HEAT TRANSFER FOR A PLANAR AIR JET STRIKING A CONCAVE SURFACE

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Results are presented from an experimental study of the local and mean heat transfer rates when a rectangular jet enters a relatively short concave body of parabolic profile.

Appropriate experimental evidence is lacking, so it is usual to calculate the heat transfer for a jet entering a concave region via relationships deduced for circular and rectangular jets striking planes perpendicular to the flow [1, 2]. It has been shown [3] that this approach can produce considerable errors for the heat-transfer coefficients, even as averaged over the surface.

These experiments were done with a jet striking a parabolic concave slot having R = 5 mm for the radius of the circle inscribed at the vertex (Fig. 1), which had been machined from a rod of tufnol 60 mm thick. This concave space had $h/H = 2.5$, and in the plane of symmetry were placed rectangular nozzles having waisting factors of 2.8-4.0, widths b of 2.5-3.5 mm, and distances h to the critical point of 5.2-32.5 mm. Parallel to the axis there were grooves containing copper-constantan thermocouples 0.1 mm in diameter. The surface bore four strips of permalloy 0.1 mm thick. The heat-transfer coefficients were determined at the midpoints of these strips, 17 mm wide, which had identically disposed thermocouples. The

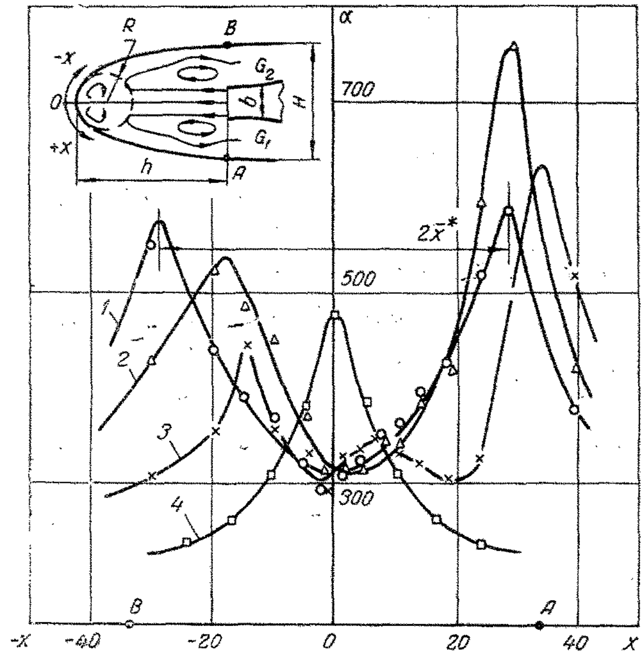

Fig. 1. Typical distribution of the local heat-transfer coefficients, W/m^2-deg, over a parabolic concave surface (distances in mm, $Re_0 = 2.5 \cdot 10^{-4}$, h = 11.2) receiving a rectangular jet with k of: 1) 1.0, 2) 0.71, 3) 0.64; 4) rectangular jet striking a plate perpendicular to the flow [1].

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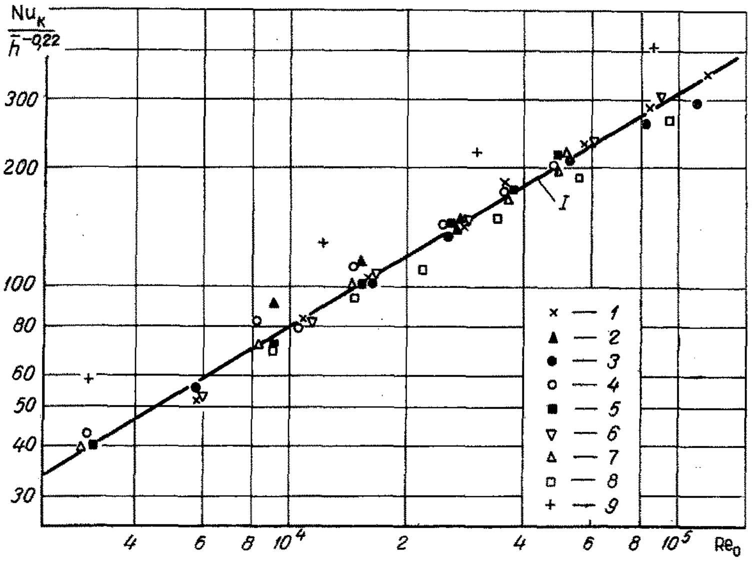

Fig. 2. Heat transfer at the critical point on a concave surface for: 1) $\bar{h}$ = 2.64; 2) 4.17; 3) 4.89; 4) 7.2; 5) 11.2; 6) 11.4; 7) 13.0; 8) h/d = 2.11 (round jets); 9) $\bar{h}$ = 14.0 [1]; I) from (1).

two edge strips acted as thermal guards. The current needed in each strip was measured with a grade 0.5 wattmeter of type D-529, while the thermocouple emf was recorded with a PPTN-1 potentiometer employing a M 195/2 meter as null detector. The air temperature at the entry to the nozzle was monitored by copper-constantan thermocouples. At the exit from the device was a mixing chamber having a perforated copper disc that gave the mass-mean temperature.

