Numerical Investigation of Natural Convection Heat Transfer in Partially Filled Porous Enclosure Subjected to Constant Heat Flux

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ABSTRACT

Steady natural convection in a square enclosure with wall length (L= 20 cm) partially filled by saturated porous medium with same fluid (lower layer) and air (upper layer) is investigated. The conceptual study of the achievements of the heat transfer is performed under effects of bottom heating by constant heat flux ($q=150,300,450,600 W/m^2$) for three heaters size (0.2,0.14,0.07)m with symmetrically cooling with constant temperature on two vertical walls and adiabatic top wall. The relevant filled studied parameters are four different porous medium heights (Hp=0.25L,0.5L, 0.75L, L), Darcey number (Da1) 3.025×10^{-8} and (Da2) 8.852×10^{-4}) and Rayleigh number range (60.354 - 241.41), ($1.304\times10^6 - 5.2166\times10^6$) for Da1 and Da2 cases respectively. Numerically, COMSOL Multiphysics 5.5a® based on the Galerkin finite element method is used for solving the governing equations with depending Brinkman- Darcy extended mode for porous media region. The results show that, effects of increasing the Rayleigh number on the temperature profile besides the progressively increasing the average Nusselt number. Moreover, symmetrical distribution of local Nu along the bottom heated wall and it is be minimum at midpoint of bottom. Also, the heat transfer and fluid flow are affected by thickness of porous layer and are maximum at porous layer thickness (0.25L) which clearly observed with large heater size to be approximately (93%) for the average Nu. Generally, the heat transfer is enhanced for large Darcy number (8.852×10^{-4}) and influenced by the convection regime improvement while it is mainly conduction mode for (Da1) for all Raleigh number with a little effect of convection when increase (Ra).

KEYWORDS

Natural Convection, porous media, heat flux, partially filled, Brinkman- extended Darcy, enclosure, numerical.

INTRODUCTION

Natural convection in a cavity saturated or partially filled with porous media is still one of the researcher's interest studying because it's a distinguishing tool to augment the heat transfer for some natural and engineering applications[1-5], also it is used as exquisite isolation for other applications[6]. The motion of fluid due to difference in density resulting from temperature gradients is the main rule of natural convection [7-9].The term (natural) comes from the absence of foreign effects such as mixer machine, compressor, pump, etc. The effect of a porous medium in natural convection has received widespread attention, especially in recent time. So, a group of researchers studied cases of fulling the cavity [10-18],some of these works are investigated enhancement the heat transfer by effect of discrete heating[12, 19]. Another investigators analysis of same cases but for tilted rectangular enclosure [20-22]. They showed enhancements(held) of heat transfer when increasing in Rayleigh number which evident by augmentation in Nusselt number. This behavior is also observed with increasing the inclination angle Mansour et al [23]. Kalaoka et al [24] showed there is an increase in the intensity of fluid motion when increasing Darcy and Grashof number.

Also, heat convection inside cavity partially filled by porous medium was studied [25-35]. Numerical and experimental works have been studied the effects of different parameters that included height of porous layer inside the cavity [36-38]. They showed that inverse relation between height ratio and average heat transfer, same with Darcy number which similar action in uniform heater base. The natural convection was investigated when there were bodies inside the cavity, with different parameters [39-44]. Recently, studies have developed in convection, where [45-47] has worked to improve heat transfer inside the cavity with different thickness of porous layers. The current study illustrates the steady natural convection resulting in cavity heated below by constant heat

flux and symmetrically cooling under an unchanged temperature at vertical side walls and insulating the top wall. It consists of single horizontal layer of porous medium (glass bead) and upper layer of free air.

METHODOLOGY

The physical geometry of the 2-D natural convection model in a square cavity. The enclosure is filled partially fluid-saturated porous medium (glass beads) with free air at remaining part, is shown in figure (1). The top wall is insulated and it is cooled at vertical sidewalls by constant temperature (Tc)and heated bottom wall with constant heat flux (q). The Boussinesq's approximation has been depended at the density of fluid and the Darcy-Brinkman model for flow inside the porous domain is used also neglecting the permeable interface with walls of the enclosure.

The governing equations are given as [48, 49]:

Continuity equation

A- For air region

$$\frac{\partial u_{air}}{\partial x} + \frac{\partial v_{air}}{\partial y} = 0 \tag{1}$$

B- For porous/air region

$$\frac{\partial u_p}{\partial x} + \frac{\partial v_p}{\partial y} = 0 \tag{2}$$

Momentum equation

A-For air region

$$\left[u_{air} \frac{\partial u_{air}}{\partial x} + v_{air} \frac{\partial u_{air}}{\partial y} \right] = -\frac{1}{\rho_{air}} \frac{\partial p}{\partial x} + \frac{\mu_{air}}{\rho_{air}} \left(\frac{\partial^2 u_{air}}{\partial x^2} + \frac{\partial^2 u_{air}}{\partial y^2} \right) \tag{3}$$

$$\left[u_{air} \frac{\partial v_{air}}{\partial x} + v_{air} \frac{\partial v_{air}}{\partial y} \right] = -\frac{1}{\rho_{air}} \frac{\partial p}{\partial y} + \frac{\mu_{air}}{\rho_{air}} \left(\frac{\partial^2 v_{air}}{\partial x^2} + \frac{\partial^2 v_{air}}{\partial y^2} \right) + \beta_{air} g (T - T_\circ) \tag{4}$$

