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

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Abstract

Local turbulent mixed convection heat transfer in inclined (from $\varphi = 0°$ (horizontal position) till $\varphi = 90°$ (vertical position)) flat channels for opposing flows was investigated for the case when only bottom wall is heated (unstably stratified flow conditions). Wide ranges of airflow parameters are covered: $Re = 4 \times 10^3$–6.6×10^4, $Gr_q = 4.7 \times 10^7$–6.3×10^{10}, pressure $p = 0.1$; 0.2; 0.4; 0.7; 1.0 MPa. Correlation for calculation of heat transfer in inclined flat channels was suggested for the region without buoyancy instabilities. The experimental data were compared with the recent experimental data for inclined flat channels when upper wall is heated (stably stratified flow conditions).

Keywords: Heat transfer; Turbulent mixed convection; Inclined flat channel; Opposing airflows; Unstable stratification; Experiments

1. Introduction

Internal mixed convection heat transfer in different channels can be applied to nuclear power technology, chemical process heat transfer, electronic cooling, etc. Significant heat transfer enhancement may be realized by mixed convection in some situations, especially in opposing flows. Thermally stratified flows, in which buoyancy plays an important role in turbulent transport of heat, mass and momentum, occur in the ocean, the atmospheric boundary layer, and also in many industrial operations.

Due to the importance of the problem for engineering applications much attention has been concentrated by researchers to the turbulent mixed convection heat transfer investigations in vertical circular tubes. Reviews on mixed convection have been published by Metais and Eckert [1], Jackson et al. [2], Petukhov and Poliakov [3], Vilemas and Poskas [4], Jackson [5] etc. An interesting fact disclosed is that for aiding flows the effectiveness of heat transfer can be seriously impaired as a result of buoyancy forces modifying the production of turbulence and laminarizing the flow. However, if higher buoyancy parameters are applied, heat transfer recovers and becomes even higher than forced convection heat transfer. For opposing flows heat transfer increases when the effect of buoyancy increases.

Number of correlations has been proposed to describe mixed convection heat transfer in vertical tubes. First of all correlations for mean heat transfer have been proposed, for example, by Charchil [6], Petukhov [7], Swanson and Catton [8] (for vertical channel) etc. Later on, when investigations of local heat transfer coefficients

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Nomenclature

Bo	buoyancy parameter, $Bo = Gr_q/(Re^{3.425} Pr^{0.8})$
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, $d_e = 2hb/(h + b)$ (m)
Gr_q	Grashof number defined by the heat flux specified on the surface, $Gr_q = g\beta d_e^4 q_w/\nu^2\lambda$
g	acceleration due to gravity (m/s^2)
h	channel height (m)
Nu	Nusselt number, $Nu = d_e/\lambda$
Pr	Prandtl number, $Pr = \mu c_p/\lambda$
p	pressure (Pa)
q	heat flux density (W/m^2)
Re	Reynolds number, $Re = u_f d_e/\nu$
T	temperature (K)
u	flow velocity (m/s)
x	distance from the heating origin (m)

Greek symbols

α	heat transfer coefficient, $\alpha = q_w/(T_w - T_f)$ (W/(m^2 K))
β	volumetric expansion coefficient (1/K)
λ	thermal conductivity (W/(m K))
ν	kinematics viscosity (m^2/s)
μ	dynamic viscosity (Pa s)
φ	angle (degree) or (rad) (in correlations)

Subscripts

f	in the flow
T	forced turbulent convection
w	at the wall
in	at the inlet
ψ	temperature factor

have been performed correlations for stabilized heat transfer have been proposed, by Petukhov and Poliakov [3], Jackson et al. [2], Poskas and Poskas [9] (for vertical flat channel) etc.

In Vilemas et al. [10] correlations for calculation of local heat transfer coefficients along the tube for aiding airflow for different temperature factors was proposed based on authors data. In Poskas et al. [11] 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 $x/d_e = 50$ 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 mean heat transfer in vertical tubes for aiding water flows taking into account effect of the length of the tube have been proposed in Aicher and Martin [12] 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 seams no influence of the L/d ratio on the heat transfer rate. In Celata et al. [13] the new correlation for calculation of mean 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.

