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FEATURS OF HEAT TRANSFER IN A FLOW OF AIR IN A PLANE CHANNEL WITH UNSTAGGERED HALF-CYLINDRICAL PROJECTIONS

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A study was made of local heat transfer in a flow of air with Re = $(1.5-170)\cdot 10^3$, $s_1/d = 1.27$; $s_2/d = 5.33$; 3.04; 2.13. It was found that heat transfer is nonsymmetrical on opposite sides of the channel.

In boiler design, great importance is attached to questions related to intensifying heat exchange and ensuring the reliable operation of convective heating surfaces. Experience accumulated in the operation of such surfaces shows the significant design, processing, and operational advantages of membrane panels used as convective and shielding elements. The studies [1-5] present data from investigations of local heat transfer, this data showing that to ensure reliable operation of a membrane structure, it is necessary to consider the effect of the nonuniform distribution of heat transfer on the temperature regime of the surface.

As in normal bundles, the pipes in such systems can be arranged in staggered or unstaggered fashion. The use of a given arrangement is dictated by the flow conditions and should be substantiated by special technicoeconomic calculations for each case. For example, it was noted in [6] that several design problems are encountered in achieving "economical" flue-gas velocities, but the authors also noted that these problems can be overcome by using unstaggered membrane bundles.

Calculations performed in [7] showed that an unstaggered bundle of tubes with solid fins is more efficient than a similar bundle of smooth tubes in the range $Re_d \leq 10^4$.

Along with this, as in the case of flow over a straight double ledge [8], flow and heat transfer may be nonsymmetrical under certain conditions in the case of a membrane surface with an unstaggered tube arrangement. These effects may have a significant influence on the temprature regime of the membrane panel.

These considerations interested us in taking a closer look at membrane heating surfaces with an unstaggered tube arrangement. To do this, we reconstructed the test section described in [7]: measurements were made only for an unstaggered arrangement of the projections, with a transverse relative step $s_1/d = 1.27$ and three lengthwise steps $s_2/d = 5.33$, 3.04, and 2.13.

Heat transfer was measured on a flat wall and on semicylindrical calorimeters for both sides of the channel. Although realization of this reconstruction required many design

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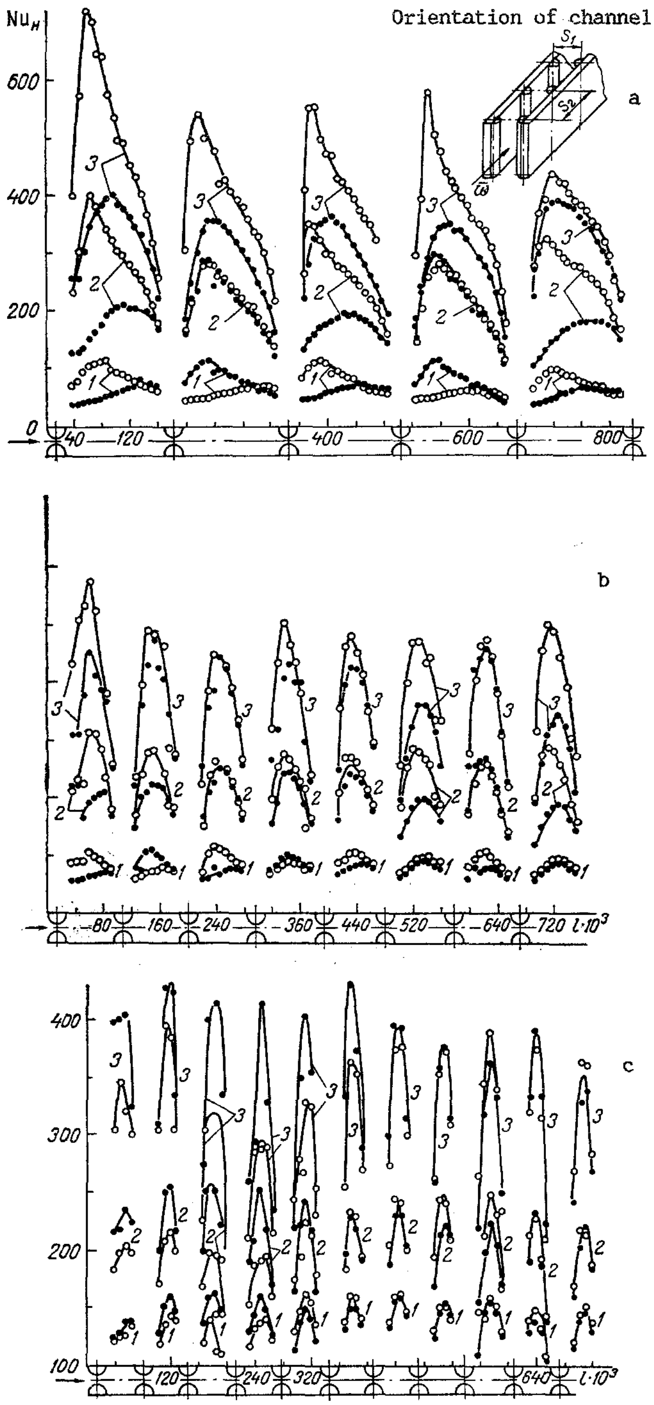

