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TURBULENT HEAT AND MASS TRANSFER IN CONFINED TWISTED FLOWS

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Results are presented from a study of aerodynamics and heat and mass transfer in turbulent rotating flows. An examination is made of flow in the initial section of a pipe, flows with peripheral twisting, and stabilization of an axial jet by a twisted flow in a swirl chamber.

Vortex flows are widely used in different areas of modern technology. As a rule, twisting of a flow is connected with a need to intensify transport processes in power plants or chemical processing equipment. However, in several cases, twisting can also be used to reduce heat and mass transfer – such as in the stabilization of plasma jets and flames.

Many studies have already examined twisted flows. Most of these studies have investigated general laws governing the flow, laminar flow, and flow stability. Most investigations of turbulent transport have been conducted for fully developed pipe flow.

Vortex flows are characterized by wide variety even in regard to the qualitative flow pattern, which is determined mainly by the geometric and discharge characteristics. The may methods available for twisting a flow – swirl vanes at the inlet of a channel, tangential gas feed, belt- and screw-type swirlers, etc. – seriously complicates analysis and generalization of experimental results. The most common approach here has been to generalize test data in relation to the initial geometric and discharge conditions of the specific equipment used in the experiment. A detailed examination of questions related to heat transfer in channels with the use of different methods of flow twisting is presented in [1].

The studies [2, 3] and certain later investigations used the so-called principle of streamline rectification to calculate turbulent heat and mass transfer and friction. This approach involves converting relations for the heat and mass transfer coefficients and friction coefficient in a plane boundary layer to the case of a twisted flow, assuming that the twisted flow is examined along the helical streamline with the corresponding parameters on the external boundary of the boundary layer. Here, in essence, the investigator is considering the intensification of heat transfer and friction which occurs only as a result of an increase in the velocity vector on the external boundary of the boundary layer and the longitudinal coordinate. It should be noted that allowing for only these factors does not always permit generalization of the available empirical data.

At the same time, it is known that the body forces due to curvature of streamlines can also affect the turbulence characteristics of a flow. It follows from the theory of flows with curvilinear streamlines [4-6] that turbulence is suppressed in a boundary layer on a

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convex surface and intensified in a boundary layer on a concave surface. This has been proven to be the case in experiments conducted by different authors [7, 8].

To allow for the effect of body forces on turbulence in curvilinear flows, the authors of [9, 10] used a method based on the Monin–Obukhov analysis for atmospheric turbulence under conditions of density stratification. These studies propose that the effect of body forces on turbulence characteristics be accounted for through a certain modification of the expression for the mixing length $\ell = \ell_0(1 - \beta \mathrm{Ri})^{n_1}$. In this expression, the Richardson number characterizes the ratio of the body forces to frictional forces (in Bradshaw's terms, the ratio of the generation of turbulence energy by centrifugal forces to the generation of same by shear stresses). In accordance with Prandtl, $n_1 = 0.5$ and $\beta = 0.5$; in accordance with Bradshaw, $n_1 = 0.25$ and $\beta = 18$. According to the data of different authors [8-13], the empirical coefficient β varies within a fairly broad range.

Here we present some results of studies of turbulent heat and mass transfer in vortex flows. We will examine a fully-twisted flow in a cylindrical channel, a partially twisted flow (twisted gas screen), and flow in a swirl chamber.

1. Fully Twisted Flow. It follows from the above that the intensification of transport processes in the turbulent boundary layer of a twisted flow relative to an untwisted flow is due mainly to two factors:

1) a change in the magnitude and direction of velocity on the external boundary of the twisted flow compared to an axial flow (we will refer to this as the effect of twisting on friction and heat transfer through the mean characteristics of the flow);

2) the effect of body forces, resulting from curvature of the streamlines at the wall, on the level of turbulence.

