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CONJUGATE NATURAL CONVECTION IN A POROUS ENCLOSURE SANDWICHED BY FINITE WALLS AND SUBJECTED TO CONVECTION COOLING CONDITION

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Abstract: Steady conjugate natural convection heat transfer in a two-dimensional enclosure filled with fluid saturated porous medium is studied numerically. The two vertical boundaries of the enclosure are kept isothermally at same temperature, the horizontal upper wall is adiabatic, and the horizontal lower wall is partially heated. The Darcy extended Brinkman Forcheimer model is used as the momentum equation and Ansys Fluent software is utilized to solve the governing equations. Rayleigh number ($1.38 \leq Ra \leq 2.32$), Darcy number ($3.9 * 10^{-8}$), the ratio of conjugate wall thickness to its height ($0.025 \leq W \leq 0.1$) , heater length to the bottom wall ratio ($1/4 \leq \varepsilon \leq 3/4$) and inclination angle (0°, 30° and 60°) are the main considered parameters. The presented results show the effect of these parameters on the heat transfer and fluid flow characteristics. These results include streamlines, isotherm patterns, and local and average Nusselt number for different values of the governing parameters. It is found that either increasing the Rayleigh number and the ratio of conjugate wall thickness to its height (d/H) or decreasing the ratio of heat source width to bottom wall (l/L), the average Nusselt number is increased. Also, it was observed that the average Nusselt number does not change substantially with inclination angle.

Keywords: *Heat transfer, Natural convection, Porous media, Conjugate, CFD.*

1. Introduction

During the past few decades, the heat transfer in the systems including fluid-saturated porous media has received significant attention because of the varied range of geophysical and engineering applications. These include great enactment isolation for structures, grain storage, energy effective freshening methods, spreading of pollutants, etc.. Descriptive reviews of these applications and others convective heat transfer applications in porous medium may be found in the latest books by [1-5].The natural convection in porous media has been studied widely using various models; see, for example, [6-15]. The influence of the conjugate solid walls confined the porous media is received comparatively fewer attention. Solid–porous layer in conjugate natural convection–conduction heat transfer can be found in many engineering applications.

Nield and Kuznetsov [16] analyzed forced convection in a flat channel filled with a saturated porous medium. Their results show that the effect of the limited thermal resistance due to the panels is to reduce both the heat transfer to the porous medium and the degree of local thermal imbalance. Then, Nield and Kuznetsov [17] extended their investigation [16] for forced convection in a flat channel filled with a saturated bi-disperse porous medium, as well as conduction in flat plates surrounding the channel using a two-speed, two-heat model. The results were presented to show the reliance of Nusselt number on Biot number associated with the boundary plate, the heat exchange parameter between the phase, the ratio of the thermal conductivity between the phase, the ratio of the effective permeability between the phase, and the ratio of the macroscopic vacuum.

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Al-Amiri et al. [18] investigated numerically the wall heat conduction effect on the natural-convection heat transfer within a two dimensional cavity, which is filled with a fluid-saturated porous medium. Their problem configuration consists of two insulated horizontal walls of finite thickness and two vertical walls which are maintained at constant but different temperatures. Their results showed that as the wall thickness increases, the temperature difference between the interface temperature and the cold boundary reduces, which accordingly brings about a reduction in the overall Nusselt number. Besides, Al-Farhany et al. [19] studied the influence of Non-Darcy on natural convection conjugate double-diffusive in a variable porous layer bounded by finite thickness conjugate partitions. They applied constant temperature for heating and cooling along the vertical sidewalls of the enclosure while the horizontal two walls were insulated. While, Zargartalebi et. al. [20] studied unsteady natural convection in a porous cavity boarded by two conjugate vertical walls. Their results show that, the amplitude of periodic fluid Nusselt number is increasing as a function of non-dimensional frequency and wall thickness. Additionaly, Mehryan et al., [21] examined the conjugate natural convection by using local thermal non-equilibrium style inside a porous square enclosure filled with micropolar nanofluid. Their results show that the Nusselt number became completely certified and vortex-free at a value higher and lower than the Riley number, respectively. Also, Bondareva et al. [22] studied numerical investigation of heating conduction wall of finite thickness and conductivity for natural convection of a rectangular enclosure filled with alumina-water nanofluid with heat boundary condition from the left wall. They observed that that for the studied models; the ratio of effective thermal conductivity, dynamic viscosity, the increase in the size of nanoparticles reduces heat transfer low fluid flow rate.

In the other hand, Gao et. al., [23] they suggested Boltzmann's modified capillary model for heat transfer for simulating conjugate heat transfer in porous media. Their modified LB model has been well validated by simulating four different conjugate heat transfer problems, steady state conjugate heat conduction within two-layer solid medium, transient conjugate heat conduction in infinite composite solid, porous/fluid conjugate heat transfer and porous/solid conjugate heat transfer. The numerical results predicted by their model agree well with the analytical and numerical solutions reported in previous studies. Chen [24] Simulated the heat transfer associated with a fluid-saturated porous medium and conjugate finite wall thickness and proposed a new numerical approach, based on the lattice Boltzmann (LB) method. He ensured that the accuracy, simplicity and stability of his current model are better. Hu et. al., [25] studied double diffusive convection in partially filled with a porous layer with spatially uniform internal heat generation and a solid layer in a square enclosure. In the present paper buoyancy driven convection heat transfer in a confined fluid saturated porous media sandwiched between two solid vertical sidewalls is investigated numerically. These vertical walls are set at constant temperature while the top wall is thermally insulated and the bottom wall is partially heated. The variation of several parameters and their effect on the heat and flow in the enclosure are studied. According to the authors' knowledge and the survey made for this case no recent references that study the present heat transfer problem and the parameters variation examined in this paper were found.

