-140 m/sec; Re = $(5.62-23.5) \cdot 10^4$ (the governing temperature is the stream temperature at the channel entrance and the governing size is equivalent to the channel diameter); and β = 0-1.026. The change in stream temperature during cooling did not exceed 10° in the experimental section. Hence, the stream temperature at the channel entrance was used as the reference temperature [$T_{f,\tau}$ in (1)].

The heat-exchange coefficients at different sections of the channel almost reached quasistationary values in a 30-50-sec outflow.

Quasistationary values of α corresponding to the time τ = 50 sec were later taken for the analysis.

The results of the experimental determination of the local heat-exchange coefficients of the pure gas with the concave, convex, and flat surfaces of the curvilinear channel for four values of the Re number are shown in Fig. 3. The data on the mean heat-exchange coefficients for each channel section are superposed here. The averaging is performed taking into account the relationship between the areas of the individual walls under the assumption that the heat-exchange coefficients on the flat surfaces are identical.

As is seen from the graphs, the heat-exchange coefficients on the channel surfaces are distinct: They have the greatest values on the concave surface in the major part of the channel ($\varphi > 30°$), and they are least on the convex part.

The magnitudes of the local heat-exchange coefficients and the nature of their distributions over the separate surfaces and along the length (angle of rotation) in a short curvilinear square channel are determined by the total action of the three fundamental effects due to the influence of mass centrifugal forces.

1. Reconstruction of the velocity profile at the concave and convex walls occurs in the entrance section of the channel. The flow is convergent in character upon entering the curvilinear section at the convex wall (the velocity increases to the appropriate constant circulation law) and is divergent in character (the velocity diminishes) at the concave wall. Hence, the heat-exchange coefficients on the convex wall can be greater in the entrance section ($\varphi < 30°$) than on the concave wall.

2. The nature of the flow at the concave and convex walls is distinct. Centrifugal forces exert an active effect in the boundary layer on the concave wall: By perturbing the stream, Taylor–Goertler vortices can occur at the concave wall. Centrifugal forces exert a conservative effect at the convex wall; they stabilize the boundary layer. As the results of a theoretical and experimental investigation show, the heat-exchange intensity on a concave wall is higher than on a flat plate, while it is lower on a convex wall [3].

3. An important singularity of the fluid flow in curvilinear channels with circular or almost square cross section (the influence of the side walls is felt in such channels) is the origination of a vortex pair occupying the whole space of the cross section. Secondary flows increase the heat-exchange intensity on all the channel surfaces. A rise in the heat-exchange coefficients with the increase in the angle of rotation (channel length), which is observed in the majority of experiments, is apparently due to the development of secondary flows.

The local heat exchange in turbulent fluid flow in short curvilinear channels has been studied inadequately at this time; hence, the data obtained were compared with computations of the local heat-elimination coefficients in a straight pipe by means of the formula [4]

$$Nu_f = 0.022 Re_f^{0.8} Pr_f^{0.43} \cdot 1.38 \left(\frac{x}{d}\right)^{-0.12}. \tag{2}$$

The spacing x was measured along the channel bending axis, and the equivalent channel diameter was used as the governing dimension.

As is seen from the graphs, the local heat-exchange coefficients on the separate walls and the heat-exchange coefficients averaged with respect to the cross-section perimeter are higher for $\varphi > 80°$ than in a straight pipe. In this part of the channel secondary flows evidently already exert influence on the heat-exchange intensity in a curvilinear channel.

The relative influence of the solid phase on the total level and distribution of the heat-exchange coefficient on the concave wall of a curvilinear channel is shown in Fig. 4. Here Nu is the Nusselt number for the two-phase stream and Nu_0 is the Nusselt number for a pure gas at the same Re number.

As an analysis of the graphs shows, the presence of solid particles in the stream results in a substantial rise in the heat-exchange intensity at the concave wall. Thus, a sixfold increase in the heat-exchange coefficient is observed for $\beta = 0.966$-1.026 and $Re = (5.62$-$8.4) \cdot 10^4$ (Fig. 4a, b). Hence the influence of the particles on the heat exchange is magnified with the increase in the angle of channel rotation.

The nature of the change in temperature of the "hot" concave wall surface in the tests with a gas suspension was the same as the nature of the change in Nu/Nu_0, while this temperature was approximately identical on the whole surface in tests with a "pure" gas.

