Al-Khwarizmi Engineering Journal, Vol. 5, No. 2, PP 32 - 43 (2009)

Al-Khwarizmi Engineering Journal

Simulation of Free Convection from an Inclined Heated Thin Plate in a Square Enclosure

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(Received 25 November 2008; accepted 7 April 2009)

Abstract

Simulation of free convection heat transfer in a square enclosure induced by heated thin plate is represented numerically. All the enclosure walls have constant temperature lower than the plate's temperature. The flow is assumed to be two-dimensional. The discretized equations were solved stream function, vorticity, and energy equations by finite difference method using explicit technique and Successive Over- Relaxation method. The study was performed for different values of Rayleigh number ranging from 10^3 to 10^5 for different angle position of heated thin plate(0°, 45°, 90°). Air was chosen as a working fluid (Pr = 0.71). Aspect ratio of center of plate to the parallel left wall $\boldsymbol{A_2}$ take a constant and is equal to 0.5. The effect of the angle position of the heated thin plate on heat transfer and flow were addressed. With the increase of Rayleigh number heat transfer rate increased in both vertical and horizontal position of the plate. For the vertical situation (γ=90°) of thin plate, heat transfer becomes more enhanced than for the horizontal situation (γ=0°) and the inclined situation (γ=45°) especially when the value of aspect ratio $\boldsymbol{A_1}$ is equal to 0.25.

Keywords: *Free Convection, Heat Transfer, Inclined Heated Thin Plate, Square Enclosure.*

1. Introduction

Enhancement and depression of heat transfer through a differentially heated cavity are of significance for industrial applications such as solar collectors and nuclear reactors. Considerable studies have been devoted to the problems of this aspect. Many techniques of enhancing or depressing heat transfer have been experimentally and numerically investigated. One technique which has attracted significant research attention is to place a fin on the sidewall in order to enhance or depress heat transfer. Ahmed and Yovanovich [1] used numerical finite difference technique based on the Marker and Cell (MAC) method to obtain solutions of a two-dimensional model of a square enclosure with laminar natural convection heat transfer from discrete heat sources. A discrete heat source is located in the center of one vertical side representing a high-power integrated circuit (IC). The conservation equations are solved using the primitive variables: velocity, pressure, and temperature. Computations are carried out for Pr = 0.72. Bilgen and Oztop [2] investigated a numerical study on inclined partially open square cavities, which are formed by adiabatic walls and a partial opening. The surface of the wall inside the cavity facing the partial opening is isothermal. Steady-state heat transfer by laminar natural convection in a two dimensional partially open cavity is studied by numerically solving equations of mass, momentum and energy. Streamlines and isotherms are produced; heat and mass transfer is calculated. Rossano and Saulo [3] investigated the heat transfer coefficient "h" of a isothermal vertical plate with H=0.15 m. The neighborhood surface influence in that coefficient is aimed at simulation and standard experimentation. A novel technology to measure the heat flux, called "Tangential Heat Flux meter" is applied and simulation with a CFD commercial code was performed. Five heat flux meters were glued on the vertical plate, heated 20°C over the air temperature. The neighborhood and air temperature were maintained constant. The distance between the plate and the base wall (floor) was changed as well as the distance between the plate and the back side wall. Simulation results will be compared with experimental. The result expected is an increasing of heat transfer coefficient, very useful in heat exchange devices. Kandaswamy and others [4] studied the buoyancy induced flow and heat transfer inside a square cavity due to a heated thin plate placed vertically/horizontally. The flow motion in cavity depends on the heated plate itself. For an increase in Grashof number, heat transfer increased in both the vertical and the horizontal situations. As the aspect ratio of heated thin plate is increased the heat transfer also increases. Heat transfers become more enhanced in vertical situation than in horizontal situation. Nithyadevi and. Kandaswamy [5] made a numerical study on the effect of aspect ratio on the natural convection of a fluid contained in a rectangular cavity with partially thermally active side walls. The active part of the left side wall is the higher temperature than that of the right side wall. The top and bottom of the cavity and inactive part of the side walls are thermally insulated. Nine different relative positions of the active zones are considered. The equations are discretized by the control volume method with a power law scheme and are solved numerically by the iterative method together with a successive over relaxation (SOR) technique. The results are obtained for Grashof numbers between 10^3 and 10^5 and the effects of the aspect ratio on the flow and temperature fields and the rate of heat transfer from the walls of the enclosure are presented.

The heat transfer rate is high for the bottom–top thermally active location while the heat transfer rate is poor in the top–bottom thermally active location. The heat transfer rate is found to increase with an increase in the aspect ratio. Xu and others [6] investigated Direct numerical simulations of unsteady natural convection in a differentially heated cavity with a thin fin of different lengths on a sidewall at the Raleigh number of 3.8×10^9. It is found that the fin length significantly impacts the transient thermal flow around the fin and heat transfer through the finned sidewall in the early stage of the transient flow development. The results also indicate that the oscillations of the thermal flow around the fin in the quasi-steady stage are very sensitive to the length of the fin. Avedissian and Naylor [7] investigated a numerical study for the free convection in a tall vertical enclosure with an internal louvered metal blind, the study considers the effect of Rayleigh number, enclosure aspect ratio, and blind geometry on the convective heat transfer. The numerical model has been validated against experimental measurements and the results have been presented in terms of an empirical correlation for the average Nusselt number; the correlation is applicable to an internal metal blind. The Nusselt number correlation can be combined with a simple one-dimensional model to closely predict the value of dimensionless velocity U. The aim of the present work is to study free convection in an enclosure with an inclined heated thin plate located at three situations(by changing angle position of the plate (0°, 45°, 90°) for different positions by changing the aspect ratio of plate position A_1.

