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Numerical Simulation of Natural Convection in a Rhombic Enclosure with Heated Sidewall Coated with Metal Foam

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ABSTRACT

A natural convection heat transfer inside rhombic square cavity partially filled with porous material have been numerically investigated. A constant heat flux has been applied to the left wall with a right wall kept in constant cold temperature while thermally insulated the top and bottom walls. Finite volume technique with Simple algorithm have been used to simulate the governing equations of fluid flow and heat transfer coupled with Darcy-Brinkman model to simulate the flow of the air inside the main cavity and the open cells of the porous media. Three factors were chosen to study their effects on the natural air velocity and the mechanism of the free convection inside the enclosure. The inclined angle of the sidewall of the rhombic (q = 90º, 80º and 70º), the thickness of the metal foam (t = 5 cm, 10 cm, and 15 cm) and the amount of heat flux (q = 150 to 600 w/m^2). Copper metal foam with 0.9 porosity was chosen as porous media with open cell filled by air (Prandtl number =0.7) and 10 as pore density. The results showed that using a layer of porous metal foam with open cells will increase the heat transfer rate. It was 41.3% enhancement when use 5 cm of porous media and 68% for 15 cm. Acute inclined angle will decrease local Nusselt number and led to form vorticities. Furthermore, high heat flux increased the average Nusselt number and improved the heat transfer rate.

1. Introduction

Heat transfer through free convection is a significant phenomenon which play a big role in many engineering and industrial applications. This role is reflected in the performance of whole systems [1, 2]. In natural convection the buoyancy force which comes from densities differences is responsible for the mass transfer and fluid motion which in turn leads to heat transfer [3-5]. Recent years, the expansion of using the porous media which is offers big features lead to emergence the need to fully understand how heat is transfer in these media [6-8]. Nuclear reactors, oil production industries, modern cooling techniques, geothermal applications, chemical electrolytic batteries, and many other fields are now commonly used the porous media in their components [9-11].

Last two decades, using of metal foams were so expanded in many of engineering fields and attracted more significance [12]. A metal with porous structure of closed or opened cavities with high thermal conductivity is so suitable and compatible to varies of heat transfer application. This motivate a lot of researchers to focus and expand their efforts on study in deep the natural heat convection in metal foam structure [13, 14]. Khashan, et al. [15] studied natural convection

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within a closed cavity filled with a porous media saturated with incompressible liquid. The case was steady state laminar flow analyzed by finite volume method for two equations model of solid and liquid state with non Darcian effects. The results approved that the effect of Forchhiemer term high is strengthen at high Rayleigh number cases with reinforcement of natural circulation and local thermal non-equilibrium and therefore increase average Nusselt number. Sathiyamoorthy, et al. [16] numerically tested the natural heat transfer inside square enclosure uniformly heated from the bottom and linearly heated from the laterals. The cavity is filled with porous media and thermally insulated from the top. They used Darcy model with Forchheimer term to predict the flow within the porous media. The results showed emergence of a strong secondary flow circulation in the half bottom of the cavity. The existence of these flow circulation cause a fluctuation in the values of local Nusselt number. The Increasing of Prandtl number led to weaken this secondary flow circulation. Chen, et al. [17] inspected the natural flow of heat and air transfer inside a square cavity partially filled with porous material saturated with fluid. The cavity was heated from one side and cooled from the other with thermal insulated of the top and bottom. They used Brinkman-Forchheimer extended to modulate the porous part of the flow with SIMPLE algorithm. They recommended focusing more attention on the region of the line between the fluid and the porous media which may affect the whole heat transfer rate and average Nusselt number variations. Sivasankaran and Bhuvaneswari [18] investigated a two-dimensional square cavity heated from the two sides and thermally insulated from top and bottom. The enclosure was filled with porous media saturated with liquid. The heated sides was subjected to a temperature with non-uniform sinusoidal distribution. The results approved that the heat transfer was in high values when the applied temperature was harmonic and vice versa. A natural convection through a porous media packed in a cavity have been explored by Chamkha and Ismael [19]. The porous media was partially filled with nanofluid. The cavity was subjected to heat transfer from one side to the other with thermally insulated the roof and the bottom. The heated side was covered with porous media. The results showed that use of the nanofluid enhanced the heat transfer rate even with existence of the porous layer. However, increased the thickness of the porous media layer decreased the average Nusselt number. Su, et al. [20] performed an experimental and numerical investigation to inspect the ability of increasing heat transfer by natural convection of a cavity filled with water by adding porous media. They tested inserting copper metal foam over the bottom or below the roof or both to know how this will change the natural velocity of the fluid inside the cavity which is heated from below and cooled from the top. The results revealed that the existence of the porous layer beside on of the heated or cooled wall enhance the transfer of the heat and this is due to the high thermal conductivity of the copper compared to water. A three-dimensional study of cubic cavity filled with porous material has been achieved by Guerrero-Martínez, et al. [21]. The system consists of three layers with constant physical properties for the external layers and varied permeability and thermal conductivity for the internal one. The cavity is heated from the bottom and cooled from the roof. They showed that only a significant decrease of the permeability will lead to drop of the Nusselt number which is also inversely proportional with the thickness of the porous material. Using lattice Boltzmann method Chen, et al. [22] investigated free convection inside open side enclosure partially filled with porous material. They explored the effects of thickness of the porous media, permeability, and the ratio of thermal conductivity of the liquid to the solid material on heat transfer by natural convection inside the cavity. They found that all these factors have their own influences in the mechanism of the heat transfer and to fully understand the impact must find the value of this factor which change it optimally. Chordiya and Sharma [23] studied a case of unsteady natural convection in a square cavity heated from one side and cooled from the other with thermally insulated top and bottom walls. The cavity is filled with porous media saturated with liquid with and without two verticals inside partition walls. The outcomes of the cavity without the partitions showed that the regions of high heat transfer is located at the top right and bottom right of the cavity. In a numerical study using finite element method the natural air circulation by free convection inside an encloser has been analyzed by Abdulwahed and Ali [24]. A square cavity filled with air and heated bottom coated with a layer of copper metal foam was put under test. The encloser has two cooled sides and a thermal insulated roof. The effect of the cooper metal foam thickness was inspected. The results showed the effect of porous media in reduction the velocity of air circulation and confirmed the increasing of Nusselt number with the increasing of the heater size.

