Journal of Engineering Physics and Thermophysics, Vol. 64, No. 5, 1993

HEAT TRANSFER IN LIQUID MOTION THROUGH A RECTANGULAR CHANNEL

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UDC 536.244; 536.246

The results of an experimental study of heat transfer in water motion through slit channels are presented and compared with data obtained by other authors.

Slit channels are widely used in various fields of engineering (chemical machine construction, refrigeration engineering, solar energy technology, nuclear reactors). A great number of works have been devoted to the investigation of the heat transfer and hydrodynamics in such channels, and many publications continue to appear in different journals and proceedings of scientific conferences.

The heat transfer and hydrodynamics have been theoretically studied for the conditions of laminar developed and developing flows, when the Reynolds number Re < 1000 [1-3]. Keis and London [1] graphically presented the results of the analytical solution obtained for laminar flow through rectangular cross-section pipes with different ratio of the sides, beginning from the square and ending in motion between parallel plates. These solutions are valid only for fully formed velocity and temperature profiles. The solutions for the heat-transfer problem are presented for a constant temperature of the wall or a constant heat flux; in the case of a constant heat flux that is fully stabilized, the solution coincides with that for heat transfer at a constant temperature difference.

In the turbulent-flow regime, the data in [1] are presented only for a circular pipe; it is, however, noted that substituting the hydraulic diameter into the Re expression as the governing linear dimension, one can use the plots for calculating the heat transfer over noncircular cross-section pipes. In [3] and [4], analytical solutions for a fully developed laminar flow are presented. These results coincide with those in [1].

Convective heat transfer under the conditions of developing laminar flow in short elliptic channels was considered in [5]. The influence of the channel geometrical shape, the change in physical properties, and free convection on heat transfer were defined.

To investigate the effect of channel geometrical dimensions on the heat transfer rate, three elliptical channels were made with 10.16 × 7.62 cm, 10.16 × 5.08 cm, and 10.16 × 2.54 cm axes and 0.61 m length. Two segments of 1.22 m length with the same dimensions and geometry as the test section were installed upstream and downstream relative to the test section in order to provide the developing laminar flow in the test section. For account of the mean temperatures of the heat-releasing surface inside the channel, 18 thermocouples were distributed over the periphery. They were positioned at different parts of the channel at a depth equal to the half-width of the channel wall, six thermocouples being placed equidistantly within the central part of the test section. This allowed one to obtain positive values for the mean temperature of the surface inside the channel.

The test section was heated from all sides by an elliptical electrical heater made in the form of Nichrome coils. This kind of heating made it possible, in the opinion of the authors of [5], to maintain a constant temperature of the walls. In the tests they made use of water, a water-glycerin mixture (40, 60, and 80%), and pure ethylene glycol (98%), which permitted them to have a Prandtl number range from 3 to 155. The heat transfer coefficient α was determined on the basis of the arithmetic mean temperature difference. All physical properties of the heat transfer agents indicated in the Colborn factor J were defined at the mean temperatures of these coolants.

The predicted values of the Colborn factor J as a function of the Reynolds number are plotted in the form of a diagram in [5] only for water as the heat transfer agent, since the diagrams for the other coolants are similar. All

Kiev Polytechnical Institute, Kiev. Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 64, No. 5, pp. 515-522, May, 1993. Original article submitted May 13, 1992.

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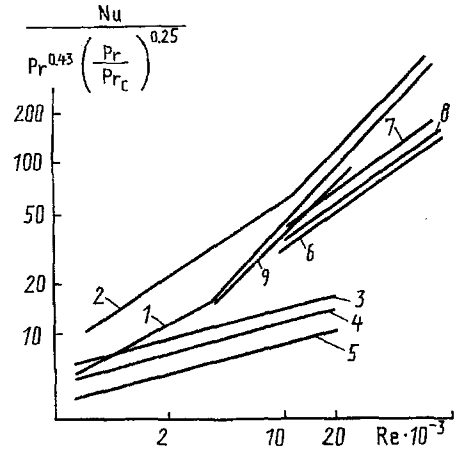


Fig. 1. Comparison of the results of the heat transfer investigation obtained by various authors: 1, 2) [6]; 3, 4, 5) [5]; 6) [7]; 7) [8]; 8) [1, 11-13]; 9) our data.

experimental data obtained in the three elliptical channels are represented by three parallel lines with the identical angular coefficient 0.73.

