HEAT TRANSFER ON THE SURFACE OF A PLATE WITH SIMULTANEOUS MASS BLOWING AND SUCTION

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Results are given of measurements of heat transfer from a heated surface with simultaneous blowing and suction of mass through performations in the surface. Formulas are derived for computing the heat-transfer coefficients and the temperature profiles in these conditions.

Reference [1] reported measurements of friction coefficients on a flat plate with simultaneous blowing and suction.

The present authors have investigated heat transfer on a flat plate with simultaneous blowing and suction of air in a turbulent boundary layer.

The experimental layout is shown in Fig. 1. The main air flow is created by a subsonic continuous wind tunnel with an open working section. The air velocity was 30 m/sec and the air temperature 300°K.

A heated flat plate model with a permeable perforated surface 1 was mounted parallel to the main air flow in the lower part of the tunnel working section in a zero-gradient flow region.

At the front of the plate a wire turbulence agent [7] was mounted. In the experimental conditions the thermal boundary layer was immersed in the dynamic boundary layer.

The model was a plate made of Textolite of length 580 mm and width 200 mm. A heater element of graphite cloth was pressed into the surface. The heater area was 420 × 130 mm^2.

Part of the heater surface, of dimension 200 × 80 mm^2, was perforated with apertures of diameter 1 mm and spacing 4 mm. The total perforated surface comprised 5.2% of the total area.

One series of apertures was joined to the blowing cavity, and the other series to the suction cavity, as shown in Fig. 1 (I).

Blowing and suction of air was carried out by means of a single fan 5, which provided equal flow rates of blown and sucked air. Additionally, the air in the blowing channel could be heated by the spiral heater 6. The entire channel from the blowing cavity to the suction cavity, including the fan, was carefully sealed.

Variation in the blowing–sucking intensity was accomplished by controlling the fan speed.

During the experiments measurements were taken of the flow rate and temperature of the blown and sucked air, the electric power of the heaters, the plate surface temperature at 15 points along the longitudinal axis, and the distribution of velocities and temperatures through the boundary layer and along the longitudinal axis.

The longitudinal air velocity above the permeable plate was measured by means of a Pitot tube, and the temperature profiles were measured with a moving differential Chromel–Copel thermocouple. The temperatures of the plate surface and of the blown and sucked air were measured with Chromel–Copel thermocouples.

The thermocouples were mounted on the plate surface both ahead of and behind the blowing and suction apertures.

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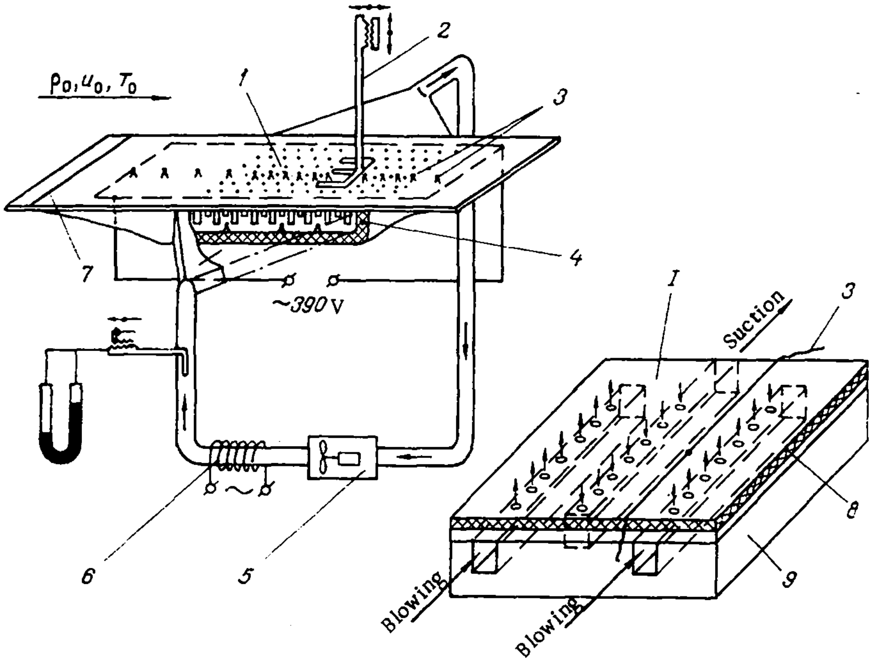

Fig. 1. The experimental layout: I) arrangement for simultaneous blowing and suction; 1) permeable heated plate; 2) sensor for measuring the air temperature and velocity; 3) surface thermocouple; 4) thermal insulation; 5) fan; 6) air heater; 7) turbulence generator; 8) plate heater element; 9) distribution box.

