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TURBULENT MIXED CONVECTION IN CHANNELS FOR DIFFERENT ORIENTATIONS OF THEM IN SPACE

P. Poškas

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Investigations of heat transfer and turbulent transfer in mixed convection in tubes and plane channels for different orientations of them in space are reviewed.

1. Introduction. The interest in investigations of turbulent transfer and heat transfer under the combined action of forced and natural convections is due to the manifestation of the latter both in nature and in various technical devices. Study of this problem is particularly pressing as applied to technical devices that operate at increased heat loads or at increased gas pressure. In the technical devices, flow of a heat transfer agent very often occurs not just in vertical channels of different flow area, where the processes of heat transfer in mixed convection have been investigated in greatest detail. Therefore in the present review, heat transfer and turbulent transfer in channels for different orientations of them in space are analyzed.

2. Stabilized Mixed Convection in Vertical Tubes. Different aspects of heat transfer and turbulent transfer far from the initial section of heating in vertical tubes are dealt with in detail in the monograph [1] and the review article [2]. In vertical tubes, we can distinguish two characteristic cases of flow: 1) the directions of forced flow and the vector of thermogravitational forces are opposite (the upward flow of a cooled liquid or the downward flow of a heated liquid); 2) the directions of forced flow and the vector of thermogravitational forces coincide (the upward flow of a heated liquid or the downward flow of a cooled liquid).

In the first case, the liquid velocity decreases near the wall and increases in the flow core as compared to the forced flow. However the enhanced turbulent transfer causes the enhancement of heat transfer as compared to the case of forced convection (Fig. 1, curve 1). To calculate heat transfer under these conditions, there are fairly reliable recommendations.

Thus, in [1], the dependence

$$\mathrm{Nu} = \mathrm{Nu}_{\mathrm{turb}} \exp\left[-0.02\mathrm{Bo}_{\mathrm{lim}}\right] + \mathrm{Nu}_{\pi}\left[1 - \exp\left(-0.04\mathrm{Bo}_{\mathrm{lim}}\right)\right], \tag{1}$$

is given, where $\mathrm{Nu}_{\mathrm{turb}}$ is the heat transfer in forced convection; $\mathrm{Nu}_{\pi} = 0.5\sqrt{\mathrm{Pr}}(\mathrm{Gr}_q^{1/4}(1 + \sqrt{\mathrm{Pr}})^{-1}$ and characterizes heat transfer in the regime of thermal turbulence; $\mathrm{Bo}_{\mathrm{lim}} = \mathrm{Gr}_q/\mathrm{Gr}_{q,\mathrm{lim}}$ is the thermogravitation parameter; $\mathrm{Gr}_{q,\mathrm{lim}} = 9\cdot10^{-5}\mathrm{Pr}^{1.15}\mathrm{Re}^{2.75}$ is the limiting Grashof number, which defines the boundary for the onset of the influence of thermogravitation as a 1% deviation of the Nu number from $\mathrm{Nu}_{\mathrm{turb}}$.

Formula (1) describes experimental data obtained for $0.7 \le \mathrm{Pr} \le 10$, $3\cdot10^3 \le \mathrm{Re} \le 10^5$, $\mathrm{Gr}_q \le 10^{11}$, and $x/d \ge 30$ under the conditions $q_w = \mathrm{const}$.

