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FORCED CONVECTION IN A PLANE CHANNEL WITH RECESSES

Yu. A. Gavrilov, G. N. Dul'nev, and A. V. Sharkov

UDC 536.253

An approximate method is suggested for estimating the coefficient of convective heat exchange in fluid flow in a flat channel with rectangular recesses in the walls.

We will consider convective heat exchange in steady, formed, fluid flow in a flat channel with rectangular recesses in the walls (Fig. 1). These recesses are identical, are equally spaced apart, and have dimensions B and H commensurate with the width h of the channel. The convective heat-exchange coefficients are to be determined.

We could not find the solution to such a problem in the literature, although much attention is paid to the effect of roughness on heat exchange. Sandy roughness has been investigated in relatively great detail [1]. There are reports where roughness differing from the sandy kind is considered. For example, vortex flow in small recesses is studied in [2]. Their dimensions are small in comparison with the channel width, and they have almost no effect on the character of the main fluid flow. A coarser roughness in the form of a spiral protuberance of Nichrome wire on the inner surface of a round pipe is adopted in [3]. The height of the protuberance is about one-tenth the pipe diameter. The experimentally derived dependence for calculating the convective heat-exchange coefficient is valid for the particular case. A set of empirical equations for concrete forms of surfaces in heat exchangers is given in [4, 5]. Data of an experimental investigation of convective heat exchange in laminar and turbulent air flow in a channel with the geometrical parameters B = 13 mm, D = 15 mm, H = 5 mm, and h = 1-10 mm are presented in [6]. In addition to transverse recesses, the walls also had longitudinal recesses. The experimental results for these channels were generalized in the form of criterial dependence $\overline{\mathrm{Nu}} = f(\mathrm{Re})$.

The longitudinal flow of a stream over a surface with a single recess or protuberance is analyzed in a number of reports, such as [7-11]. A simplified flow model is chosen in this case and one or another approximate solution of the problem is given in accordance with the adopted assumptions. In [12] the flow model is extended to a surface with protuberances arranged in a series.

Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 35, No. 5, pp. 812-819, November, 1978. Original article submitted October 5, 1977.

 0022-0841/78/3505-1294 $07.50

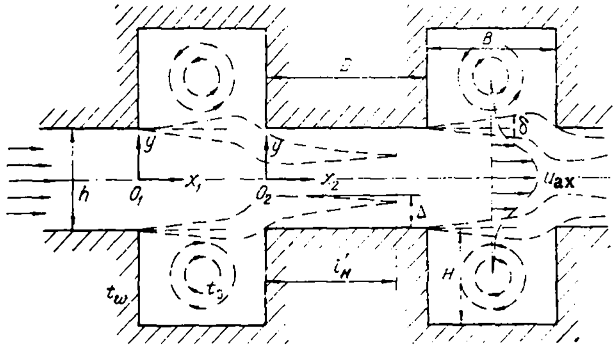

Fig. 1. A channel with rectangular recesses.

In the present report a similar flow model is adopted to analyze the heat exchange in a flat channel for which the wall profile is close in form to that of the surface considered in [12], and is represented in Figs. 1 and 4. The functions obtained for the simplified flow model are refined on the basis of experimental investigations and a method is suggested for calculating the average heat-exchange coefficient for flat channels with protuberances.

The average convective heat-exchange coefficient is determined by the equation

$$(B+D)\,\overline{\mathrm{Nu}} = B\overline{\mathrm{Nu}}_B + D\overline{\mathrm{Nu}}_D. \tag{1}$$

Here each of the numbers $\overline{\mathrm{Nu}}$, $\overline{\mathrm{Nu}}_B$, and $\overline{\mathrm{Nu}}_D$ pertains to sections of the channel with lengths B + D, B, and D, respectively, and denotes the dimensionless complex

$$\overline{\mathrm{Nu}} = \frac{\bar{\alpha} d_e}{\lambda};\ d_e = 2h;\ \bar{\alpha} = \frac{q}{\overline{\Delta t}};\ \overline{\Delta t} = \frac{1}{L}\int_0^L (t_w - t)\,dx. \tag{2}$$

The quantity $\overline{\mathrm{Nu}}_D$ will be calculated with the aid of the approximate functions obtained in [13], which allow for the effect of the initial section l_H of heat exchange in a flat channel with smooth walls. We will therefore assume that the distance D between the recesses is greater than the channel width h, and a section D can be treated as a plane channel with smooth walls. The equations presented in [13] consist of functions of the type $\mathrm{Nu} = \mathrm{Nu}(l_H)$, in which the quantity l_H is the argument and is calculated for uniform velocity and temperature profiles of the stream at the channel entrance. In the case under consideration the value of l_H will later be corrected with allowance for the presumed velocity and temperature profiles of the stream at the entrance to the smooth section D of the channel.