We recalculated the data of [5] and present them in Fig. 1 (lines 3, 4, 5) in the form of the dependence of the complex $Nu/Pr^{0.43}(Pr/Pr_c)^{0.25}$ on the Reynolds number, up to Re = 20,000.

The equation which correlates the data on the heat transfer obtained under the conditions of a laminar developing flow in short elliptical channels with the properties of the heat transfer agents is of the form [5]

$$J = 3{,}19\,\mathrm{Re}^{-0{,}73}\left(\frac{l}{d}\right)^{-0{,}47}(1 + 0{,}015\,\mathrm{Gr}^{1/3})^{0{,}024}, \tag{1}$$

where $J = St\, Pr^{2/3}(\mu/\mu_c)^{0.14}$ is the Colborn factor; d is the equivalent diameter (within the ranges Re < 20,000 and Pr = 3—155).

Novikov and Shcherbakov investigated of the heat transfer and hydraulic resistance in the air flow through narrow rectangular channels. By varying the air pressure in channels of various dimensions, they were able to eliminate almost all natural convection and to study the heat transfer during an exclusively forced flow of air through the channels.

Investigations were conducted in the narrow channel formed by cooled parallel copper plates with distance between them b = 1.5, 3, 5, and 8 mm. The width of the plates in all channels was the same and equaled a = 100 mm. The pre-enclosed stabilization region at the inlet to the experimental channels was absent, and the plate inlet edges were sharp. To determine the change in the mean heat transfer coefficients over the channel length, heat exchangers with various relative channel lengths were used: l/b = 20, 40, 80, 133, and 266.

In processing the experimental data, the physical quantities included in the similarity criteria were determined by the mean air temperature at the test region of the channel. The distance between the plates was taken as the characteristic dimension.

The data of [6], which we recalculated applying the equivalent diameter, are presented in Fig. 1 (curves 1 and 2).

Within the range of Reynolds numbers Re = 600-3600 for channels with b = 1.5, 3, and 5 mm, the experimental data of [6] are correlated by the equation

$$\mathrm{Nu} = 0{,}1\,\mathrm{Re}^{0{,}56}K_l, \tag{2}$$

where K_l is factor which takes into account the effect of the initial region of the plate.

The experiments conducted in the channels with distance between the plates 1.5, 3, and 5 mm showed that at Grashof numbers 0.25-1000 free convection does not exert any influence on the heat transfer in the forced flow of air. (In determining the Grashof number, the distance between the plates was taken as the characteristic dimension.) At the same time, for the channel with b = 8 mm, even at Gr = 1000, a certain increase in the heat transfer coefficient was noted. In [6], for the channel with b = 8 mm the following relation was obtained:

$$Nu = \frac{Gr^{0,1}}{Re^{0,05}} + 0,1\, Re^{0,56} K_l. \quad (3)$$

It is also noted that, with an increase in the Reynolds number, the effect of free convection on the heat transfer somewhat decreases.

In the range of Reynolds numbers above 3600, a sharp increase in the heat transfer coefficient is observed. At the same time, for short channels ($l/b < 20$) in the range Re = 3600-11,000 the test data reported in [6] can be generalized by an equation characterizing the laminar regime of a liquid flow. For longer channels ($l/b > 100$), even for Re > 3600, the test data are described by an equation characterizing the transitional region of the flow. In the range Re = 3600-11,000, the test data of [6] for different relative channel lengths are generalized by the relationship

$$Nu = \left[10 - \left(\frac{l}{b}\right)^{0,5}\right] + 4,37 \cdot 10^{-3}\, Re^{0,98}, \quad (4)$$

and when Re > 11,000, by

$$Nu = 0,013\, Re^{0,86} K_l. \quad (5)$$

Moreover, in [6] it is also noted that for Re > 32,000, when the forced motion velocity values in channels exceeded 70 m/sec, the values of the mean Nusselt numbers depend on the Reynolds factor to a greater degree than according to Eq. (5). This can be attributed to a very strong flow agitation destroying the structure of the near-wall layer at such high velocities of the forced motion.

The results of the experimental study of the heat transfer in water motion through rectangular cross section channels are presented in [7]. The heat transfer was defined with the help of an alpha calorimeter. The test channel with 7 mm wall width was sectional, which allowed one to vary the cross section from 15 × 50 up to 7 × 50 mm. Three alpha calorimeters were located along the 720-mm-long working segment, with step 210 mm.