B-For porous/air region

$$\frac{1}{\varepsilon^2} \left[u_p \frac{\partial u_p}{\partial x} + v_p \frac{\partial u_p}{\partial y} \right] = -\frac{1}{\rho_{air}} \frac{\partial p}{\partial x} + \frac{\mu_{air}}{\varepsilon \, \rho_{air}} \left(\frac{\partial^2 u_p}{\partial x^2} + \frac{\partial^2 u_p}{\partial y^2} \right) - \frac{\mu_{air} u_p}{\rho_{air} K_p} \tag{5}$$

$$\frac{1}{\varepsilon^2} \left[u_p \frac{\partial v_p}{\partial x} + v_p \frac{\partial v_p}{\partial y} \right] = -\frac{1}{\rho_{air}} \frac{\partial p}{\partial y} + \frac{\mu_{air}}{\varepsilon \, \rho_{air}} \left(\frac{\partial^2 v_p}{\partial x^2} + \frac{\partial^2 v_p}{\partial y^2} \right) - \frac{\mu_{air} v_p}{\rho_{air} k_p} + \beta_{air} g (T - T_\circ) \tag{6}$$

Energy equation

A-For air region

$$u_{air} \frac{\partial T}{\partial x} + v_{air} \frac{\partial T}{\partial y} = \alpha_{air} \left(\frac{\partial^2 T}{\partial x} + \frac{\partial^2 T}{\partial y} \right) \tag{7}$$

B-For porous/air region

$$u_p \frac{\partial T}{\partial x} + v_p \frac{\partial T}{\partial y} = \alpha_{eff.} \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} \right) \tag{8}$$

Eeffective thermal diffisivety is defind by

$$\alpha_{eff.} = \frac{K_{eff.}}{(\rho c_p)_{fluid}} \tag{9}$$

Where the effective conductivety ($k_{eff.}$)

$$k_{eff.} = \varepsilon (k_f) + (1 - \varepsilon) k_s \tag{10}$$

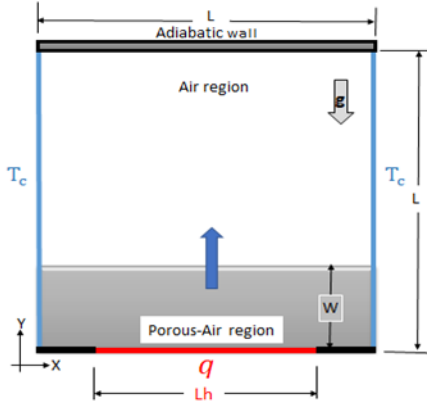(k_f, k_s):fluid and solid conductivey respectively

Figure 1. The physical Geometry of the problem

Enclosure boundary conditions

The boundary conditions which is used in the present work as follows;

Cold Vertical Sides

$$x(0, L), 0 \leq y \leq L], T = T_c, u = 0, v = 0 \tag{13}$$

Top Side Wall

$$[x(0, L), y(L)], \frac{\partial T}{\partial y} = 0, \ u = 0, v = 0 \tag{14}$$

Bottom Side Wall

$$[\frac{(L-L_H)}{2} \leq x \leq \frac{(L+L_H)}{2}, \text{y (0)}] \text{ q.= h (ΔT), where}(L_H) \text{ is the heater length}$$

$$q_{conduction} = q_{convection} = -k_f \frac{\partial T}{\partial y} = h(T - T_\circ), u = 0, v = 0 \tag{15}$$

At Interface of Fluid -Porous Domain

$$k_f \frac{\partial T_f}{\partial y} = k_s \frac{\partial T_p}{\partial y} \quad \text{and } T_f = T_s \tag{16}$$

Elsewhere $\frac{\partial T}{\partial y}|_{y=0} = 0$

The local Nusselt (Nu_l) and average $(Nu_{average})$ along the heat flux side of the enclosure are used as shown below:

$$Nu_l = \frac{heat\ transfer\ by\ covection}{heattransfer\ by\ conduction}, \ Nu_l = -(\frac{K_s}{K_{eff.}})\frac{\partial T}{\partial y} \tag{11}$$

$$Nu_{average} = \frac{1}{L}\int_0^L Nu\ dy$$

Grid Independence Test (GIT)

The main aim of reducing processing time and cost for the PC with maintaining the high quality is GIT. Therefore, several attempts were done to choose the appropriate mesh size which gives accurate results and also a treatment for mesh of enclosure boundaries and in common area of pure fluid domain and porous layers was taken place. GIT has been proceeded for both types of porous media Da1 and Da2 respectively as figure (2) which shows the change in average Nusselt number along the bottom hot wall for all predefined mesh sizes. The parameters of GIT were heat flux (450 Watt/m²), horizontal porous layer thickness (0.1m), Darcy number (Da1 and Da2). Obviously, the convenience mesh for current cases was (Extra Fine Mesh) which has number of elements 17442 with good

quality (0.84/1) as compared with Extremely fine grid of (25770 elements). So, the extra fine grid has been adopted for present investigation with error nearly (10^{-5}).
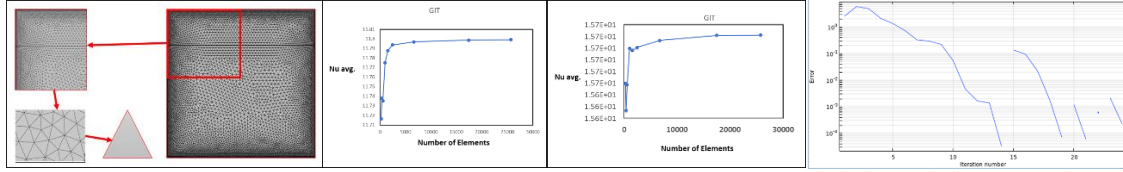

Figure 2. Triangular Mesh Distribution, Average Nusselt Number Various Number of Elements for Case of Small Porosity (0.418) and (0.568) Respectively, and Error Percentage for Convergence.