The effect of these factors on transport processes can be analyzed fairly simply on the basis of the Kutateladze–Leont'ev asymptote theory for a turbulent boundary layer [14]. Shown below are expressions for turbulent flow and the heat flux along a streamline in accordance with the Prandtl hypothesis for a sufficiently thin ($\delta/R \ll 1$) three-dimensional boundary layer

$$\tau_\Sigma = \rho l^2 \left(\frac{\partial U_\Sigma}{\partial y}\right)^2, \tag{1}$$

$$\tau_x = \tau_\Sigma \cos\varphi_0 = \rho l^2 \left(\frac{\partial U_x}{\partial y}\right)^2 \frac{1}{\cos\varphi_0}, \tag{2}$$

$$q = \rho c_p l^2 \left(\frac{\partial U_\Sigma}{\partial y}\right)\left(\frac{\partial T}{\partial y}\right) = \rho c_p l^2 \left(\frac{\partial U_x}{\partial y}\right)\left(\frac{\partial T}{\partial y}\right)\frac{1}{\cos\varphi_0}. \tag{3}$$

In accordance with the asymptotic theory of a turbulent boundary layer, Eqs. (2) and (3) lead to the following relative limit law [15] for heat and mass transfer and friction in the case of a constant angle of twist of the flow across the boundary layer $\varphi = \mathrm{arctg}\,(U_\varphi/U_x) = \mathrm{const}$ (which follows from the experiment in [1]):

$$\Psi_{\mathrm{Re}^{**}\to\infty} = \frac{1}{\cos\varphi_0^{0.75}}\left[\int_0^1 \sqrt{\rho/\rho_0}\,(l/l_0)\,d\omega\right]^2. \tag{4}$$

Here, $\Psi = (c_f/c_{f_0})_{\mathrm{Re}^{**}} = (\mathrm{St}/\mathrm{St}_0)_{\mathrm{Re}^{**}}$ is the relative coefficient of friction or heat and mass transfer; $\mathrm{Re}^{**} = \rho_0 U_{x0}\delta^{**}/\mu_0$ is the Reynolds number calculated from the momentum thickness and longitudinal velocity on the external boundary of the boundary layer.

Equation (4) includes the ratio of the mixing ℓ in the twisted flow to ℓ_0 in an untwisted flow under standard conditions. If we assume that twisting of the flow has no effect on its turbulence characteristics $\ell = \ell_0$, then it follows from Eq. (4) at $\rho_0/\rho = \psi + (1 - \psi)\omega$ that:

$$\Psi_{\mathrm{Re}^{**}} = \Psi_\varphi \Psi_T = \frac{1}{\cos\varphi_0^{0.75}}\left(\frac{2}{\sqrt{\psi} + 1}\right)^2, \tag{5}$$

where $\psi = T_{st}/T_0$ is the temperature factor.

Thus, the effect of twisting on the coefficients of heat and mass transfer and friction, manifest through the average parameters of the flow, can be accounted for by means of the relative function $\Psi_{\varphi} = 1/\cos \varphi_0^{0.75}$. A similar result is obtained from analysis of the solutions of the integral relations for the moment and energy of a three-dimensional boundary layer [15].

A twisted flow in a pipe can be represented as flow along a curved surface with an equivalent radius of curvature $R_c = R/\sin^2\varphi_0$, where R is the radius of the pipe and φ_0 is the angle of twist of the flow at the wall. As indicated above, the body forces due to streamline curvature will also have an effect on the turbulence characteristics of the flow. To determine this effect, we used the Prandtl mixing-length model. Here, in place of the change in linear momentum, we examine the change in angular momentum in the transport of a mole of liquid in the radial direction [16]. Here, it was assumed that the centrifugal force field affects only the radial component of fluctuation velocity. The primary effect of body forces on transverse fluctuation velocity was demonstrated in experiments involving a curved boundary layer [7, 8] and twisted mixing layers [17]. The model which was developed – in contrast to the currently used Bradshaw method of modifying the mixing length – makes it possible to consider the effect of flow curvature without the use of additional constants: for flow on a concave surface