2. Modelling and CFD Simulation

2.1. Computational Condition

The geometry of present study is a square cavity occupied by the porous media of uniform porosity and permeability; air saturated with Oberbeck-Boussinesq equation. The two-dimensional square cavity under investigation is with dimensions of (L × H) in

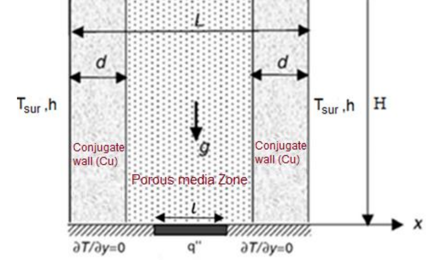

Figure 1. Schematic diagram for the model

2.2. Governing Equation

In order to simplify the problem, the following assumptions were applied to governing equations [4].

- Steady, laminar, and all physical properties for solid and convective fluid are constant.
- The convective fluid and the solid material are in local thermal equilibrium in all places.
- The convective fluid remains in a single phase.
- Chemical reactions, compression work, inertia effects, viscous dissipation, and thermal dispersion in the system are negligible.
- Homogenous isotropic media.
- No internal heat generation.
- Incompressible fluid (due to low velocity flow).
- Forchheimer – Brinkman - extended Darcy model is applied.
- The effect of pressure due to body force on the density of fluid is negligible .
- In considering the body force term, the density of fluid is assumed a linear function of temperature and concentration, i.e., the effect of pressure on the density of the fluid is negligible.
- $\rho = \rho_0[1 - \beta(T - T_0)]$.

By considering the assumptions mentioned above, the free convection heat transfer process in porous medium is governed by the basic conservation principles of mass, momentum (Forchheimer – Brinkman - extended Darcy model), and energy equations. Therefore, the general governing equations are given as [26]:

Continuity Equation:

$$\nabla \cdot \vec{v} = 0 \qquad (1)$$

Momentum Equation:

$$\rho_f \left[\frac{1}{\varepsilon}\nabla\left(\frac{\vec{v}.\vec{v}}{\varepsilon}\right)\right] = -\nabla P + \frac{\mu}{\varepsilon}\nabla^2\vec{v} - \frac{\mu}{K}\vec{v} - \frac{C_F\rho_f}{K^{1/2}} \qquad (2)$$

Energy Equation:

$$(\rho c)_f\, \vec{v}\cdot\nabla T = \nabla\cdot\left(k_{eff}\nabla T\right) \qquad (3)$$

Where

$$k_{eff} = (1-\varepsilon)k_s + \varepsilon k_f \qquad (4)$$

Meanwhile, the transport process within the wall can be represented as:

$$\nabla^2 T = 0 \qquad (5)$$

The associated boundary conditions for the problem under consideration can be expressed as:

$$x = 0 \text{ and } 0 \le y \le H: u = 0, T = T_c \qquad (6)$$

$$x = L \text{ and } 0 \le y \le H: u = 0, T = T_c \qquad (7)$$

$$y = 0: v = 0 \text{ and } \frac{\partial T}{\partial y} = \begin{cases} 0 & for \quad 0 \le x \le d \\ \frac{q''}{k_{eff}} & for \quad d \le x \le L-d \\ 0 & for \quad L-d \le x \le L \end{cases} \qquad (8)$$

$$y = 1 \text{ and } 0 \le x \le L: v = \frac{\partial T}{\partial y} = 0 \qquad (9)$$

At the interface ($x = d$ and $x = L - d$), the following condition can be applied :

$$k_{eff}\left.\frac{\partial T}{\partial x}\right)_{porous} = k_s\left.\frac{\partial T}{\partial x}\right)_{solid} \qquad (10)$$

The physical quantities of interest in this investigation are the Darcy number ($Da = K/L^2$), the Rayleigh number ($Ra = g\beta Kq''L/\alpha_{eff}\nu$), the local Nusselt number and the average Nusselt number that are defined respectively as:

$$x = d: Nu = \frac{k_s}{k_{eff}}\left.\frac{\partial T}{\partial x}\right)_{solid} \qquad (11)$$

$$x = L-d: Nu = \frac{k_s}{k_{eff}}\left.\frac{\partial T}{\partial x}\right)_{solid} \qquad (12)$$

$$Nu_{ave} = \int_0^H Nu\, dy \qquad (13)$$

2.3. Mesh Generation

Standard CFD methods require a mesh that fits the boundaries of the computational domain and the nonlinearity of the geometry due to the difference in sizes and dimensions. For the present case, the type of mesh suggested is structured quadrilaterals for straight wall domain used in the enclosure. In this work, mesh independency test was performed in which several sizes of mesh were tested to display the effect of meshing size on the average Nusselt number in the steady-state solution. Fig. (2) Shows that better range of elements can be achieved when the mesh size was between 2500 to 3000 elements, with an average skewness of cells 0.1 to 0.4.

