Effect Of Vertical Partition On Natural Convection Flow In Enclosure With Adjacent Inclined Single Wall

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Abstract :

In this paper a numerical study for the natural convection in an enclosure with adjacent single wall using vertical adiabatic partition attached to bottom wall. The vertical walls of the enclosure are isothermal while the bottom and top horizontal walls are insulated. The temperature of the left vertical wall is higher than the temperature of the right vertical wall. The study included a change in the inclination of the cold wall of ($f=15°$, $30°$ and $45°$), Rayleigh number of ($Ra=10^3$ to 10^5), length of partition of ($Y_P= 0.2$, 0.3 and 0.4) and location of partition of ($X_P=0.25$ and 0.5). A finite difference method was applied to solve the governing equations using Gauss-siedel iterative technique. Results for the mean Nusselt number, and contour maps of the streamlines and isotherms line are presented. It has been found that Nusselt number increases with an increase in Rayleigh number, inclination of cold wall with percentage of (10-16%) and location of partition with percentage of (3-14%) while decreases with increasing in partition length with percentage of (5.4-7%).

Key words: Enclosure , Adiabatic partition , Natural convection

تأثير الحاجز العمودي على جريان الحمل الطبيعي داخل حيز ذو جدار مائل

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الخلاصة

في هذا البحث تم إعداد دراسة عددية للحمل الطبيعي لحيز فيه احد الجدران المتجاورة مائل باستخدام حاجز عمودي ملاصق للجدار الأسفل. الجدارين العموديين للحيز لهم درجة حرارة ثابتة بينما الجدارين الأفقيين الأعلى والأسفل معزولين. درجة حرارة الجدار الأيسر أعلى من درجة حرارة الجدار الأيمن. تضمنت الدراسة تغيير في ميلان

الجدار البارد عند (°15 و °30 و °45) و عدد رالي (10^3 الى 10^5) و طول الحاجز عند (0.2 و 0.3 و 0.4) وموقع الحاجز عند (0.25 و 0.5) . طبقت طريقة الفروق المحددة لحل المعادلات المتحكمة باستخدام تقنية كاوس – سيدل المتناوبة قدمت النتائج بطريقة الخرائط الكنتورية لخطوط ثبوت درجات الحرارة وخطوط دالة الانسياب ومعدل عدد نسلت. أظهرت النتائج إن عدد نسلت يزداد بزيادة عدد رالي وزاوية ميلان الحاجز البارد بنسبة (10−16%) وموقع الحاجز بنسبة (14-3%) بينما يقل بزيادة طول الحاجز بنسبة (5.4−7%).

الكلمات المفتاحية: حيز ، حاجز أدياباتي ، الحمل الطبيعي

Nomenclature

Symbol	Definition	Dimensions
Cp	Heat capacity at constant pressure	J/kgK
k	Thermal conductivity of fluid	W/mK
L	Bottom horizontal and vertical lengths	M
L_1	Top horizontal length	M
Nu	Nusselt number = Q_{conv}/Q_{cond}	---
P	Fluid pressure	N/m^2
Pr	Prandtl numbers = ν/α	---
$Q_{cond.}$	Conduction heat transfer	W
$Q_{conv.}$	Convection heat transfer	W
Ra	Rayleigh number	---
t_p	Thickness of partition	M
T	Temperature	K
U	Velocity in x direction	m/s
V	Velocity in y direction	m/s
X	Horizontal distance	M
x_P	Distance of Partition location	M
X_P	Dimensionless partition location = x_p/L	---
Y	Vertical distance	
y_P	Partition length	M
Y_P	Dimensionless partition length = y_p/L	
ψ	Stream function	m^2/s
ω	Vorticity	---
Ω	Dimensionless velocity = $(\omega L^2 Pr/\nu)$	---
θ	Dimensionless temperature = $(T-T_c)/(T_h-T_c)$	---
ρ	Density	Kg/m^2

ν	Kinematic viscosity	m^2/s
ϕ	Angle of inclination of cold wall	
α	Thermal diffusivity $=k/(\rho Cp)$	m^2/s
β	Thermal expansion coefficient	K^{-1}
$\hat{\psi}$	Dimensionless stream function $= \psi Pr/\nu$	---
Subscript		
C	Cold	
H	Hot	

1. Introduction:

Natural convection flow and heat transfer in different geometrical enclosures have been the topic of many research engineering studies. These studies consist of various technological applications such as solar collectors, building heating and ventilation, cooling of electronic devices [1].

Numerical studies of natural convection heat transfer and flow in closed enclosures has been shifted in the last decade towards the study of natural convection in partitioned enclosures and those enclosures with discrete heat sources attached to its adiabatic walls. On the other hand, natural convection from heated elements within enclosures has received attention recently.