Studies on turbulent mixed convection in horizontal channels are not numerous and mostly have been performed in horizontal tubes (Petukhov and Poliakov [3]). Due to effect of secondary flows and stratification of the flow significant variation of heat transfer around the perimeter of the tube was detected.

There are no secondary flows related to natural convection in horizontal flat channels and this facilitates an analysis of the effects of both stable and unstable stratification of density on the turbulent transport. The unstable stratification leads to an augmented heat transfer, while the stable stratification leads to a decreased heat transfer and ends with a transition to the laminarized mode of the heat transfer (we are also applying this terminology for inclined channels). The correlation for the calculation of heat transfer in horizontal flat channels under such conditions was suggested in (Petukhov and Poliakov, [3]), based on the experimental data in air flow at atmospheric pressure.

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 by Poskas et al. [14]. It was revealed that in case of stable density stratification and two side heating conditions, the maximum decrease of heat transfer does not depend on the orientation of the flat channel, but the critical buoyancy parameter is closely related to the inclination angle. For an unstable density stratification case, the relative heat transfer rate is loosely related to the inclination angle of the channel in the interval of $\varphi = 0-60°$.

Experimental data on heat transfer for opposing flows under stable stratification have been presented in Poskas and Poskas [15]. 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.

The aim of this paper is to present experimental results on local heat transfer in the inclined flat channels for unstably stratified flow (only bottom 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.

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 0.3–0.8 MPa, comes from compressors (1) to the first group of receivers (2). Here it loses some moisture. After that air is directed to 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. The high stability of the generated voltage, which also means the stability of the supplied heating power, was attained by a special electronic control device with feedback consisting of the voltage drop at the test section.

A special test section (Fig. 2) for the studies of heat transfer for mixed convection in a flat channel was constructed at the Laboratory of Nuclear Engineering, Lithuanian energy institute. 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 laminarizing grids. The calorimeter is 3890 mm ($x/d_e \approx 50$) long. The test section contains two central and two lateral duraluminium plates. The internal side of this duraluminium rig is insulated by 120 mm thick asbestos strips on the central plates and by 60 mm thick asbestos-cement strips on the lateral plates. The calorimetric heated surfaces of the two central plates are made of stainless steel foil 0.38 mm thick and 370 mm wide, fixed at the inlet to the brass contacts.

In order to achieve higher values of Gr_q and higher effects of buoyancy pressurized air up to 1.0 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.

The temperature of the calorimeter test-surface (heated foil) was measured by 25 chromel–alumel

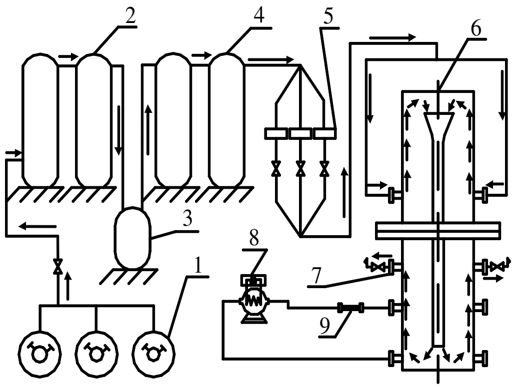

Fig. 1. The aerodynamic setup: 1—compressors; 2—the first group of receivers; 3—dryer; 4—the second group of receivers; 5—flow metering orifices; 6—experimental rig; 7—bypass valves; 8—electric generator; 9—shunt.