$$l/l_0 = \sqrt{f} = \sqrt[4]{1-(\xi/\bar{l}_0)^2 \mathrm{Ri}}\,, \tag{6}$$

while on a convex surface

$$l/l_0 = \sqrt{f} = 1/\sqrt[4]{1+(\xi/\bar{l}_0)^2 \mathrm{Ri}}\,. \tag{7}$$

The quantity $\bar{\ell}_0 = \bar{\ell}_0/\delta$ was determined from the familiar relation for standard conditions; f is a function which considers the effect of curvature on the turbulence characteristics of the flow.

Given a constant angle of twist over the thickness of the layer and similarity of the profiles of axial and circumferential velocity, the Richardson number reduces to the form [15]:

$$\mathrm{Ri} = \pm \frac{\delta}{R} \sin^2\varphi_0 \left(2\omega \frac{\partial\omega}{\partial\xi} + \frac{1}{\rho}\frac{\partial\rho}{\partial\xi}\omega^2\right)\Big/ (\partial\omega/\partial\xi)^2. \tag{8}$$

The velocity profiles are determined from the relation

$$\frac{\partial\omega}{\partial\xi} = \frac{\partial\omega_0}{\partial\xi}\left(\frac{\Psi\tilde{\tau}}{f\tilde{\tau}_0}\right)^{1/2}, \tag{9}$$

which follows from a comparison of Prandtl turbulent friction under the conditions examined here and under standard conditions. In this expression, $\omega_0 = \xi^n$ is the velocity profile in the boundary layer of the untwisted flow.

The relative coefficients of heat and mass transfer and friction were calculated from Eqs. (4)-(9) by the numerical method of successive approximations. As the first approximation in determining the Richardson number (8) and the relative function (4), we used the velocity profile without allowance for the effect of flow curvature. The results of numerical calculations performed in the range $0 < \delta^{**}/R < 0.025$ for twisted flow in a pipe (concave surface) are approximated well by the relation [15]

$$\Psi_{\mathrm{Re}^{**}} = \Psi_t \Psi_\phi \Psi_{cr} = \Psi_t \Psi_\phi \left\{1 + 1{,}8\cdot 10^3 \frac{\delta^{**}\sin^2\varphi_0}{R}\left[1 + \frac{\psi - 1}{2(\psi n + 1)}\right]\right\}^{0,162}. \tag{10}$$

In the first approximation here, we can take $n = n_0 = 1/7$. Figure 1 compares results alculated with this formula and experimental data in [18] on mass transfer. The experimental data was obtained in a burning graphite channel under nonisothermal conditions $\psi \approx 7$, with an angle of twist of the flow at the inlet $\varphi_0 \approx 32°$. The figure illustrates the effect of twisting on mass transfer due to the change in the magnitude and direction of velocity Ψ_ϕ and the body forces, which alter the turbulence characteristics Ψ_{cr}. It is evident that, for the conditions investigated, Ψ_ϕ is even lower than Ψ_{cr}. Under nonisothermal conditions, at $T_{st}/T_0 > 1$, the effect of body forces on mass transfer is greater than under isothermal conditions. In the former case, the contribution of these forces to the overall heat and

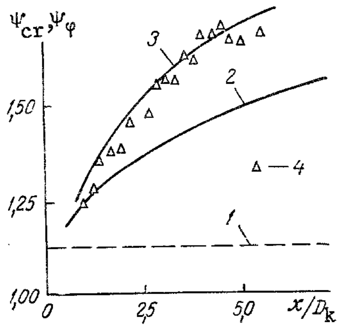

Fig. 1. Heat and mass transfer in a twisted flow (φ_0 = 32°): 1) effect of the three-dimensional character of the flow; 2, 3) effect of body forces at ψ = 1 and 7; 4) experiment, ψ = 7.

