EXPERIMENTAL STUDY OF HEAT TRANSFER IN A PLANE CHANNEL WITH SEMICYLINDRICAL PROJECTIONS

A. S. Sukomel, V. I. Velichko,
and V. A. Pronin

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An experimental study was made of local heat transfer in a plane channel with semicylindrical projections arranged in staggered and unstaggered fashion. The study was conducted in the range of Reynolds numbers $(1.5\text{-}170)\cdot 10^3$ for the air flow.

In channels with hard-to-streamline projections, flow separation may occur, and the flow may subsequently be attached to the boundary surface. Here heat transfer in the region of attached flow, according to the data in [1-3], may be about six times greater than outside this region for a rectangular projection. This increase in heat transfer is not related to an increase in frictional resistance on the wall. The frictional resistance, in fact, approaches zero at the attachment point.

The above situation draws attention to channels of complex geometry, in which flow separation and attachment may occur, for study of their possible application as the through portion of heat exchangers.

Figure 1a illustrates the basic design of the test section. In the case of unstaggered arrangement of the tubes, the relative transverse and longitudinal spacings were as follows: $(s_1/d)_{uns} = 1.21$, $(s_2/d)_{uns} = 5.32$; for a staggered arrangement of the tubes, the respective values were $(s_1/d)_{stag} = 2.42$, $(s_2/d)_{stag} = 2.66$. The range of Reynolds numbers for the unstaggered arrangement was $Re_d = (7\text{-}170)\cdot 10^3$, while the range for the staggered arrangement was $Re_d = (15\text{-}38)\cdot 10^3$.

The temperature of the hot surface of the plates and the half-cylinders was no higher than 50-60°C, which allowed us to ignore the effect of variability of the physical properties of the air and radiative heat transfer on overall heat transfer.

The local heat transfer on the plane surface and half-cylinder was measured with the boundary condition q_{wa} = const. The plate and half-cylinder were heated separately. The method of local thermal modeling was used to study heat transfer on the half-cylinder. Local heat transfer was measured about the perimeter of a single semicylindrical calorimeter located in the fourth (in the direction of gas travel) row of each tube arrangement.

The heating element for the semicylindrical calorimeter was a stainless-steel tube with ∅31/30 mm. The tube was mounted on a textolite base and equipped with copper—Constantin thermocouples, which were embedded in its central section (Fig. 1b). The heating element for the flat surface was six stainless-steel plates 1.5 mm thick and connected in series to an electrical circuit. Thirteen Chromel—Copel thermocouples were located along the central line of the middle plate. The two outermost plates served as protective heaters.

The local heat-transfer coefficient $\alpha = q_{conv}/(t_{wa} - t_{air})$, where t_{air} was found from the reading of the thermocouple at the inlet, with allowance for the supply of heat from the heating elements; $q_{conv} = q_{el} + q_{los} + q_{calc}$.

Heat loss to the environment for the plane channel q_{los} was determined in special calibration tests during which the test section was heated without an air flow [4]. For semicylindrical calorimeters, $q_{los} \approx 0$, since local heat transfer was measured on them during heating of the plates in the channel, which in this case served as protective heaters [4]. With a characteristic temperature distribution about the semicylindrical calorimeter for a test with Re_d = 30,145, in which q_{el} = 5000 W/m², the maximum value of q_{los} = 3.42% of q_{el}. The value of q_{los} decreased as Re increased and amounted to 0.5-2.5% of q_{el}.

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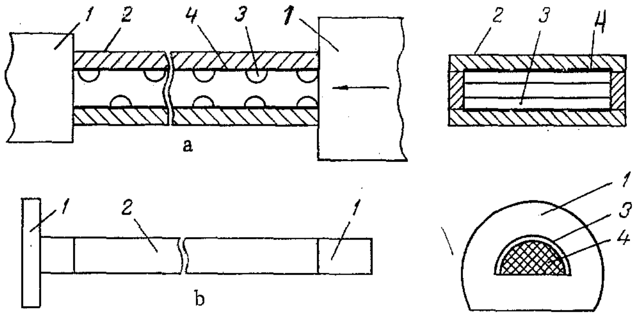

Fig. 1. Basic diagram of the design: a) test section [1) inlet and outlet chambers; 2) walls of plane channel 3) semicylindrical projections; 4) flat heating sections]; b) semicylindrical calorimeter [1) current leads; 2) calorimeter assembled; 3) semicylindrical heating element; 4) textolite base].