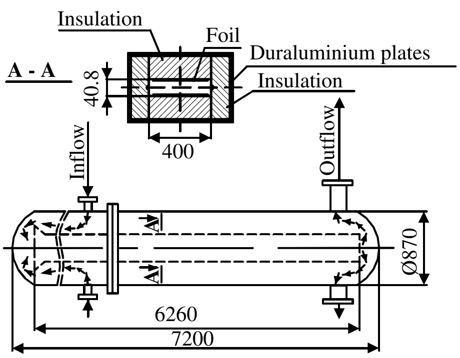

Fig. 2. Test section geometry (not to scale).

thermocouples of a 0.3 mm diameter wire, which are fixed to the outer surface of the test foil. Longitudinal voltage drop on this heated foil was measured by the same thermocouples. Thermocouples also measured the temperatures in the insulation layers, which were used to define the losses.

The above 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 the channel, so the experiments were accomplished on downward flow with one-side (bottom wall) heating of the channel. All experiments were performed at the limiting condition $q_w = \text{const}$ and a wall-fluid temperature difference up to 150 K.

The air pressure was measured using manometers. Well-type differential manometers filled with distilled water measured the pressure drops across the orifices.

Electric signals from thermocouples and pressure probes as well as voltage drops on the calorimetrically measured surface and shunt were all recorded automatically by a computer-based data acquisition system, and processed on line.

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.

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 were used as reference values.

Experimental uncertainties were as follows: Re—2.6–3.1%, Nu—2.8–3.8%, Gr_q—2.6–5.5%. The methodology presented by Schenck [16] was applied for the evaluation of uncertainties.

The more detailed description of experimental rig and methodology is presented in Poskas and Poskas [15].

3. Results

Experimental investigations have been performed in the wide range of parameters ($Re = 4 \times 10^3$–6.6×10^4, $Gr_q = 4.7 \times 10^7$–6.3×10^{10}) for different air pressure (0.1; 0.2; 0.4; 0.7; 1.0 MPa) provided for one-side (bottom wall) heating.

Variation of stabilized (quasistabilized) heat transfer upon Re number for different air pressures is presented in Figs. 3 and 4.

Figs. 3 and 4 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 heat transfer rates are approaching 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. In case of inclined channels for the smallest Re numbers at all pressures Nu numbers are

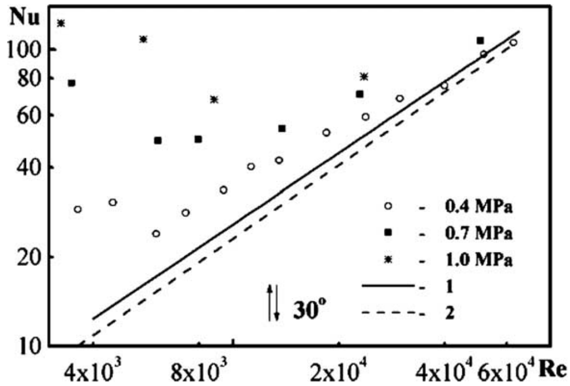

Fig. 3. The relation between the heat transfer and Re for the opposing flows in one-side heated inclined ($\varphi = 30°$) flat channel ($x/d_e = 42$). 1—forced convection for constant fluid-physical properties, 2—forced convection for variable fluid-physical properties (experimental conditions, $\psi = T_w/T_f = 1.2$–1.3).

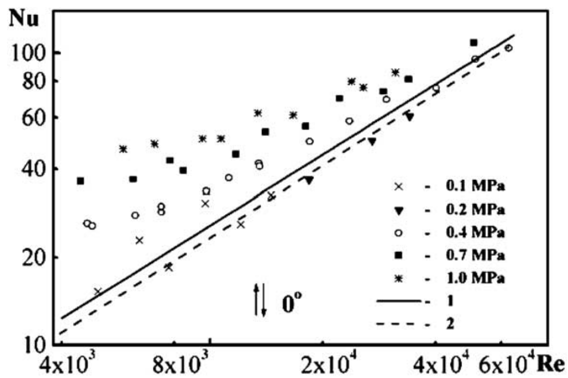

Fig. 4. The relation between the heat transfer and Re for the opposing flows in one-side heated inclined ($\varphi = 0°$) flat channel ($x/d_e = 42$). 1—forced convection for constant fluid-physical properties, 2—forced convection for variable fluid-physical properties (experimental conditions, $\psi = T_w/T_f = 1.2$–1.3).

increasing suddenly (Fig. 3). Such changes at the smallest Re numbers is related with the changes of heat transfer regimes. 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. Similar results were received also for higher inclination angles ($\varphi = 60°$) and in vertical channel ($\varphi = 90°$). Such instabilities were revealed also in case of stable stratification (Poskas and Poskas [15]).