2. Mathematical Formulation and Boundary Conditions

A schematic diagram of the two-dimensional square enclosure of length and height L filled with a fluid under investigation is shown in Fig. 1. The top and bottom walls of the enclosure are adiabatic and the two vertical walls have constant temperature T_c lower than plate's temperature T_h. The Cartesian coordinates (x, y) with the corresponding velocity components (u, v) are indicated in Fig.1.a. The gravity g acts normal to the y-direction. In figure 1.b, h is the length of plate, h_1 the distance of the center of plate to the perpendicular lower wall, and h_2 is the distance of the center of plate to the parallel left wall. Representing the position through Cartesian coordinate system and assuming all other fluid properties to be constant, the flow of an incompressible Boussinesq viscous fluid under the above specified geometrical and physical conditions is governed by the equations:

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

$$u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}=-\frac{1}{\rho_o}\frac{\partial p}{\partial x}+\nu\nabla^2 u, \qquad \ldots(2)$$

$$u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}=-\frac{1}{\rho_o}\frac{\partial p}{\partial y}+\nu\nabla^2 v-\frac{\rho}{\rho_o}g, \qquad \ldots(3)$$

$$u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}=\alpha\nabla^2 T, \qquad \ldots(4)$$

where

$$\rho=\rho_o\left[1-\beta\left(T-T_c\right)\right] \qquad \ldots(5)$$

The appropriate boundary conditions are:

- u=v=0, $T=T_c$ at the lower and upper walls and
- u=v=0, $T=T_h$ for thin hated plate.

Introducing the following non-dimensional variables
$\Psi=\frac{\psi}{\upsilon}$, $\zeta=\frac{\omega}{\upsilon L^2}$, $U=\frac{u}{\upsilon L}$, $V=\frac{v}{\upsilon L}$, $X=\frac{x}{L}$, $Y=\frac{y}{L}$, $\theta=\frac{T-T_c}{T_h-T_c}$, with T_h > T_c, we get the vorticity-stream function formulation of the above problems (2)-(3) as

$$\nabla^2\Psi=-\zeta \quad \ldots(6)$$

$$\frac{\partial\Psi}{\partial Y}\frac{\partial\zeta}{\partial X}-\frac{\partial\Psi}{\partial X}\frac{\partial\zeta}{\partial Y}=\Pr\nabla^2\zeta+Ra\cdot\Pr\cdot\frac{\partial\theta}{\partial X} \quad \ldots(7)$$

$$\frac{\partial\Psi}{\partial Y}\frac{\partial\theta}{\partial X}-\frac{\partial\Psi}{\partial X}\frac{\partial\theta}{\partial Y}=\nabla^2\theta \quad \ldots(8)$$

We can obtain the horizontal and vertical velocities (U and V) from

$$U=\frac{\partial\Psi}{\partial Y},\quad V=-\frac{\partial\Psi}{\partial X} \quad \ldots(9)$$

With

$$\xi=\frac{\partial V}{\partial X}-\frac{\partial U}{\partial Y} \quad \ldots(10)$$

The boundary conditions in the dimensionless

- Ψ= θ =0 at the all walls and thin plat
- ζ=0 and θ=1 at the thin plat

and at all walls, we can get the value of vorticity by using

$$\xi=-\frac{\partial^2\Psi}{\partial n^2} \quad \ldots(11)$$

where ***n*** is the normal direction on the wall.

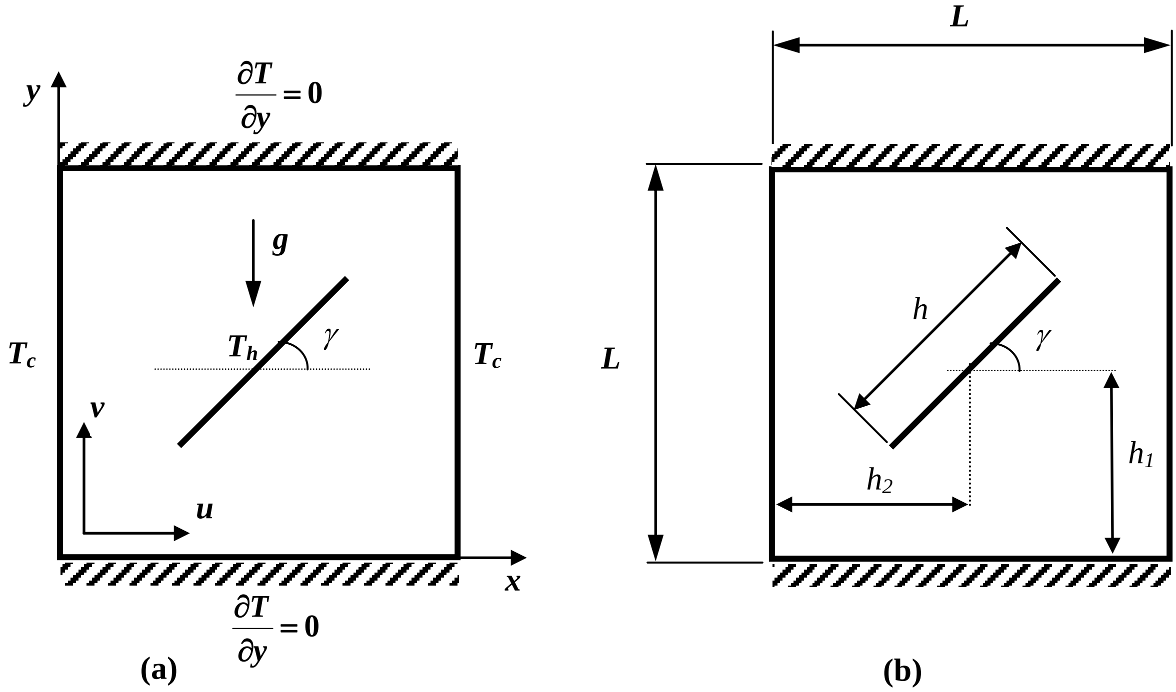


Fig.1. Geometric of Present Work and Boundary Conditions

The non-dimensional parameters that appear in the equations are ***Ra,*** the Rayleigh number, ***Pr,*** the Prandtl number, ***g,*** the acceleration due to gravity, ***υ,*** the kinematic viscosity, ***α,*** the thermal diffusivity, ***β*** coefficient of thermal expansion, and ***θ*** dimensionless temperature.
The local Nusselt number is defined by Nu_X

$$Nu_X = \frac{Q\,L_{plate}}{k(T_h - T_c)} \text{ , } L_{plate}=h=A*L \qquad \ldots(12)$$

Where Q is the total heat transfer from thin heated plate

$$Q = q_{tup} + q_{t\,down} \qquad \ldots(13)$$

$$Nu_X = \frac{0.5*(q_{tup} + q_{t\,down})}{k(T_h - T_c)} \qquad \ldots(14)$$

$$\Rightarrow Nu_X = 0.5*\left(Nu_{X\,up} + Nu_{X\,down}\right) \qquad \ldots(15)$$

