

Heat transfer from a channel wall with discrete heating

© I.A. Davletshin, N.I. Mikheev, R.R. Shakirov

Institute of Power Engineering and Advanced Technologies, Federal Research Center, Kazan Scientific Center of the Russian Academy of Sciences, Russia

E-mail: davlet60@mail.ru

Received March 25, 2026

Revised April 21, 2026

Accepted April 21, 2026

Heat transfer from a wall of a rectangular channel with discrete (in streamwise direction) heating was estimated experimentally. Distributions of heat transfer coefficient were obtained at different ratios of heated to unheated areas in developed turbulent air flow. Discrete arrangement of heat transfer was demonstrated to promote the heat transfer coefficient in the entrance section compared to uniform heat transfer.

Keywords: discrete heating, entrance length, thermal boundary layer.

DOI: 10.61011/TPL.2026.07.63842.20692

The distribution of heat flows on the heat exchange walls is generally uneven. In particular, the initial heat transfer phase is characterized by increased values of the heat transfer coefficient (HTC) [1]. The HTC values decrease downstream and reach stable values after reaching the number of diameters $x/d \sim 50$. This situation is typical of both laminar and turbulent flow modes of coolants. In particular, the values averaged over the length of $x/d = 1$ directly behind the inlet section exceed the corresponding stabilized values (up to $\sim 90\%$ for laminar modes and up to $\sim 65\%$ for turbulent modes) [2]. At the same time, it can be expected that this excess can be manifold directly near the entrance edge [3].

In practice, many types of heat transfer intensifiers (protrusions, wells) are based on the principle of thermal boundary layer destruction [4–7]. From the same point of view, discrete heating (cooling) of the wall can be considered: each subsequent heating site will form its own thermal boundary layer with increased HTC values. Such an organization of heat transfer can be characteristic, for example, in electronics, when it is necessary to cool discretely located heat sources [8]. In modern conditions, this task is especially large-scale and relevant for data storage and processing centers (data centers) [9]. Thus, this paper is aimed at identifying the features of heat transfer during discrete (intermittent) heating of the heat exchange wall of the channel.

The study was conducted using an experimental setup (Fig. 1), which consisted of a flat rectangular channel with a width of 180 mm and a height of $H = 20$ mm. The hydraulic diameter of the channel was $d_h = 36$ mm. The channel consisted of pre-connected 1 and heat exchange 3 sections with lengths $L_0 = 2.5$ m and $L_T = 455$ mm, respectively. The presence of a pre-connected section with a length $8 \times 8 \times E$ made it possible to form a developed turbulent velocity profile at the entrance to the heat exchange section. There was an infra-red transparent window 2 with a diameter of 120 mm on the opposite (upper) wall of the

heat exchange section, designed for optical measurements of the wall temperature, which were performed using a stationary VarioCam HD head thermal imager. Thus, it was possible to obtain information about the temperature state of both heated and non-heated wall sections.

The air consumption in the installation was created by a turbocharger operating on suction. A nozzle block with a set of critical flow nozzles was installed between the duct and the turbocharger. The air flow (velocity U) values were set by opening a set of nozzles with different bore diameters. The extended relative uncertainty of the flow through the nozzles with a coverage factor of 2 did not exceed 0.25 %. Thus, the air was supplied to the working area of the experimental installation from the environment (laboratory premises). There were no other effects on the incoming air in the experiments. There was a large-volume free space with still air in front of the entrance device.

The measurements of temperature T_f of the ambient air were carried out using a platinum Pt100 resistance thermometer, which was installed at the channel entrance. To organize the heat exchange process between the wall and the air flow, the experiments involved heating the measuring wall with direct electric current from batteries with EMF $E = 24$ V.