Zimmerman and Acharya [2] had studied numerically the natural convection heat transfer in a cavity with perfectly conducting horizontal end walls and finitely conducting baffles. Results were obtained at lower Rayleigh numbers, and no flow separation in front of partial divider was noted.

Acharya and Jetli [3] had investigated numerically the heat transfer and flow patterns in a partially divided differentially heated square box. Rayleigh numbers studied were in the range of 10^5- 10^6. The flow was weak in this stratified region and a tendency for flow separation behind the divider was noted.

Sun and Emery [4] studied numerically two-dimensional conjugate natural convection heat transfer in an enclosure containing discrete internal heat sources and an internally conducting baffle.

Türkoğlu and Yücel [5] also studied numerically fluid flow and heat transfer in partially divided square enclosures. It was observed that the mean Nusselt number increased with increasing Rayleigh number and decreased with increasing number of partitions; however, the decline in the mean Nusselt number was much less at low Rayleigh numbers. Increasing the partition height resulted in a decrease in the mean Nusselt number.

Dağtekin and Öztop [6] studied numerically the natural convection heat transfer and fluid flow of two heated partitions in a rectangular enclosure for Rayleigh number range of 10^4-10^6. The partitions', attached to the bottom wall, the length and the location were varied while the enclosure was cooled from two walls.

Bilgen [7] numerically studied 2D square differentially heated cavities, with a partition is attached on the active wall. The effects of Rayleigh number (10^4 to 10^9), dimensionless partition length (0.10 to 0.90), dimensionless partition position (0 to 0.90), and dimensionless conductivity ratio (10 to 60) were examined. It is found that Nusselt number increased with Rayleigh number and decreased with partition length and conductivity ratio, and an optimum partition position existed.

Altaç and Kurtul [8] numerically studied 2D natural convection in tilted rectangular enclosures with a vertically situated hot plate placed at the center. The plate was very thin and isothermal. The enclosure was cooled from a vertical wall only. Rayleigh numbers and the tilt angles of the enclosure ranged from 10^5 to 10^7 and from 0 to 90 degrees. The flow pattern and temperature distribution were analyzed, and steady-state plate-surface-averaged Nusselt numbers were correlated.

The present paper is concerned with a numerical study of the effect of inclined single side wall on heat transfer in enclosure. The enclosure is subjected to a boundary conditions as isothermal vertical wall, the left is hot and the right wall is adjacent inclined at cold temperature while the horizontal walls are insulated. An adiabatic vertical partition is attached to the lower horizontal insulated wall. The study include the effect of the inclination angle of the cold wall and the adiabatic partition (location and height).

2. Physical model:

The schematic configuration of the physical model is plotted in **Figure (1),** which show a two dimensional enclosure subjected to the boundary conditions, a isothermal vertical walls, the left wall is maintain at hot temperature (T_h) and the right wall is adjacent inclined at cold temperature (T_c) while the horizontal walls are insulated. An adiabatic vertical partition is attached to the lower insulated horizontal wall.
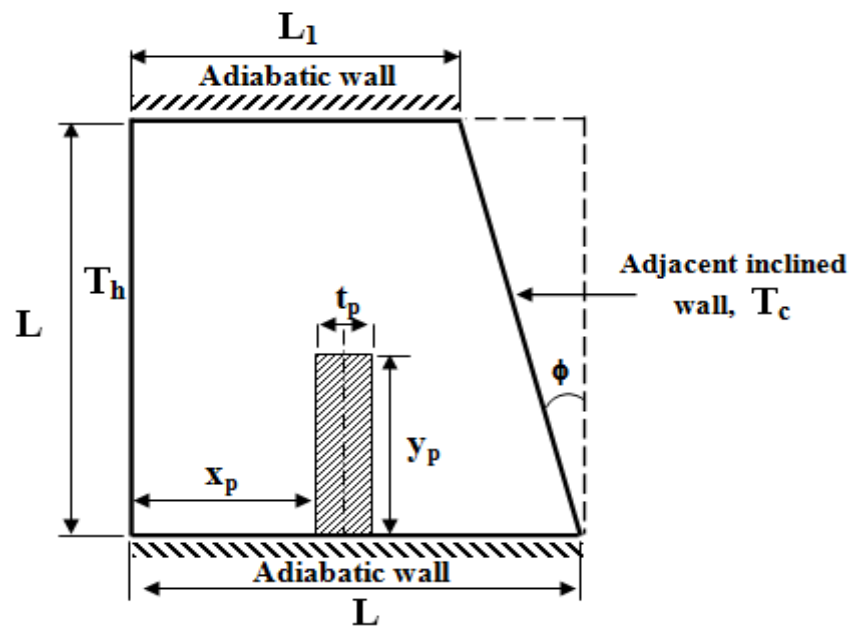

Fig. (1) Geometry of enclosure with adjacent inclined single wall with vertical partition

To simplify the analysis, the following assumptions is considered:

1. A two-dimensional distribution of fluid flow and temperature distribution .
2. The fluid is incompressible, steady and laminar.
3. The fluid is Newtonian and with Boussinesq approximation.
4. The fluid have constant properties.
5. The radiation effect is negligible.

