HEAT TRANSFER AT THE INITIAL SEGMENT OF A TUBE UNDER CONDITIONS OF NATURAL AIRSTREAM TURBULIZATION

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The quantitative relationships governing heat transfer and the features in the development of a mixed boundary layer at the initial segment of a tube are considered for the case in which a collector designed on the basis of the Vitoshinskii curve is mounted at the inlet.

Most recommendations in the literature on calculating the transfer of heat in the initial segment of a tube (see, for example, [1]) pertain to the case of monotonic reduction in the heat-transfer coefficient over tube length. Variation in heat-transfer intensity is accounted for in this case through correction factors ε_l and $\bar{\varepsilon}_l$ representing the ratio of the heat-transfer criteria (or coefficients) at the initial segment relative to their values in stabilized flow.

At the same time, a number of references [2–6] suggest the possibility of establishing a laminar layer at the forward end of the tube, this layer subsequently being transformed into a turbulent layer with increasing distance from the inlet. The existence of a mixed boundary layer is responsible for a nonmonotonic variation in the heat-transfer coefficients over the length of the initial segment.

It is only in [3] that we find recommendations for the calculation of the average heat transfer in the presence of a mixed boundary layer at the initial segment of the tube, but to be able to use these recommendations, we must have at our disposal data on the coordinates for the transition point which serves in the place of the region of transition from laminar to turbulent flow.

At the same time, it was demonstrated in [7] that the extent of the transition region and the stability of the laminar flow are functions not only of the Reynolds number (proportional to the mean flow velocity), but also of the degree of flow turbulization; the transition region is shifted downstream as the degree of flow turbulence diminishes and it is shifted upstream as the turbulence is increased.

The results given in [2, 3, 5, 6], as well as our earlier experiments [4], indicate the need for further study of the effect exerted by the level of the initial flow turbulence on heat transfer in the initial segment of the tube.

For the purposes of this investigation we used the test stand described in detail in [4]; the input device of this installation was removed and replaced with one designed on the basis of the Vitoshinskii curve.

The velocity profile with such an inlet device was completely straightened across the inlet cross section ($l/d = 1$) and the thickness of the boundary layer over a velocity range of 4–35 m/sec did not exceed 1 mm.

The level of the turbulent fluctuations in velocity at the flow core was below the sensitivity limit of the electrothermoanemometer (0.5% when $\mathrm{Re} = 10^4$ and 0.2% when $\mathrm{Re} = 10^5$)(the experiments were carried out with an electrothermoanemometer using dc current, with a tungsten-filament sensor 5 μm in diameter and 2.5 mm in length. The thermoanemometer employed an amplifier with a maximum amplification factor of $1.6 \cdot 10^4$). Discrete low-frequency bursts (up to 3%) were seen only near the wall. The mean-square axial pulsation is tentatively estimated at approximately 0.5%.

The heat-transfer coefficients were determined for this tube over a Reynolds number range $1.5 \cdot 10^3$–10^5. The tube-wall temperature was measured at 18 points along the length, while the average airstream temperatures at these points were determined by calculation—from the temperature measured at the inlet and from the heating of the air in the subsequent segments. The specific heat flow through the tube wall was determined on the basis of the difference between the electrical load per unit heating surface and the heat losses to the ambient space as a result of the convection and radiation calculated in accordance with the recommendations found in [1].

The resulting temperature differences for the walls and air over the length of the tube were used to calculate the heat-transfer criteria (the Stanton or Nesselt numbers).

An indication of a change in flow regime over the length of the initial segment is the nonmonotonic change in the heat-transfer coefficient (Fig. 1) noted in our research over the Reynolds number range 0.83–$4.8 \cdot 10^4$ at points from 40 to 1 diameters from the inlet. The transition from one flow regime to another shows up also in the form of "maxima" and "minima" of wall temperatures.

As the Reynolds number increases, the extent of the transition zone along the length of the tube initially increases sharply, reaching a maximum (20 diameters) when $\mathrm{Re} = 1.5 \cdot 10^4$, and then diminishing.

The above-described quantitative relationship governing the development of the boundary layer is confirmed qualitatively for certain other conditions of flow entry into the tube. In Fig. 2 we see similar results [4], derived on entry of the flow into the tube through a collector with a radius of 170 mm, behind which was mounted a conical device 75/51 mm in diameter. An inlet device of this type ensures a rather uniform velocity profile $w/\overline{w_{\max}} = 0.93$ for a thicker boundary layer (up to 2 mm) in the inlet section and with approx-

