AVERAGE AND LOCAL CHARACTERISTICS OF THE HYDRODYNAMICS AND HEAT EXCHANGE IN CHANNELS CONTAINING SPHERICAL ELEMENTS

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A study is made of the hydraulic resistance and the average and local heat-treatment coefficients in channels packed with spheres, as well as the hydrodynamics of the flow over a spherical element.

The analysis of data on the investigation of processes of hydrodynamics and heat exchange in channels packed with spheres [1-4] has shown the inadequate degree of study of the hydraulic resistance and heat exchange as well as the total absence of information on the local heat exchange and hydrodynamics in such channels. In the present work, therefore, we investigated the hydraulic resistance and average heat exchange in channels is packed with spheres in the range of variation of coolant velocity from 0.5 to 50 m/sec, which corresponds to variation of the Reynolds number in the range of $5 \cdot 10^2 - 7 \cdot 10^4$, the local heat-transfer coefficients from spherical elements of different packings, and the structure of the gas stream in a channel packed with spheres to clarify the mechanisms of the processes of hydrodynamics and heat exchange.

The hydraulic resistance of channels containing spheres was studied in channels of eight standard sizes from 32 to 150 mm in diameter and up to 1800 mm long. Spheres of 17 standard sizes with diameters from 12.7 to 76.2 mm were used. The channels and spheres were finished to classes 6-8.

The experimental data on the hydraulic resistance of packings of spheres in channels were analyzed with respect to the equation

$$\xi = \frac{2\Delta P d_e^{av}}{\rho v^2}. \tag{1}$$

The experimental investigation of the hydraulic resistance was preceded by a study of the sphere packings which develop in channels. It was established that there is a one-to-one correspondence between the character of the packing and the ratio n = D/d. The existence of four types of packings, differing in structure, was detected in the channels as a function of the ratio n:

1) straight-line packing, created artificially with spacing ribs, with n from 1 to ∞, characterized by the arrangement of the centers of the spheres and the contact points on the channel axis;

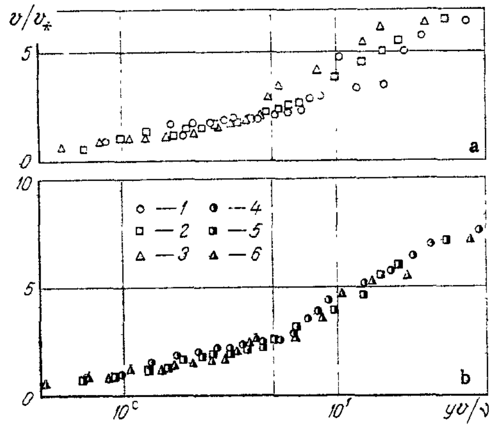

Fig. 1. Velocity profiles at the surface of sphere No. 7 of straight-line packing (the angle φ is measured from the front point): a) cross section No. 1, $\varphi = 60°$; 1) $Re_\infty = 1.92 \cdot 10^4$; 2) $1.32 \cdot 10^4$; 3) $0.92 \cdot 10^4$; b) cross section No. 4, $\varphi = 90°$; 4) $Re_\infty = 1.92 \cdot 10^4$; 5) $1.32 \cdot 10^4$; 6) $0.92 \cdot 10^4$.

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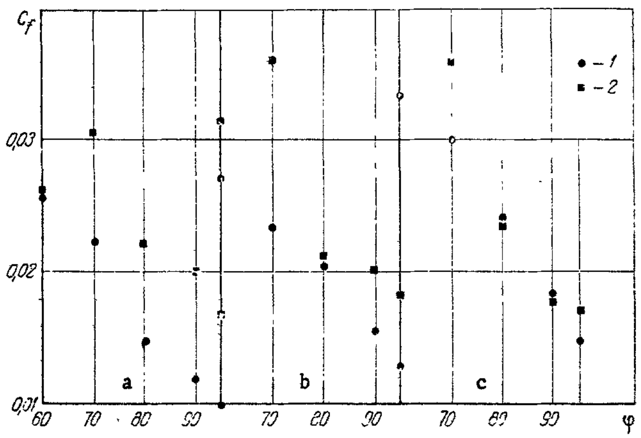

Fig. 2. Distribution of coefficient of friction over the surface of sphere No. 7 of straight-line packing: a) $Re_\infty = 2.08 \cdot 10^4$; b) $Re_\infty = 1.30 \cdot 10^4$; c) $Re_\infty = 0.9 \cdot 10^4$; 1) based on data of hydrodynamic studies; 2) based on heat-exchange data.