Ten thermocouples were used to monitor the heat loss from the outside of the device, as well as an ITP-4 heat-loss indicator [4]. The two methods of measuring the loss agreed very closely. There was 4-15% difference between the amounts of heat (as corrected for the loss) as deduced from the electrical input and from the enthalpy change in the air between inlet and outlet.

The air speed at the exit from the nozzle ranged from 20 to 320 m/sec. The surface temperature did not exceed 80°C.

Most of the nozzles provided a symmetrical distribution of the air flow after division of the jet at the critical point ($k = G_1/G_2 = 0.9$-1.1).

Two of the nozzles were such as to give k of 0.71 and 0.64 (unsymmetrical flow division). Also, the rectangular nozzle in one case was replaced by one with three holes 5.6 mm in diameter whose axes were 16 mm apart.

Figure 1 shows that the local heat-transfer coefficient has a characteristic U distribution with a pronounced minimum at the critical point and two peaks at a certain distance between the critical point and the end of the nozzle.

Curve 1 is for the case k = 1.0, with the plane of symmetry of the jet coincident with that of the slot. If the nozzle is turned through 5° relative to the longitudinal axis, the curve remains U-shaped but the peaks become unequal, with a rise to the side to which the nozzle is turned and a fall on the other side (curve 2). Larger angles produce a third peak, evidently because of marked flow constriction by the nozzle itself (curve 3).

Curves 1, 2, and 4 show that the distribution of the local heat-transfer coefficient α differs substantially as between a shallow concavity and a plate perpendicular to the flow, which is due to the hydrodynamic conditions in the two cases. A plate turns the jet through about 90°, whereas a shallow concavity produces nearly twice this angle, and so one expects a stagnant zone near the critical point in the latter case. Also, the flow speed is increased appreciably after rotation through 180°, in part from reduction in the effective area of the boundary layers at the boundary between the oppositely directed flows.

The following empirical relation represents the heat transfer at the critical point (Fig. 2) to ±10%:

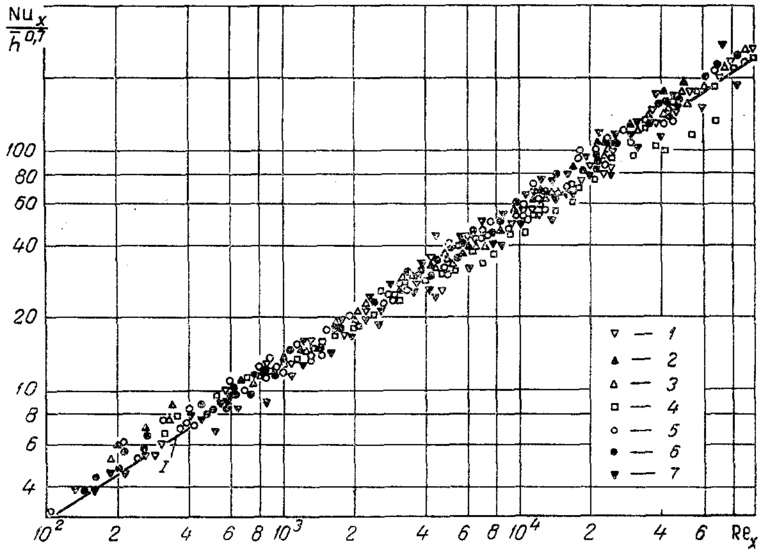

Fig. 3. Local heat transfer on a concave surface: $\bar{h}$ = 2.64; 2) 4.17; 3) 4.89; 4) 7.2; 5, 6) 11.2; 7) 13 (for k = 0.71); I from (2) and (2a).

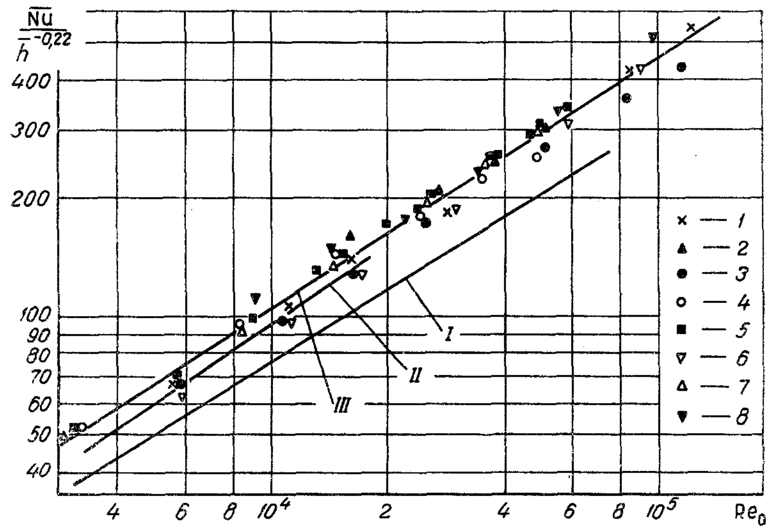

Fig. 4. Mean heat transfer on a part of length $2\bar{x}^*$: 1) $\bar{h}$ = 2.64; 2) 4.17; 3) 4.89; 4) 7.2; 5) 11.2; 6) 11.4; 7) 13.0; 8) h/d = 2.11 (round jets); I from $\overline{Nu} = 0.298\, Re_0^{0.62} \bar{h}^{-0.31}$ for $\bar{h} > 7$ [1]; II from $\overline{Nu} = 0.14\, Re_0^{0.7} \bar{h}^{-0.22}$ for $\bar{h} < 8$ [7]; III from (4) and (4a).