Solution procedure and Validation

To solve the partial differential equations, COMSOL Multiphysics 5.5a®-CFD has been used in the present work due to the accuracy and effectiveness of the program for solving scientific and engineering problems. The governing equations are solved and the results are shown as the stream function, isotherms, local, and average Nusselt numbers (Nu). The error plots convergence criterion for each case was less than 10^{-5} due to the execute of the required iteration number to solve the cases figure (2). To make sure the program is working properly for studying and investigating the current work, some results have been compared with published approved researched papers in different cases and models for isotherms and streamlines. Figure (3.a,b,c and d) showed the comparison of the present model with Orhan Aydin et, al [9] , Sathiyamoorthy et al [10] and[30] Chen, Yu et al.2009 for enclosure filled with free fluid only, porous media and free fluid with porous layers respectively and (d)comparing with porous medium heated by constant heat flux[12](Noor et, al 2020)at(middle)and [19](Sivasankaran et.al. 2011) at right column.
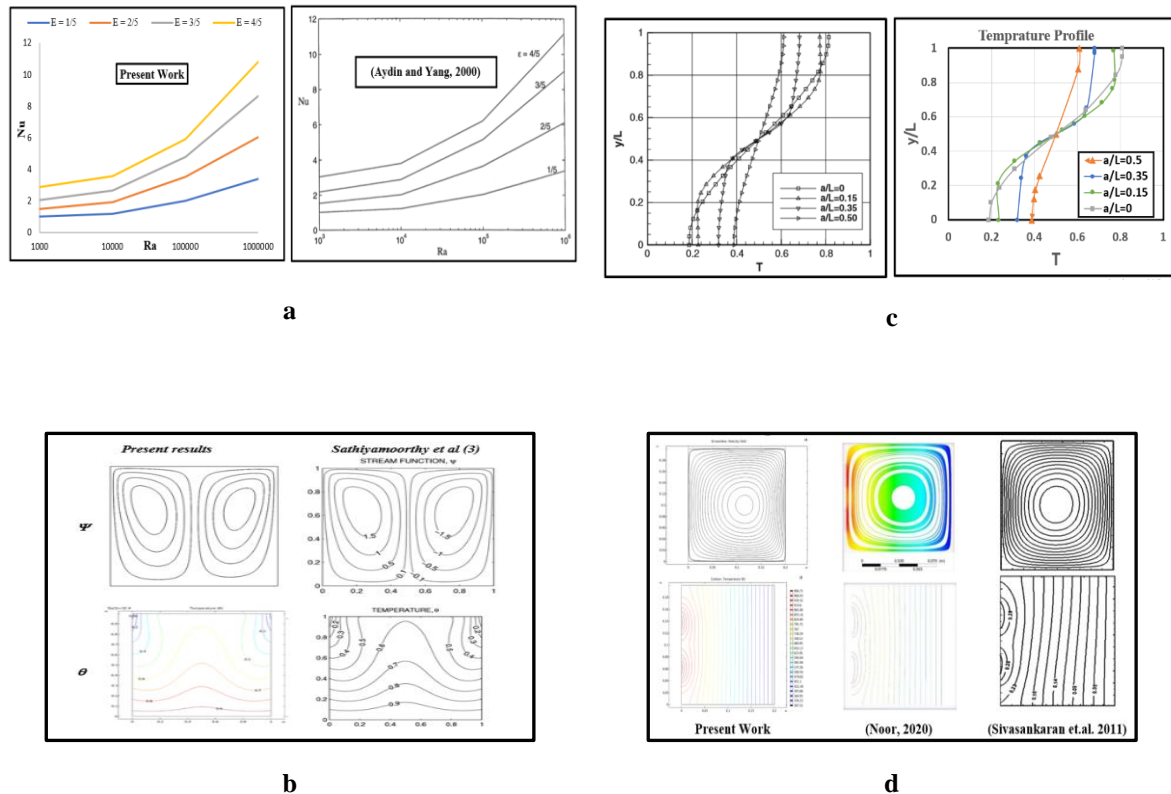

a

c

b

d

Figure 3. Variation of (a)(fluid only), $Nu_{average}$ for different heat source length (E)[9] (b)(Fully Porous media),Ψ(stream-function) and θ(isotherm plot)[10],(c)(Partially Filling Porous Media), Temperature Profile at Mid -Width of the Cavity (Present Work at Right. Side) [30] (d)(District Heating), Stream Lines and Thermal Behaviors of Present Work at left Column, [12] (middle) and [19] (Right Column).

385

Table 1. Thermophysical properties of air and porous medium

Physical properties	Air	Glass bead
ρ (kg/m^3)	0.99	2510.52
Cp(J/kg.K)	1006	670
k(W/m.K)	0.030	0.87

RESULTS AND DISCUSSION

The natural convection was studied inside a cavity filled with horizontal porous medium layer saturated with same fluid in lower region and free air for remaining enclosure. The Prandtl number (Pr=0.71) has been used for air with a different value of heat flux and the physical properties of air and glass beads is illustrated in (table 1). The two vertical walls are symmetrically cooled at constant temperature (Tc=35C°), bottom wall is heated by constant heat flux and adiabatic boundary condition for top wall is applied. The parameters that are investigated three lengths for heater (δ=20, 14, 7cm), four heights of porous medium (Hp=0.25L,0.5L, 0.75L, L), heat flux (q=150, 300, 450, 600W/m^2) and Darcey number (Da1) 1.570086×10^{-9} and (Da2) 3.541×10^{-5}) with porosity (0.418) and (0.569), permeability (1.570086×10^{-9}) and (3.541×10^{-5}) m^2 for Da1 and Da2 respectively.