2) checkerboard packing with n from 1 to 1.867, characterized by the arrangement of the centers of the spheres and the contact points in a diametral plane;

3) helical packing with n from 1.867 to 2, characterized by the arrangement of the centers of the spheres along a helical line, the pitch of which is determined by the ratio n;

4) circular packing with different numbers of spheres (from 2 and 5) for n from 2 to 3, characterized by the arrangement of the centers of the spheres of one array in one plane perpendicular to the channel axis.

The experimental studies of the hydraulic resistance were carried out under isothermal conditions. The following criterial equations were obtained to calculate the coefficient of hydraulic resistance:

straightline. packing

$$\xi = 11.64 \mathrm{Re}^{-0.24} n^{-1.11}, \tag{2}$$

checkerboard

$$\xi = 30 \mathrm{Re}^{-0.116} n^{-3.98}, \tag{3}$$

helical

$$\xi = 0.0088 \mathrm{Re}^{-0.21} n^{11.5}, \tag{4}$$

circular

$$\xi = 0.66 \mathrm{Re}^{-0.04} n^{3}. \tag{5}$$

In all the types of packings one observes a monotonic decrease in the coefficient of hydraulic resistance with an increase in the Reynolds number, which is connected with a decrease in the coefficient of friction at the

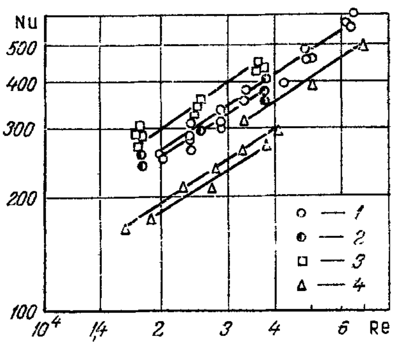

Fig. 3. Dependences of coefficient of heat transfer on Reynolds number for different packings of spherical elements: 1) checkerboard; 2) helical; 3) circular; 4) straight line.

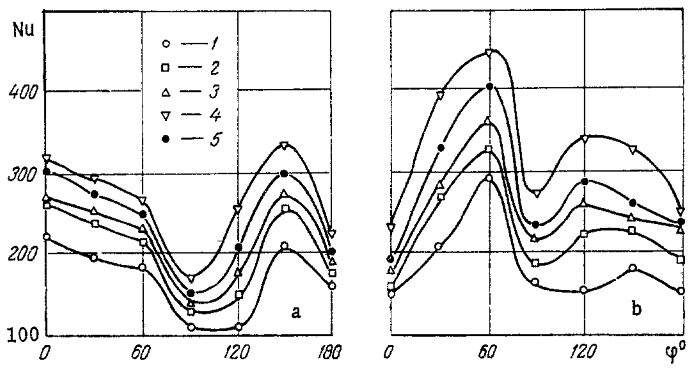

Fig. 4. Distribution of local heat-transfer coefficient over the surface of a sphere in straight-line packing: a) sphere No. 1; b) sphere No. 7; 1) Re = $1.58 \cdot 10^4$; 2) $2.25 \cdot 10^4$; 3) $2.74 \cdot 10^4$; 4) $3.16 \cdot 10^4$; 5) $4.0 \cdot 10^4$; $N_{loc} = \alpha_{loc} d_e^{av}/\lambda_b$.

surfaces of the wall and the spheres and with a decrease in the coefficient of induced drag. This kind of variation in ξ is evidently explained by a decrease in the size of the associated vortices and an increase in the frequency of their separation with the coefficient of pressure drag remaining practically constant in the investigated range of Reynolds numbers. The quantity n has different effects on the coefficient of hydraulic resistance. With its increase in straight-line and checkerboard packings, the coefficient of hydraulic resistance decreases as a result of an increase in the through cross section and a decrease in the coefficient of pressure drag, while in helical and circular packings it increases because of blocking of the stream by the layer of spheres and an increase in the pressure coefficient.