In general, most of the research and articles dealing with the natural heat convection inside closed cavity were focused on specific geometry which was square enclosure or in few of them rectangular. The rhombic cavity, which is an interesting geometry for many engineering applications has been overlooked and it didn't receive enough of investigations efforts. In rhombic cavity, there will be the effect of the sidewall angle which is expected to influence the mechanism of the natural heat convection inside the cavity. However, past literature mostly focused on heating the bottom side of the cavity with the least efforts on the side heating. This emphasize the importance of investigation of the rhombic cavity with side heating and cooling. This work focuses on the natural convection inside a two-dimensional square rhombic enclosure with heated sidewall coated with copper metal foam and cooled the other side while keeping the top and bottom thermally insulated. The system of Navier-Stokes equation will be solved with Darcy-Brinkman model using Simple algorithm and finite volume method (FVM). The main objective is to find the effects of the thickness of the porous layer, the angle of the side wall and the amount of the heat flux on rate of over whole the heat transfer.

2. Mathematical formulation

The physical description of under study case is a rhombic square cavity with L side length exposed to a natural convection heat transfer as shown in Figure 1. The cavity is subjected to a constant heat flux (q) in the left side wall which is covered with a layer of a copper metal foam of different thickness (t) and the right-side wall is subjected to constant cold temperatures. The top and bottom of the cavity is thermally insulated. The fluid filled the cavity is air with no heat generation. A thermal equilibrium has been assumed with laminar flow.

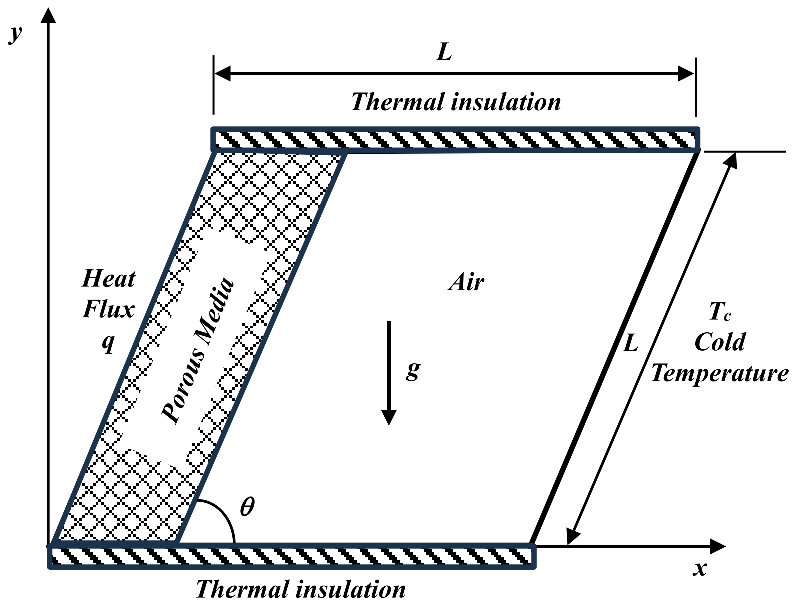


Figure 1. Configuration geometry and boundary condition of the studied cavity

The conservation of mass and energy equations have been numerically solved with Boussinesq's approximation and the model of Brinkman-extended Darcy. The mathematical mode describes the problem is as follows [25]:

The conservation of mass (continuity equation) in the cavity region (air zone) is:

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

the porous region (copper metal foam zone) is:

$$\frac{\partial u_{por}}{\partial x} + \frac{\partial v_{por}}{\partial y} = 0 \tag{2}$$

The conservation of momentum (momentum equation) in the cavity region (air zone) in the x -direction is:

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

the y-direction is:

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

In the above equations u_{air}, v_{air} represent the velocity components of the air in the empty zone in the x and y directions respectively, while u_{por} and v_{por} are the velocity components of the air in the porous media zone in x and x direction respectively, m_{air} is the dynamic viscosity of the air and r_{air} is the density of the air, g is the gravitational acceleration and b_{air} is the coefficient of thermal expansion of the air. The conservation of momentum (momentum equation) in the porous region (copper metal foam zone) in the x -direction is:

$$\frac{1}{\varepsilon^2}\left[u_{por}\frac{\partial u_{por}}{\partial x} + v_{por}\frac{\partial u_{por}}{\partial y} = -\frac{1}{\rho_{por}}\frac{\partial p}{\partial x} + \frac{\mu_{por}}{\varepsilon\rho_{air}}\left(\frac{\partial^2 u_{por}}{\partial x^2} + \frac{\partial^2 u_{por}}{\partial y^2}\right)\right] - \frac{\mu_{por} u_{por}}{\rho_{por} K} \tag{5}$$

the y-direction is:

$$\frac{1}{\varepsilon^2}\left[u_{por}\frac{\partial v_{por}}{\partial x} + v_{por}\frac{\partial v_{por}}{\partial y} = -\frac{1}{\rho_{por}}\frac{\partial p}{\partial y} + \frac{\mu_{por}}{\varepsilon\rho_{air}}\left(\frac{\partial^2 v_{por}}{\partial x^2} + \frac{\partial^2 v_{por}}{\partial y^2}\right)\right] - \frac{\mu_{por} v_{por}}{\rho_{por} K} + \beta_{por} g (T - T_o) \tag{6}$$

where e and K represent the permeability and porosity of the porous media (copper metal foam). The following equation can be used to determine the permeability (K) of the porous media [25]:

$$K = \frac{\varepsilon^3 D^2}{150(1-\varepsilon)^2} \tag{7}$$

The conservation of energy (energy equation) in the cavity region (air zone) in the x -direction is:

$$u_{air}\frac{\partial T}{\partial x} + v_{air}\frac{\partial T}{\partial y} = \alpha_{air}\left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2}\right) \tag{8}$$

the porous region (copper metal foam zone) is:

$$u_{por}\frac{\partial T}{\partial x} + v_{por}\frac{\partial T}{\partial y} = \alpha_{eff}\left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2}\right) \tag{9}$$

where a_{eff} is the effective thermal diffusivity which can be determined as [24]:

$$\alpha_{eff} = \frac{k_{eff}}{(\rho c_p)_{air}} \tag{10}$$

while k_{eff} the effective thermal conductivity can be found from:

$$k_{eff} = (\varepsilon) k_{air} + (1-\varepsilon) k_{por} \tag{11}$$

To calculate the Rayleigh number, local Nusselt number and average Nusselt number the following equation can be used, respectively [26]:

$$Ra = \frac{K g \beta_{air} L^2 \rho_{air} q}{k_{eff}\, \mu_{air} \alpha_{eff}} \tag{12}$$

$$Na = -\left(\frac{k_{por}}{k_{eff}}\right)\frac{\partial T}{\partial x} \tag{13}$$

$$Nu_{av} = \frac{1}{L}\int_0^L Nu\, dy \tag{14}$$

The boundary conditions of the cavity under study is as follows:

1- An adiabatic bottom wall of the rhombic cavity:

$$y = 0, \quad 0 \le x \le L, \frac{\partial T}{\partial y} = 0, u = 0, v = 0$$

2- An adiabatic Top wall of the rhombic cavity so

$$y = L \sin\theta, \quad L \cos\theta \le x \le (L + L\cos\theta), \frac{\partial T}{\partial y} = 0, u = 0, v = 0$$

3- The right wall of the rhombic cavity subjected to cold temperature:

$$0 \le y \le L \sin\theta, \quad L \le x \le (L + L\cos\theta), T = T_c, u = 0, v = 0$$

4- The left wall of the rhombic cavity subjected to a constant heat flux:

$$0 \le y \le L \sin\theta, \quad 0 \le x \le L\cos\theta, q = h(T - T_o), u = 0, v = 0$$

where q is the heat flux, T_o is the temperature of the ambient and h is the coefficient of the heat transfer.

5- The line of interface between the air and the porous copper

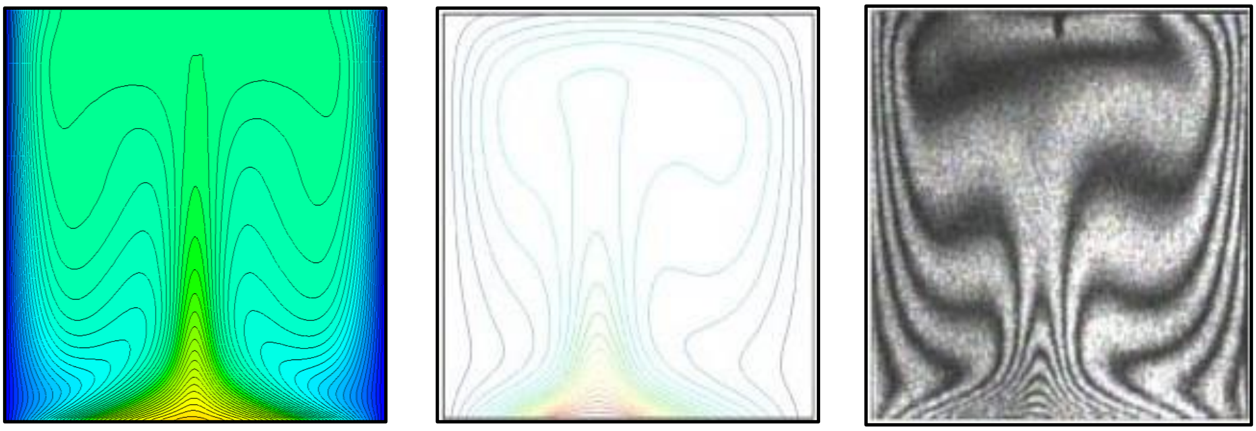
$$k_{air}\frac{\partial T_{air}}{\partial x} = k_{por}\frac{\partial T_{por}}{\partial x}, then\ T_{air} = T_{por}$$

where k_{air} and k_{por} are the thermal conductivity of the air and porous copper respectively.

3. Results and discussion

A numerical analysis has been used to solve the equation of conservation of mass and energy to simulate the free convection inside the cavity. Finite volume technique with optimized number of triangular mesh have been utilized to predict the effects of three factors on heat transfer rate and the mechanism of the thermal convection through the enclosure. These factors include the applied heat flux, the inclined angle, and the thickness of the porous metal media.

To ensure the validity and reality of the current methodology and simulation whether it can be used to predict of the under-study case, a samples of the results have been compared to the experimental and numerical work of Corvaro and Paroncini [27] and those of Shim and Hyun [28]. Also, the current simulation was applied to reproduce the results of Abdulwahed and Ali [24] work. An acceptable agreement between these studies with the current investigation could be noticed as seen in Figures 2 and 3.



a) Current study

b) Numerical result of Corvaro and Paroncini [27]

c) Experimental result of Corvaro and Paroncini [27]

Figure 2. Comparison of the current study with Corvaro and Paroncini [27] numerical and experimental research

