Laminar Natural Convection of Air In a Square Cavity With a Vertically/Horizontally Located Heated Plate Inside It

الحمل الحر الطباقي داخل فجوه مربعه مملؤه بالهواء تحتوي على صفيحه مسخنه مثبته بصوره عموديه/افقيه

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Abstract

A numerical study has been performed for laminar natural convection in a square cavity containing a heated plate located at vertical and horizontal situations by using finite difference method. Top and bottom of the cavity are adiabatic, the two vertical walls of the cavity have constant temperature lower than the plate's temperature. The flow is assumed to be two-dimensional**.**The significant parameters are Grashof number(Gr),aspect ratio(A) and position of the thin plate(A1,A2).Grashof number is varied from10^3,10^5,10^8, while aspect ratio and positions of the heated plate were studied(A=0.25,0.5,0.75),(A1=0.25,0.5,0.75) respectively.Air was chosen as a working fluid(Pr=0.7).In present study, the effect of the position and aspect ratio of heated plate on heat transfer and fluid flow are reported.From the present analysis it is found that with the increase of Gr, the heat transfer rates(Nusselt number) increases in both vertical and horizontal positions of the plate. When aspect ratio of the thin plate decreases, $\overline{Nu}$ also decreases.For vertical situation of the plate, heat transfer is more enhanced than for horizontal. The numerical results of average Nusselt number values have been confirmed by comparing it to similar known previous works using the same boundary conditions. Good agreement is obtaine.

الملخص

تم اجراء دراسه عدديه لانتقال الحراره الطباقي بالحمل الحر داخل فجوه مربعه تحتوي على صفيحه مسخنه موضوعه بصوره عموديه وافقيه بطريقة الفروقات المحدده.الجدران العلويه والسفليه للفجوه تكون معزوله حراريا والجدران العموديه تمتلك درجة حراره ثابته اقل من درجة حرارة الصفيحه.يكون الجريان ثنائي البعد والعوامل التي تؤثر على الجريان هي رقم كرشوف الذي يتغير من10^3 الى 10^8,نسبة الشكل(A=0.25,0.5,0.75)وموقع الصفيحه المسخنه (A1=0.25,0.5,0.75).تم استخدام الهواء(Pr=0.7) كمائع جريان. في البحث الحالي تم دراسة تأثير موقع ونسبة الشكل للصفيحه المسخنه على انتقال الحراره وجريان المائع. من خلال النتائج وجد بان معدل انتقال الحراره(رقم نسلت) يزداد بزيادة رقم كرشوف للموقع العمودي والافقي وعند نقصان نسبة الشكل فان معدل انتقال الحراره يتناقص.كما ان معدل انتقال الحراره يكون اكثر في الموقع العمودي من الافقي. لقد تم مقارنة النتائج العدديه لعدد نسلت المتوسط لهذه الدراسة مع الدراسات السابقة باستعمال نفس الظروف الحدية, وتم الحصول على تطابق جيد للنتائج مع الدراسات السابقهز

Keywords: Natural Convection ,Laminar, Heated plate

Nomenclature

A aspect ratio of the heated plate($\frac{h}{H}$)

A1 position of the horizontal heated plate($\frac{h_1}{H}$)

A2 position of the vertical heated plate($\frac{h_2}{H}$)

g acceleration due to gravity(m/s^2)

h_1 distance of the plate to parallel left wall for vertical situation ,distance of the plate to the parallel upper wall for horizontal situation
h_2 distance of center of plate to the perpendicular upper wall for vertical situation, and to the perpendicular left wall for horizontal situation
t dimensional time(s)
Gr Grashof number($\frac{g\beta(T_h - T_{c)}L^3}{\upsilon^2}$)
$\bar{N}u$ average Nusselt number
T temperature [K]
u, v dimensional velocity components(m/s)
H height of the cavity (m)
U,V dimensionless velocity components
L length of the cavity(m)
x, y dimensional coordinates(m)
X,Y dimensionless coordinates
Pr Prandtl number (υ / α)

