AN EXPERIMENTAL STUDY OF TRANSPIRATION AND COMBINATION COOLING WITH TURBULENT FLOW OF AIR THROUGH A ROUND TUBE

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We present the results from an experimental investigation into transpiration and combination cooling in the turbulent flow of air through a round tube; in addition, we present the empirical relationships derived on the basis of experimental data.

Particular attention is currently being devoted to the protection of thermally stressed surfaces by means of transpiration and film cooling methods. However, very few data are available on heat transfer under such cooling conditions, when the flow of air through round tubes is turbulent [1-5].

The investigation of heat transfer in a porous tube has been undertaken in the presence of a hydrodynamic stabilization segment, connected prior to the test. Cold air is injected through the permeable walls of the tube. A detailed description of the installation is presented in [6]; below we offer only a brief description of the test stand and of the measurement methods. The porous tube was fabricated from a fireclay-type ceramic [7]. The dimensions of the working sections are: d_{out} = 49.2 mm, d_{in} = 45.7 mm, and the length was varied from test to test, i.e., 43 mm, 125 mm, and 238 mm, respectively. It is established through preliminary tests that the selected material exhibits excellent homogeneity and uniformity of its porous structure through the length of the channel. Its porosity was 31-33% and the roughness of the inside tube surface was 12 μm, which makes it possible to treat the streamlined surface as aerodynamically smooth. This is confirmed by measurements of the static pressure difference across the length of the working section. The following quantities were measured at the beginning of the test: the flow rate and temperature of the primary and secondary air flows, the temperature of the inside wall at three-five points – with four thermocouples at each point, the velocity and temperature profiles at the outlet from the porous section, the static pressure difference across the porous section, the barometric pressure and the moisture content of the ambient medium, and the presence of the secondary air at the inlet to the porous section. The range of variations for the primary regime parameters was the following: $Re_d = (9.5\text{-}45) \cdot 10^3$, $T_0 = (343 \pm 1)$°K, $T_{in} = (293\text{-}296)$°K, $m = \rho_w v_w/\rho_0 u_0 = (0.2354\text{-}48) \cdot 10^{-3}$ is the injection parameter.

Preliminary calibration tests were used to determine the unsimulated heat losses due to leakage through the flange joints, these losses being accounted for in the determination of the heat-transfer coefficients.

Below we give the experimental heat-transfer results and the results from the determination of the cooling efficiency of the porous tube.

The heat-transfer coefficient is determined from the relationship

$$\bar{\alpha} = \frac{q - q_x}{T_0 - \bar{T}_w}. \tag{1}$$

Here q_x is taken from the calibration curve $q_x = f(m, Re_d, \Delta T = T_{fl} - T_{in})$, while

$$q = \frac{Gc_p(\bar{T}_w - T_{in})}{F}.$$

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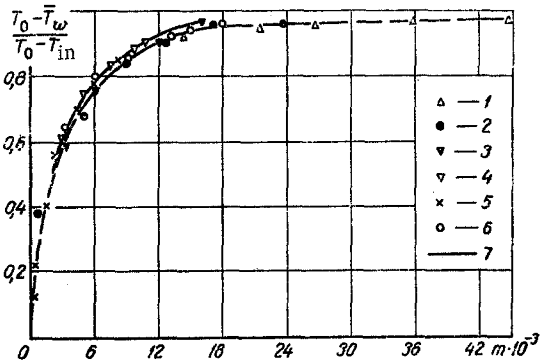

Fig. 1

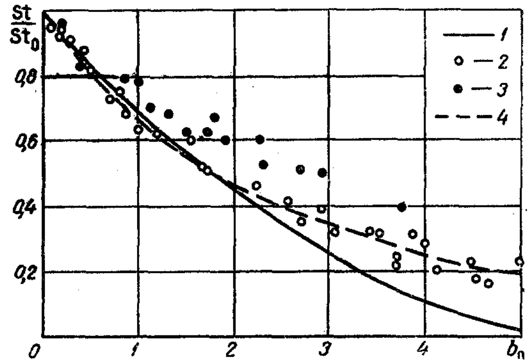

Fig. 2

Fig. 1. Cooling efficiency as a function of m and Re_d: 1) $Re_d = 10350$; 2) 16300; 3) 24300; 4) 36000; 5) 44500; 6) the data of [2]; 7) theoretical calculation [1].

Fig. 2. Effect of injection on convection heat transfer in a porous tube: 1) theoretical calculation [1]; 2) the given experiment; 3) the data of [3]; 4) empirical curve $St/St_0 = 1 - 0.36 b_0 + 0.042 b_0^2$.

The cooling efficiency is determined from the formula

$$\theta = \frac{T_0 - T_w}{T_0 - T_{in}}. \tag{2}$$

Figure 1 shows the data for the cooling efficiency of a porous wall as a function of the injection parameter m for l d = 5.2. Comparison with the experimental data of [2] and with the theoretical calculations of [1] demonstrates that these are in good agreement. The experimental data for τ in the case of shorter tubes ($l/d = 2.84$ and $l/d = 0.94$) lie below the theoretical curve and, moreover, we see that these data are more distinctly layered in terms of the Re_d number.

