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# Local turbulent opposing mixed convection heat transfer in inclined flat channel for stably stratified airflow

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**Abstract**

Local turbulent mixed convection heat transfer in inclined flat channels ($\varphi = 20$–$90°$ from horizontal position) for opposing flows was investigated for the case when only upper wall is heated (under stably stratified flow conditions). Wide ranges of airflow parameters are covered: $Re = 4 \times 10^3$–$4 \times 10^4$, $Gr_q = 1.7 \times 10^8$–$1.4 \times 10^{10}$, pressures; $p = 0.2$; 0.4; 0.6; 0.8 MPa. Based on analysis of local heat transfer data and existing information in the literature three characteristic regions in the buoyancy parameter range investigated were identified: region without buoyancy instabilities, transition region and region with buoyancy instabilities in whole heated section. For the region without buoyancy instabilities correlation for calculation of heat transfer in inclined flat channels was suggested.


## 1. Introduction

Mixed convection heat transfer exists provided that forced flow velocities and natural convection currents are of the same order of magnitude. In internal vertical flows the buoyancy forces may be in the same direction as the forced flows (‘‘aiding flows’’), or directed opposite to the forced flow (‘‘opposing flows’’). Air stratification or distribution of the temperature of air over the height, which determines equilibrium conditions in the air that are favourable or not favourable to the development of the vertical displacements of air. There are two types of air stratification in horizontal channels: stable air stratification (upper wall heated), with which the vertical motion decrease; the unstable air stratification (bottom wall heated), which supports or strengthens the ascending movements of air and serves as the necessary condition for the development of the convection. We are also applying this terminology for inclined channels.

Internal mixed convection heat transfer in different channels can be applied to nuclear power technology, chemical process heat transfer, some aspects of electronic cooling, etc. Significant heat transfer enhancement may be realized by mixed convection in some situations, especially in opposing flows.

Numerous investigations have been performed for understanding of the effect of natural convection on forced convection heat transfer in vertical tubes. There have been experimental and numerical studies investigating heat transfer coefficients, velocity and temperature profiles, velocity and temperature pulsations and their correlations. An interesting fact disclosed is that for turbulent aiding flows the effectiveness of heat transfer can be seriously impaired as a result of buoyancy forces modifying the production of turbulence and laminarising the flow. However, if higher buoyancy parameters are applied, heat transfer recovers and becomes even higher than forced convection heat transfer. For opposing flows turbulent transport and heat transfer are increasing when the effect of buoyancy increases.

Summaries on mixed convection have been published by Metais and Eckert [1], Petukhov and Poliakov [2], Jackson et al. [3], Vilemas and Poškas [4] etc. Number of correlations has been proposed to describe mixed convection heat transfer in vertical tubes. First of all correlations for average heat transfer have been proposed, for example, by Charchil [5], Petukhov [6], Swanson and

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**Nomenclature**

| | |
|---|---|
| $Bo$ | buoyancy parameter, $Bo = Gr_q/(Re^{3.425} Pr^{0.8})$ |
| $Bo_1$ | buoyancy parameter, $Bo_1 = Gr_q/(Re^{2.75} Pr^{1.12})$ |
| $Bo_2$ | buoyancy parameter, $Bo_2 = Gr_q/(Re^{2.5} Pr)$ |
| $b$ | channel width [m] |
| $c_p$ | specific heat [J/(kg K)] |
| $d_e$ | equivalent diameter of the channel, $d_e = 2h$ [m] |
| $Gr_q$ | Grashof number defined by the heat flux specified on the surface, $Gr_q = g\beta d_e^4 q_w/v^2\lambda$ |
| $g$ | acceleration due to gravity [m/s$^2$] |
| $h$ | channel height [m] |
| $Nu$ | Nusselt number, $Nu = \alpha d_e/\lambda$ |
| $p$ | pressure [Pa] |
| $Pr$ | Prandtl number, $Pr = \mu c_p/\lambda$ |
| $q$ | heat flux density [W/m$^2$] |
| $Re$ | Reynolds number, $Re = u_f d_e/v$ |
| $T$ | temperature [K] |
| $u$ | flow velocity [m/s] |
| $x$ | distance from the heating origin [m] |

*Greek symbols*

| | |
|---|---|
| $\alpha$ | heat transfer coefficient, $\alpha = q_w/(T_w - T_f)$ [W/(m$^2$ K)] |
| $\beta$ | volumetric expansion coefficient [K$^{-1}$] |
| $\lambda$ | thermal conductivity [W/(m K)] |
| $v$ | kinematics viscosity [m$^2$/s] |
| $\mu$ | dynamic viscosity [Pa s] |
| $\varphi$ | angle [°] |

*Subscripts*

| | |
|---|---|
| f | in the flow |
| T | forced turbulent convection |
| w | at the wall |
| in | at the inlet |

Catton [7] (for vertical ducts) etc. Later on, when investigations of local heat transfer coefficients have been performed correlations for stabilized heat transfer have been proposed, for example, Petukhov and Poliakov [2], Jackson et al. [3], Poškas and Poškas [8] (for vertical ducts) etc.

In Vilemas et al. [9] correlation for calculation of local heat transfer coefficients along the tube for aiding air flow for different temperature factors was proposed based on authors data.