– For the horizontal position

$$Nu_{X\,up} = -\left.\frac{\partial\theta}{\partial Y}\right|_{thin\,plate}^{+} \qquad \ldots(16)$$

$$Nu_{X\,down} = -\left.\frac{\partial\theta}{\partial Y}\right|_{thin\,plate}^{-} \qquad \ldots(17)$$

$$Nu_{av} = \int^{h} Nu_X\, dX \qquad \ldots(18)$$

– For the inclined position

The heat transfer from nodes source (on the heated thin plate) is shown in fig.(2)

$q_{t\,up} = q_X + q_Y$ and $q_{t\,down} = q_X + q_Y$

By using equation of Nusselt number

$$\Rightarrow Nu_{X\,up} = -1\left.\left(\left(\frac{\partial\theta}{\partial X}\right) + \left(\frac{\partial\theta}{\partial Y}\right)\right)\right|_{inclined\,thin\,plate}^{+} \qquad \ldots(19)$$

$$\Rightarrow Nu_{X\,down} = -1\left.\left(\left(\frac{\partial\theta}{\partial X}\right) + \left(\frac{\partial\theta}{\partial Y}\right)\right)\right|_{inclined\,thin\,plate}^{-} \qquad \ldots(20)$$

$$Nu_{av} = \int^{h} 0.5*\left(Nu_{up} + Nu_{down}\right)dX \qquad \ldots(21)$$

– For the vertical position

$$Nu_{X\,right} = Nu_{X\,left} = -\left.\frac{\partial\theta}{\partial X}\right|_{thin\,plate} \qquad \ldots(22)$$

$$Nu_Y = 0.5*\left(Nu_{Y\,right} + Nu_{Y\,left}\right) \qquad \ldots(23)$$

$$Nu_{av} = \int^{h} Nu_Y\, dY \qquad \ldots(24)$$

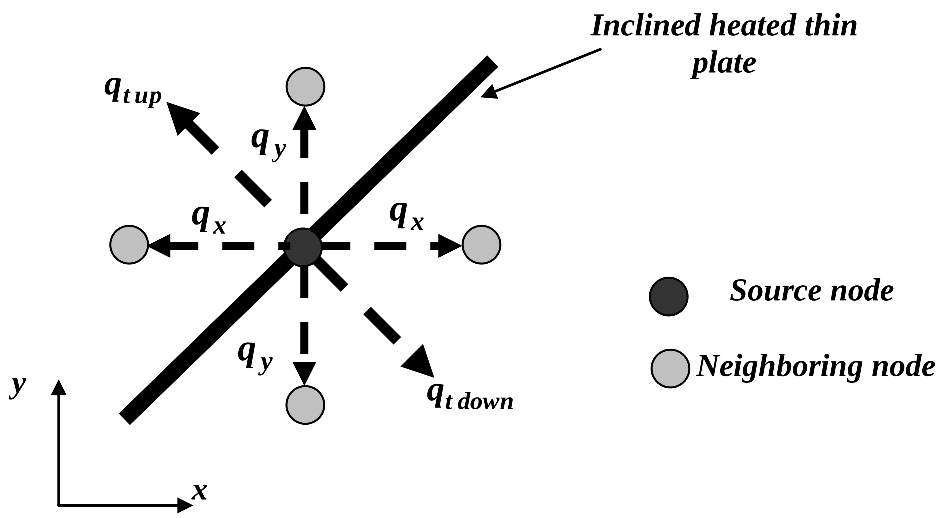


Fig.2. Heat Transfer from Inclined Thin Plate

3. Method of Solution

A finite difference method based on successive over relaxation iterative method is used to solve numerically the non-dimensional governing equations (6)–(8). A computational program was written in Fortran-90 language to compute the values of the required variables. The region of interest was covered with vertical and horizontal uniformly spaced grid lines equal to N. The vorticity and temperature distributions are obtained from Eqs. (7) and (8), respectively. The stream function distribution was obtained from Eq. (6) using successive over relaxation (SOR) and a known vorticity distribution. An iterative process is employed to find the stream function, vorticity and temperature fields. The process is repeated until the following convergence criterion is satisfied

$$\left|\frac{\phi new_{i,j} - \phi old_{i,j}}{\phi new_{i,j}}\right| \leq 10^{-5} \qquad \ldots(25)$$

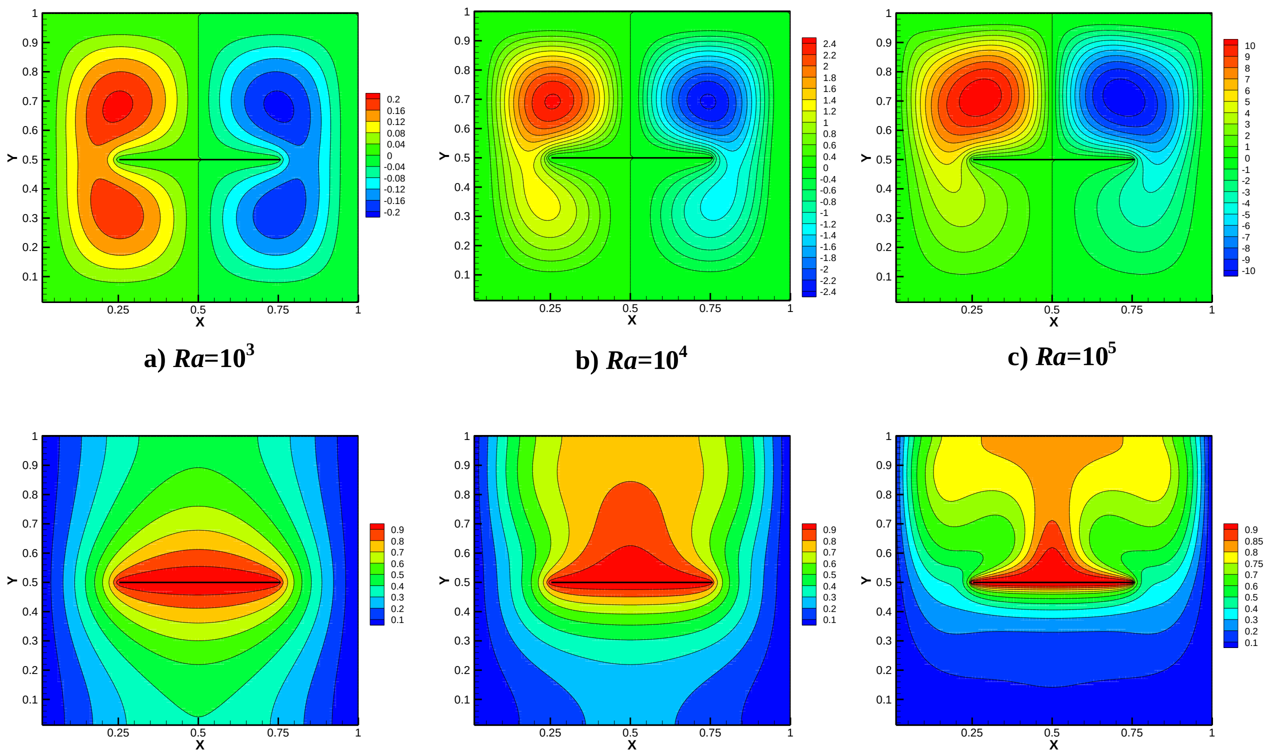
The numerical solutions are found for different grid systems from 100 x 100 to 120 x 120.

