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ANALYSIS OF TRANSIENT MIXED CONVECTION IN A HORIZONTAL CHANNEL PARTIALLY HEATED FROM BELOW

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The heat convection phenomenon has been investigated numerically (mathematically) for a channel located horizontally and partially heated at a uniform heat flux with forced and free heat convection. The investigated horizontal channel with a fluid inlet and the enclosure was exposed to the heat source from the bottom while the channel upper side was kept with a constant temperature equal to fluid outlet temperature. Transient, laminar, incompressible and mixed convective flow is assumed within the channel. Therefore, the flow field is estimated using Navier Stokes equations, which involves the Boussinesq approximation. While the temperature field is calculated using the standard energy model, where, Re, Pr, Ri are Reynolds number, Prandtl number, and Richardson number, respectively. Reynolds number (Re) was changed during the test from 1 to 50 (1, 10, 25, and 50) for each case study, Richardson (Ri) number was changed during the test from 1 to 25 (1, 5, 10, 15, 20, and, 25). The average Nusselt number (Nuav) increases exponentially with the Reynold number for each Richardson number and the local Nusselt number (NuI) rises in the heating point. Then gradually stabilized until reaching the endpoint of the channel while the local Nusselt number increases with a decrease in the Reynolds number over there. In addition, the streamlines and isotherms patterns in case of the very low value of the Reynolds number indicate very low convective heat transfer with all values of Richardson number. Furthermore, near the heat source, the fluid flow rate rise increases the convection heat transfer that clarified the Nusselt number behavior with Reynolds number indicating that maximum Nu To. are 6, 12, 27 and 31 for Re No. 1, 10, 25 and 50, respectively

Keywords: mixed convection, channel, uniform heat flux, Richardson number, open cavity

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1. Introduction

Recently, the study of convection heat transfer is an important topic, which is due to its great importance in many sectors. Including considerable applications in various fields such as the PV solar modules cooling due to the temperature drop that has a positive effect on energy production, as well as the cooling has a benefit in reduction of modules degradation [1, 2]. In addition, deposition of chemical vapor is some of the concerned fields [3]. In the field of scientific literature and research, many numerical and analytical approaches, as well as experimental work, take up the study of the mixed heat convection in ventilated geometries in the radiation absence [4–6]. Furthermore, big efforts are being made in studying convection heat transfer in order to save energy and money depending on many factors such as the physical geometry taken into account. Thus, the kind of geometry affects significantly the hydrodynamic and thermal distributions, as well as the heat transfer enhancements. An example of this is the heat transfer enhancement techniques corresponding to “double pipe, square duct, rhombus duct, wavy channel, flow around a hexagonal cylinder, center-trimmed twisted tape, diamond shape cylinder, corrugated tube with spring tape, inclined tabulator, and twisted tape insertion” [7–9].

Therefore, studies that are devoted to this issue neglected the problem of mixed convection heat transfer in horizontal channels and using the possibility of generating periodic flows under certain conditions in heat transfer enhancement. Indeed, buoyant flows during parallel plate channels are a significant subject that can be found in electronic cooling devices and heat sinks of micro-electronics.