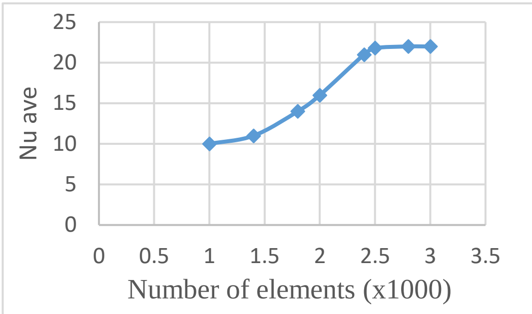

Figure 2. Mesh independency test

3. Validation for Convection Heat Transfer

The numerical results of natural convection heat transfer streamlines and isotherm for the present work is compared with the results of [26] as shown in Fig. (3). Apparently, the results from both works are in good agreement. For Model 1 in both cases, streamlines regions were observed in the whole enclosure are two regions one is with clockwise (the right swirl) and anticlockwise (the left swirl). While for Model 2, from the temperature contour it appears that there is gradient in the temperature since the heat source position in the middle of the bottom wall for the two studies.

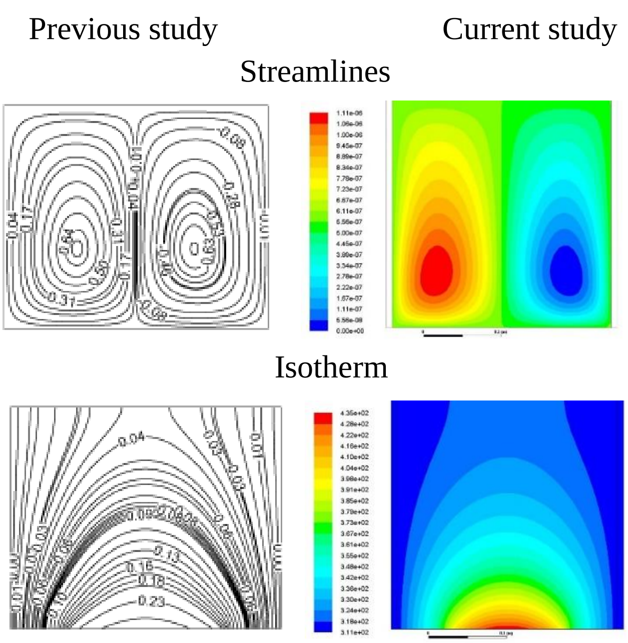

Figure 3. Streamline and Isotherm for the present work and for the Current Study) [26].

4. Results and Discussion

In the present work, free convection heat transfer is investigated numerically, by ANSYS 17.1 software, for laminar flow through porous media in cubic cavity heated partially with constant heat flux at the middle of the bottom wall, the working fluid is selected as air. The values of Rayleigh number, Ra, are taken to be 1.38, 1.68, 2.02, and 2.32. The ratio of conjugate wall thickness to its height values, W, are 0.025, 0.075, and 0.1, The ratio of heater width to the bottom wall ratio, ε , are 1/4 ,1/2 and 3/4 ,finally the inclination angle values, γ, are 0°, 30°, and 60°.

4.1. Effect of Conjugate wall thickness to the its height ratio (W)

W is defined as the conjugate wall thickness to its height W=d/H. The effect of conjugate wall thickness on the streamlines and isotherms within the cavity is showed in Fig. 4. The following parameters were fixed to muteness their effects: Ra =2.32, and ε= 1/2. The flow circulation in the porous medium is divided into two equal vortices, the right vortex is with clockwise and the left vortex is with counterclockwise with a flow upward along the center line of the enclosure since the heater in the center of the bottom wall and downward at the right and left cold walls. Further, two vortices appear as the main characteristic of the flow for all the considered values of the dimensionless conjugate wall thickness. From isothermal contours, we can discover that the heat transfer system manifests itself in the conjugate wall while the contribution of convection to energy transfer in general is live in the porous cavity. When the conjugate wall thickness increases, the strength of the circulation within the porous region increase. This is because, with the increase in conjugate wall thickness, the effective temperature difference that sets up the flow in the porous medium, increase and the flow strength also increase. This can be noticed by the change in the isotherms strengths within the porous medium with the increase in the wall thickness. Apparently, such a trend impacts the average Nusselt number predictions which tend to rise with the increase in conjugate wall thickness (the value of the average Nusselt number is between parentheses for each case in Fig. 4.