4. Results and Discussion

Free convection of low Prandtl number fluid (0.71) corresponding to air is investigated numerically in the presence of a heated thin plate. The computations are carried out for a wide range of Rayleigh number Ra varying from 10^3 to 10^5. The study was conducted for three different positions of the thin heated plate (by change $\boldsymbol{A_1}$) located at horizontal ($\gamma=0°$), vertical ($\gamma=90°$) and inclined ($\gamma=45°$) positions for all the cases which studied the value of aspect ratio $\boldsymbol{A_2}$ (aspect ration of the center of plate to the parallel left wall) equal to 0.5. The results are depicted as streamlines and isotherms plots. Hence the problem is symmetrical about the center of the enclosure for the horizontal and vertical positions. We have plotted both isotherms and streamlines in to study the effect of the different locations of the plate. The rate of heat transfer across the enclosure is calculated in terms of the local and average Nusselt number.

4.1. Thin Plate Located Horizontally

When the heated thin plate is located horizontally the results are obtained by changing the Rayleigh numbers and aspect ratio of the position of the plate ($\boldsymbol{A_1}$).

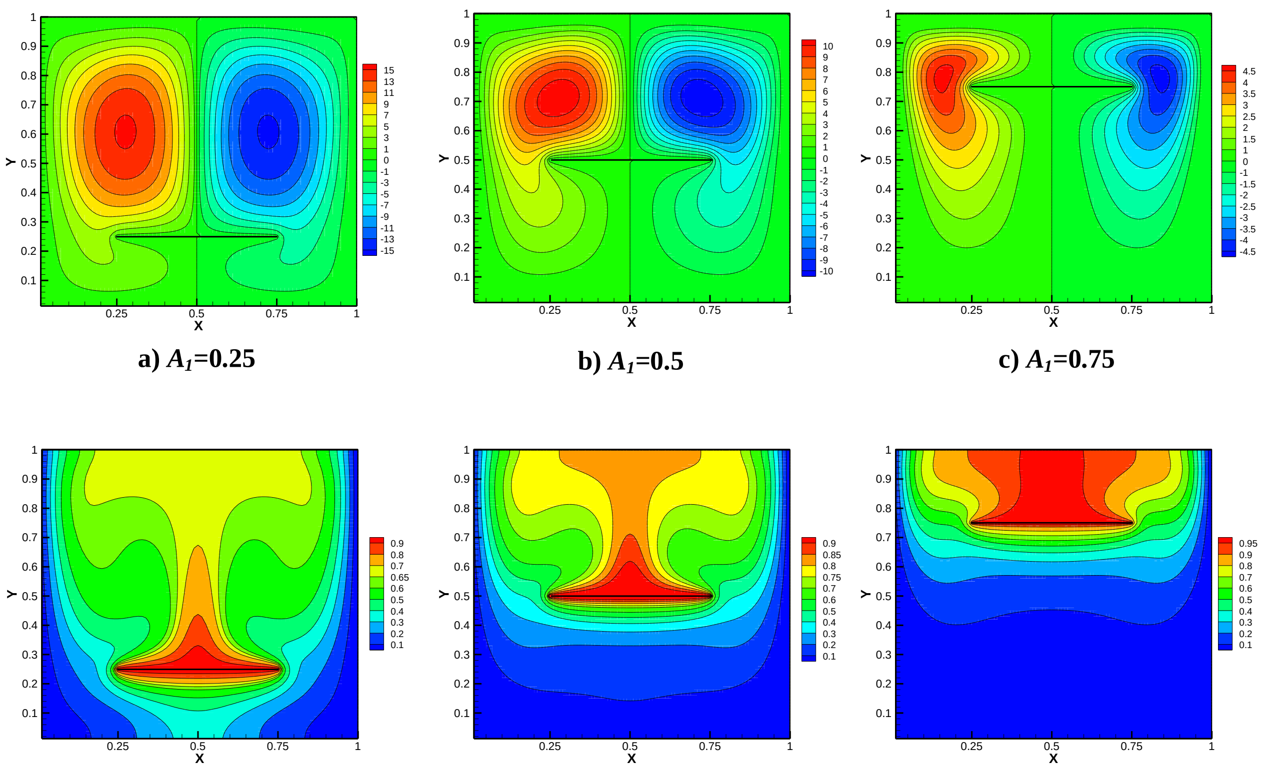


a) $Ra=10^3$ b) $Ra=10^4$ c) $Ra=10^5$

d) $Ra=10^3$ e) $Ra=10^4$ f) $Ra=10^5$

Fig.3. Effect of Rayleigh Number on the Stream Lines and Temp. Distrib. for Horiz. Location (A=0.5, A_1=0.5)