Different situation is in the horizontal channel (Fig. 4). At small air pressures (0.1–0.4 MPa) even for the smallest Re numbers Nu number is still dependent on Re number and no evidence of the change in heat transfer process. When the air pressure is higher heat transfer becomes practically not dependent on Reynolds number at $p = 0.7$ from $Re \approx 7000$ and at $p = 1.0$ MPa from $Re \approx 8000$. So really we have natural convection here and there is no sudden "jump" as in case of inclined channels.

Variation of the relative heat transfer along inclined channel ($\varphi = 30°$) is presented in Fig. 5. For normalization of mixed convection heat transfer (Nu), Nusselt

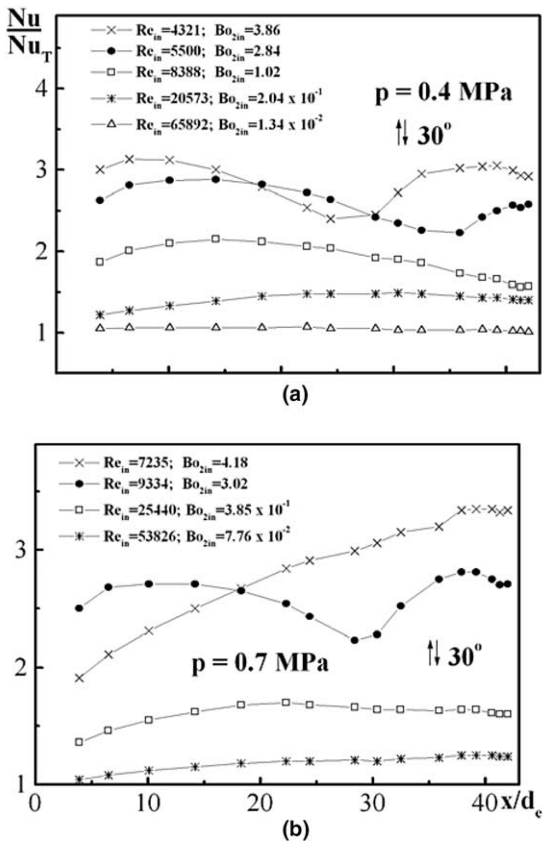

Fig. 5. Longitudinal variation of relative heat transfer, for different Re_{in} ($\varphi = 30°$).

number, which stands for heat transfer in the case of a pure forced convection (Nu_T) was evaluated for the same Re numbers, for which the experiments were performed. To evaluate the effect of variable physical properties of the fluid (in these experiments it is not very significant) a technique suggested for annuli (in the limiting case–flat channel) by Vilemas et al. [17] was applied

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

here $a = -0.53n_\rho - \frac{1}{3}n_\lambda - \frac{1}{4}n_c$,

$$f = 1 - \exp(-0.1\tilde{x}),$$

$$\Phi = \frac{1.25 \cdot (0.01\tilde{x})^2}{1 + (0.01\tilde{x})^2},$$

$$\tilde{x} = \frac{x}{d_e},$$

$$K_f = \frac{q_w d_e}{\lambda_f T_f Nu_{\psi=1}},$$

$n_\rho, n_\lambda, n_c, n_\mu$, 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: $a = 0.26$, $n_\rho = 0.70$. 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 (Petukhov and Poliakov [3])

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

As it was shown by Poskas and Poskas [15] in case of opposing mixed convection under stable stratification conditions in inclined channel or in vertical channel three types of heat transfer regimes can be distinguished

1. Regimes with monotonic increase of heat transfer with increasing of x/d_e due to buoyancy and with some decrease after some x/d_e due to decrease of buoyancy parameter Bo_2 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.