A rise in the heat-exchange level on the concave wall because of the presence of particles in the stream is due to the following. Inertial displacement of the particles to the concave wall surface – separation of the particles – occurs in a curvilinear channel with a gas-suspension flow. Particles incident on the wall exert on effect on the heat-exchange process because of mechanical and thermal interaction with the boundary layer and the wall surface. The heat-exchange intensity between a gas-suspension stream and a concave wall surface changes in comparison to the heat exchange of a pure gas because of the perturbations which the particles passing through induce in the boundary layer and because of heat transfer to the wall by colliding particles. Moreover, because of the lack of a thermal interphasal equilibrium, the particles penetrating the boundary layer raise its temperature as compared to the pure-gas temperature. A rise in the local particle concentration occurs because of inertial incidence of particles at the concave wall, and this can significantly exceed their concentration on the convex and flat walls. After the particles fall on the concave wall, they continue to move in the near-wall zone. Hence an increase in the angle of rotation is accompanied by an increase in the local particle concentration, which can exceed 10-fold the concentration at the entrance to the channel.

Therefore, a substantial rise in the heat-exchange intensity can occur because of the presence of particles in the stream even for comparatively low particle concentrations if inertial incidence of particles on the wall is observed.

In the range of variation of the solid-phase concentration investigated, the presence of particles in the stream resulted, in the majority of experiments, in a reduction in the heat exchange on the convex and flat walls, the values of Nu/Nu_0 for these walls being 0.7-1.09. Hence, the test data have a substantial spread, reaching 10%. This is explained principally by the particles adhering to the cold surface of the convex and flat walls. Visual examination of the channel surfaces, which was carried out after each experiment, showed that there were actually outcrops of aluminum oxide particles on these surfaces, while the concave surface remained clean. This surface was evidently cleansed by the abrasive action of the inertial stream of particles incident on the concave wall and then moving along it.

The distinct nature of the influence of a solid phase on the heat exchange of concave, convex, and flat walls affords a foundation for assuming that the mechanical and thermal action of the inertial stream of particles on the stream structure in the near-wall zone and the thermal resistance of the viscous sublayer, as well as the increase in the true particle concentration at the wall surface, are of primary value in the intensification of the heat exchange on a concave wall. Hence, an investigation of the influence of the separate factors on the heat-exchange intensification because of the presence of particles and the extension of the test data should be based on a study of the structure of a two-phase gas flow with suspended particles in a curvilinear channel with an estimation of the velocity, temperature, and density of the particles and the gas, the density of the inertial mass flux of the particles incident on the separate sections of the concave wall surface, and the local particle concentration in the near-wall zone.

NOTATION

d, pipe diameter; d_s, particle diameter, G, G_s, mass flow rates of the gas and particles; n, normal to the heat-exchange surface; p, pressure; T, temperature; $(dT/dn)_{n=0}$, normal temperature gradient at the wall on the heat-exchange surface; T_f, T_w, stream and wall temperatures; w, stream velocity; x, distance to the section under consideration; α, local heat-exchange coefficient; $\beta = G_s/G$, coefficient of solid-phase particle discharge concentration; τ, time; λ, coefficient of wall heat conduction; φ, central angle characterizing the channel length; Nu, Nusselt number; Pr, Prandtl number; Re, Reynolds number.

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HEAT TRANSFER DURING THE MOTION OF COLD GASEOUS NITROGEN IN A POROUS TUBE

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Heat-transfer processes taking place during the motion of nitrogen in a tube are studied experimentally under conditions in which some of the gas filters through the wall. The effect of suction on the distribution of wall temperature and the intensity of heat transfer is examined.

A number of papers devoted to problems of heat transfer and hydrodynamics during the motion of liquid in channels subject to suction and injection have recently appeared in the literature. These papers have appeared because of the use of porous heat-exchangers in various fields of the chemical and power industries.

During the motion of liquid in a porous tube, suction has a turbulizing effect on laminar flow in the tube, this effect increasing with increasing suction [1-9]; if there is an external flow around the porous walls, suction through the wall laminarizes the external flow.

Heat transfer during the laminar flow of a liquid in a porous tube was analyzed in [1] over a wide range of the filtration rate through the porous wall. It was found that as a result of injection the heat-transfer coefficient at the boundary between the flow and the wall diminished, while as a result of suction it increased, i.e., suction intensified the heat-transfer process. For both suction and injection the section of steady-state

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