Fig.3. shows the streamlines and isotherms for $\boldsymbol{A}$=0.5 and the plate is located at the center ($\boldsymbol{A_1}$=0.5), for different values of Rayleigh numbers. It is observed that four cell centers are formed in the enclosure when Rayleigh number is small. Fluid motion is symmetric in the enclosure due to dominance of conduction mechanism. As Rayleigh number increases convection becomes stronger and rotation over the plate becomes stronger in comparison to the one below the plate. In addition, as the space between the heated vertical boundary and plate is small, the rotation of fluid downward is prevented. The fluid flow below the plate produces stagnation point depending on the increase of Rayleigh number. When looking at temperature distribution, it is observed that the temperature gradient becomes steeper and bigger at the top side of vertical boundaries depending on Rayleigh number. As cooling process is symmetric, flow and temperature disturbances show asymmetric behavior according to X-axis. Fig.4. shows the effect of aspect ratio of the location of the thin plate on the streamline and temperature distributions for $\boldsymbol{A}$=0.5, and Ra=10^5, for $\boldsymbol{A_1}$=0.25; the resulting flow pattern is found to be bi-cellular and the corresponding isotherms show a weak convection type. As the plate is closer to the bottom wall the fluid under the plate is heated moderately and temperature gradient is steeper along the vertical side walls. When the position is changed ($\boldsymbol{A_1}$=0.5), the flow pattern remains the same, but corresponding isotherms show that convection becomes strong and when ($\boldsymbol{A_1}$=0.75), the convection becomes very strong. The heat transfer is increased to decrease the aspect ratio of plate and the convection mechanism is very effective. As can be seen, when the heated thin plate is on the upper of enclosure, the fluid under the thin plate is not affected considerably and recirculation is restricted to the upper corner. Another expected result is that if the hot plate is on the upper wall of the enclosure there is no motion under the enclosure at a very low small motion relatively small motion, compared to the motion over the plate.



a) A_1=0.25 **b) A_1=0.5** **c) A_1=0.75**

d) A_1=0.25 **e) A_1=0.5** **f) A_1=0.75**

Fig.4. Effect of Aspect Ratio on the Stream Lines and Temperature Distribution for Horiz. Location (Ra=10^5, A=0.5).

4.2. Thin Plate Located Inclined (γ=45°)

In this case calculations were performed to see the effect of Rayleigh number in an enclosure with an inclined heated thin plate *(γ=45°)* built in for different aspect ratio of location thin plate. Figure(5) shows stream lines and temperature distributions for $\boldsymbol{A}$=0.5, $\boldsymbol{A_1}$=0.5 and Rayleigh number change from 10^3 to 10^5. As Rayleigh number is small conduction, mechanism is dominant. As it is seen when Rayleigh number is small, the distribution of stream lines up and below the thin plate will be approximately symmetric as shown in figure (5.a). As increase in Rayleigh number shows non symmetrical behavior, it is note the maximum value of stream lines is up of thin plate because the thin plate will not obstruct the flow above the plate but will obstruct the flow lower the plate, also we note that the weak convection will decrease with increasing the angle of position. Figure (5.d, e & f) shows the isotherms, as Rayleigh number increased the change of temperature distribution above the inclined plate increases more than the change of below the plate because of the high convective flow above the plate.

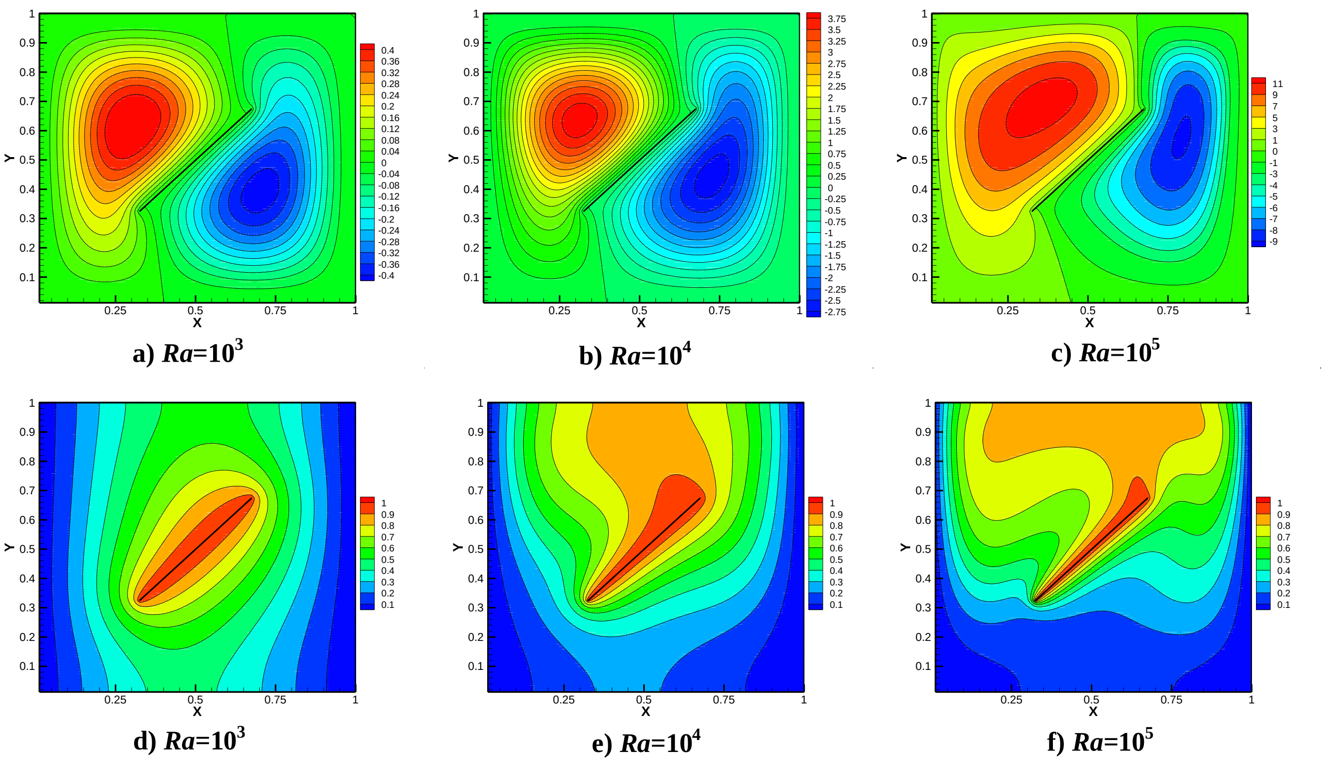
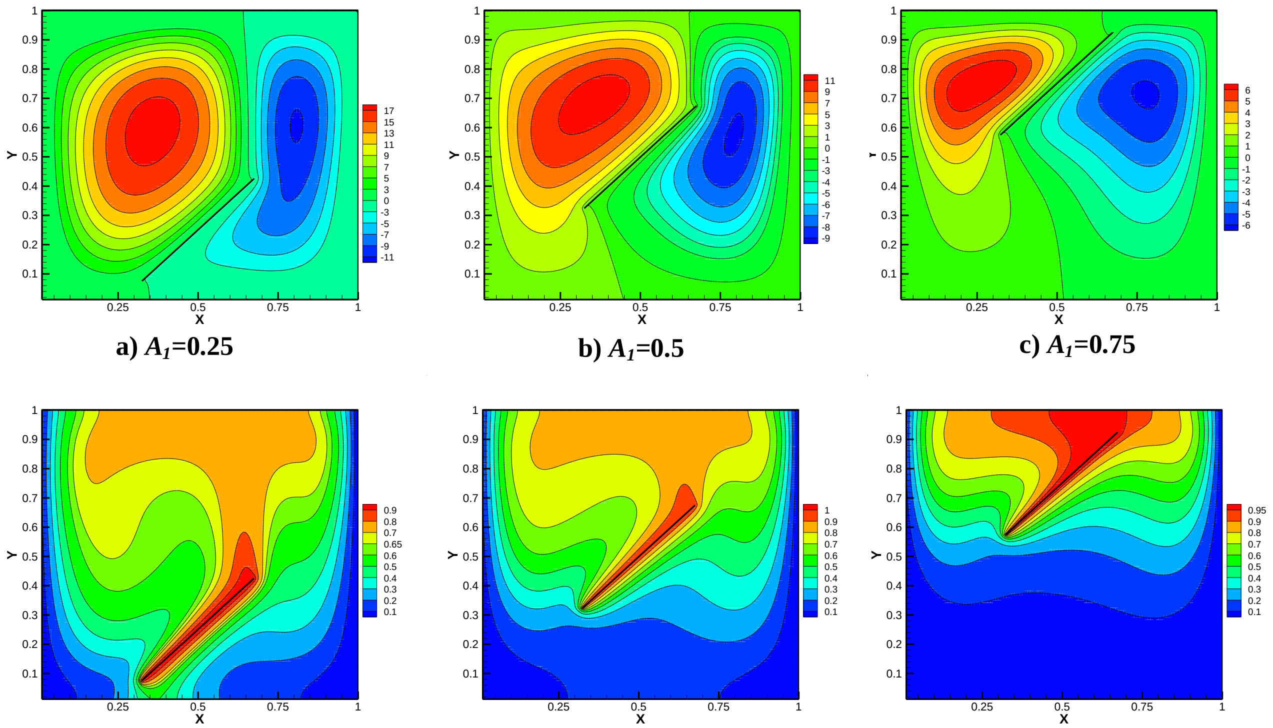