The dimensionless governing equations in streamline-vorticity form can be obtained via introducing dimensionless variables as follows:

$$X = \frac{x}{L}, \quad Y = \frac{y}{L}, \quad \hat{\psi} = \frac{\psi \, Pr}{\nu}, \quad \Omega = \frac{\omega(L)^2 \, Pr}{\nu} \qquad \ldots(1)$$

$$\theta = \frac{T - T_c}{T_h - T_c}, \quad U, V = \frac{(u, v)L}{\alpha} \qquad \ldots(2)$$

$$u = \frac{\partial \psi}{\partial y}, \quad v = -\frac{\partial \psi}{\partial x}, \quad \omega = \left(\frac{\partial v}{\partial x} - \frac{\partial u}{\partial y} \right) \qquad \ldots(3)$$

$$Ra = \frac{\beta g (T_h - T_c) L^3 \, Pr}{\nu^2}, \quad Pr = \frac{\nu}{\alpha} \qquad \ldots(4)$$

Based on the dimensionless variables above governing equations (stream function, vorticity and energy equations) can be written as:

$$-\Omega = \frac{\partial^2 \psi}{\partial X^2} + \frac{\partial^2 \psi}{\partial Y^2} \qquad \ldots(5)$$

$$\frac{\partial^2 \Omega}{\partial X^2} + \frac{\partial^2 \Omega}{\partial Y^2} = \frac{1}{Pr} \left(\frac{\partial \hat{y}}{\partial Y} \frac{\partial \Omega}{\partial X} - \frac{\partial \hat{y}}{\partial X} \frac{\partial \Omega}{\partial Y} \right) - Ra \left(\frac{\partial q}{\partial X} \right) \qquad \ldots(6)$$

$$\frac{\partial^2 q}{\partial X^2} + \frac{\partial^2 q}{\partial Y^2} = \frac{\partial \hat{y}}{\partial Y} \frac{\partial q}{\partial X} - \frac{\partial \hat{y}}{\partial X} \frac{\partial q}{\partial Y} \qquad \ldots(7)$$

The boundary conditions are illustrated in the physical model and they can be defined as follows:

On the inclined wall:

U=0 , V=0 , $\theta = 0$

On the bottom and top walls:

U=0   ,   V=0   , $\dfrac{\partial q}{\partial y}=0$

On the hot wall:

U=0   ,   V=0   , $\theta = 1$

3. Numerical approach:

The governing equations for steady state, laminar, two dimensional natural convection heat transfer in an enclosure with an adiabatic partition are solved using finite difference method based on Taylor series. In this method, all partial derivatives in governing equations are converted to forms that can be solved with aid of computer.

The mesh consists of (M) in vertical and (N) horizontal lines positioned at intervals of (Δx) and (Δy), respectively. For any unknown variable points ϕ (x,y), the continuous first and second order derivatives at grid point (i,j), which are $(\partial\phi/\partial x)_{i,j}$ and $(\partial^2\phi/\partial x^2)_{i,j}$ respectively may be expressed in three ways:

The forward difference approximation for the one side three points, the first and second derivatives are:

$$\left(\frac{\partial\phi}{\partial x}\right)_{i,j} = \frac{-3\phi_{i,j} + 4\phi_{i+1,j} - \phi_{i+2,j}}{2\Delta x} + O(\Delta x)^2 \qquad \qquad \dots(8)$$

$$\left(\frac{\partial^2\phi}{\partial x^2}\right)_{i,j} = \frac{\phi_{i,j} - 2\phi_{i+1,j} + \phi_{i+2,j}}{(\Delta x)^2} + O(\Delta x)^2 \qquad \qquad \dots(9)$$

The backward difference approximation for the one side three points, the first and second derivatives are:

$$\left(\frac{\partial\phi}{\partial x}\right)_{i,j} = \frac{-3\phi_{i,j} + 4\phi_{i-1,j} - \phi_{i-2,j}}{2\Delta x} + O(\Delta x)^2 \qquad \qquad \dots(10)$$

$$\left(\frac{\partial^2\phi}{\partial x^2}\right)_{i,j} = \frac{\phi_{i,j} - 2\phi_{i-1,j} + \phi_{i-2,j}}{(\Delta x)^2} + O(\Delta x)^2 \qquad \qquad \dots(11)$$