The tests were performed in cold and hot water within the temperature range from 30° to 90°C. The magnitude of the error in determining the Nusselt numbers, due to the nonuniformity of the temperature field over the calorimeter core, did not exceed 18%.

The experimental results in [7] on the main heat transfer over a flat channel in the range $Re = 8 \cdot 10^3 - 9 \cdot 10^4$ are presented in Fig. 1 (curve 6) and generalized by the relation

$$Nu = 0,0187\, Re^{0,8} Pr^{0,43} \left(\frac{Pr}{Pr_c}\right)^{0,43}. \quad (6)$$

In [7] the equivalent diameter was taken as the deciding linear dimension, and the mean fluid temperature at the transverse cross section of the channel was taken as the deciding temperature.

A numerical analysis of the heat transfer over the thermal initial region of rectangular channels was performed in [8]. The author employed a two-dimensional mathematical model under the following assumptions: the fluid flow is hydrodynamically stabilized; the flow temperature at the channel inlet is constant and uniformly distributed over the cross section; there are no internal heat sources in the flow; the heat due to friction is negligibly small; the thermophysical properties of the fluid are constant.

The solution to the energy equation in rectangular channels is obtained under the condition of constant wall temperature and constant heat flux density at the channel wall.

Calculations of temperature fields were performed in the range $Re = 10^4 - 10^5$ for air and water flow in channels with the ratio of the sides $\gamma = b/a = 1$, 0.5, 0.33, 0.25, and 0.125. From the obtained temperature distribution the local heat transfer rate over the perimeter and length was determined. Heat transfer coefficients are referred to the difference between the wall temperature and mean mass fluid temperature at the given cross section. The equivalent diameter was used as the deciding dimension.

Data on the heat transfer behind the region of thermal and hydrodynamic stabilization, averaged over the perimeter, are correlated by the equation

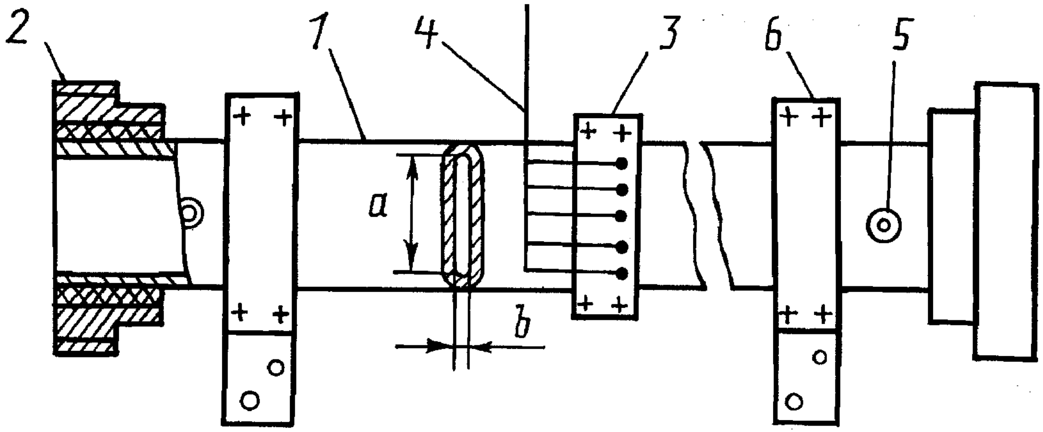


Fig. 2. Test segment.

$$\mathrm{Nu} = \left(0{,}0176 + \frac{0{,}001}{\gamma}\right) \mathrm{Re}^{0,8} \mathrm{Pr}^{0,43}. \quad (7)$$

This relation is shown in Fig. 1 (curve 7) at $\gamma = 0.125$ (at $\gamma = 1$ curve 7 merges with line 6).

It was noted in [8] that, in a turbulent flow, the heat transfer stabilization in rectangular channels is completed at a length of 15d. For the Reynolds numbers Re > $5 \cdot 10^4$, the error value for the thermal initial region depends weakly on Re and is defined by the distance from the beginning of the heated region.

The heat transfer rate at the thermal initial region is affected by the geometric peculiarities of the channel transverse cross section. Rectangular profile channels have the largest region of thermal stabilization. With a decrease in $\gamma = b/a$, the length of the thermal stabilization region decreases.

