2. F. L. Sayakhov, M. A. Fatykhov, and O. L. Kuznetsov, "Study of electromagnetic-acoustic action on temperature distribution in oil-water saturated rock," Izv. Vyssh. Uchebn. Zaved., Neft' Gaz, No. 3 (1981).
3. Ngoi Hai Jung, A. G. Kutushev, and R. I. Nigmatulin, "On the theory of liquid filtration in a porous medium upon volume heating by high frequency electromagnetic field," Prikl. Mat. Mekh., 51, No. 1 (1987).
4. Ngoi Hai Jung, N. D. Musaev, and R. I. Nigmatulin, "Self-similar solutions of the heat-mass transport problem in a saturated porous medium with volume heat source," Prikl. Mat. Mekh., 51, No. 6 (1987).
5. A. A. Kislitsyn and R. I. Nigmatulin, "Numerical modeling of the process of heating a petroleum stratum by high frequency electromagnetic radiation," Prikl. Mekh. Tekh. Fiz., No. 4 (1990).
6. F. L. Sayakhov, G. A. Babalyan, and A. N. Al'met'ev, "One method for extracting viscous oils and bitumens," Neft. Khoz., No. 12 (1975).
7. A. A. Samarskii, Difference System Theory [in Russian], Nauka, Moscow (1977).
8. V. M. Paskonov, V. I. Polezhaev, and L. A. Chudov, Numerical Modeling of Heat-Mass Transport Processes [in Russian], Nauka, Moscow (1984).
9. F. G. Arzhanov, A. S. Kuvshinov, Yu. B. Fani, et al., "Estimating possible petroleum output from development of Russian fields by thermal methods," Neft. Khoz., No. 2 (1977).
10. A. T. Akhmetov, A. A. Kislitsyn, A. M. Fadeev, and A. A. Chebakov, "Experimental study of dielectric properties of petroleum technology materials," in: Studies of the Multiphase System Mechanics Institute, Siberian Branch, Academy of Sciences of the USSR [in Russian], No. 2, Tyumen' (1990).
11. L. Yu. Tashlykova and N. V. Yudina, "Effect of high molecular weight components on petroleum system rheology," in: Solution and Suspension Structure: Properties of Colloidal Systems and Petroleum Solutions [in Russian], Nauka, Novosibirsk (1988).
12. L. D. Landau and E. M. Lifshits, Electrodynamics of Continuous Media [in Russian], Nauka, Moscow (1982).

FORCED-CONVECTION HEAT TRANSFER FROM A HEATED SURFACE IN A CAVITY FORMED BY TWO HIGH FINS

A. I. Gnyrya, V. I. Terekhov,
S. P. Tret'yakov, and N. I. Yarygina

UDC 536.24

Interest in the investigation of detached flows is due to the extensive engineering applications of such streams. It is well known [1] that the presence on a surface over which flow occurs of macroscopic roughness in the form of steps, fins, and grooves, resulting in detaching of the stream, can cause either intensification or a decrease in intensity of convective heat transfer. Intensification of heat transfer is caused by the additional turbulizing of flow in vortex zones, as well as the small thickness of the boundary layer in the zone of its reattachment to the wall and in the region of its further development downstream. A decrease in heat transfer is due to the small transfer coefficient in closed circulation zones ahead of and behind the obstacles. The influence of this factor increases with increasing height of the obstacles, when its relative value is $h^+ = hv^*/\nu \geq 10^2$-10^3 (h is the height of an obstacle, $v^* = \sqrt{\tau_w/\rho}$ is the frictional velocity, and ν is the kinematic viscosity).

The combined manifestation of these factors considerably complicates the process of heat and mass transfer, and a theoretical solution of the problem becomes very problematic. Experiment acquires primary importance in the analysis of such flows.

Despite the large number of papers that have been devoted to the experimental study of heat transfer in detached flows, the mechanism of heat transfer in a cell between two high fins has hardly been studied. Heat transfer behind a single high fin was mainly investigated in [2, 3]. A large number of papers have been devoted to the experimental study of hydrodynamics and heat transfer when there are roughness elements on a surface in the form of a system of

Novosibirsk. Translated from Prikladnaya Mekhanika i Tekhnicheskaya Fizika, No. 3, pp. 103-108, May-June, 1993. Original article submitted April 8, 1992.