Figure (4) describes the effects of (Hp) on the natural convection by the stream function (Ψ) and the isotherm lines (θ).
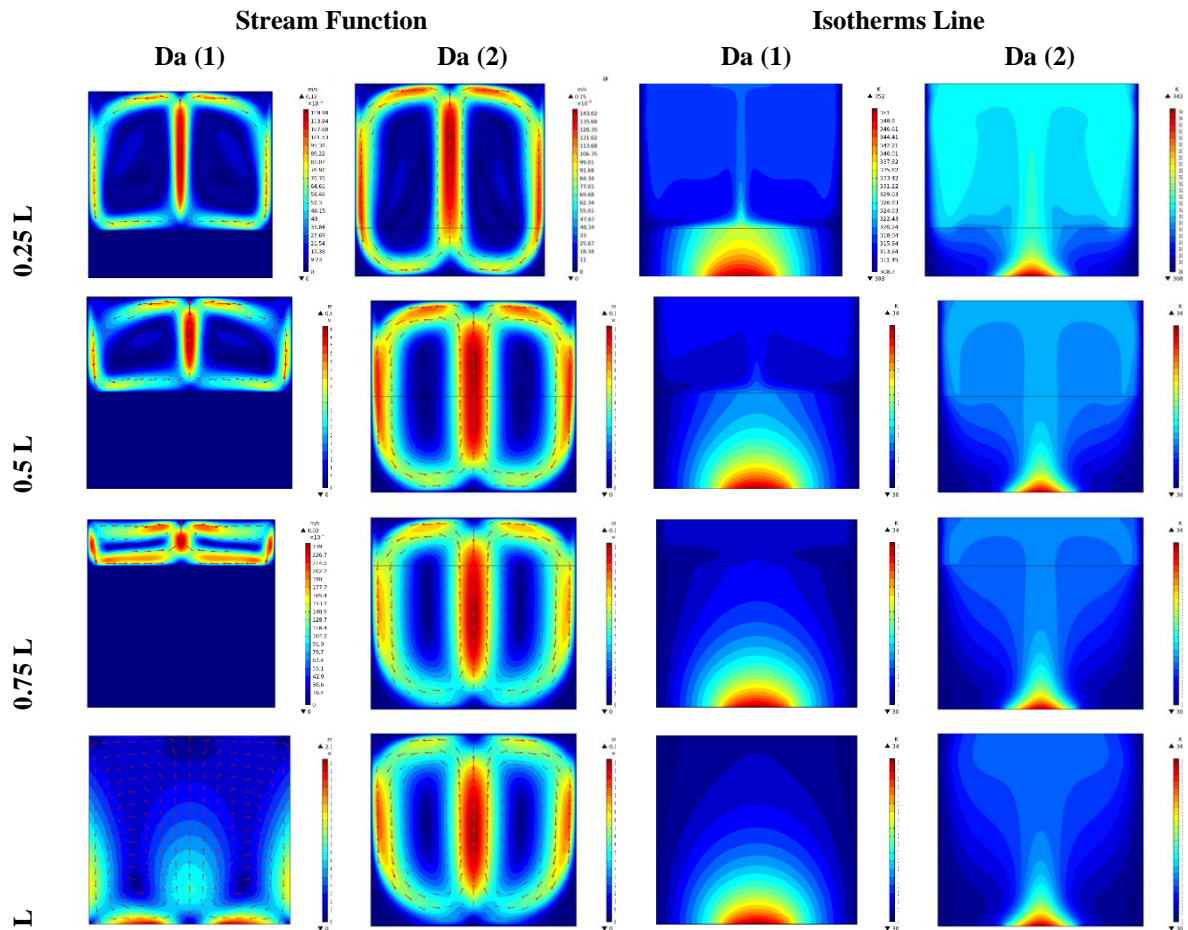
Figure 4. The Effects of Porous Layer Thickness on Stream Function and Isotherm contours for (Da1&Da2, δ=7 Cm, q=450Watt/m2.

The fluid flow appears with two large symmetric cells (one flow in the anticlockwise direction and the other in the clockwise direction). The cell behaviour is dominated along the enclosure due to high permeability (Da2) and less

drag in free fluid domain than porous layer but for greater drag (Da1) with drastic resistance to flow in porous domain. The flow circulation is decreased and the convection cell occupied the free fluid domain only with negligible penetration in porous layer for all cases. Nearly, same behaviour for all heater size so only (δ = 7Cm) has been chosen to display the effects of all studied range of (Hp). It is evident that the reduction of the fluid flow circulation in porous domain for all cases as compared with free fluid domain and the maximum velocity is located at midplane of circulation cells interface. Moreover, it is clear observed there is an increasing in intensity of stream line near the cold and isolation walls specially at (Da2) because of buoyancy force effects and irregularity of the glass beads at the vertical walls which results in a high value for permeability and thus its facilities the movement of fluid in that area.