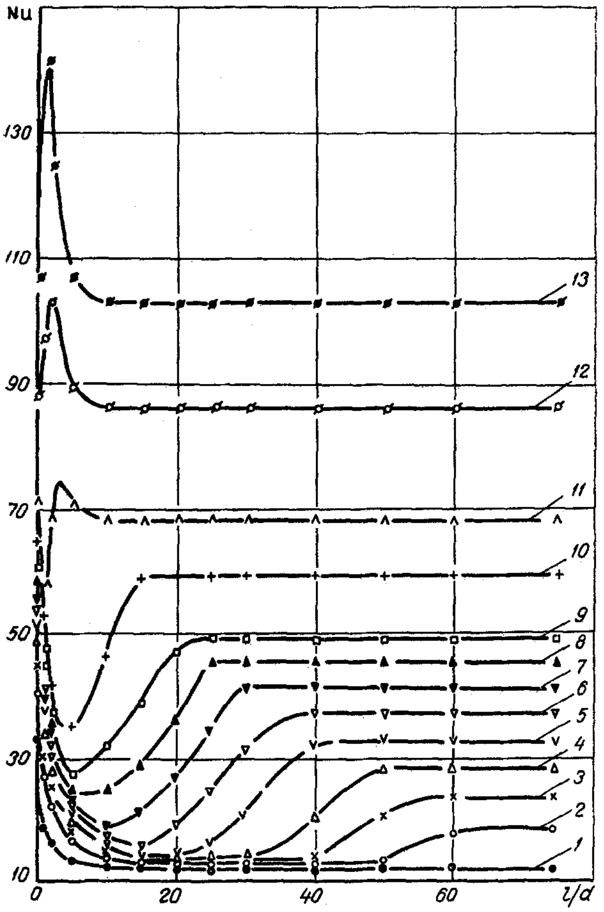

Fig. 1. Variation of local Nusselt number along tube length with natural flow turbulization: 1) Re $= 3 \cdot 10^3$; 2) $6 \cdot 10^3$; 3) $8 \cdot 10^3$; 4) 10^4; 5) $1.2 \cdot 10^4$; 6) $1.4 \cdot 10^4$; 7) $1.6 \cdot 10^4$; 8) $1.8 \cdot 10^4$; 10) $2.5 \cdot 10^4$; 11) $3 \cdot 10^4$; 12) $4 \cdot 10^4$; 13) $5 \cdot 10^4$.

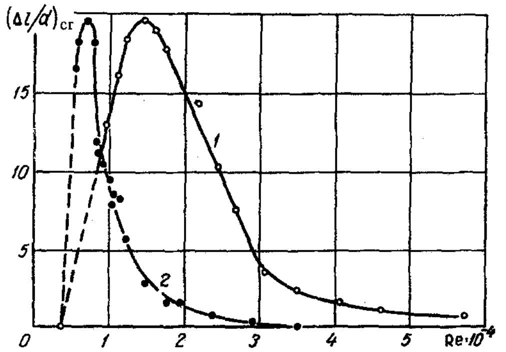

Fig. 2. Influence of Reynolds number on zone length with transitional flow regimes under conditions of natural flow turbulization: 1) according to data of present study; 2) according to [4].

Transition-zone Boundaries and Values for the Correction Factors ε_l and $\bar{\varepsilon}_l$ in Turbulent Flow Regimes

Heat transfer	Quantity	l/d						
		1	2	5	10	20	40	75
Local	$\mathrm{Re}_{\mathrm{trans}}$	30000 48000	26000 39000	19000 30000	15500 26000	12600 20200	8300 12000	3640
	ε_l	1.37	1.21	1.03	1.00	1.00	1.00	1.00
Average	$\mathrm{Re}_{\mathrm{trans}}$	33000 52000	29000 50200	22400 44000	16800 38000	13600 32000	10300 26800	7300 25000
	$\bar{\varepsilon}_l$	—	1.25	1.16	1.08	1.05	1.02	1.00

imately the same level of turbulent fluctuations in velocity.

The maximum length of the transition region in this case also amounted to 20 diameters, but was seen at substantially smaller Reynolds numbers—on the order of $6.5 \cdot 10^3$. For Reynolds numbers higher than $3.5 \cdot 10^4$, turbulent flow was set up over the entire length of the tube, whereas in this study a mixed boundary layer existed even when $\mathrm{Re} = 10^5$.

We should point out the difficulty and a certain arbitrariness in establishing the boundaries for the transition zones, whether on the basis of the Reynolds number, or with respect to tube length. For local heat transfer the boundaries of the transition zone can be determined either from the points of the "maxima" and "minima" of the wall temperatures, or from the points at which the straight lines $\mathrm{Nu} = f(\mathrm{Re})$ or $\mathrm{St} = f(\mathrm{Re})$ (in a logarithmic coordinate system) intersect for various flow regimes. The second method may be used also to determine the extent of the transition zone for average heat transfer. The magnitudes of the critical Reynolds numbers derived in this manner are presented in the table.

The experimental data derived for turbulent flow are virtually coincident with the known generalization [1]

$$\mathrm{Nu}_\infty = 0.018\,\mathrm{Re}^{0.8}. \qquad (1)$$

The correction factors ε_l and ε_l, needed to calculate the heat transfer at the initial segment of the tube under conditions of turbulent flow, are presented in the table.

The extent of the thermal-stabilization segment for local heat transfer amounted only to 6 diameters, extending to 50 diameters for average heat transfer, which is in agreement with the results of [2, 5, 8].

The studies that we carried out confirmed that to ensure the possibility of a sufficiently accurate calculation of heat transfer in the case of a mixed boundary layer in the initial segment of a tube, we must have at our disposal reliable data on the relationship between the boundaries of the transition region and the parameters of the flow in the inlet section or on the relationship between the former and the conditions for entry into the channel.

Our further studies must therefore be directed, first of all, at the establishment of these relationships.

Here we were able to derive the data needed to calculate heat transfer in a mixed layer for a uniform velocity profile at the tube inlet in the case of a low degree of initial flow turbulization (on the order of 0.5%).

For laminar and turbulent flow regimes it is possible to calculate heat transfer on the basis of generally accepted recommendations. The calculation of heat transfer in the transition region can be accomplished on the basis of an interpolational straight line (in a logarithmic coordinate system) drawn through the values of the heat-transfer criteria (Nusselt, Stanton) at the critical Reynolds numbers for specified distances from the inlet or for given tube lengths.

NOTATION

Re is the Reynolds number based on tube diameter and mean velocity; Nu is the Nusselt number; St is the Stanton number; d is the tube diameter, m; l is the distance from input or tube length for local or mean heat transfer, m; $\bar{w}$ is the mean flow velocity, m/sec; $w_{\max}$ is the maximum velocity in the cross section, m/sec.

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