EXPERIMENTAL INVESTIGATION OF TURBULENT HEAT TRANSFER IN A BOUNDARY LAYER ON A COOLED FLAT PLATE

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Results are presented of an experimental investigation of the effect of the temperature factor on the law of heat transfer in turbulent boundary layers.

In spite of the large volume of experimental investigations of the effect of nonisothermal conditions on the heat-transfer coefficient in a turbulent boundary layer, the effect of strong wall cooling on the heat-transfer coefficient is still in dispute. The reason is that, for the relatively mild wall cooling obtaining in most experimental investigations, the effect falls within experimental error. The tests with strong wall cooling, as a rule, are not sufficiently accurate to allow final quantitative conclusions to be drawn.

In the present work the investigations were made on a flat plate in subsonic flow, thus eliminating the effect of things like pressure gradient and compressibility on the parameters to be measured. The plate (Fig. 1) was made of sheet brass of thickness 3.5 mm, in the form of a sealed box of length 295 mm, width 160 mm, and thickness 14.0 mm. Transverse partitions divided the inside space into 7 sections of length 25, 30, 35, 40, 45, 50, and 55 mm. The plate had graphite fairings front and back to achieve unseparated flow. The length of the forward fairing was chosen to achieve a fully turbulent boundary layer on the test plate. This was verified by the variation of the heat-transfer coefficient along the plate. The plate was mounted in a brass tube of diameter 150 mm. The tube had its own cooling, which was used to keep the temperatures of the tube inside wall and the plate surface identical.

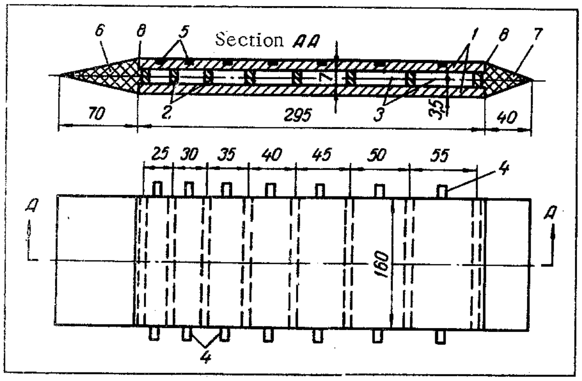

Fig. 1

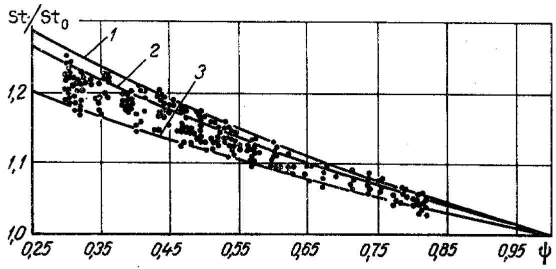

Fig. 2

Fig. 1. Diagram of the cooled brass plate: 1) brass plates; 2) partitions between sections; 3) channels for water cooling; 4) water inlet and outlet tubes; 5) thermocouples for plate surface temperature measurement; 6) front fairing; 7) rear fairing; 8) mica.

Fig. 2. The dependence of $St/St_0 = f(\psi)$ (the solid lines are calculated by the Kutateladze and Leont'ev formula [1]; the points are the test data): 1) for $Re_T^{**} = 9000$; 2) 5000; 3) 1000.

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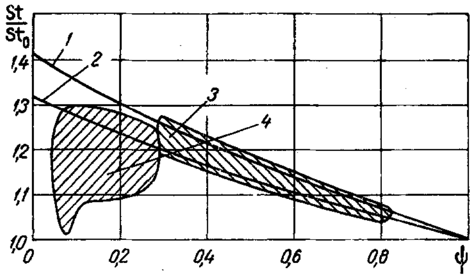

Fig. 3. Dependence of $St/St_0 = f(\psi)$ (the solid lines are as calculated by the Kutateladze and Leont'ev formula [1]): 1) for $Re_T^{**} = 5000$; 2) 1000; 3) the present test region; 4) Belyanin's test region [3];

The plate was exposed to a stream of combustion products of a turbojet engine with its own feed and control system. The mass flow of gas could be varied from 1.5 to 4.0 kg/sec.

The quantities measured during a test were: the mass flow rate and temperature rise of the cooling water for each section; the temperature of the wall exposed to the gas in each section; the temperature of the gas stream, and the gas mass flow rate. In addition, a number of auxiliary measurements were made, required for operational control of the experimental rig. The water flow rate was measured by a volume method.

For measuring the temperature rise of the cooling water in each section we used a two-electrode differential thermocouple, of copper–Copel wires of diameter 0.5 mm. A thermocouple was mounted in a special insulated pocket at the entrance and exit of each section. The wall temperature measurements were made by means of thermocouples embedded in the wall at a distance of 0.2 mm from its outer surface. They were made from Chromel–Alumel wires of diameter 0.2 mm in double insulation (enamel and fiberglass). The thermocouple electrodes were fastened down in a transverse channel in the middle of each plate section, carefully insulated with mica, and soldered to foil.