To evaluate heat flow about the perimeter of the half-cylinder or flat surface, we used the solution of the heat conduction equation for an element of constant cross section. In calculating q_{calc} we took the empirical temperature distributions over the surfaces. It should be noted that the resulting value of q_{calc} for the flat surface was about 0.02% of q_{el}, i.e., was negligibly small; for the half-cylinders, the value of q_{calc} was 2-4.5% of q_{el}.

Heat transfer on the flat surface and the semicylindrical calorimeter was measured separately on one side of the channel. Heat transfer on the flat surface was measured after the first, second, and third projections in the staggered and unstaggered arrangements.

Figure 2a, b shows test data on the distribution of the heat-transfer coefficients along the flat surface after the third half-cylinder. It follows from the figure that in both the staggered and unstaggered arrangements there are characteristic maximums of heat transfer at the flow attachment points on the flat surface. The drop in local heat transfer upstream and downstream of this point is in qualitative agreement with the predictions of boundary-layer theory.

We should note the higher level of heat transfer on the flat surface with the unstaggered arrangement (Fig. 2a) compared to the staggered setup (Fig. 2b). This fact can be explained by the different flow conditions for the two arrangements. For example, with an unstaggered arrangement the flow undergoes greater constriction than with a staggered arrangement: $(F/f)_{uns} = 5.77$, $(F/f)_{stag} = 1.7$. As follows from [3], this has a significant effect on the rate of heat transfer in the attachment zone.

Figure 3a, b shows a comparison of the local heat-transfer coefficients on the flat surface of the channel beyond the third semicylindrical projection for the unstaggered and staggered arrangements, respectively, with heat transfer far from the inlet in the plane channel. For comparison we used the following dependence for heat transfer in a plane channel far from the inlet [5]

$$\mathrm{Nu}_{d\mathrm{eqv}} = 0.024\,\mathrm{Re}_{d\mathrm{eqv}}^{0.8}\mathrm{Pr}^{0.43}. \tag{1}$$

It is apparent from Fig. 3 that heat transfer is greater in a plane channel with semicylindrical projections than in a smooth plane channel by a factor of from four to nine in the case of an unstaggered arrangement and by a factor of from two to five in the case of a staggered arrangement.

Figure 4 shows test data on the local heat-transfer coefficient of the semicylindrical projections with unstaggered and staggered arrangements. Heat transfer about the perimeter of the half-cylinders in the unstaggered arrangement increases from the frontal generatrix ($\varphi = 0°$) to $\varphi = 60$-$70°$ (except for the regimes with $\mathrm{Re}_d = 96{,}046$ and $\mathrm{Re}_d = 107{,}398$). In the case of a staggered arrangement (Fig. 4b), this increase takes place up to $\varphi \approx 20°$. The shift of the heat-transfer maximum from the frontal generatrix ($\varphi = 0°$) can be explained by the presence of the boundary layer on the flat surface ahead of the half-cylinder and the formation of a stagnant zone in the region of the frontal generatrix of the half-cylinder.

The increase in heat transfer in the aft zone of the half-cylinder in the cases of both

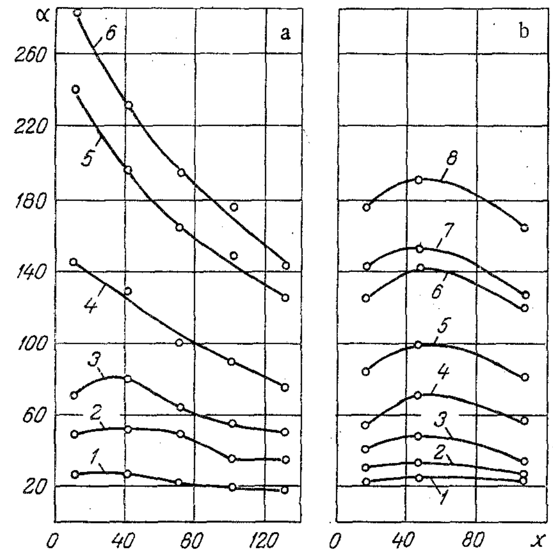

Fig. 2. Distribution of heat-transfer coefficients α (W/m^2 · deg) along the plane surface with unstaggered (a) and staggered (b) arrangements: a) 1) Re_h = 1107; 2) 2824; 3) 4844; 4) 8958; 5) 21,490; 6) 27,051; b) 1) Re_h = 1072; 2) 2763; 3) 4769; 4) 7416; 5) 15,159; 6) 16,957; 7) 21,422; 8) 27,085; x, mm.