In Poškas et al. [10] it was clearly demonstrated that for aiding flows the position of the minimum heat transfer, authors called it critical value of buoyancy parameter $Bo_{cr}$, and the level of heat transfer at this point $(Nu/Nu_T)_{cr}$ are both closely related to $x/d_e$ in the thermal entrance region. And this gave the explanation why there is such big difference on heat transfer data from different investigations. Correlation for calculation of $Bo_{cr}$, minimum heat transfer, local heat transfer in the region of the decreasing heat transfer ($Bo < Bo_{cr}$) within $x/d_e \leqslant 50$ have been proposed. For $x/d_e > 50$ these parameters are not more $x/d_e$ dependent. It was also demonstrated that in the region of recovering heat transfer ($Bo > Bo_{cr}$) relative heat transfer in airflow is actually independent of $x/d_e$ and heat load.

Correlation for average heat transfer in vertical tubes for aiding flows taking into account effect of the length of the tube have been proposed in Aicher and Martin [11] based on authors data and some data available in the literature. For opposing flows the new correlation for heat transfer calculations was also proposed but it was indicated that in this case it seems no influence of the $L/d$ ratio on the heat transfer rate. In Celata et al. [12] the new correlation for calculation of average heat transfer for aiding water flows in the vertical tubes was proposed based on authors data. This correlation takes into account dependence of heat transfer rate on the relative tube length $L/d$.

Investigations of turbulent mixed convection in inclined flat channels are very limited. Investigations of the local heat transfer for different inclination angles of the flat channels for aiding flows have been performed at Lithuanian Energy Institute [13]. Some preliminary data on heat transfer for opposing flows have been presented in Poškas and Poškas [14].

The aim of this paper is to present experimental results on local heat transfer in the inclined flat channels for stably stratified flow (only upper wall heated) in case of opposing turbulent mixed convection, with special emphasis on the analysis of the possible flow modes based on local heat transfer data and existing information in the literature. Finally, correlation is presented which describes the experimental data for different inclination angles of the channel in the region where buoyancy instabilities still not occur.

## 2. Experimental rig

Tests were performed on the rig built for the mixed convection investigations at the Laboratory of Nuclear Engineering, Lithuanian energy institute. Fig. 1 presents a scheme of the aerodynamic setup. Atmospheric air, reduced under a pressure of up to 0.2–0.8 MPa, comes from compressors (1) to the first group of receivers (2). Here it loses some moisture. After that air is directed to

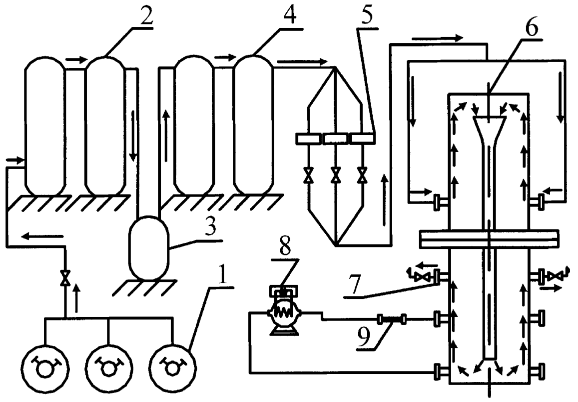


Fig. 1. The aerodynamic setup: (1) compressors; (2) the first group of receivers; (3) dryer; (4) the second group of receivers; (5) flow metering orifices; (6) experimental rig; (7) bypass valves; (8) electric generator; (9) shunt.

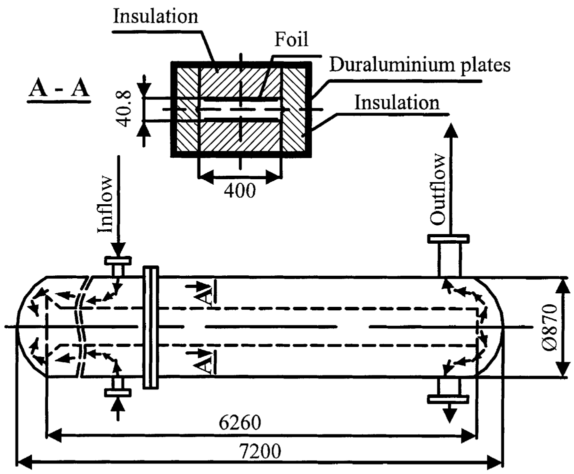


Fig. 2. Test section geometry (not to scale).

the dryer (3) and to the second group of receivers (4). Then it passes one of the three parallel lines with paired flow metering orifices (5) having different flow cross-sections, and intakes into experimental rig (6), then the air is released into the atmosphere via bypass valves (7).

The air pressure and flow rates are controlled by bypass valves (7). Direct electric current generator (8) was used for heating the elements of the experimental section. The current strength was regulated by varying the voltage, applied to an excitation winding of the generator, and determined from the voltage drop on shunts (9) of accuracy class 0.5. A high stability of the generated voltage and, hence, the heat release stability were attained via a particular stabilizer of exciting voltage of generator with a feedback that was taken to be the voltage drop on the calorimetrically measured elements.

The air pressure was measured using the manometers. Well-type differential manometers filled with distilled water measured the pressure drops across the orifices for flow rate measurement. The thermocouples were fabricated from wires of 0.3 mm in diameter covered with thermal insulation and were employed for measuring the wall temperature and the voltage drop along the channel.

All electric signals from thermocouples and voltage drops on the calorimetrically measured surface and shunt were measured via the automated measuring data-collection system connected to the IBM computer.

