HEAT TRANSFER ON A FLAT PLATE IN CONDITIONS OF HYDRODYNAMICALLY STABILIZED TURBULENT BOUNDARY LAYER FLOW

V. M. Legkii and Yu. D. Koval'

UDC 536.244

Results are presented of an experimental investigation of heat transfer on a plate in turbulent flow of air, by the method of regular thermal regime. Formulas are given for calculating local and average heat-transfer coefficients, reflecting an improvement in the relations currently used.

Experimental data [1] on heat transfer on a heat plate with longitudinal flow of air show evidence of the existence of an unstable regime in the region $Re_{x_1} < 100{,}000$, in which the distribution of local heat-transfer coefficients along the surface has quite a complex, and at first sight a random, nature. Since the experiment in [1] was performed on a relatively short plate, the variation in the heat-transfer law for $Re_{x_1} < 100{,}000$ could be a consequence of two quite different phenonema. There is no doubt that the nature of the heat transfer in the entrance section is dictated to some extent by the shape of the entrance, even if the boundary layer is assumed to be turbulent everywhere. However, under conditions of developed turbulent motion of an air stream, near the leading edge of the plate a laminar boundary layer can exist, which is unstable at the stagnation point, and later undergoes transition to a turbulent boundary layer [2]. Unfortunately, the results given in [1] do not afford a sufficiently clear answer to the question as to which of the above factors plays the dominant part in generating the transition.

In the work described in this paper an investigation of heat transfer on a plate in a hydrodynamically stabilized boundary layer flow is accompanied by a special series of tests to establish the region of action and the nature of the entrance effects. The tests were performed in a closed low-head wind tunnel, which has been briefly described earlier in [3]. Figure 1 shows a schematic of the working section of the wind tunnel. The collector 1 has four plates forming a square channel of section 280×280 mm^2, in which the shaped inserts 2 are located. The inserts reduced the cross section to 140×280 mm^2. At the beginning of the working section there is a pressure tap 3, connected to a standard type MMN micromanometer, and a type ÉTAM-3A hot wire anemometer, 4. The thermocouple 5 and a mercury thermometer are used to control the stream temperature. Nonuniformity in the velocity field in the central part of the working section, limited to dimension 250×250 mm, was no more than 0.5%.

The experimental Textolite plate of length 300 mm and width 220 mm was mounted vertically on the bracket 7. The bracket passed through a cutout in the top plate of the working section and was fixed to the support plate 6, which was isolated from the top plate with soft felt, at the points of contact. Horizontal movement of the lower part of the plate was eliminated by the rod, 8, sliding in the guide rings 9. The total plate was made up of several carefully fitted parts.

The heat-transfer coefficients were measured using the method of regular thermal regime of the first kind on the plates with an identical rectangular entrance of thickness $\delta = 5$ mm (plate No. 1) and thickness $\delta = 12$ mm (plate No. 2). By way of example, Fig. 2 and Table 1 show the construction, orientation, and technical parameters of the α calorimetric plate No. 2, composed of five parts (50 mm + 50 mm + 50 mm + 25 mm + 125 mm).

Calorimeters Nos. 1, 2, and 3, each of size $8 \times 15 \times 2$ mm^3, were mounted flush with the surface in Section I of this plate. The felt thermal insulation in which the calorimeters were buried was isolated from the flow by a Textolite plate of thickness 0.7 mm.

Polytechnic Institute, Kiev. Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 16, No. 1, pp. 22-29, January, 1969. Original article submitted March 13, 1968.

