

Development of a Convective Regime Within a Closed System

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Article Info

Article history:

Received , 02/11/2025
Revised , 05/05/2026
Accepted , 11/05/2026

Keywords:

Numerical investigation;
Rectangular chamber ;
Non-uniform grid ;
Fluid circulation ;
Heat transition.

ABSTRACT

In the current work, a numerical investigation of Newtonian fluid flow within a rectangular chamber partially heated from the bottom with T_h and having movable side lids, which are kept at a cold temperature T_c is conducted. The partial differential equations have been discretized employing a numerical approach based on the finite difference method, considering a non-uniform grid system. The numerical calculations are carried out for the Reynolds number ($Re=100$). The resulting flow patterns, temperature eddies, and heat transition have been illustrated for relevant values of the Richardson parameter between 12.4 and 12.6. It is also detecting that the mixed convection effects have a very important role in the loss of symmetry for the internal fluid circulation.

I. Introduction

The energy transmission inside cavities with different boundary conditions has increasingly been reported in several industrial sectors, such as the electrical field [1], nuclear reactor and oil extraction [2] and energy transition over the photovoltaic systems [3] ... etc. This topic has reported in the literature. A. Jobby et al. [4] have done a review on natural convection in rectangular geometries with various internal objects. They observed that the internal objects' inclination and the difference in their heating prevent the heat transmission and the convective behaviour. Dalila et al. [5-8] have investigated mixed convection across a saturated porous square cavity for high porosity. They also have noted the occurrence of instability of the flow structure for non-Newtonian fluid when Darcy's value is equal to 0.1 [5]. However, they addressed the influence of the boundary conditions on the internal fluid flow in the work [6], and through their contribution [7]; they have carried out a numerical simulation of laminar regime flow in a 2D cavity. Their findings show that the fluid motion results from the complex interaction within the geometric configuration. Additionally, they have examined the combined effects of the heat source and the cooling air in [8]. They concluded that an unsteady state streamlines and temperature contours inside a porous material. Many works in the literature have focused on hybrid systems. For example, Al-Kaby et al [9] have presented the influence of porosity on the heat transfer of a nano-fluid within a 2D cavity. They noted that the rate of heat inside the process has been increasing with lower values of porosity. Meria et al. [10] have investigated the effect of porous medium on the PV system energy. The obtained results reveal that the porous material led to an improvement the energy efficiency of the process from 10% to 28%.

A literature review, relevant to the subject of the present study, can be summarised in experimental and numerical studies taking into account the different parameters' effect on the energy efficiency and fluid motion. Our problem presents a new proposal to the work of Aydin and Yang. [11], which appears in considering an upward movement of the left and right lids of the chamber. Our particular objective to highlight the impact of the Richardson number on the mixed flow behaviour within the proposed model.

II. Geometry of the Model

We consider a square cavity as shown in Fig. 1, containing a fluid initially at rest. The lower part is subjected to uniform heating, while the vertical walls are cooled by a circulated air at a constant velocity (V_0). The other parts are considered adiabatic. We assume a laminar and two-dimensional flow.

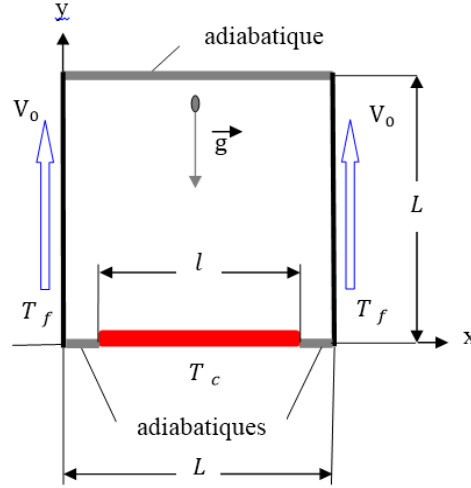


Figure 1. Cavity diagram of the studied problem

III. The governing equations

A. Continuity equation

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \quad (1)$$

B. Equation of motion according to x

$$\frac{\partial u}{\partial t} + u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} = -\frac{1}{\rho} \frac{\partial P^*}{\partial x} + \nu \left[\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} \right] \quad (2)$$

C. Equation of motion according to y

$$\frac{\partial v}{\partial t} + u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} = -\frac{1}{\rho} \frac{\partial P}{\partial y} + \nu \left[\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} \right] + g\beta(T - T_f) \quad (3)$$

D. Equation of energy

$$\frac{\partial T}{\partial t} + u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} = \lambda \left[\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} \right] \quad (4)$$