The differences in the readings of the thermocouples were less than 1°.

Experiments were carried out with a constant specific heat flux from the wall, equal to 5.9 kW/m^2, and blowing–suction intensities of $\bar{j}_W$ = 0-0.0185.

The temperatures of the blown and sucked gas were kept the same by heating the blown air in the heater 6; the temperature varied from 35 to 41°C.

Figure 2 compares the local measured heat-transfer coefficients (St_0) as a function of Re_T^{**} with theoretical values, for the case of no blown and sucked mass. The satisfactory agreement between the test data and the well-known heat-transfer law [2]

$$St_0 = 0.0128\, Re_T^{**-0.25}\, Pr^{-0.75} = 0.0166\, Re_T^{**-0.25} \tag{1}$$

indicates the reliability of this method of experimental investigation and the accuracy of construction of the heated plate.

Analytical estimates show that the heat leakage under steady conditions due to conduction to the plate body and radiation were no more than 2% of the power supplied.

Reference [1] gave a formula for calculating the resistance of a plate with simultaneous blowing and suction of mass through a perforated surface

$$\Psi_\pm = (c_{f\pm}/c_{f0})_{Re^{**}} = 1 + (b_\pm/b_{cr})^2. \tag{2}$$

Figure 3 compares the average measured drag coefficients ($\Psi_\pm$) in the present experiments with values obtained using Eq. (2). The relative friction law $\Psi_\pm$ was calculated from the measured velocity and temperature profiles by an integral method, using the technique described in [1].

It can be seen from Fig. 3 that Eq. (2) is satisfactorily confirmed up to $b_\pm$ = 5.3, and thereafter $\Psi_\pm$ takes values close to 2.

The results of the measured heat transfer on a plate with simultaneous blowing and suction of mass were also processed in the form of a correlation between the relative heat-transfer law $\Psi_{\pm T} = (St_\pm/St_0)_{Re_T^{**}}$ as a function of the permeability parameter $b_\pm$.

The quantity $St_\pm$ was determined from the relation

$$St_\pm = \frac{q_W}{\rho_0 u_0 c_p (\bar{T}_W - T_0)\Psi_T}, \tag{3}$$

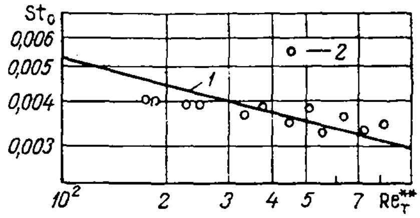

Fig. 2. Heat-transfer law on an impermeable surface: 1) theory, Eq. (1); 2) test data.

in which the local wall temperatures (T_w) were averaged over the entire blowing–suction length:

$$\frac{1}{\bar{T}_w - T_0} = \int_{\bar{x}_1}^{\bar{x}_2} \frac{d\bar{x}}{T_w - T_0}. \tag{4}$$

The value of St_0 was calculated from the heat-transfer law of Eq.(1), in which the number $Re_T^{**} = u_0\delta_T^{**}/\nu_w$ was determined from the velocity at the edge of the thermal boundary layer and the calculated energy loss thickness (δ_T^{**}) from the expression

$$\delta_T^{**} = \int_0^{\delta_T} \frac{\rho u}{(\rho_0 u_0)_{\delta_T}} \left(1 - \frac{T_w - T}{T_w - T_0}\right) dy. \tag{5}$$

The values of St_0 were also averaged over the entire mass blowing–suction section using the formula

$$St_0 = \frac{0.0128\,(Re_{T2}^{**0.75} - Re_{T1}^{**0.75})}{0.75\,(Re_{T2}^{**} - Re_{T1}^{**})\,Pr^{0.75}}. \tag{6}$$

The coefficient Ψ_T in Eq. (3), which accounts for the influence of nonisothermal conditions, was determined from [2]

$$\Psi_T = \sqrt{T_0/T_w} \tag{7}$$

and varied from 0.93 to 0.96.

The permeability parameter $b_\pm$ was determined from the relation

$$b_\pm = b_{\pm T} \frac{Re^{**0.25}}{Re_T^{**0.25}\,Pr^{0.75}}. \tag{8}$$

It can be seen from Fig. 3 that with this processing the test heat-transfer data for simultaneous mass blowing and suction can be described very well by Eq. (2). After the value $b_\pm = 5.3$ is reached, the value of $\Psi_{\pm T}$ also remains equal to 2 with further increase in $b_\pm$.