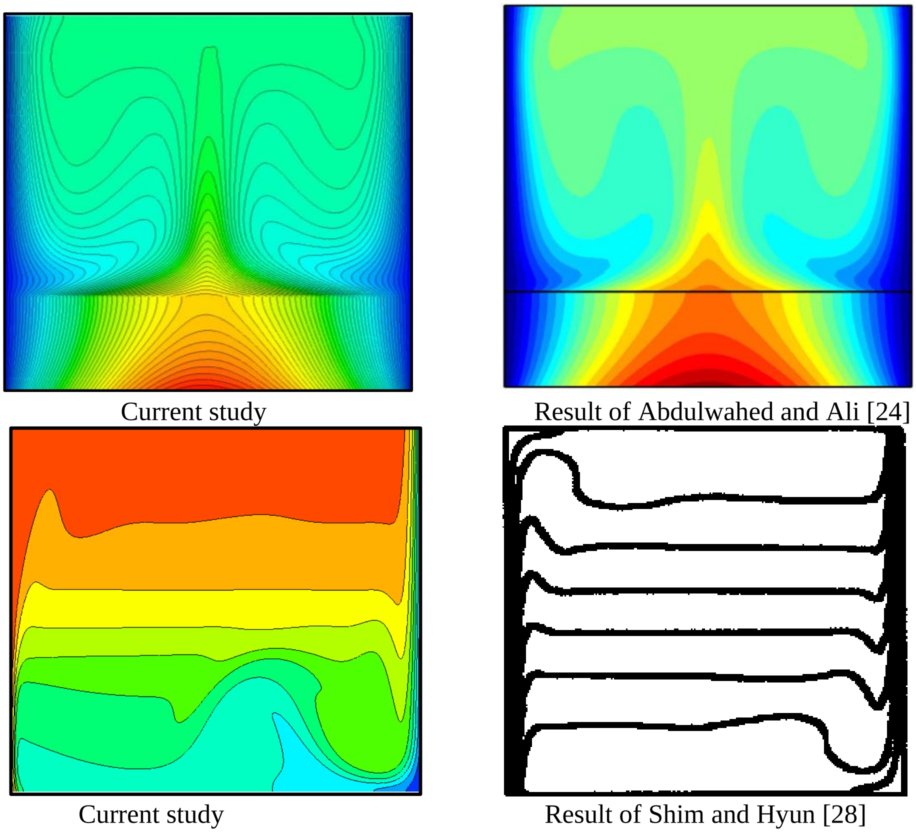


Figure 3. Comparison of the current study with the results of Abdulwahed and Ali [24] and Shim and Hyun [28]

Figure 4. presented the effects of the sidewall angle of the cavity on the heat transfer by natural convection inside it. In all its parts it can be seen two big whirls of air flow one at the below right corner and the other at the top left corner. The formation of these two vortices are due to the geometry and the conditions of the cavity. The asymmetric situation of the boundary conditions is represented by the existence of the heat source on one side and the heat sink on the other side with thermal insulated roof and bottom. Obviously air flow with natural effect due to density difference is headed from below to top driven by buoyancy force. When the hot air touch the top wall it will not lose its heat because of the heat shield so it is forced to redirect down forming the vorticity. The movement of this whirl will induce the circulation of the lower layers of the air which are near and adjacent to the thermal insulated bottom until these movement reach the right cold wall to losses their energy. Increasing the thickness of the porous material (t) to 10 cm for the same value of the applied heat flux as can see from figure 5. led to clarify the formation of the upper whirl. It can be easily seen as the velocity streamline headed to the heat sink of the right wall. This can be attributed to the shorter distance between the heat source and the heat sink for this case compared to the previous case (5 cm). However, it can be noticed also increasing in the maximum value of the air velocity when increasing the porosity material thickness (t) due to the high thermal conductivity of the copper compared to air. This behavior continue when increasing the thickness of the metal foam (t) to 15 cm. Decreasing the inclined angle of the cavity sidewall causes impede of the heat transfer from the source (heated wall) to the sink (cold wall). This is supported by the decrement of the maximum values of air velocity due to decreasing of the inclined angle and converted it to be more as acute angle. This conversion created a narrow passage for the movement of the air molecules. The slow motion of the air particles and the obstruction of its area led to accumulated them then inverse the phenomena of heat transfer from dominated natural heat convection combined with a low portion of heat conduction to dominated heat conduction with a little bit of heat convection.