TABLE 1. Thermophysical Characteristics of the Calorimeters of Plate No. 2

Calorimeter No.	Heated length $X \cdot 10^3$, m	Calorimeter material	Mass of calorimeter $M \cdot 10^3$, kg	Heat transfer area $H \cdot 10^4$, m^2	Calorimetric constant $Mc_{k_2}/H \cdot 10^{-2}$, $J/m^2 \cdot deg$
1	7,92	St. 35	18,49	1,170	7,59
2	7,96	St. 35	18,41	1,175	7,51
3	7,96	St. 35	18,43	1,175	7,52
4	8,03	St. 35	18,25	1,204	7,26
5	8,07	St. 35	19,16	1,210	7,57
6	8,10	St. 35	19,23	1,215	7,58
7	30,0	Lead	124,30	18,00	9,09
8	30,0	St. 35	84,38	18,00	22,47
9	30,0	St. 35	35,30	9,00	18,80

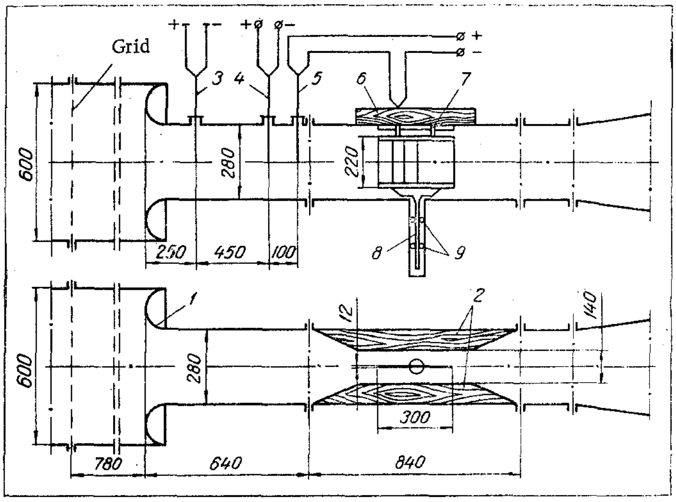

Fig. 1. Schematic of wind tunnel working section.

Calorimeters Nos. 4, 5, and 6, also of size $8 \times 15 \times 2$ mm^3, mounted in a cast block of polyurethane foam plastic PU-3, were located in the second section. Section III contained calorimeters Nos. 7 and 8, supported and insulated with felt, having two heated edges, and of size $30 \times 30 \times 12$ mm^3, and also calorimeter No. 9, of single-sided form, of size $30 \times 30 \times 5$ mm^3.

All the calorimeters had a hole of diameter 0.3-0.4 mm, in which Chromel–Copel thermocouples, with wire diameter 0.12 mm, were fastened with Rose metal. The thermocouple wires were brought out through the plate body and the support, and were differentially connected through a switch to thermocouple 5, located in the wind tunnel stream. A type M 17/5 mirror galvanometer was used to measure the differential thermocouple emf.

Plate No. 1, similar in structure to plate No. 2, had seven double-sided α calorimeters of size $20 \times 20 \times 5$ mm^3.

The sequence of operations during a test was as follows. The plate was withdrawn from the wind tunnel using the support plate, and each of the calorimeters was rapidly heated (over 3-5 sec) to 18-20°C above the stream temperature, due to contact with the hotter body. After the plate was inserted in the exposed part of the rig, and the overheat temperature was reduced to $\vartheta = 10$°C, a record was made of the change of excess calorimeter temperature with time, $\vartheta = f(\tau)$, as is usually done in experiments with regular regime. Then the cooling rate was measured from a graph of $\ln\vartheta = f(\tau)$. The temperature of the incident stream was maintained at the level $t_{flow} = 18$-20°C and 48-52°C.

$$m = \frac{\ln\vartheta - \ln\vartheta_1}{\tau_1 - \tau} \ \mathrm{deg}^{-1}. \qquad (1)$$

It was convenient to conduct the experiment with the same excess temperature range. Then for $\tau = 0$, $\ln\vartheta_0 = \mathrm{const}$, and for $\tau = \tau_1$, $\ln\vartheta_1 = \mathrm{const}$, and Eq. (1) takes the form