The central difference approximation for the internal nodes can be written as:

$$\left(\frac{\partial\phi}{\partial x}\right)_{i,j} = \frac{\phi_{i+1,j} - \phi_{i-1,j}}{2\Delta x} + O(\Delta x)^2 \qquad \qquad \dots(12)$$

$$\left(\frac{\partial^2\phi}{\partial x^2}\right)_{i,j} = \frac{\phi_{i-1,j} - 2\phi_{i,j} + \phi_{i+1,j}}{(\Delta x)^2} + O(\Delta x)^2 \qquad \qquad \dots(13)$$

Where $O(\Delta x)^2$ is the truncation error of second order.

In the same way the derivatives in y-direction can be solved.

The heat transfer from the walls of the enclosure can be obtained using the integration of temperature gradient along the vertical wall.

After computing the heat transfer in the enclosure, Nusselt number can be calculated as follows:

$$Nu = \frac{Q_{conv.}}{Q_{cond.}} \qquad \qquad …(14)$$

After transferring Nusselt number to dimensionless form we get :

$$Nu = \int_0^1 \left(\frac{\partial \theta}{\partial x}\right)_{x=0} dy \qquad \qquad …(15)$$

In order to examine the accuracy with mesh size, the mesh size is varied in the range 40x40 to 120x120. Figure (2) shows the Nu values obtained. Optimum grid dimensions is chosen at 100x100.
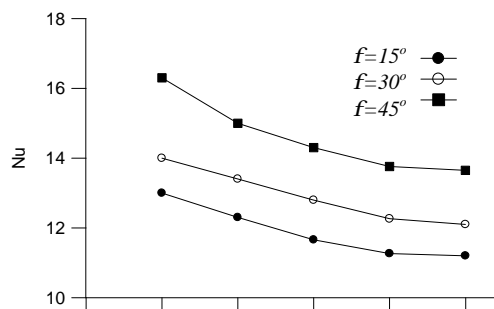

Fig. (2) variation of Nusselt no. with No. of Grid at Ra=10^5 , Xp=0.5, Yp=0.2

4. Results and discussion:

In this section, numerical results for the streamline and temperature contours for various values of the cold wall inclination angle θ=(15°, 30° and 45°), dimensionless length of partition of Y_P=(0.2, 0.3 and 0.4), dimensionless location of partition at X_P=(0.25 and 0.5) and Rayliegh number Ra=(10^3, 10^4 and 10^5) will be reported. All results are computed for a Prandtl number Pr=0.74. Furthermore, The results for the average Nusselt number will be presented and discussed.

The effects of Rayliegh number on the flow field and the temperature distribution are presented in **Figure (3)** for (10^3, 10^4 and 10^5) respectively at location of partition at X_P of (0.25), length of partition Y_P of (0.4) and inclination angle θ (15°). It is clear that the increase in Rayliegh number leads to increase in value of ψ_{max} in counter clockwise because the increase in flow velocity and then increasing in the natural heat transfer, also the boundary layer thickness near the hot and cold walls increased due to the increase in the buoyancy force. The effects of the isotherm is to be clear near the lower part of the hot wall and the temperature gradient will be increased because the high difference in the fluid density, this causes increasing in the bouncy force and more heat transferred to the fluid to the cold wall.


(a)   (b)   (c)
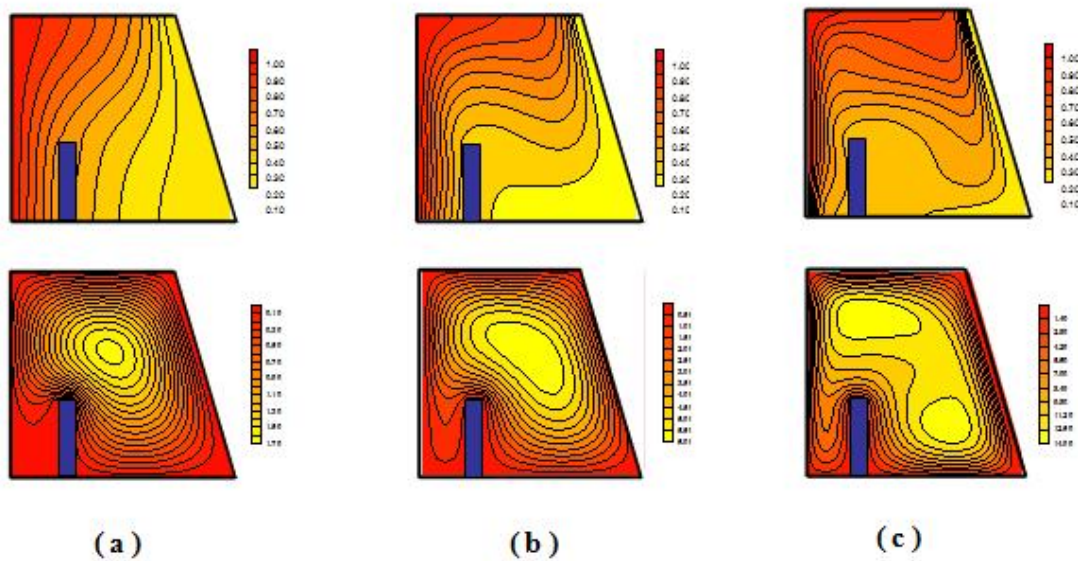
**Fig. (3,a-b-c) Isotherms lines & Stream functions for   Xp = 0.25 , Y$_P$=0.4, f=15°
(a) Ra=10^3 , (b) Ra=10^4 , (c) Ra=10^5**