Fig.5 Effect of Rayleigh Number on The Stream Lines And Temp. Distrib. for Inclined Location (a=0.5, a_1=0.5)

Figure (6) represents the effect of aspect ratio of the location of the thin plate on the stream lines and temperature distributions; we note that as the inclined thin plate is close to the lower wall($\boldsymbol{A_1}$=0.25) the value of stream lines will be higher because the space between the heated vertical boundary and the inclined thin plate is large and the value of stream lines will be lower when the inclined thin plate close to the upper wall ($\boldsymbol{A_1}$=0.75). Figure (6.d, e & f) shows temperature distribution; we note that at the lower aspect ratio ($\boldsymbol{A_1}$=0.25), the change in temperature distribution will be higher and that the hot fluid will penetrate more through the enclosure, As the plate is closer to the bottom wall, the fluid under the plate is heated moderately and the temperature gradient is steeper along the vertical side walls. As aspect ratio increases the temperature gradient and the hot fluid will penetrate less through the enclosure and the weak convection will be higher.

4.3. Thin Plate Located Vertically

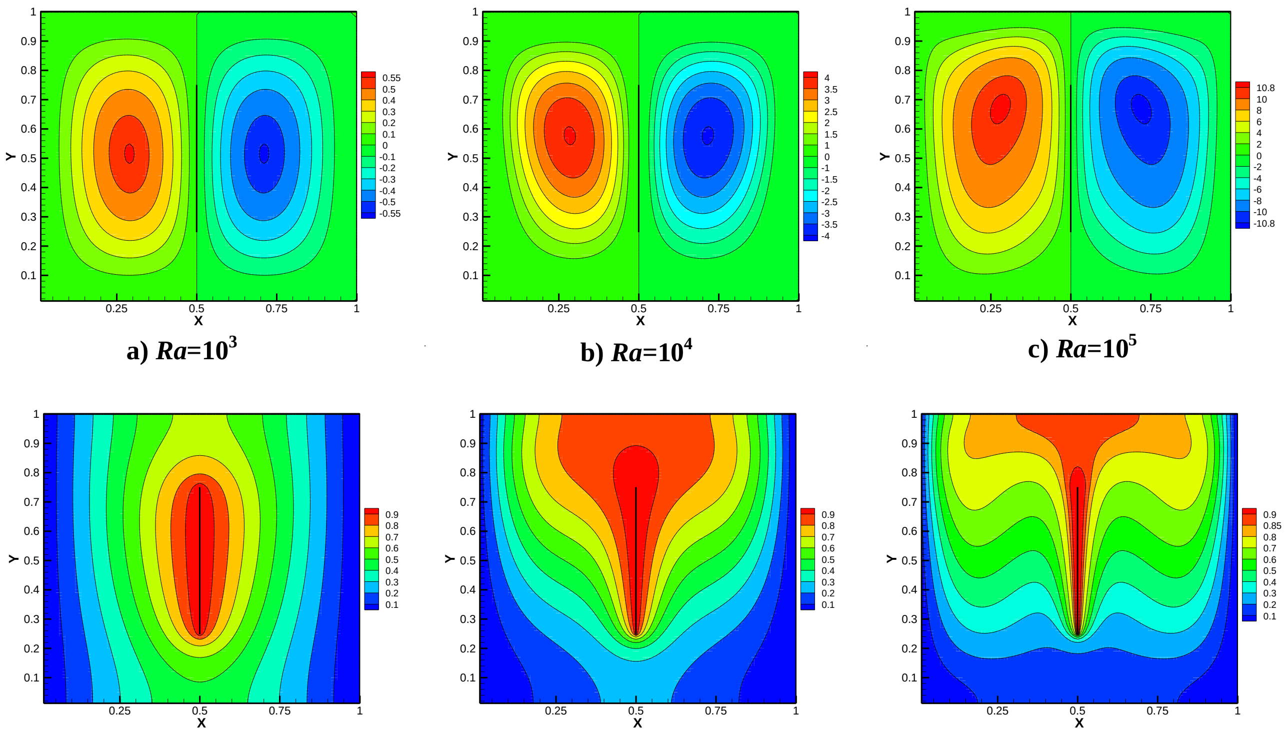
In this case, calculations were performed to see the effect of Rayleigh number in an enclosure with a vertically heated thin plate built in for different of aspect ratio of location of thin plate.