$$m = \frac{\mathrm{const}}{\tau_1} \ \mathrm{deg}^{-1}. \qquad (2)$$

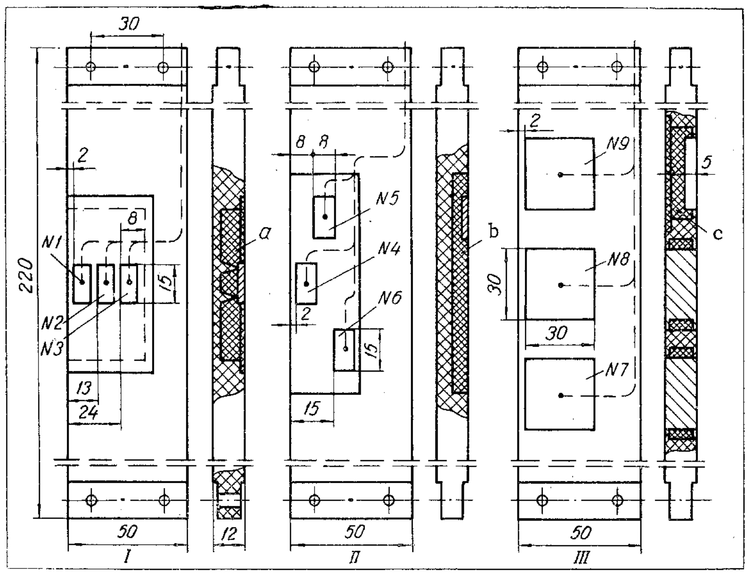

Fig. 2. Construction of α calorimetric plate No. 2 [I-III) sections; a,c) felt; b) foam plastic].

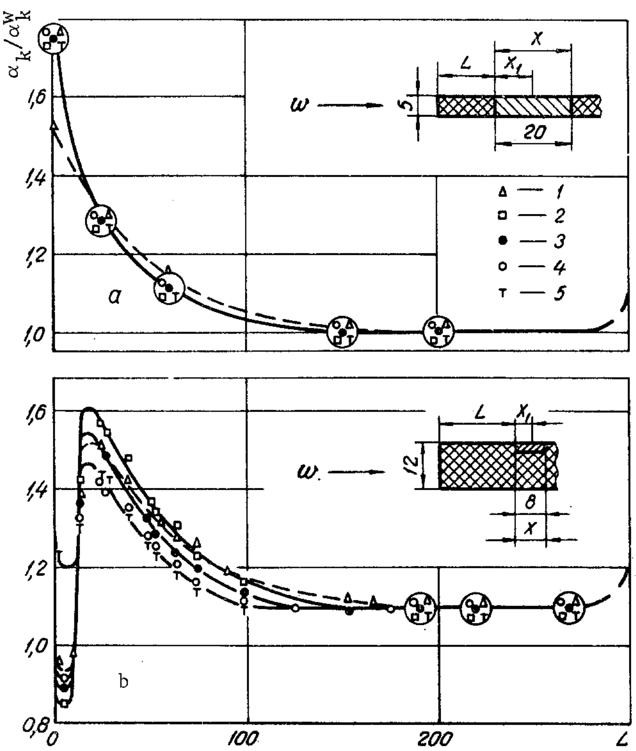

Fig. 3. Distribution of heat-transfer coefficient along the plate length (mm). [a) Plate No. 1, δ = 5 mm; b) plate No. 2, δ = 12 mm]: 1) 2 m/sec; 2) 10; 3) 20; 4) 50; 5) 80 m/sec.

Because the quantity $\vartheta_0 - \vartheta_1$ was relatively small, and constant from test to test, the heat drain to the insulation can be described, with sufficient accuracy, by some contact thermal resistance $1/\bar{\alpha}_{ins}$, referred to the area of contact between the calorimeter and insulation. Then the cooling rate is related to the heat-transfer coefficients by the expressions [6]:

$$m = \frac{(\alpha_k + \alpha_r) H + \bar{\alpha}_{ins} H_{ins}}{M c_k} \psi \quad \text{deg}^{-1}; \tag{3}$$

$$\alpha_k = \frac{m M c_k}{\psi H} - \bar{\alpha}_{ins} \frac{H_{ins}}{H} - \alpha_r \quad \text{W/m}^2 \cdot \text{deg}. \tag{4}$$