To estimate $\overline{\mathrm{Nu}}_B$ we use the flow model from [7-12], the results of the investigation in [7], and the simplified scheme of a flooded jet [14-18]. In section B of the channel one can distinguish three zones [7, 8]: a zone of circulation flow (the recess), a zone of jet flow, and an intermediate zone. We will consider the flow properties in each of them.

The parameters of a flat flooded jet which flows out of a slot with a width h are known from the theory of jet flows [1, 14, 15]. A simplified diagram of a jet in free space is presented in Fig. 2. In the initial section l_0 of the jet in its central part (the core of the jet) the particle velocity remains the same as at the exit from the slot. Its size is

$$l_0 = 5h. \tag{3}$$

The boundary layer occupies the rest of the jet. The thickness δ of its inner part at $x \leq l_0$ is roughly estimated from the equation [1, 15]

$$\delta = 0.1x. \tag{4}$$

In [17, 18] it is shown that the conclusions drawn for a free jet are also practically retained for a jet propagating in a confined space.

The flow of a fluid bathing a surface with a rectangular recess (Fig. 3a, b, c) is discussed in [7, 8]. Circulation flow, which is confirmed by smoke photographs, forms in a recess. If one of the dimensions B or H of the recess exceeds the other by several times, then secondary circulation flows form. The temperature t_0 of the fluid in the region of the recess is close to uniform. The average temperature $\overline{t}_w$ of the wall of the recess is connected with t_∞ and t_0 by the dependence [7]

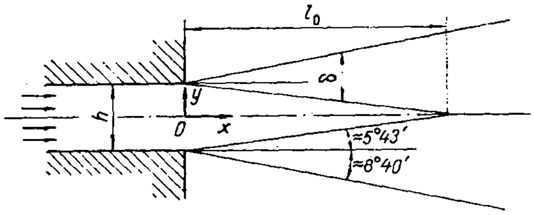

Fig. 2. Simplified diagram of a free jet.

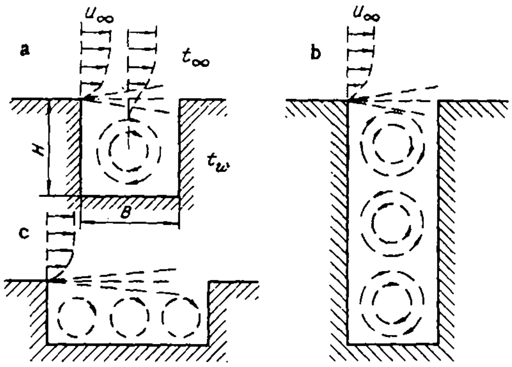

Fig. 3. Fluid flow in the region of a rectangular recess.

$$\frac{t_0 - t_\infty}{t_w - t_\infty} = \left[A \left(1 + \frac{B}{H} \right) \right]^{-0.5}, \tag{5}$$

where A is an empirical coefficient; A = 2 for laminar and A = 3 for turbulent fluid flow.

To solve the stated problem we will use the experimentally confirmed dependence (5) and supplement the flow model in accordance with [10].

The flow in the region of a recess in the channel will be treated as a plane flooded jet flowing out of a slot with a width h into the recess (h + 2H) and, as shown in Fig. 1, the intermediate zone will be divided into two parts by thickness: upper and lower. The lower part of the flow is adjacent to the vortical core in the recess. In this part the flow turns downward near the back rim of the recess and curls into a vortex. The upper part of the flow (as well as the fluid particles penetrating into it from the vortical core) forms a boundary region of flow at the entrance to the smooth section D of the channel. As a result, a boundary layer with a leveled temperature and velocity profile forms in this section; in accordance with (4) at x = B its thickness Δ is

$$\Delta = \delta|_{x=B} = 0.1\, B. \tag{6}$$

Since dependence (4) is valid only in the initial section of the jet ($x < l_0$), a limit emerges to the dimensions B and H of the channel, viz., the dimension B must not significantly exceed the value of l_0 and, in accordance with (3),

$$B < 5h, \tag{7}$$

or else the model adopted above will not be observed.

