EFFECT OF THE TRANSVERSE MASS FLOW ON THE HEAT EXCHANGE AND DYNAMICS OF A STREAM IN TURBULENT HEATED AIR FLOW IN AN AXISYMMETRIC DIFFUSOR WITH A PERMEABLE WALL

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Results are presented of an experimental investigation of the heat exchange and dynamics of a turbulent flow in an axisymmetric permeable diffusor with uniform blowing of coolant through the wall.

In order to design porous cooling correctly, it is necessary to have reliable methods of determining the heat fluxes on the streamlined surface as mass is supplied to the gas stream. The influence of coolant injection on the heat exchange between the main gas stream and the wall depends to a considerable extent on what kind of boundary layer (laminar or turbulent) is formed on the wall surface. The case of gas injection into a laminar boundary layer on a wall for a gradient flow of the main gas stream has been examined in a number of papers [1-4]. The results of these papers show that the pressure gradient exerts substantial influence on the coolant discharge in order to maintain a given wall temperature for definite main stream parameters.

For negative pressure gradients the heat flux at the wall increases almost twice as compared with a flat plate, while for small positive pressure gradients it diminishes 25%.

The influence of the pressure gradient on heat exchange during coolant injection in a turbulent boundary layer has been studied slightly up to now. The results of experimental investigations of the influence of injecting different gases through a porous wall into a turbulent boundary layer on the heat exchange and friction are presented in [5, 6] for a gradient-less and a gradient hot air flow on a flat plate. It is shown that within the limits of variation of the pressure gradient which occurs in tests, its effect on the heat exchange is only slight when a definite gas is injected. As for gradient-less gas flows, the heat exchange on a porous surface depends only in the blowing intensity and the physical parameters of the gas being injected. No experimental work has been published on investigating the heat exchange and dynamics of a stream in a turbulent hot air flow in axisymmetric diffusors with coolant injected through a porous wall.

We conducted an experimental investigation of the heat exchange and dynamics of the turbulent flow in an axisymmetric permeable diffusor with uniform injection of coolant through the wall. The experimental

TABLE 1. Test Data on the Influence of Injection on Re_x for x/d = 2.28

Re_0	$\bar{m}$	Re_x	Re_0	$\bar{m}$	Re_x
$0{,}98 \cdot 10^5$	0	$1{,}95 \cdot 10^5$	$1 \cdot 10^5$	0,0140	$3{,}24 \cdot 10^5$
$1{,}03 \cdot 10^5$	0,00057	$2{,}1 \cdot 10^5$	$4{,}02 \cdot 10^5$	0	$9{,}11 \cdot 10^5$
$0{,}99 \cdot 10^5$	0,00124	$2{,}16 \cdot 10^5$	$4{,}06 \cdot 10^5$	0,00058	$9{,}2 \cdot 10^5$
$1{,}01 \cdot 10^5$	0,00248	$2{,}24 \cdot 10^5$	$4{,}03 \cdot 10^5$	0,0012	$9{,}24 \cdot 10^5$
$0{,}99 \cdot 10^5$	0,00490	$2{,}55 \cdot 10^5$	$4{,}17 \cdot 10^5$	0,00241	$9{,}85 \cdot 10^5$
$1{,}02 \cdot 10^5$	0,00918	$2{,}94 \cdot 10^5$	$4{,}12 \cdot 10^5$	0,00478	$10{,}4 \cdot 10^5$

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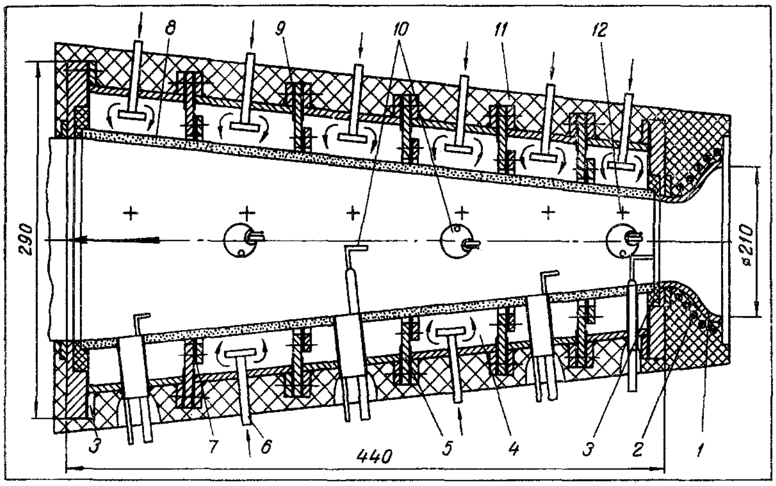