0021-8944/93/3403-0387$12.50

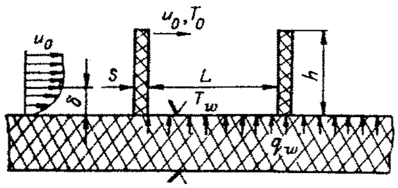

Fig. 1

obstacles — fins of small height (h^+ = 5-50) that are used to intensify heat transfer [4-9]. On the whole, flow between obstacles of small height, whose linear scale is considerably less than the thickness of the boundary layer, differs fundamentally from the problem we are considering.

The present paper is devoted to an experimental investigation of forced convective heat transfer from a heated surface between two high fins. Considerable attention is devoted to visualization of the flow pattern.

1. EXPERIMENTAL APPARATUS. MEASUREMENT METHOD

The tests were carried out in the channel of a subsonic wind tunnel with a 200 × 200 mm cross section. On the lower wall of the channel (Fig. 1) we set up fins of a material with a low thermal conductivity (plastic) and a thickness S = 3 mm. In these tests the fins were the same height (h = 60 mm) and the distance between them was L/h = 1, 2, and 3 calibers.

The lower wall of the channel consisted of a textolite plate with a thickness d = 20 mm. Heat was supplied by a ribbon of aluminum foil with a thickness ~20 μm, mounted on the plate on the inside of the channel. The heater ran the entire width of the channel in the form of a continuous current-carrying track with a width 5 mm. The distance between successive tracks was ~0.5 mm, and the total electrical resistance was R ≈ 7.6 Ω.

The condition q_w = const was satisfied in the tests, with q_w being defined as the ratio of the supplied electrical power to the entire heated surface, including the gaps between the current-carrying tracks, which is valid because of their smallness. The surface temperature was determined with 36 chromel—copel thermocouples with a wire diameter 0.2 mm, located below the ribbon heater along one straight line in the middle of the working section. The air temperature was measured with a similar chromel—copel thermocouple located in the forechamber of the apparatus. The maximum temperature difference between the wall and the oncoming air in the tests was $\Delta T = T_w - T_0 \approx 50°$, so that the temperature factor did not exceed $\psi = T_w/T_0 \leq 1.2$ and the flow was nearly isothermal. In analyzing the experimental data we did not allow for the temperature factor, and the physical properties of the air were taken at the mean temperature between the wall and the core of the flow.

Heat leaks through the plate on which the ribbon heater was mounted were determined experimentally. For this, thermocouples were fixed to the outer side of the working plate (see Fig. 1) along its entire length. Heat leaks in the tests were 10-30% and were taken into account in analyzing the results.

The tests were carried out in the range of velocities of the oncoming stream u_0 = 4-21 m/sec, so that the Reynolds number, calculated from the distance between the fins, was in the range $Re_L = u_0 L/\nu = 10^4\text{-}3\cdot10^5$. The characteristic velocity u_0, from which Re_L was determined, was taken from measurements of the potential flow over the first fin. As measurements showed, u_0 hardly varied over the entire length of the cell.

The turbulence intensity in the core of the flow was ~1%. A turbulent boundary layer with a thickness δ ≈ 7-10 mm was formed at the lower wall by the time the position of the first fin was reached; the thermal boundary layer began immediately after the first fin.

Our series of tests showed that in the absence of fins on the surface, the experimental data on heat transfer agree with the law of heat transfer for a flat, turbulent boundary layer. Their deviation did not exceed ±10%.

We paid much attention to visualization of the flow, which was done by the shadow method using an IAB-451 schlieren instrument. The field of optical inhomogeneities was produced by the heating of the heat-transfer surface. In addition, to obtain a visualization pattern of better quality, thin wires heated by an electric current were mounted at different heights on the surface of the fins, producing clear thermal wakes.