The same situation is also in inclined channels with unstable stratification (Fig. 5). Data for the highest Re numbers ($Re = 5.4 \times 10^4$–6.6×10^4, Fig. 5(a) and (b)) demonstrate that for the conditions of these experiments the Nu/Nu_T ratio is close to 1.0 and we have here the forced convection regimes or regimes close to forced convection. For smaller Re numbers ($Re = 8300$–21000 at $p = 0.4$ MPa (Fig. 5(a)) and $Re = 25000$ at $p = 0.7$ MPa (Fig. 5(b))) the effect of buoyancy on heat transfer is expressed very well: due to turbulization of the flow under unstable stratification conditions heat transfer is increasing with x/d_e until $x/d_e \approx 15–20$ is reached. 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_2 along the channel. For smaller Re numbers such decrease is more significant. However for the smallest Re numbers ($Re = 4300–5500$, Fig. 5(a); $Re = 9300$, Fig. 5(b)) character of the heat transfer variation along the channel is completely different. In these cases at some x/d_e sharp increase in heat transfer is noticed. In case of $Re = 7200$ at $p = 0.7$ MPa (Fig. 5(b)) the sharp increase of the heat transfer is starting already from the beginning of the heating section.

Results for horizontal channel ($\varphi = 0°$) are presented in Fig. 6. Data for the highest Re numbers ($Re = 5.4 \times 10^4–6.6 \times 10^4$) are close to the forced convection regimes. For smaller Re numbers due to turbulization of the flow heat transfer is increasing with x/d_e until $x/d_e \approx 10–15$ is reached (Fig. 6(a) and (b)). After that the relative heat transfer is decreasing slowly. So, in case of horizontal channel only regimes with monotonic increase of heat transfer exist.

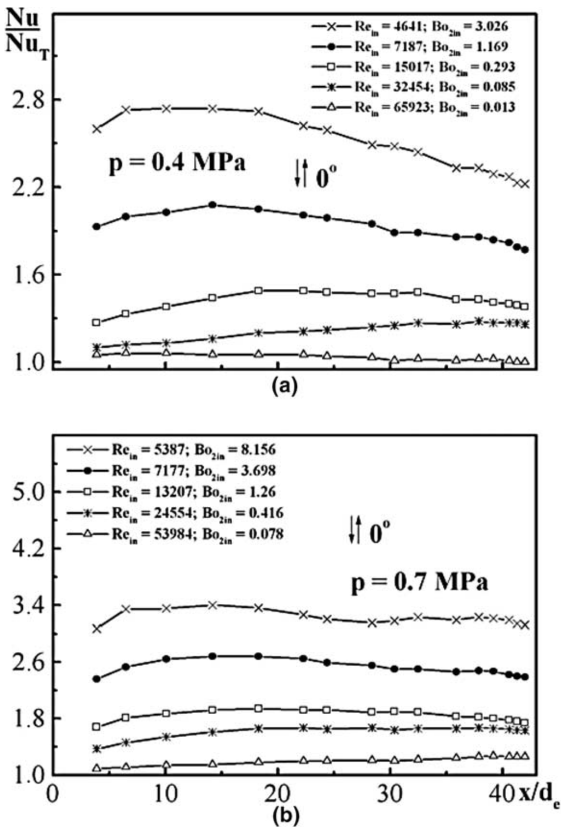

Fig. 6. Longitudinal variation of relative heat transfer, for different Re_{in} ($\varphi = 0°$).