Fig. 1. Distribution of heat transfer along flat surfaces: a) $s_2/d = 5.33$; 1) $Re_n = 5200$, 2) 26,400, 3) 46,500; b) $s_2/d = 3.04$; 1) $Re_n = 5200$, 2) 23,500, 3) 46,700; c) $s_2/d = 2.13$; 1) $Re_n = 14{,}500$, 2) 24,300, 3) 45,100, ℓ, m.

changes, heat release conditions remained the same as before. The study [7] presented results of an evaluation of heat transfer in plane and semicylindrical surfaces. There was no redistribution of the warming electric current due to the temperature dependence of electrical resistivity for the plate. For the half-cylinders, the redistribution was estimated as being no more than $\sim 1\%$. For the semicylindrical calorimeters, q_{pot} was determined as $q_{pot} = f\ (\Delta t = t_c - t_{bse}) = (\lambda/\delta)\Delta t$, where Δt is the temperature difference between the heater t_c and textolite base (t_{bse}). Thermal conductivity λ/δ was determined from special calibration tests in which a semicylindrical calorimeter was located in the plane temprature field of the heating plate. In this case, for a half-cylinder at $\varphi° = 90°$, we can determine the value of λ/δ for an annular layer from the known heat flux and the measured temperature difference $\Delta t = (t_c - t_{bse})$, just as for a plane layer of the thickness δ. For other values of $\varphi°$, we took the quantity q_{pot} to have the same value as at $\varphi° = 90°$. This did not produce large errors, since q_{pot} is small relative to $q_{e\ell}$ ($\sim$4%). Heat losses for the plane channel were determined by the same method as used in [7].

Heat transfer on the flat walls of the channel was measured after each semicylindrical projection. Here, the number of projections was different for different lengthwise steps: $n = 6$ at $s_2/d = 5.33$; $n = 9$ at $s_2/d = 3.04$; $n = 12$ at $(s_2/d) = 2.13$.

The semicylindrical calorimeters were positioned a fixed distance from the inlet. This distance corresponded to 5th, 8th, or 11th row, depending on the lengthwise step. We assumed that heat transfer was stabilized on the half-cylinders for these rows, since, according to the data in [9], stabilization begins with the 5th row for unstaggered membrane bundles.

Heat transfer on the semicylindrical calorimeters was measured by the method of local thermal modeling. It follows from [10, 11] that in the case $s_2/d \geq 2$ and when the mixing temperature in front of the calorimeter is chosen as the flow temprature, this method yields the same result for heat transfer as the method of complete thermal modeling.

Figures 1 and 2 show test data on heat transfer for the plane and semicylindrical surfaces. The following conclusions can be made from the figures.

For $s_2/d = 5.33$

1) flow and heat transfer on the flat channel surface are asymmetrical; asymmetrical flow is manifest in short and long attachment of the flow, while asymmetrical heat transfer is manifest in different levels of heat transfer on opposite walls of the channel; heat transfer is higher with short attachment and lower with long attachment.

2) the asymmetrical flow and heat transfer on the flat walls are related to the corresponding asymmetry on the cylinders;

3) at low and moderate Reynolds numbers Re, short and long attachment of the flow on the walls alternate along the channel, i.e., wavy flow occurs in the channel.

4) the wavy flow degenerates as Re increases and disappears at large Re, i.e., short attachment always occurs on one wall and long attachment always occurs on the other wall;

5) heat transfer on the half-cylinders has a maximum at $\varphi° \sim 40\text{-}50°$ and a minimum at $\varphi° \sim 140\text{-}150°$; here, for short attachment, separation begins earlier, and heat transfer is higher in the rear region;

6) the character of heat transfer changes on the half-cylinders with an increase in Re; along with separation of the laminar boundary layer, there is a transition to more turbulent flow and later turbulent separation.

For $s_2/d = 3.04$

1) the character of flow and heat transfer at low Re is the same as with the step $s_2/d = 5.33$. However, at large Re, flow asymmetry disappears and only heat transfer remains asymmetrical; the position of the heat-transfer maxima (large and small) correspond roughly to the middle of the plane section;

2) the frontal heat-transfer maximum on the half-cylinders is shifted to $\varphi° \sim 0°$ with an increase in Re.

For $s_2/d = 2.13$

1) the flow is symmetrical at nearly all Re, and heat transfer is appreciably asymmetrical only at large Re. However, heat-transfer asymmetry in this case decreases over the rows;

2) the frontal heat-transfer maximum remains at $\varphi° \sim 40°$at all Re on the half-cylinders.