Fig. 1. Diagram of the experimental section: 1) Vitoshinskii nozzle; 2) compensating electrical heater; 3) Asbotextolite bottoms; 4) injection chamber; 5) steel baffle; 6) union; 7) compression ring of heatproof rubber; 8) porous diffusor; 9) paronite spacer; 10) moving probe; 11) heat insulation; 12) site of built-in thermocouple.

apparatus is an open, continuous action wind tunnel consisting of a B-6/800 air blower, a 75 kW power electrical heater, systems of reducers and equilibrating cascades, an experimental section (Fig. 1), and two VVN-3 vacuum pumps operating in the blower mode for injection. The experimental section is an axisymmetric diffusor with a porous wall whose porosity is 40% and the coefficient of heat conduction is 38.3 W/m · deg. The diffusor is enclosed in a steel shell producing the volume for the injection which is separated into six insulated chambers along the length. Coolant is injected into these chambers and its discharge is determined by a calibrated and volume method by measuring diaphragms. The diffusor length is 400 mm, and its entrance and exit diameters are 65 and 147 mm, respectively. Its aperture is 12° and the wall thickness is 7.5 mm. Sintering the diffusor by the method of free filling assured its uniform permeability.

The main stream was turbulent at the entrance. The turbulence at the entrance to the section was measured by a UTA-5B type thermoanemometer. The degree of turbulence of the longitudinal velocity in the boundary layer was 0.5-0.6% ahead of the entrance edge of the diffusor. At 2 mm downstream from the entrance edge it was approximately 1%. This indicated the absence of a transition flow mode in the boundary layer and the presence of a turbulent boundary layer in the beginning of the flow into the diffusor [7]. The distribution of longitudinal velocity pulsations in the diffusor was measured in seven sections downstream in an isothermal flow. The test data show that the longitudinal pulsations increase intensively downstream, which agrees with test results in [8]. The static pressure distribution was measured in twelve sections in the tests, the dynamic pressure and temperature of the gas in seven sections along the diffusor length, the temperatures of the inner and outer surfaces of the porous wall in seven sections, the coolant at the chamber, the inner and outer surfaces of the steel shell, the outer surface of the insulation, and the outer surface of the nozzle. The thickness of the dynamic boundary layer at the entrance did not exceed 1.4 mm. The thermal boundary layer was developed from the entrance edge, which was assured by heating the nozzle wall to the temperature in the core of the stream at the entrance into the section. The velocity distribution at the entrance was uniform. The discharge of the main stream was determined by means of the velocity profile at the entrance to the diffusor, and the thermocouple readings were recorded by a portable PP-63 type potentiometer. A Chromel and Copel wire of 0.2 mm diameter insulated by fiberglass was used as thermocouple.

The velocity and temperature fields were determined by specially fabricated moving combination probes which permitted taking readings across the boundary layer to 0.05 mm accuracy at 0.25 mm from the wall in two mutually perpendicular planes along the diffusor length. The range $Re_0 = 0.5 \cdot 10^5–4 \cdot 10^5$ was encompassed by the tests. The main stream temperature varied between 348 and 423°K. Injection was realized according to the law $\bar{m} = (\rho_w v_w)/\rho_1 u_1 = \mathrm{const}$. The intensity of injection $\bar{m}$ varied between 0.00058 and 0.292, the diffusor wall temperature between 293.4 and 417°K, and the free stream velocity at the diffusor

TABLE 2. Test Data on the Influence of Injection on Heat Exchange

Re_0	$\bar{m}$	Coefficient of heat exchange, W/m²·deg					
		$\frac{x}{d}=0{,}42$	$\frac{x}{d}=0{,}95$	$\frac{x}{d}=1{,}43$	$\frac{x}{d}=1{,}89$	$\frac{x}{d}=2{,}28$	$\frac{x}{d}=2{,}57$
$2{,}02\cdot10^5$	0,00058	166,5	111,5	77,2	62,2	51,2	45,9
$1{,}99\cdot10^5$	0,00123	129,0	78,5	49,8	43,1	38,0	31,6
$2{,}04\cdot10^5$	0,00245	85,1	60,5	45,4	37,0	29,8	26,2
$2{,}02\cdot10^5$	0,00480	60,7	39,6	29,3	24,7	20,6	18,0
$2{,}04\cdot10^5$	0,0072	54,1	36,4	25,5	20,3	17,1	13,8