In [2], the simpler dependence

$$\frac{\mathrm{Nu}}{\mathrm{Nu}_{\mathrm{turb}}} = \left[\left|1 + 8.3\cdot10^4\,\mathrm{Bo}\left(\frac{\mathrm{Nu}}{\mathrm{Nu}_{\mathrm{turb}}}\right)^{-2}\right|\right], \tag{2}$$

is proposed, where $\mathrm{Bo} = \mathrm{Gr}_q/\mathrm{Re}^{3.425}\cdot\mathrm{Pr}^{0.5}$. As follows from expressions (1) and (2), the exponent m of Re differs rather widely in the characteristic thermogravitation parameters Bo and $\mathrm{Bo}_{\mathrm{lim}}$ (m = 2.75 and 3.425). However, as

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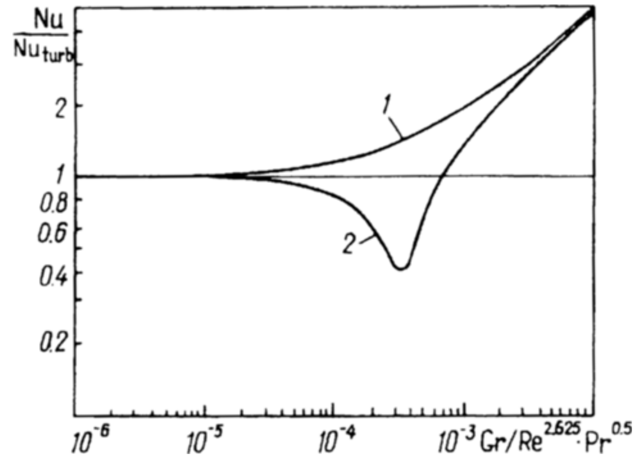


Fig. 1. Schematic representation of the relative heat transfer in turbulent convection under conditions of opposite (1) and coincident (2) directions of forced and natural convections [2].

shown in [3], the experimental data for water can be generalized successfully by using various characteristic thermogravitation parameters (m = 2.7–3.636).

When the directions of forced flow and the vector of thermogravitational forces coincide, the flow velocity increases near the wall and decreases at the center of the tube (the M-profile of the velocity) as compared to the case of forced flow. In turbulent flow, heat transfer first decreases under the growing influence of thermogravitational forces and, for large thermogravitation parameters, it starts to increase as compared to the case of forced convection (Fig. 1, curve 2) since not only the profile of the averaged velocity changes but also the turbulent transfer. To calculate heat transfer under these conditions, a number of generalizing dependences are proposed. In [4], the relation

$$\frac{\mathrm{Nu}}{\mathrm{Nu}_{\mathrm{turb}}} = 8.84\mathrm{Bo}_A^{0.263}, \tag{3}$$

generalizes data obtained in flows of air, water, and mercury in the region of the strong influence of natural convection, when, after decreasing, the heat transfer begins to be restored (Fig. 1). Here $\mathrm{Bo}_A = \mathrm{Gr}_q/4\mathrm{Re}^3\mathrm{Pr}$ is the thermogravitation parameter. In [5], a similar dependence is used for calculating heat transfer in an air flow for different heat loads:

$$\frac{\mathrm{Nu}}{\mathrm{Nu}_{\mathrm{turb}}} = 17\mathrm{Bo}_A^{0.27}(q^+)^{0.1}, \tag{4}$$

where $q^+ = q_w/h\bar{u}q$ is the parameter of the heat flux.

Rather simple relations are proposed in [6, 7] in generalizing data obtained in a helium flow in the near-critical region; however their accuracy is low: the spread in the experimental data attains 25–30%. Here, too, Bo_A is used as the characteristic thermogravitation parameter. A unified dependence for determining heat transfer in the range from the regime of forced flow to the regime of natural convection that is represented as St = f(Re, Pr, E) ($E = \mathrm{Gr}_q/\mathrm{Re}^4\mathrm{Pr}$) for Pr = 0.6–8, Re = 300 $-5\cdot 10^4$, and $\mathrm{G}_q = 0-10^{11}$ is given in [1]. However when calculating heat transfer it is essential to know the coefficient of resistance ξ under conditions of mixed convection. Unfortunately, the proposed formulas for determining ξ under these conditions are not sufficiently verified by experiment. In [2], for coincident directions of the convections, the data are noted to be satisfactorily described by dependence (2) if a minus sign is substituted for the plus sign.