In certain works [1, 11-13] it was recommended that, for predicting the heat transfer in a flat channel, one should make use of the equations for a round tube, employing the equivalent diameter as the characteristic dimension. In Fig. 1, curve 8 is plotted according to the formula

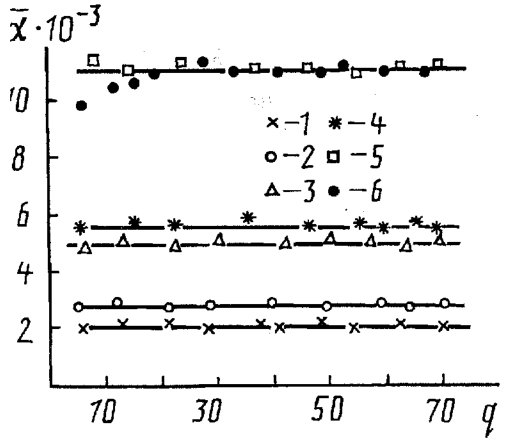
$$\mathrm{Nu} = 0{,}021 \cdot \mathrm{Re}^{0,8} \mathrm{Pr}^{0,43} \left(\frac{\mathrm{Pr}}{\mathrm{Pr}_c}\right)^{0,25} \quad (8)$$

The authors of the present paper have conducted a series of investigations from which the coefficients of local heat transfer were determined in the forced motion of the fluid through a rectangular smooth and transverse-finned channel with side ratios a:b = 36:2, 36:3, and 30:9 mm within the variation ranges Re = 600-2.5 $\times$ 10^4 and Pr = 4-7.

The experimental plant [14] is represented by a closed contour composed of a series-connected working element, circulating pump, regulating rectifier, heat exchanger-cooler, and two flow meters. Distilled water was used as the heat transfer agent. The test segment was heated by alternating current. Investigations were performed over the working segments, the total length of which comprised 0.6 m, and the length of the heated segment varied within the limits 0.12-0.35 m.

The working segment is the flat tube 1 (Fig. 2), the ends of which are fixed at the tips 2 permitting one to hermetically connect the working element to the hydraulic loop of the test stand. The measuring unit 3 is fixed at the central cross section of the element, at its external side, enabling one to measure the temperature at the external side of the working element at five points over the half-perimeter of the channel. Chromel-Copel thermocouples 4 made of 0.15 mm wide thermoelectrodes were used as the temperature gauge. Before installation, the thermocouples were calibrated using a thermostat and laboratory thermometers with 0.1°C scale value.

The measuring unit provided safe thermal contact between the thermocouple junctions and the slit channel wall, preserving, however, their reliable electrical insulation from each other. The external sides of the measuring unit and slit channel were thoroughly heat-insulated. The working element has two pressure takeoffs 5 located at its ends and intended for measuring hydraulic losses over the slit channel. In addition, the slit element is equipped with two current inputs 6 which can move along the channel length, changing thereby the length of the heated part of the



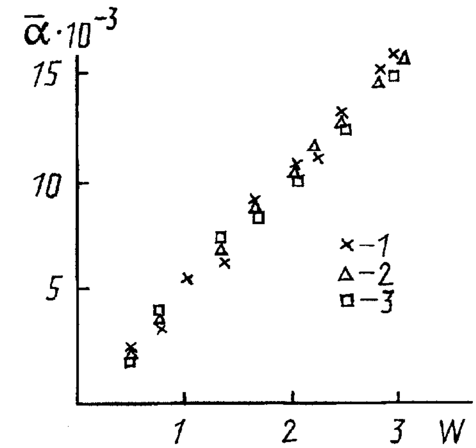


Fig. 3. The effect of heat flux density variation on heat transfer: 1) channel b = 2 mm, W = 0.5 m/sec; 2) 3 and 0.5; 3) 2 and 1; 4) 3 and 1; 5) 2 and 2; 6) 3 and 2. q, kW/m^2; $\overline{\alpha}$, $W/(m^2 \cdot K)$.

Fig. 4. The effect of the channel slope on the heat transfer: 1) $\varphi = 0$; 2) 45°; 3) 90°. W, m/sec.

working element. The construction of the bracing elements of the working segment enables one to change its orientation from the horizontal position to the vertical one.

The following were measured in the experiments: the local heat transfer coefficients α at the point of wall-temperature measurement, and the mean local coefficients $\overline{\alpha}$ at the cross section, where the temperatures of the wall were measure. The equivalent diameter was used as the characteristic dimension.