2. Literature review and problem statement

Convection heat transfer inside enclosures has been taken the big effort in the literature. For instance, for the problem of mixed convection with enclosures, [10] conducted a numerical study to investigate mixed convection in a covered square cavity of various wall temperatures, in the presence of four rotating cylinders that have harmonic motion. Also, the full and harmonic rotation was compared to each other in cases of transient and steady conditions to obtain a better visualization of the harmonic rotation effect. Furthermore, this study investigated governing parameters such as solid volume fraction ($0 \leq \phi \leq 0.03$) ,Richardson number ($0.1 \leq Ri \leq 10$). Consequently, the results were presented in terms of average values of entropy generation profiles, PEC, velocity profiles, streamlines contours, isotherm contours, and Nusselt numbers. The obtained results showed that there are some different parameters such as the angular velocity of the cylinder, rotation type, and the concentration of nanoparticles that can cause a considerable improvement in the heat transfer rates. [11] reported results of another study on the effects of the presence of two oscillating fins on the nanofluid flow characteristics and the mixed convection heat transfer rates inside a square enclosure. The results displayed that the heat transfer rate can be increased substantially due to the fins oscillation. Furthermore, it was found that as the ratio of the thermal conductivity of fins relative to the nanofluid increases, there is a clear growing trend in the heat transfer rate. Indeed, the heat transfer rate was found to be enhanced by increasing the thermal conductivity ratio; however, it was shown to be weakened as the viscosity parameter increases. Besides, a numerical study was performed on the mixed convection heat transfer within a cavity of square dimensions vertically vented by [12]. The Reynolds number used was varied within the values of (50, 100, 200, and 250), and the Richardson number was varied within the values of (1, 5, and 10). The results showed that the agreement relation between the Reynolds number and Nusselt number, as well as the Richardson number value, rises at the same value of Reynolds number leading to the Nusselt number rise within the cavity. An experimental investigation of fluid flow and heat transfer in the cavity of a cubic geometry with different heated side, and large solid spheres that create a tough porous medium filling the cavity was conducted by [13]. The results of the variations of Nusselt numbers, which were calculated for different Rayleigh numbers, found that the average Nusselt number is reduced due to forming stratification layers of the spheres next to the cold or hot walls generally. These layers can change the temperature stratification and consequently cause a reduction in the overall heat transfer. [14] made a numerical investigation on the mixed convection heat transfer in a cavity of trapezoidal geometry. The cavity was insulated from the bottom and heated from the left inclined wall. The local and average rates of heat transfer, isothermal, and stream functions were compared and found that the average Nusselt number decreases with the increase of orientation angle. Another study done by [15] dealt with mixed convection flow during a rectangular enclosure with two facing identical open cubic cavities subject to separated heating. Comprehensive features of the fluid flow in the form of velocity, temperature, and vorticity were presented. The results obtained demonstrate that the aspect ratio of the cavity affects the fluid flow and temperature fields. Besides, it was found that the Prandtl, Reynolds numbers and; the channel wall heat losses have sufficient influences on the system dynamical response for heat loss parameter values. In addition, the phenomena of mixed convection heat transfer inside a channel with an open trapezoidal enclosure under the influence of different lengths of the heat source, embedded at the bottom of the enclosure, were investigated numerically by [16]. The airflow enters the channel horizontally (laminar flow) at a constant velocity and temperature. The results clarified that the isotherms distribution and the heat transfer rate depend significantly on the heat source length. Thus, it was shown that both the average and local Nusselt numbers rise with the increase in the heat source length. Moreover, the maximum temperature occurred near the heat source. Combined free and forced convection in enclosures was also studied by [17, 18].

For the mechanism of natural convection inside enclosures, [19] conducted a numerical study to test natural convection flow in a square cavity. The cold and hot walls were tilted from the vertical and rising the density uniform grids for various Rayleigh numbers. Also, the tilt angle was changed as 5 and 3 deg. with rotation of out and inside direction. The obtained results indicated that the performance increased by up to 7 %. [20] reported the mechanism of natural convection flow inside a cavity of trapezoidal geometry and partially heated. The cavity was filled with non-Newtonian Casson fluid. It was found that the velocity component of X and Y directions exhibits the strongest behavior with decreasing Casson fluid parameter. [21] studied the enclosure free convection heat transfer characteristics numerically using a rectangular model enclosure. [22] and [23] examined natural convection heat transfer within horizontal elliptic and inclined, respectively, enclosures with a circular interior heat source. [24] performed an experimental investigation of natural convection heat transfer in a concentric vertical cylinder and the effect of forced vibration on heat transfer augmentation. The findings reveal that the local heat transfer coefficient is clearly affected by the amount of heat input and axial distance of the cylinder, indicating a positive relationship with the first and an inverse relation to the latter, while an obvious increase is observed in the local Nusselt number along the cylinder axis from bottom to top. Natural convective flows in an enclosure filled with porous medium were investigated by [25–27] to show the effect of porosity on the heat transfer characteristics. Comprehensive reviews on natural convection heat transfer in enclosures and cavities have been reported in [28, 29].