Generally, fluid flow raises with decreasing the porous layer thickness and it will be maximum at Hp=0.25L. The isothermal lines display at the right column of the figure. It is noticed improvement the heat transfer when decrease the porous thickness specially at cases of (Da2) owing to two mean reasons, firstly high permeability comparing with (Da1) and secondly larger size of free fluid domain which supports heat transfer figure (5). There is a slightly increase 2.8% in maximum temperature in case (Da1) than (Da2) for hot side as result of porous layer obstruction and low conductivity for glass beads so the porous media acts as a buffer to vertical heat transfer. Conduction mode dominate the porous domain for all used heater size at (Da1) which is clearly illustrated by vertical isotherm lines while it changes to convection mode along enclosure for (Da2) affected by large size of free fluid region with high temperature gradient specially near the heat source(bottom) in porous domain. Moreover, it is clearly for all cases; same thermal plume as shape of isothermal pattern with wide base in porous domain for (Da1) and as umbrella for other (Da).
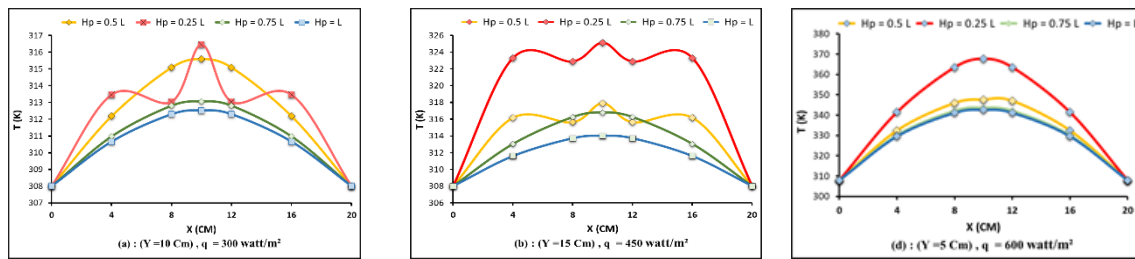

Figure 5. Temperature Distribution with X-axis for Enclosure with (Da1= 4.025×10-8) for Different Heat Flux, Porous Layer Thickness and Thermocouple locations. (a) δ=7cm, (b) δ=14cm, (c) δ=0cm (d) δ=20cm

The main parameter that affects natural convection is buoyancy force which is informed by the change of density due to temperature gradients (Boussinesq's approximation). So, the formed cells of flow rotation start from lower heated side of enclosure and rise above then it splits into two different directions due to the effect of the adiabatic upper wall to complete its cycle downwards under the effect of the cold walls until complete the cycle this behaviour is distinctly observed in present work. Figure (6) (left column) demonstrates the streamline plots at four values of heat flux (150,300,450,600) Watt/m2 for Hp=0.25L, δ = 20cm, Da1 and Da2.The symmetrical cooling on the side wall leads to symmetric in fluid movements about the vertical centre line of cavity, two symmetric cell flow on porous and free fluid domain due to high Darcy(Da2) value which leads to allowing greater movement of fluid through all enclosure but for low Darcy number (Da1) movement obstruction is detected.

Hence, the penetration of fluid in porous domain decreases and leads to raise the viscous forced comparing with buoyancy force. For all cases, the vertical mid area of enclosure includes the maximum fluid flow velocity and the increasing in heat flux leads to boost the fluid flow for all studied porous layer thickness. Also, it may be noticed that the value of the stream function is higher in the air layer for example (155 x10^{-3} m/s) at (300 W/m), while it is lower than that in the porous medium region, where it drops to (115 x10^{-3} m/s). When the heat flux is increased to (600 W/m^2), the heat transfer increases, which leads to an increase in the buoyancy force and streamlines intensity, where in the air layer it increases to (239 x10^{-3} m/s) while in the contact layer it increases to (170 x10^{-3} m/s)
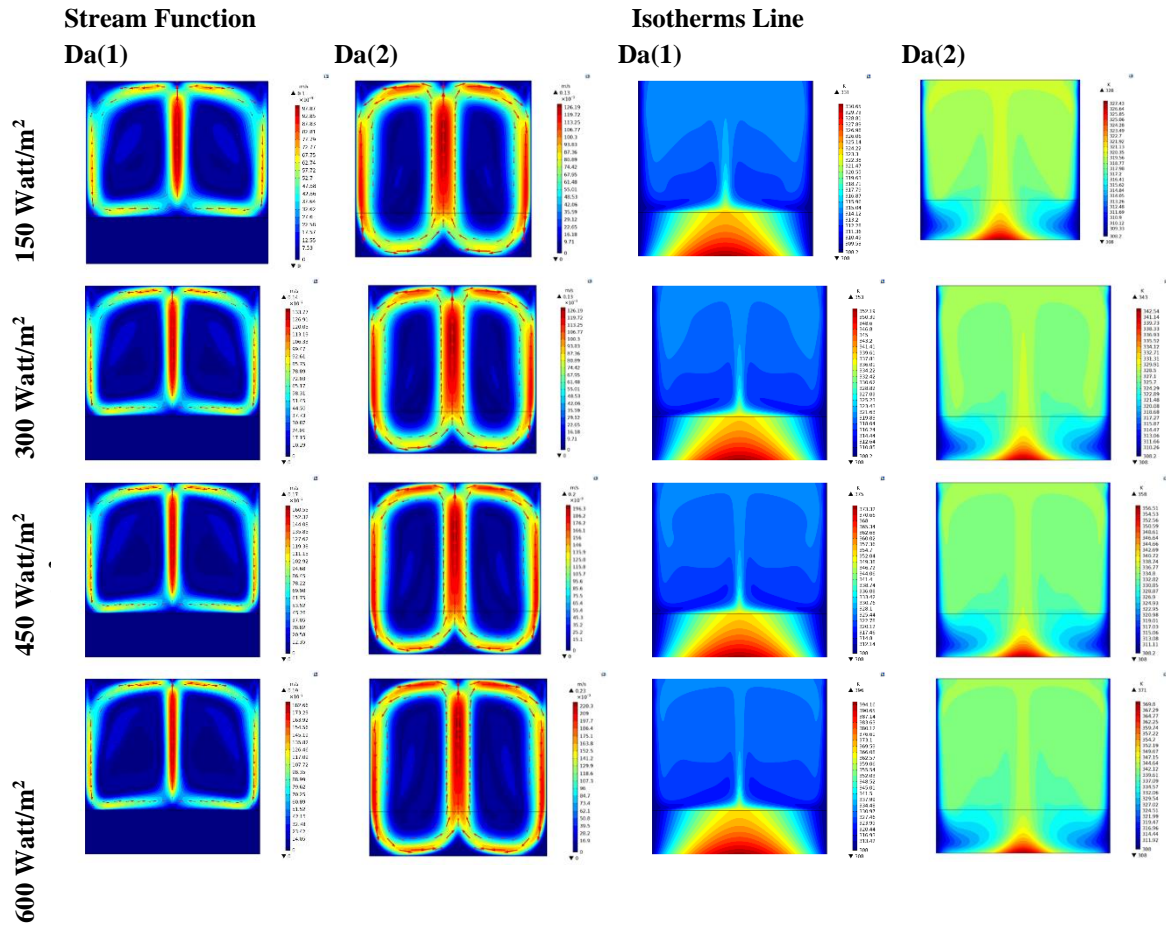
Figure 6. The Effects of Heat Flux on Stream Function and Isotherm contours for (δ=20 Cm, Hp = 0.25L, Da1 and Da2)