Furthermore, the scheme of the jet changes if its outer boundary touches the bottom of the recess. In order to prevent this, we limit the dimension H:

$$H > 0.1\, B. \tag{8}$$

Now let us apply the dependence (5) to the flow model under consideration. For this we must change from the temperature t_∞ to the temperature t_{ax} along the channel axis averaged over the section B. We assume that the structure of Eq. (5) is retained in this case, i.e.,

$$\frac{t_0 - t_{ax}}{t_w - t_{ax}} = \frac{1}{\sqrt{A}} \; \frac{1}{\sqrt{1 + \frac{B}{H}}}. \tag{9}$$

We make one more assumption: on the left-hand side of Eq. (9) the average temperature t_{ax} in the section $0 \le x \le B$ can be replaced by the average-mass temperature $\bar{t}_a$ of a flooded jet in this section, i.e.,

$$\frac{t_0 - t_{ax}}{t_w - t_{ax}} = \frac{t_0 - \bar{t}_a}{t_w - \bar{t}_a} = \frac{t_0 - \bar{t}_a}{\Delta \bar{t}}. \tag{10}$$

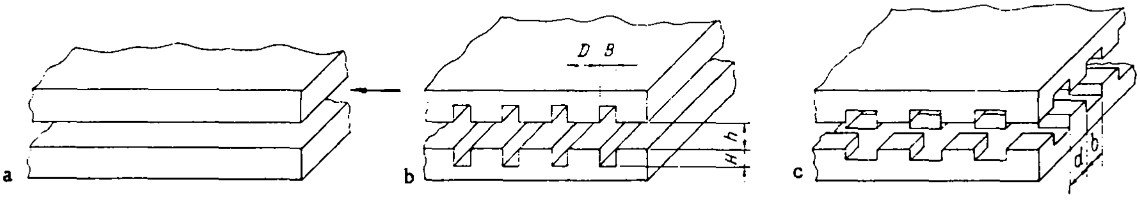

Fig. 4. Schematic diagrams of test channels: a) No. 1; b) Nos. 2-9; c) Nos. 10 and 11.

On the basis of Eqs. (2), (9), and (10) we write the expression for the average heat-exchange coefficient $\bar{\alpha}_B$ in the section B:

$$\bar{\alpha}_B = \frac{q}{\bar{t}_w - \bar{t}_a} = \frac{qK}{t_0 - \bar{t}_a}; \quad K = \frac{1}{\sqrt{A\left(1 + \frac{B}{H}\right)}}. \tag{11}$$

By definition the local heat-exchange coefficient α_D at x = 0 is

$$\alpha_D|_{x=0} = \alpha_{D0} = \left.\frac{q}{t_w - \bar{t}_a}\right|_{x=0}.$$

We note that at x = 0 the temperature of the boundary of the jet must equal the temperature t_0 of the core, since the thickness h' of the intermediate layer equals zero at this point. Suppose that the thickness of the boundary layer is also equal to zero at the rim; then $t_w = t_0$ at x = 0 and α_{D0} can be represented in the form

$$\alpha_{D0} = \frac{q}{t_0 - \bar{t}_a}, \tag{12}$$

while (11) can be rewritten as

$$\bar{\alpha}_B = K\alpha_{D0}.$$

We introduce the average Nusselt number in the section B and the local Nusselt number at x = 0,

$$\overline{\mathrm{Nu}}_B = \frac{\bar{\alpha}_B h}{\lambda}; \quad \mathrm{Nu}_{D0} = \frac{\alpha_{D0} h}{\lambda}, \tag{13}$$

and we establish the connection between them on the basis of (11) and (12):

$$\overline{\mathrm{Nu}}_B = \frac{\mathrm{Nu}_{D0}}{\sqrt{A\left(1 + \frac{B}{H}\right)}}; \quad \sqrt{A} \approx 2. \tag{14}$$

It was indicated that $1.4 \leq \sqrt{A} \leq 1.8$ in Eq. (5), but in (14) the value of $\sqrt{A}$ is taken as equal to two. Some overstatement of the value of the factor $\sqrt{A}$ partially compensates for the error due to the assumption (10) and gives better agreement with the experimental data.

So, on the basis of (1) and (14), the solution of the problem is reduced to the search for the local and average Nusselt numbers Nu_D and $\overline{\mathrm{Nu}}_D$ in the smooth section D of the channel. For this one must know the temperature and velocity distribution of the flow at the entrance to this section. This distribution will be nearly uniform if the boundary layer Δ fills the entire cross section and, on the basis of (6),

$$\Delta = 0.1\,B \quad 0.5\,h. \tag{15}$$

Then the values of Nu_D and $\overline{\mathrm{Nu}}_D$ are determined through the equation from [13].

