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 Reynolds number for each Richardson number and the local Nusselt number (Nu) 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

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 ≤ φ ≤ 0.03), Richardson number (0 ≤ Ri ≤ 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 substanc-
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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 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 \( u_0 \) 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 calculated 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:

\[
\begin{align*}
\frac{\partial U}{\partial X} + \frac{\partial V}{\partial Y} &= 0, \\
\frac{\partial U}{\partial t} + \left( U \frac{\partial U}{\partial X} + V \frac{\partial U}{\partial Y} \right) &= \frac{1}{Re} \left( \frac{\partial^2 U}{\partial X^2} + \frac{\partial^2 U}{\partial Y^2} \right) - \frac{\partial P}{\partial X}, \\
\frac{\partial V}{\partial t} + \left( U \frac{\partial V}{\partial X} + V \frac{\partial V}{\partial Y} \right) &= \frac{1}{Re} \left( \frac{\partial^2 V}{\partial X^2} + \frac{\partial^2 V}{\partial Y^2} \right) - \frac{\partial P}{\partial Y} + \frac{\partial \theta}{\partial Y}, \\
\frac{\partial \theta}{\partial t} + \left( U \frac{\partial \theta}{\partial X} + V \frac{\partial \theta}{\partial Y} \right) &= \frac{1}{Re} \frac{\partial^2 \theta}{\partial X^2} + \frac{\partial \theta}{\partial Y}, \\
Re &= \frac{u_o H}{\mu}, \quad Pr = \frac{\nu}{\alpha}, \quad Ri = \frac{Gr}{Re^2}, \\
Gr &= g \beta \rho_f \cdot H \cdot (T_h - T_o) / \mu^2. 
\end{align*}
\]

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.

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.
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.

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.
5.4 Transient average Nusselt number

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