The right column of figure (6) shows the effects of studied heat flux on isotherm contour for Darcy number (Da2) with heat size (20 Cm). The maximum temperature is found in central vertical line for all cases and increases towards the top of enclosure, symmetrically distributed to the left and right of contour and then decreases downward which affected by cold wall and circulation cell of fluid flow. The temperature increases with heat flux raising and the best heat distribution is attended at heat flux (600) Watt/m^2 for Hp=0.25cm and heater size 20cm with an increase of (16.41% and 11.595%) over (q=150 Watt/m^2) for (Da1) and (D2) respectively. For all heat flux of Da2, indentation in the isotherms line is found inward affected by the ends of circulation cells of fluid flow which is located below the interface line of two domain due to good penetration of fluid in porous layer, while for Da1 this indentation is located at the overlying fluid (above interface line) with reason of obstruction of permeability which begins at this area. The isotherm plot for (Da1) is essentially conduction dominate mode with all heat flux values due to a week circulatory fluid motion while the heat transfer enhancement for same heat flux with (Da2) due to diffusion dominate (specially in free fluid section) by increasing the penetrative convection into porous media layers. To investigate the effect of heat source size, three heater sizes were studied (20,14,7cm). Expecting the flow is symmetry about the vertical midplane as discussed. Generally, for all cases of (Da2), the recirculation expands and enters the porous domain stream due to the increase of the convection intensity.
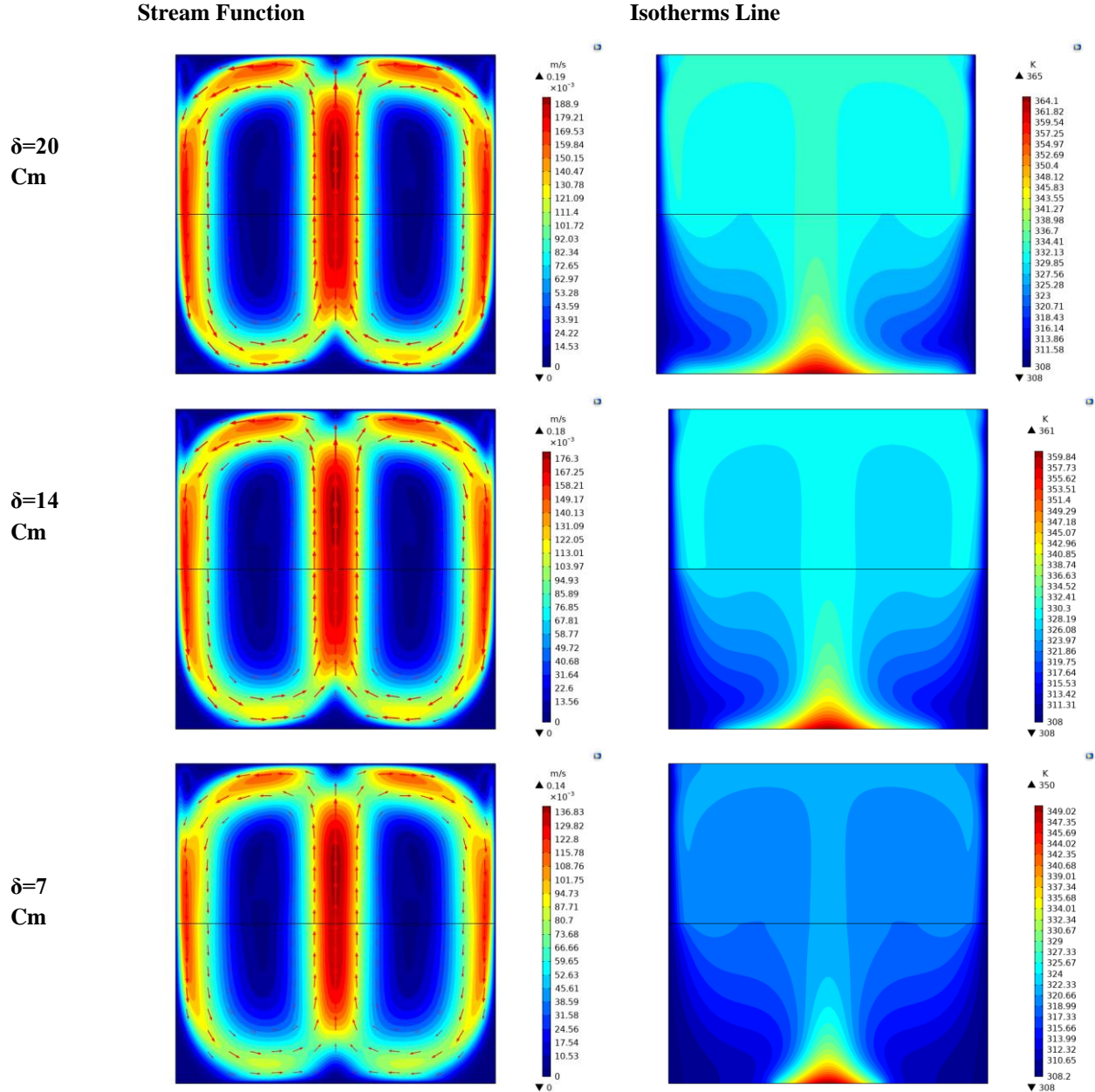
Stream Function   **Isotherms Line**