a) A_1=0.25 **b) A_1=0.5** **c) A_1=0.75**

d) A_1=0.25 **e) A_1=0.5** **f) A_1=0.75**

Fig.6. Effect of Aspect Ratio on the Stream Lines and Temp. Distrib. for Inclined Location (Ra=10^5, A=0.5)



a) Ra=10^3 **b) Ra=10^4** **c) Ra=10^5**

d) Ra=10^3 **e) Ra=10^4** **f) Ra=10^5**

Fig.7. Effect of Ra Number on the Stream Lines and Temp. Distrib. for Vertical Location (A=0.5, A_1=0.5)

Fig.(7) shows stream lines and temperature distributions for $\boldsymbol{A}$=0.5, $\boldsymbol{A_1}$=0.5 Ra=10^3-10^5. Symmetrical flow behavior is observed as expected. As Rayleigh number is small conduction mechanism is dominant. As it is seen in figure (7.a-f), stream lines and temperature distributions also show symmetrical behavior and the temperature gradient has steeper gradient along the vertical side walls and the heated thin plate. As Rayleigh numbers are increased, the recirculations intensify around the center of recirculation zone, especially on the upper part of enclosure. In the core region the fluids stay motionless. Looking at the temperature distribution in figure (7.d, e& f), it can be seen that depending on the increase of the Rayleigh number, temperature gradient increases and stratification is formed due to convection mechanism is effective on heat transfer. In figure(8), which stream lines and temperature distributions are given for $\boldsymbol{A}$=0.5, Ra=10^5, $\boldsymbol{A_1}$=0.25,0.5,0.75 it can be seen that when increasing aspect ratio the value of the stream lines decreases because the space between the heated vertical boundary and the plate is small. In addition to that, the temperature distribution in the cold region will increase as shown in figure (8.d). Therefore the maximum value of stream line when the plate is in the vertical location (γ=90°, $\boldsymbol{A}$=0.5, $\boldsymbol{A_1}$=0.25) is equal to 14.589.

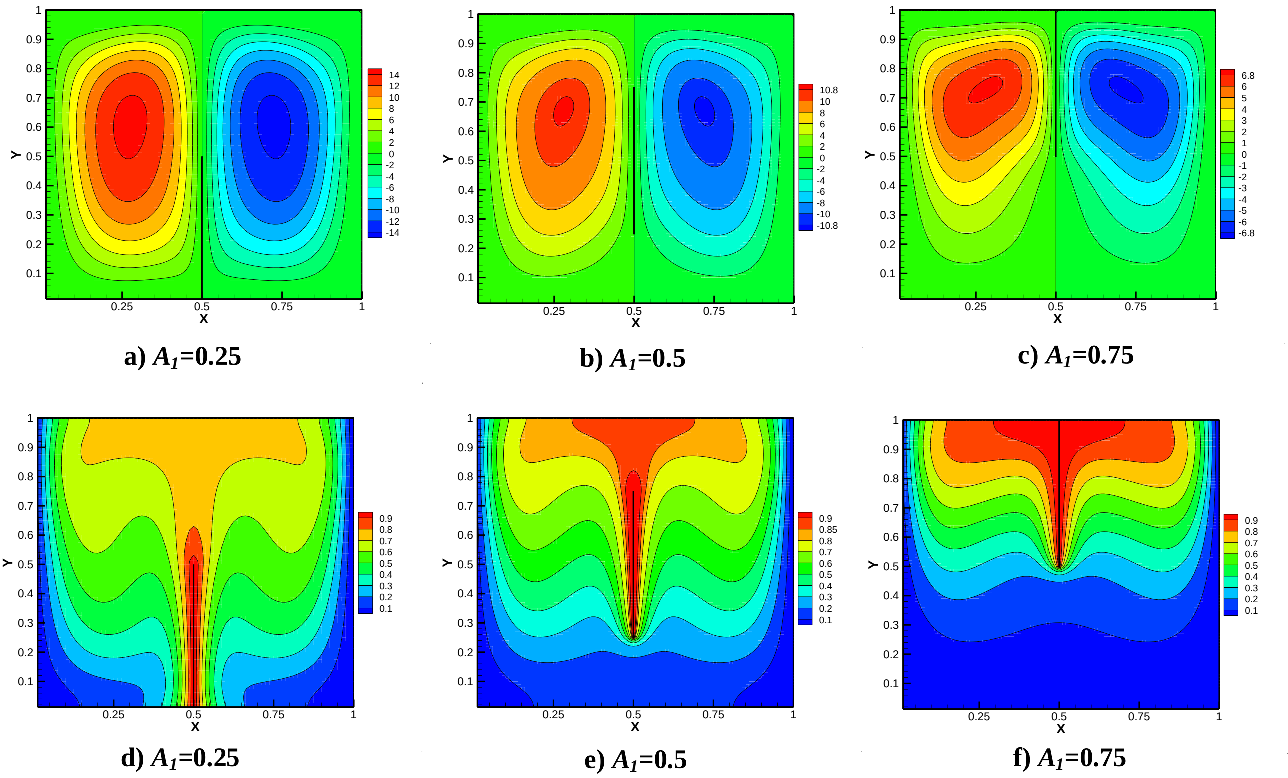
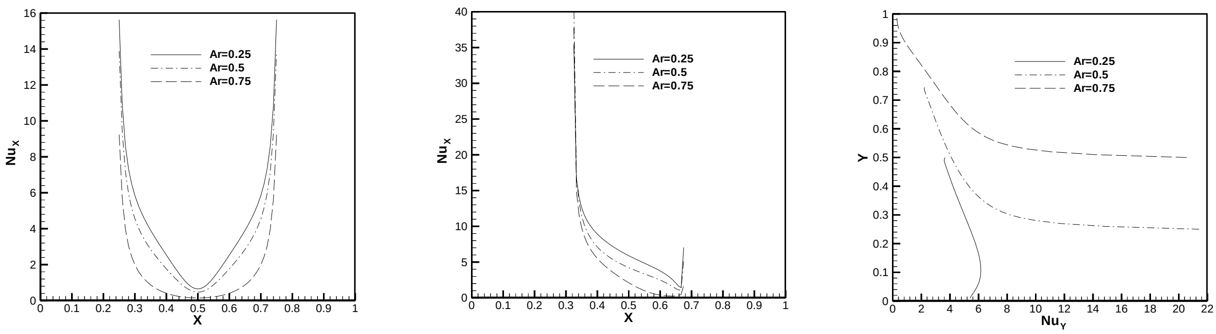