As follows from the above review, here, too, different researchers use different thermogravitation parameters in which the exponent of Re varies from m = 3 [4-7] to m = 4 [1].

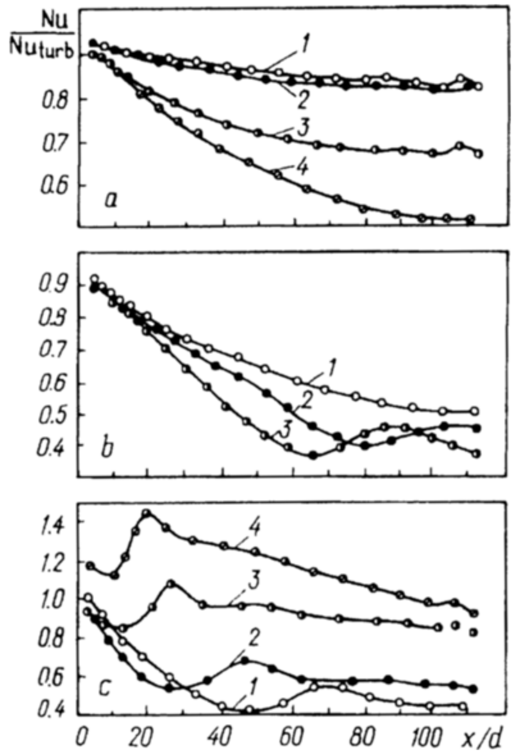


Fig. 2. Change in the relative heat transfer over the tube length for coincident directions of forced and natural convections [5]. Inlet parameters: a: 1) $\mathrm{Re_{in}} = 2.42 \cdot 10^4$, $(\mathrm{Bo}_A)_{\mathrm{in}} = 1.38 \cdot 10^{-5}$; 2) $1.94 \cdot 10^4$ and $2.21 \cdot 10^{-5}$; 3) $1.61 \cdot 10^4$ and $3.14 \cdot 10^{-5}$; 4) $1.45 \cdot 10^4$ and $3.58 \cdot 10^{-5}$; b: 1) $1.45 \cdot 10^4$ and $3.58 \cdot 10^{-5}$; 2) $1.33 \cdot 10^4$ and $3.63 \cdot 10^{-5}$; 3) $1.24 \cdot 10^4$ and $3.79 \cdot 10^{-5}$; c: 1) $1.14 \cdot 10^4$ and $4.85 \cdot 10^{-5}$ 2) $0.885 \cdot 10^4$ and $9.47 \cdot 10^{-5}$; 3) $0.5 \cdot 10^4$ and $36.8 \cdot 10^{-5}$; 4) $0.37 \cdot 10^4$ and $77 \cdot 10^{-5}$. $\mathrm{Nu_{turb}}$ is the heat transfer in forced convection.

In [8], in investigating heat transfer for different air pressures, it is shown that the data cannot be generalized by using one thermogravitation parameter since the thermogravitation parameter Bo is characteristic in the zone of a decrease in the heat transfer whereas Bo_A is characteristic in its restoration zone.

3. Development of Heat Transfer and Turbulent Transfer over the Length of a Vertical Tube in Mixed Convection. Whereas for opposite directions of the convections, heat transfer stabilizes rapidly over the tube length, for coincident directions of the forced and natural convections characteristic maxima of the wall temperature are noted in certain regimes. In [9], it was shown for the first time that in the ascending motion of water under conditions of near-critical parameters there is a significant local increase in the wall temperature in vertical heated tubes. In the downward flow of water, these phenomena were not observed, which made it possible to establish the fact of the influence of free convection on the deterioration of the heat transfer in an ascending flow. In more recent works, similar maxima of the wall temperature were obtained in water flows at $p = 5$ bar [10] and at atmospheric pressure [6].