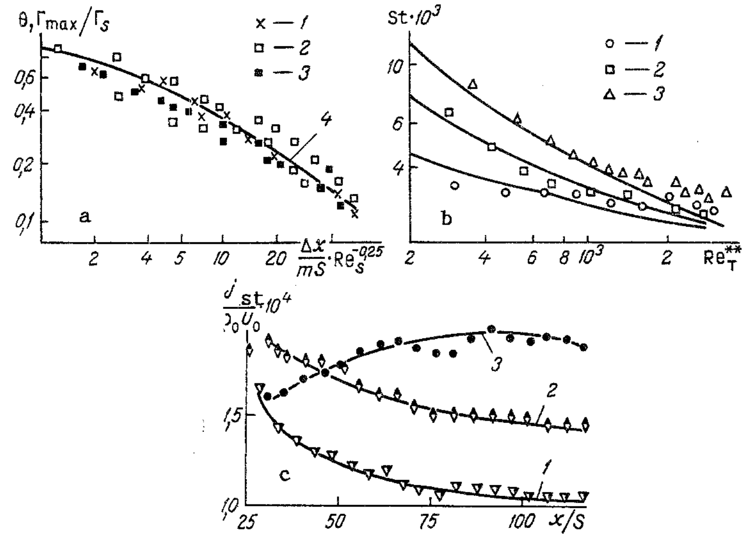

Fig. 2. Efficiency and heat and mass transfer in a twisted screen: a) efficiency and decay of the screen (1 – untwisted screen; 2, 3 – twisted screen θ and Γ_{max}/Γ_S; 4) calculation from Eq. (11)); b) heat transfer with twisting of the screen, $\psi \approx 1$, m = 0.5; experiments in [19] (1) φ_S = 0°C; 2) 58°; 3) 74°), lines – calculation; c) mass transfer in a channel with a twisted screen ψ = 7, φ_S = 58° (1) m = 0.2; 2) 0.5; 3) 0.9).

mass transfer coefficient increases along the channel because the parameter δ^{**}/R – characterizing the effect of the body forces on turbulence – also increases.

2. Partially Twisted Flows (Screens). By a partially-twisted flow, we mean a twisted jet which develops in a cylindrical channel in the presence of an untwisted co-current flow in the center of the channel. THe twisted wall jet is injected through a slit of the width S and has an initial angle of twist φ_S. The basic difference between this flow and a fully-twisted flow can be described as follows:

1) the angle of twist decreases rapidly along the channel due to mixing with the untwisted central flow;

2) the longitudinal component of velocity increases monotonically with increasing distance from the wall, and the profiles of tangential velocity and circulation $\Gamma = U_\varphi$ have a point of inflection.

It follows from analysis of the stability of rotational flows [5] that the condition of stability against unidimensional perturbations is the inequality $d\Gamma^2/dr > 0$. This was demonstrated by Rayleigh and Karman, and in the investigations of Chandrasecar [6] it was extended to rotational–translational motion with arbitrary profiles of longitudinal and tangential velocity. Thus, the boundary layer of a partially twisted flow contains a region in which turbulent transport is intensified, at $y < \delta_m$ (where $d\Gamma/dr < 0$), and another region at $y < \delta_m$, where turbulence is suppressed ($d\Gamma/dr > 0$). The magnitude of the effect of turbulence intensification and suppression on heat and mass transfer and friction in the boundary layer is determined from Eqs. (6) and (7). Here, the condition $d\Gamma/dr > 0$ corresponds to flow over a concave wall, while $d\Gamma/dr > 0$ corresponds to flow over a convex wall. The presence of the region of turbulence suppression in the external, jet part of the flow has a significant effect on the profile of longitudinal velocity, making it less full. Thus, for the conditions of the experiments conducted in [19], the exponent in the power velocity profile was equal to $n \approx 0.25$. Good agreement with the experimental results [15] was obtained from a theoretical analysis based on a model similar to that used in Part I to describe heat and mass transfer in boundary layers with longitudinal curvature.