Figure (4,a-b-c) represents the effect inclination of the cold wall θ=(15°, 30° and 45°) at constant location of partition at X_P of (0.5) and length of partition Y_P of (0.4) and Rayliegh number Ra of (10^5). For the inclination of (15°), it is clear that the flow circulate towards the cold wall and continue over the partition. As the inclination becomes (30°), the enclosure will be a trapezoidal shape. This mean less flow being behind the partition and most of the flow move toward the hot wall. Finally, when the inclination of the cold wall is being (45°), the enclosure convert to triangular shape and no flow behind the partition, as it being clear, to the adjacent wall and flow return directly to the hot region. As an effect those flow conditions, the increase of the inclination angle the isotherm lines pushed towards the left zone between the hot and the cold walls and less temperature gradient behind the partition.
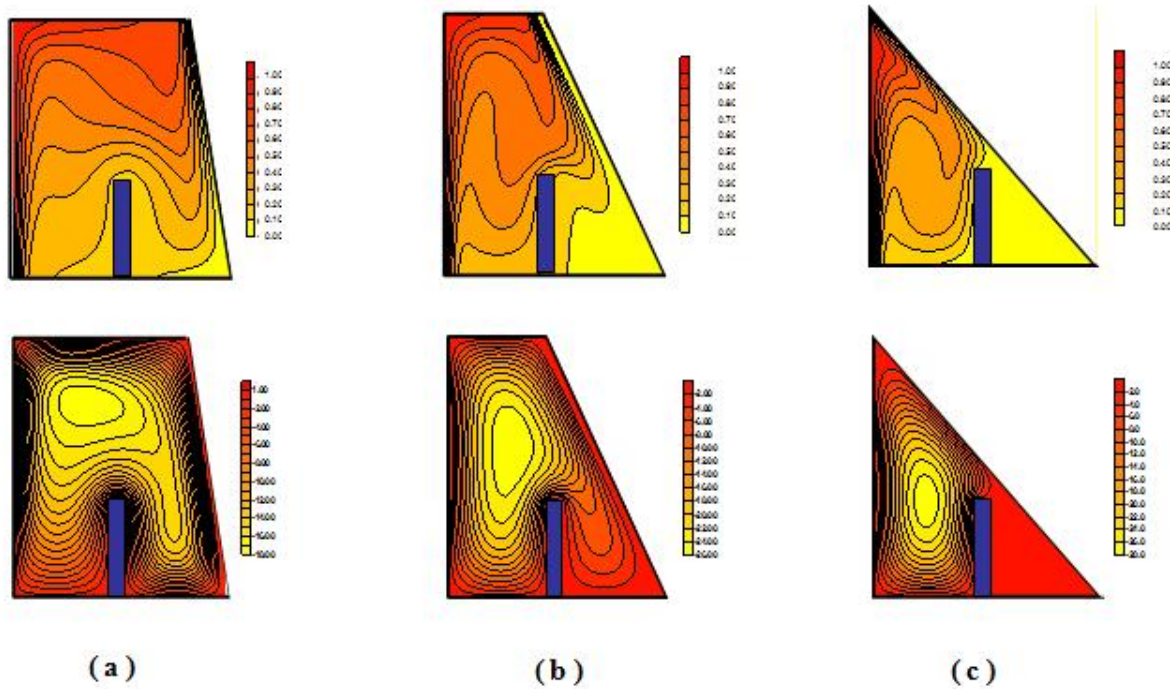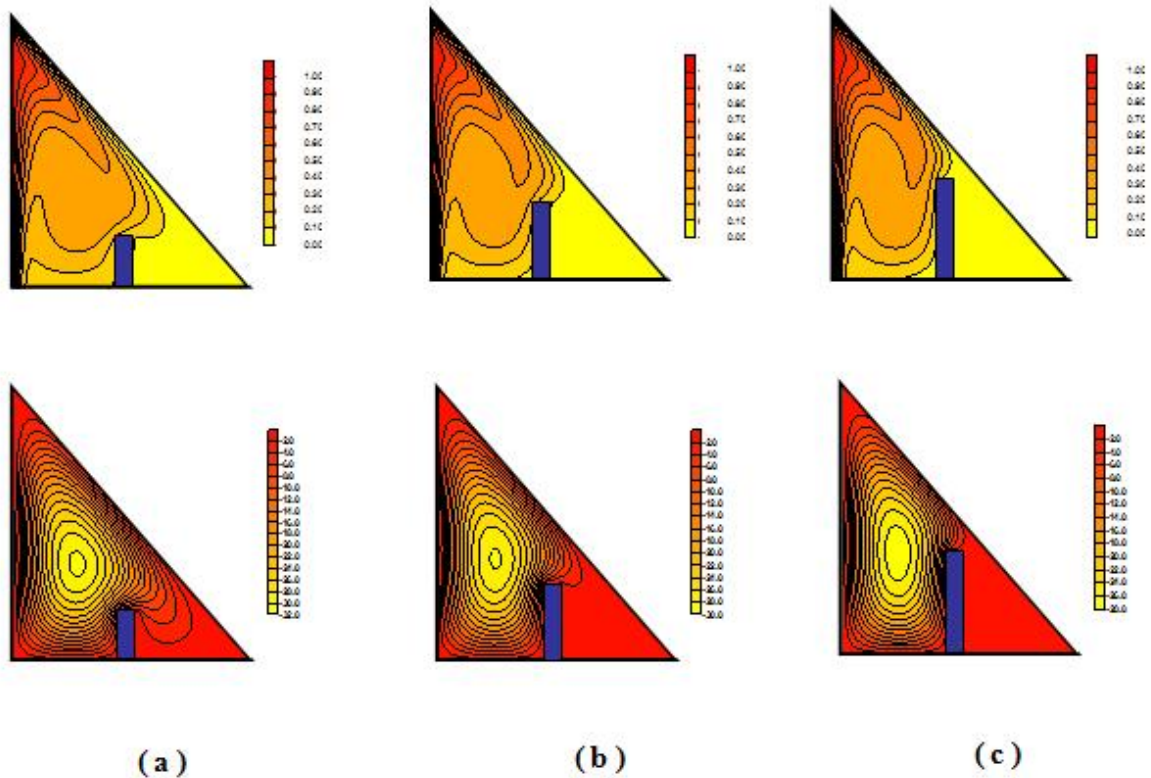
Fig. (4,a-b-c) Isotherms lines & Stream functions for   Xp = 0.5 , Y_P=0.4, Ra=10^5
(a) f=15° , (b) f=30° , (c) f=45°