Figure 4 shows the temperature profiles through the thermal boundary layer for various intensities of mass blowing and suction. With increase in intensity of the mass blowing and suction the dimensionless temperature profiles become less full, and after the critical value of the blowing–suction parameter is reached ($b_{\pm cr} = 5.3$), they become self-similar relative to $\bar{j}_\pm$.

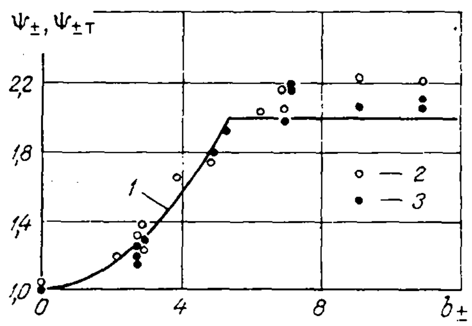

Fig. 3. Relative friction and heat-transfer coefficients for a flat plate with simultaneous blowing and suction of air: 1) theory according to Eq. (2); 2) experiments on friction ($\Psi_\pm$); 3) experiments on heat transfer ($\Psi_{\pm T}$).

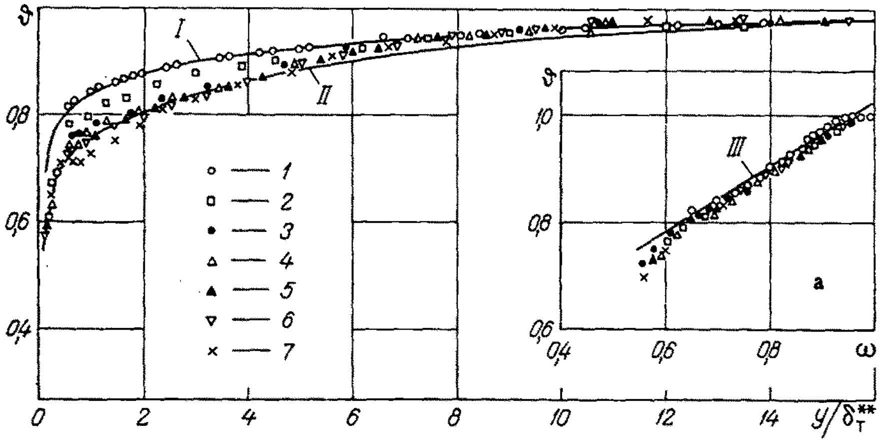

Fig. 4. Temperature profiles in a turbulent boundary layer with simultaneous blowing and suction of air: a) nonsimilarity of the velocity and temperature profiles; I) theory using Eq. (10); II) theory using Eq. (13); III) theory using Eq. (10); 1) $\bar{j}_{\pm} = 0$; 2) $\bar{j}_{\pm} = 0.005$; 3) $\bar{j}_{\pm} = 0.01$; 4) $\bar{j}_{\pm} = 0.011$; 5) $\bar{j}_{\pm} = 0.0121$; 6) $\bar{j}_{\pm} = 0.016$; 7) $\bar{j}_{\pm} = 0.0185$.

Because the thermal boundary layer in these experiments is immersed in the dynamic boundary layer and $Pr \neq 1$, there is a breakdown of similarity in the distribution of dimensionless velocity and temperature profiles.

With a power-law approximation for the profiles of velocity (ω) and temperature (ϑ), these are related by the formula [2]

$$\vartheta = \varepsilon_1 \omega^{n_T/n}. \tag{9}$$

For the conditions of the experiment ($\varepsilon_1 = 1.025$, $n = 1/8$, $n_T = 1/16$) we obtain the formula

$$\vartheta = 1.025\sqrt{\omega}. \tag{10}$$

Figure 4a compares Eq. (10) with the experimental data. It can be seen that Eq. (10) is confirmed quite well not only for an impermeable plate, but also for a plate with simultaneous mass blowing and suction. Using Eq. (10), we can obtain the following formula for the temperature profile on an impermeable plate:

$$\vartheta_0 = 0.824\left(\frac{y}{\delta_T^{**}}\right)^{1/16}. \tag{11}$$

If we use an expression for the velocity profile on a flat plate with simultaneous blowing and suction from [1], and allow for Eq. (10), we can obtain an expression for the dimensionless temperature profile.