Indeed, the problem of mixed convection inside channels is a very important topic, and might be distinguished in various modern feasible implementations like electronic cooling devices and heat sinks of micro-electronics. The up-to-date research on this problem has been made by [30, 31], but inside porous channels. Hence, [30] tested the impacts of the thermal boundary condition and the Darcy number on the flow and thermal behaviors. Whereas [31] investigated the aided-flow and the opposed-flow by the buoyancy in a vertical porous channel.

However, the literature reveals that there is no up-to-date work about mixed convection inside channels without porous media. Therefore, the current study is to conduct an up-to-date investigation to numerically analyze temporal mixed convective flows throughout an empty horizontal channel under the action of finite heating from the lower wall. In addition, the previous old works in this regard have not reported unsteady periodic flow behavior in the channels. Therefore, in the present study, we will use high Reynolds and Richardson numbers for an attempt to generate periodic flows.

3. The aim and objectives of the study

The aim of the present study is to examine the influence of some parameters such as Reynolds number and Richardson number on the flow and temperature distributions within the channel, as well as the rates of heat transfer. The results will be useful to show the effects of such parameters on the thermal performance of cooling applications.

To achieve this aim, the following objectives are accomplished:

– studying the influence of the Reynolds number and Richardson number on the average Nusselt number;

– studying the influence of the Reynolds number and Richardson number on the local Nusselt number along the heat source;

– studying the impact of the Reynolds number and Richardson number on the flow and thermal behavior inside the channel;

– studying the effect of the Reynolds number and Richardson number on the transient period of the temporal average Nusselt number.

4. Materials and methods

The physical problem under consideration is shown in Fig. 1. In this figure, the two-dimensional horizontal channel is heated from below at constant temperature T_h. The channel height is H and its length is $L=21H$. The flow enters the channel through the inlet port at a uniform axial velocity u_o and temperature T_o, and exits by the outlet port. The length of the heating element is assumed to be three times the channel height $S=3H$, and the remaining bottom wall of the channel is assumed to be insulated, whereas the top wall is preserved at the same inlet cooled temperature $T_w=T_o$. The computational domain is shown in Fig. 2, which indicates the mesh clustering near the walls. Transient, laminar, incompressible and mixed convective flow is assumed within the channel. Therefore, the flow field is estimated using Navier Stokes equations, which involves the Boussinesq approximation. While the temperature field is calculated using the standard energy model, where, Re, Pr, Ri are Reynolds number, Prandtl number, and Richardson number, respectively. They are expressed as follows:

$$\frac{\partial U}{\partial X}+\frac{\partial V}{\partial Y}=0, \tag{1}$$

$$\left(\frac{\partial U}{\partial t}\right)+\left(U\frac{\partial U}{\partial X}+V\frac{\partial U}{\partial Y}\right)= =\frac{1}{\mathrm{Re}}\left(\frac{\partial^2 U}{\partial X^2}+\frac{\partial^2 U}{\partial Y^2}\right)-\left(\frac{\partial P}{\partial X}\right), \tag{2}$$

$$\left(\frac{\partial V}{\partial t}\right)+\left(U\frac{\partial V}{\partial X}+V\frac{\partial V}{\partial Y}\right)= =\frac{1}{\mathrm{Re}}\left(\frac{\partial^2 V}{\partial X^2}+\frac{\partial^2 V}{\partial Y^2}\right)-\left(\frac{\partial P}{\partial Y}\right)+\mathrm{Ri}\theta, \tag{3}$$

$$\left(\frac{\partial \theta}{\partial t}\right)+\left(U\frac{\partial \theta}{\partial X}+V\frac{\partial \theta}{\partial Y}\right)= =\frac{1}{\mathrm{Re}\cdot\mathrm{Pr}}\left(\frac{\partial^2 \theta}{\partial X^2}+\frac{\partial^2 \theta}{\partial Y^2}\right), \tag{4}$$

$$\mathrm{Re}=\frac{u_o\rho_f H}{\mu_f},\quad \mathrm{Pr}=\frac{\nu}{\alpha},\quad \mathrm{Ri}=\frac{\mathrm{Gr}}{\mathrm{Re}^2}, \tag{5}$$

$$\mathrm{Gr}=\frac{g\cdot\beta_f\cdot\rho_f^2\cdot H^3\left(T_h-T_o\right)}{\mu_f^2}. \tag{6}$$