A special test section for the studies of heat transfer for mixed convection in a flat channel was constructed at the Lithuanian Energy Institute. In order to achieve higher values of *Gr* and higher effects of buoyancy pressurized air up to 0.8 MPa was used. For this purpose the whole test section had to be placed in a pressure vessel of an 870 mm diameter and 7200 mm long. The vessel consists of two halves fixed together by flanges (Fig. 2). The flat channel has a height-to-width ratio of about 1:10 (40.8:400 mm) and it is 6260 mm long. The channel consists of a fluid-dynamic stabilizer and a calorimeter. The stabilizer is 2370 mm long. Its height and width are the same as that of the calorimeter. The entrance to the channel is fitted with a well-designed smooth intake with an array of laminarising grids.

The calorimeter is 3890 mm ($x/d_e \approx 50$) long. The test section is made from duraluminium. The internal side of this duraluminium rig is covered by 120 mm thick insulating strips on top and bottom and by 60 mm thick insulating strips on the cheeks. The calorimetric heated surfaces are made of stainless steel foil 0.38 mm thick and 370 mm wide.

The temperature of the calorimeter test surface was measured by 25 chromel–alumel thermocouples of a 0.3 mm diameter wire, which are fixed to the outer surface of the test foil. Longitudinal voltage drop on the heated foil was measured by the same thermocouples.

This construction enables experiments on heat transfer in a flat channel for different pressures of the heat carrier (air). The air was supplied through the top of channel, so the experiments were accomplished on downward flow with one side (upper wall) heating of the channel. All experiments were performed at the limiting condition $q_w$ = const and a wall-fluid temperature difference up to 150 K. Turbulent mixed convection was studied in the range of *Re* from $4 \times 10^3$ to $4 \times 10^4$, and of $Gr_q$ up to $1.4 \times 10^{10}$. In the primary interpretation of the data, the values of *Nu*, *Re*, $Gr_q$ and *Pr* were evaluated. Local bulk flow velocity, temperature and equivalent diameter of the channel $d_e = 2hb/(h + b) \approx 2h$ were used as reference values.

When calculating convective heat flux, losses of the heat through the insulation, conductive flux along calorimeter wall and losses of the heat due to radiation were taken into account. Special calibration experiments were performed to determine the heat loss through the insulation. During these experiments channel's cross-section was filled with insulation material to avoid heat transfer due to natural convection. So, in this case the heat from the calorimetric foil was directed only through the thermal insulation to the environment. The temperature difference in the insulation was fixed by paired thermocouples. Calibration values ($q_{is} = f(\Delta T_{is})$ were approximated by the second order polynomial and used in basic experiments for evaluation of the heat losses. Experimental uncertainties: *Re*: 4–6%, *Nu*: 4–5.5%, $Gr_q$: 3.5–6.5%. For evaluation of uncertainties methodology presented in Schenck [15] was applied.

## 3. Results

Experimental investigations have been performed in the wide range of parameters ($Re = 4 \times 10^3$–$4 \times 10^4$, $Gr_q = 1.7 \times 10^8$–$1.4 \times 10^{10}$) for different air pressure (0.2, 0.4, 0.6, 0.8 MPa) provided for one-side (upper wall) heating.

Variation of the relative heat transfer rate along inclined channel ($\varphi = 60°$) is presented in Fig. 3. For normalisation of mixed convection heat transfer (*Nu*), Nusselt number $Nu_T$, which stands for heat transfer in the case of a pure forced convection was evaluated for same *Re* numbers, in which the experiments were performed. To evaluate the effect of variable physical properties of the fluid (in this experiment it is not very significant) a technique suggested for annuli (in the limiting case—flat channel) by Vilemas et al. [16] was applied:

$$\frac{Nu_T}{Nu_{\psi=1}} = 1 - 0.744\left\{1 - \exp\left[-K_f(af + n_\mu \Phi K_f)\right]\right\}, \qquad (1)$$

here

$$a = -0.53n_\rho - \tfrac{1}{3}n_\lambda - \tfrac{1}{4}n_c, \quad f = 1 - \exp(-0.1\tilde{x}),$$

$$\Phi = \frac{1.25(0.01\tilde{x})^2}{1 + (0.01\tilde{x})^2}, \quad \tilde{x} = \frac{x}{d_e}, \quad K_f = \frac{q_w d_e}{\lambda_f T_f Nu_{\psi=1}},$$

$n_\rho$, $n_\lambda$, $n_c$, $n_\mu$ are exponents in the functions, which evaluate temperature-dependent properties of individual gases. For calculations approximate values can be applied. For diatomic gases they are: $a = 0.26$, $n_\mu = 0.70$. The value of $Nu_{\psi=1}$ for constant fluid-physical properties ($\psi = T_w/T_f = 1$) and one-side heating was found from the following correlation [2]:

$$Nu_{\psi=1} = 0.01935 Re^{0.8} Pr^{0.6}\left(0.86 + 0.8(x/d_e)^{-0.4}\right). \qquad (2)$$