E. Equation of vorticity

$$\frac{\partial \omega}{\partial t} + u \frac{\partial \omega}{\partial x} + v \frac{\partial \omega}{\partial y} = \nu \left[\left(\frac{\partial^2 \omega}{\partial x^2} \right) + \left(\frac{\partial^2 \omega}{\partial y^2} \right) \right] + g\beta \left(\frac{\partial \theta}{\partial x} \right) \quad (5)$$

F. Flow function equation

$$\frac{\partial^2 \psi}{\partial x^2} + \frac{\partial^2 \psi}{\partial y^2} = -\omega \quad (6)$$

$$\omega = \frac{\partial v}{\partial x} - \frac{\partial u}{\partial y} \quad (7)$$

Velocity components

$$u = \frac{\partial \psi}{\partial y}, \quad v = -\frac{\partial \psi}{\partial x} \quad (8)$$

The average Nusselt number along the bottom:

$$Num = -\frac{1}{\varepsilon} \int_0^\varepsilon \frac{\partial \theta}{\partial Y} dX \quad (9)$$

IV. Numerical resolution method

The discretization of the time term has been performed using the fourth-order Runge-Kutta method. The convective terms have been resolved by the third upwind scheme [12]. The fourth-order scheme is employed to discretize the velocity equations. The iterative method of non-linear over relaxation has been employed to resolve the obtained stream function equation. A code in the Fortran language has been developed to solve the considered model equations.

V. Grid size chosen

Four different non-uniform grid sizes, namely 61x61, 81x81, 101x101 and 121x121, were taken into consideration in order to verify the independence of the solution grid. Table I shows the corresponding tests, which present the maximum stream function for the independence of the third grid (101x101), which was chosen to accomplish the conducted simulations, because it provides a good compromise between the duration of the calculation time and the precision of these tests.

Table 1. Maximum flow function for various mesh sizes

Grid size	Max of flow function
61x61	0.1677
81x81	0.1674
101x101	0.1672
121x121	0.1672

VI. Tests for validation

The numerical code was validated by taking into account the variation of the average Nusselt as a function of Richardson number from the obtained results of Aydin and Yang [11] in the case of a rectangular enclosure containing air (Fig. 2). The comparison presents an excellent agreement. The dimensionless transformation makes it possible to specify the flow conditions with a restricted number of parameters in order to make the solution more general and speed up calculations.

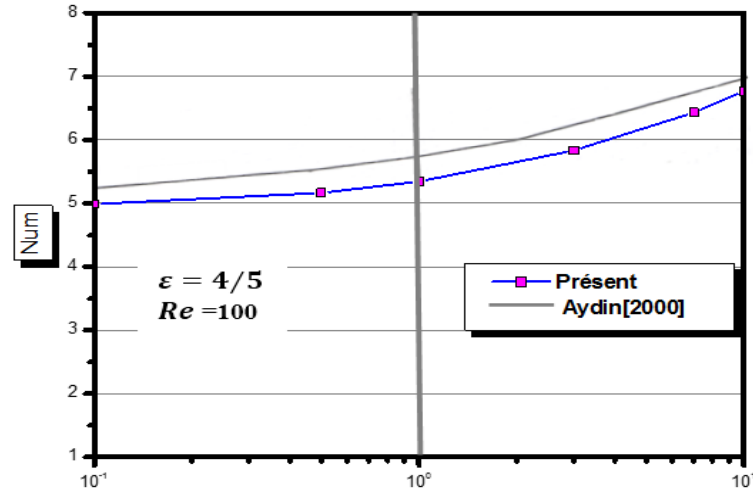


Figure 2. Average Nusselt profile between the present study and that of Aydin. [11]

VII. Obtained results and Discussions

In all cases of numerical tests, both Reynolds and Prandtl numbers are assumed constant, their values are set respectively at 100 and 0.7. Richardson value ranges from 12.4 and 12.6. Temperatures and fluid dynamics curves, as well as thermal transmission for the heated part were analysed. The obtained results allowed us to detect three structures, all of which are different. Figures 3, 4, and 5 clearly illustrate their respective arrangements.

VII.1. Isotherms

The temperature contours are shown in Figure 3. It appears from this figure that the temperature distribution inside the chamber is characterized mainly by the loss of symmetry. High temperatures occupy about two-thirds of the studied domain. It should be noted that practically all the heat recovered from the hot source is evacuated either through the right part for $Ri=12.4$, or through the left part when the value of this number increases only by 0.1. Demonstrating here the loss of symmetry of the flow. This fact is due to the interaction between the forces induced by the hot source and those caused by moving lids.