Greek Symbols
α thermal diffusion coefficient (m^2/s)
β volumetric thermal expansion coefficient[K^{-1}]
θ dimensionless temperature
υ kinematic viscosity (m^2/s)
Ψ stream function (m^2/s)
ψ dimensionless stream function
τ dimensionless time
ω dimensional vorticity(m^2/s)
ζ dimensionless vorticity

Subscripts
c cold
h hot

Introduction
Natural convection in fluid-filled enclosures and cavities has received considerable attention in recent years since this phenomenon often affects the thermal performance of various systems[1]. Natural convection cooling is desirable because it doesn't require energy source, such as a forced air fan and it is maintenance free and safe[2]. Laminar and turbulent natural convection heat transfer from rectangular enclosures that are heated and cooled from lateral wall has been extensively examined. In the last decade, the attention has shifted to the study of natural convection in partitioned cavities and enclosures with discrete heat sources attached to its adiabatic walls. Dagtekin and Oztop[3] numerically studied 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 partition, attached to the bottom wall, the length and the location were varied while the enclosure was cooled from two walls. Shi and Khodadadi[4] studied the effect of a thin fin on the hot wall in a differentially heated square cavity. The range of the Rayleigh number was $10^4 - 10^7$ and length of fin equals to 20, 35 and 50 percent of the side wall. They reported the two competing mechanisms that were responsible for the flow and thermal modifications were identified. One was due to the blockage effect of the fin and the other one was extra heating of the fluid. The extra heating effect was promoted as the Ra increases. For high Ra, the flow field was enhanced regardless of the fin's length and position. Desai et al. [5] considered rectangular enclosures with multiple heaters

mounted on one side wall, with the top wall being cooled while the other walls were insulated. Their predictions also compared well with previous experimental and numerical work. Kandaswamy et al. [2] studied natural convection heat transfer in a square cavity induced by heated plate numerically. The study was performed for different values of Grashof number ranging from 10^3 to 10^5 for different aspect ratios and position of heated plate. The effect of the position and aspect ratio of heated plate on heat transfer and flow were addressed. With increase of Gr heat transfer rate increased in both vertical and horizontal position of the plate. When aspect ratio of heated thin plate was decreased the heat transfer also decreases.Yucel and Turkoglu[6] also numerically studied 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. Tasnim and Collins[7] determined the effect of a horizontal baffle placed on hot (left) wall of a differentially heated square cavity. It has been found that adding baffle on the hot wall can increase the rate of heat transfer by as much as 31.46 percent compared with a wall without baffle for $Ra = 10^4$. When $Ra = 10^5$ the increase in heat transfer was 15.3 percent for the same baffle length and the increases in heat transfer was 19.73 percent, when the longest baffle was attached at the middle of the cavity. Frederick [8] study numerically natural convection in an air filled, differentially heated, inclined square cavity with a diathermal partition placed at the middle of its cold wall for Rayleigh numbers 10^3 to 10^5. It was observed that due to suppression of convection, heat transfer reductions up to 47 percent in comparison to the cavity without partition was observed by. Altac and Kurtul [9] 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 105 to 107 and from 0 to 90 degrees. The flow pattern and temperature distribution were analyzed, and steady-state plate-surface-averaged Nusselt numbers were correlated. Alami et al [10]study numerically laminar natural convection from a two dimensional horizontal channel with rectangular heated blocks. Oztop and Bilgen [11] have numerically investigated the natural convection in a differentially heated, partitioned, square cavity containing heat generating fluid. Nansteel and Grief [12] conducted an experimental study at higher Rayleigh numbers (10^9- 10^{11}) and an aspect ratio of ½. Water was the working fluid and the horizontal end walls were made of Plexiglass. Bilgen [13] numerically studied 2D square differentially heated cavities, with a thin fin is attached on the active wall. The effects of Rayleigh number (10^4 to 10^9), dimensionless thin fin length (0.10 to 0.90), dimensionless thin fin position (0 to 0.90), dimensionless conductivity ratio (0 to 60) were examined. It is found that Nusselt number increased with Rayleigh number and decreased with fin length and conductivity ratio, and an optimum fin position existed. Barozzi and Corticelli [14] performed 2D numerical simulation of two vertical plates with uniform heat generation and a rectangular heating block with uniform wall temperature, placed at the center of the enclosure. The air-filled cavity was cooled by the vertical walls. Grashoff number was varied from 4×10^4 to 10^8. From the literature review it is clear that the case of a baffle attached to the vertical walls was addressed widely. In this paper, we present a numerical study of natural convection in an air filled square cavity with a heated thin plate located at vertical and horizontal situation for a wide range of Grashof number(10^8). The top and bottom wall of the cavity were adiabatic and the two vertical walls were kept at constant temperature lower than plate ′s temperature. The objective of this paper is to study the effects of the varying Gr, positions and aspect ratio of heated plate on flow and temperature fields and on the heat transfer characteristics of the cavity.