To investigate the effect of free convection on the heat transfer, a series of tests was performed, in which for three fixed liquid flow rates the density of the heat flux q varied in the range 6-70 kW/m^2. The results of these tests for various channels and fluid velocities are shown in Fig. 3.

In another series of tests, the problem was stated as that of defining the effect of the channel slope. The results for channels 2 and 3 mm wide and slopes 0, 45°, and 90° are presented in Fig. 4. As is seen from the figure, neither the change in the heat flux density nor the channel orientation in space exerts any influence on the heat transfer. This allows us to conclude that, in the range of values of q and W studied, forced convection has no effect on the heat transfer.

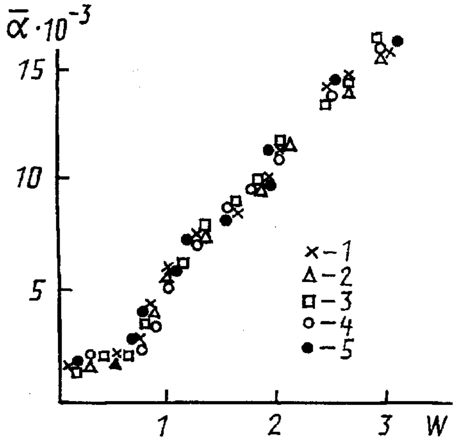
Figure 5 presents our results for heated segments of different length. It is obvious that in our tests the size of the heated segment has no influence on the heat transfer. In the same figure are shown data on the mean heat transfer in the case where the segment was turned over by 180°, when the temperatures were measured at the lower wall of the channel. We observed no difference in the results in this case.

The typical character of the α distribution over the height of the channels with width b = 2 and 3 mm as a function of the mean velocity of the liquid (water) at q = 70 kW/m^2 and at the measured cross section distance from the channel inlet x = 0.15 m is shown in Fig. 6. Analysis of the data shows that, for the channel with a:b = 36:2 for all W, the character of the α distribution over the channel width does not change, whereas, for the channel with a:b = 36:3, this character changes when W < 0.9 m/sec and W > 1.28 m/sec. At W = 0.9 m/sec the curve has a peak at the channel center, and then with an increase in the velocity W > 1.28 m/sec it acquires a saddle form.

In the channel with a:b = 30:9 mm, during the investigation a maximum fluid velocity W = 1.13 m/sec was obtained, the character of the α distribution over the channel width (b) being the same as for the channel with a:b = 36:2.

For a complete explanation of the heat transfer characteristics, a study of the velocity profile and hydraulic resistance in fluid motion through plane-oval slit channels is required.

For Reynolds numbers $Re > 4 \cdot 10^3$, the test data for all channels investigated lie on one line (curve 9 in Fig. 1). Within the range $Re = 4 \cdot 10^3 - 2.5 \cdot 10^4$, our data for smooth channels are correlated by the equation



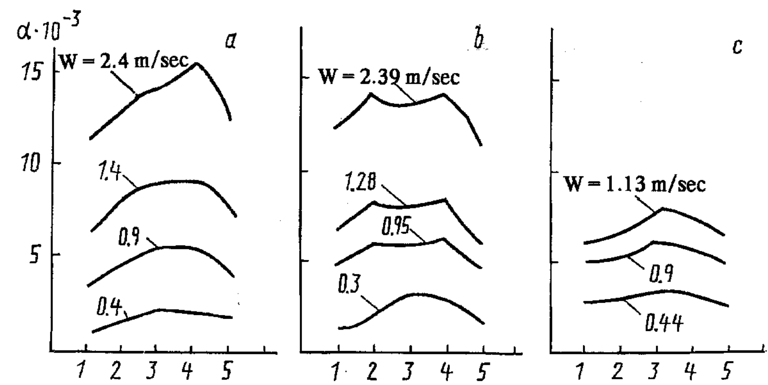


Fig. 5. The effect of the heated segment length on the heat transfer: 1) $x = 0.15$ m, $l = 0.35$ m; 2) 0.12 and 0.32; 3) 0.91 and 0.291; 4) 0.032 and 0.232; 5) 0.15 and 0.35, the channel is inverted by 180°.