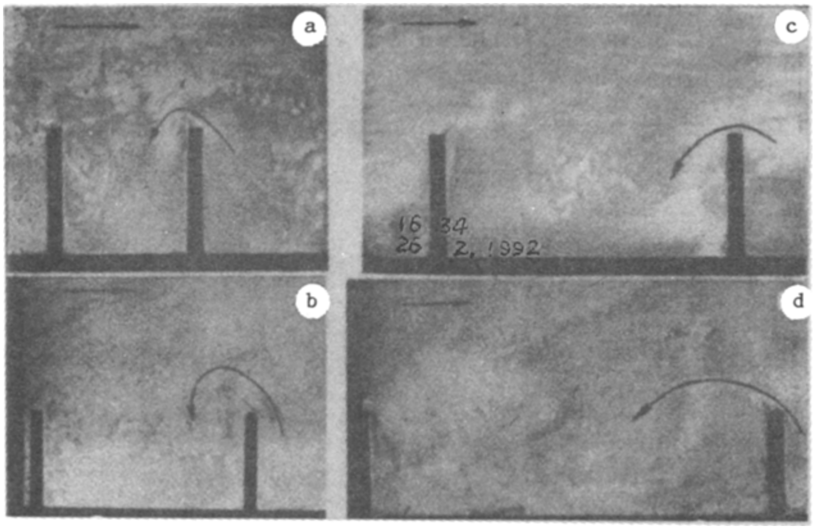

Fig. 2

A series of visualization experiments was carried out on a plane hydrodynamic bench with a 3 × 200 mm cross section by the method of an optically active fluid. The geometrical dimensions of the cell between the fins corresponded completely to the case of three-dimensional flow, and the values of Re_L were the same as in the investigation of heat transfer. The flow patterns obtained by the two methods were qualitatively similar, on the whole. It must be noted that vortex formation was observed at considerably higher Re_L in the plane model.

2. EXPERIMENTAL RESULTS AND DISCUSSION

Photographs of the flow pattern in the cavity between two fins in the plane model with an optically active fluid are given in Fig. 2 for four cavity widths: L/h = 1, 2, 3, and 4 correspond to a-d. It is seen that the oncoming stream detaches from the upper edge of the first fin, forming a shear layer behind it, as occurs in a rectangular groove cavity. Rarefaction is created in the space between fins, as a result of which secondary flow is generated, in the direction opposite to the main stream above the second fin. No such secondary flows have been noted in the literature in flow over grooves and systems of fins of small height; they evidently appear in the interaction of streams with a system of high fins.

The presence of secondary flows has considerable influence on the overall flow pattern in the space between fins, which is thus determined by the interaction of the two opposite streams. For L/h = 1, the secondary counterflow reaches the first fin. The resulting vortex rotates in the direction opposite to that of the vortex observed in flow over a rectangular cavity [10]. As the distance between fins increases (Fig. 2b), the flow pattern becomes more complicated. The secondary flow, penetrating into the region between the fins, forms a vortex ahead of the second fin. Some of the gas from that vortex is continuously detached from its outer boundary by the main stream. Another vortex is formed in the region behind the first fin, with the same direction of rotation as that ahead of the second fin. When the distance between fins is even larger (Fig. 2c), the vortices break down into a number of smaller ones, and the secondary flow can reach the heat-transfer surface between the fins.

The indicated features of the flow structure in the space between fins affect the distribution of local heat-transfer coefficients. Those data are given in Fig. 3, where $\bar{\alpha}$ is the integral-mean value of the heat-transfer coefficient in the space between fins and Re_h is the Reynolds number calculated for the fin height.

For L/h = 1 (Fig. 3a), the maximum of local heat transfer falls in the region immediately adjacent to the front fin, where the cool secondary flow first penetrates. A monotonic decrease in heat transfer is then observed as we move toward the second fin.

A different pattern in the distribution of local heat transfer occurs in flow over a groove with a heated bottom (also with L/h = 1) [10]. Its maximum value is reached at the downstream wall and its minimum is reached in the vicinity of the front wall. It is obvious that such a significant difference in the qualitative behavior of the heat-transfer coefficient is introduced by the secondary flows, which are typical of flow over a system of high fins.