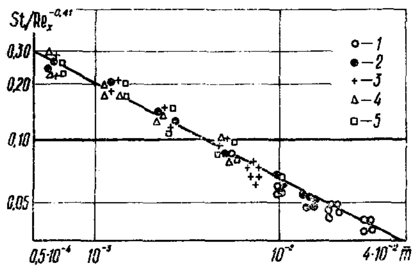

Fig. 2. Dependence of $St/Re_x^{-0.41}$ on $\bar{m}$ in dimensionless form: 1) $Re_0 = 0.5\cdot10^5$; 2) 10^5; 3) $2\cdot10^5$; 4) $3\cdot10^5$; 5) $4\cdot10^5$.

entrance between 19 and 118 m/s. The separation of the stream was at 220–240 mm from the entrance edge, was nonsymmetric [9], and displaced slightly upstream under the injection intensity occurring in the tests, although the velocity and temperature profiles were deformed considerably.

The density of the heat flux entering the wall in this section was determined from the heat balance equation

$$q_w = \frac{G}{F} c_p (T_w - T_{cool}) + q_r + q_x,$$

$$q_r = 5.76\ \varepsilon_{reduced} \left[\left(\frac{T'}{100}\right)^4 - \left(\frac{T''}{100}\right)^4\right],$$

$$q_x = -\lambda\delta \frac{d^2t}{dx^2},$$

where q_r is the radiant heat flux from the diffusor wall in W/m² and q_x is the longitudinal heat overflow in W/m².

In the case under consideration $\varepsilon_{reduced}$ is expressed by the equation

$$\varepsilon_{reduced} = \frac{1}{\frac{1}{\varepsilon_1} + \left(\frac{1}{\varepsilon_2} - 1\right)\cdot\frac{F_2}{F_1}}.$$

Here ε_1 and ε_2 are the emissivities of the porous diffusor surface and the inner surface of the steel shell; F_1 and F_2 are the porous diffusor and steel shell surfaces, respectively, which take part in the mutual irradiation in the considered injection chamber.

The local Stanton number

$$St = \frac{q_w}{c_{p1}\rho_1\ u_1(T_1 - T_w)},$$

was computed by means of a definite heat flux q_w. The experimental results on the stream heat exchange and dynamics were generalized as the dependences:

$$St = f(Re_x, \bar{m}); \tag{1}$$

$$St = f\left(Re_0, \bar{m}, \frac{x}{d}\right); \tag{2}$$

$$Re_x = f\left(Re_0, \bar{m}, \frac{x}{d}\right); \tag{3}$$

$$Eu = f\left(Re_0, \bar{m}, \frac{x}{d}\right). \tag{4}$$

The test data were processed using least squares. The dependence 1 in Fig.2 is described by the power-law equation

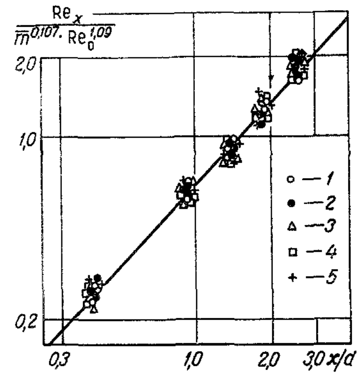

Fig. 3

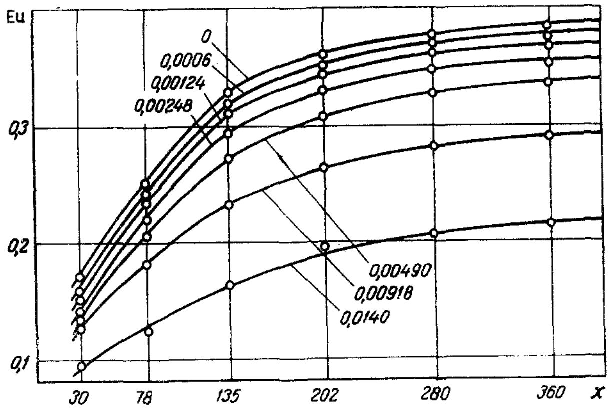

Fig. 4

Fig. 3. Dependence of $Re_x/(\overline{m}^{0.107} Re_0^{1.09})$ on x/d in dimensionless form: 1) $Re_0 = 0.5 \cdot 10^5$; 2) 10^5; 3) $2 \cdot 10^5$; 4) $3 \cdot 10^5$; 5) $4 \cdot 10^5$.

