EXPERIMENTAL STUDY OF THE TURBULENT BOUNDARY LAYER ON A SMOOTH FLAT PLATE WITH STEPWISE HEAT ADDITION

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Results are presented of an experimental study of heat transfer with stepwise heat addition on a flat wall. The experimental temperature profiles and heat transfer data obtained confirm the previously suggested hypothesis that the conventional heat-transfer law is valid in this case if the calculation is based on the difference of the equilibrium and actual wall temperatures.

The experimental setup consisted of a subsonic wind tunnel with rectangular working section dimensions 110 × 110 × 1300 mm. The air entered the horizontal working channel through a profiled nozzle. The working channel had a horizontal heat-transfer segment consisting of nine brass plates. The plate dimensions were 120 × 95 mm. Transverse slots were provided in the plates to prevent longitudinal heat crossflow. There were 26 nichrome-constantan thermocouples made from 0.2-mm-diameter wire embedded in the surface of the plates along the segment length (axially). At certain sections three thermocouples were installed across the width of the plate. A heating element was mounted below each plate. The heating-element power could be regulated to provide arbitrary variation of the heat fluxes to the wall. The heat transfer segment was thermally insulated on the bottom and sides. The clearances (~8 mm) between the plate and the side walls of the working channel were filled with asbestos. The plates with heating elements were mounted on asbestos-cement slabs and insulated below by a 140-mm-thick layer of foam.

Calibration experiments were first conducted to study the dynamic characteristics of the gas flow under isothermal conditions. The tests were made with air flow over the plate with velocities $W_0 = 20-120$ m/sec (the Reynolds number was $R_x = 4 \cdot 10^5 - 5 \cdot 10^6$). A total head tube and a static pressure pickup were used to measure the velocity. The total head tube has a rectangular section with height 0.5 mm and width 1.5 mm. The tube wall thickness is 0.1 mm. The pressure was recorded on an MMN cistern micromanometer at low speeds, with a U-tube water manometer being used at high velocities. The measurements showed good uniformity of the velocity fields at the entrance to the working channel (velocity nonuniformity did not exceed 3%). The velocity variation along the channel axis was about 2%. The velocity profiles in the boundary layer were also measured in the course of the dynamic study. The values of the local friction coefficient c_f were determined from the velocity profiles measured by the Clauser method [1, 2]. The experimental results agreed to within 5% with the calculated value of the local friction coefficient, determined using the formula [3]

$$c_f = \frac{0.0256}{R^{**0.25}} \tag{1}$$

where R^{**} is the Reynolds number based on the momentum thickness δ^{**}. This indicates the presence of a developing turbulent boundary layer on the plate.

Experiments were then conducted to determine the heat losses, which amounted to ~15% of the total amount of heat supplied to the wall.

The thermal calibration experiments involved determination of the heat transfer coefficient on the smooth plate. The tests were conducted under quasi-isothermal conditions with constant wall temperature and with constant heat flux at the wall. In the experiments the wall temperature varied in the range $t_w = 70-150°$ C and the thermal fluxes amounted to $q_w = 3000-10{,}000$ kcal/m$^2 \cdot$ h. The main flow temperature was $t_0 = 20-60°$ C. The air flow velocity varied in the range $W_0 = 10-125$ m/sec ($R_x = 2.5 \cdot 10^5 - 6 \cdot 10^6$). The Stanton number S was determined from the quantities measured in the experiments (heating-element power, wall and undisturbed stream temperatures, undisturbed flow velocity). In this calculation a correction (4–9%) was made for nonisothermicity and account was also taken of the radiation heat losses, which did not exceed 4% of the magnitude of the convective heat flux. The experimental point scatter was ±10%. The mean curve satisfying the experimental points is described by the formula

$$S_0 = 0.0184 \, R_T^{**\,-0.25} \qquad (P = 0.71) \tag{2}$$

where R_T^{**} is the Reynolds number based on the energy thickness. This relation corresponds to the formula obtained as a result of analysis of all the experimental data on convective heat transfer on a plate and in a tube, presented in [3], and also to the Seban formula, used in [5].