Mathematical Analysis

Natural convection heat transfer from heated plate placed horizontally /vertically inside a square cavity of height H. The assumptions of the present study are the top and bottom walls of the cavity are adiabatic and the two vertical walls have constant temperature T_c lower than plate's temperature which is kept at T_h. The geometry and the coordinate system are illustrated in Fig.(1). The equations governing the laminar two-dimensional incompressible flow of the fluid under Boussinesq approximation are[1,2]:

for the vorticity:

$$\frac{L^2}{\upsilon^2}(\frac{\partial\omega}{\partial t})+u(\frac{L^2}{\upsilon^2})(\frac{\partial\omega}{\partial x})+v(\frac{L^2}{\upsilon^2})(\frac{\partial\omega}{\partial y})=\frac{g\beta L^2}{\upsilon^2}\frac{\partial(T-T_h)}{\partial y}+\frac{L}{\upsilon}\nabla^2\omega \qquad ....(1)$$

For the energy:,

$$\frac{L^2}{\upsilon}\frac{1}{(T_h-T_c)}\frac{\partial(T-T_c)}{\partial t}+u\frac{L^2}{\upsilon}\frac{1}{(T_h-T_c)}\frac{\partial(T-T_c)}{\partial x}+v\frac{L^2}{\upsilon}\frac{1}{(T_h-T_c)}\frac{\partial(T-T_c)}{\partial y}=\frac{\alpha}{\upsilon}\frac{L}{(T_h-T_c)}\nabla^2(T-T_c) ...(2)$$

And for the stream function

$$L\frac{\partial^2\Psi}{\partial x^2}+L\frac{\partial^2\Psi}{\partial y}=-\omega \qquad .....(3)$$

The dimensionless forms of the governing equations will be as follow;

$$\frac{\partial\zeta}{\partial\tau}+U\frac{\partial\zeta}{\partial X}+V\frac{\partial\zeta}{\partial Y}=Gr\frac{\partial\theta}{\partial Y}+\nabla^2\zeta \qquad .....(4)$$

$$\frac{\partial\theta}{\partial\tau}+U\frac{\partial\theta}{\partial X}+V\frac{\partial\theta}{\partial Y}=\frac{1}{\Pr}\nabla^2\theta \qquad .....(5)$$

$$\nabla^2\psi=-\zeta \qquad .....(6)$$

Where the velocity components and the vorticity are defined as :

$$U=-\frac{\partial\psi}{\partial Y} \quad ,V=\frac{\partial\psi}{\partial X},\ \zeta=\frac{\partial U}{\partial Y}-\frac{\partial V}{\partial X} \qquad ....(7)$$