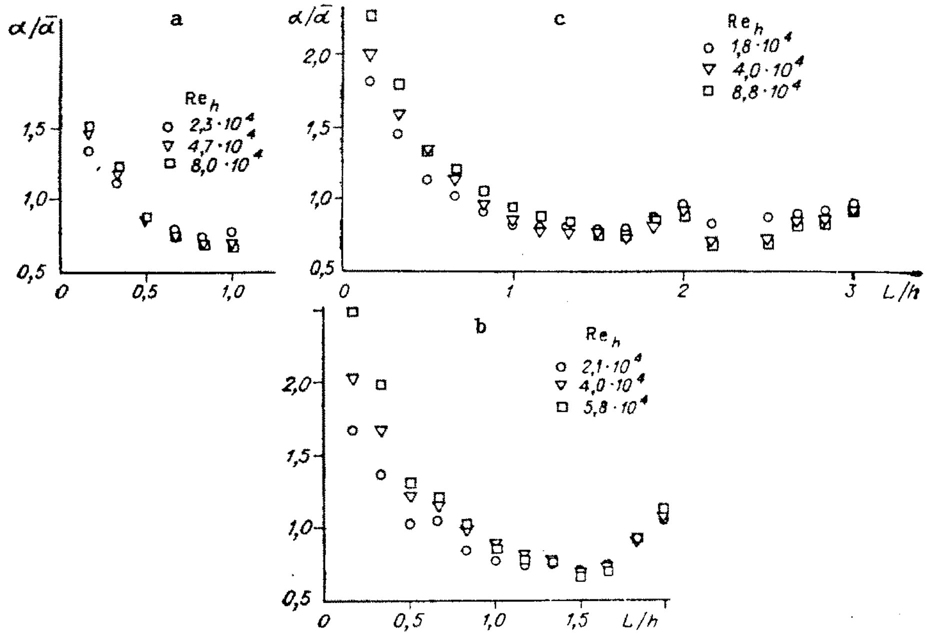

Fig. 3

With an increase in the number of calibers between fins (Fig. 3b, L/h = 2), local heat transfer near the base of the second fin begins to increase, which is due to the weakening and gradual breakup of the vortex near the second fin. The minimum of the heat-transfer coefficient thus shifts into the space between the fins. For L/h = 3 (Fig. 3c) an additional maximum in heat transfer appears at $x/h \sim 2$, due to attachment of the counterflow to the heated wall, clearly observed in the visualization pictures (Fig. 2b, c).

We note another important feature that is typical of all of the investigated geometries of the space between fins. The distributions of the relative heat-transfer coefficient $\alpha/\bar{\alpha}$, normalized to the mean value, turned out to be fairly insensitive to the value of Re_L. As seen from Fig. 3, the experimental points obtained for different Re_L differ insignificantly.

In analyzing the experimental results on heat transfer, we used the Nusselt number averaged over the cavity, $\overline{Nu}_L = \bar{\alpha}L/\lambda$, and the Reynolds number calculated from the distance between the fins, $Re_L = u_0L/\nu$, as the determining criteria. In Fig. 4 we give experimental data on the average heat transfer in flow over fins spaced at different numbers of calibers (L/h = 1, 2, and 3: points 1-3). The experimental data are not generalized in such a representation, and the heat-transfer intensity increases with increasing numbers of calibers between the fins. The character of their behavior for L/h = const has similar features, however. In Fig. 5 we show the ratio of the average heat transfer in the cavity between fins to the analogous quantity in the absence of fins, $\overline{Nu}_L/\overline{Nu}_{L,sm} = f(Re_L)$. The influence of the fins on the heat-transfer intensity was thus analyzed in pure form.

The following conclusions must be drawn from Fig. 5. The experimental curves obtained for different numbers of calibers (L/h = 1 and 3: points 1 and 3) are similar to each other. They have two characteristic regions: a straight descending section at low Re_L and then an abrupt increase with the formation of a fairly smooth maximum. The most interesting result from a practical standpoint is that the average heat transfer between the fins was 1.5-3 times lower than for flow over a smooth surface. As one would expect, the heat-transfer intensity increases with increasing distance between fins.

The experimental data on average heat transfer for different numbers of calibers of the cell between fins were generalized using the ratio L/h as the characteristic parameter. The results of such generalization are shown in Fig. 6. The experimental data are grouped about two lines. Line 1 corresponds to the equation

$$\overline{Nu}_L = 0{,}306 Re_L^{0{,}5} (L/h)^{0{,}4}, \tag{1}$$

in which the exponent to the Reynolds number corresponds to laminar heat transfer. Line 2, describing the second group of experimental points, is