In analyzing the thermally protective properties of gas screens, we will use the concept of its efficiency on an adiabatic wall $\theta = (T_0 - T^*)/T_0 - T_s)$. This quantity characterizes the rate of mixing of the injected gas with the main flow. It was originally assumed that the efficiency of a twisted screen would be considerably lower than that of an untwisted screen due to an intensification of transport processes. However, as shown by the tests in [19], in the case of a twisted screen, only the length of the initial section x_0 (where $\theta = 1$) is significantly reduced as a result of the intensification. On the main section of the flow, at $x > x_0$, turbulent transport is intensified only in the boundary region. In this region, $dT/dy \approx 0$ due to the adiabatic nature of the wall. Thus, there is almost no effect on thermal efficiency. Figure 2a generalizes experimental data from [19] on the thermal efficiency of twisted and untwisted screens. It is evident that, with allowance for the length of the initial section, the experimental results for the twisted screen can be generalized and are described by the well-known relation [14]

$$\theta = \left[1 + 0{,}25\,\frac{\Delta x}{mS}\,\mathrm{Re}_s^{-0,25}\right]^{-0,8}. \tag{11}$$

As already indicated above, an important characteristic of partially-twisted flows is the decay of maximum tangential velocity along the channel. This occurs mainly as a result of mixing of the injected gas with the main axial flow, and friction against the wall plays a much smaller role. To calculate heat and mass transfer in the boundary layer, we need to know the local values of velocity on the external boundary of the layer.

The integral relation of the boundary layer for angular momentum can be written in the form

$$\frac{d}{dx}\left[R\rho_0 U_{x0}\Gamma_{\max}\int\limits_0^{\delta}\frac{\rho U_x}{\rho_0 U_{x0}}\,\frac{\Gamma}{\Gamma_0}\left(1 - \frac{y}{R}\right)dy\right] = \tau_{\varphi_{st}} R^2. \tag{12}$$

In a flow in a pipe with $R \approx$ const, if we ignore friction against the wall in the tangential direction we can write Eq. (12) in a form similar to the equation for energy on an adiabatic surface [20]:

$$\frac{d}{dx}(\Gamma_{\max}\delta_\varphi^{**}) = 0. \tag{13}$$

It follows from the similarity condition that the decrease in circulation along the channel should obey the same law as the law for the decrease in the thermal efficiency of the screen: $\theta = \bar{\Gamma}_{max} = \Gamma_{max}/\Gamma_s$. Test data on the decay of twisting, shown in Fig. 2a, confirms this hypothesis.

Twisting of the screen has only a slight effect on its thermal efficiency on an adiabatic wall. Under nonadiabatic conditions, it intensifies heat and mass transfer at the wall (Fig. 2b). In this case, twisting may reduce the protective properties of the screen to zero.

Figure 2c shows test data on the rate of combustion of a graphite channel in an air flow with the injection of a twisted nitrogen screen through a tangential slit. It is evident

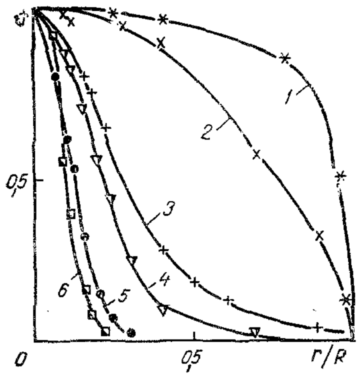

Fig. 3. Radial profiles of temperature in a swirl chamber G_0 = 1 g/sec; 1) G_k/G_0 = 0; 2) 1; 3) 2; 4) 3.5; 5) 6; 6) 15.

