HEAT TRANSFER ON THE INLET SECTION OF A TUBE AT INCREASED LEVELS OF INITIAL FLOW TURBULENCE

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The authors report the results of an experimental study of heat transfer on the initial section of a tube with uniform velocity profile at the inlet and a degree of flow turbulence up to 20% in the range of Reynolds numbers from 10^4 to 10^5.

The heat transfer on the initial sections of tubes in the presence of a turbulent air flow is usually calculated with the aid of correction coefficients ε_l and $\bar{\varepsilon}_l$ determined as the ratio of the heat transfer coefficients or corresponding Nusselt or Stanton numbers in a given cross section (for local heat transfer) or on a given length of tube (for average heat transfer) to their stabilized-flow values (see, for example, [1–4]).

It has been shown [2, 3] that the flow inlet conditions have an important influence on the coefficients ε_l and $\bar{\varepsilon}_l$, and recommendations have been made regarding some of the most commonly used inlet devices.

On the basis of general ideas concerning the mechanism of boundary layer formation on the initial section of a tube it may be assumed that the geometry of the inlet device affects the rate of heat transfer primarily by modifying the flow velocity profile and the degree of turbulence.

We have attempted to establish a quantitative relation between the initial level of flow turbulence and the values of the heat transfer coefficients on the inlet section.

The experiments were conducted on the apparatus described in [4] using a redesigned inlet device whose chief element is a Vitoshinskii nozzle with a contraction ratio of 9.

The results of experiments on heat transfer in a tube with a smoothed velocity profile and a low degree of turbulence (less than 0.5%) were reported in [7]. The results obtained when the turbulence is varied from 1 to 18% are presented below.

The level of turbulence was controlled by installing in the inlet section of the header (and not the tube as in [4]) a perforated plate with openings 13.5 mm in diameter, varying in number from 36 to 6.

The initial level of turbulence was determined in a section at a distance of one diameter from the inlet and estimated from the mean-square relative fluctuation of the axial velocity averaged over the area of the inlet cross section. This choice of turbulence criterion was conditioned by the dominant role of the longitudinal component on the initial section [5].

The axial velocity fluctuations were measured with a hot-wire anemometer operating in the constant-current mode with single-wire probes using tungsten wire 5–6 μ in diameter. The anemometer calibration curve was constructed as described in [6].

As the measurements showed, in the inlet cross section of the tube (l/d = 1) the velocity profile was only slightly affected by changing the number of openings in the perforated plate. The ratio of the maximum velocity in the inlet section to the mean flow-rate value was not less than 0.9 at a boundary layer thickness of the order of 1 mm.

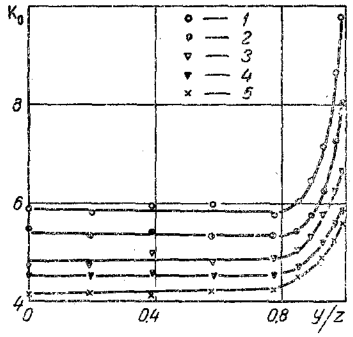

Fig. 1. Typical distribution of axial component of velocity fluctuations in the inlet section of the tube (l/d = 1) in the presence of artificially created turbulence (perforated plate with 12 openings 13.5 mm in diameter): 1) W_{max} = 5.1 m/sec; 2) 7.4; 3) 12.2; 4) 17.5; 5) 23.8.

The radial distribution of the relative mean-square velocity fluctuations was almost independent of the number of openings in the plate and of the absolute level of the velocity and the fluctuations (Fig. 1). As the level of flow turbulence increased, the relative velocity fluctuations also grew both in the flow core and in the boundary layer. In the experiments we distinctly observed a decrease in the relative fluctuations with increase in Reynolds number with a simultaneous increase in the absolute value of the fluctuation. The following values were obtained for the Karman number, averaged over the cross section, at Reynolds numbers of 10^5 and 10^4, respectively (in %): without the perforated plate—less than 0.5; with 36 openings—1–3; with 12 openings—4–7; with 18 openings—6–11; with 9 openings—9–12; with 6 openings—13–18.

The local and mean values of the heat transfer coefficients were determined using the measuring apparatus described in [4]. The method of analyzing the data is discussed in [7].

As the experiments showed, in the range of Reynolds numbers 10^4–10^5 increasing the initial level of turbu-