To validate the current numerical approach used, a comparison has been done between streamlines of air-jet of the result obtained by Wong and Saeid [32] with the result obtained from the current work. However, Wong and Saeid investigated numerically using the finite volume method. The mixed convection occurs under local thermal non-equilibrium conditions, from jet impingement cooling of an isothermal heated surface immersed in a confined porous channel. The characteristics of the heat transfer are found with different study parameters. Fig. 3, *a*, *b* indicates the mixed convection flow when Pe=40. However, the isotherms in Fig. 3 show the match between *a*, *b* but the overall gradients temperature along the heated portion decreases, as well as the flow velocities *u* above the wall (Y=0.002) at the first grid point are examined to investigate the reason. Certainly, the magnitude value of velocities *u* over the heated surface influences heat transfer from the heated surface source, i.e. the fluid velocity increase gives rise to heat transfer rate rise. We can conclude, when the overall gradients temperature along the heated portion decreases, the fluid velocity increase gives rise to heat transfer rate rise.

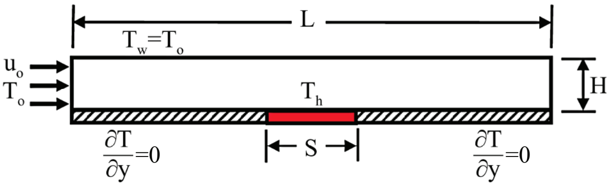


Fig. 1. Physical problem schematic diagram

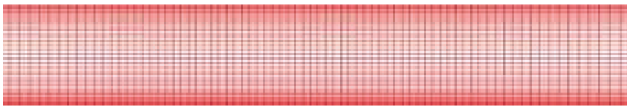
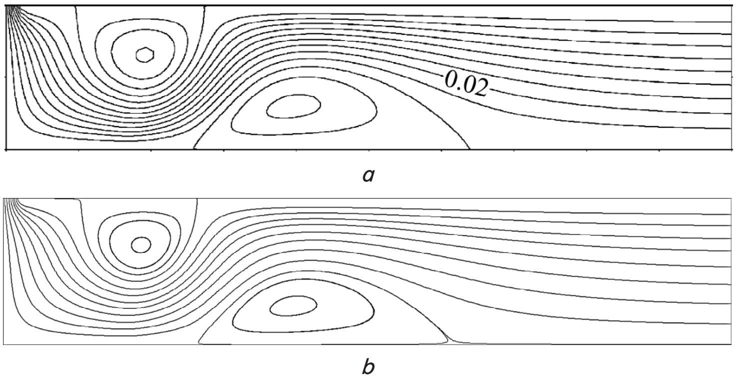


Fig. 2. Computational domain



a

b

Fig. 3. Comparison between the obtained results for streamlines of air jet impingement at Peclet number Pe=40 of: *a* – Wong and Saeid (2009) (Top); *b* – those estimated by the current code (Bottom)

5. Research results of transient mixed convection in a horizontal channel partially heated from below

5. 1. Variation of average Nusselt number

The fluid flow in the current work is considered as a laminar flow with a low range of Reynolds number of (Re≤100). Fig. 4 shows the variations of the average Nusselt number against the Reynolds number for different Richardson numbers.

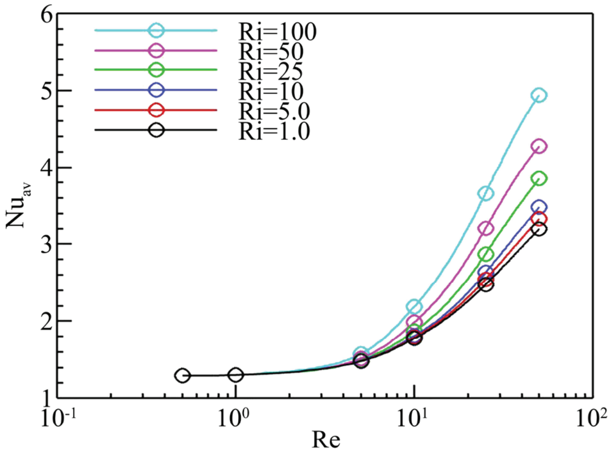


Fig. 4. Average Nusselt number variance with Reynolds number at different Richardson numbers

5. 2. Variation of local Nusselt number

Fig. 5–8 present the variations of the local Nusselt number along the heat source at different Reynolds and Richardson numbers. The Reynolds number varies over the values of 1, 10, 25 and 50 for each case study, and the Richardson number changes over the values of 1, 5, 10, 15, 20 and 25.

