EFFECT OF INITIAL TURBULENCE ON THE EFFICIENCY OF COOLING A PERMEABLE WALL

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The effect of turbulence in the outer stream (ε = 1.7, 5.3, 9.3%) on convective heat transfer is analyzed in the case of flow around impermeable and permeable plates.

In most studies made for the determination of equilibrium temperatures and heat-transfer coefficients under conditions of streamlining flow around permeable walls, the experiments were performed with a relatively low initial intensity of turbulence in the oncoming gas stream: ε = 1-3% [1-5]. In many engineering devices the intensity of turbulence in the gas stream can be much higher. Available published data [6-10] do not provide a sufficient basis, however, for drawing definitive conclusions as to how the magnitude of the turbulence parameter ε affects convective heat transfer between a turbulent gas stream and a wall.

The main object of this study was to obtain comparative data on the magnitudes of heat-transfer coefficients in the case of streamlining air flow with various intensities of turbulence around various plates.

The plates in this study were heated from an independent infrared radiation source, while the temperatures of both the boundary stream and the main stream of air were maintained at the same level ($T_1 = T_0$). In this case, with constant intensities of the radiant flux and constant averaged parameters of both streams, the ratio of excess temperatures $\Delta T_W/\Delta T_{W_0} = (T_W - T_0)/(T_{W_0} - T_0)$ (with negligible relative heat losses) was equal to the ratio α_0/α of heat-transfer coefficients, ΔT_{W_0} and α_0 denoting, respectively, the excess temperature and the heat-transfer coefficient at a low initial turbulence level.

The overall scheme of the test apparatus is shown in Fig. 1. Test plates, forming the upper wall of our model, were placed in a uniform air stream discharged from a rectangular nozzle with a 120 × 100 mm orifice. The geometrical dimensions of this nozzle and of the test plates had been selected so as to ensure retention of a stream core with uniform velocities and temperatures at the end of the plate.

The cooling air was supplied through the lower model wall. For jet cooling (Fig. 1a) there was a clearance set between the test plate and the baffle of the nose cone at the entrance. The baffle edge had been machined to a thickness of 0.5 mm. During cooling of a perforated plate the coolant entered through the holes in the plate (Fig. 1b). The area of the perforated part of the plate was 120 × 180 mm, with 4 holes 1.2-0.85 mm in diameter per 1 cm^2 of plate surface (permeability $\bar{F}$ = 3%) and with the plate 1.5 mm thick. The plate temperature was measured with a model ÉPP-09 potentiometer of class 0.5 accuracy.

The test models were positioned in the sweeping air stream so as to minimize the effect of prior flow conditions. The thickness of the boundary layer at the front section of the plate was ~1 mm.

The independent source of radiant energy (radiator) was placed above the test models. As radiating elements served KG-220-1000 quartz lamps. The electric power W drawn by these lamps was varied with a regulating transformer and measured with an ammeter and a voltmeter.

The outside surface of the model walls had been coated black with printer's ink, while the outside surface of the nose cone and its baffle (Fig. 1) had been covered with aluminum foil so as to reduce heating of these surfaces by radiation. As a result, a thermal boundary layer formed in this experiment almost at the front end of the test plates.

The test models and the geometrical characteristics of a permeable plate made of grade