This is attributed to the fact that, with an increase in the conjugate wall thickness, the temperature difference between the solid–porous interface and the cold boundary (i.e., right and left walls) increase as illustrated in Fig. 5 and, thus, increasing the magnitude of the average Nusselt number as shown in Fig. 6 . It is worth mentioning that the temperature at the interface decrease vertically along the wall and this is associated with a fact that the clockwise and anticlockwise rotating fluid carries energy from the center of the enclosure to the right and left walls, consequently, becomes hot as it rises up against the bottom wall. On the opposing, as the fluid flows downward along the right and left walls, its temperature increase gradually which is reflected on the interface temperature at Y = 0

W = 0.025 (Nu_{ave} = 5.9041) W = 0.075 (Nu_{ave} =6.2755) W = 0.1 (Nu_{ave} =7.3446)

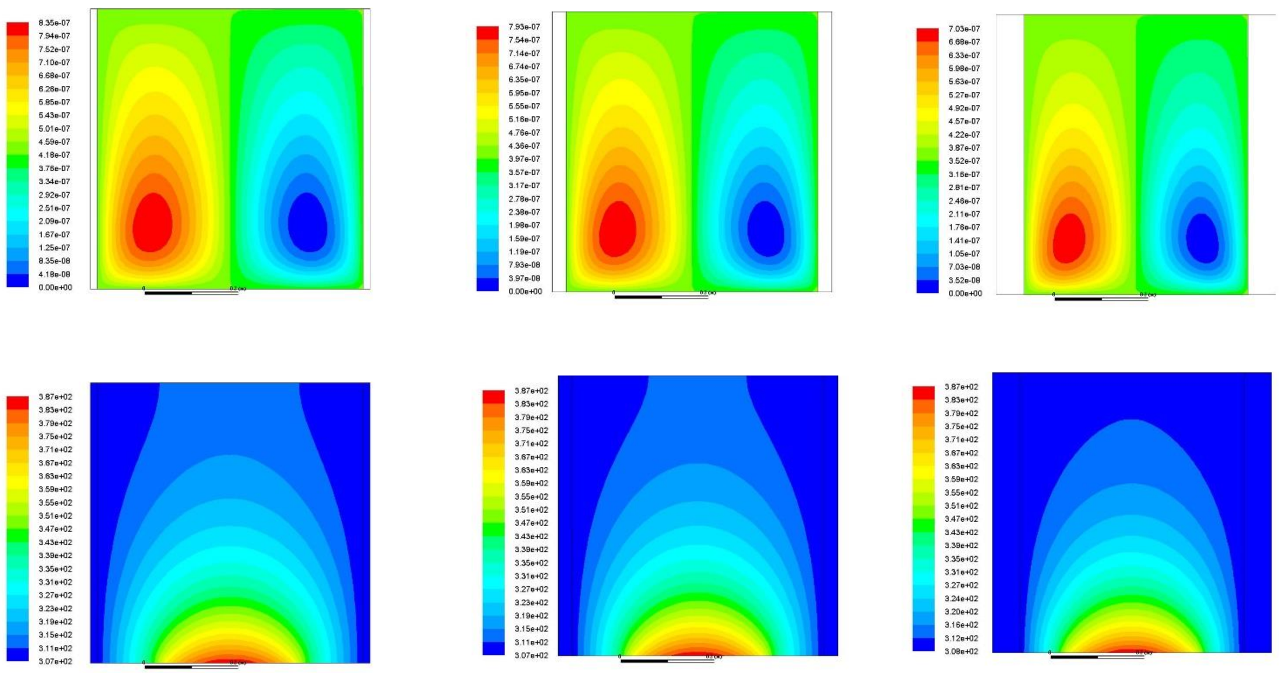

Figure 4. Effect of varying wall thickness on the streamlines and isotherms using ε =1/2 and Ra =2.32.

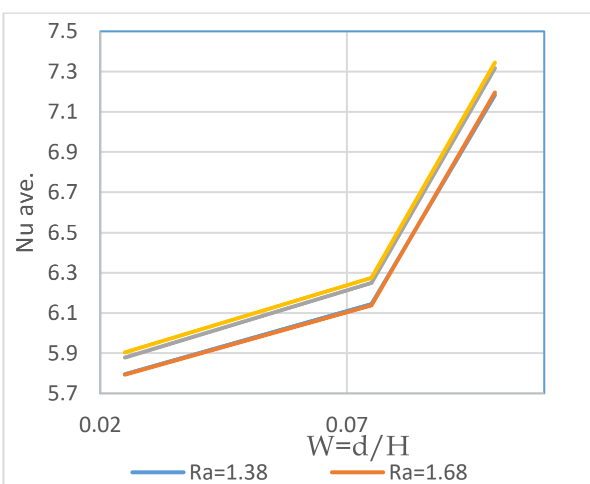

Figure 5. Temperature distribution at the bottom wall–porous interface for different wall thickness using ε =1/2 and Ra = 2.32.