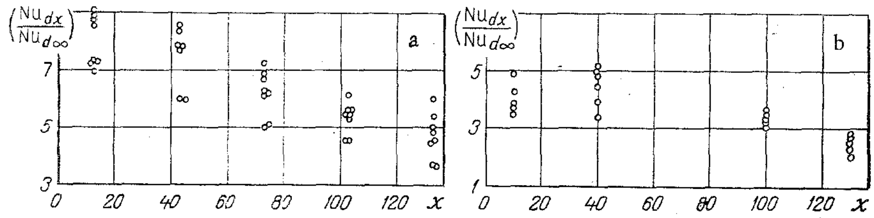

Fig. 3. Comparison of local values of heat-transfer coefficient on the flat surface of the channel with heat transfer far from the inlet: $(Nu_{deqvx}/Nu_{deqv\infty}) = f(x)$ in the case of an unstaggered (a) and staggered (b) arrangement.

staggered and unstaggered arrangements can be explained by separation of the boundary layer from the surface of the half-cylinder.

The above is valid for all curves except the two topmost curves in Fig. 4a, which correspond to Reynolds numbers Re_d = 135,193 and Re_d = 170,000. The character of change in heat transfer here in the aft region of the half-cylinder can be explained by the possible appearance of transitional and turbulent flow regimes in the boundary layer, with its subsequent separation.

If we compare our data with the heat transfer after separation of a laminar boundary layer for a single cylinder, it becomes obvious that the free space in the aft region of the cylinder promotes more intensive heat transfer after separation. This can be explained by the fact that a reverse flow develops on the surface of the projection from the attachment point, and the growing boundary layer prevents intensive heat transfer.

The perfection of the heat-exchanging surface from an energy point of view can be characterized by the relationship between the quantity of heat transmitted through the given heating surface and the energy spent by the moving fluid to overcome resistance.

An adequately full representation of an optimum design of heat-exchanging surface composed of tubes in a transverse flow gives a heat-transfer efficiency coefficient which, following the recommendations in [6], is calculated from the formula

$$E = \frac{\bar{\alpha}}{(f/F_h)\,\bar{w}\Delta p}, \quad \frac{1}{\text{deg}}. \tag{2}$$

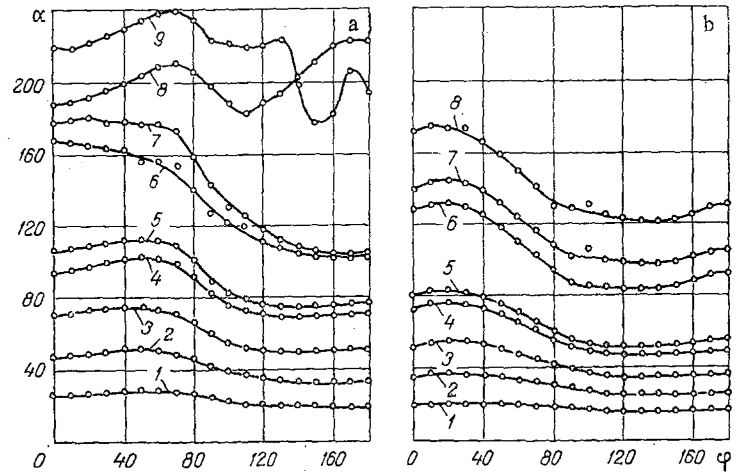

Fig. 4. Local heat transfer α of semicylindrical projections with an unstaggered (a) and staggered (b) arrangement: a) 1) Re_d = 6928; 2) 17,759; 3) 30,175; 4) 47,449; 5) 56,365; 6) 96,046; 7) 107,398; 8) 135,193; 9) 170,000; b) 1) Re_d = 1569; 2) 4061; 3) 6943; 4) 10,912; 5) 13,002; 6) 24,903; 7) 31,447; 8) 37,883; φ, deg.

Equation (2) was also used to calculate the heat-transfer efficiency E_0 and E_x, respectively, for an unstaggered bundle of tubes without fins and for a staggered bundle of tubes equipped with solid longitudinal fins. In both cases the tube diameter was 31 mm and the relative transverse and longitudinal spacings had the values: $(s_1/d) = 1.21$; $(s_2/d) = 5.32$. The coefficient E_x was calculated using experimental data on heat transfer and hydraulic resistance.