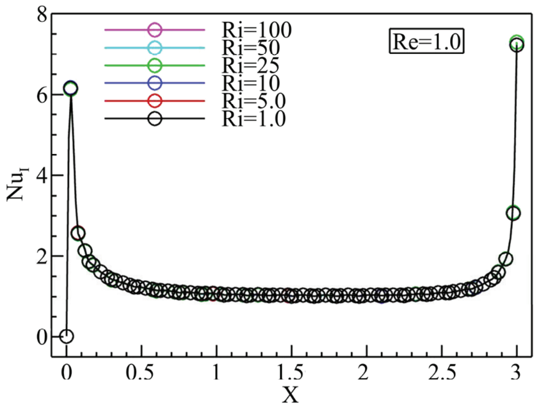


Fig. 5. Variation of local Nusselt number over the heat source at different Richardson numbers and at Re=1.0

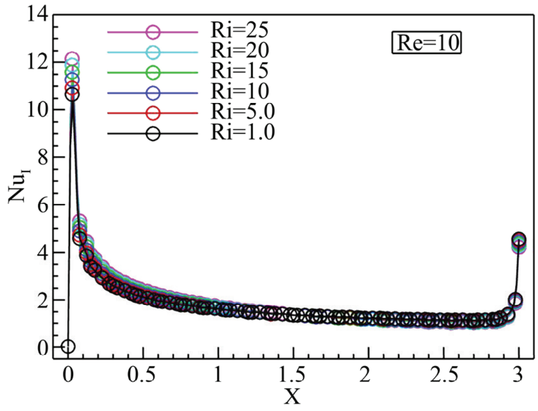


Fig. 6. Variation of local Nusselt number over the heat source at different Richardson numbers and at Re=10

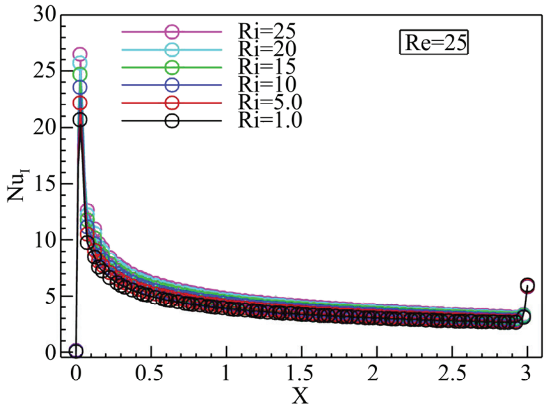


Fig. 7. Variation of local Nusselt number over the heat source at different Richardson numbers and at Re=25

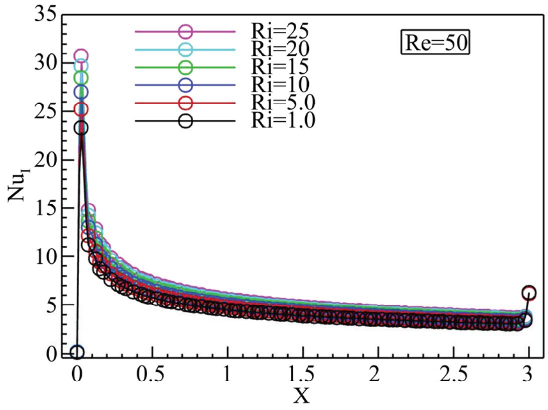
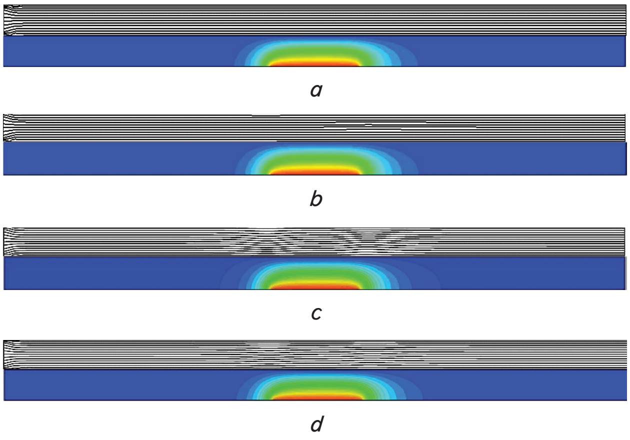