The measurement of the local values of the heat transfer coefficient $\alpha = q_w / (T_w - T_f)$ was performed using a technique that allows heating the heat exchange surface while simultaneously determining its local temperatures T_w based on measuring the associated electrical resistances of the heating elements. For this purpose, the lower part of the wall of the working section was a printed circuit board with a length of 455 mm, a width of 180 mm and a thickness of 1.5 mm. On the side of the board facing inside the channel, the copper coating was etched in the form of zigzag tracks. The tracks were localized on rectangular sections measuring $14 \times 14 \times E$ mm. The number of such sections, located densely one after the other in the longitudinal direction with a step of 3 mm, was 150 pieces. Each

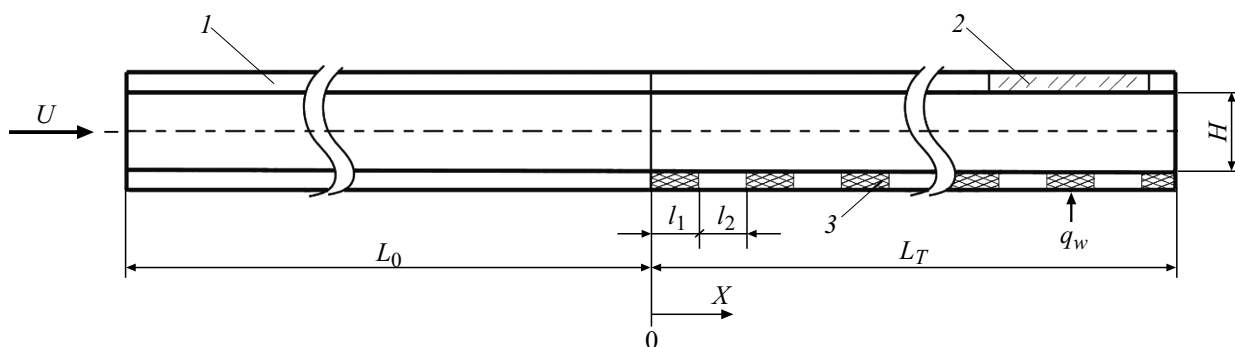


Figure 1. Experimental setup. 1 — upstream section, 2 — optical window, 3 — heat exchange section.

track had its own current leads, which allowed them to be connected to a current source in a serial or parallel circuit. In the extreme case, each element could be connected to its own current source. The central parts of the tracks, measuring 3×80 mm, had electrical taps running along the back of the board to measure the voltage drop in the circuit section. Electrical voltage measurements were carried out in order to determine the heat generation and electrical resistances of the sections tracks and then the temperatures of the corresponding wall sections based on the temperature dependence of the resistances. The outer surface of the heat exchange wall was insulated to minimize heat loss. The error limit for measuring the heat transfer coefficient was $\pm 5\%$. A detailed description of the HTC measurement method is presented in Ref. [3].

The experiments were conducted in various configurations of wall heating. First, heat transfer measurements were carried out during continuous heating of the wall. Discrete heating of the wall was investigated in two variants:

- option 12-12: the lengths of the heated sections $l_1 = 12$ mm, non-heated $l_2 = 12$ mm (this option is shown in Fig. 1, the heated sections of the wall are shaded);
- option 12-24: $l_1 = 12$ mm, $l_2 = 24$ mm.

In all variants, the width of the heated wall sections was 150 mm. The experiments were carried out in the range of Reynolds numbers $Re = U d_h / \nu = (0.63 - 6.50) \cdot 10^4$.

The values of $Nu = \alpha d_h / \lambda$ for both continuous and discrete heating were obtained by averaging the local values over a 12 mm long wall section along the axis x . With discrete heating, the Nu values correspond to the average heat transfer of the heated elements, and with continuous heating, — to the average local values for elements of the same area. The average area temperature of the mentioned elements was used as the wall temperature. The results obtained for the three heating options of the heat exchange wall are shown in Fig. 2. The data is normalized by the value Nu_0 for the developed turbulent flow in the channel

$$Nu_0 = 0.023 Re^{0.8} Pr^{0.4}. \quad (1)$$

As can be seen from Fig. 2, with continuous heating, the local HTC in the initial section of the thermal boundary layer formed under conditions of a developed dynamic