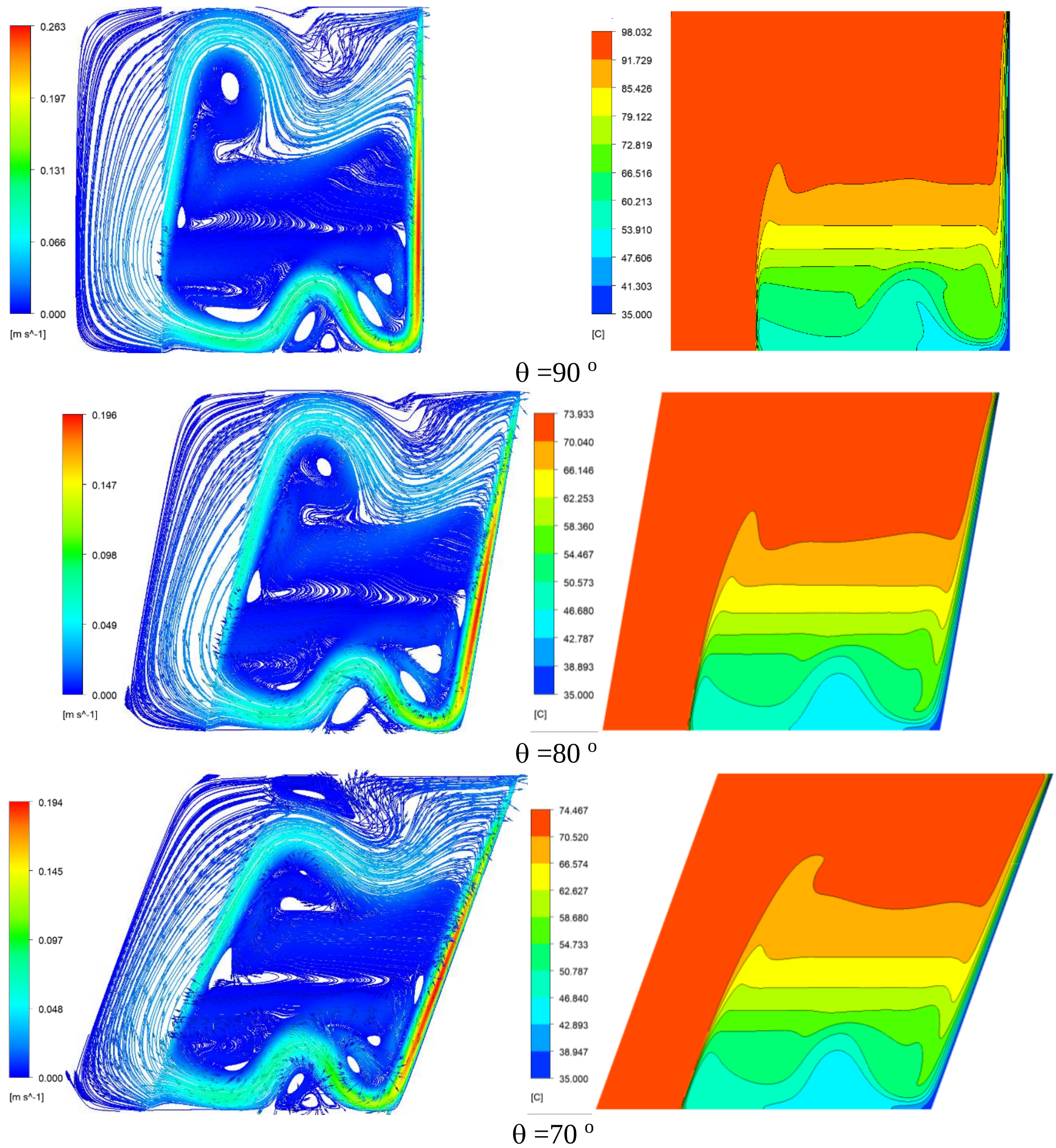
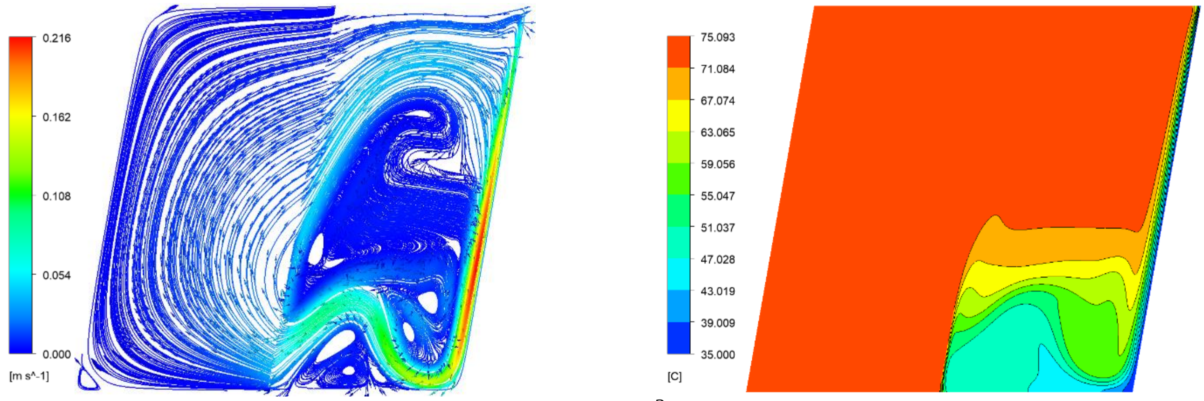
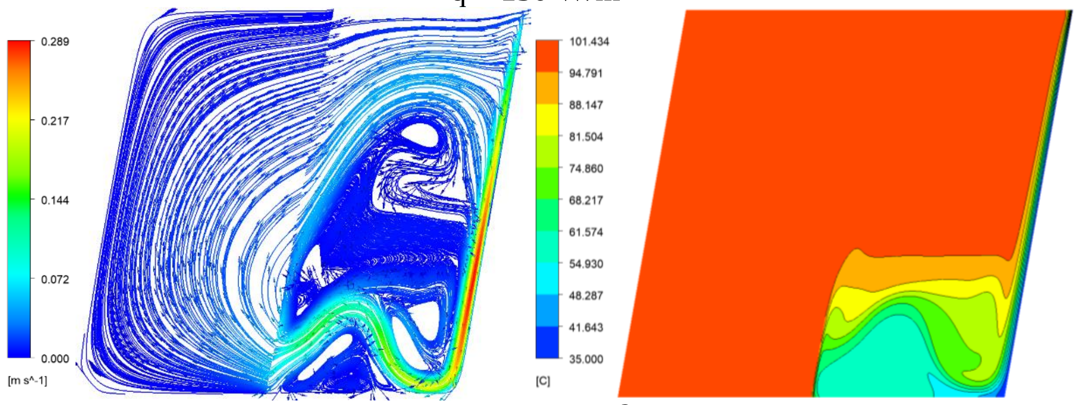


Figure 4. Effect of inclined angle (q) on the velocity streamlines and isotherms for q =150 W/m^2 and t = 5 cm

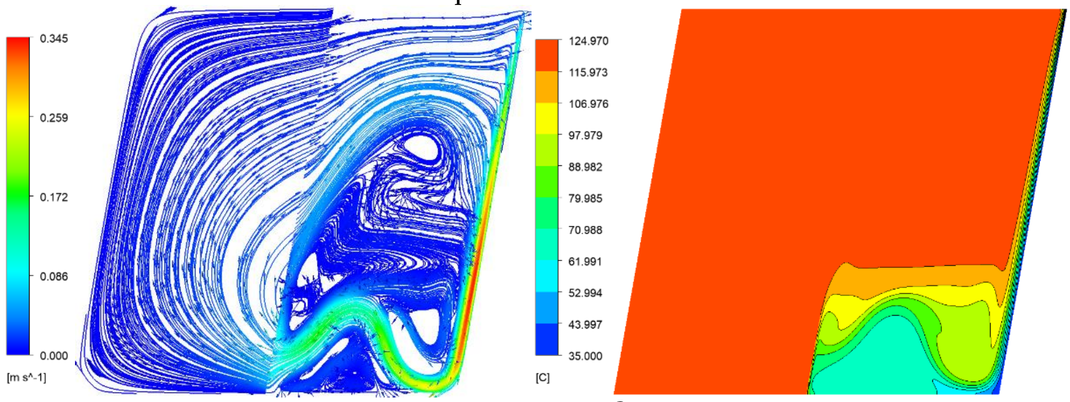
The effects of increasing the heat flux on the heat transfer and natural air movement inside the cavity have been shown in figure 5. It can be noticed that regardless of the thickness of the porous media (t) and the value of the inclined angle of the sidewall, increasing heat flux caused increasing the air velocity which in turn lead to enhancement in the heat transfer rate. This is attributed to the growing of the Rayleigh number. It can also be noticed that rise of the maximum temperature inside the cavity and this due to the increase in dominance of the heat conduction at the expense of heat convection because of the sidewall inclination. This is clearly noticed from the superiority of the red color in the isotherms contours.