The effect of location of partition shown in the **Figures (5, A-B)** for X_P of (0.25 and 0.5) at inclination of (30°), length of partition Y_P of (0.4) and Rayliegh number of (10^5). For the partition location of X_P of (0.5) most of the circulation is near to the cold wall and a two cells circulation occur at the core due to the variation of flow velocity in the upper and lower zone due to temperature variation caused by the partition. As the partition moved toward the cold wall (X_P) of 0.5, the stream lines becomes close to the hot wall and over the partition near the cold wall with single cell flow and less fluid motion behind the partition as the space being smaller. The change in the partition location leads to increase the value of maximum stream function as the circulation zone become smaller. The isotherms affected by this variation of flow velocity and less stratification in temperatures and less temperature gradient in temperatures behind the partition as it become close to the adjacent cold wall.
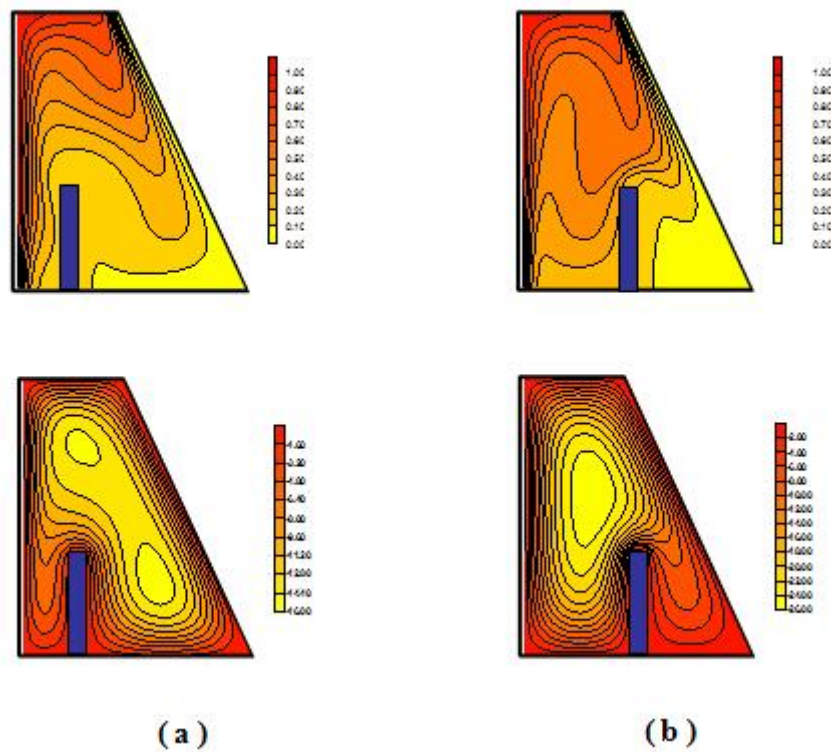
Figure (5,a-b) Isotherms lines & Stream functions for $f=30°$, $Y_P=0.4$, $Ra=10^5$
(a) Xp = 0.25, (b) Xp = 0.5