In particular, for the critical blowing and suction mass, when [1]

$$\omega_{\pm cr} = \frac{3\omega_0^2 - \omega_0^4}{2}, \tag{12}$$

we obtain the critical dimensionless temperature profile

$$\vartheta_{\pm cr} = \frac{\vartheta_0^2}{\varepsilon_1}\sqrt{\frac{3 - \frac{\vartheta_0^4}{\varepsilon_1^4}}{2}}. \tag{13}$$

As can be seen from Fig. 4, there is satisfactory agreement between the measured dimensionless temperature profiles on the plate with simultaneous mass blowing and suction and the calculations using Eq. (13).

NOTATION

$\bar{j}_+$ and $\bar{j}_-$, relative mass velocity of blown and sucked gas, respectively; δ_T^{**}, energy loss thickness, m; u_0, flow velocity, m/sec; St, Stanton number; $b_{\pm}$, permeability parameters with blowing and suction; c_f, friction coefficient; $Re_T^{**} = u_0\delta_T^{**}/\nu_w$, Reynolds number; $\Psi_{\pm} = c_{f\pm}/c_{f0}$; $\Psi_{\pm T} = St_{\pm}/St_0$, relative laws for friction and heat transfer with blowing and suction; $\vartheta = (T_W - T)/(T_W - T_0)$, $\omega = u/u_0$, relative temperature and velocity, respectively; δ, δ_T, thickness of the dynamic and thermal boundary layers, respectively; $\varepsilon_1 = (\delta/\delta_T)^{n_T n}$, coefficient of nonsimilarity of velocity and temperature profiles; n and n_T, indices for power law approximation to the velocity and temperature profiles, respectively; T_W and T_0, temperatures of the wall and of the main air flow.

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NUMERICAL ANALYSIS OF LAMINAR FLUID FLOW IN A "POROUS PIPE IN A PIPE" HEAT EXCHANGER

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The change in the velocity of liquid overflow through the wall of a porous tube along the length of a system as a function of the input flow parameters and the permeability of the inner tube walls is investigated.

A flow with variable rate along pipes of different configurations was recently the theme of a considerable number of publications, which has certainly been caused by the demands of engineers analyzing such flows. Motion with mass varying along the path is realized in collector systems, e.g., in heat exchangers with porous elements, in heat pipes, etc. Laminar fluid flow in a porous pipe with constant suction along the length has been studied in [1-5]. A flow with rate of mass delivery or extraction varying according to a known law has been studied considerably less [6-8]. A more complex case is examined in this paper — flow with variable mass in two channels separated by a permeable baffle. The velocity of overflow from one channel to the other is not known and is determined from the solution of the problem. The viscosity and density of the fluid are hence considered constant, mechanical energy dissipation into heat is neglected, and the geometric dimensions of the system are such that the channel length is substantially greater than its diameter.

Laminar incompressible fluid flow in a circular pipe with permeable walls of a porous material inserted in a coaxial pipe of larger radius with impermeable walls is considered.

Under the effect of a pressure drop on the porous pipe wall, the fluid will be sucked out of the inner channel into the outer or conversely. The pressure drop on the wall and the overflow velocity depend on the axial coordinate. Let us take Darcy's law for porous material as the dependence of the overflow on the pressure drop. The suction rate for a homogeneously porous wall with constant thickness can be expressed as follows according to Darcy's law:

$$v_{|r=a} = \frac{K(p_{|r=a} - p_{|r=b})}{\mu a \ln \frac{b}{a}}. \tag{1}$$

The flows in the inner channel and in the annular gap are described by the stationary Navier-Stokes equations:

$$\begin{aligned} u^{(i)}\frac{\partial u^{(i)}}{\partial x} + v^{(i)}\frac{\partial u^{(i)}}{\partial r} &= -\frac{1}{\rho}\frac{\partial p^{(i)}}{\partial x} + \nu\left(\frac{\partial^2 u^{(i)}}{\partial x^2} + \frac{1}{r}\frac{\partial u^{(i)}}{\partial r} + \frac{\partial^2 u^{(i)}}{\partial r^2}\right), \\ u^{(i)}\frac{\partial v^{(i)}}{\partial x} + v^{(i)}\frac{\partial v^{(i)}}{\partial r} &= -\frac{1}{\rho}\frac{\partial p^{(i)}}{\partial r} + \nu\left(\frac{\partial^2 v^{(i)}}{\partial x^2} + \frac{1}{r}\frac{\partial v^{(i)}}{\partial r} + \frac{\partial^2 v^{(i)}}{\partial r^2}\right), \end{aligned} \tag{2}$$

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