The investigation of the sections of hydrodynamic stabilization in the channels under consideration and of the aerodynamics of the flow over individual elements in the packings was carried out with a spherical probe 60 mm in diameter with inlet openings 0.6 mm in diameter. The pressure distribution over the surface of a sphere was determined with the aid of this probe. The pressure P_0 in the stream was sampled with a Prandtl tube at a distance of 150 mm from the first array of spheres. The experimental results were treated in the form of the dependences of the dimensionless pressure coefficient on the position of the test point on the sphere surface.

The stream velocity in the unencumbered channel ahead of the bed of spheres was taken as the determining velocity in the equation

$$\overline{P} = (P_i - P_0)/0.5\,\rho\, v_\infty^2 \tag{6}$$

for the pressure coefficient.

From the pressure distributions over the surfaces of spheres located at different points over the height of the bed which we obtained we determined the coefficients of pressure drag of the spherical elements, the sections of hydrodynamic stabilization, and the velocities at the limit of the boundary layer. In channels with a straight-line arrangement of spheres, the section of hydrodynamic stabilization is confined to the first two layers of spheres.

The following function can be recommended for calculating the coefficient of pressure drag for the first sphere of a straight-line packing:

$$C_{x\mathrm{p}} = C_{x0} \left(\frac{n^2}{n^2 - 1}\right)^{2.4}. \tag{7}$$

The sections of hydrodynamic stabilization in the other types of packings, which comprise the first three arrays for checkerboard packing, the first five arrays for helical packing, and the first two arrays of spheres for circular packing, were established by analogous investigations.

The regions of existence of the boundary layer at the surfaces of the spheres in the straight-line packing were determined by the liquid-film method using a suspension of aluminum dioxide in kerosene. Visualization of the flow showed that in the vicinities of the contact points there are stagnant zones whose size increases with an increase in n. The region of existence of the boundary layer moves downstream along the surface of a sphere with a decrease in n. The movement comprised 6-8° in the channels investigated. No influence of the Reynolds number on the position of the region of existence of a boundary layer on a sphere was detected in the investigated range of its variation.

The study of the boundary layer at the surface of a sphere in the region of its existence was carried out with a DISA thermoanemometric complex by the method of a constant filament temperature. The investigations were made for straight-line packing in the stabilized section of flow in the range of variation of the Reynolds number from 10^4 to $4 \cdot 10^4$ with isothermal flow.

With allowance for the thermal influence of the sphere surface on the magnitude of the filament signal of the thermoanemometer, we obtained the following dependence for the velocity:

$$v = \sqrt[0.45]{(U_0^2 + 0.135\,U_0'^2 - U_{*0}^2)\,/B}. \tag{8}$$

The expression for the pulsation of the longitudinal velocity component has the form

$$\sqrt{\overline{v'^2}} = 4.44\;\frac{v\,U_0\,dU_0}{U_0^2 + 0.135\,U_0'^2 - U_{*0}^2}. \tag{9}$$

The distance of the initial point from the surface of the sphere was ~10 μm. The detector filament was calibrated before each series of tests in the calibration device of the DISA complex.

The velocity profiles in the boundary layer obtained as a result of the treatment of the experimental data are presented in generalized coordinates in Fig. 1. An analysis of the results indicates the presence of three regions in the boundary layer: a region of laminar sublayer ~20 μm thick, a power-law region ~80 μm thick, which comprises ~70% of the thickness of the boundary layer, and a region of the universal velocity-defect law, The thickness of the boundary layer, the displacement thickness, and the thickness of momentum loss were determined by the method of numerical integration.

From the value of the form factor $H_{1,2} = \delta_1/\delta_2$, which varies from 1.5 to 2, it was concluded, in accordance with the recommendations of Schlichting, that the boundary layer has a transitional character in the given case. At the separation point the form factor is $H_{1,2} = 2$, which corresponds to the data of [5].