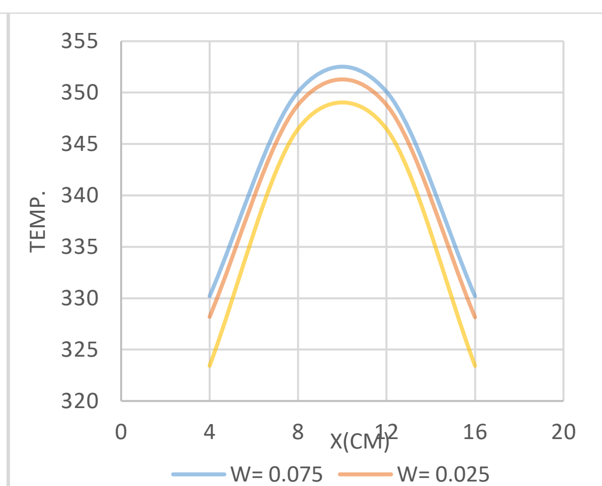

Figure 6. Effect of conjugate wall thickness on the average Nusselt number using ε =1/2 with different Ra and W.

4.2. Effect of heater length to the bottom wall ratio (ε)

The dimensionless heater width is defined as ε = l/L.The effect of heater width on the fluid motion and isotherms within the cavity is depicted in Fig. 7. This is endorsed to the fact that, with an increase in the heater width with presence conjugate wall, the temperature difference between the solid–porous interface and the cold boundary (i.e., right and left walls) increase as illustrated in Fig.8 due to the fact that increasing heater width for the same Rayleigh number, the strength of the circulation within the porous region decrease. The effective temperature difference that sets up the flow in the porous medium, decrease and the flow strength also decrease and, thus, decreasing the magnitude of the average Nusselt number as shown in Fig. 9.

ε =1/4 (Nu_{ave} = 11.7901) ε =1/2 (Nu_{ave} =6.2958) ε = 3/4 (Nu_{ave} =4.87926)

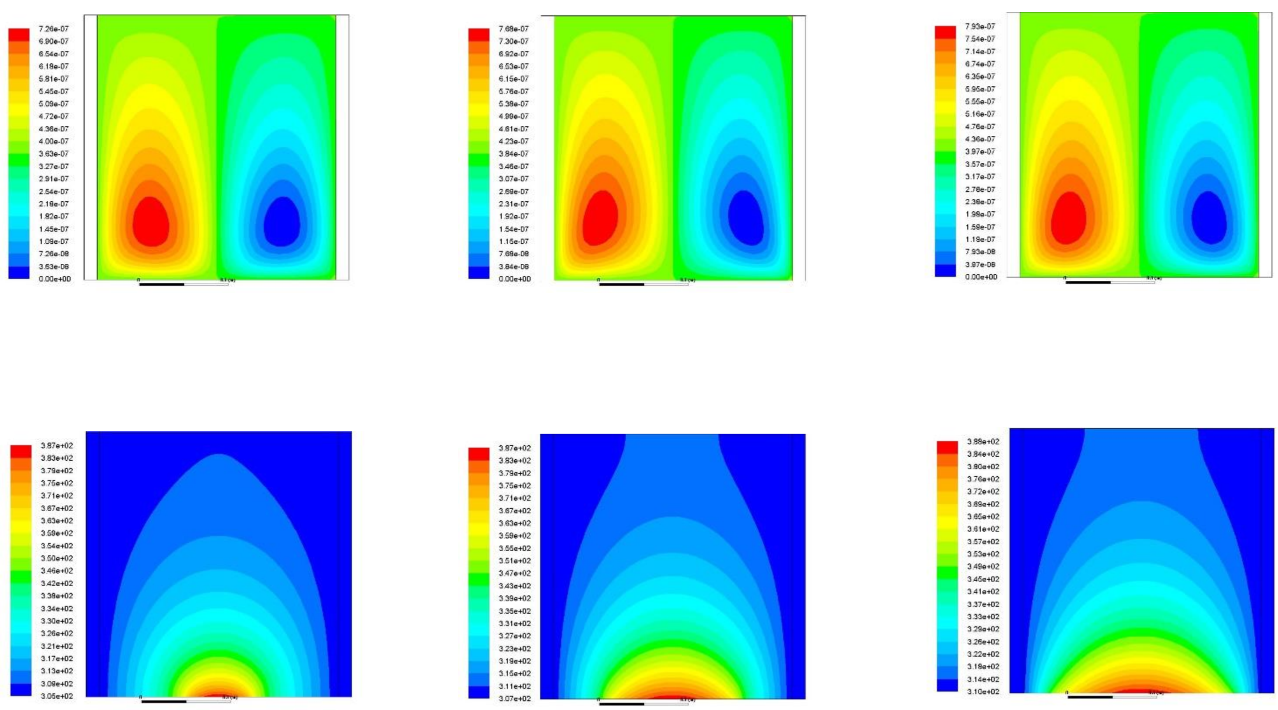

Figure 7. Effect of varying heater width on the streamlines and isotherms using W=0.075 and Ra =2.32