Figure 2 show the heat-transfer data for $l/d = 5.2$. Here $\overline{St} = \alpha/c_p \gamma u_0 \cdot 3600$ is the Stanton number in the case of injection; $\overline{St}_0 = 0.0306\overline{X}^{-0.2}/Re_d^{0.2} Pr^{0.6}$ is the Stanton number when there is no injection; $b_0 = \rho_w v_w / \rho_0 \overline{u}_0 \overline{St}_0$ is the reduced injection parameter. For purposes of comparison, here we also find the results from [1, 3]. Curve 1 has been plotted for $b_{cr} = 5.98$, determined in accordance with the recommendations of [1] for finite Re_X^{**} numbers

$$Re_X^{**} = \frac{\int\limits_0^x q_w dx}{\mu_0 c_p (T_0 - T_w)}$$

(the Reynolds number, compiled from the energy thickness δ_X^{**}), which prevails in our experiments ($Re_X^{**} = 2 \cdot 10^3$). As we can see, the agreement with the theoretical curve up to $b_0 < 3$ is excellent. The nonagreement for $b_0 > 3$, apparently, can be explained by the reduced accuracy in the determination of the heat-transfer coefficient as the injection parameter increases. The experimental points for $b_0 \leq 5$ are approximated by the following relationship:

$$\overline{St}/\overline{St}_0 = 1 - 0.36 b_0 + 0.042 b_0^2. \tag{3}$$

The divergence between our data and the experiments of [3] is slight and falls within the limits of experimental accuracy. Moreover, it should be noted that in [3] no provision was made for the influx of heat into the clearance of the housing, into which the cooling air enters.

The calculation of heat transfer for the external problem in the presence of a gas screen, according to [4, 5], is possible with the formulas that are usual for the boundary layer, if we assume the difference $T_{ad}^W - T_W$ as the characteristic temperature head.

Data on the measurement of the adiabatic wall temperature in the case of turbulent air flow in a round tube with a gas screen have been published in [6]. The tests were carried out in the same installation as in [6], but with T_W = const. The heat flows were determined both from the heat balances of the various sections and by integration of the temperature and velocity fields at the various sections, over the length of the tube. A detailed description of the measurement method is presented in [6]. First, the heat-transfer tests

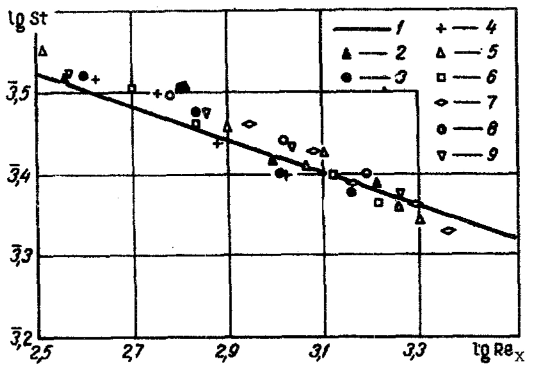

Fig. 3

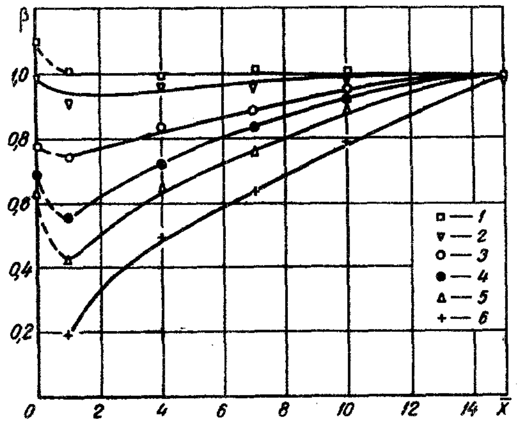

Fig. 4

Fig. 3. Local Stanton number as a function of the local Re_x^{**} number: 1) $\mathrm{St} = 0.0118/\mathrm{Re}^{**0.25} \cdot \mathrm{Pr}^{0.75}$ [1]; 2) $m = 0$, $\mathrm{Re_d} = 41700$; 3) $m = 0$, $\mathrm{Re_d} = 36300$; 4) $m = 0$, $\mathrm{Re_d} = 24100$; 5) $m = 0.0094$, $\mathrm{Re_d} = 41700$; 6) $m = 0.006058$, $\mathrm{Re_d} = 41700$; 7) $m = 0.004151$, $\mathrm{Re_d} = 41700$; 8) $m = 0.007507$, $\mathrm{Re_d} = 36700$; 9) $m = 0.002838$, $\mathrm{Re_d} = 41700$.