The asymmetry of heat transfer in a geometrically symmetrical channel can be explained as follows. The effect of free convection is excluded by the orientation of the channel in space. The plane horizontal channel was positioned so that both sides on which heat transfer was studied were perpendicular to the ground and were subject to the same conditions. In our opinion, the asymmetry of heat transfer was thus connected with asymmetry of the velocity

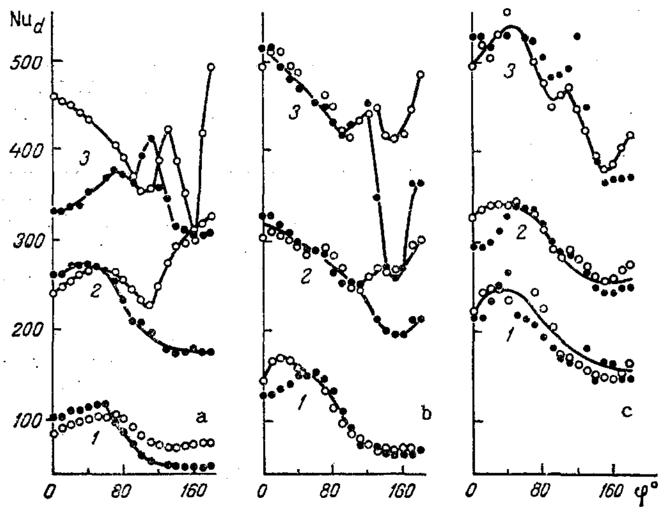

Fig. 2. Distribution of heat transfer on semicylindrical projections: a) $s_2/d = 5.33$; 1) $Re_d = 1860$, 2) 95,200, 3) 169,100; b) $s_2/d = 3.04$; 1) $Re_d = 18{,}700$, 2) 84,800, 3) 170,200; b) $s_2/d = 2.13$; 1) $Re_d = 52{,}200$, 2) 87,300, 3) 161,100.

profile and the turbulent structure of the flow. However, a more exact representation of the situation will be possible after further experimental studies of the flow structure.

In analyzing the data obtained on heat transfer in a plane channel with semicylindrical projections, it is important that we point out the following fact: the boundary layer (laminar or turbulent) separates from the surface of the half-cylinders and subsequently reattaches to the walls of the plane channel over short and long distances. This allows us to regard a channel with semicylindrical projections in the same light as channels having different types of ledges and projections — tubes with diaphragms, a plane channel with double ledges, etc.

The study [8] presented data on the lengths associated with short and long attachment of the flow as a function of the degree of separation in the flow of air in a symmetrical plane channel with a double ledge. In our tests, the degree of separation (the ratio of the maximum and minimum through sections of the symmetrical plane channel F/f), with allowance for the actual geometry of separation from the surface of the half-cylinder, was $F/f = 1.64$ for $s_2/d = 5.33$ and $Re_H = 26{,}400$. Here, the length of the short attachment $\ell_{sho} \cong 30$ mm, while the length of the long attachment $\ell_{\ell g} \cong 90$ mm (see Fig. 1). The analogous lengths obtained in [8] with the same degree of separation were equal to $\ell_{sho} = 35$ mm and $\ell_{\ell g} = 75$ mm.

The results in [8] also make it possible to evaluate the limiting permissible value of transverse spacing for the tubes of the membrane surface. Below this value, the flow and heat transfer becomes asymmetrical.

According to the data in [1], as in our case, the position of the separation region on the half-cylinder corresponds to $\varphi° \sim 145°$ for unstaggered membrane bundles with transverse steps $s_1/d = 1.52$ and $s_1/d = 3.05$. Using the data in [8] on flow asymmetry in a plane channel with a double ledge and taking $\varphi° \sim 150°$ for the position of the separation region on the half-cylinder, we obtain the limiting transverse step $(s\ /d)_{\ell imn} \cong 1.50$. Thus, with transverse steps $s_1/d > (s_1/d)_{\ell imn}$, asymmetry of flow and heat transfer should decay.

The experimental data currently available does not touch on this question, being limited to measurements of mean or local heat transfer at $s_1/d > 1.5$.

NOTATION

$Re_d = \bar{w}_4 d/\nu$, $Re_H = \bar{w}H/\nu$, Reynolds for the tube bundle and plane channel; d, H, diameter of semicylindrical projections and height of plane channel, m; $\bar{w}_{nar}$, $\bar{w}$, mean air velocity in the narrow section and between the plane walls of the channel, m/sec; ν, kinematic viscosity, m^2/sec; s_1/d, s_2/d, relative transverse and lengthwise steps; ℓ, length of short and long attachment of the flow to the plane surface, m.

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