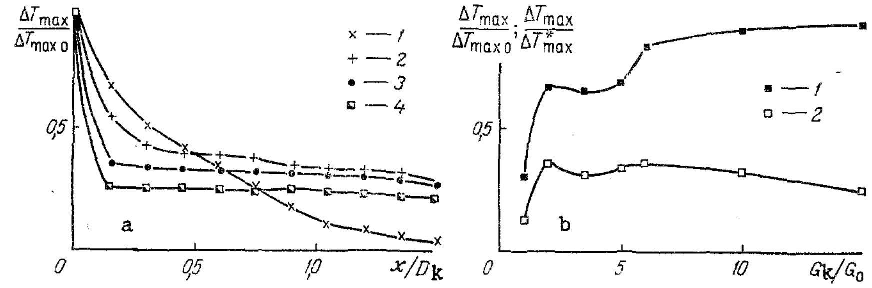

Fig. 4. Stabilization of an axial jet in a swirl chamber: a) distribution of the minimum temperature of the jet over the height of the chamber G_0 = 1 g/sec (1 – G_k/G_0 = 1; 2 – 2; 3 – 10; 4 – 15); b) effect of rotation rate on gas temperature on the axis, G_0 = 1 g/sec, x/D_k = 1 (1 – without mixing of the end layer; 2 – with allowance for the layer).

that an increase in injection of the inert gas leads to a deterioration in the protective properties of the screen. This can be attributed to the fact that an increase in injection (given the same initial angle of twist) is accompanied by an increase in the local angles of twist of the flow at the wall.

3. Swirl Chambers. The main difference betwen the flow in a swirl chamber and a partially twisted flow in a cylindrical channel is the reduction in the size of the outlet section. This change leads to fundamental restructuring of the flow. Here, an important role is played by the radial component of velocity, as well as by the boundary layers on the end surfaces. We will examine the results of studies of the thermal mixing of axial jets in a swirl chamber. A twisted flow of gas is fed along the periphery G_k, while an untwisted jet with the discharge G_0 is fed along the axis. The effects of stabilization in the axial region can be observed in certain flow regimes.

Figure 3 shows temperature profiles in a swirl chamber [21]. The profiles indicate that the axial jet stabilizes with an increase in the discharge of the peripheral twisted flow G_k. The tests were conducted under conditions close to isothermal (when the density gradient is negligible) in a swirl chamber of the diameter D_k = 100 mm and length L_k = 150 mm. The figure shows the distribution of dimensionless temperature $\vartheta = (T - T_{st})/(T_0 - T_{st})$ over the radius of the chamber. It is evident that stabilization of the axial jet occurs at values of the ratio of the discharges of the peripheral and axial flows $G_k/G_0 > 6$. In this case, there is almost no mixing of these two flows, and the walls are fully protected from the heat-conducting flow. Such localization of an axial jet was regarded in [22] as the result of laminarization of the flow under the influence of a gaseous vortex.

Figure 4 shows the effect of the discharge ratio G_k/G_0 on the character of mixing of the axial jet along the chamber; $T_{max} = (T_0 - T_{mi})/(T_0 - T_{mi})_{in}$ is the temperature associated with complete mixing. It is evident that with small peripheral discharges, temperatures on the axis decreases monotonically along the chamber as a result of turbulent mixing. In the case of large discharges $G_k/G_0 > 6$, temperature decreases sharply near the inlet section. This can be attributed to the effect of the end flow [23]. In the region outside the end layer, there is almost no change in temperature along the axis. This is evidence of the nearly complete absence of mixing.

The dependence of the maximum temperature on the axis T_{max} on the discharge ratio is complex (Fig. 4b). The clear points in this figure were obtained by analyzing data on the temperature of the jet at the chamber inlet. In the form presented in the figure, the experimental results characterize the mixing of the jet over the entire chamber — including its cooling as a result of dilution by the cold end layer. The relative temperature has a small maximum on the axis at a discharge ratio $G_k/G_0 \approx 6$.