4. Discussion of results

We are analyzing turbulent flow ($Re \geqslant 4000$) in the hydrodynamic stabilization region and in the heated section. But it is known from recent investigations (Poskas and Poskas [15], Poskas et al. [18]) that at high Bo_2 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 some x/d_e. When $Bo_2 > 2$ heat transfer minimum is shifted to the beginning of the heated section of the channel. So Fig. 7 represents three typical regimes. Until $Bo_2 \leqslant 0.7$ there are regimes without buoyancy instabilities. In the region $0.7 \leqslant Bo_2 \leqslant 2$ 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. For $Bo_2 > 2$ buoyancy instabilities exists in all heated section and heat transfer is increasing sharply starting from the beginning of the heated section.

It is evident from Fig. 7 that under unstable stratification conditions local heat transfer minimum and herewith flow separation is shifted towards the higher buoyancy parameter values in comparison with stable stratification case. This could be explained by the specific of the velocity profiles in stably and unstably stratified flows. Experimental investigations by Petukhov and Poliakov [3] performed in horizontal channels with both sides heating indicate that under effect of stable (upper wall heated) and unstable (bottom wall heated) stratification velocity and temperature profiles become asymmetrical. Under effect of unstable stratification

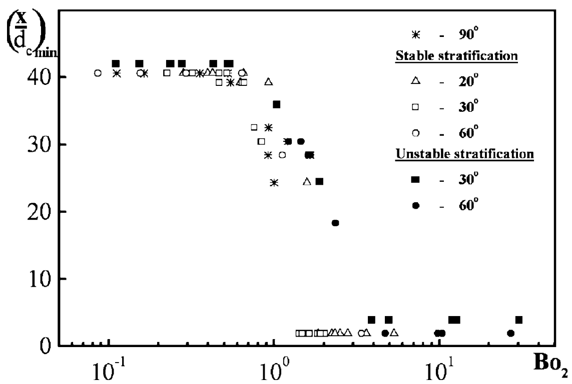

Fig. 7. The relation between the position f the local heat transfer minimum and buoyancy parameter Bo_2.

turbulence of the flow is increasing and velocity and temperature profiles become flatter then in forced convection (unstratified flow) case. In the region of stable stratification the flow is laminarized and velocity and temperature profiles are approaching profiles typical for laminar flow. Accordingly, the velocity (and temperature) gradient at the surface and therefore the surface shear stress is larger for unstable stratification than in case of stable or unstratified flow. Separation of the flow from the surface occurs when velocity gradient at the wall becomes zero. So, this condition for unstable stratification case will be meet at higher x/d_e or Bo_2 parameter values than in case of stable stratification.

Experiments show more strong correlation between $(x/d_e)_{min}$ and Bo_2 in the transition region ($0.7 \leqslant Bo_2 \leqslant 2$) for unstably stratified flow than for stably stratified one (Fig. 7). It seams local flow separation (position of the heat transfer minimum) is very sensitive not only to Bo_2 variations.

Calculations by Poskas et al. [18] performed using k–ε turbulence models confirm existence of such regimes. Experiments show that at high Re numbers wall temperature is monotonically increasing with x/d_e. Modelling results demonstrate the same tendency. The velocity in the whole cross-section is downward oriented. For smaller Re numbers experiments show sharp decrease of wall temperature starting from x/d_e (Nu is increasing sharply). Numerical results do not follow this but air flow at the heated wall is oriented already upward when in the most part of the cross-section it is oriented downward (direction of forced flow). Such velocity profile has been measured experimentally by Jackson et al. [19] but for much smaller Re number ($Re = 2000$) when at the inlet to the heated section the flow is still laminar.

5. Correlation of the data

In Poskas et al. [20] 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 Poskas et al. [11] for aiding flow in vertical tube. In Poskas and Poskas [9], [15] for opposing flows in vertical channel buoyancy parameter Bo_2 was justified.