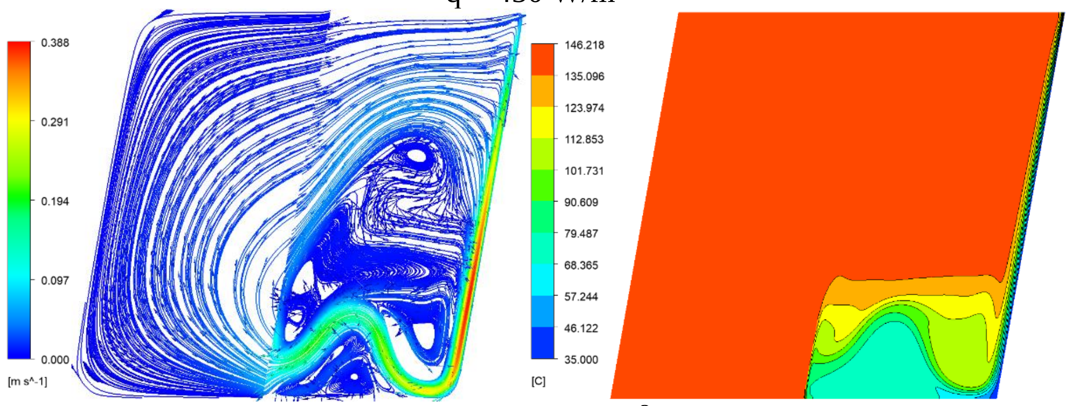
q = 150 W/m^2



q = 300 W/m^2



q = 450 W/m^2



q = 600 W/m^2

Figure 5. Effect of heat flux (q) on the velocity streamlines and isotherms for q =80° and t = 10 cm

Figure 6. discuss the relation of the thickness of the copper foam (t) with the mechanism of convection heat transfer inside the cavity. It can be shown that at a constant inclined angle and constant heat flux increasing the porous layer thickness (t) will raise the velocity of the air in the cavity so increase the Rayleigh number and rate of heat transfer. This make the whirls more prominent and this so clear at 10 cm. For more increasing at 15 cm, it can be noticed that disappearing of the upper vorticity because of lack of required space to form it due to proximity of the heat sink (cold wall) from the heat source (hot wall). Consequently, the velocity streamline will transfer in an approximately inclined lines from left to right headed too slightly up.

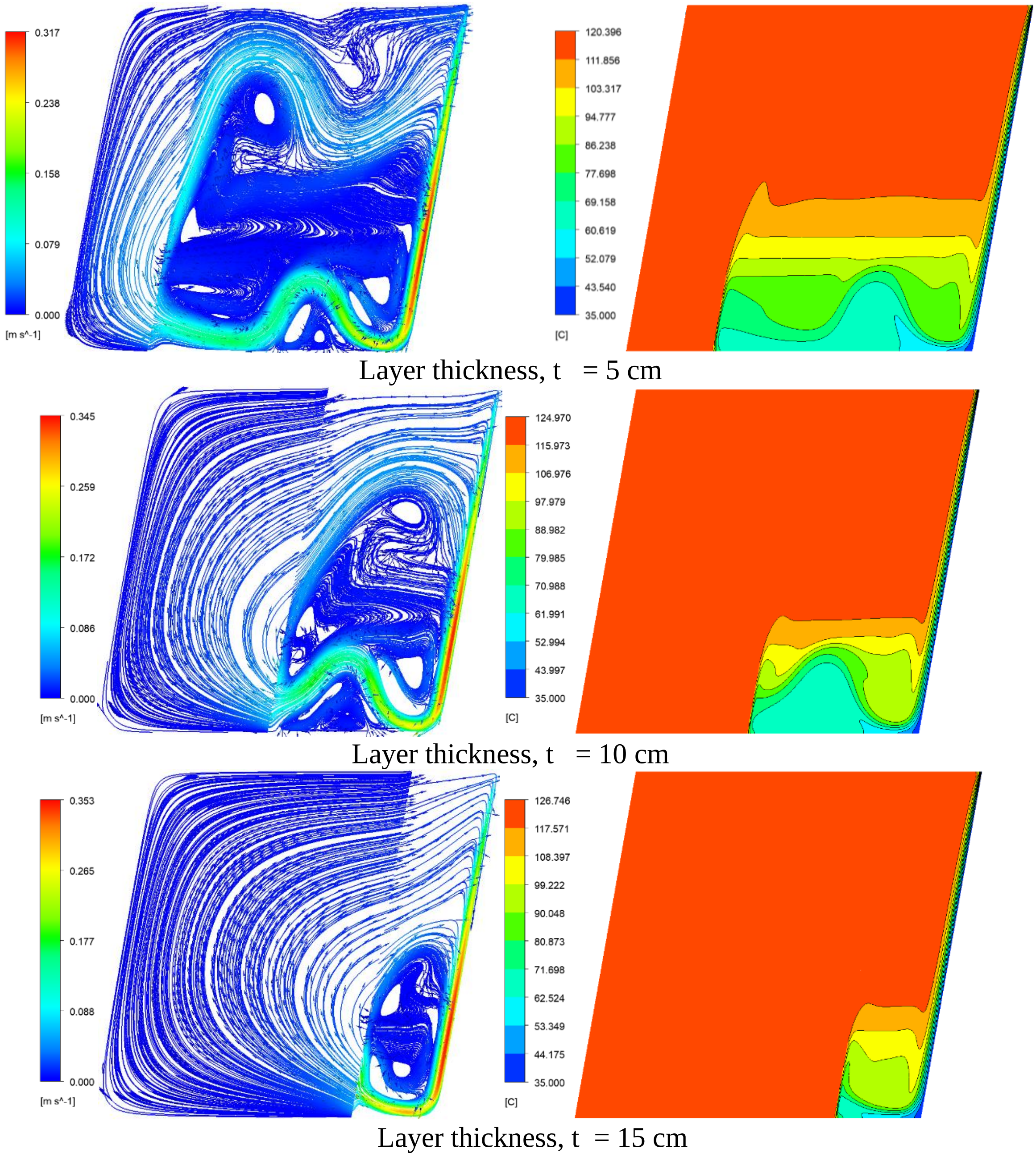
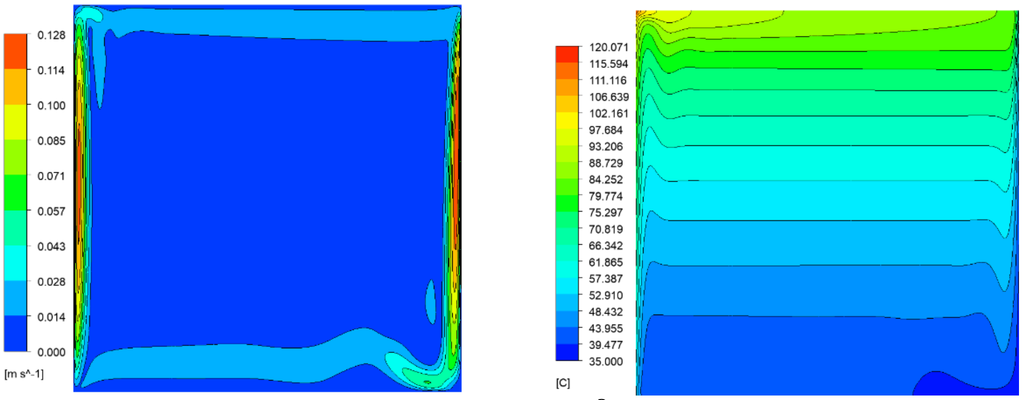
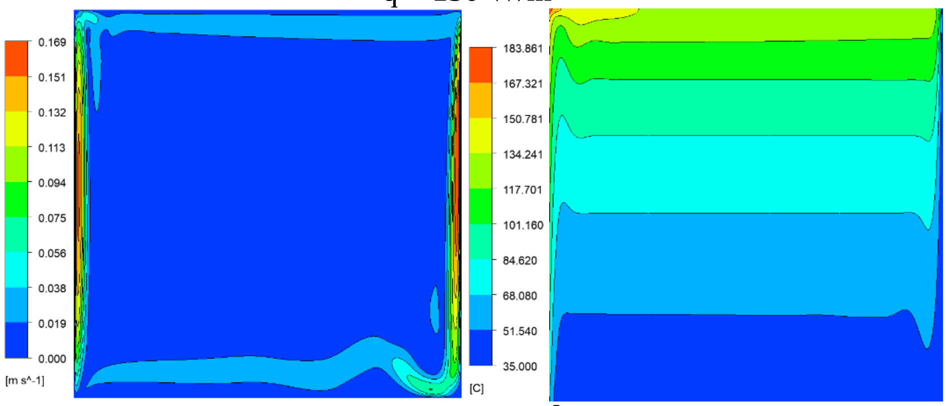