It follows from the results that an unstaggered arrangement of tubes of the above geometry, equipped with longitudinal ribs and used in the Reynolds number region $Re_d \leqslant 10^4$, is more effective than a similar arrangement of smooth-tube bundles.

NOTATION

H, height of plane channel, m; d, diameter of projection, m; S_1, S_2, transverse and lateral spacing, m; $Re_d = \overline{w}_{nar}d/\nu$, $Re_h = \overline{w}_{wid}H/\nu$, $Nu_d = \alpha d/\lambda$, corresponding Reynolds and Nusselt numbers; q_{wa}, q_{conv}, q_{el}, q_{los}, q_{calc}, heat fluxes on the wall, by convection, by electrical heating, in heat loss to the environment, and in heat flow due to the thermal conductivity of the metal, respectively, W/m²; t_{wa}, t_{air}, temperature of the surface and the air in the section of interest, °C; F, f, area of maximum and minimum cross sections, m²; F_h, area of heating surface, m²; α, $\overline{\alpha}$, local and mean heat-transfer coefficients, W/m²·deg K; $\overline{w}$, mean velocity, m/sec; Δp, pressure drop, N/m²; E, efficiency coefficient, 1/deg; d_{eqv} = 2H, equivalent diameter, m.

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EXPERIMENTAL INVESTIGATION OF LOCAL HEAT TRANSFER IN A TURBULENT BOUNDARY LAYER AT SUPERSONIC SPEED

N. F. Ragulin and Yu. G. Shvalev

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The authors present results of an experimental investigation of coefficients of local heat transfer in a turbulent boundary layer over a wide range of M and Re numbers and values of the temperature ratio.

Most of the presently known methods of calculating convective heat transfer in a turbulent boundary layer at supersonic speeds are based on experimental data. However, these data span heat-transfer conditions rather fully only for comparatively low Reynolds numbers and for surface temperatures close to adiabatic. There has been little experimental study of the influence of large Reynolds numbers over a wide range of values of temperature ratio and Mach number.

The aim of the present investigation is to obtain the heat-transfer laws in a turbulent boundary layer over a wide range of the governing criteria. The tests were made in a wind tunnel on airfoil models with a 5% profile and a cylindrical body of revolution with ogive nose section and length equal to 4. The experimental values of local heat-transfer coefficients span the ranges $Re_\delta = 2.5 \cdot 10^6$–$84 \cdot 10^6$, $M_\delta = 1.8$–6.2 and temperature ratio $\overline{T}_w = T_w/T_r = 0.28$–$1.37$.

To obtain values of the temperature ratio $\overline{T}_w < 1$ on the model surface we used special equipment to cool the model skin with a mixture of liquid and gaseous nitrogen and to maintain its temperature at a given level. The flow rate of liquid nitrogen was determined from the condition of obtaining a given average temperature on the model surface, and the gaseous flow rate was determined by the need to maintain rather high flow velocities of two-phase mixture in the internal channels of the model to avoid separation of the liquid phase.

To measure local heat-transfer coefficients with $\overline{T}_w > 1$, we heated the model prior to the test by means of a special electrical heater. In the tests with hot flow ($M_\delta = 6.2$) the model was heated immediately before the flow was brought on.

The surface temperatures and the local heat-transfer coefficients were measured by means of special thermal sensors located on the inner surface of the model skin [1]. With these one can measure local heat-transfer coefficients to the model surface by both steady and unsteady methods. The total error in determining local heat-transfer coefficients did not exceed 15%.

The test heat-transfer data were reduced to obtain a power-law relationship between the governing criteria of the form

$$\mathrm{St}_\delta = A\mathrm{Re}_\delta^n (1 + 0.2rM_\delta^2)^m \left(\frac{T_w}{T_r}\right)^k,$$

where A, n, m, and k are constants; r = 0.89 is the flow temperature recovery factor.

In determining the Reynolds number one must know the conditions of boundary-layer development. The test models had a pressure gradient and therefore the conditions differed from flow over a flat plate, which made it difficult to compare the results of different experiments directly, and therefore the flow conditions for the test models were reduced to conditions for flat plate flow.

We allowed for the influence of the pressure gradient on the development of the boundary

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