$$Nu_k = 0.31\, Re_0^{0.6} \bar{h}^{-0.22}\,, \tag{1}$$

which applies for $3 \cdot 10^3 \leq Re_0 \leq 1.2 \cdot 10^5$ and $2.6 \leq \bar{h} \leq 13$. The decisive quantity in Nu_k and Re_0 is the hydraulic diameter of the rectangular nozzle; the speed was calculated for the end of the nozzle, while the physical parameters were deduced from the air temperature at the exit from the nozzle.

Figure 2 shows results [1] for heat transfer at the critical point when a rectangular jet strikes a plate perpendicular to it. The results for equal $\bar{h}$ show that the stagnant zone near the critical point in the concavity reduces the heat-transfer rate by about 30% relative to that at the same point on a plate.

Figure 2 also shows results for a nozzle with three circular holes; as definitive dimension in Nu_k and Re_0 we took [5] the separation (16 mm) of the holes. No systematic study has been made on a system of round jets entering a concavity, so we can only assume that there is no essential change in the form of the equation; but the heat-transfer rate in the jet system is higher by about 40% than that in a rectangular jet having the same flow rate and same distance (h = 11.5 mm) to the critical point.

We processed the results on α for the concavity on the assumption that the flow speed varies linearly from zero at the critical point to its peak value at the section coincident with the end of the nozzle (Fig. 1).

It is clear that this is justified from results [6] on the speed along a stream line at the wall of a rectangular concavity, and the decisive dimension in this case is the distance x along the concave surface from the critical point. Figure 3 shows that about 80% of the points for the ranges $10^2 \le Re_x \le 10^5$ and $2.6 \le \bar{h} \le 13$ can be represented as follows to ±15%:

$$\text{for } \bar{h} > 4.7 \quad Nu_x = 0.167\, Re_x^{0.63} \bar{h}^{0.7}, \tag{2}$$

$$\text{for } \bar{h} \leqslant 4.7 \quad Nu_x = 0.493\, Re_x^{0.63}. \tag{2a}$$

There is an elevated spread at high Re_x on account of error in determining the peak α. Also, (2) and (2a) describe satisfactorily the distribution of α for k = 0.71 (unsymmetrical jet division).

Our conditions gave $\bar{x}^*$ (the coordinate for the peak α) as

$$\bar{x}^* \simeq 1.45\, \bar{h}^{0.83}. \tag{3}$$

We averaged the α between the peaks over a range of $2\bar{x}^*$. All the observed points for the mean transfer are shown in Fig. 4 and may be represented by

$$\text{for } \bar{h} > 7 \quad \overline{Nu} = 0.26\, Re_0^{0.65} \bar{h}^{-0.22}, \tag{4}$$

$$\text{for } \bar{h} \leqslant 7 \quad \overline{Nu} = 0.17\, Re_0^{0.65}. \tag{4a}$$

These equations are correct for the ranges of the parameters used in (1).

Figure 4 shows the results from [1] and [7], as converted to the definitive parameters used in $\overline{Nu}$ and Re_0, in order to compare the mean transfer rates for concavity and plate. The mean transfer rate in the former case is 35-40% greater than that in the latter [1].

The results of [7] lie within the spread of our values.

The above equations have been derived for a shallow concavity having a radius of curvature of 5 mm at the vertex, whose curvature may be characterized by 2R/H. This curvature may affect the local and mean heat-transfer rates. Additional experiments are needed to elucidate this point and other features of the hydrodynamics of jets.

NOTATION

b	is the nozzle width;
d	is the diameter of the round nozzle;
$\bar{h}$	is the distance from the nozzle to the critical point as a ratio to nozzle width;
H	is the width of the concave slot;
R	is the radius of curvature at the top;
x	is the curvilinear coordinate on the surface;
$\bar{x}^*$	is the dimensionless coordinate of the peak local heat-transfer coefficient;
α	is the local heat-transfer coefficient;
G	is the air flow rate;
Re_0	is the Reynolds number calculated from the exit speed and the hydraulic diameter of the nozzle;
Re_x	is the Reynolds number calculated from the local flow velocity and the current x;
Nu_k	is the Nusselt number calculated from the hydraulic diameter of the nozzle for the critical point;
Nu_x	is the Nusselt number calculated from the current x;
$\overline{Nu}$	is the Nusselt number calculated from the average heat-transfer coefficient and the hydraulic diameter of the nozzle.

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