In gaseous flows, maxima of the wall temperature are revealed at supercritical pressure [11]. Up to now, one failed to notice these maxima when the pressure of the gases is small or atmospheric since short tubes ($x/d < 65$) were used in the experiments and the thermogravitation parameter varied within insufficiently wide limits [12-16]. Recent investigations in the Lithuanian Power Institute showed that in gaseous flows at moderate gas pressures ($p = 7$ bar), too, characteristic maxima of the wall temperature can appear in certain regimes in sufficiently long tubes.

When the influence of thermogravitational forces is weak, there is a monotonic decrease in the heat transfer over the tube length (Fig. 2a, curves 1 and 2). At the beginning of the heated section, the heat transfer corresponds

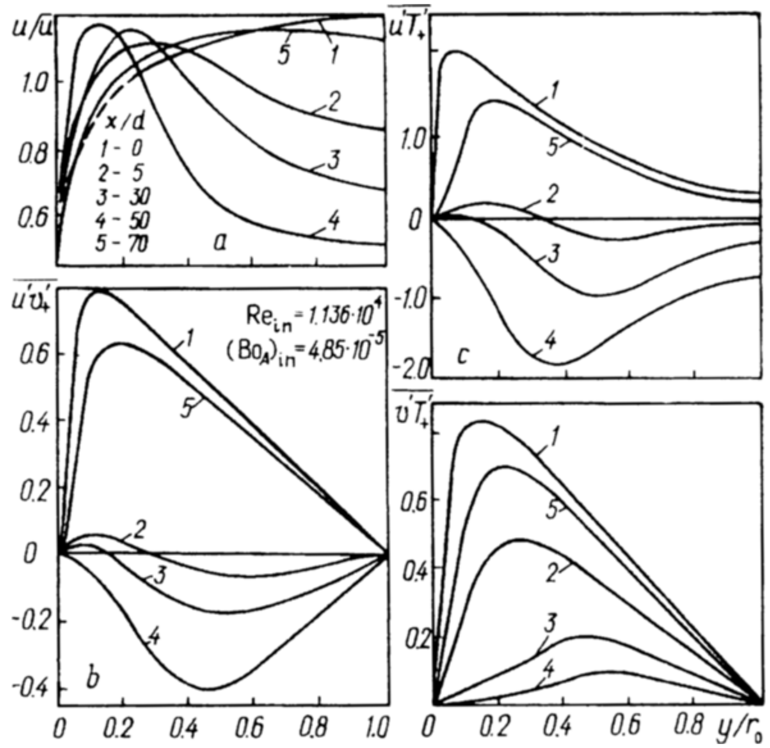


Fig. 3. Development of the profiles of velocity (a), turbulent tangential stresses (b), and turbulent axial (c) and radial (d) heat fluxes in the case of nonmonotonic development of heat transfer [18].

to that in forced convection, and it is minimum at the end of the tube. As the action of thermogravitational forces increases, the heat transfer begins to change more sharply but in this case, too, its minimum corresponds to the end of the tube (Fig. 2a, curves 3 and 4; Fig. 2b, curve 1). Next, as these forces grow, the heat transfer decreases gradually on the initial thermal section and, for $x/d \approx 80$, a local minimum is noted behind which there is enhancement of it again (Fig. 2b, curve 2). It is not only the minimum but also a pronounced local maximum of the heat transfer that appears with a further insignificant increase in the thermogravitation parameter (Fig. 2b, curve 3). As the action of thermogravitational forces increases further, the minima (maxima) of the heat transfer begin to shift toward the inlet section, and the total level of the heat transfer increases (Fig. 2c, curves 2-4).

The change in the local heat transfer over the tube length reflects the complex hydrodynamic processes that occur under these conditions. Most of the experimental investigations of turbulent characteristics of a flow are performed in one cross section far from the initial section of heating. These results are reviewed and analyzed in [1, 2].