If we increase the working parameter towards the value 12.6, the heat recovered from the hot part is transported up in the middle of the enclosure by the pair of cells in the center. This is what explains the relatively high temperatures in the central part of the considered model. The heat is evacuated equally through both vertical walls.

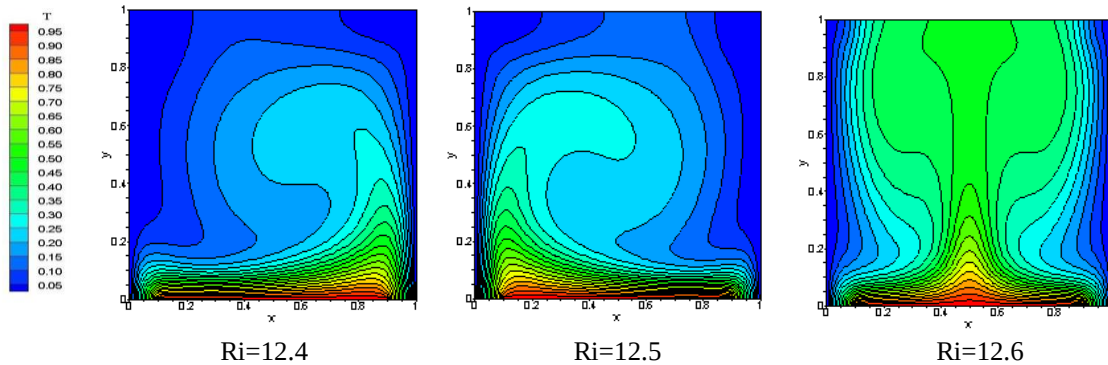


Figure 3. Temperature contours for $Ri=12.4$, 12.5 and 12.6 respectively

VII.2. Flow contours

The air circulation inside the model studied is illustrated in Figure 4. It's characterized by two counter-rotating cells, but non-symmetric for the Richardson values equivalent to 12.4 and 12.5. Top from there, when $Ri=12.6$; it's noted that two new identical vortices that are generated by the heat source appeared, accompanied by two main others due to forced convection, which are retersted towards the two vertical parts of the cavity.

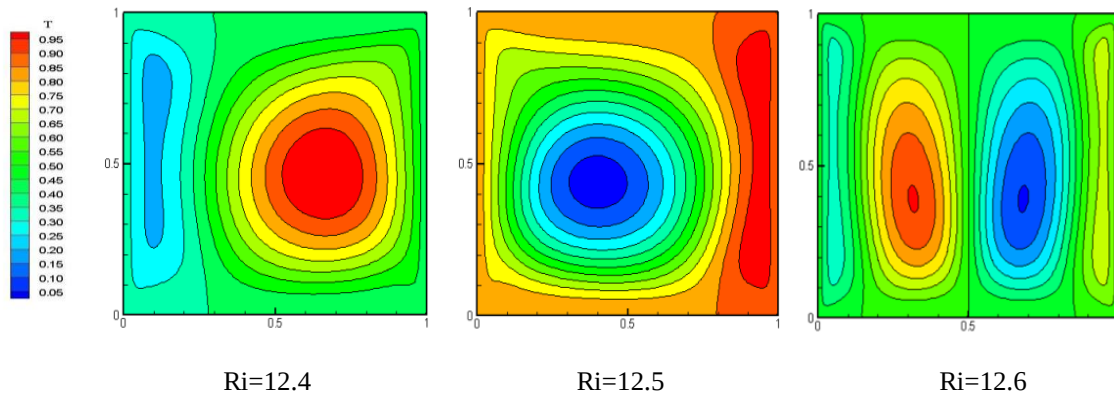
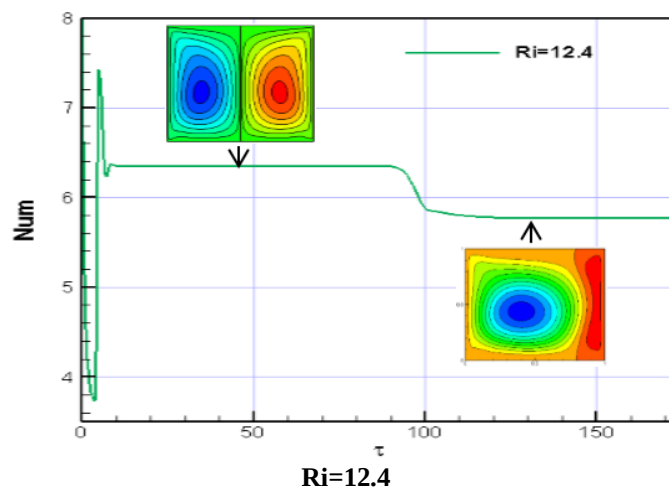


Figure 4. Flow contours for $Ri=12.4$, 12.5 and 12.6 respectively

VII.3. Heat transmission rate

The temporal evolution of the average Nusselt number is shown in Figure 5. The appearance of this number represents remarkable fluctuations during the first moments, when the Richardson parameter takes the values of 12.4 and 12.5. Then it presents a sudden peak in order to stabilize at a constant value. Putting here a regime formed of two strictly identical cell.