Performed analysis showed that for opposing flows in inclined channels under unstable stratification conditions, buoyancy parameter Bo_2 is also most suitable. Data in Fig. 8 indicates that relative heat transfer is

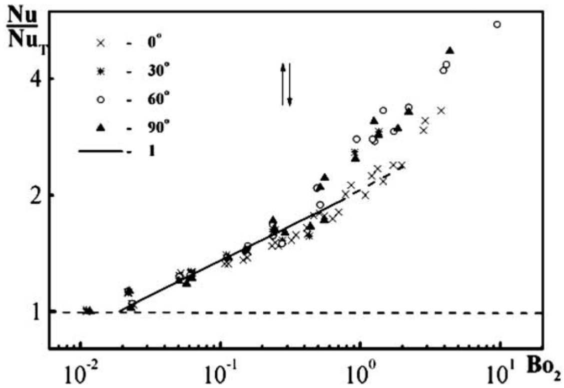

Fig. 8. The relation between stabilized ($x/d_e = 42$) heat transfer and Bo_2 for the opposing flows in a one-side heated inclined flat channel for different inclination angles φ: 1—Eq. (4).

increasing with increasing of buoyancy parameter. There is no evidence of heat transfer dependence on inclination angle at relatively small Bo_2 parameters (till $Bo_2 = 0.7$). This is in contradiction with stable density stratification case (Poskas and Poskas [15]) where clear dependence on inclination angle was defined. At small Bo_2 values $Nu/Nu_T = 1$, and we have forced convection regime here. 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. The heat transfer dependence on inclination angle also is evident there.

In case of stable density stratification for calculation of stabilized heat transfer at $\varphi = 20°$–$90°$ in the region of buoyancy parameter $Bo_{2rib} \leqslant Bo_2 \leqslant 7 \times 10^{-1}$ correlation was suggested (Poskas and Poskas [15])

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

here φ—in rad.

As we already discussed in case of unstable stratification the heat transfer dependence on inclination angle for $Bo_2 < 0.7$ is not significant therefore stabilized heat transfer can be calculated from (3) when $\varphi = 90°$

$$\frac{Nu}{Nu_T} = 2.05Bo_2^{0.18}, \tag{4}$$

It correlates experimental data with the uncertainty no more than 12%.

Bo_{2rib} is the value of Bo_2 at which $Nu/Nu_T = 1.0$. In this case $Bo_{2rib} = 0.018$.

When $Bo_2 > 0.7$ the character of heat transfer is changing for all inclination angles but for horizontal channel this happens at higher buoyancy parameter (Bo_2). This means that no flow separation occurs in horizontal channels and our data show that correlation (4) can be extended till $Bo_2 \approx 2$. For $Bo_2 \geqslant 2$ the dependence of Nu/Nu_T on Bo_2 is also changing in horizontal channel but this is related with the transition to natural convection regime as discussed earlier.

It is necessary to indicate that additional investigations of local heat transfer are necessary at small inclination angles of the channel (close to horizontal position) where the regimes with and without flow separation from the wall can exist.

6. Conclusions

Analysis of the experimental data on turbulent mixed convection heat transfer in inclined flat channels with one-side heating under unstably stratified flow conditions over wide ranges of the parameters ($Re = 4 \times 10^3$–6.6×10^4, $Gr_q = 4.7 \times 10^7$–6.3×10^{10}, pressure $p =$ 0.1; 0.2; 0.4; 0.7; 1.0 MPa) lead to the following conclusions

1. Three characteristic regions were identified for turbulent opposing mixed convection in inclined flat channels with unstably stratified flows (like in stably stratified flows Poskas and Poskas [15]) based on local heat transfer data:
   (a) Region without buoyancy instabilities until $Bo_2 \leqslant 0.7$.
   (b) In the region $0.7 \leqslant Bo_2 \leqslant 2$ 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 $Bo_2 > 2$ there are buoyancy instabilities in all heated section and heat transfer is increasing sharply starting from the beginning of the heated section.
2. There are no such local flow separations from the wall in the horizontal channel and the changes in heat transfer are related with the transition to natural convection regime.
3. Correlation (4) for calculation of stabilized heat transfer in inclined flat channels was suggested for the region without buoyancy instabilities. Heat transfer rate do not depend on inclination angle of the channel in this region.

