STUDY OF THE HEAT TRANSFER AND OF THE HYDRAULIC RESISTANCE ASSOCIATED WITH THE FLOW OF AIR THROUGH NARROW RECTANGULAR CHANNELS

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Results are shown of an experimental study concerned with the heat transfer during various modes of air flow through narrow channels (Re = 360-32,000). The channel dimensions as well as the conditions are established which correspond to a definite effect of natural convection on the heat transfer.

Heat exchangers for use in certain applications of modern technology must satisfy rather stringent requirements dictated by the need for achieving precise thermal operating conditions with optimum weights and sizes as well as with a minimum waste of energy.

Heat exchangers designed in thin-layer stages are most suitable for such applications.

There are not sufficient data available on the heat transfer during a forced air flow through narrow channels [1-3], on the basis of which contact-type heat exchangers operating outside the gravitation field could be designed.

By varying the air pressure in channels of various dimensions, the authors were able to eliminate almost all natural convection and to study the heat transfer during an exclusively forced flow of air through the channels. At that time, the channel dimensions and the critical Grashof number were established at which the effect of natural convection must be considered in the heat transfer calculations.

The tests were performed in a narrow channel made up of parallel copper plates h = 1.5, 3, 5, or 8 mm apart with provisions for cooling. The width of the plates was the same in all channels: b = 100 mm.

As has been noted in [1], a variation in the width of a channel with a large b/h ratio has no appreciable effect on the magnitude of the heat transfer coefficient. The lateral Textolite walls in our channels were almost inactive in the heat transfer, since the heat leakage through them did not exceed 1% of the total thermal flux. There was no duct stage inserted before the entrance to a test channel for hydrodynamic stabilization, rather the front edge of the plates were sharp so as to approximately simulate the entrance conditions in actual heat exchangers. Before the entrance to a test channel and behind it were installed mixers, in order to ensure an adequate stirring of the air ahead of the temperature measurement. The

TABLE 1. Values of the Correction Factor in Formulas (1) and (4) for Calculating the Nusselt Number when 1/h < 100

	Laminar region, Re = 360-1,800					Turbulent region, Re = 5,600-32,000				
l/h	20	40	60	80	100	20	40	60	80	100
K_l	1,86	1,26	1,10	1,03	1,00	1,21	1,08	1,03	1,01	1,00

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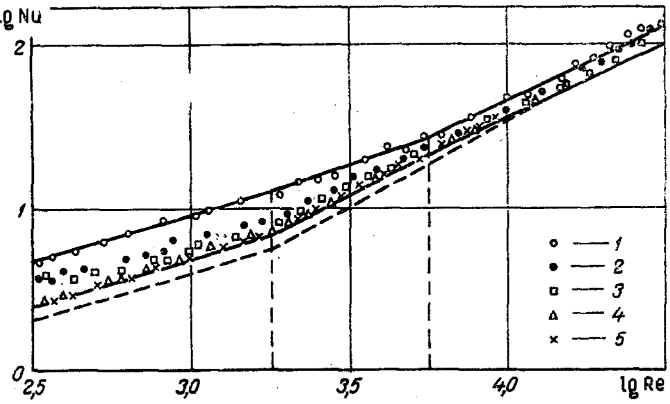

Fig. 1. Test data on the heat transfer in channels with different relative lengths: 1/h = 20 (1), 40 (2), 80 (3), 133 (4), 266 (5).

temperature in the mixer was measured at three points on a transverse section across the air stream: the differences between the thermocouple readings were insignificant. The air temperature was defined as the average readings of the three thermocouples.

The temperature at the walls was measured with copper–constantan thermocouples sealed to the inside plate surfaces. The temperature at the walls was maintained almost constant by means of circulating water with constant temperature through the cooling system at a high velocity. Essentially, all tests were run with the cooled plates in a horizontal position and only for exploring the effect of natural convection on the heat transfer were the plates set up in a vertical position.