Figure 6. Effect of thickness of the copper foam (t) on the velocity streamlines and isotherms for q = 80° and q = 450 W/m^2

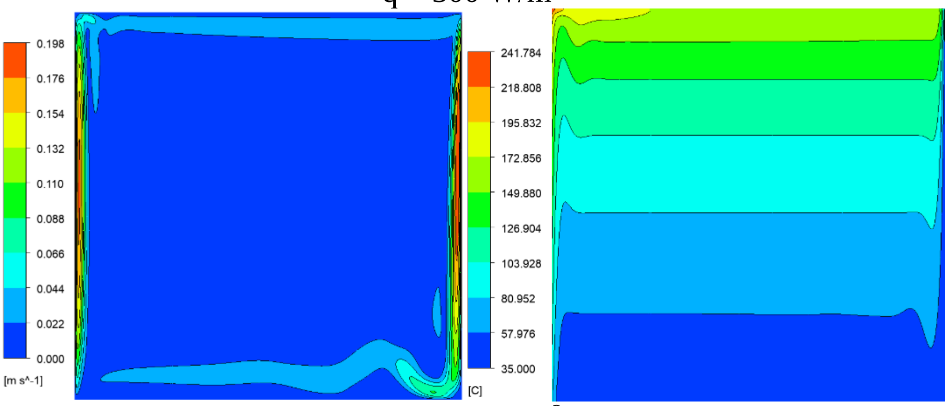
For more discussion about the above-mentioned results the effects of the heat flux value on the natural heat convection inside the cavity have without inclined angle (orthogonal) and without porous media have been studied in figure 7. Through quick look at this figure, it can be shown that the pattern support the above discussion where the behavior of the curves confirmed the increasing of heat transfer according to increasing in the applied heat flux due to accelerate the speed of the air particles so rise Rayleigh number. Because of the nonexistence of the porous media, the movement has the required space without obstacles and the whirls were in their smallest size.



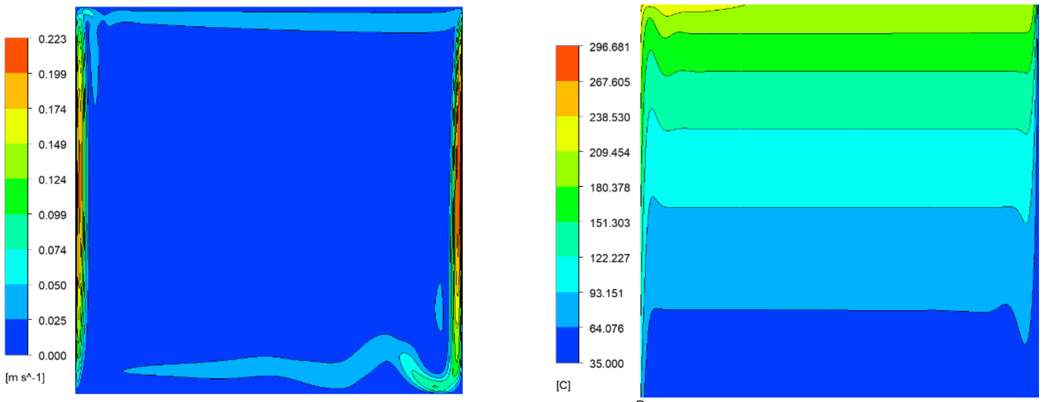
q = 150 W/m^2



q = 300 W/m^2



q = 450 W/m^2



q = 600 W/m^2

Figure 7. Effect of heat flux (q) on the velocity streamlines and isotherms for q =90º and no porous media

The change of local Nusselt number affected by heat flux value changed, inclined angle (q) and the thickness of the porous media have all been discussed in figure 8. It can be noticed that the value of local Nusselt numbers are very high down the hot wall and begin to decrease toward up till reach the minimum value. This behavior is supported by what have mentioned before where at the down section near the hot wall the heat is almost merely convection. Due to densities difference the buoyancy force bush the air particle up to the thermally insulated top wall. Then because the air will not lose the energy, it will accumulate there trying to find another way to flow. The stuck air there serve as thermal conduction media and prevent or reduce the convection currents. So, the conduction heat transfer became the dominated heat transfer phenomena. This can be seen in all the figures where the red color prevails over the top of the contours. The figure also confirmed that Nusselt number is raised by increasing the applied heat flux value and increasing the thickness of the porous media.