The cases of measuring the characteristics of the flow over the tube length are extremely few [13, 15, 17]. They show that, with distance from the initial section of heating, the velocity profile becomes M-shaped. Strong acceleration of the flow is observed near the wall, especially before the first minimum of the heat transfer [17]. Velocity and temperature pulsations decrease and then are restored [17]. More detailed information on the change in the turbulent characteristics of the flow is obtained in numerical investigations [17]. Turbulent transfer becomes strongly deformed because of deformation of the velocity profile (Fig. 3b-d). With increasing x/d the Reynolds stresses $\overline{u'v'_+}$ and the axial turbulent heat flux $\overline{u'T'}$ become negative on an ever increasing part of the tube cross section until the position of the first minimum of the heat transfer is attained ($x/d \approx 40-50$). The radial turbulent heat flux also decreases with increasing x/d but remains positive. As x/d increases further, the turbulent heat transfer is restored (Fig. 3b-d, curves 5) and the heat transfer is enhanced. As calculations [18] show, this results from the heightened importance of the generation of turbulence due to the thermogravitational forces.

This significant change in the character of the heat transfer over the tube length affects, of course, the position of the heat transfer minimum (Fig. 1, curve 1), which was called the critical value of the thermogravitation parameter (Bo_{cr}) in [8]. Whereas, for a large distance from the initial section of heating ($x/d > 50$), $Bo_{cr} \approx 2 \cdot 10^{-6}$, with a decrease in it Bo_{cr} increases distinctly (Fig. 4). The relative heat transfer is also enhanced at this point. Consequently, in short tubes and for a sufficiently strong influence of thermogravitational forces, the superheating hazard for the wall is substantially lower than in long tubes since in them the degree of the decrease in the heat transfer is much lower at Bo_{cr}, too. In [8], for calculating Bo_{cr} and $(Nu/Nu_{turb})_{cr}$ in the region x/d = 0–50, the dependences

$$Bo_{cr} = 1.8 \cdot 10^{-6} + 1.5e^{-(11.7+0.06x/d)}, \tag{5}$$

$$(Nu/Nu_{turb})_{cr} = 2.16\,(x/d)^{-0.4}, \tag{6}$$

are proposed, and for $x/d \geq 50$, we can take $Bo_{cr} \approx 2 \cdot 10^{-6}$ and $Nu/(Nu_{turb})_{cr} \approx 0.42$.

As was noted, to calculate stabilized heat transfer in mixed convection, a number of generalizing dependences have been proposed. Equations for calculating heat transfer over the entire tube length are obtained only in [19] for water and in [5, 20] for air.

It is often said that the conditions for mixed convection to manifest itself are created only for small Reynolds numbers of a forced flow. In [21], these conditions are shown to be entirely possible in purely natural convection, too (or in circulation of a flow because of natural convection). Depending on the magnitude of the resistance to the flow at the inlet or outlet of the tube and on the thermogravitation parameter Bo (or Bo_A), the relative heat transfer can change in natural convection in the same manner as in the case of coincident directions of forced and natural convections (Fig. 1, curve 2).

4. Turbulent Mixed Convection in Vertical Plane Channels. In systematic investigations of heat transfer in mixed turbulent convection, vertical tubes have been primarily used. Heat transfer in channels of a noncircular cross section is studied to a much lesser degree. In [22-25], investigations were performed in plane vertical channels. In [22], experiments were performed at atmospheric pressure of air in a channel of small dimensions (a 0.21 × 0.02 m cross section) and therefore the influence of thermogravitational forces was small. It was obtained that, for coincident directions of the convections, the heat transfer decreases as compared to that in forced convection, and it is enhanced when the directions are opposite. The friction resistance changes inversely: it decreases for opposite directions of the convections and is enhanced for coincident directions. The deformation of the measured velocity and temperature profiles as well as of the longitudinal velocity and temperature pulsations is small since the experiments were performed with a weak influence of thermogravitational forces.