Figure 7. The Effects of Different Heat Flux length on Stream Function and Isotherm contours for (Da2= 8.8525×10^{-4}, q=600Watt/m² (Ra = 5.2166 ×10⁶), Hp = 0. 5)

a result improvement of permeability. However, the two symmetric cells rotate in opposite direction and becomes more intense at vertical mid-area of cavity with increasing the fluid flow velocity at cold walls and slightly near the top. Moreover, the maximum flow strength is found at heater size (20cm) for all cases because of the exposure of the larger area of the bottom to uniform heat flux also increasing the intensity of circular convention cell due to temperature gradient at bottom edge from local heater which is appeared as big size cells in stream function at fluid domain. However, the fluid flow decreases with lowering the length of heat flux to be minimum at (δ= 7cm). At the same time, it is noted that the increasing in the average of maximum flow velocity raise with heater size and thickness of porous layer because of effect of porous as discussed. Also, it is observed, for low permeability (Da1) with all heater size, two symmetric velocity cells that occupies the free fluid domain only with slightly or negatable spread of flow penetration through porous domain as discussed. Furthermore, the intensity of circulation decreases with all heater size comparing with Da2 with maximum increasing average of fluid flow velocity related heater sizes was at heater size (20Cm) and porous thickness (Hp=0.75L).

The impact of heater size in isotherms line for (Da2) are displayed in right column of figure (6), it is found an increase the bottom area of thermal plume with (δ) which boost the porous region that is exposed to heat flux and this led to a grow in heat intensity and rise in the maximum temperature for (δ=20cm) in ratio (2.35%) comparing with lower heater size for this case. Generally, the temperature gradient increases with heater size near the location of heater at porous domain which enhances the heat transfer by convection while it is reduced at cold wall specially at (δ=7cm) being moving away from the heat source for all cases. Same effect of heater size for cases of (Da1) with same shape (Dome) for isothermal contour and the conduction is dominant the heat transfer as the vertical isothermal lines in porous domain. Significant difference is found between the plots for high permeability (Da2) and low permeability for (Da1) and variation behaviors for fluid in porous domain and free fluid domain as shown in previous figures for stream lines and isotherm patterns.

Obviously, decreasing the Darcy number leads to raising the flow obstruction in porous domain so the symmetric flow circulation cells that was noted in the previously discussed cases concentrates on free fluid domain only with no penetration in porous layer also it is still rotated at opposite direction in the left and right upper part of cavity due to buoyancy force. Moreover, the increase in velocity is clearly noted in mid plane of free fluid domain owing to the confluence of the rotational movement's direction of symmetrical cell upward in this part of enclosure, this supports the fluid flow specially when increasing the heat flux and size of free fluid domain. The large void of porous domain(D2) rises the permeability and then decreases the fluid flow resistance. Consequently, perspicuous effects of buoyancy force on raising the fluid circulation. Also, the air penetration has been boosted and propagated underlying porous layer with improvement the bouncy force intensity and convection flow acceleration. As obvious note, the increase in fluid flow velocity in central vertical plane of system is due to air movement at same direction for two symmetrical convection cells. Moreover, the minimum flow velocity is found at central area of two convection cell and four enclosure corners because of the influence of buoyancy force.

It is evident that isotherm patterns are changed with increasing the permeability as shown. For (Da1), the temperature gradient (Dome shaped) is clearing concentrated near the hot side and conduction dominates the most porous domain as presented in vertical isotherm lines specially near the cold sides. While the effect of convection is pronounced in free fluid region. The enhancement of heat transfer is found at (Da2) with different thermal shape (heat plume) which has wide base and vertical extended column with more temperature distribution at lower region of enclosure affected by hindrance of porous. The system receives heat from bottom and distributes to top and side walls over the enclosure. Generally, maximum temperature is presented near the heat flux source that gradually decreases to be minimum at top due to effects of natural convection and cold walls. Local Nusselt number at bottom wall with investigated values of heat flux is illustrated in figure (8). Symmetrical distributed about central line is observed because of presence two symmetrical side cold wall with constant heat flux in bottom.

It is evident all figures that the local Nu increases with heat flux downstream of hot wall coordinates (x) due to the rise the amounts of energy that received from bottom which enforces the heat transfer by convection to be greater than conduction mode .Also, it is noted that the irregularity of the Nusselt number values along the heat flux wall is due to the effect of cold walls on heat transfer inside the cavity .Generally, minimum local Nu is found at center area of heat flux due to locate minimum flow velocity at this region which found under confluence the symmetric rotation cells, so the conduction was dominate heat transfer and the local Nu has been minimum obtained value. Moreover, it is clearly that the effect of increasing the flow velocity at mid area of enclosure rises the convection; therefore, the local Nu be greater in two half around vertical centerline of cavity as relation($Nu = \frac{h_x X}{k_p}$). Also, the heater size influences the movement of fluid through formed vortices in area around the ends of smaller heater so the convection become more stronger; for this reason, the local Nu increases in this portion at heater size (δ = 7cm) for same porous layer thickness, heat flux and (Da2) while this effect decreases in low porosity (Da1) which obstacle the fluid flow as discussed earlier.