Fig. 6. Typical distribution of local heat transfer coefficients α over the channel height depending on the mean velocity of the fluid: a) the channel with a:b = 36:2; b) a:b = 36:3 mm; c) a:b = 30:9 mm. $\alpha \cdot 10^{-3}$, W/(m^2·K), is the local heat transfer coefficient.

$$\mathrm{Nu} = 0{,}0066\,\mathrm{Re}^{0,937}\mathrm{Pr}^{0,43}\left(\frac{\mathrm{Pr}}{\mathrm{Pr_c}}\right)^{0,25}. \tag{9}$$

In [9], a fully developed turbulent flow was experimentally studied in rectangular cross section channels with one rough wall of smaller height. Mean velocities and turbulent stresses were measured in detail with the help of a thermoanemometer. Although the main attention was focused on the effect of roughness which manifests itself in the form of large secondary vortices at the angles between the rough and smooth walls, the velocity profiles presented in the figures allow one to speak of the absence of similarity in the fluid flow in a rectangular channel and a round tube, which, in its turn, calls into question the use of the equivalent diameter for predicting the hydraulic resistance and heat transfer in a flat channel.

In [10], it was convincingly shown that the discrepancy between the experimentally obtained resistance factors for round and noncircular channels is specified by closely related effects of transition and the choice of a linear dimension. The problem of the choice of linear dimension in the process of determination of the resistance factor and the Reynolds number as applied to the flow in round tubes is solved fairly clearly. For noncircular cross section channels, the problem of the choice of the proper linear dimension arises. As a rule, a hydraulic diameter is employed, although such a choice is widely believed to be incorrect. This opinion (not necessarily true) is based on the following arguments [10]: the Reynolds number represents a similarity parameter; therefore, if the flow through two channels is characterized by one and the same Reynolds number, they are dynamically similar and, accordingly, have almost identical resistance factors. This statement is valid for smooth round tubes. Successful application of the similarity criterion for round tubes and the failure in the attempts to use it for correlating experimental data for noncircular channels has led to the assumption [10] that in these trials significant peculiarities of the fluid flow through channels were not taken into account. In studying flows through noncircular cross section channels, the problem of the choice of a linear dimension is critical.

Hence, in [10] a new method is proposed - the so-called method of critical resistance factor. It is based on the assumption that the deciding factor is the transition from the laminar to the turbulent flow regime and also the equality of the critical values of the resistance factor f and the Reynolds number Re for round and noncircular channels. By introducing the scale coefficients

$$\Psi_R = \frac{\mathrm{Re}_{c,c}}{\mathrm{Re}_{n,c}}, \quad \Psi_f = \frac{f_{c,c}}{f_{n,c}}$$

the equations of modified or reduced Reynolds numbers $\overline{Re}$ and the resistance factor $\bar{f}$ for isothermal conditions

$$\overline{Re} = \Psi_R Re_n, \ \bar{f} = \Psi_f f_n.$$

have been obtained. Here, $Re_{c,c}$ and $f_{c,c}$ are the critical Reynolds number and critical resistance factor for a circular channel, and $Re_{n,c}$ and $f_{n,c}$ are the same for a noncircular channel. From here, it is obvious that $\bar{f}$ and $\overline{Re}$ do not depend on the characteristic length for a noncircular channel. Thus, emphasizing the fact that the validity of the application of the hydraulic diameter has been debated problem for a long time, Obot [10] suggests to "bypass" it, using the method of critical resistance factor.

So, considering in Fig. 1 the data obtained by various authors, one can notice certain qualitative and quantative disagreements. According to the data of [6], at Re = 20,000, the heat transfer exceeds by 10 times the heat transfer calculated using the data of [5]. Moreover, there is no unanimity in the choice of the characteristic linear dimension. We have recalculated the data of [6, 7, 14] using the slit width as the deciding parameter; however this did not result in a qualitative change of the plot presented in Fig. 1. Proceeding from the fact that such a significant divergence in experimental data can hardly be due to the incorrectness of the tests of all the authors mentioned, it is natural to assume certain peculiarities of the hydrodynamics in slit channels which were not taken into account. Thus, a detailed study of the hydrodynamics in different tubes is of paramount importance.

NOTATION

Nu, Nusselt number; Pr, Prandtl number; Re, Reynolds number; Gr, Grashof number; St, Stanton number; μ, coefficient of dynamic viscosity, Pa·sec; l, channel length; d, equivalent diameter; a, b, height and width of the channel; K_l, correction coefficient accounting for the effect of the initial segment of the channel; γ, side ratio of the channel; α, heat transfer coefficient, $W/(m^2 \cdot K)$; W, fluid velocity, m/sec. Subscripts: c, parameters determined based on the wall temperature.

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