Data for the highest *Re* numbers ($Re = 20{,}000$–$30{,}000$, Fig. 3b and c) demonstrate that for the conditions of these experiments ratio $Nu/Nu_T$ is close to 1.0 and we have here forced convection regimes or regimes close to forced convection. For smaller *Re* numbers ($Re = 6800$–$20{,}000$) the effect of buoyancy on heat transfer is expressed very well: due to turbulisation of the flow heat transfer is increasing with $x/d_e$ until $x/d_e \approx 15$–$20$ is reached (Fig. 3). After that the relative heat transfer is decreasing slowly. Analysis showed that such decrease of relative heat transfer is related to the decrease of buoyancy parameter $Bo = Gr_q/(Re^m Pr^n)$ along the channel. For smaller *Re* numbers such decrease is more significant. However for the smallest *Re* numbers ($Re = 4800$, Fig. 3b; $Re = 5200$, Fig. 3c) character of the heat transfer variation along the channel is completely different. In case of $Re = 4800$ ($Bo_{in} = 7.13 \times 10^4$, Fig. 3b) until $x/d_e \approx 30$ character of heat transfer variation along the channel is the same as for previous cases but after $x/d_e \approx 30$ is reached, sharp increase in heat transfer is noticed. In case of $Re = 5200$ at $p = 0.6$ MPa (Fig. 3c) the sharp increase of the heat transfer is existing already starting from the beginning of the heating section.

Similar results are for inclination angle $\varphi = 30°$ presented in Fig. 4, for $\varphi = 20°$, and also for vertical channel ($\varphi = 90°$). So, three types of heat transfer regimes can be distinguished in the region of *Re* and $Gr_q$ investigated:

1. Regimes with monotonic increase of heat transfer with increasing of $x/d_e$ due to buoyancy and with some decrease after $x/d_e \approx 30$ due to decrease of buoyancy parameter *Bo* along the channel.
2. Regimes with local heat transfer minimum at some $x/d_e$ and sharp increase of heat transfer after that.
3. Regimes with sharp increase of heat transfer starting from the beginning of the heating section.

Variation of stabilized (quasistabilized) heat transfer upon *Re* number for different air pressures is presented in Figs. 5 and 6.

Figs. 5 and 6 show the close dependence of the heat transfer rate on air pressure in the channel. The higher the pressure (the higher $Gr_q$) is, the more intensive the heat transfer rates are. For highest *Re* numbers in all cases heat transfer rates approach the level of forced convection heat transfer. Mixed convection is not important in this turbulent region, because the hydrodynamic turbulence at these Reynolds numbers is much stronger than the natural convection mechanism and dominates the heat transfer. For the smallest *Re* numbers at 0.4 and 0.6 MPa pressures *Nu* number becomes independent of *Re* (Fig. 5) or even is increasing suddenly (Fig. 6). Such changes at the smallest *Re* numbers is related with the changes of heat transfer regimes anal-

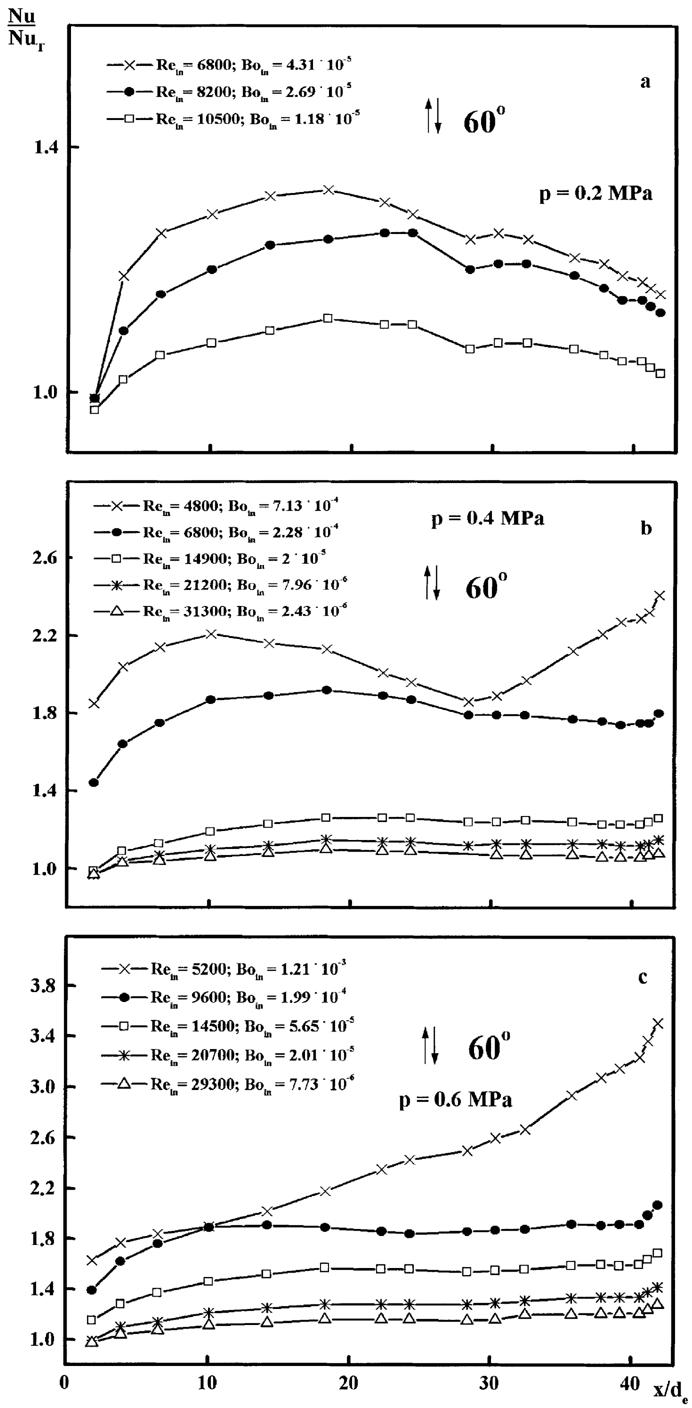


Fig. 3. Longitudinal variation of relative heat transfer, for different $Re_{in}$ ($\varphi = 60°$).

ysed above. It is necessary to notice also that for such *Re* numbers rather significant pulsations of surface temperatures were observed during experiments that are related to the buoyant instabilities.