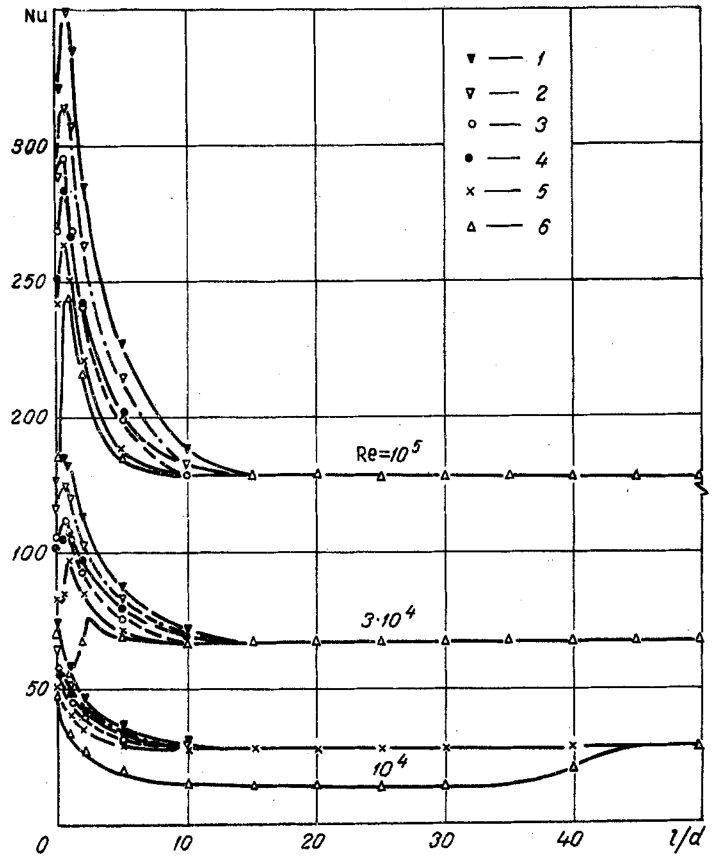

Fig. 2. Variation of level of local heat transfer along length of tube at increased levels of initial flow turbulence (in %): 1) $\overline{K}_0$ = 13–18; 2) 9–12; 3) 4–7; 4) 6–11; 5) 1–3; 6) ≤0.5.

lence leads to a marked increase in the rate of local heat transfer on the initial sections of the tube (Fig. 2).

For example, at Re = 10^4 and $\bar{K}_0$ = 18% up to l/d = 35. the local heat transfer coefficient is 70% higher than for a flow with $\bar{K}_0 \le 0.5\%$, whereas at Re = 10^5 and K $\bar{K}_0$ = 13% in the section l/d = 1 the increase in the local heat transfer coefficient is only 37%. This increase at Re = 10^4 is associated chiefly with elimination of the mixed boundary layer as the turbulence increases and the development of turbulent flow over the entire tube.

The erosion of the mixed boundary layer is indicated by the monotonic increase in the wall and air temperatures without the "maxima" and "minima" observed at a low level of turbulence [7]. At Re = $3 \cdot 10^4$ the introduction of artificial perturbations is not capable of completely changing the original flow structure, and near the inlet (l/d = 0.5–0.9) there is a transition with a sharp change of wall temperature. However, on the rest of the initial section the heat transfer coefficients decrease monotonically to the values corresponding to stabilized flow.

On the stabilized-flow section ($l/d > (l/d)_t$), over the entire investigated range of values of the Karman number in the inlet section, the Stanton numbers are in good agreement with the known generalization.

$$St_\infty = 0.0257\ Re^{-0.2}. \tag{1}$$

The experimental data, obtained with an error of not more than ±10%, are described by the following empirical relations:

for local heat transfer at $1 \le l/d \le (l/d)_t$:

$$\varepsilon_l = \frac{1.35 + 0.04\bar{K}_0\%}{(l/d)^{0.17+0.006\bar{K}_0\%}}; \tag{2}$$

for average heat transfer at $1 \le l/d \le 5$:

$$\bar{\varepsilon}_l = \frac{1.3 + 0.05\bar{K}_0\%}{(l/d)^{0.07+0.005\bar{K}_0\%}}; \tag{3}$$

for average heat transfer at $l/d \ge 5$:

$$\bar{\varepsilon}_l = 1 + \frac{0.8 + 0.15\bar{K}_0\%}{l/d}. \tag{4}$$

The length of the thermal stabilization section is then given by the following equations:

for local heat transfer (when $\varepsilon_{l\infty} = 1$):

$$(l/d)_t = (1.35 + 0.04\bar{K}_0\%)^{\frac{1}{0.17+0.006\bar{K}_0\%}}; \tag{5}$$

for average heat transfer (when $\bar{\varepsilon}_{l\infty} = 1.02$):

$$(l/d)_t = 40 + 7.5\bar{K}_0\%. \tag{6}$$

Thus, the experiments show that the initial level of flow turbulence has an important effect on heat transfer on the initial section of the tube.

Further research should be directed toward establishing a relation between the Karman number and the geometry and operating conditions of the inlet device.

NOTATION

Re is the Reynolds number based on the tube diameter; Nu is the Nusselt number; St is the Stanton number; $K_0 = \Delta w/W$ is the Karman number, %; ε_l and $\bar{\varepsilon}_l$ are the correction coefficients for tube length; d is the tube diameter, m; l is the distance from inlet or relative length of tube for local and average heat transfer, m; W is the averaged flow velocity at a given point, m/sec; Δw is the mean-square fluctuation of axial velocity, m/sec; the subscript t stands for thermal.

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