Experiments were then conducted with stepwise heat addition. The heat-input scheme is shown in Fig. 1. All the experiments were conducted with two thermal steps. The length of the steps was 360 and 240 mm. In Fig. 1 curve 1 is for calculation with formula (2), the experimental points 2* (0.3, 20.4), 3 (0.5, 76.9), 4* (1.0, 121.5), 5* (3.1, 9.9), 6 (6.5, 14.9), 7 (6.7, 15), 8 (6.7, 15.4), 9 (3.5, 35.0), 10 (3.5, 38.5), 11 (2.8, 78.8) correspond to the specific heat flux ratios q_{W1}/q_{W2} (first values in the parentheses) and the air flow velocities W_0 m/sec (second values in parentheses). The experiments marked with an asterisk were conducted with an adiabatic segment 0.24 m long between the steps. The experiments were performed in two stages. Initially heat was supplied only to the first step q_{W1}. In this case the second step, where $q_{W2} = 0$, was thermally screened from the first step. The temperature on the adiabatic wall T_W^*, or the so-called "equilibrium" temperature, was measured. The measured equilibrium temperature corresponded to the calculated values obtained using the formula [6]

$$\frac{T_w^* - T_0}{T_w' - T_0} = \left(1 + 15.5\,\frac{x - x_0}{x_0}\right)^{-0.8} \qquad [T^\circ \mathrm{K}] \tag{3}$$

Here x_0 is the length of the heated segment, T_W' is the temperature at the end of the heated segment. Then heat was supplied to the second step and the wall temperature T_W was measured in the screen zone. The data from the experiments with stepped heat addition were analyzed in two ways.

1. The Stanton number in the screen zone was determined from the difference between the wall and undisturbed stream temperatures $(T_W - T_0)$. We see from comparison of the experimental points with the curve in Fig. 1 that only those experiments in which the thermal flux ratio $q_{W1}/q_{W2} < 1$, or in which there is an adiabatic segment between the steps, are in satisfactory agreement with the curve. In such experiments, owing to the small heat flux in the first step and the presence of the adiabatic segment ahead of the second step the equilibrium wall temperature T_W^* is close to the undisturbed stream temperature T_0. In those experiments in which the initial heat flux exceeds the following heat flux the experimental points deviate from the curve. This deviation increases with increase of the ratio q_{W1}/q_{W2}. The maximum deviation of the points from the curve (up to 2.5 times) is observed in the experiments with the largest heat flux ratio $(q_{W1}/q_{W2} \sim 6.7)$. Thus, we see from this figure that the heat transfer law in the form of (2) cannot be used for the case of stepwise heat addition if the heat transfer coefficient is calculated using the difference $(T_W - T_0)$.

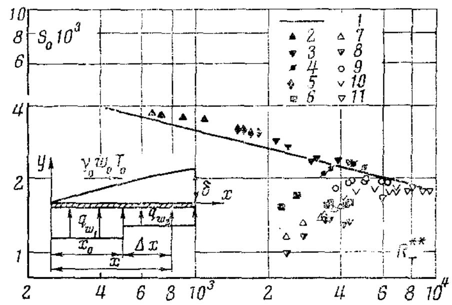

Fig. 1

2. The heat transfer coefficient was calculated from the difference of the actual and equilibrium wall temperatures; in this case

$$S_a = \frac{q_w}{\gamma_0 W_0 c_{p_0} (T_w - T_w^*)} \tag{4}$$

In this analysis the experimental data on heat transfer in the screen zone for stepwise heat addition are shown in Fig. 2, in which the notation is the same as in Fig. 1. Here S_{a0} is the experimental value of the Stanton number, reduced to standard conditions,

$$S_{a0} = \frac{S_a}{\Psi_t}\,\varphi(x, x_0) \tag{5}$$

Here

$$\Psi_t = \left(\frac{2}{\sqrt{\psi^*} + 1}\right)^2, \quad \psi^* = \frac{T_w}{T_w^*}, \quad \varphi(x, x_0) = \left(\frac{x}{x - x_0}\right)^{0.114}$$

Ψ_t accounts for the influence of nonisothermicity on the heat transfer [3]; $\varphi(x, x_0)$ accounts for the influence of the previously activated segment on the heat transfer [3, 4]; x_0 is the preactivated segment of the second step. The Reynolds number was also based on the difference of the actual and equilibrium wall temperatures:

$$(R_T^{**})_a = \frac{1}{gC_{p_0}\mu(T_w - T_w^*)} \int_{x_0}^{x} q_w dx \tag{6}$$

We see from the figure that the experimental data agree satisfactorily with calculation with (2). The experimental point scatter is ±15%. Formula (2) will also be valid for stepwise heat addition if the heat transfer coefficient is based on the difference $(T_W - T_W^*)$.