Figure (**6, a-b-c**) show the effect of partition length Y_P as (0.2,0.3 and 0.4) at a location X_P of (0.5), inclination angle ϕ of (45°) and Rayliegh number of (10^5). For all cases most of the flow are between the hot wall and the cold wall over the partition. Increasing the partition length make the space between cold wall and partition narrower ,which prevent fluid motion behind the partition . The isotherms deflect upward towards the hot wall and the thermal boundary layer increase near the hot and the cold wall as the partition elongated. The reduction of flow behind the partition with its elongation cause a decrees in the temperature differences of the fluid .

Fig. (6,a-b-c) Isotherms lines & Stream functions for   Xp = 0.5 , f=45°, , Ra=10^5
(a) Y$_P$=0.2 , (b) Y$_P$=0.3 , (c) Y$_P$=0.4

The Nusselt number will be considered as an indication of the heat transfer rate for different parameter considered . The Rayliegh number variation demonstrates a significant effect on Nusselt number. These effects are presented in **figure (7)** which shows the variation of Nusselt number with Rayliegh number for different parameters studied. All figures indicate that the Nusselt number increased when Rayliegh number increase for all studied cases. This may be due to the increase in the fluid flow rate and eventually would the heat transfer rate. The figures also show the effect of different partition location (X$_P$=0.25 and 0.5), partition length (Y$_P$=0.2,0.3 and 0.4), inclination of cold wall (ϕ=15°, 30° and 45°) at different Rayliegh number (Ra=10^3 to 10^5) .

Figure (7-a) shows the effect of partition location at length Y$_P$ of (0.2) and inclination angle of ϕ of (15°, 30° and 45°), the increase in the partition location leads to enlarge the area of the fluid which result in heat transfer rate increase and ultimately leading to an increase in Nusselt number as a ratio (3%-14%) depending on Rayligh number and inclination angle ϕ. At ϕ = 15°, the percentage of Nu increased from (4% to 11%) at Rayligh number varied from 1000 to 10^5 . At ϕ = 30° the percentage of Nu increased from (4%- to 2.6%) at Rayliggh number varied from 1000 to 10^5 . At ϕ = 30° the percentage of Nu increased from (4% to 14%) at Rayliggh number varied from 1000 to 10^5 . The increase in the inclination angle ϕ cause an increase in the flow velocity due to the reduction in the flow behind the partition.

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Figure (7-b) shows the variation of Nusselt number with Rayliegh number and the effect of variation of partition length Y_P of (0.2,0.3 and 0.4) at X_P of (0.5) and different inclination angle ϕ. It was seen that the Nusselt number decrease as the partition height increase, causing a reduction in heat transfer area at the cold wall and more stagnation region behind the partition. The reduction ratio of Nusselt number is (5.4% - 7%)depending on location of partition at high Rayligh number 10^5 . The reduction ratio of Nusselt number is very small at low Raghligh Number . However, the inclination angle increase will cause the Nusselt number increase too due to velocity increase caused by the significant change of the buoyancy force.