If condition (15) is not satisfied, i.e., $\Delta < 0.5$ h, then in the central part of the entrance cross section the velocity and temperature profile corresponds to or is close to the profile for stabilized flow. As a result of the stabilization the heat exchange will take place over a smaller length l'_H of the initial section (Fig. 1) than the length l_H with the condition (15).

Adopting a proportional dependence between l_H and l'_H,

$$\frac{\Delta}{0.5h} = \frac{l'_H}{l_H}.$$

TABLE 1. Geometrical Parameters of Channels with h = 5.4

Channel No.	Geometrical dimensions, mm				
	B	D	H	b	d
1	—	—	—	—	—
2	8,0	11,0	2,0	—	—
3	8,0	12,0	4,0	—	—
4	8,0	11,0	6,0	—	—
5	1,5	20,0	2,0	—	—
6	1,5	20,0	4,0	—	—
7	1,0	1,5	6,0	—	—
8	9,0	1,5	6,0	—	—
9	19,5	1,5	6,0	—	—
10	1,5	20,0	4,0	4,0	15,0
11	8,0	12,0	4,0	8,0	12,0

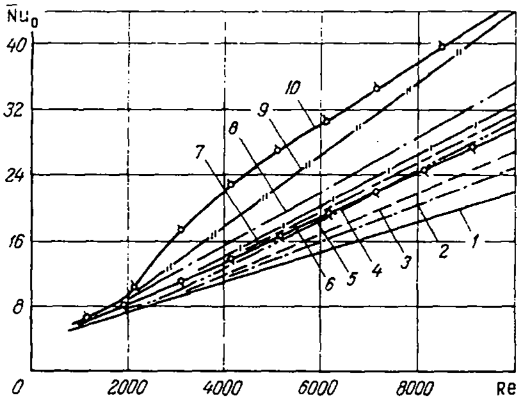

Fig. 5. Experimental results: 1) Channel No. 1; 2) No. 5; 3) No. 6; 4) No. 2; 5) No. 10; 6) No. 3; 7) No. 4; 8) No. 7; 9) Nos. 8 and 9; 10) No. 11.

with allownace for (6) we obtain

$$l_H' = 0.2 \frac{B}{h} l_H. \tag{16}$$

We use this value of l_H' instead of l_H for $\Delta < 0.5h$ to calculate Nu_D and $\overline{Nu}_D$ through the equations from [13], where the quantity l_H is the argument.

In order to allow for the effect of the conditions at the channel entrance on the heat exchange, we find the Nusselt number Nu_{sm} in a channel with smooth walls for the corresponding x coordinate. Then we replace Nu_∞ by Nu_{sm} in the equations from [13]. To calculate the average Nusselt number $\overline{Nu}$ over the entire extent of a channel with recesses, we must replace the quantity Nu_∞ from [13] by the average Nusslt number $\overline{Nu}_{sm}$ for a smooth channel. Such a replacement lets us allow for the variation of the flow velocity and temperature profile along the channel when not disturbed by recesses.

To sum up, Eqs. (1), (14), (16), and the equations from [13] allow us to estimate the convective-heat-exchange coefficient in a channel with rectangular recesses.

The proposed method of estimating $\overline{Nu}$ does not have a rigorous foundation. The calculating equations are extremely simplified owing to a number of assumptions and a priori suppositions. But a calculation of the average convective heat-exchange coefficient gave satisfactory agreement with the results of an experiment in [6]. In addition, we measured the average heat-exchange coefficients $\bar{\alpha}_0$ and $\bar{\alpha}$ (or the criteria Nu_0 and $\overline{Nu}$), normalized to the initial and mean-flow-rate temperature difference as a function of Re with a constant channel wall temperature. The measurement error did not exceed 5% for $\bar{\alpha}_0$ and 9% for $\bar{\alpha}$. In the tests the Reynolds numbers were varied in the range of $5 \cdot 10^2 \leq Re \leq 10^4$. The investigations were conducted in the entrance sections of flat channels 200 mm long and 150 mm wide; the profiles of the heat-transfer surfaces of the channels and their geometrical parameters are presented in Fig. 4 and in Table 1 [19]. The results of the investigations are presented in Fig. 5; curve 1 pertains to a smooth channel; the results of the calculations through the equations of [13] agree well with the tests. We adduce some general conclusions.

At Re > 3000 the recesses intensify the heat exchange and one cannot use the equations for a smooth channel with the equivalent diameter introduced into them. The error in such a calculation exceeds 100% for large Re $\approx 10^4$.