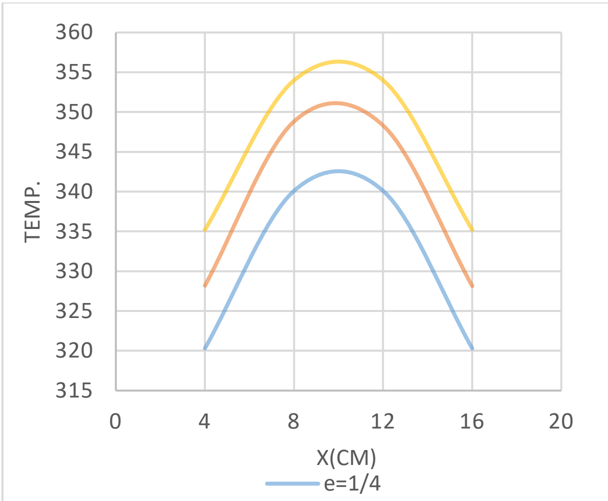

Figure 8. Temperature distribution at the bottom wall–porous interface for different wall thickness using W=0.075 and Ra =2.32

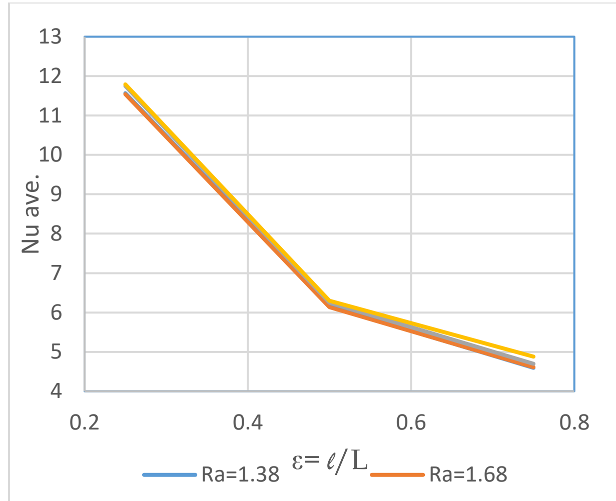

Figure 9. Effect of heater length on the average Nusselt number using w= 0.075 with different Ra and ε.

4.3. Effect of inclination angle (γ)

The evolution of the flow and thermal fields in the cavity with increasing inclination are shown in Fig. 10 for a representative cases of different inclination angles and fixed the Ra = 3.23 , W = 0.075 and ε = 1/2. For $\gamma = 0°$,The flow circulation in the porous medium is divided into two equal vortices, the right vortex with clockwise and the left vortex with counterclockwise with a flow upward along the center line of the enclosure since the heater in the center of the bottom wall and downward at the right and left cold walls. Further, a central vortices appear as the characteristic of the flow for $\gamma = 0°$, when increase the inclination angle to $\gamma = 30°$ its observed that the left recirculating vortex becomes dominating in the cavity while the right vortex is squeezed thinner and move to the downward right corner, while for $\gamma = 60°$ the minor vortex will disappear and the enclosure will including only the big vortex due to the change in the heat source location.

The isotherms are still the same without any adjustment indicating the presence of a small temperature gradient there. In addition, it was observed that the left side of the enclosure temperature decreased while it was increased in the right side as shown in Fig. 11. this difference happen due to the inclination adjusted anticlockwise ,so the left side will be lower than the heater , as result the hot air move upward to the right side. The variation of the Average Nusselt number, along the inclination angles for different Ra number is shown in Fig. 12. It was observed that average Nusselt number increase with increasing the inclination angle in sensible way.

ϒ= 0° (Nu $_{ave}$ = 11.7901) ϒ= 30° (Nu $_{ave}$ =6.2958) ϒ= 60° (Nu $_{ave}$ =4.87926)