As well, the ease of movement of fluid in cases of (Da2) boost the heat transfer by convection in the sense of raising the local Nu. Finally, perspicuous effect has been presented for porous layer thickness in local Nu for example, at case (Hp=0.25L),(Da2) and (q=600)Watt/m^2, enhancement in ratio about (94% and 30%) for (Hp=L)

and (Hp= 0.5L) respectively because of increase the area of free fluid which lead to promote the convection and therefore raising the local Nu, same behavior with (Da1) with ratio (92% and 28%) for same parameter. While Figure (9), shows the effect of average Nusselt number for different values of (Ra) at studied porous layer thickness. It may be noticed that there is increasing the average Nu linearly with (Ra) or heat flux for all cases and becomes greater in cases of heater length (δ=20 cm) due to the larger area of porous media that exposed to heat flux. Also, it is found that the enhancement the heat transfer when decrease the Hp and be maximum at porous thickness (Hp=0.25L). Generally, the results of average Nu is found nearly (Nu$_x$=1) for low permeability (Da1) at different (Ra) or heat flux which explains the conduction mode is dominant in enclosure as discussed previously.
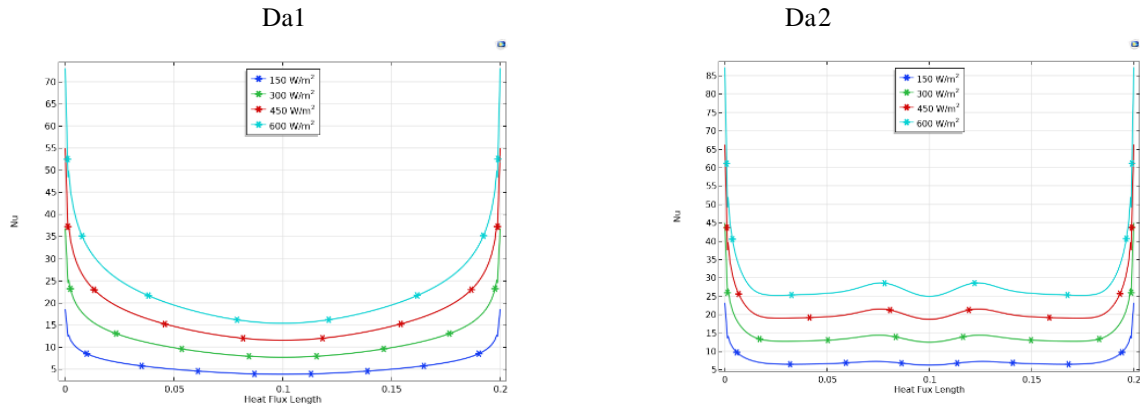
Da1   Da2

Figure 8. The Variation of local Nusselt Number with heater length for δ=20Cm, Hp=0.25L with different heat flux values.

Da1   Da2

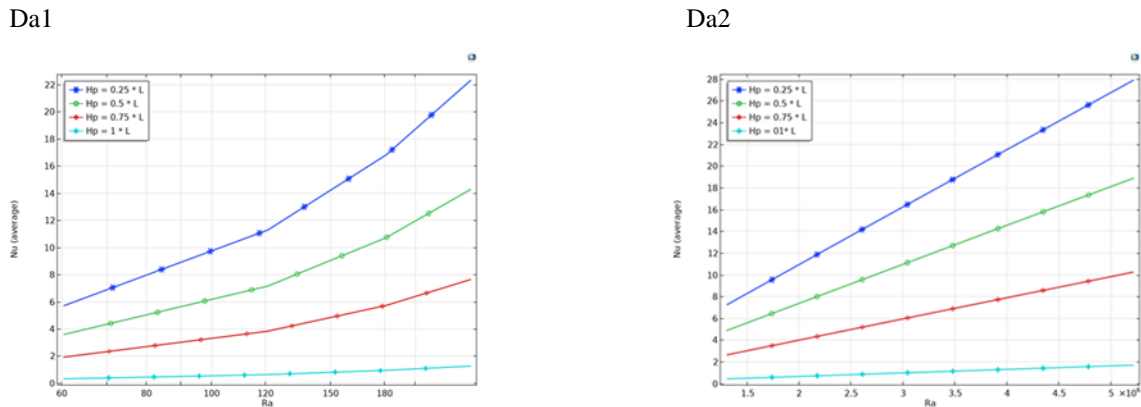**Figure 9.** The Variation of average Nusselt Number with heater length for δ=20Cm, Hp=0.25L with different Rayleigh values.

CONCLUSION

The present numerical study shows the effects of investigated parameters (four porous thickness, three heater size and porosity) on heat transfer by natural convection as follows:

1. The partial filling by porous media affects temperature distribution especially for porous layer thickness equal to (0.25L) which represented the best case comparing with other (Hp)for all parameters.
2. The maximum temperature is obtained at small porosity and increases with increasing heater size and decreasing porous thickness where the maximum temperature increasing is about (5) % for case (Da1) with (Da2). So, for drying applications (for example), it is better to decrease the distances between granules or piece of the required material.
3. Darcy number clearly affects the heat transfer enhancement at high porosity.

4. Nusselt number increases with the porous thickness and best results are obtained at cases of (Hp=0.25L and Hp=0.5L). The maximum increasing ratio is obtained in the case of porous layer thickness(5cm) and it is approximately (93%).

5. Good enhancement in heat transfer is achieved with increasing the heat flux or (Ra) value which reach to (71%) for Nusselt number in case of ($q=600Watt/m^2$).

6. Increasing the Nusselt number with length of heater and best result is found nearly to (21%) at (δ=20cm.

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