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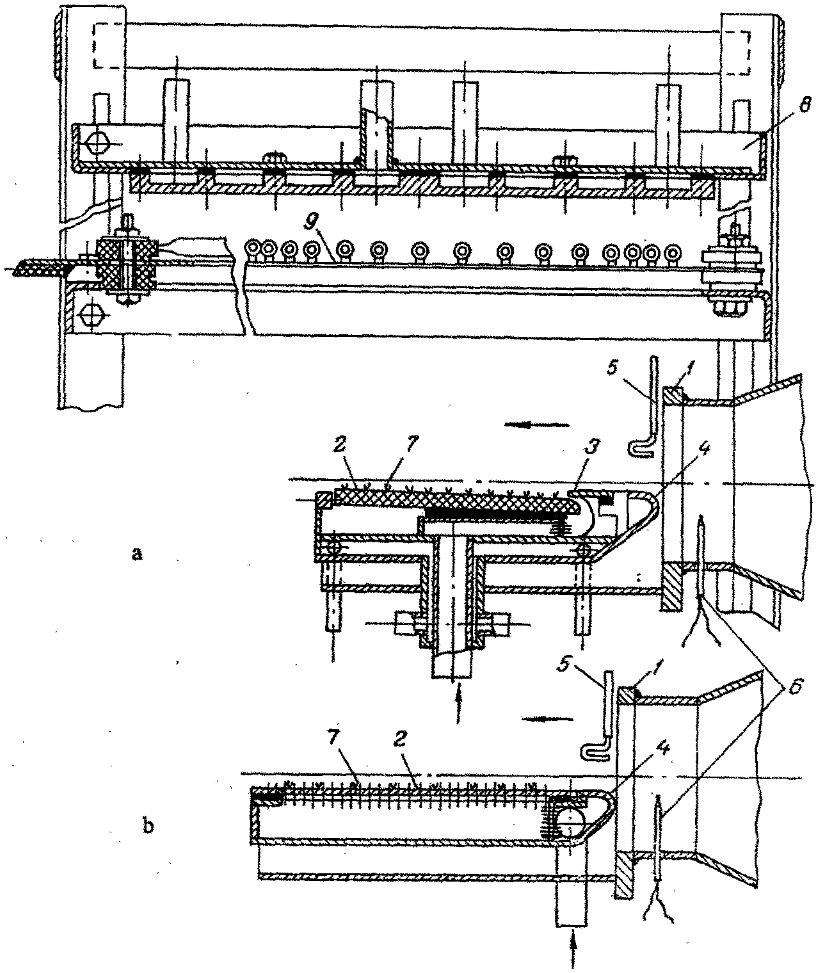

Fig. 1. Schematic diagram of the apparatus: 1) nozzle; 2) test plate; 3) baffle; 4) nose cone; 5) Pitot tube; 6) thermocouple; 7) thermocouples for measuring the wall temperature; 8) independent radiator; 9) assembly with an array of quartz lamps.

TABLE 1. Intensity of Turbulence along the Stream

x', nm	ε, %		
	without a mesh	mesh	
		$\bar{F}$=50%	$\bar{F}$=20%
40	1,65	15,3	36,5
50	1,65	10,0	21,4
60	1,70	7,5	9,4
70	1,70	6,2	11,0
80	1,70	5,3	9,3
90	1,80	4,8	8,2

ÉYaIT steel have been described more thoroughly in another report [11]. The other plates were made of Textolite and 10 mm thick so as to reduce the heat overflow along their surfaces.

Different turbulence levels in the main air stream were attained by placing meshes with holes 10 mm in diameter and a permeability $\bar{F}$ = 50 or 20% in the nozzle outlet.

The intensity of turbulence was measured with a thermoanemometer. At the same time, the mean velocity of the stream was also determined. Measurements were made along the stream at a distance y = 10 mm from the test plate (beyond the boundary layer) and at a distance x' = 80 mm from the nozzle, in the directions normal and transverse with respect to the test plate. According to the experimental evidence, placing the test model in the air stream did

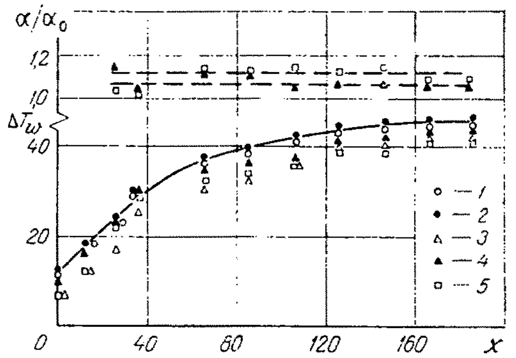

Fig. 2

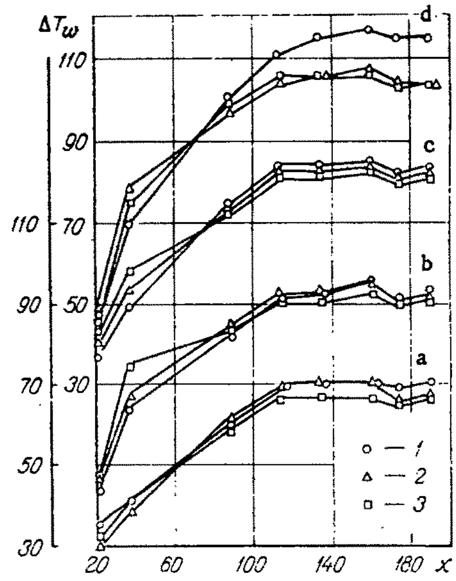

Fig. 3

Fig. 2. Variation of the excess temperature ΔT_w, °K, at the wall and of the relative heat-transfer coefficient α/α_0 along a smooth plate (x, mm), for $T_0 = T_1 = 293$°K, W = 6.5 kW, and $(\rho u)_0 = 147$ kg/m^2·sec: 1, 2) tests performed with $(\rho u)_0 = 187$ and 110 kg/m^2·sec without a mesh; 3, 4) $(\rho u)_0 = 176$ and 92 kg/m^2·sec with an $\overline{F} = 50\%$ mesh; 5) $(\rho u)_0 = 147$ kg/m^2·sec with an $\overline{F} = 20\%$ mesh.