Fig. 4. Change in β as a function of $\overline{X}$ and m for combination cooling: 1) $m = 0.01164$, $\mathrm{Re_d} = 24300$; 2) $m = 0.0094$, $\mathrm{Re_d} = 41700$; 3) 0.006058, $\mathrm{Re_d} = 41700$; 4) $m = 0.003077$, $\mathrm{Re_d} = 24300$; 5) 0.002838, $\mathrm{Re_d} = 41700$; 6) $m = 0$, $\mathrm{Re_d} = 41700$.

were performed in the initial segment to which the hydrodynamic-stabilization segment had been connected in advance. The comparison of our measurements with the literature data [1] known to us shows satisfactory agreement. The variations of the parameters in the tests for combination cooling covered the following ranges: $\mathrm{Re_d} = (10\text{–}50) \cdot 10^3$, $m = \rho_w v_w/\rho_0 u_0 = (2\text{–}35) \cdot 10^{-3}$, $T_{in} = (293\text{–}298)$°K, $T_0 = 343 \pm 1$°K, $T_w = (295 \pm 1)$°K.

The experimental results are shown in Fig. 3, where we have denoted

$$\mathrm{St} = \frac{0.0118}{\mathrm{Re}_x^{**0.25}\,\mathrm{Pr}^{0.75}}, \tag{4}$$

$$\mathrm{Re}_x^{**} = u'\delta_x^{**}/\nu, \tag{5}$$

$$\delta_x^{**} = \int_0^R \frac{\rho u}{\rho_0 u'}\left(\frac{T - T^*}{T_{ad}^w - T_w}\right) dR, \tag{6}$$

α is the heat-transfer coefficient,

$$\alpha = \frac{q_w}{T_{ad}^w - T_w}; \tag{7}$$

q_w is the specific heat flux at the wall,

$$q_w = G_{\mathrm{water}} c_p \Delta t_{\mathrm{water}}/F,$$

where t_{water} denotes the heating of the water in the cooling jacket.

Analysis of the derived data shows that calculation of heat transfer in accordance with the recommendations of [4], as regards the conditions of the tests, is possible only when $m < 0.01$.

The parameters δ^{**}, δ_x^{**}, and $\beta = \delta_x^{**}/\delta^{**}$ (Fig. 4) as functions of the variables m and $\overline{X}_1$ indicate that when $m < 0.01$ the transfer of heat is governed by the quantitative relationships that are characteristic of the initial thermal segment. When $m > 0.01$ the injection of the secondary air sets up conditions at the outlet from the porous section that are characteristic for developed turbulent flow when $\beta \geq 1$. Calculation of the heat transfer by the above-indicated method when $m > 0.01$ yields exaggerated values for the heat-transfer coefficient α. As demonstrated by our measurements, under these conditions heat transfer can be calculated from the generally accepted formula

$$\mathrm{Nu} = 0.023\mathrm{Re}_d^{0.8}\mathrm{Pr}^{0.4}. \tag{8}$$

Here the quantity $Re_d = u_{01}d/\nu_{01}$ is calculated from the mean-mass velocity at the outlet from the porous section, while the physical properties are determined from the mean-mass temperature in that section.

The condition $m < 0.01$ has been derived for a porous section length of $l/d = 5.2$. For other lengths of the section connected in advance of the test, this condition will be different. The calculations for β at the outlet from the porous sections show that when $l/d \leq 5.2$ the following equality is maintained:

$$\beta = 4.17\,(l/d)^{0.54} m^{0.5}. \tag{9}$$

Then the inequality $m < 0.01$ is replaced by the condition $\beta < 1$.

NOTATION

St	is the Stanton number;
$\tilde{\alpha}$	is the average heat-transfer coefficient over the tube length, $W/m^2 \cdot deg$;
$Re_d = u_0 d/\nu_0$	is the Reynolds number for the mean-mass velocity at the inlet to the porous section;
$Re_d = u_{01}/\nu_{01}$	is the Reynolds number for the mean-mass velocity at the outlet from the porous section;
$Re_d = u_0 \delta_x^{**}/\nu$	is the Reynolds number for the mean-mass velocity with respect to the energy thickness;
v	is the transverse velocity component at the wall, m/sec;
u'	is the longitudinal velocity along the tube axis, m/sec;
T_{ad}^{w}	is the adiabatic wall temperature, °K;
T^*	is the temperature at a given point on a thermally insulated surface, °K;
T	is the temperature at a given point, on an isothermal surface, °K;
q	is the total specific heat flux, W/m^2;
q_x	is the specific heat flux, characterizing the heat influx through the flanges, W/m^2;
q_w	is the theoretical specific heat flux, W/m^2;
G	is the flow rate of the secondary air, kg/h;
G_{water}	is the flow rate of the cooling water, kg/h;
T_{fl}	is the flange temperature, °K;
T_0	is the axis temperature, °K;
T_w	is the average wall temperature, °K;
T_{in}	is the secondary air temperature, °K;
F	is the inside tube surface, m^2;
$\bar{X} = l/d$	is the relative length of the porous section;
$\bar{X}_1 = x/d$	is the relative length of the working section.

Subscript

w refers to the parameters at the wall.

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