## 4. Discussion of results

There are some investigations of flow and heat transfer instabilities in opposing mixed convection flows.

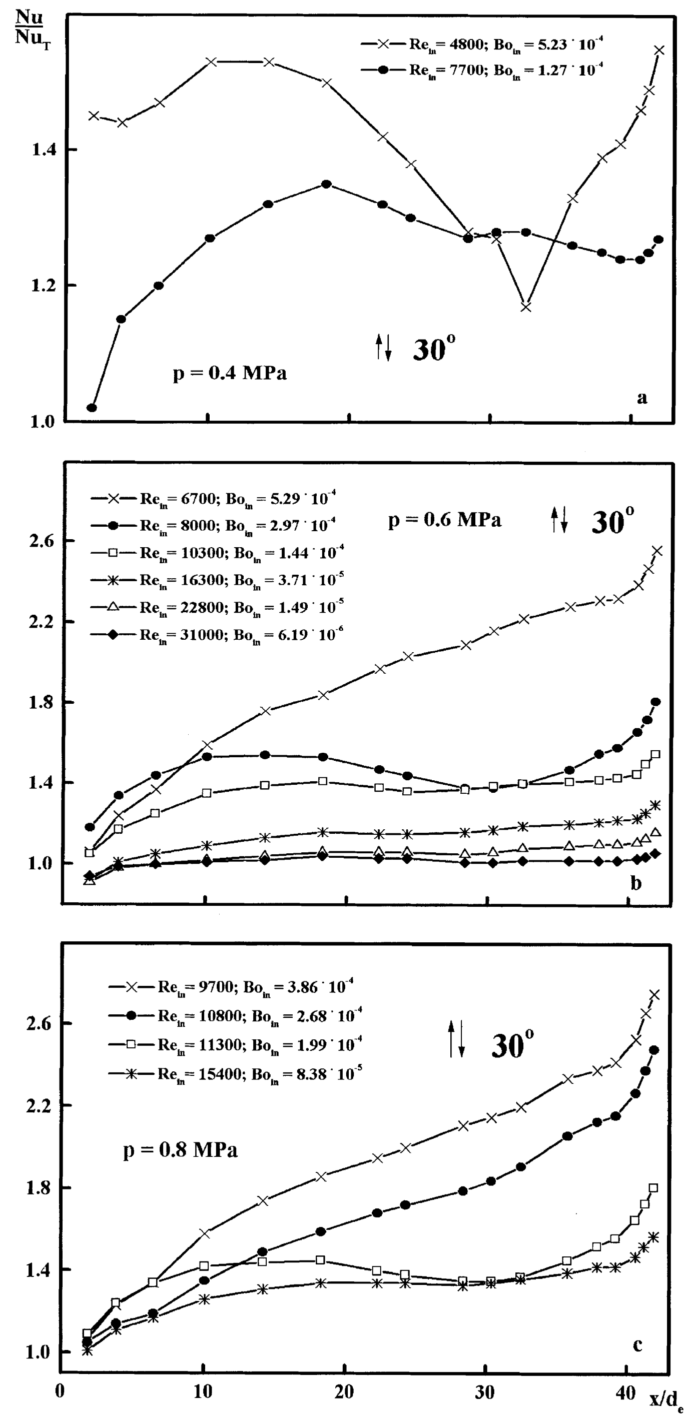


Fig. 4. Longitudinal variation of relative heat transfer, for different $Re_{in}$ ($\varphi = 30°$).

Recently, time dependent opposed mixed convection flow in a vertical channel with specified heat flux conditions has been studied numerically by number of authors at $Re$ number equal to few hundreds, for example Chang and Lin [17], Lin et al. [18] and Evans and Greif [19].

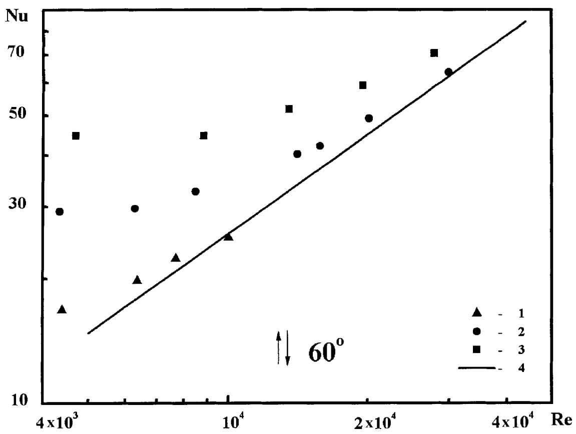


Fig. 5. The relation between the heat transfer and $Re$ for the opposing flows for one-side heated inclined ($\varphi = 60°$) flat channel ($x/d_e = 41.9$). (1) $p = 0.2$ MPa, (2) 0.4 MPa, (3) 0.6 MPa, (4) forced convection for constant fluid-physical properties.