Fig. 3. Variation of the excess temperature ΔT_w, °K, at the wall along a perforated plate (x, mm), for $T_0 = T_1 = 290$°K, $(\rho u)_0 = 60$ kg/m^2·sec, W = 11.2 kW, and $(\rho u)_w/(\rho u)_0$: (a) 0.0246, (b) 0.007, (c) 0.0117, (d) 0.0035; 1) without a mesh; 2) with an $\overline{F} = 50\%$ mesh; 3) with an $\overline{F} = 20\%$ mesh.

hardly affect the results of measurements and a core 50 × 40 mm in cross section with ε = const as well as u = const had formed in the stream at a distance x' = 80 mm. The magnitude of the turbulence parameter ε tapered appreciably downstream, decreasing with increasing distance from the mesh. In the absence of a mesh, on the other hand, it remained uniform (ε = const). The readings of the intensity of turbulence taken along the stream are presented in Table 1. In discussing the results of this study, we will henceforth tentatively characterize the intensity of turbulence in the air stream by the value of parameter recorded in the section x' = 80 mm. These values of ε were 9.3, 5.3, and 1.7%, respectively, with an $\overline{F} = 20\%$ mesh, with an $\overline{F} = 50\%$ mesh, and without a mesh. By the way, section x' = 80 mm coincided with section x = 0 where the coolant entered for jet cooling (Fig. 1a) and was about 40 mm away (x = 40 mm) from where the coolant began to enter for cooling a perforated plate (Fig. 1b).

Let us now proceed to analyze the effect of initial turbulence in a gas stream on the heat-transfer coefficients.

We first consider streamlining flow around a smooth plate. Excess temperatures ΔT_w at the wall along a plate, with and without turbulizing meshes standing in the stream, have been plotted in Fig. 2. These tests were performed at temperatures $T_0 = T_1 = 293$°K and with an incident radiant flux of q = 13.4 kW/m^2 density (W = 6.5 kW). The mass flow rate of the sweeping stream was $(\rho u)_0 = 147$ kg/m^2·sec. On the same diagram have also been plotted values of the relative heat-transfer coefficient $\alpha/\alpha_0 = \Delta T_{w0}/\Delta T_w$. Here α_0 and ΔT_{w0} correspond to conditions without turbulizing meshes. It is to be noted that for ε = 1.7% the values of ΔT_w correspond to $(\rho u)_0 = 187$ and 110 kg/m^2·sec, while for ε = 5.3% the values of ΔT_w correspond to $(\rho u)_0 = 176$ and 92 kg/m^2·sec. The ΔT_w values in Fig. 2 have been referred to $(\rho u)_0 = 147$ kg/m^2·sec. A proportionality of ΔT_w to $(\rho u)_0^{-0.8}$ was assumed for this conversion. An increase of ε is found to cause an insignificant decrease of ΔT_w, resulting in some increase of the relative heat-transfer coefficient. Over the segment x = 20-150 mm a change of ε from 1.7 to 5.3% is followed by an ≈7% increase of α/α_0 and a change of from 1.7 to

TABLE 2. Mean Values of the Excess Temperature and of the Relative Heat-Transfer Coefficient at a Perforated Plate with Cooling

$\frac{(\rho u)_w}{(\rho u)_e} Re_0^{0,2}$	Without a mesh ΔT_W, °K	Mesh $\bar{F}=50\%$ ΔT_w, °K	Mesh $\bar{F}=50\%$ α_e/α_{e_0}	Mesh $\bar{F}=20\%$ ΔT_w, °K	Mesh $\bar{F}=20\%$ α_e/α_{e_0}
0,04	113	104	1,085	104	1,085
0,077	92	90	1,025	90	1,025
0,13	82,5	80	1,03	81,5	1,012
0,28	69	66	1,045	68	1,015