In deriving equations (4-7), the following dimensionless variables where introduced[3]

$$X=\frac{x}{L},\quad Y=\frac{y}{L},$$

$$U=\frac{u}{\upsilon/L},\quad V=\frac{v}{\upsilon/L},$$

$$\tau=\frac{t}{L^2/\upsilon},\quad ,\theta=\frac{T-T_c}{T_h-T_c},$$

$$\psi=\frac{\Psi}{\upsilon},\qquad \zeta=\frac{\omega}{\upsilon}$$

The initial and boundary conditions in the dimensionless form as:

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

$$\tau\succ 0, U=V=0, \frac{\partial\theta}{\partial X}=0 \qquad \textit{on the top and bottom walls}$$

$$U=V=0, \theta=0 \qquad \textit{on the right and left walls}$$

$$U=V=0, \theta=1 \qquad \textit{on the hot plate surface}$$

The average Nusselt number a long the heated plate is defined by[2]:,

$$\overline{N}u=\int_0^1\left.\frac{\partial\theta}{\partial X}\right|_{X=0}dY \qquad ....(8)$$

Numerical Solution

Equations (4-7) are solved numerically using finite difference method with a regular cartesian space grid (Fig.(2.a)). Suppose that all quantities $T_{i,j,n}, \zeta_{i,j,n}, \psi_{i,j,n}$ are known at a time $n\Delta\tau$. An alternating direction implicit (ADI) method is employed to find the temperature and vorticity values at the interior grid points in the next time level $(n+1)\Delta\tau$.For this, forward difference approximation is used for time derivative and central difference approximations are used for all spatial derivatives. The method of successive over relaxation (SOR) is then used in conjunction with the newly computed temperatures $T_{i,j,n+1}$ and vorticities $\zeta_{i,j,n+1}$ to solve the stream function equation for the new improved stream function $\psi_{i,j,n+1}$. After finding the stream function, the values of U and V at the current time level are computed using the central difference approximations to $U = -\dfrac{\partial\psi}{\partial Y}$ and $V = \dfrac{\partial\psi}{\partial X}$. This procedure is repeated at each time step.

$$\frac{\zeta_{i,j,n+1/2} - \zeta_{i,j,n}}{\Delta\tau/2} + U_{i,j,n+1/2}\left(\frac{\zeta_{i+1,j,n+1/2} - \zeta_{i-1,j,n+1/2}}{2\Delta X}\right) + V_{i,j,n}\left(\frac{\zeta_{i,j+1,n} - \zeta_{i,j-1,n}}{2\Delta Y}\right) = Gr\left(\frac{\theta_{i,j+1,n} - \theta_{i,j-1,n}}{2\Delta Y}\right) \quad ...(9)$$
$$+\left(\frac{\zeta_{i-1,j,n+1/2} - 2\zeta_{i,j,n+1/2} + \zeta_{i+1,j,n+1/2}}{\Delta X^2} + \frac{\zeta_{i,j-1,n} - 2\zeta_{i,j,n} + \zeta_{i,j+1,n}}{\Delta Y^2}\right)$$

$$\frac{\zeta_{i,j,n+1} - \zeta_{i,j,n}}{\Delta\tau/2} + U_{i,j,n+1/2}\left(\frac{\zeta_{i+1,j,n+1/2} - \zeta_{i-1,j,n+1/2}}{2\Delta X}\right) + V_{i,j,n+1}\left(\frac{\zeta_{i,j+1,n+1} - \zeta_{i,j-1,n+1}}{2\Delta Y}\right) = Gr\left(\frac{\theta_{i,j+1,n+1} - \theta_{i,j-1,n+1}}{2\Delta Y}\right) \quad ....(10)$$
$$+\left(\frac{\zeta_{i-1,j,n+1/2} - 2\zeta_{i,j,n+1/2} + \zeta_{i+1,j,n+1/2}}{\Delta X^2} + \frac{\zeta_{i,j-1,n+1} - 2\zeta_{i,j,n+1} + \zeta_{i,j+1,n+1}}{\Delta Y^2}\right)$$