An analysis of the results of the investigation of the velocity pulsations shows that in the boundary layer at a sphere one observes a high turbulence intensity, extending up to the laminar sublayer at 16% at a distance of 500 μm from the surface of the sphere in the same cross section. The maximum of the turbulence intensity is noted in the cross section located at an angle $\varphi = 80°$ from the front point of the sphere. These results agree with the data of [5, 6].

The frictional stresses at the surface of a sphere in the region of existence of the boundary layer were determined by graphic differentiation of the velocity profiles in the laminar sublayer. The character of the variation in the dimensionless coefficient of friction is presented in Fig. 2. The decrease in the frictional stress as the separation point is approached is seen from the figure.

The heat transfer from a bed of spheres was studied by the method of local thermal modeling in the steady state by calorimeter probes 60 mm in diameter made from L-62 brand brass. The calorimeter probe was isolated from the steel spheres by a layer of Teflon spheres of the same diameter. The sections of thermal stabilization were investigated by placing the probe at different points over the height of the bed.

The results are presented in Fig. 3 and are generalized by the following criterial functions:

for straight-line packing

$$\mathrm{Nu} = 0.23\,\mathrm{Re}^{0.7} n^{-0.6}, \tag{10}$$

for helical and checkerboard packings

$$\mathrm{Nu} = 0.25\,\mathrm{Re}^{0.7}, \tag{11}$$

for circular packing

$$\mathrm{Nu} = 0.29\,\mathrm{Re}^{0.7}. \tag{12}$$

An analysis of the experimental data shows that in all the types of packings the heat transfer from the spherical elements grows monotonically at the same rate with an increase in the Reynolds number.

An influence of the ratio n on the coefficients of heat transfer is detected only in the straight-line packing. The intensity of heat transfer decreases with an increase in n, which can be explained by an increase in the size of the stagnant zones in the vicinities of the contact points with a simultaneous decrease in the intensity of circulation of the attached vortices and thickening of the boundary layer at the heat-exchange surfaces.

The local heat-transfer coefficient was investigated with local heat-flux detectors. The heat-flux detectors were set in recesses made in a spherical probe every 30° from the front point. The diameter of a detector of the spherical probe is 5 mm. A detector consists of a battery of copper — Constantan thermocouples 0.1 mm in diameter, connected in series. The principle of its operation is based on the summing of the emf of single differential thermocouples having an intermediate wall, the role of which is filled by a celluloid film 2.5 mm wide and 0.5 mm thick. The thermocouple junctions were located exactly at the faces of the film. The film with the battery of thermocouples uniformly distributed on it was rolled into a spiral, placed in a cap at the center of which the button of a Chromel — Alumel thermocouple was welded, and fastened with epoxy resin in the recess under a detector. A detector is only sensitive to a heat flux penetrating it perpendicularly to the ends. A detector was calibrated under steady thermal conditions by the radiation method. The sensitivity of a detector in the temperature range from 0 to 150°C was determined as a result of the calibration. The experimental data on the local heat-transfer coefficient were treated using the equation

$$\alpha_{loc} = q_{loc} \Big/ \left(t_d - \frac{t_{in} - t_{out}}{2} \right). \tag{14}$$

The results of a study of the local heat-transfer coefficient in straight-line packing are presented in Fig. 4 as an illustration of the experiments performed.

An analysis of the results shows that the minima of the heat-transfer coefficient are observed in the zones of the points of contact of the spheres with neighboring spheres and with the channel wall and in the zones of separation of the boundary layer. The maxima of the heat-transfer coefficient are located in the zones of existence of the boundary layer. The values of the heat-transfer coefficient of nonuniformity over the sphere surfaces are:

straight-line packing, $K_\alpha = 1.4$–2.6;

checkerboard packing, $K_\alpha = 2.2$–3.0;

helical packing, $K_\alpha = 3.0$–3.2;

circular packing, $K_\alpha = 3.0$–4.0.