$$\overline{Nu}_L = 0{,}0668 Re_L^{2/3} (L/h)^{0{,}4} \tag{2}$$

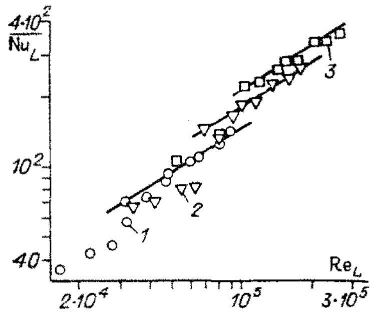

Fig. 4

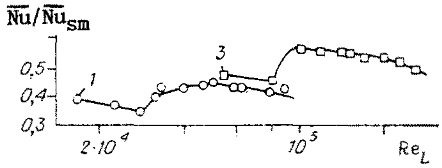

Fig. 5

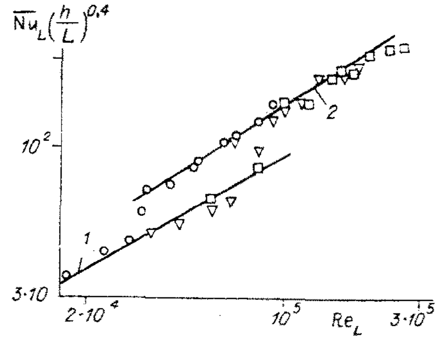

Fig. 6

with an exponent 2/3 to the Reynolds number, and it characterizes heat transfer in detached turbulent flows [1], when diffusional heat transfer makes the main contribution to the transfer processes.

For a fixed distance between fins (L/h = const), the experimental points, judging from Fig. 6, at first satisfy the laminar function (1) for small Re_L, and then after the transition, they satisfy the turbulent function (2). The Reynolds numbers of the transition from laminar to turbulent heat transfer are not fixed, but increase with increasing distance between fins.

Test results on heat transfer of a qualitatively similar nature have been established in an investigation of convective heat transfer in rectangular cavities with a heated bottom [10]. It does not seem possible to make a detailed comparison and bring out common relationships of heat transfer at present, however, mainly because of the significant differences between the aerodynamic patterns formed in flow over a recess and over the cavity between two fins. The mechanism of formation of the counterflows that penetrate into the cavity between fins remains unclear. A detailed experimental study of the aerodynamics of the flow is needed to clarify it. It also seems advisable to further expand the field of research on heat transfer in a cavity between fins for other values of L/h, with variation of the absolute fin height h.

An important aspect is the study of the influence of the fin thickness S on the aerodynamics of detached flows in the cell to determine the conditions for a transition from the functions describing heat transfer in the space between fins to the analogous equations for a rectangular cavity.

LITERATURE CITED

1. P. K. Chang, Separation of Flow, Pergamon Press, Oxford—New York (1970).
2. M. G. Ktalkherman, "Heat transfer to a plate behind an obstacle," Prikl. Mekh. Tekh. Fiz., No. 5 (1966).
3. A. I. Gnyrya, External Heat and Mass Transfer in Concreting with an Electrically Heated Mixture [in Russian], Tomsk. Gos. Univ., Tomsk (1977).
4. V. I. Gomelauri, R. D. Kandelaki, and M. E. Kipshidze, "Intensification of convective heat transfer by artificial roughness," in: Problems of Convective Heat Transfer and the Purity of Steam [in Russian], Metsniereba, Tbilisi (1970).
5. É. B. Kalinin, S. A. Yarkho, and G. A. Dreitser, Intensification of Heat Transfer in Channels [in Russian], Mashinostroenie, Moscow (1972).
6. M. Kh. Ibragimov, V. I. Subbotin, V. P. Bobkov, et al., Structure of a Turbulent Stream and Mechanism of Heat Transfer in Channels [in Russian], Atomizdat, Moscow (1978).
7. F. Durst and A. K. Rastogi, "Turbulent flow behind two-dimensional obstacles," in: Turbulent Shear Flows. 2 [Russian translation], Mashinostroenie, Moscow (1983).
8. F. Durst and M. Founti, "Experimental and computational investigation of the two-dimensional channel flow over two fences in tandem," Tr. ASME, J. Fluids Eng., 110, No. 1 (1988).
9. Liu, Chan, and Huan, "Experimental and numerical investigation of turbulent flow in a channel containing two pairs of turbulizers on the walls," Sovrem. Mashinostr., Ser. A, No. 2 (1991).
10. Yamamoto, Seki, and Fukusako, "Heat transfer by forced convection from the heated bottom of a cavity," Tr. ASME, J. Heat Transfer, 101, No. 3 (1979).