The quantities ψ and α_{rad} were calculated from [4] and [7], and the value of the correction ψ did not go outside the range 0.96-1.0. The specific heat of steel was taken to be c_k = 460 J/kg · deg and that of chemically pure lead as c_k = 130 /kg · deg [11, 12]. The heat-transfer coefficient $\bar{\alpha}_{ins}$ was determined by calibration tests in which the calorimeter surfaces normally cooled by the air stream were covered by felt or foam-plastic, appropriately. By treating the calibration curve in sections we were able to find the average cooling rate m_{ins} in the range $\tau = 0$, $\tau = \tau_1$, and therefore

$$\alpha_{ins} = \frac{\bar{m}_{ins} M c_k}{H + H_{ins}} \quad \text{W/m}^2 \cdot \text{deg}. \tag{5}$$

Equation (5) agrees qualitatively with the results of a theoretical analysis of heat losses made in [8].

It can be seen from Table 2, which gives $\bar{\alpha}_{ins}$ for calorimeter No. 5 of plate No. 2 as a function of test time, that the relative part played by heat drain rapidly drops with decrease of τ_1. The condition with τ_1 = 80 sec and the least favorable ratio $\bar{\alpha}_{ins}/(\alpha_k + \alpha_{rad})$ refers to the case of natural convective cooling of the calorimeter. For a stream velocity w > 25 m/sec the test duration decreased to τ < 20 sec, and the heat transfer to the insulation did not exceed 10% of the basic measured value.

TABLE 2. Heat-Transfer Coefficient $\bar{\alpha}_{ins}$ as a Function of Time

τ_1, sec	10	20	30	40	50	60	80
$\bar{\alpha}_{ins}$, W/m^2·deg	20,0	19,2	18,8	18,5	18,2	17,9	17,3
$\bar{\alpha}_{ins}/(\alpha_k + \alpha_r)$	0,047	0,109	0,158	0,228	0,308	0,590	0,940

The stream velocity varied in the range w = 0.5-90 m/sec at a turbulence level not exceeding 3-4%.

Analysis of the distribution of the convective heat-transfer coefficient along the plate length at constant velocity shows that in the neighborhood of the leading edge the heat transfer is higher, and depends appreciably on the extent of the adiabatic forward section. Downstream the results of the calorimetry were repeatable, and at the end of the washed surface, where the boundary layer separates from the plate, an "outlet" effect was generated, and the heat-transfer coefficient again increased.

Figure 3 shows typical curves of $\alpha_k/\alpha_k^{st} = f(L)$. The distribution of relative heat transfer intensity over the initial section of plate No. 1 was the same over all the velocity range investigated. Only in the region where w < 3 m/sec, was the curve $\alpha_k/\alpha_k^{st} = f(L)$ gradually changed in shape, degenerating to the straight line $\alpha_k/\alpha_k^{st} = 1$, as $w \to 0$. The absence of any indication of transition is evidence that the boundary layer is in a homogeneous turbulent state at the initial section, or, more correctly, that it has been instantaneously rendered turbulent at the leading edge.

On plate No. 2, a deterioration of the aerodynamic quality of the profile led to separation of the wall flow, accompanied by a discontinuous increase in the heat-transfer level.

The repeatability of the measured results at some distance from the leading edge is an indication that the flow in the boundary layer is hydrodynamically stabilized. In what follows we discuss only data relating to this region. The experimental data were correlated on the basis of the relation $St = cRe^{-n}$. Initially the average heat-transfer coefficient over the surface H was considered localized at the calorimeter center, and the Reynolds number was referred to the coordinate X_1. Figure 4A shows the test results reduced in this way. Also shown are the data of [1], obtained with $Re_{X_1} \geq 100{,}000$, and curve 1 calculated from Eq. (6), equivalent to Eq. (7) of [1],

$$St_{X_1} = 0.0364\, Re_{X_1}^{-0.2}, \qquad (6)$$

$$Nu_{X_1} = 0.0255\, Re_{X_1}^{0.8}. \qquad (7)$$

In agreement with Eq. (6) at $Re_{X_1} > 20{,}000$, the experimental points are located along a smooth curve whose slope gradually changes with reduction of Reynolds number.