The heat transfer in channels with clearances of 2, 5, 6, 18, and 50 mm in a water flow was investigated in [24]. As the authors point out, the channel length was insufficient for attaining fully developed flow. Under these conditions, the heat transfer is enhanced as compared to that in forced convection for both opposite and coincident directions of the convections.

Heat transfer in plane transformer channels cooled by an air flow was studied in [23]. The velocity profiles at the inlet to the heat-transfer zone were flat rather than developed, as in the majority of the previous investigations. It is found that the heat transfer is enhanced for opposite directions of the convections and it first decreases and then is enhanced as compared to that in forced convection when the directions coincide. The decrease in the heat transfer for coincident directions of the convections is observed in a wider region of the thermogravitation parameter than was the case in tubes.

In [21], it was shown that for two-sided symmetric heating in a plane channel with opposite directions of the convections the effect of thermogravitational forces is the same as in a tube. The heat transfer under these conditions was calculated using the relation

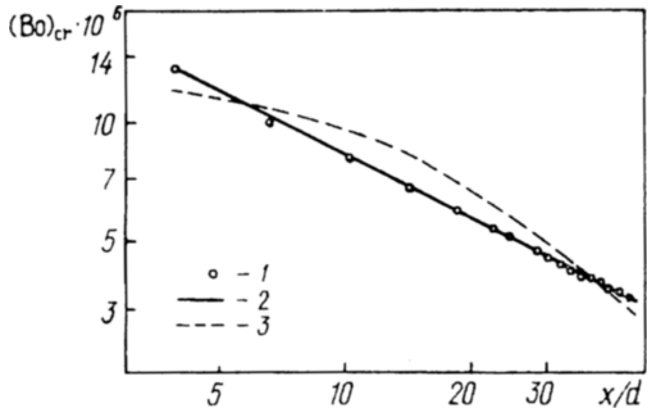


Fig. 4. Bo_{cr} vs the dimensionless distance from the initial section of heating x/d: 1) plane channel [27]; 2) tube [8].

$$Nu = 0.0011\, Re^{0.8}\, Pr^{0.5} \left\{ 1 + \left[1 - \frac{696}{Re^{0.8}} + 8300 \frac{Gr_{\Delta t}}{Re^{2.6}\,(Pr^{0.5} + 1)} \right]^{-0.39} \right\}, \tag{7}$$

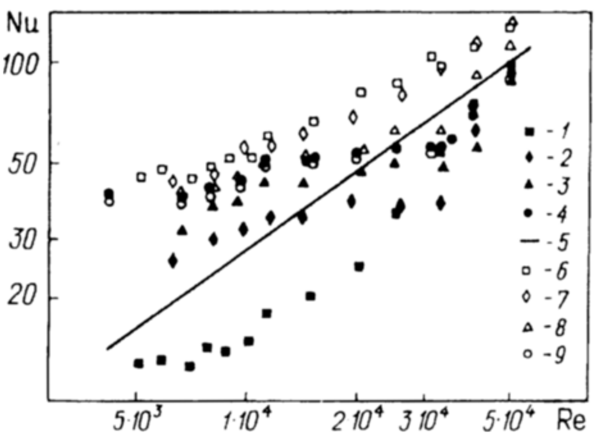
which describes the results of works of various researchers for air, water, and Freon-113 with an error of no more than 7% in the region $Re = 10^4 - 2\cdot 10^4$, $Pr = 0.7-7$, and $Gr_{\Delta t} = 10^6 - 2\cdot 10^9$. The most informative data on the local heat transfer in the case of coincident directions of forced and natural convections in a plane channel with two-sided symmetric heating were obtained in [26, 27]. Here it was shown that just as in tubes characteristic maxima and minima of the wall temperature appear in certain regimes. As Fig. 4 shows, the change in the critical thermogravitation parameter Bo_{cr} as a function of x/d in a plane channel is the same as in a tube. For calculating Bo_{cr}, the generalizing dependence