The heat-transfer coefficient of nonuniformity increases with an increase in the Reynolds number and in n for all the packings. A comparison of the values of the nonuniformity of the heat-transfer coefficients in the channel variant and in the AVR type of packing [1] indicates a decrease of nonuniformity by 2-3 times in the channel variant.

The local dimensionless coefficients of friction were calculated through the Stanton numbers from the results of the study of the local heat transfer from the surface of a sphere in straight-line packing.

Good agreement (within the limits of the experimental errors) was obtained between the calculated values of the dimensionless coefficient of friction and the results obtained from experiments on hydrodynamics (Fig. 2), which indicates that the Reynolds analogy is satisfied in the region of existence of a boundary layer on a sphere.

The optimum range of n for channels containing spherical elements was chosen on the basis of an estimate of the energy efficiency of the investigated channels. They are channels with checkerboard packing in the range of n from 1.65 to 1.85.

NOTATION

ξ, coefficient of hydraulic resistance; ΔP, pressure drop in channel; v, velocity in a channel with an average equivalent diameter d_e^{av} equal to the diameter of a cylindrical channel whose volume equals the free volume of the channel filled with spheres and whose length equals the length of the channel being studied; ρ, gas density; D, channel diameter; d, diameter of a spherical element; n = D/d; Re, Reynolds number; $\bar{P}$, pressure coefficient; P_i, pressure at a point of a spherical element; P_0, pressure ahead of the bed of spheres; C_{x_p}, coefficient of pressure drag for the first sphere of straight-line packing; C_{x_0}, coefficient of drag of a sphere in a free stream; U_0, voltage of output of thermoanemometer; U_0', apparent zero voltage at output of thermoanemometer; B, filament constant (proportionality factor), determined by calibrating the filament in an airstream; U_{*0}, voltage at output of thermoanemometer in the absence of gas motion near the wall; v', magnitude of longitudinal velocity pulsation; δ_1, displacement thickness of boundary layer; δ_2, thickness of momentum loss; $H_{1,2} = \delta_1/\delta_2$, form factor of boundary layer; Nu, Nusselt number; α_{loc}, local heat-transfer coefficient of a spherical element; q_{loc}, local heat flux from surface of a spherical element; t_d, detector temperature; t_{in}, t_{out}, air temperatures at inlet and outlet of channel; K_α, heat-transfer coefficient of nonuniformity over surface of a spherical element.

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INTENSIFICATION OF CONVECTIVE HEAT EXCHANGE BY SPIRAL SWIRLERS IN THE FLOW OF ANOMALOUSLY VISCOUS LIQUIDS IN PIPES

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The results of an experimental investigation of the intensification of convective heat exchange in the flow of anomalously viscous liquids are presented. An estimate is made of the thermodynamic and energy efficiencies of the use of longitudinal spiral swirlers.

One of the well-known means of intensification of heat exchange in pipes consists in acting on the boundary region of flow using spiral wire swirlers, very effective technologically in preparation and utilization. Investigators have now accumulated extensive experimental material on heat exchange in pipes containing various types of intensifiers, including spiral wire swirlers [1-3]. It should be noted that all the papers have been devoted to the intensification of heat exchange during the motion of viscous liquids in pipes.

Now from an analysis of the known work it is seen that the use of spiral wire swirlers intensifies the heat exchange in viscous liquids up to 3 times [2], with the greatest increase in the heat-transfer coefficients being observed in the region of Reynolds numbers from 3500 to 8000. This is connected with the formation and development of vortices intensifying the process of heat exchange. With the gradual development of turbulence the quantity Nu/Nu_0 decreases somewhat, nevertheless remaining considerably higher than unity, since at $Re > 8000$ turbulence begins to have the dominant effect on the heat exchange in a viscous liquid while the role of vortices gradually decreases.

Unfortunately, data on the intensification of convective heat exchange in the flow of anomalously viscous liquids, which find the widest application, are entirely absent at present.

On the basis of the fact that the use of spiral wire swirlers gives a considerable gain in heat transfer in the flow of viscous liquids, we attempted to experimentally determine the possibilities of the intensification of heat exchange in anomalously viscous media using the indicated swirlers and to estimate the efficiency of their use.

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