HEAT TRANSFER WITH TANGENTIAL FLUID INJECTION

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The paper presents experimental data on heat transfer to a smooth plate with tangential fluid injection, and compares data on heat transfer to a tubular surface with and without injection of cooling gas.

Notation

x - coordinate;	q - heat flux;
w - velocity;	μ - dynamic viscosity;
t - temperature;	d - tube diameter;
γ - density;	α - heat transfer coefficient;
s - height of slit;	λ - thermal conductivity;
Δ - height of tube;	R - Reynolds number;
θ - "effectiveness" of injection cooling;	N - Nusselt number;
	S - Stanton number;

$$R_x = \frac{\gamma_\infty w_\infty x}{\mu_\infty}, \quad m = \frac{\gamma_0 w_0}{\gamma_\infty w_\infty}, \quad N_x = \frac{\alpha x}{\lambda}, \quad S_t = \frac{\alpha}{\gamma_\infty c p_\infty w_\infty},$$

$$\theta = \frac{t_{aw} - t_\infty}{t_0 - t_\infty}, \quad R_s = \frac{\gamma_0 w_0 s}{\mu_\infty}.$$

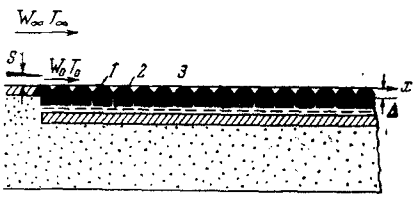

Fig. 1

1. The experiments were carried out on the set-up described in [1]. The main part of the set-up, with the experimental heat-transfer panel, is shown in Fig. 1. The heat-transfer plate 1 consisted of a 10-mm thick steel plate with transverse slits filled with epoxy resin, which eliminated heat transfer in the longitudinal direction. The plate was mounted on an electric heating palte 2 composed of several sections. The length of each section was 112 mm and the electrical resistances of the sections were equal to within 8%. The electric heating plate was insulated from the heat transfer plate by a sheet of 0.5-mm asbestos paper. The tubular surface was formed by tin-plated steel half-cylinders 3 with a diameter 2Δ = 28 mm attached to plate 1. The contact surfaces between the half-cylinders and the plate were polished. The sides of the experimental panel were insulated with 10 mm of asbestos on each side. The whole assembly was insulated from below with 200 mm of slag wool. The conductive heat loss was measured, and constituted up to 20% of the total heat supplied.

The experimental method was described in [1]. In addition the electric power supplied to each couple of heater sections, which were connected in parallel, was measured. During the experiment, the power supplied to each couple was maintained at a constant value within 2%. The flux q varied between $2 \cdot 10^3$ and $4.5 \cdot 10^3$ W/m^2. The main air stream had the ambient temperature t_∞. The temperature t_0 of the injected air was 0–130° above the ambient.

2. Figure 2(a) shows the results of experiments with a smooth plate and without the secondary stream, s = 0. The experimental points lie near line 1, whose equation is $N_{x0} = 0.0263\, R_x^{0.8}$, which represents the heat transfer coefficient in a turbulent boundary layer on a flat plate with constant heat flux [2]. The scatter of the experimental points is apparently due to the nonuniformity of the heating. The good agreement between the experimental data and the accepted formula indicates that the set-up is well-suited to the present investigation.

Figure 3 shows the variation of the reduced Nusselt number for the injection cooling of a smooth plate as a function of the distance from the injection slit for $m = \gamma_0 w_0 / \gamma_\infty w_\infty < 1$. The main governing parameters of the experiments are given in the table. The heat transfer coefficient for the injection cooling experiments was defined as

$$\alpha = \frac{q}{t_w - t_{aw}} \qquad (2.1)$$

Here t_w is the measured surface temperature in the experiment, and t_{aw} is the adiabatic wall temperature under the corresponding conditions. The value of t_{aw} was calculated from the formula [3]

$$\theta = \left[1 + 0.24 \frac{\Delta x}{ms} R_s^{-0.25}\right]^{-0.8},$$

$$\left(\theta = \frac{t_{aw} - t_\infty}{t_0 - t_\infty}\right), \qquad (2.2)$$

which correlates the results of experiments carried out on this set-up [1]. The values of N_{x0} were taken from Fig. 2(a). It can be seen that for x/s > 50 the heat transfer coefficient calculated on the basis of (1) coincides with the heat transfer coefficient for a smooth plate without injection. This result has already been obtained by Kutateladze and Leont'ev in their boundary-layer theory of injection cooling [2], as well as by other investigators [3, 4].