For energy equation

$$\frac{\theta_{i,j,n+1/2} - \theta_{i,j,n}}{\Delta\tau/2} + U_{i,j,n+1/2}\left(\frac{\theta_{i+1,j,n+1/2} - \theta_{i-1,j,n+1/2}}{2\Delta X}\right) + V_{i,j,n}\left(\frac{\theta_{i,j+1,n} - \theta_{i,j-1,n}}{2\Delta Y}\right) = \quad ....(11)$$
$$\frac{1}{\Pr}\left(\frac{\theta_{i-1,j,n+1/2} - 2\theta_{i,j,n+1/2} + \theta_{i+1,j,n+1/2}}{\Delta X^2} + \frac{\theta_{i,j-1,n} - 2\theta_{i,j,n} + \theta_{i,j+1,n}}{\Delta Y^2}\right)$$

$$\frac{\theta_{i,j,n+1} - \theta_{i,j,n+1/2}}{\Delta\tau/2} + U_{i,j,n+1/2}\left(\frac{\theta_{i+1,j,n+1/2} - \theta_{i-1,j,n+1/2}}{2\Delta X}\right) + V_{i,j,n+1}\left(\frac{\theta_{i,j+1,n+1} - \theta_{i,j-1,n+1}}{2\Delta Y}\right) = \quad ...(12)$$
$$\frac{1}{\Pr}\left(\frac{\theta_{i-1,j,n+1/2} - 2\theta_{i,j,n+1/2} + \theta_{i+1,j,n+1/2}}{\Delta X^2} + \frac{\theta_{i,j-1,n+1} - 2\theta_{i,j,n+1} + \theta_{i,j+1,n+1}}{\Delta Y^2}\right)$$

Where ϕ is used for(ζ, θ, ψ).The boundary vorticities at the solid walls can be derived considering Taylor's series expansions for stream function in the vicinity of the walls. This computational cycle is repeated till steady state solution is obtained, that is, when the following convergence criteria[2]

$$\left|\varphi_{i,j,n} - \varphi_{i,j,n+1}\right| \prec 10^{-5} \quad ....(14)$$

for temperature, vorticity and stream function is met.Numerical experiments with different grid sizes which correspond to 21 × 21, 41 × 41, 61 × 61, and 81 × 81 are conducted for the Gr (10^4) in a square cavity with heated thin plate located at the middle of the cavity. The length of plate is set to be half of the cavity length. The maximum value of the stream function($\psi_{max.}$) is commonly used as a sensitivity measure of the accuracy of the solution. Fig.(2.b) shows the dependence of quantity ($\psi_{max.}$) on the grid size. Comparison of the predicated ($\psi_{max.}$) values among four different cases suggests that the two grid distributions 61×61, and 81 × 81 gives nearly identical results. Considering both the accuracy and the computational time, the following calculations were all performed with 61×61 grid system.A computational program was written in Fortran-90 language to compute the values of the required variables.