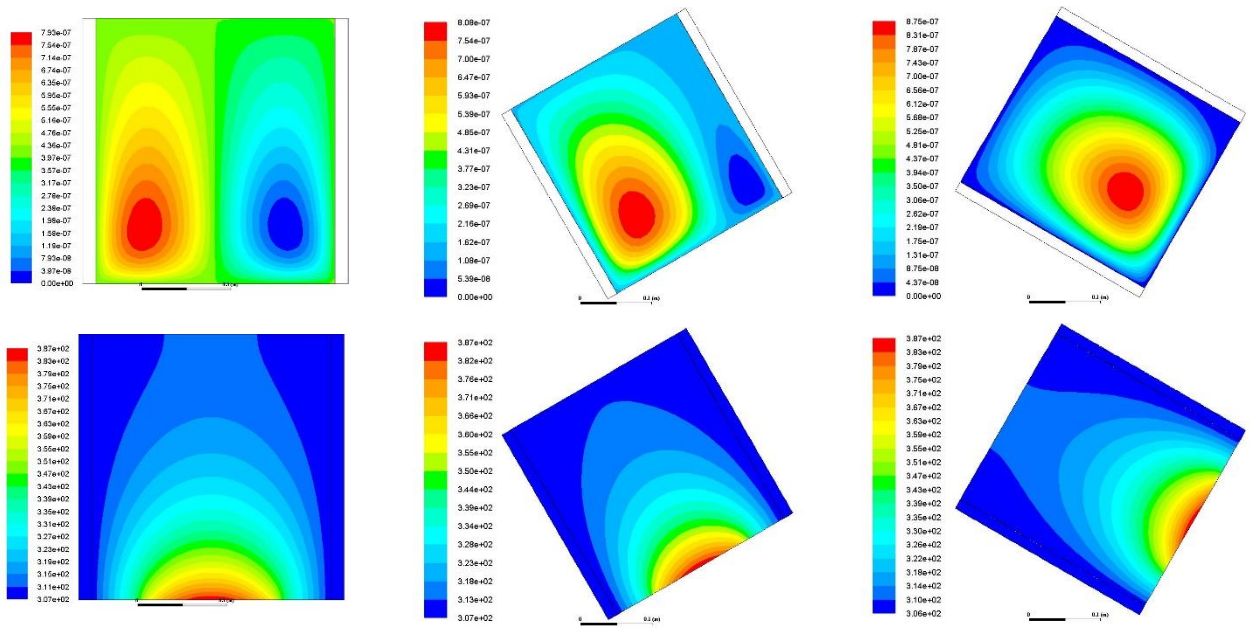

Figure 10. Effect of varying inclination angle on the streamlines and isotherms using W=0.075 and Ra =2.32

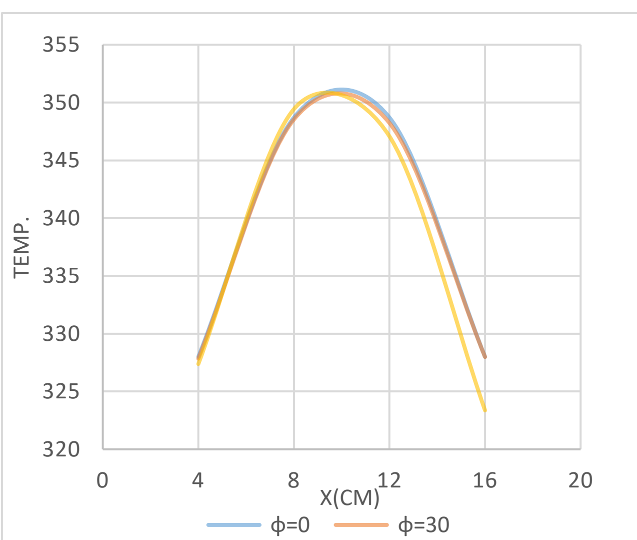

Figure 11. Temperature distribution at the bottom wall–porous interface for different wall thickness using W=0.075 and Ra =2.32

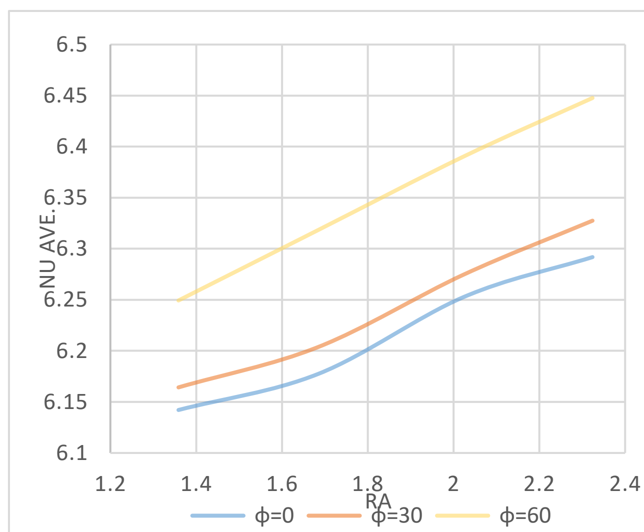

Figure 12. Effect of inclination angle on the average Nusselt number using w= 0.075 with different Ra and ϒ.

4.4. Effect of Rayleigh Number

The effect of Rayleigh number on the streamlines and isotherms is shown in Fig. 13 using w=0.075 and ε= 1/2 . For low, Rayleigh number values, the streamlines are characterized by two vortices with that occupies the entire cavity body. When increasing the Rayleigh number it mean increase the power input to the enclosure this will increase the temperature as shown in the Fig 14 for the same heater width and conjugate wall thickness because of the porous media and low Darcy number the effect will not be sensible and the average Nusselt number will increase also as per the equation Nu =Ra*Pr*Da

Ra =1.38 (Nu_{ave}=6.137) Ra =1.68 (Nu_{ave}=6.14) Ra =2.02 (Nu_{ave}=6.248) Ra =2.32 (Nu_{ave}=6.275)