The temperature field in the gas stream was measured by means of shielded thermocouples of outer diameter 6 mm. They were mounted along two sections, one ahead of and one behind the plate, each section having three thermocouples at angles of 120° relative to one another. The temperature rise of the water, and the temperatures of the plate and gas stream were recorded by means of a type PPTV1 high impedance dc potentiometer.

The gas flow rate in the tube was measured by means of two rakes mounted along two mutually perpendicular diameters. The actual flow rate through the control section was calculated as the sum of the rates through the conventional annular sectors containing the individual rake sensors:

$$G_0 = \left(\frac{2k}{k+1}\,\frac{g}{RT_0}\right)^{1/2} \sum p_0\, q\, \Delta F,$$

$$q = \lambda \left(1 - \frac{k-1}{k+1}\lambda^2\right)^{\frac{1}{k-1}}.$$

All the parameters were measured after the rig operating conditions became steady, i.e., for fixed gas stream and plate wall temperatures. At the test conditions, the wall temperature along the plate was held constant by varying the water flow rate in the cooling sections.

All the measurements were made in the 61 condition [sic].

The temperature factor was varied from 0.824 to 0.295; the gas stream temperature from 373 to 1323°K; the gas speed from 180 to 300 m/sec; and Re_T^{**} from 1000 to 6000. The test data were reduced by the method described by Leont'ev and Fedorov [2].

The local heat-transfer coefficient is

$$\alpha_i = \frac{q_i}{\Delta T_i} = \frac{G_{Wi}\, c_{pWi} \Delta t_{Wi}}{2F_s\, (T_0 - T_{wi})}.$$

The characteristic Reynolds number for the thermal boundary layer at the end of the section is

$$Re_{Ti}^{**} = \frac{\sum_{i=1}^{7} q_i\, \Delta x_i}{g\, c_{p0}\, \mu_0 (T_0 - T_{wi})},$$

where c_{p0} and μ_0 are the specific heat and dynamic viscosity of the gases. (They were determined from a thermodynamic calculation of the combustion products for each condition.)

We found the Stanton number at the center of each section from the expression:

$$\mathrm{St} = \frac{\alpha_i}{c_p \gamma_0 w_0}; \quad \gamma_0 w_r = \frac{G_0}{F_{\mathrm{free}}}.$$

The relative variation of St with Re_T^{**} = idem is $\Psi = (\mathrm{St}/\mathrm{St}_0)_{\mathrm{Re}_T^{**}}$, where $\mathrm{St}_0 = 0.0128/\mathrm{Pr}^{0.75}\,\mathrm{Re}_T^{**0.25}$.

An analysis of the errors showed that the maximum error in determining the heat-transfer coefficient in the conditions examined is ±7%.

Reduction of the heat-transfer test data to fit the form $\mathrm{St}/\mathrm{St}_0 = f(\psi)$ (Fig. 2) shows that the temperature factor has an appreciable effect on the heat-transfer law. The effect becomes substantial for ψ less than 0.5. The solid lines on Fig. 2 show the results of calculation using the Kutateladze and Leont'ev theory, allowing for the finite length and Re_T^{**} for $\mathrm{Re}_T^{**} = 5000$ and 1000. It can be seen that the test data are in satisfactory agreement with this theory [1].

In the range of variation of ψ from 0.5 to 0.295 the Kutateladze and Leont'ev relation

$$\frac{\mathrm{St}}{\mathrm{St}_0} = \frac{4}{(\sqrt{\psi - 8{,}2(\psi - 1)\sqrt{c_{f0}}} + 1)^2}; \quad c_{f0} = \frac{2}{(2.5 \ln \mathrm{Re}_T^{**} + 3{,}8)^2}.$$

was confirmed very well by the tests.

Figure 3 shows the results of tests by Belyanin with gas flow in a strongly cooled tube [3]. It can be seen that the results from the theory of [1] are in good agreement with the test data up to the value $\psi \approx 0.05$.

NOTATION

$T_0, p_0, w_0, c_{p0}, \gamma_0$ are the temperature, pressure, speed, specific heat, and specific weight of the combustion products;
ΔF is the area of the annular sector in the gas flow rate measurement;
G_W is the water flow rate in the cooled sections;
c_{pW}, t_W are the specific heat and the temperature rise of the water in the section;
F_s is the area of the plate section washed by the gas;
Δx_i is the length of the i-th section of the plate;
F_{free} is the area of cross section of the tube not occupied by the plate;
St_0 is the heat-transfer coefficient for isothermal flow over an impermeable plate;
$\psi = T_W/T_0$ is the temperature factor;
c_{f0} is the friction factor for isothermal flow;
Re_T^{**} is the characteristic Reynolds number for the thermal boundary layer.

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