$$Bo_{cr} = 1.54\,(x/d)^{-0.38}\,. \tag{8}$$

is proposed. The characteristic thermogravitation parameter in the zone of a decrease in the heat transfer (Fig. 1, curve 2) for plane channels is, just as in tubes, is $Bo = Gr_q/Re^{3.425}Pr^{0.5}$. In the zone of restoration of the heat transfer, the exponent of Re in the characteristic thermogravitation parameter for a plane channel is found to be somewhat smaller ($Bo_2 = Gr_q/Re^{2.5}Pr^{0.5}$) [27] than for a tube ($Bo_A = Gr_q/Re^3Pr$) [8, 20] but this is due to the wider region of thermogravitation parameters in the case of a plane channel, for which the experimental data were obtained.

5. Turbulent Mixed Convection in Horizontal and Sloping Channels. A large complex of investigations of local heat transfer and turbulent transfer in plane channels and tubes was also performed at the Institute of High Temperatures of the former Academy of Sciences of the USSR [1]. Therefore we will not analyze in detail the processes that are characteristic of these conditions. We just note that it is laminarization of the flow and a decrease in heat transfer that are characteristic of the conditions of stable stratification of density. Unstable stratification of density, on the other hand, is characterized by turbulence of the flow and enhancement of heat transfer as compared to the case of forced convection. In [1], it is shown that for generalizing data on the heat transfer to air in both stable and unstable stratification of density, we can use the same thermogravitation parameter Gr_q/Gr_{q_0}, where $Gr_{q_0} = 1.2\cdot 10^{-4} Pr^{1.12} Re^{2.75}$ defines the bound for the onset of the influence of thermogravitation on turbulent flow and heat transfer as a 1% deviation of Nu from the Nu_{turb} characteristic of neutral stratification (forced convection). However investigations performed at various air pressures [28] showed that, in generalizing data on heat transfer, different thermogravitation parameters should be used for stable stratification of density and the unstable one.

Investigations of mixed turbulent convection in sloping channels are very few. We know only two works in which experimental data on the local heat transfer in a sloping plane channel [29] and the average heat transfer in a sloping annular channel with an internal heated tube [30] are presented.



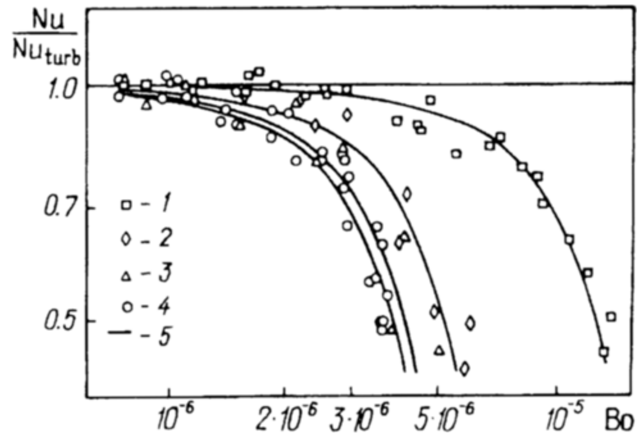


Fig. 5. Heat transfer vs Re in a plane channel on the upper (light symbols) and lower (dark symbols) walls at an air pressure $p = 0.7$ MPa and $x/d = 38$ for different slopes [29]: 1 and 6) 0°, 2 and 7) 30°, 3 and 8) 60°, 4 and 9) 90°, 5) forced convection.

Fig. 6. Relative heat transfer from the upper wall of a plane channel in the region of a decrease in heat transfer for $x/d = 38$ and different slopes [29]: 1) 0°, 2) 30°, 3) 60°, 4) 90°, 5) by the dependence of [29].