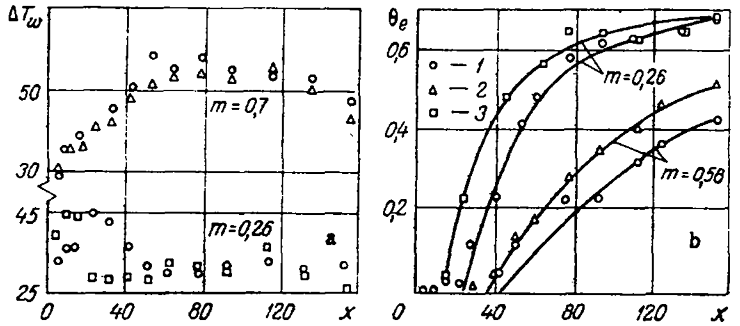

Fig. 4. Variation of the excess temperature ΔT_W, °K, at the wall and of the equilibrium temperature θ_e along a plate with jet cooling and an h = 3.8-mm-wide clearance: 1) without a mesh; 2) with an $\bar{F}$ = 50% mesh; 3) with an $\bar{F}$ = 20% mesh; (a) $T_0 = T_1$ = 290°K, $(\rho u)_0$ = 60-150 kg/m²·sec, and W = 6.5 kW; (b) T_0 = 360°K, T_1 = 290°K, $(\rho u)_0$ = 58 kg/m²·sec, and W = 0.

9.3% is followed by an ≈13% increase of α/α_0. At distances x < 20 cm the effect of changes in ε is stronger, but the accuracy of α/α_0 determinations within this range is too low for establishing a functional relation between both parameters.

The test results describing the effect of initial turbulence on the cooling of a perforated plate are shown in Fig. 3. These tests were performed with $(\rho u)_0$ = 60 kg/m²·sec, $T_0 = T_1$ = 290°K, q = 25.3 kW/m² (W = 11.2 kW), and various flow rates of cooling air. It is evident here that changes in ε have a small effect on ΔT_W and thus also on the heat-transfer coefficients. Mean over the x = 120-180-mm segment values of ΔT_W and α_e/α_{e_0} are given in Table 2 for various values of the injection parameter $(\rho u)_W Re_{,0}^{0.2}/(\rho u)_0$, with $(\rho u)_W$ denoting the flow rate of coolant per unit area of cooled surface and $N_{Re,0} = (\rho u)_0 x/\mu_0$. The value of α_e/α_{e_0} is seen to decrease somewhat with increasing injection, the maximum value of α_e/α_{e_0} never exceeding 1.1.

The data in Figs. 2 and 3 indicate that the excess temperature of plates rises with increasing distance and, moreover, the fastest change of ΔT_W occurs at the front end of a plate. This is indirect evidence that the thermal boundary layer forms almost at the front end of a plate.

Let us next analyze the data on the effect of initial turbulence on the parameters at equilibrium during jet cooling.

The readings of excess temperatures at the wall, with an h = 1-mm-wide clearance, indicate that changes in ε have almost no effect on the heat-transfer coefficient. These tests were performed over a wide range of ratio m ≤ 3.6, ratio of air velocity in the clearance to air velocity in the sweeping stream.

Changes in the intensity of turbulence had a stronger effect in tests with an h = 3.8-mm-wide clearance. For illustration, excess temperatures ΔT_W along a plate are shown in Fig. 4a for m = 0.26 and 0.7. These tests were performed at $T_0 = T_1$ = 290°K with q = 13.4 kW/m² (W = 6.5 kW). An analysis of these data had revealed a wide spread of ΔT_W values, depending on the value of ε, over a relatively short segment of the plate: 20 ≤ x ≤ 45 mm for

$m = 0.26$ and $50 \leq x \leq 75$ mm for $m = 0.7$. Moreover, $m = 0.26$ $(\Delta T_{w_0}/\Delta T_w)_{max} \approx 1.7$ in the first case and $m = 0.7$ $(\Delta T_{w_0}/\Delta T_w)_{max} = 1.2$ in the second case. For explaining the causes of such a change in $\Delta T_{w_0}/\Delta T_w = \alpha_e/\alpha_{e_0}$, additional comparative tests were performed and the excess temperature at the wall was measured when $q = 0$ and $T_0 > T_1$.

We note that the wall temperature is equal to the temperature of the gas mixture when no heat transfer occurs between the wall and the ambient medium and when no heat overflow occurs along the wall. In these tests the measured temperatures of a plate were near the equilibrium temperatures, except over short wall segments with large longitudinal temperature gradients.

The results of tests with $m = 0.2$ and 0.58 ($T_0 = 360°K$, $T_1 = 290°K$, and $u_0 = 60$ m/sec) are shown in Fig. 4b in terms of the parameter $\theta_e = (T_w - T_1)/(T_0 - T_1)$.