When the value of this parameter becomes equal to 12.6, the temporal evolution of mean heat transmission rate shows significant fluctuations of varying magnitudes, which fade after a certain period, to then stabilize at a fixed value. Bifurcation towards this new regime is easier to obtained, since the time required for each simulation is significantly less.



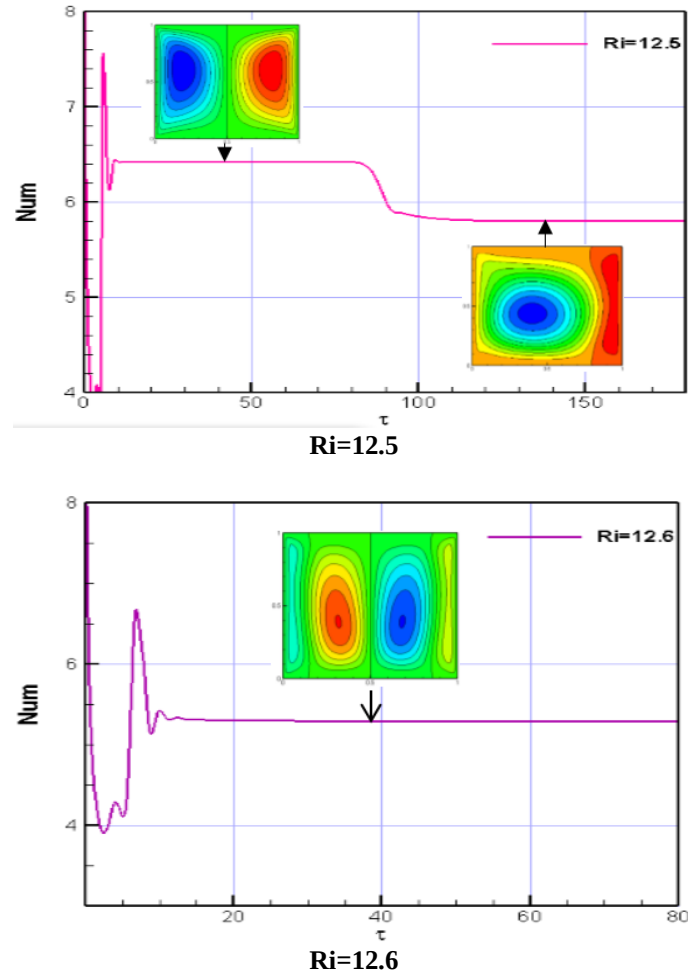


Figure 5. Heat transmission rate corresponding to the considered Richardson values

VIII. Conclusion

This study constitutes a contribution in numerical solutions of energy transport from a heat source in a confined and externally cooled environment. This mixed flow led us to study the characterization of the developed structure of fluid circulation and its temperature, as well as the evolution of the heat transmission rate, taking into account the importance of free convection compared to forced convection.

From obtained results, it is also indicates that:

- The critical value for the appearance of symmetry loss from the Richardson value is equal to 12.4.
- The transition from an antisymmetric flow (Ri=12.5), to a new regime formed by two symmetrical pairs of cells at (Ri=12.6) indicates the presence of an unsteady state of the air movement inside the cavity.

Nomenclature

g	Acceleration due to gravity (ms^{-2})
Gr	Grashof number, $Gr = g\beta (T_h - T_c) L^3 / \nu^2$
l	Heat source length (m)
L	Cavity height (m)
Nu	Number of Nusselt
Num	Average Nusselt number
Pr	Prandtl number, $Pr = \nu / \alpha_e$

Re	Parameter of Reynolds, $Re = V_0 L / \nu$
Ri	Number of Richardson, $Ri = Gr / Re^2$
u, v	Dimensionless velocity components
V_0	Side walls velocity (ms^{-1})
x, y	Dimensionless coordinates

Greek

ν	Cinematic viscosity ($Pa\ s^{-1}$)
θ	Temperature non-dimensional
τ	Dimensionless time
ψ	stream function non-dimensional
ω	Dimensionless vorticity

Subscripts

c	Refers to cold temperature
h	Refers to hot temperature

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