As Fig. 5 shows, the heat transfer of the upper wall of a plane channel depends strongly on its slope whereas, for the lower wall, this dependence is much weaker. But on both walls there is a region in which heat transfer is lower than in forced convection. This region is particularly wide (in the Re number) for the upper wall, i.e., for conditions of stable stratification of density ($\varphi = 0^\circ$). However not for all Re is the heat transfer under conditions of stable stratification of density minimum. All this depends on the magnitude of the thermogravitation parameter. When the position of the channel is vertical ($\varphi = 90^\circ$), the heat transfer is most sensitive to manifestation of the action of thermogravitational forces (Fig. 6); in this case (for $x/d = 38$), $Bo_{cr} = 4 \cdot 10^{-6}$. For $\varphi = 60^\circ$, $Bo_{cr} = 4.3 \cdot 10^{-6}$, for $\varphi = 30^\circ$, it is equal to $5.5 \cdot 10^{-6}$, and for $\varphi = 0^\circ$ (stable stratification of density) it is only to $1.3 \cdot 10^{-5}$. Despite different critical thermogravitation parameters Bo_{cr}, the maximum decrease in the heat transfer as compared to forced convection is practically the same in all cases (Fig. 6), and it is only the abruptness of the restoration of the heat transfer with a further increase in the thermogravitation parameter that is different [29]. This abruptness, which is due to the generation of turbulence because of the action of thermogravitational forces, decreases noticeably as the slope of the channel decreases (as conditions of stable stratification of density are approached).

The heat transfer of the lower wall increases as the action of thermogravitational forces increases, and it changes very slightly in the region $\varphi = 0-60^\circ$ [29]. In the vertical position, we observe laminarization of the flow and, as a consequence, a decrease in the heat transfer with its subsequent restoration. To determine the heat transfer in the region $\varphi = 60-90^\circ$, further investigations are needed since a decrease in the heat transfer in the channel is to be expected not only in its vertical position but also when the latter is approached. As shown in [30], with a deviation from the vertical position the average heat transfer in an annular channel in turbulent mixed convection in the region $5000 < Re < 20{,}000$ decreases for both opposite and coincident directions of the convections, and it becomes minimum for its horizontal position. Data for the plane channel (Fig. 5) demonstrate the same trend in the region $Re \approx 5000$ since the levels of the heat transfer in the vertical channel and from the lower wall of the horizontal channel are similar whereas the heat transfer from the upper wall of the horizontal channel is much smaller. With increasing Re there is a change in the character of the change in the average heat transfer in the plane channel as its slope changes and, in the region of the most pronounced laminarization of the flow in the vertical channel ($Re = (3-4) \cdot 10^4$), the largest heat transfer will occur in the horizontal channel.

CONCLUSIONS

1. Investigations performed at different pressures of the air showed that, in generalizing data on heat transfer, use should be made of different characteristic thermogravitation parameters for the regions of its decrease and restoration (or intensification).

2. The maximum decrease in heat transfer and the critical thermogravitation parameter depend on x/d provided that $x/d < 50$. For larger values of x/d, they change insignificantly and $Bo_{cr} = (Gr_q/Re^{3.425}Pr)_{cr} \approx 2 \cdot 10^6$.

3. The regularities of laminarization of the flow and the change in heat transfer in mixed convection are similar for forced flow of a heat-transfer agent for small Reynolds numbers and in the case of natural convection, provided that the thermogravitation parameters are the same.

4. The maximum decrease in the heat transfer on the upper wall of a plane channel because of flow laminarization under the action of thermogravitational forces is the same for all slopes (from the vertical slope to the horizontal one) but the critical thermogravitational parameters Bo_{cr} are different; the smallest Bo_{cr} occur for a vertical channel and the largest occur are for a horizontal channel.

5. On the lower wall of a plane channel, the heat transfer changes little in the region $\varphi = 0-60^{\circ}$ ($\varphi = 0^{\circ}$ is the horizontal position). As the vertical position is approached a decrease in the heat transfer is observed.

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