It is evident here that, as the intensity of turbulence ε becomes higher, the length of the initial jet segment ($\theta_e = 0$) becomes shorter and the wall temperature begins to rise nearer to the clearance section. In the case of $m = 0.2$ the values of parameter θ_e with an $\overline{F} = 20\%$ mesh are analogous to those without a mesh, if the latter values are referred to a section shifted toward the clearance section through a distance equal to the decrease in the length of the initial segment. This conclusion, based on the pertinent tests, remains valid up to $m \approx 0.4$. When $m > 0.4$, then this conclusion does not agree so well with experimental data (as is evident in Fig. 4b for $m = 0.58$) and additional studies are needed for establishing the effect of changes in ε on the value of parameter θ_e.

A comparison of data in Fig. 4a and b suggests that the intensity of turbulence affects the heat-transfer coefficient and the equilibrium temperature in the vicinity of transition from the initial segment of the boundary jet to the core of the jet. The earlier mentioned large change in α_e/α_{e_0} at low values of ratio m is due to a change in the mass flow rate of the gas mixture near a plate.

We may conclude, on the basis of this study, that initial turbulence has a relatively small effect on convective heat transfer during streamlining flow around smooth or perforated plates.

The magnitude of initial turbulence can largely influence the magnitude of the heat-transfer coefficient near the beginning of boundary-layer formation, and also the length of the initial flow segment in the case of jet cooling.

NOTATION

x', x, longitudinal coordinates measured from the nozzle throat and from the coolant entrance, respectively; h, clearance width; u, stream velocity; T, temperature; ρ, density; G, flow rate; α, heat-transfer coefficient; q, radiant energy flux; μ, dynamic viscosity; W, electric power supplied to the radiator; $\Delta T_w = T_w - T_0$, excess temperature at the wall; ε, intensity of turbulence; $\theta_e = (T_w - T_1)/(T_0 - T_1)$, referred excess temperature; $\overline{F}$, wall (plate) permeability; m, ratio of air velocity in the clearance to velocity of the oncoming stream; $N_{Re,0} = (\rho u)_0 x/\mu$, Reynolds number; subscripts: w, wall; 0, oncoming stream or to a low turbulence level; 1, boundary stream (layer); and e, equilibrium conditions.

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CLOSED EQUATION FOR THE PROBABILITY DISTRIBUTION OF VELOCITY AND TEMPERATURE DIFFERENCES BETWEEN TWO POINTS IN AN ISOTROPICALLY TURBULENT STREAM OF AN INCOMPRESSIBLE FLUID

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A closed equation is derived for the characteristic function describing the joint probability distribution of velocity and temperature differences between two points in an isotropically turbulent stream of an incompressible fluid.

Almost all theories of isotropic turbulence are based either on the formalism of moment equations in any form whatever or on the equation for the characteristic functional. The outcomes of these theories, viz., the closed systems of equations each yields for the spectral energy, are quite similar [1]. An analysis of these equations reveals that they are all incompatible with the mechanism of stagewise energy transfer over the spectrum of fluctuations characterized by successively different scales. The expression for the spectral energy within the inertial range, which can be derived from these equations, contains the mean-square energy of the velocity field. At very high values of the Reynolds number the discrepancy between theory and experiment widens without bounds. This deficiency is overcome by using certain procedures which ensure correct solutions for the inertial range of scales. It is quite doubtful, however, whether the equations thus derived remain valid over the entire universal range. Under consideration here are approximations of the Kraichnan "trial field" kind [2].

The lack of decisive progress made in deriving an equation for the simplest two-point characteristic of isotropic turbulence along conventional lines suggests that new approaches to the problem may have to be tried. One of such approaches could be the formalism based largely on the equations for finite-dimensional functions describing the probability distributions of turbulence fields (F-DFPD formalism, for short). Some developments have already been made so far with regard to this formalism: an array of F-DFPD equations has been derived [3-6], several methods of deriving closed systems of F-DFPD equations have been proposed [7-13], and attempts have been made to analyze the equations for a two-point distribution function covering the inertial scale range, whereupon the corresponding approximate equations have been found to be compatible with the Kolmogorov–Obukhov law [9,11,13]. Closed F-DFPD equations have, furthermore, been derived [14-16] on the basis of a semiempirical theory. The feasibility of developing an analytical theory on the basis of F-DFPD equations has not yet been established, although the results of some studies in this direction [9,11,17,18] are promising.

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