boundary layer decreases very rapidly, already at a distance of $x/d_h = 10$ approaching Nu_0 . With discrete heating, the temperature of the heated areas also decreases with an increase in x , but the rate of decrease is much lower, especially with the longer of the lengths of the non-heated areas (24 mm). A trend for stabilization of Nu is seen with increase of x but at the level of higher than Nu_0 . The experimental change in Nu along the longitudinal coordinate is consistent with our ideas about the formation of a thermal boundary layer at the discrete heating site. The thickness of the thermal boundary layer below the heating start point continuously increases until the boundary layers close, and with unilateral heating — to a value corresponding to the height of the channel at which the HTC should stabilize. In the case of discrete heating in an unheated area in the wall area, the temperature of the coolant decreases, which, under conditions of a thermally insulated wall, occurs mainly due to convective heat transfer from the wall area to the core of the flow. The role of the thermal conductivity of the wall is small compared to convection: according to the calculated estimate, the transfer of heat from the heated area along the wall is more than an order of magnitude lower than the convective heat flow. This estimate, performed on the basis of Fourier's law using data from Fig. 3 to determine the wall temperature gradient, showed that the proportion of heat transferred by thermal conductivity along the wall from the heated area to the non-heated area is within $\sim 5\%$ of the heat transferred from the wall to the flow in the heated area. Thus, with discrete heating, the thermal boundary layer in the wall region remains nonequilibrium even after the formation of a thermal boundary layer over the entire height of the channel. This means that a new thermal boundary layer begins to form on the next heated element, which partially collapses again in the area without heating. The further away the heated areas are from each other, the more fully the conditions for updating the thermal boundary layer on each of them are created.

Since the intensity of convective heat transfer is related to the thickness and condition of the thermal boundary layer, the nature of the change in the HTC along the longitudinal coordinate could be predicted even before the experiments. The experiment allowed obtaining quantitative

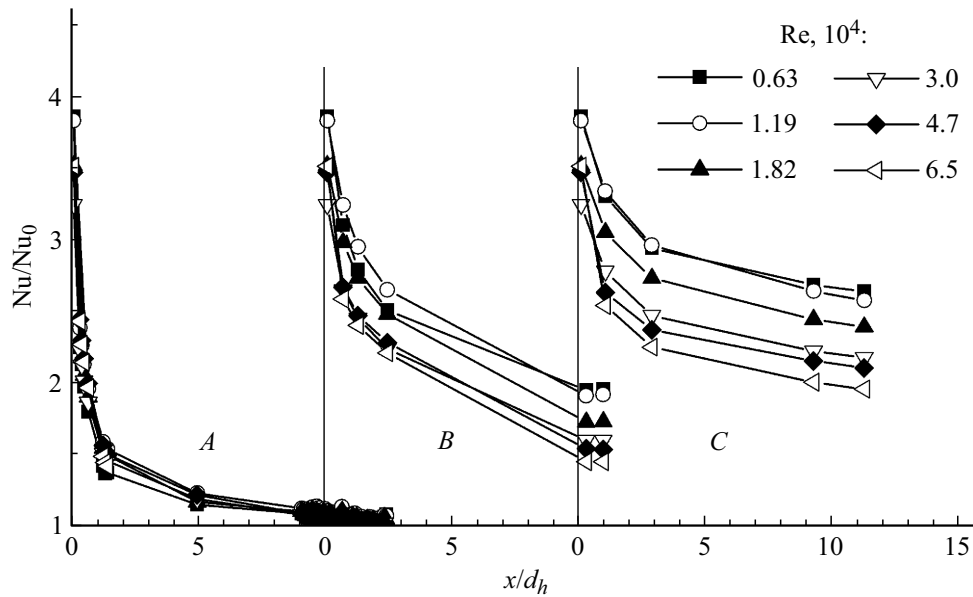


Figure 2. Distribution of the heat transfer coefficient in the initial thermal section of the channel for various wall heating options: *A* — continuous, *B* — 12-12, *C* — 12-24.