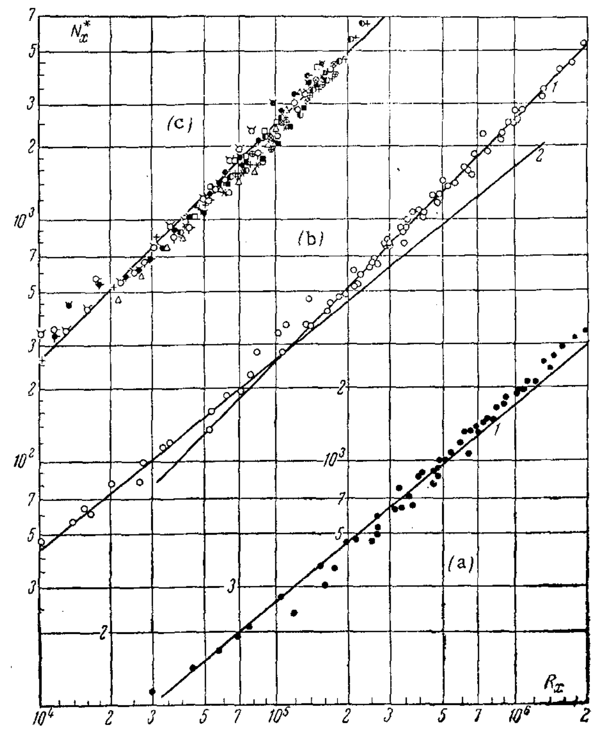

Fig. 2

3. In the experiments with the tubular surface the heat flux q was calculated on the basis of the whole wetted surface. Figure 2(b) shows the results of experiments with s = 0. In the range of Reynolds numbers R_x between $5 \cdot 10^4$ and $2 \cdot 10^6$ the experimental data are correlated by the formula (curve 1)

$$N_x^* = 0.0026 R_x, \qquad (3.1)$$

i.e., the heat transfer coefficient α^* is independent of the linear dimension. The points which satisfy (3.1) are located at distances $x/\Delta > 6$ from the upstream edge of the panel. The first few tubes are correlated by the formula $N_x^* = 0.0263\, R_x^{0.8}$ (curve 2). An analogous type of roughness was investigated by Nunner [5] in the case of a round tube. Nunner's results also indicated a very weak dependence of the heat transfer coefficient on the linear dimensions ($\sim d^{-0.07}$) under conditions of fully developed turbulence. In his experiments the Stanton number $S = \alpha^*/\gamma C_p W$ was, to within 10%, $3.7 \cdot 10^{-3}$, which coincides with the value of the Stanton number in the present experiments. It seems that beginning from some value of the ratio of boundary layer thickness to tube radius (which is a measure of roughness), the heat transfer process is determined only by the nature of the flow near the tubes and becomes self-similar with respect to the boundary layer thickness. The region of self-similarity is established very fast, on the first few tubes.

Point	s, mm	W_∞ m/sec	W_0, m/sec	m
1	3.5	19.7	10	0.41
2	3.5	22	9.7	0.44
3	6.5	21.7	13.1	0.42
4	6.5	21.1	15.7	0.74
5	13	18	6.8	0.29
6	13	29.8	16.9	0+47
7	13	29.2	26.3	0.75
8	3.5	16.7	7.1	0.36
9	3.5	16.7	6.1	0.37
10	3.5	10.1	7.7	0.76
11	3.5	18.5	20	0.92
12	2	29.7	9.8	0.33
13	6.5	29.2	9.4	0.32
14	6.5	15.6	9	0.57
15	13	21.4	4.6	0.21
16	13	20.2	6.3	0.26
17	13	20.3	10.1	0.5
18	13	20.1	11.2	0.56
19	13	19.9	14.9	0.75
20	13	20	20.9	1.05

Figure 2(c) shows experimental data on heat transfer to a tubular surface with a tangential jet at $m \le 1$. The governing parameters of the experiments are given in the table. The heat transfer coefficient was calculated from (2.1), as in the case of a smooth surface. The values of t_{aw} were taken from Fig. 8 of [1]. The points with $x/s > 30$ are shown in the figure. The solid line represents (3.1). It can be seen that even in the case of a surface with large-scale roughness the heat transfer coefficient for the case with a wall jet, as defined by (2.1), practically coincides with the heat transfer coefficient without injection.

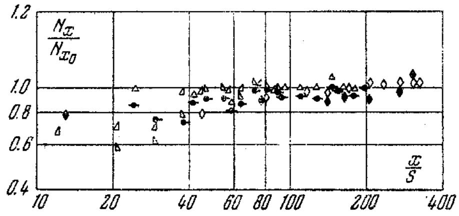

Fig. 3

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