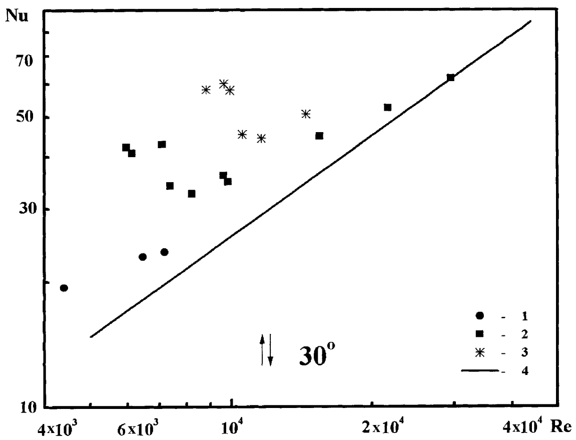


Fig. 6. The relation between the heat transfer and $Re$ for the opposing flows for one-side heated inclined ($\varphi = 30°$) flat channel ($x/d_e = 41.9$). (1) $p = 0.4$ MPa, (2) 0.6 MPa, (3) 0.8 MPa, (4) forced convection for constant fluid-physical properties.

In Evans and Greif [19] unsteady two-dimensional simulation of the downward flow in a vertical heated (isothermal) parallel plate channel with an upstream cold section have been performed. Buoyant instabilities in a symmetrically heated channel have been analyzed for $Re = 219.7$, $Pr = 0.7$ and three different Grashof numbers ($Gr/Re^2 = 1.83$, 8.0 and 13.7). It was demonstrated that for all cases there's an upward buoyant flow near the wall that turns downward at the top of the heated section. Near the walls, the axial flow reverses direction periodically. The lateral flow reverses direction both at the centerline and near the walls. The temperature and axial component of velocity along the centerline of the channel are nonmonotonic and oscillatory. The results are nonsymmetric, periodic and exhibit increasing complexity and frequency for increasing buoyancy. The temperature and axial component of velocity half a period apart show symmetric reflections (of the asymmetric fields) while the lateral component of velocity shows antisymmetric reflections. The spatially averaged Nusselt numbers show a periodic oscillation. The local Nusselt numbers at fixed time exhibit nonmonotonic variations along the channel. Time averaged spatially averaged Nusselt number is increasing with increasing buoyancy.

In Joye and Jacobs [20] flow patterns generated by buoyancy effects in opposing mixed convection heat transfer to water in a vertical tube have been investigated experimentally. Axially moveable thermocouple probes were used to obtain local, dynamic and steady state temperature measurements in the flow. Forced flow Reynolds numbers ranged from 0 to 2400 at Grashof number $5 \times 10^7$ based on tube diameter and bulk fluid properties. It was demonstrated that backflow of hot, large-scale turbulent eddies existed in both the buoyant layer and the bulk flow near the entrance (top) of the heated section of the tube. The intensity of these flow patterns decreased as forced-flow Reynolds number increased. It was indicated that the turbulent backflow causes a large mixing and heat transfer effect at the entrance region, leading to very high heat transfer enhancement.

In our case we have turbulent flow in the hydrodynamic stabilization region and in the heated section. But it is possible to expect that for some $Bo$ parameter values local flow separation can occur like in laminar flows, and this creates buoyancy instabilities.

Dynamics of the position of local minimum heat transfer based on our experimental results is presented in Fig. 7. There is conditionally accepted that minimum heat transfer is at the end ($x/d_e \approx 40$) of channel when impact of buoyancy is rather small. When buoyancy forces increases ($Bo_2$ up to 0.7), there are regimes with clear expressed minimum heat transfer at $x/d_e \approx 25$–35. When $Bo_2 > 1.7$ heat transfer minimum is shifted to the beginning of the heated section of the channel. So Fig. 7 represents three typical regimes discussed above:

1. Until $Bo_2 \leqslant 0.7$ there are regimes without buoyancy instabilities.
2. In the region $0.7 \leqslant Bo_2 \leqslant 1.7$ at the beginning of the heated section there is no buoyancy instabilities but at some $x/d_e$ local separation of the flow occur and buoyancy instabilities are generated that sharply increase heat transfer rate.
3. For $Bo_2 > 1.7$ in all heated section buoyancy instabilities exists and heat transfer is increasing sharply starting from the beginning of the heated section.

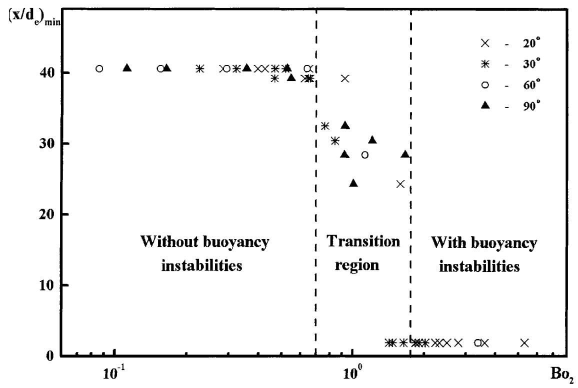


Fig. 7. The relation between the position of the local heat transfer minimum and buoyancy parameter $Bo_2$.

Experiments do not show strong correlation between $(x/d_e)_{min}$ and $Bo_2$ in the transition region. It seams local flow separation (position of the heat transfer minimum) is very sensitive not only to $Bo_2$ variations.

Preliminary calculations performed using $k$–$\varepsilon$ turbulence models confirm existence of such regimes.