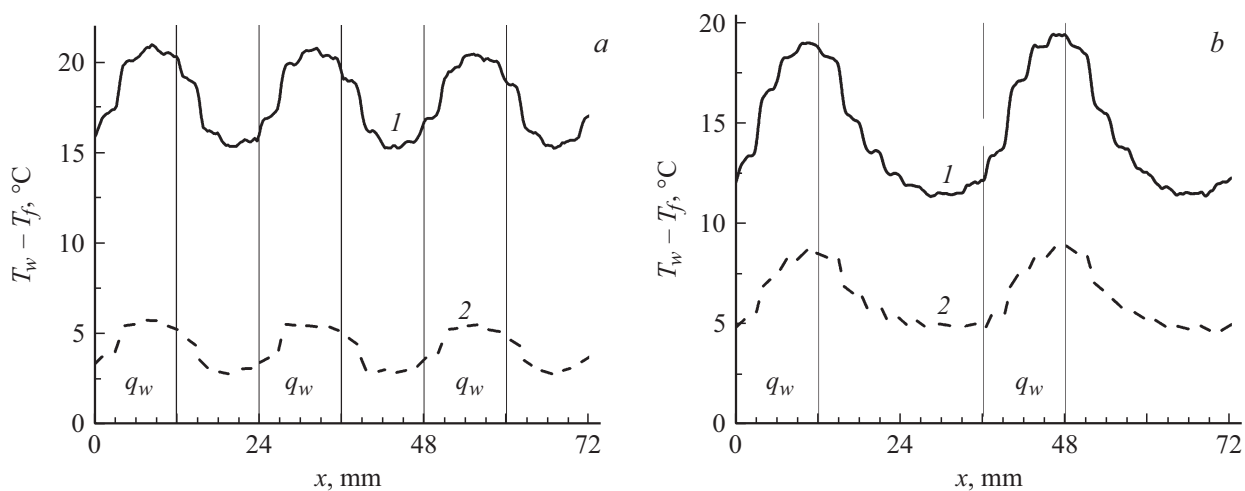


Figure 3. The wall temperature for heating options 12-12 (*a*) and 12-24 (*b*) for different modes according to Re . $Re = 0.63 \cdot 10^4$ (1) and $6.5 \cdot 10^4$ (2).

characteristics. With the length of the section used in the work for averaging the $l_1 = d_h/3$ ratio, the Nu/Nu_0 value for the first heated section in the region of low experimental values of the Reynolds number was 3.9. As the Reynolds number increased, this ratio decreased monotonously to 3.4. Obviously, with discrete heating in the limit at $l_2 \gg l_1$, the same Nu/Nu_0 values can be expected. As follows from Fig. 2, even at $l_2 = 2l_1$, the ratio Nu/Nu_0 quite closely approaches the values indicated above. At the same time, the trend towards a decrease in Nu/Nu_0 remains with an increase in the Reynolds number. It should be noted that in the classical initial section of the channel with the simultaneous formation of dynamic and thermal boundary layers, the influence of the Reynolds number on the Nu/Nu_0 ratio occurs in the same direction [2].

The results of thermal imaging measurements of the wall temperature during discrete heating are shown in Fig. 3. The measurements were carried out in the end part of the heat exchange wall $x/d_h > 10$. The data is presented as a temperature pressure ($T_w - T_f$), the distribution of which at a constant value of the flow temperature T_f can be considered as the distribution of the wall temperature T_w . The coordinate system ($0x$) here is associated only with the measuring area. The presence of a heat flow q_w is shown in the heated areas (their boundaries are represented by vertical lines). It can be seen from the distributions that the temperatures of the non-heated areas have intermediate values (from the flow temperature to the temperatures of the heated areas). The highest wall temperature corresponds to the end (downstream) of the heated area, where the

minimum local heat transfer coefficient is realized due to the greatest thickness of the newly formed thermal boundary layer. The gas in the trickle closest to the wall has a wall temperature, but this trickle of high-temperature gas in the unheated area gradually erodes, while the temperature in the wall area and, accordingly, the temperature of the unheated wall decrease.