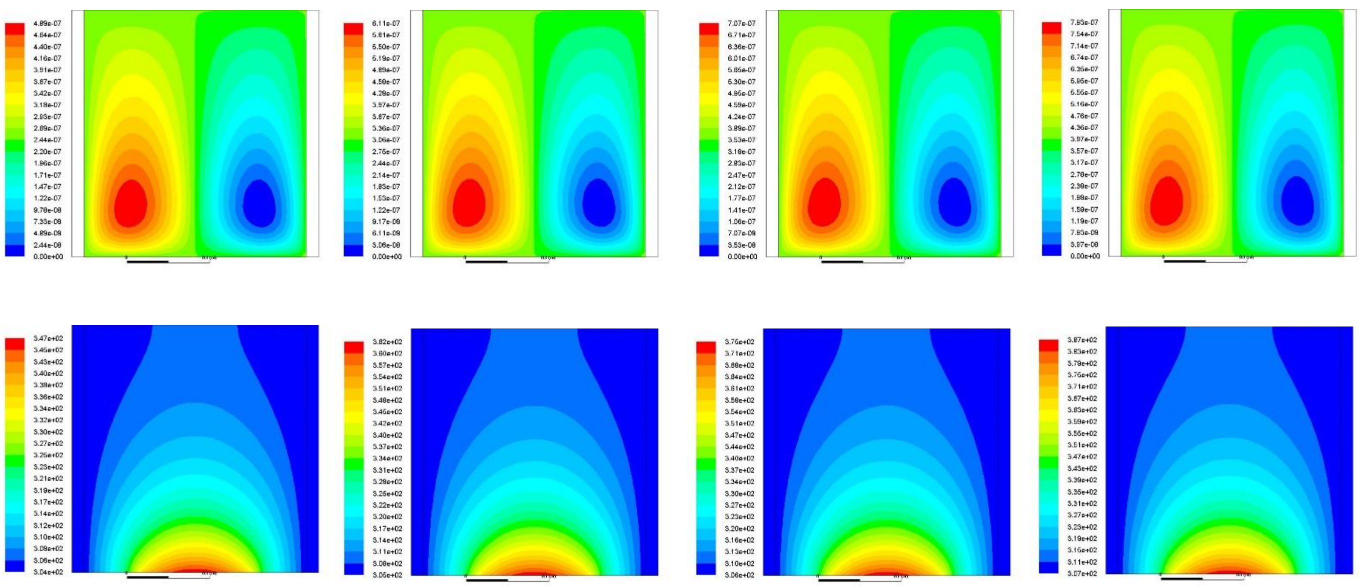

Figure 13. Effect of varying Ra number on the streamlines and isotherms using W=0.075 and ε =1/2.

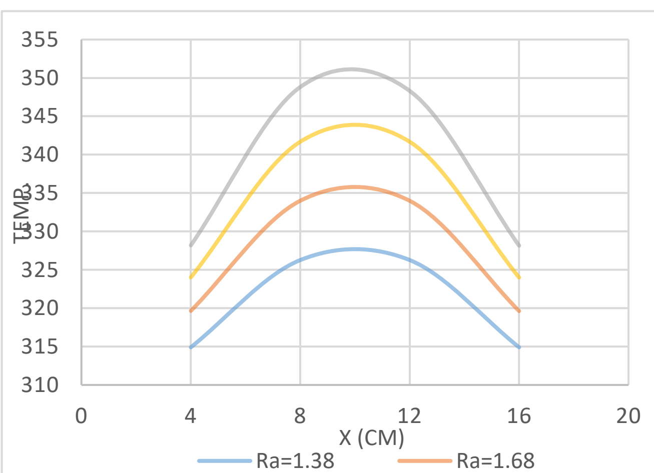

Figure 14. Temperature distribution at the bottom wall–porous interface for Ra number using W=0.075 and ε =1/2.

5. Conclusion

The main conclusions that can be drawn from the present investigation that includes a numerical analysis based on ANSYS Workbench (flaunt) 17.1 results can be summarized as follows:

1. It is observed that for enclosures with two conjugate walls subjected to constant heat flux boundary condition; two symmetrical cells are formed into the cavity and in different directions. The negative sign of stream function denotes the clockwise circulation. In addition, from the streamline schemes, streamlines become unsymmetrical and the streamline values of which with the clockwise direction decrease while they increase for the counterclockwise direction.

2. As the angle of inclination of the enclosure increases, the streamlines become one cell and the direction of the streamline is counterclockwise. In addition, the streamline values keep increasing with the increase of the inclination angle.
3. The conjugate walls thickness play an important role; when increasing the walls thickness the heat transfer will increase proportionally.

Nomenclature

c	$J/kg.℃$	Fluid specific heat
C_F	-	Form-drag constant
d	m	Solid wall thickness
g	m/s^2	Gravitational Acceleration
H	m	Enclosure height
k_{eff}	$W/m \cdot ℃$	Effective thermal conductivity
k_f	$W/m \cdot ℃$	Fluid thermal conductivity
k_s	$W/m \cdot ℃$	Solid thermal conductivity
K	m^2	Permeability
L	m	Enclosure width
P	N/m^2	Pressure
q''	W/m^2	Heat flux
T	℃	Temperature
T_0	℃	Reference temperature
T_c	℃	Cold temperature
u	m/s	Velocity component in x -direction
v	m/s	Velocity component in y -direction
$\vec{v}$	m/s	Velocity vector
x	m	Dimensional length of x -axis
y	m	Dimensional length of y -axis
α_{eff}	m^2/s	Effective thermal diffusivity
β	$1/K$	Volumetric Expansion Coefficient
ε	-	Porosity
μ	$kg/m.s$	Dynamic viscosity
ρ	kg/m^3	Density
ρ_o	kg/m^3	Density at reference temperature
ρ_f	kg/m^3	Fluid density
v	m^2/s	Kinematic viscosity

Conflict of interest

There are not conflicts to declare.

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