Despite the increase in the heat-transfer surface area of channels with recesses, at Re < 2000 the heat-exchange intensity does not differ from that for smooth channels with the same d_e. For channels differing only in the depth H of the recesses (Nos. 2-6) an increase in $H > 2 \cdot 10^{-3}$ m has an unimportant effect on the heat-exchange intensity.

A comparison of the results of the calculation of $\overline{Nu}$ through Eqs. (1), (14), and (16) and the values of $\overline{Nu}$ for smooth channels with the measurement data leads to the following recommendations.

The proposed calculating scheme is not applicable for channels in which D ≪ B (such as channels Nos. 8 and 9), in which case the faces D cannot be treated as sections of smooth walls, which is assumed in the model.

For channels with only transverse recesses with B ≤ D the proposed method leads to good agreement with experiment. For channels with longitudinal and transverse recesses (such as 10 and 11) the proposed scheme can be set at the basis of the calculation of Nu, but in this case one must calculate the length l'_H from the equation

$$l'_H = 0.2 \left(1 \quad 4\frac{b}{d}\right)\frac{B}{H}\, l_H, \quad 0 \leqslant b/d \leqslant 2, \tag{17}$$

while the equivalent hydraulic diameter must be calculated with allowance for the longitudinal recesses.

The disagreement between the calculated and test data does not exceed 20% in this case.

NOTATION

x	is the longitudinal coordinate;
B and H	are the width and depth of a recess;
D	is the distance between recesses;
h	is the distance between channel walls;
d_e	is the equivalent hydraulic diameter of channel;
l_0	is the length of initial section of jet;
l_H	is the length of initial section of stabilization of heat exchange;
δ	is the thickness of boundary layer of jet;
Δ	is the thickness of flow with a uniform temperature and velocity;
q	is the average heat-flux density normalized to the channel dimension B + D;
α	is the convective heat-exchange coefficient;
λ	is the thermal conductivity of fluid;
u	is the average flow velocity;
Re and Nu	are the Reynolds and Nusselt numbers;
t and t_w	are the average-mass temperature of fluid and wall temperature for the chosen channel cross section.

Indices

upper bar	is the average value;
∞	is beyond the limits of the initial section;
sm	is the channel with smooth walls;
B, D	are for channel sections B or D;
ax	is the jet axis;
0	is the heat-exchange coefficient or criteria normalized to the initial temperature difference.

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INFLUENCE OF THE ASPECT RATIO AND DIAMETER OF THE PROBE FILAMENT OF A THERMOANEMOMETER ON ITS READING

E. U. Repik, A. S. Zemskaya, and V. N. Levitskii

UDC 532.526.4:551.508.5

The results are presented of an experimental investigation of the influence of the aspect ratio of the probe filament of a thermoanemometer and of the absolute value of its diameter on the magnitude of the heat losses from the probe filament to the holders when measuring the average velocity of a gas stream.

The use of a thermoanemometer to measure the average velocity of a gas stream is possible only in the case when the dependence of the heat emission of the probe filament of the thermoanemometer on the physical parameters of the stream and its geometrical dimensions is known. An exact determination of the heat-exchange law of a filament by theoretical means under the actual conditions of flow over it does not seem possible, which leads to the necessity of using calibration equations in practical measurements.

The heat exchange between a filament and a gas stream for infinitely long filaments was studied experimentally by Collis and Williams [1], who suggested the equation

$$\mathrm{Nu}\left(\frac{T_m}{T_\infty}\right)^{-0.17} = 0.24 + 0.56\,\mathrm{Re}_w^{0.45}, \tag{1}$$

where

$$\mathrm{Nu} = \frac{V^2}{\pi\lambda_t l \Delta T R_w};\quad \mathrm{Re}_w = \frac{Ud}{\nu_m};\quad T_m = \frac{T_w + T_\infty}{2}.$$

Filaments of finite length are used in practical measurements, which leads to the necessity of introducing corrections to the heat-exchange law (1) allowing for the heat loss from the proble filament to the probe holders. In recent years the efforts of experimenters have been directed toward the establishment of that minimum admissible relative filament length $(l/d)_{adm}$ for which the measurement error due to the finite filament length is unimportant or can be neglected in practical measurements. The available test data [1-4], however, are in poor agreement with each other, which hinders their practical use. For example, in [1] it is recommended to use filaments with $(l/d)_{adm} \geq 2000$, in [2] $(l/d)_{adm} \geq 700$, while in [3] $(l/d)_{adm} \geq 400$. In [4],

Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 35, No. 5, pp. 820-826, November, 1978. Original article submitted October 5, 1977.

 0022-0841/78/3505-1300 $07.50