Although the comparison made in Fig. 4A confirms the results of [1] as a whole, the validity of the comparison is itself doubtful. There is no physically valid justification for identifying the mean heat-transfer coefficient with the local value at the calorimeter center. If the surface is divided, going downstream, into a number of short independent segments, the average heat-transfer coefficient is at least 13% above the local value at the center, within the first segment.* The possibility that systematic positive errors are introduced in carrying out tests by the above method has been noted by a number of authors who have used elongated calorimeters [10, 13].

The measured results are shown in Fig. 4B in the form of the correlation $\log St_X = f(\log Re_X)$, usual for average heat-transfer coefficients. Curve 2 was calculated from the mean heat-transfer equation, Eq. (8), which, like Eq. (6), is based on the data of [1]. In the region Re_X = 20,000-120,000 the experimental points lie 10% below curve 2:

$$St_X = 0.0457\, Re_X^{-0.2}. \qquad (8)$$

Figure 4B also shows the summary relations from [13], in which an isothermal plate of length 1650 mm was investigated. In [13] a relation for local heat transfer was assumed:

$$St_{X_1} = 0.0296\, Re_{X_1}^{-0.2} Pr^{0.4} (T_w/T_{flow})^{0.4}. \qquad (9)$$

*The error is reduced to 4-5% in the second and third segments, but it can be neglected only at a distance of 150-200 mm from the start of the heated section. The calculations are based on the assumption that n = 0.2.

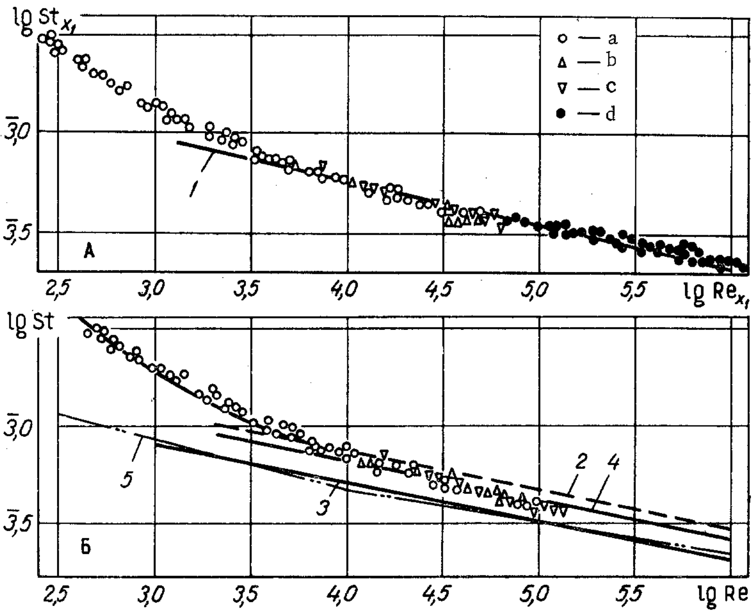

Fig. 4. Results of correlation of test data on heat transfer on the plate in the hydrodynamically stabilized region: 1) from Eq. (6); 2) Eq. (8); 3) Eq. (10), 4) Eq. (11); 5) Eq. (13); a) calorimeters of size 8 × 15 × 2 mm; b) 20 × 20 × 5; c) 30 × 30 × 12 and 30 × 30 × 5 mm; d) test data of [1].

By introducing Pr = 0.71 and a value for the temperature factor $T_{st}/T_{flow} \approx 1.1$ equal to that in [13], we arrive at the expression (curve 3):

$$St_{X_1} = 0.0325\, Re_{X_1}^{-0.2}. \tag{10}$$

Correspondingly, for the mean values (curve 4)

$$St_X = 0.0410\, Re_X^{-0.2}. \tag{11}$$

The divergence of the test data from Eq. (11) in the region of Reynolds number greater than 20,000 does not exceed ±2%. For $Re_X < 20{,}000$ the experimental data can be approximated, to an accuracy of ±5%, by the equation

$$St_X = 1.93 \frac{Re_X^{0.3}}{(\lg Re_X)^{6.1}}. \tag{12}$$

We convert to a relation for local heat transfer by differentiating Eq. (12)

$$St_{X_1} = St_{X=X_1} + X\, St'_{X=X_1} = 1.93 \frac{Re_{X_1}^{0.3}}{(\lg Re_{X_1})^{6.1}} \left[1.3 - \frac{2.6}{\lg Re_{X_1}}\right]. \tag{13}$$

Curve 5 in Fig. 4B calculated from Eq. (13) in the range $Re_{X_1} = 10^3$–$3 \cdot 10^5$, practically coincides with curve 3, deviating from it by a maximum of 7%.