The heat transfer coefficient was determined from the expression

$$\alpha = \frac{G_{\Sigma p}(t_1 - t_2)}{F_\Sigma \Delta t_{\log}}.$$

The error of this determination did not exceed 7%.

The basic apparatus and the test procedure have been thoroughly described in [4].

Our studies were conducted over the following range of parameters: air temperature at the heat exchanger entrance t_1 = 30-100°C, air pressure p = 1,330-100,000 N/m^2, wall temperature t_w = 8-12°C, Reynolds number Re = 360-32,000, and Grashof number Gr = 0.25-2,300. The variation of the mean heat transfer coefficient along a channel was determined by using heat exchangers with different relative channel lengths: 1/h = 20, 40, 80, 133, and 266.

The test data for channels with different relative lengths are shown in Fig. 1.

In the evaluation of test data, the physical quantities entering into the similarity ratios were determined from the mean temperature in the test segment of a channel. The distance between plates was taken as the characteristic dimension. With it, the correlation between the heat transfer equation for a circular and for a rectangular channel section was most satisfactory. With the equivalent diameter d_e = 4F/U as the usual characteristic dimension, the variance between heat transfer data for pipes and for rectangular channels was more appreciable.

For this reason, as had been already noted in [5], it was rather difficult to generalize the test data on the heat transfer in channels of various shapes by universal criterial relations.

As can be seen in Fig. 1, for values of the Reynolds number in the 300-1800 the test data for various channel lengths, plotted in logarithmic coordinates, follow approximately straight parallel lines. The test data for channels 1.5, 3, and 5 mm wide can in this range be generalized by the equation:

$$\mathrm{Nu} = 0.1\,\mathrm{Re}^{0.56} K_l, \tag{1}$$

where K_l denotes a coefficient accounting for the effect of the entrance stage. Its values are given in Table 1.

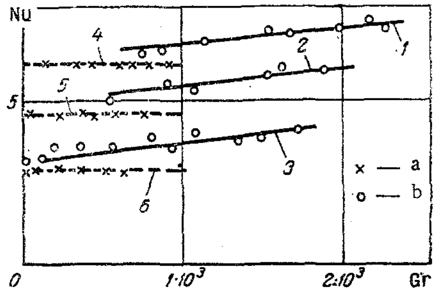

Fig. 2. Relation Nu = f(Gr) for various values of the Reynolds number: Re = 1,550 (1,4), 840 (2,5), 360 (3, 6); h = 1.5, 3, 5 mm (a), h = 8 mm (b).

Our test data indicate that, at relative channel lengths $l/h > 100$, the mean-over-the-length Grashof number remains almost constant and depends on the Reynolds number only. We note that, as in the case of heat transfer in small pipes with d = 1.5-6.0 mm studied in [4], the Reynolds number was much higher here than in the critical relations for channels with large characteristic dimensions. It was quite difficult to detect a significant variation in the exponent of the Reynolds number over such a narrow range of the characteristic dimension in our tests. We will only indicate the existence of such a variation, therefore, by comparing our test data with those of other authors [1, 2]. The tests performed on channels with the plates 1.5, 3, and 5 mm apart have shown that natural convection has no effect at all on the heat transfer during a forced air flow at a Grashof number from 0.25 to 1000. The variation of the Grashof number over this range was effected here by reducing the air pressure in the channel from 100,000 to 1,000 N/m^2. The distance between plates was used as the characteristic dimension in defining the Grashof number. A change in the heat exchanger position relative to gravitational forces had also no effect on the rate of heat transfer.

The test data obtained with the Grashof number varying from 1000 to 0.25 in channels 1.5, 3, and 5 mm wide are shown in Fig. 2 for several values of the Reynolds number.

At the same time, some increase in the heat transfer coefficient for the 8 mm channel was noted even at a Grashof number Gr = 1000. An effect of natural convection on the rate of heat transfer during a forced air flow was revealed only with the plates in a horizontal position. With the plates turned into a vertical position, no such effect could be observed.