Fig. 8. Variation of local Nusselt number over the heat source at different Richardson numbers and at Re=50

5. 3. Streamlines and isotherms patterns

Fig. 9–12 demonstrate the streamlines and isotherms patterns in an empty channel heated from below at different Richardson numbers in the range of 1, 5, 10, 15, 20 and 25 at Reynolds numbers varying as 1, 10, 25 and 50 for each case study.



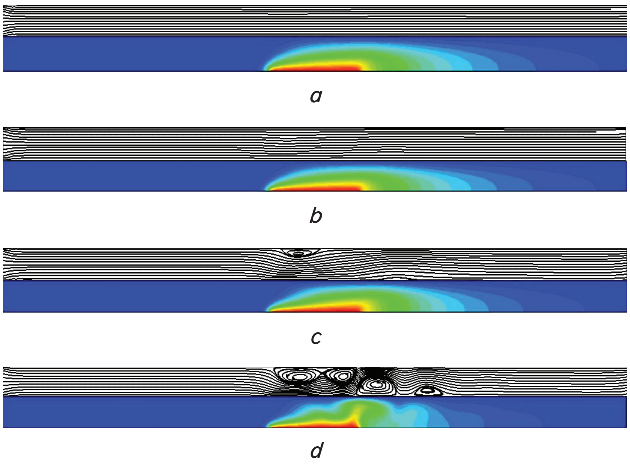
a

b

c

d

Fig. 9. Streamlines and isotherms patterns in an empty channel heated from below at Re=1 and at different Richardson numbers: *a* – Ri=1.0; *b* – Ri=10; *c* – Ri=50; *d* – Ri=100



a

b

c

d

Fig. 10. Streamlines and isotherms patterns in an empty channel heated from below at Re=10 and different Richardson numbers: *a* – Ri=1.0; *b* – Ri=10; *c* – Ri=50; *d* – Ri=100

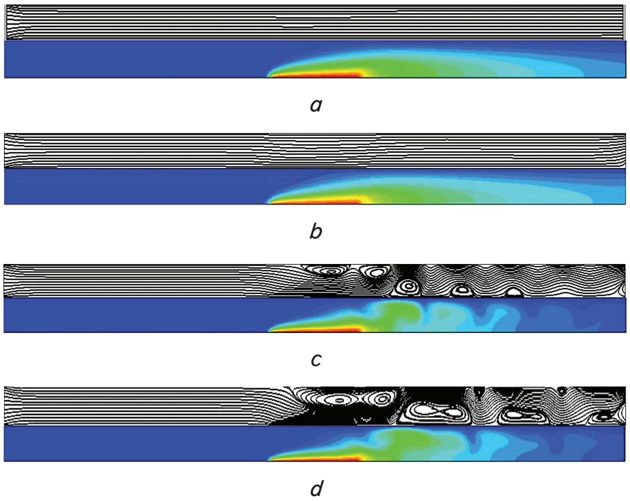


Fig. 11. Streamlines and isotherms patterns in an empty channel heated from below at Re=25 and different Richardson numbers: *a* – Ri=1.0; *b* – Ri=10; *c* – Ri=50; *d* – Ri=100

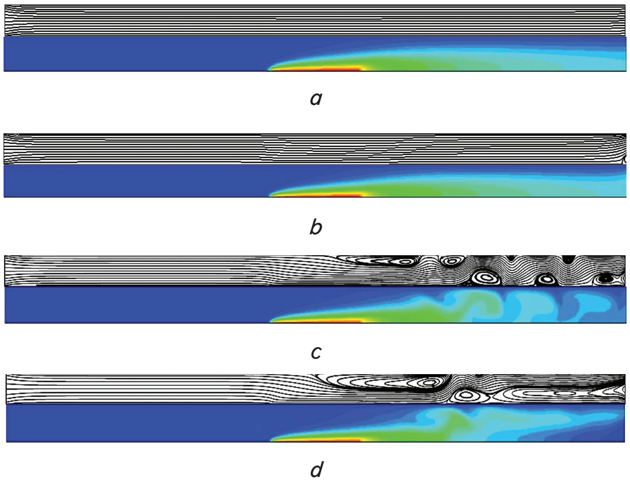


Fig. 12. Streamlines and isotherms patterns in an empty channel heated from below at Re=50 and different Richardson numbers: *a* – Ri=1.0; *b* – Ri=10; *c* – Ri=50; *d* – Ri=100