The accuracy of calorimetry by the method of unsteady thermal conditions is determined principally by errors in the cooling rate m and m_{ins}, since weighing of the calorimeters and determination of their geometrical parameters can be accomplished quite carefully. In the range of values m = 0.05–0.1 sec^{-1}, prevailing in the tests, the error in cooling rate did not exceed 1.5–2.0%. The error in the value of c_k appearing in Eq. (4) was 2%. The greatest total error in the experimental determination of the heat-transfer coefficient was ~6%.

NOTATION

T_{flow}, t_{flow} — are the flow temperature, °K, °C;
T_w, t_w — are the calorimeter temperature, °C;

ϑ	is the excess calorimeter temperature, °C;
τ	is the time, sec;
m	is the calorimeter cooling rate, sec^{-1};
α_k	is the convective heat-transfer coefficient, $W/m^2 \cdot deg$;
$\underline{\alpha}_r$	is the coefficient of heat transfer by radiation, $W/m^2 \cdot deg$;
α_{ins}	is the conventional heat-transfer coefficient for calorimeter with insulation, $W/m^2 \cdot deg$;
α_k^{st}	is the heat-transfer coefficient by convection in the stabilized region, $W/m^2 \cdot deg$;
H	is the surface area for heat transfer with air, m^2;
H_{ins}	is the area of contact of calorimeter with insulation, m^2;
M	is the calorimeter mass, kg;
c_k	is the specific heat of calorimeter material, $J/kg \cdot deg$;
ψ	is the correction for nonuniform temperature field at the calorimeter;
w	is the velocity of airstream, m/sec;
δ	is the plate thickness, m;
X	is the total calorimeter length, m;
X_1	is the ambient coordinate along heated length, m;
L	is the length of adiabatic section preceding calorimeter, m;
$Nu = \alpha_k X/\lambda$	is the Nusselt number;
$St = \alpha_k/\rho w C_p$	is the Stanton number;
$Re = wX/\nu$	is the Reynolds number;
$Pr = \nu/a$	is the Prandtl number.

LITERATURE CITED

1. B. S. Petukhov, A. A. Detlaf, and V. V. Kirilov, Zh. Tekh. Fiz., 24, No. 10 (1954).
2. H. Schlichting, Boundary Layer Theory [Russian translation], IL (1956).
3. V. I. Tolubinskii and V. M. Legkii, Énergomashinostroenie, No. 8 (1963).
4. G. M. Kondrat'ev, Regular Thermal Regime [in Russian], GITTL (1954).
5. V. M. Legkii, Izv. Kievskogo Politekhn. Inst., 30, Part 1, 62 (1960).
6. A. I. Lazarev, Sb. Rabot Studencheskogo Nauchnogo Obshchestva, LITMO, No. 8 (1953).
7. M. A. Mikheev, Fundamentals of Heat Transfer [in Russian], Gosénergoizdat (1956).
8. L. V. Kozlov, Izv. Akad. Nauk SSSR, OTN, No. 6 (1961).
9. M. A. Mikheev, in: Convective and Radiative Heat Transfer [in Russian], Izd. AN SSSR (1960).
10. A. I. Chaplina, Inzh. Fiz. Zh., 5, No. 7 (1962).
11. D. Kaye and T. Laby, Tables of Physical and Chemical Constants [Russian translation], Fizmatgiz (1962), p. 168.
12. B. E. Neimark (editor), Physical Properties of Steels and Alloys Used in Power Engineering: Handbook [in Russian], Izd. Énergiya (1967), p. 93.
13. W. C. Reynolds, W. M. Kays, and S. L. Kline, Trans. of ASME, Ser. C, 82, No. 4 (1960).