Fig.8. Effect of Aspect Ratio on the Stream Lines and Temp. Distrib. for Vertical Location (*Ra*=10^5, *A*=0.5)

4.4. Local and Overall Heat Transfer

Local heat transfer is shown in terms of local Nusselt numbers for three cases at the heated thin plate as mentioned before. In figure (9), local Nusselt numbers were plotted at different aspect ratio of location of thin plate for three cases. We note in the horizontal situation, as shown in figure (9 a), the local Nusselt is maximum at the edges of the heated thin plate and decreases to the lower value at the center of plate. Figure (9.b) shows the local Nusselt number for the inclined situation, we note the value of Nusselt number is maximum at the left edge because the maximum gradient of temperature is at the left edge, while in the vertical situation, as shown in figure(9.c), we note that the local Nusselt number is maximum at the lower edge(for A1=0.5, 0.75) because the maximum gradient of temperature is at the lower edge. Also we note, for all cases, that with the decrease of the aspect ratio of location of the thin plate, the local of Nusselt number will increase. Generally, we note that the optimum situation for the studied cases is the vertical situation with A1=0.25; also we note that the inclined situation for the thin plate is better than the horizontal situation as shown in table(1). In the table (2) a comparison of average Nusselt number between present work and **Kandaswamy [4]** for the vertical situation and for the right side of the thin heated plate (**A**=0.5, $\mathbf{A_1}$=0.5, $\mathbf{A_2}$=0.5, **Ra**=10^5, **Pr**=0.71), shows an approximation agreement between the two results.



a b c

Fig.9. Effect of Aspect Ratio on the Local Nusselt Number for the Studied Cases (*Ra*=10^5, *A*=0.5) . a) γ =0°, b) γ =45°, c) γ =90°

Table 1
The Values of Nu_{av} for the Studied Cases at *Ra*=10^5

γ	A_1	Nu_{av}
0°	0.25	3.8985
45°	0.25	4.685
90°	0.25	5.1305
0°	0.5	3.8135
45°	0.5	4.4025
90°	0.5	4.816
0°	0.75	2.778
45°	0.75	3.225
90°	0.75	3.597

Table 2
Comparison the Values of Nu_{av} from the Vertical Thin Plate Between Present Work and *Kandaswamy [4]*, (*A*=0.5, A_1=0.5)

	Ra=10^3	*Ra*=10^4	*Ra*=10^5
Kandaswamy[4]	1.52	2.548	5.175
Present work	1.48	2.45	4.816

5. Conclusions

A numerical calculation is performed for laminar free convection heat transfer in square enclosure with an inclined heated thin plate located in three situations horizontal (γ=0°), inclined (γ=45°) and vertical (γ=90°). The plate itself behaves like an obstruction and affects the flow field considerably. With the increase of Rayleigh number local Nusselt number increased in all situations. The value of the average Nusselt number for inclined situation will be grater than the average Nusselt number for horizontal situation and lower than average Nusselt number for the vertical situation. As aspect ratio of location of the thin plate is decreases average Nusselt number, namely, heat transfer increases.

Nomenclature

A	aspect ratio of the heated plate(=***h/L***).
$\boldsymbol{A_i}$	position of heated plates (=$\boldsymbol{h_i / L}$).
g	acceleration due to gravity(m/ s^2).
$\boldsymbol{h_i}$	the distance of the center of plate to the lower or left wall (m).
L	length and height of the enclosure (m).
Nu	local Nusselt number.

p pressure (Pa).

Q total heat source (W).

Q components of heat source (W).

Ra Rayleigh number.

T dimensional temperature (K).

u, v dimensional velocity components (m/s).

U, V dimensionless velocity components.

x, y dimensional coordinates(m).

X, Y dimensionless coordinates.

Greek Symbols

α thermal diffusivity (m^2/s).

β volumetric co-efficient of thermal expansion (1/K).

Ψ dimensionless stream function.

ρ density (kg/m^3).

ρ_o density at T_o (kg/m^3).

υ kinematic viscosity (m^2/s).

ω dimensional vorticity (1/s).

ψ dimensional stream function (m^2/s).

γ angle of location of thin plate.

ζ dimensionless vorticity.

θ dimensionless temperature

Subscripts

c cold

h hot

i = 1, 2

av average

t total

Superscripts

+ upper thin plate

- lower thin plate

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محاكاة انتقال الحرارة بالحمل الحر من صفيحة مسخنة نحيفة مائلة داخل تجويف مربع

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الخلاصة

محاكاة لانتقال الحرارة بالحمل الحر داخل تجويف مربع من صفيحة مسخنة مائلة قد تم دراستها عدديا. كل جدران التجويف محفوظة عند درجات حرارة ثابتة أقل بكثير من درجة حرارة الصفيحة. الجريان فرض على أنه ثنائي البعد. الحل العددي للمعادلات أنجز بواسطة حل معادلات كل من دالة الجريان والدوامية والطاقة بإستخدام الفروق المحددة وتقنية الطريقة الواضحة مع إستخدام طريقة فوق التراخي. الدراسة أنجزت لقيم مختلفة من رقم رايلي (من 10^3 إلى 10^5) ولأوضاع مختلفة لموقع الصفيحة ولثلاث زوايا (γ =0°, 45°, 90°). الهواء قد تم اختياره كمائع عمل في الدراسة. النسبة الباعية لمركز الصفيحة للجدار الموازي الأيسر قد أخذت بقيمة ثابتة ومساوية 0.5. تأثير زاوية الوضع للصفيحة المسخنة على انتقال الحرارة والجريان قد تم دراسته. خلال النتائج تم التوصل إلى أنه الوضع العمودي للصفيحة المسخنة يكون أفضل من الوضع الأفقي والوضع المائل وخاصة عندما تكون النسبة الباعية A_1 مساوية إلى 0.25.