With a decreasing Grashof number, according to Fig. 2, the test values of the Nusselt number for the 8 mm wide channel approached the values of this number for the 1.5, 3, and 5 mm wide channels.

An analysis of the test data has yielded the following relation for the 8 mm wide channel:

$$\mathrm{Nu} = \frac{\mathrm{Gr}^{0.1}}{\mathrm{Re}^{0.05}} + 0.1\,\mathrm{Re}^{0.56} K_l. \tag{2}$$

It must be noted that the effect of natural convection becomes somewhat weaker as the Reynolds number increases.

At values of the Reynolds number Re > 1800 one observes a rapid increase of the heat transfer coefficient. In the case of short channels ($l/h < 20$), moreover, the test data for the Re = 1800-5600 range fit into an equation characteristic of a laminar fluid flow. It may be assumed, then, that a laminar flow with Re > 1800 becomes unstable at some distance from the entrance section and, as the Reynolds number increases, the flow mode changes along the entire channel. In the case of longer channels ($l/h > 100$), even at Re > 1800 the test data fit into an equation characteristic of transitional flow. Our test data for channels of various relative lengths are adequately well generalized by the following relation:

$$\mathrm{Nu} = \left[10 - \left(\frac{l}{h}\right)^{0.5}\right] + 4.37 \cdot 10^{-3}\,\mathrm{Re}^{0.98} \tag{3}$$

within the Re = 1800-5600 range.

Within the Re > 5600 range our test data for channels of various relative lengths are generalized by the relation

$$\mathrm{Nu} = 0.013\,\mathrm{Re}^{0.86} K_l. \tag{4}$$

The values of the correction factor K_l for turbulent flow are given in Table 1. It is evident here that the effect of the entrance stage on the mean Nusselt number is much weaker in the turbulent region than in the laminar region, i.e., that at the entrance to the test segment a turbulent air flow stabilizes thermally and hydrodynamically faster than a laminar air flow.

It must also be noted that at Re > 16,000, when the velocity of forced air exceeded 70 m/sec, the dependence of the Nusselt number on the Reynolds number was much stronger than according to formula (4). This peculiarity can, evidently, be explained by the very high degree of turbulence in the air stream, which affected the structure of the boundary layer at such velocities of forced air flow.

We compared our test data on the heat transfer in narrow rectangular channels with data pertaining to small-diameter pipes [4].

Test data for pipes with an inside diameter d = 1.5, 3, 4, 6 mm and with $l/d > 100$ are shown in Fig. 1 with a dashed line. As can be seen here, the test values pertaining to the heat transfer in rectangular channels are slightly higher than those for small pipes, especially in the laminar region.

The test data obtained for the hydraulic resistance in the laminar region fit into the following equation:

$$\lambda = \frac{40}{\mathrm{Re}}\left(\frac{\mathrm{Pr_w}}{\mathrm{Pr_a}}\right)^{1/3}.$$

For the flow region where Re > 5000, the test data fit into the equation:

$$\lambda = 0.29\,\mathrm{Re}^{-0.25}.$$

The mean air temperature in the test segment was used as the temperature parameter with respect to which the test data were evaluated in terms of the hydraulic resistance.

NOTATION

G	is the weight rate of air flow, per second;
c_p	is the specific heat of air at constant pressure;
t_1, t_2	is the air temperature at the entrance to and at the exit from a test channel;
F_Σ	is the total inside surface area of the cooled plates;
$\Delta t_{\log}$	is the logarithmic mean temperature difference between air and channel wall;
F	is the area of a transverse channel section;
U	is the wet perimeter of a channel;
l	is the channel length;
b	is the plate width;
λ	is the drag coefficient;
$\mathrm{Pr_w}$, $\mathrm{Pr_a}$	is the Prandtl number at wall temperature and at air temperature.

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