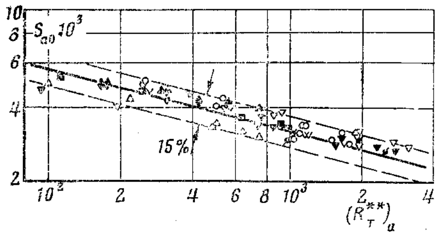

Fig. 2

In conducting experiments on staged heat addition the temperature profile in the boundary layer in the screen zone was also measured. The measurements were made with air flow velocity W_0 = 21 m/sec and the heat flux ratio for which maximum deviation of the heat transfer experimental points from the calculated value is observed (Fig. 1). This condition was repeated twice at different times. These experiments are plotted separately in Fig. 3 (filled and open symbols). Here the experimental points 2 (6000, 20.9) and 3 (3300, 39.4) correspond to constant heat fluxes q_W kcal/m^2 · h (first values in the parentheses) and flow velocities W_0 m/sec (second values in the parentheses); the points 4, 5, 6, and 7 correspond to the values Δx = 0.036, 0.083, 0.204, and 0.278 m for $q_{W1}/q_{W2} \sim 6$ and W_0 = 21 m/sec; the measured section distance Δx was reckoned from the beginning of the second step. Figure 3 shows the dimensionless temperature profile, constructed in two ways: In Fig. 3A the dimensionless profile was constructed so that the excess temperature in the boundary layer is referred to the difference of the wall and undisturbed stream temperatures. In this instance the energy thickness in the dimensionless group $Y = y/\delta_T^{**}$ was defined as

$$\delta_T^{**} = \int_0^{\delta} \frac{\rho W}{\rho_0 W_0} \left(\frac{T - T_0}{T_w - T_0}\right) dy \tag{7}$$

Figure 3 shows the temperature distribution in the boundary layer following the 1/7-th-power law in the form of the curve

$$\Theta = \frac{T - T_0}{T_w - T_0} = 1 - 0.715 Y^{1/7}, \qquad Y = y/\delta_T^{**}$$

A significant disagreement of the experimental results with this curve is observed. Also plotted here are the experimental data for T_W = const, which are in good agreement with the calculated profile. In Fig. 3B these same experimental data for stepwise heat addition were analyzed so that the excess temperature was referred to the difference between the actual and equilibrium wall temperatures. In this case the energy thickness in the dimensionless complex $Y_a = y/(\delta_T^{**})_a$ was defined as

$$(\delta_T^{**})_a = \int_0^{\delta} \frac{\rho W}{\rho_0 W_0} \left(\frac{T - T^*}{T_w - T_w^*}\right) dy \tag{8}$$

where T* is the temperature at the boundary layer point in question in the absence of heat flux on the wall.

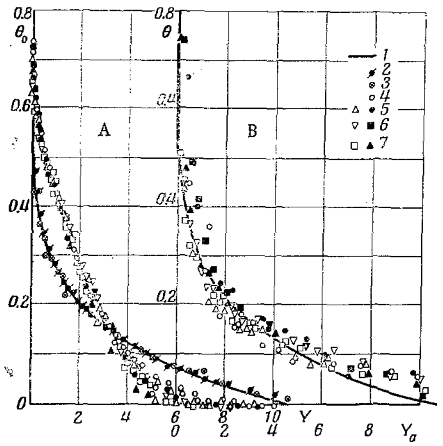

Fig. 3

We see from the figure that the experimental points agree satisfactorily with the temperature distribution curve in the boundary layer following the 1/7-th-power law.

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