References

[1] B. Metais, E.R.G. Eckert, Forced, mixed and free convection regimes, ASME, Int. J. Heat Transfer (2) (1964) 295.
[2] 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.
[3] B.S. Petukhov, A.F. Polyakov, Turbulent mixed convection heat transfer, Nauka, Moscow, 1986, pp. 192 (in Russian).
[4] J. Vilemas, P. Poškas, in: A. Žukauskas (Ed.), Effect of Body Forces on Turbulent Heat Transfer in Channels, LEI-Bigell House, Kaunas-New York, 1999, pp. 195.
[5] J.D. Jackson, Influences of buoyancy on velocity, turbulence and heat transfer in ascending and descending flows in vertical passages, in: B. Sunden, J. Vilemas (Eds.), Proceedings of the Fourth Baltic Heat Transfer Conference, Advances in Heat Transfer Engineering, LEI-Bigell House, Kaunas-New York, 2003, pp. 57–78.
[6] S.W. Churchill, Combined free and forced convection in channels, in: E.U. Schlunder (Ed.), Heat Exchanger Design Handbook, 2, Hemisphere, New York, 1983.
[7] 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, 2, Hemisphere, New York, 1977.
[8] L.W. Swanson, I. Catton, Surface renewal theory for turbulent mixed convection in vertical ducts, Int. J. Heat Mass Transfer (30) (1987) 2271–2279.
[9] P. Poškas, R. Poškas, Turbulent opposing mixed convection heat transfer in a vertical flat channel with one side heating, Int. J. Heat Transfer Eng. 25 (2) (2004) 17–23.
[10] 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.
[11] P. Poskas, 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.), Proceedings of the Second Baltic Heat Transfer Conference, Advances in Engineering Heat Transfer, Computational Mechanics Publications, UK, 1995, pp. 139–148.
[12] 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.
[13] G.P. Celata, F. D'Annibale, A. Chiaradia, M. Cumo, Upflow turbulent mixed convection heat transfer in vertical pipes, in: J.S. Lee (Ed.), Proceedings of the Eleventh International Heat Transfer Conference, 3, Taylor & Francis, Levittown, PA, 1998, pp. 287–292.
[14] P. Poškas, G. Bartkus, J. Vilemas, Turbulent mixed convection heat transfer in upward inclined flat channel gas flow, in: J.S. Lee (Ed.), Proceedings of the Eleventh International Heat Transfer Conference, 3, Taylor & Francis, Levittown, PA, 1998, pp. 275–280.
[15] P. Poskas, R. Poskas, Local turbulent opposing mixed convection heat transfer in inclined flat channel for stably stratified airflow, Int. J. Heat Mass Transfer 46 (21) (2003) 4023–4032.
[16] H. Schenck, Theories of Engineering Experimentation, Mir, Moscow, 1972, pp. 381 (Russian translation by E.G. Kovalenko, ed. N.P. Buslenko).
[17] J. Vilemas, B. Čėsna, V. Survila, Heat Transfer in Gas-Cooled Annular Channels, Hemisphere, New York, 1987, pp. 226.
[18] P. Poskas, R. Poskas, D. Sabanskis, Turbulent opposing mixed convection heat transfer in vertical flat channel with symmetrical heating, in: B. Sunden, J. Vilemas (Eds.),

Proceedings of the Fourth Baltic Heat Transfer Conference, Advances in Heat Transfer Engineering,, LEI-Bigell House, Kaunas-New York, pp. 411–418.

[19] J.D. Jackson, T.-H. Wu, Z. Xu, Proceedings of the Twelfth International Heat Transfer Conference, 2, Elsevier, Grenoble, France, 2002 (in CD).

[20] P. Poskas, 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, in: G.F. Hewitt (Ed.), Proceedings of the Tenth International Heat Transfer Conference, 5, Taylor & Francis, Bristol, PA, 1994, pp. 555–560.