## 5. Correlation of the data

Different buoyancy parameters were applied for correlation of mixed convection heat transfer data. In Jackson et al. [3] buoyancy parameter $Bo = Gr_q/(Re^{3.425}Pr^{0.8})$ was used for correlation of data in vertical tubes for aiding and opposing flows. The data of analysis presented in Jackson and Buyukalaca [21] show, that heat transfer data can be correlated successfully for water flow in case of opposing mixed convection, provided that the power $m = 2.0$–$2.636$ for $Re$ in buoyancy parameter $Bo = Gr_q/(Re^m Pr^n)$ is used. In Poškas et al. [22] analysis of heat transfer data at different pressure of the air for aiding flows in vertical flat channel was performed using three buoyancy parameters: $Bo = Gr_q/(Re^{3.425}Pr^{0.8})$, $Bo_1 = Gr_q/(Re^{2.75}Pr^{1.12})$, $Bo_2 = Gr_q/(Re^{2.5}Pr)$. It was demonstrated that different buoyancy parameters must be used for correlation of the data in heat transfer decreasing and recovery regions. For heat transfer decreasing region buoyancy parameter $Bo$ is most suitable, but in the recovery region buoyancy parameter $Bo_2$ must be applied. Similar situation has been demonstrated earlier by Poškas et al. [10] for aiding flow in vertical tube. In Poškas and Poškas [8] for opposing flows in vertical channel also buoyancy parameter $Bo_2$ was justified. Analysis showed that for opposing flows in inclined channels, buoyancy parameter $Bo_2$ is also most suitable.

Data in Fig. 8 indicates that relative heat transfer is increasing with increasing of buoyancy parameter and

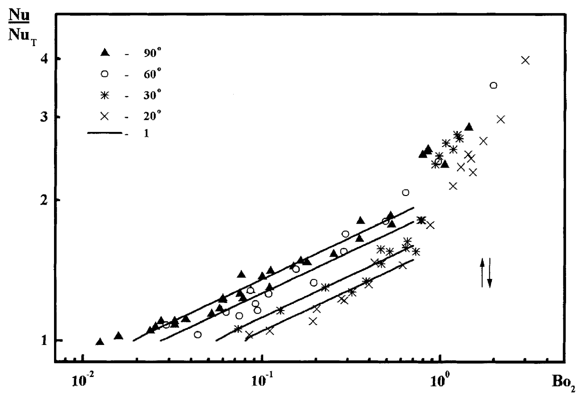


Fig. 8. The relation between the heat transfer and $Bo_2 = Gr_q/(Re^{2.5}Pr)$ for the opposing flows in a one-side heated inclined flat channel ($x/d_e = 41.9$) for different inclination angles $\varphi$: (1) Eq. (3).

decreasing with decreasing of the inclination angle. With decreasing of inclination angle the influence of buoyancy (increase of heat transfer) starts at much higher $Bo_2$ values. Different effect of buoyancy with variation of inclination angle can be explained by interaction of two phenomena: decreasing of flow turbulence due to stable stratification of the flow and increasing of flow turbulence due to opposing mixed convection. Maximum turbulisation of the flow is in case of vertical channel because of maximum effect of opposing mixed convection and no effect of stable stratification of the flow. So, in this case intensification of heat transfer is noticed at smaller buoyancy parameters $Bo_2$ than for inclined channels. With decrease of inclination angle effect of stable stratification becomes more and more expressed so turbulisation of the flow due to opposing mixed convection and as a result intensification of heat transfer is also decreasing.

Three characteristic regimes have been discussed in the previous chapter. It is possible to notice also from Fig. 8, that for $Bo_2 \geqslant 0.7$ when buoyancy instabilities appear the dependence of $Nu/Nu_T$ on $Bo_2$ is changing and intensity of heat transfer is increasing more significantly with increasing of $Bo_2$. Dependence on inclination angle is also decreasing here.

For calculation of stabilized heat transfer at $\varphi = 20$–$90°$ in the region of buoyancy parameter $Bo_{2\mathrm{rib}} \leqslant Bo_2 \leqslant 7 \times 10^{-1}$ simple correlation was suggested:

$$\frac{Nu}{Nu_T} = 1.9 Bo_2^{0.18} \varphi^{0.17}. \tag{3}$$

It correlates experimental data with the uncertainty no more than 10%. $Bo_{2\mathrm{rib}}$ is the value of $Bo_2$ at which $Nu/Nu_T = 1.0$. The correlation for calculation of $Bo_{2\lim}$ was derived from correlation (3):

$$Bo_{2\,\mathrm{lim}} = 0.028\varphi^{-0.95}. \quad (4)$$

The experiments performed at inclination angle $\varphi = 10°$ demonstrate that in some region on buoyancy parameter $Bo_2$ heat transfer is less intensive than forced convection heat transfer ($Nu/Nu_T < 1.0$). This means that laminarisation of the flow occurs already at this inclination angle like in horizontal channels.

## 6. Conclusions

Analysis of the experimental data on turbulent mixed convection heat transfer in vertical and inclined flat channel ($\varphi = 60°$, 30°, 20°) with one-side heating (upper wall is heated) over wide ranges of the parameters ($Re = 4 \times 10^3$–$4 \times 10^4$, $Gr_q = 1.7 \times 10^8$–$1.4 \times 10^{10}$, $p =$ 0.2–0.8 MPa) lead to the following conclusions:

1. Three characteristic regions were identified for turbulent opposing mixed convection in inclined flat channel with stably stratified flows based on local heat transfer data:
   (a) Region without buoyancy instabilities until $Bo_2 \leqslant 0.7$.
   (b) In the region $0.7 \leqslant Bo_2 \leqslant 1.7$ at the beginning of the heated section there is no buoyancy instabilities but at some $x/d_e$ local separation of the flow occur and buoyancy instabilities are generated that increase sharply heat transfer rate.
   (c) For $Bo_2 > 1.7$ there are buoyancy instabilities in all heated section and heat transfer is increasing sharply starting from the beginning of the heated section.
2. Correlation (3) for calculation of heat transfer in inclined flat channels was suggested for the region without buoyancy instabilities.