Fig. 4. Change in the Euler number ($Eu = \Delta p/\rho_0 u_0^2$) along the diffusor length as a function of the injection intensity; for $\overline{m} = (\rho_w v_w)/\rho_1 u_1$ = const $Re_0 = 10^5$. The numbers on the curves are values of $\overline{m}$, and x is in mm.

$$St = 0.006\ Re_x^{-0.41}\ \overline{m}^{-0.52}. \tag{5}$$

The dependence 3 in Fig. 3 is also described by a power-law equation

$$Re_x = 0.645\ Re_0^{1.09}\ \overline{m}^{0.107} \left(\frac{x}{d}\right)^{1.1}. \tag{6}$$

Substituting (6) into (5) and manipulating as necessary, we obtain a power-law equation for the dependence (2)

$$St = 0.00736\ Re_0^{-0.45}\ \overline{m}^{-0.565} \left(\frac{x}{d}\right)^{-0.45}. \tag{7}$$

The power-law equation for the dependence (4) is

$$Eu = 0.29\ Re_0^{-0.074}\ \overline{m}^{-0.088} \left(\frac{x}{d}\right)^{0.595}. \tag{8}$$

It is shown in Fig. 4 how the Euler number varies along the diffusor length as a function of the injection intensity for $Re_0 = 1 \cdot 10^5$. As the injection increases, the Euler number diminishes noticeably since the drop in static pressure on the channel wall diminishes. The change in Re described by (6) for identical $\overline{m}$ and x/d is more significant for smaller Re_0; this means that the injection exerts somewhat less influence on the flow dynamics as the free stream Re_0 increases. This is illustrated by the test data in Table 1. Test data showing the change in the local heat-exchange coefficient along the diffusor length and as a function of injection for Re_0 close to $2 \cdot 10^5$ are presented in Table 2. It is seen that a 12.4-fold increase in $\overline{m}$ implies a threefold diminution in the coefficient of heat exchange on the average. However, it is also seen well that the cooling efficiency diminishes as the relative discharge increases.

The accuracy of determining Re_x, Eu, and St in the tests is ±2.2, ±7, and ±12.43, respectively. The criterial equations (6), (7), and (8) therefore permit computation of Re_x, the heat-exchange coefficient St and the Euler number for a heated gas flow in separating axisymmetric diffusors with transverse delivery of coolant through the wall in the range of free-stream parameters and channel geometries which occurred in the tests. The lack of published test data on the heat exchange and dynamics of a stream with coolant injection through the wall in axisymmetric diffusors does not permit comparison of the results obtained.

NOTATION

$Re_0 = u_0 d_0/\nu$	is the Reynolds number based on the diffusor entrance diameter;
$Re_x = u_1 x/\nu$	is the Reynolds number based on the longitudinal coordinate x;
$Eu = \Delta p/\rho_0 u_0^2$	is the Euler number;

ν	is the coefficient of kinematic viscosity, m^2/s;
x	is the distance parallel to the channel wall measured from the entrance edge of the diffusor, m;
Δp	is the static pressure drop on the wall between the initial and next section of the channel;
$\overline{m}$	is the relative mass flow of the coolant;
ρ	is the stream density;
u, v	are the longitudinal and transverse stream velocity components, m/s;
G	is the coolant discharge in a given section, kg/h;
F	is the area of the inner surface of a given diffusor section, m^2;
c_p	is the specific heat of a gas at constant pressure in kJ/kg·deg;
T	is the gas temperature, °K;
q	is the heat flux density, W/m^2;
$\varepsilon_{reduced}$	is the reduced emissivity;
λ	is the coefficient of heat conduction of porous diffusor material, W/m·deg;
δ	is the diffusor wall thickness, m;
St	is the Stanton number;
d	is the running value of the inner diameter of the diffusor, m.

Subscripts

0	is the parameter at diffusor entrance;
1	is the stream parameter at outer border of boundary;
w	is the stream parameter on channel inner surface;
cool	is the coolant parameter at entrance to injection chamber;
'	is the value of the quantities on the outer porous wall surface;
"	is the value of the quantities on the inner steel shell surface.

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