With discrete heating, the values of $\langle \alpha \rangle = \langle q_w \rangle / (\langle T_w \rangle - T_f)$ averaged over the entire area are also of particular interest, where $\langle q_w \rangle = q_w l_1 / (l_1 + l_2)$ — is the area-averaged heat flux density, $\langle T_w \rangle$ — is the average integral wall temperature obtained by averaging the measurement results with a thermal imager for heated and non-heated areas. The values of $\langle Nu \rangle / Nu_0$ ($\langle Nu \rangle = \langle \alpha \rangle d_h / \lambda$) in the region of almost stabilized values at $x/d_h > 10$ turned out to be quite close to unity, and for the $l_2 = l_1$ heating scheme in the entire range of Reynolds numbers. However, the $l_2 = 2l_1$ heating scheme corresponded to continuous heating only in the range of minimum Reynolds numbers according to experimental conditions, and with an increase in the Reynolds number, the ratio $\langle Nu \rangle / Nu_0$ gradually decreased to a value of 0.72 at the maximum experimental value of the Reynolds number. Obviously, with a further increase in the ratio l_2/l_1 , the ratio $\langle Nu \rangle / Nu_0$ will continue to decrease.

Thus, due to the renewal of the thermal boundary layer during discrete heating, the local heat transfer of the heated elements is significantly increased compared to the case of continuous heating. At low l_2/l_1 and low Reynolds numbers in the region of stabilized heat transfer, this increase is close to the value of $Nu/Nu_0 = (l_1 + l_2)/l_1$, at which the decrease in the actual heating area is fully compensated by an increase in heat transfer from the heated elements. In general, the integrated HTC under discrete heating turned out to be close to the value of HTC under continuous heating, but it loses, and the loss appeared only for the scheme $l_2 = 2l_1$ and increases with increasing Reynolds number.

Funding

The study was financed by the grant from the Russian Science Foundation (No 22-19-00507-P).

Conflict of interest

The authors declare that they have no conflict of interest.

References

- [1] N.L. Kafengauz, M.I. Fedorov, *Atomnaya energiya*, **27** (3), 552 (1969) (in Russian).
- [2] M.A. Miheev, *Osnovy teploperedachi* (Gosenergoizdat, M., 1949) (in Russian).
- [3] I.A. Davletshin, N.I. Mikheev, A.A. Paereliy, I.M. Gazizov, *Int. J. Heat Mass Transfer*, **129**, 74 (2019). DOI: 10.1016/j.ijheatmasstransfer.2018.09.066
- [4] E.K. Kalinin, G.A. Drejcer, I.Z. Kopp, A.S. Myakochin, *Effektivnye poverhnosti teploobmena* (Energoatomizdat, M., 1998) (in Russian).
- [5] I.A. Davletshin, N.S. Dushin, N.I. Mikheev, R.R. Shakirov, *Int. J. Therm. Sci.*, **204**, 109202 (2024). DOI: 10.1016/j.ijthermalsci.2024.109202
- [6] V.I. Terekhov, N.I. Yarygina, Ya.I. Smulsky, *Thermophys. Aeromech.*, **15** (2), 205 (2008). DOI: 10.1134/S0869864308020042.
- [7] S.A. Isaev, M.S. Gritckevich, A.I. Leontiev, O.O. Milman, D.V. Nikushchenko, *Int. J. Heat Mass Transfer*, **145**, 118737 (2019). DOI: 10.1016/j.ijheatmasstransfer.2019.118737
- [8] C.J. Ho, J.Y. Chang, *Int. J. Heat Mass Transfer*, **37** (6), 917 (1994). DOI: 10.1016/0017-9310(94)90217-8
- [9] A.A. Arbatskij, V.S. Glazov, *Vestn. MEI*, No 4, 36 (2018) (in Russian). DOI: 10.24160/1993-6982-2018-4-36-43

Translated by A. Akhtyamov