## References


[1] B. Metais, E.R.G. Eckert, Forced, mixed and free convection regimes, ASME, Int. J. Heat Transfer 2 (1964) 295.
[2] B.S. Petukhov, A.F. Polyakov, Turbulent Mixed Convection Heat Transfer (in Russian), Nauka, Moscow, 1986, p. 192.
[3] J.D. Jackson, M.A. Cotton, B.F. Axcell, Studies of mixed convection in vertical tubes, Int. J. Heat Fluid Flow 10 (1989) 2–15.
[4] J. Vilemas, P. Poškas, in: A. Žukauskas (Ed.), Effect of Body Forces on Turbulent Heat Transfer in Channels, 1st ed., LEI-Bigell House, Kaunas-New York, 1999, p. 195.
[5] S.W. Churchill, Combined free and forced convection in channels, in: E.U. Schlunder (Ed.), Heat Exchanger Design Handbook, vol. 2, Hemisphere, New York, 1983.
[6] B.S. Petukhov, Turbulent flow and heat transfer in pipes under considerable effect of thermo-gravitational forces, in: D.B. Spalding, N. Afgan (Eds.), Heat Transfer and Turbulent Buoyant Convection, vol. 2, Hemisphere, New York, 1977.
[7] L.W. Swanson, I. Catton, Surface renewal theory for turbulent mixed convection in vertical ducts, Int. J. Heat Mass Transfer 30 (1987) 2271–2279.
[8] P. Poškas, R. Poškas, Turbulent opposing mixed convection heat transfer in a vertical flat channel with one side heating, Int. J. Heat Transfer Engineering, accepted for publication.
[9] J. Vilemas, P. Poškas, V. Kaupas, Local heat transfer in a vertical gas-cooled tube with turbulent mixed convection and different heat fluxes, Int. J. Heat Mass Transfer 35 (1992) 2421–2428.
[10] P. Poškas, R. Pabarčius, J. Vilemas, Experimental investigation of turbulent mixed convection heat transfer in a vertical tube for different pressure of air, in: B. Sunden, E. Blums, A. Žukauskas (Eds.), Advances in Engineering Heat Transfer, Computational Mechanics Publications, UK, 1995, pp. 139–148.
[11] T. Aicher, H. Martin, New correlations for mixed turbulent natural and forced convection heat transfer in vertical tubes, Int. J. Heat Mass Transfer 40 (1997) 3617–3626.
[12] G.P. Celata, F. D'Annibale, A. Chiaradia, M. Cumo, Upflow turbulent mixed convection heat transfer in vertical pipes, Proceedings of the Eleventh International Heat Transfer Conference, 1998, pp. 287–292.
[13] P. Poškas, G. Bartkus, J. Vilemas, Turbulent mixed convection heat transfer in upward inclined flat channel gas flow, Proceedings of the Eleventh International Heat Transfer Conference, Kyongju, Korea, vol. 3, 1998, pp. 275–280.
[14] P. Poškas, R. Poškas, Turbulent mixed convection heat transfer in an inclined flat channel with opposing flows, Proceedings of the Twelfth International Heat Transfer Conference, Grenobl, France, vol. 2, 2002, pp. 243–249.
[15] H. Schenck, Theories of Engineering Experimentation, Mir, Moscow (Russian translation by Kovalenko E.G., edited by N.P. Buslenko), 1972, p. 381.
[16] J. Vilemas, B. Čėsna, V. Survila, Heat Transfer in Gas-Cooled Annular Channels, Hemisphere, New York, 1987, p. 226.
[17] T.S. Chang, T.F. Lin, Steady and oscillatory opposing mixed convection in a symmetrically heated vertical channel with a low Prandtl number fluid, Int. J. Heat Mass Transfer 36 (1993) 3783–3795.
[18] T.F. Lin, T.S. Chang, Y.F. Chen, Development of oscillatory asymmetric recirculating flow in transient laminar opposing mixed convection in a symmetrically heated vertical channel, Int. J. Heat Transfer 115 (1993) 342–352.
[19] G. Evans, R. Greif, Buoyant instabilities in downward flow in a symmetrically heated vertical channel, Int. J. Heat Mass Transfer 40 (1997) 2419–2430.
[20] D.D. Joye, S.W. Jacobs, Backflow in the inlet region of opposing mixed convection heat transfer in a vertical tube, Proceedings of the Tenth International Heat Transfer Conference, Brighton, vol. 5, 1994, pp. 489–494.

[21] J.D. Jackson, O. Buyukalaca, Correlation of heat transfer data for buoyancy opposed turbulent mixed convection in vertical tube, Proceedings of Eurotherm 32 Seminar, Heat Transfer in Single Phase Flows, Oxford university, 1993, pp. 105–107.

[22] P. Poškas, J. E. Adomaitis, J. Vilemas, G. Bartkus, Development of turbulent heat transfer over the length of vertical flat channel under a strong influence of buoyancy, Proceedings of the Tenth International Heat Transfer Conference, Vol. 5. 1994, pp. 555–560.