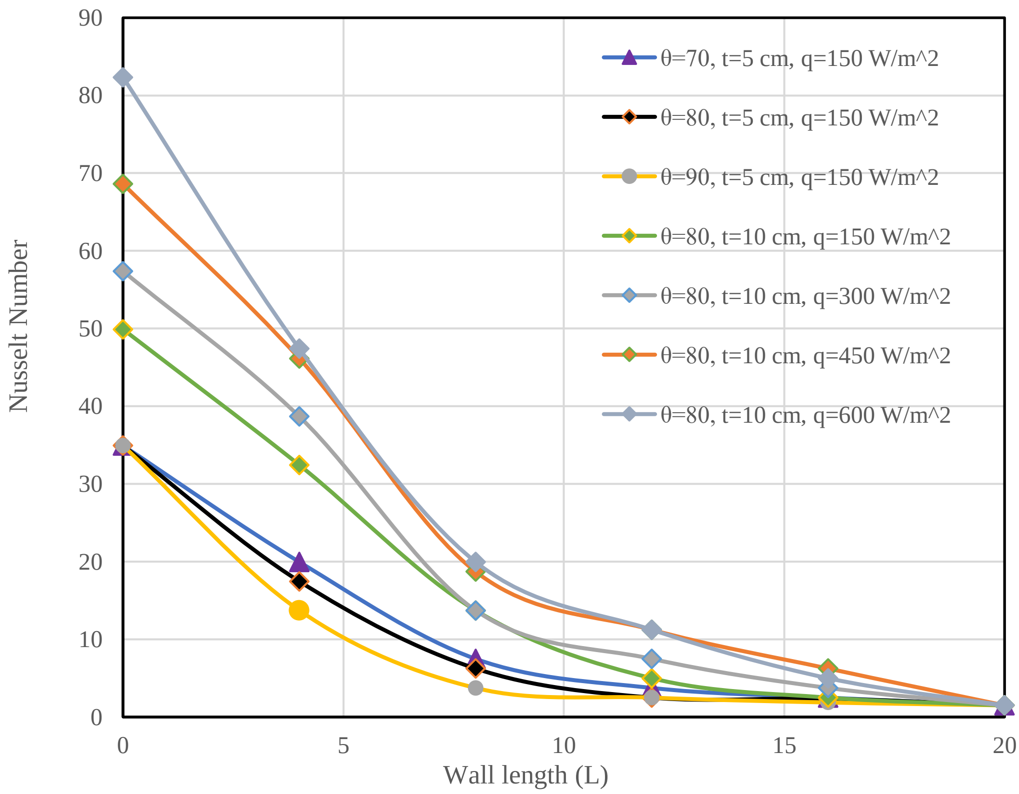


Figure 8. Local Nusselt number effect by heat flux, thickness of porous media and inclined angle

Figure 9 confirmed the effect of the heat flux on the values of average Nusselt number. It can be seen the direct relationship between them where increasing the heat flux will raise the average Nusselt number. This agrees with much of the previous research.

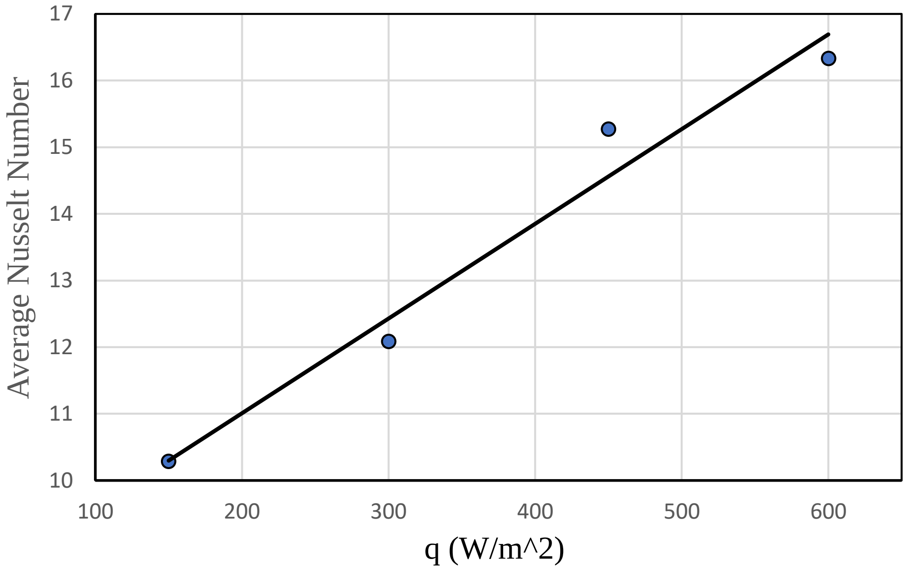


Figure 9. Effects of average Nusselt number with heat flux for q = 80º and t = 10 cm

4. Conclusions

A two-dimensional numerical analysis of Navier–Stokes system of equations have been done on a natural heat convection through a square rhombic cavity. Mass and energy equations was numerically solved with Darcy - Brinkman model and Simple algorithm using finite volume technique. The rate of heat transfer was calculated inside the cavity subjected to heat flux from the left side which coated with a copper metal foam and a constant cold temperature from the right side while the top and bottom walls were thermally insulated. The effect of heat flux, thickness of the porous media layer and the inclined angle of the cavity on Nusselt number and Rayleigh number were discussed. the following conclusions have been extracted from these investigation and discussion:

1. The existence of the porous media layer increase the heat transfer rate inside the cavity.
2. Increasing the thickness of the porous layer led to a rise in the rate of heat transfer so rise the average Nusselt number values. The enhancement of the heat transfer rate was 68.3%, 57.1%, and 41.3% when the thickness of the porous media was15 cm, 10 cm and 5 cm respectively, compared to without porous media.
3. Decreasing the inclined angle of the sidewall of the cavity from the perpendicular value will increase dominance of the conduction heat transfer over the convection heat transfer.

Increasing heat flux values will raise the average Nusselt number values thus increasing the heat transfer rate..

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