5. 4. Transient average Nusselt number

Fig. 13–16 display the transient evolution of the average Nusselt number over time for different Richardson numbers Ri=1, 10, 50 and 100 for the Reynolds number varying as 1, 10, 25 and 50.

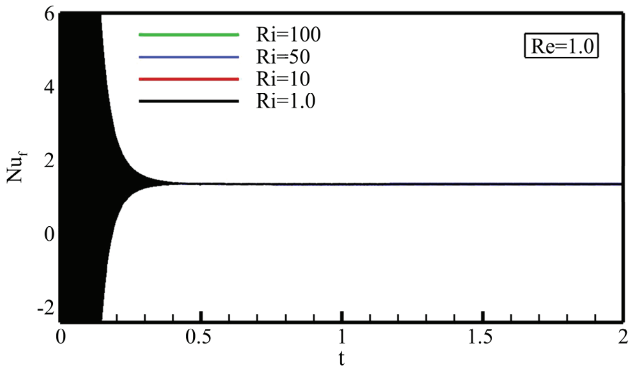


Fig. 13. Transient evolution of average Nusselt number for different Ri=1, 10, 50, 100, and at Re=1.0

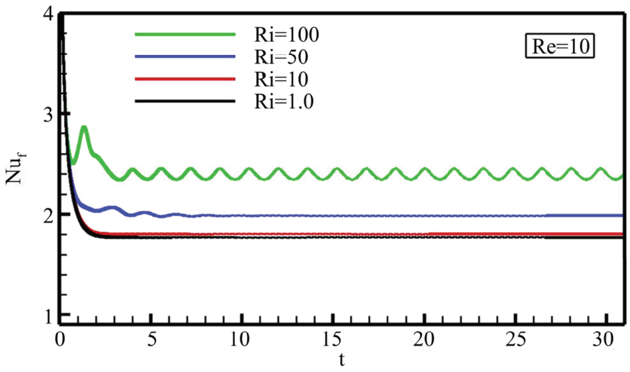


Fig. 14. Transient evolution of average Nusselt number for different Ri=1, 10, 50, 100, and at Re=10

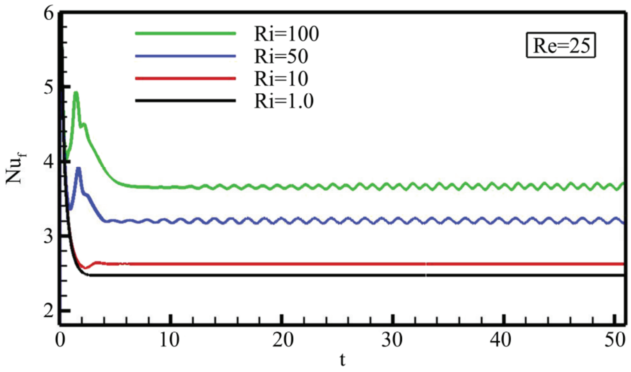


Fig. 15. Transient evolution of average Nusselt number for different Ri=1, 10, 50, 100, and at Re=25

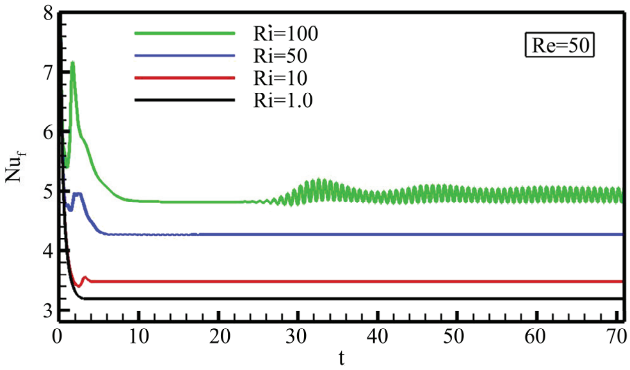


Fig. 16. Transient evolution of average Nusselt number for different Ri=1, 10, 50,100, and at Re=50