The dark points (Fig. 4b) reflect the change in gas temperature on the jet axis without allowance for mixing of the end flow into the jet. It can be seen that the relative temperature in such a representation increases with an increase in G_k/G_0 throughout the investigated range. This is evidence of reinforcement of processes which lead to suppression of turbulent heat and mass transfer with an increase in body forces caused by rotation of the gas.

NOTATION

ℓ, ℓ_0, mixing length in the twisted and untwisted flows; β, empirical coefficient in the expression for mixing length; Ri, Richardson number; ρ, c_p, U, T, density, specific heat, velocity, and temperature of the gas; j_{st}, combustion rate; $\Psi = (c_f/c_{f0})_{Re^{**}} = (St/St_0)_{Re^{**}}$, function of friction and mass transfer; $Re^{**} = \rho_0 U_{x0} \delta^{**}/\mu_0$, Reynolds number constructed from the momentum thickness; Ψ_t, Ψ_φ, Ψ_{cr}, relative functions of friction and heat and mass transfer describing the effect of nonsiothermality, the increase in the velocity vector with twisting, and body forces on turbulence; $\psi = T_{st}/T_0$, temperature factor; $\xi = y/\delta$, $\ell_0 = \ell_0/\delta$, relative distance from the wall and the dimensionless mixing length; δ, thickness of the boundary layer; $\omega = U_x/U_{x0}$, dimensionless velocity in the boundary layer; R, channel radius; R_k, radius of curvature of the streamline; $\varphi = U_\varphi/U_x$, angle of twist of the flow; $\Gamma = U_\varphi \cdot r$, circulation; $m = \rho_s U_s/\rho_0 U_0$, injection parameter; $\theta = (T - T_w^*)/(T_0 - T_s)$, screen efficiency; T_0, T_s, T_w^*, temperature in the core of the flow, in the slit, and on the surface of an adiabatic wall; δ_{max}, thickness of the boundary part in the circulation profile; x_0, length of the initial section where $\theta = 1$; $Re_s = \rho_s U_{xs} S/\mu_s$, Reynolds number from the parameters in the slit; G_k and G_0, gas discharge at the periphery and into the axial jet; $T_{mi} = (T_0 G_0 + T_k G_k)/(G_0 + G_k)$, temperature of complete mixing in the chamber; T_0 and T_k, temperature of the jet and the peripheral flow at the chamber inlet; L_k, D_k, length and diameter of the swirl chamber. Indices: 0, parameters in the core, standard conditions (untwisted flow); w, s, parameters on the wall and in the slit; x, r, φ, Σ, longitudinal, radial, circumferential, and resultant components; max, maximum value; mi, parameters associated with complete mixing; in, conditions at the inlet; k, periphery.

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EXTERNAL HEAT TRANSFER IN INFILTRATED GRANULAR BEDS. EXPERIMENTAL STUDY

V. A. Borodulya, Yu. S. Teplitskii, Yu. G. Epanov, and I. I. Markevich

UDC 66.096.5

Experimental data was obtained on external heat transfer in infiltrated granular beds of coarse particles. It was determined that calcualtions with the proposed model agree satisfactorily with the experimental results.

A two-phase model was formulated in [1] for heat transfer in stationary granular beds injected with gas. The model was ued to obtain simple expressions for the coefficient of heat transfer in the case of two different surfaces:

a) plane surface

$$\alpha = \frac{\lambda_f}{R} \frac{\lambda s + f^2\xi_0}{1 + \lambda s\xi_0}, \tag{1}$$

b) cylindrical surface

$$\alpha = \frac{\lambda_f}{R} \frac{\lambda s + f^2\xi_0/K^*}{1 + \lambda s\xi_0 K^*} K^*. \tag{2}$$

Equations (1) and (2) are suitable for calculations of α at Pe > 500 and 100, respectively. To check the validity of the above relations, we used experimental data obtained from the literature and our own tests.

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