Figure (7-c) represents the effect of inclination angle of the cold wall ϕ of (15°, 30° and 45°), for fixed value of partition location of (X_P=0.25) and different value of partition length of (Y_P=0.2, 0.3 and 0.4). It can be seen that the highest Nusselt number are obtained at inclination (ϕ=45°) and (Y_P=0.2). However, it was found that Nusselt number decrease as Y_P increase. Inclination angle decrease for all Rayliegh numbers. This effect has been discussed before that increase the angle cause an increase in flow velocity and more heat transfer while the elongation of the partition will reduce cold area to transfer heat and reduce Nusselt number.
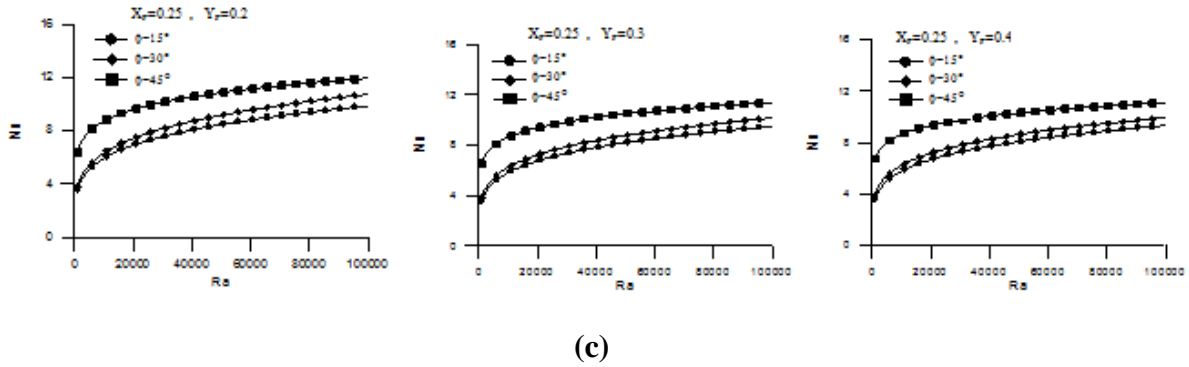

(a)

(b)

(c)
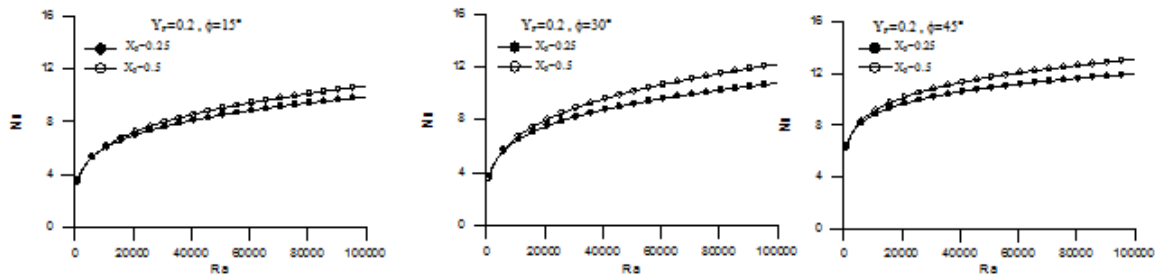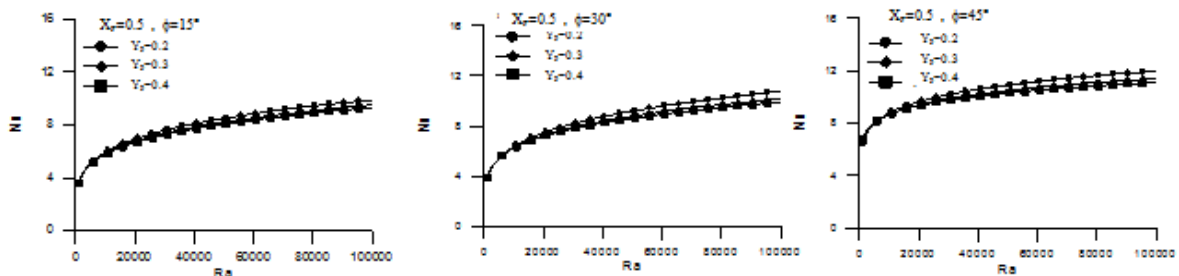
Fig. (7,a-b-c) Variation of Nusselt number with Rayliegh number for different partition length , different location and different inclination of cold wall

5. Conclusions:

The results obtained from this study indicated many conclusions:

1- Nusselt number increases with increasing the Rayliegh number in all cases.

2- Nusselt number increases with increasing the inclination of cold wall as a percentage of(10%-16%) depending on Rayligh number .

3- Nusselt number increases with increasing the location of partition as a percentage of (3%-14%) .

4- Nusselt number decreases with increasing the partition length as a percentage of (5.4% - 7%).

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