6. Discussion of the results of analysis of transient mixed convection in a horizontal channel partially heated from below

Fig. 4 demonstrates that the rates of convective heat transfer increase with the increase in Reynolds number, e.g. the flow velocity at the channel entrance. This is obvious to occur at any value of Richardson number. Also, it is demonstrated that the profiles of the Nusselt number increase exponentially with the Reynolds number for each Richardson number. Moreover, it is shown that at a low Reynolds number, there is no effect of the Richardson number on the Nusselt number; however, at a high Reynolds number, the Nusselt number increases as the Richardson number increases.

The results presented in Fig. 5–8 indicate that the Nusselt number increases with an increase in Richardson number. This explains the convection heat transfer rise due to shifting from natural convection to the forced convection. In addition, the highest value of convective heat transfer is shown to be at the beginning point of heating, which is illustrated by the value of the Nusselt number at this heating point. Also, the value of the maximum Nusselt number increases with the increase in Reynolds number. Thus, at the location near the beginning of the heat source, the maximum values of the Nusselt number are 6, 12, 27 and 31, for Reynolds numbers of 1, 10, 25 and 50, respectively. After that, the trend of the Nusselt number gradually decreases along the horizontal heated element until reaching the endpoint of it, where the heat transfer increases due to the sudden change in the temperatures. It can be shown that the values of the local Nusselt number are not affected by the Richardson number at a low Reynolds number, for example Re=1.0. However, the values of the local Nusselt number start increasing with the Richardson number as the Reynolds number increases.

Fig. 9–12 illustrate the streamlines and isotherms patterns in case of the very low value of Reynolds number have a strong thermal stratification. This indicates very low convective heat transfer with all values of the Richardson number (natural convection). Yet, the convection heat transfer increases as a result of the increase in Reynolds number. Also, the increase in Richardson numbers leads to a change in fluid layers' shapes that belong to forced convection.

The Nusselt number profiles start with a strong unsteady behavior, decrease over time, and become steady after a period of time (Fig. 13–16). In Fig. 13, the Nusselt number reaches the steady state after a very short time t=0.5. However, as the Reynolds number increases, approaching the steady state needs longer time, as shown in Fig. 14 for Re=10. Therefore, it is obvious that the decrease in the Richardson number leads to a decrease of the Nusselt number for the same value of the Reynolds number and for the same time interval. Also, importantly, the figures report that at higher Reynold numbers, increasing the Richardson numbers generates oscillations in the trend of Nusselt number. This means that fluctuating flows appear inside the channel.

The limitations of the present study are as follows: the code solves 2D geometries, we solved the problem of a single heat source, and the code becomes unsteady for very high Reynolds numbers. We have a plan in the future to develop the code to do 3D simulations, and for multi-heat sources. Also, the code needs to be improved numerically to be stable and solve simulations for very high Reynolds numbers.

7. Conclusions

1. The convection heat transfer represented by the Nusselt number increases with the increase in Reynolds number, and the trends of the Nusselt number profiles are shown to increase exponentially with Reynolds number for each Richardson number. There is no effect of the Richardson number on the Nusselt number for the low Reynolds number; however, for the high Reynolds number, the Nusselt number increases as the Richardson number increases.

2. The highest local Nusselt numbers are at the beginning and end points of the heat source; however, the trend of the local Nusselt number along the heat source is gradually minimally decreased.

3. Steady flow and thermal fields are seen in the channel for low Reynolds and Richardson numbers; however, for the higher Reynolds number, increasing the heating effect (Richardson number) generates unsteady flow behavior, which affects the isotherms patterns inside the channel.

4. When the unsteady flow is generated, the trends of the Nusselt number begin to oscillate with time, indicating to the periodic flow. As the Reynolds and/or Richardson number increases, the transient time of reaching the steady state or the perfect periodic state increases.

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