Results and Discussion

In this study, square cavity containing air (Pr=0.7), is investigated numerically in the presence of heated plate located vertically/horizontally situation.The computations are carried out for a wide range of Gr varying from $10^3 to 10^8$,different position and length of the plate are considered. Fig.(3) represents temperature and flow field when the heated thin plate is located horizontally for(A=0.25,0.75) and three plate locations(A1=0.25.0.5,0.75) for different values of Grashof($Gr = 10^3, 10^5, 10^8$).In Figures 3(a),3(b), and 3(c) for(A=0.25,A1=0.25). When Gr=10^3 , the fluid motion is not symmetric in the cavity due to dominance of conduction mechanism to, and there exist two circulating cells of different sizes around the plate. As Gr increases convection becomes stronger because the fluid moves faster and these two cells are symmetric about the hot plate.The circulation over the plate becomes stronger when Gr=10^8 , and the rolls inside the main circulation over the plate develops and gains strength.Temperature and velocity gradients increase around the plate walls and the cold walls with increasing Gr.In Figures3(d),3(e), and 3(f) the isothermal and streamlines for the plate which is located in the middle of the cavity (A1=0.5)is depicted. The flow and temperature distribution in the cavity maintain the symmetric behavior. It can be seen the circulation of fluid downward the plate is prevented because the space between the plate and vertical boundary is small.The fluid flow below the plate produces stagnation point depending on increase of Gr.Dence concentration of isotherms around the plate and cold walls because the temperature difference on vortex areas is small due to fluid circulation.This indicates that the heat transfer rate decreases as the hot plate location is vertically elveted within the cavity. In Figures 3(g),3(h), and 3(i) the plate is located at A1=0.75.At low Gr the circulating cells is unsymmetrical about the heated plate and when Gr increases the flow becomes symmetric. When (Gr=10^8)the fluid moves faster and circulation is very stronger and occurs upwards and downward of the plate. The isotherms become more packed at the upper one third of cold walls and around the plate and almost horizontal about the plate because the natural convection is more vigorous in the top and around the plate due to separation of main circulation. Similar observations are made for cases (A=0.75) and (A1=0.25,0.5,0.75)in Figure3(j)to (s),which show the similarity with the first case. Figs.(4)show the flow and temperature patterns for (A=0.25,0.75),(A1=0.25,0.5,0.75)and different values of Gr, When the hot plate is located vertically.It can be seen for (A1=0.25,0.75)(Figures 4(a),(b),(c),(g),(h),(i)(j),(k),(l),(p),(r),(s)) the circulation is not symmetry about the plate and with increasing Gr,the rotating cell in the left/right of the heated plate slightly gains strength and the streamlines and isotherms become more packed around the vertical plate and next to the cold walls.This suggest that the flow moves faster as the natural convection is intensified. Figures 4(d),(e),(f),(m),(n)(o), for (A1=0.5),the flow and temperature distribution in the cavity becomes symmetrical .The flow exhibits two circulation cells of same size around the plate each covering half of the cavity.As Gr increases the streamlines moves closer towards the vertical plate,producing a strong boundary layer effects on the plate.Furthermor, the isotherms are symmetric and almost horizontal about the heated plate and diminished on the areas bottom the plate because the temperature difference is small in this areas due to fluid circulation.

In Fig.5 the average Nusselt number is plotted for different aspect ratio of heated thin plate for horizontal situation at Gr=10^5. For increase the aspect ratio of plate ,fixing Gr the heat transfer increasres.For a fixed aspect ratio of thin vertical plate, when Grashof number is increased the average Nusselt number also increases as seen in Fig.6. The steady state average Nusselt number is plotted in Figure(7) as a function of Gr,aspect ratio of heated thin plate(A=0.25,0.5,0.75) and different positions of the horizontal thin plate(A1=0.25,0.5,0.75).For A=0.25 case, $\overline{Nu}$ increases with increasing Gr because the velocity field becomes more violent and as a result the boundary layers get thinner which in turn yield higher heat transfer rates as Gr is increasing. $\overline{Nu}$ is maximum for the horizontal hot plate when its located at A1=0.25.As the position of the plate is elveted vertically to A1=0.5 the average Nusselt number decreases slightly and when A1=0.75,the

$\overline{Nu}$ decreases sharply due to increases in boundary layer thickness below the plate.For A=0.5 and A=0.75, the average Nu similar to the case when A=0.25 and is maximum when the plate position is at A1=0.25.In Figure(8) Nusselt number is plotted for different values of Gr and different aspect ratios(A=0.25,0.5,0.75) and different positions(A1=0.25,0.5,0.75) of heated thin plate for vertical situation. The $\overline{Nu}$ for all cases increases with increasing Gr and highest heat transfer is obtained for the plate position at A1=0.5 because the circulations around the plate gets stronger and the removal of the hot fluid from the plate is easy. The minimum value of $\overline{Nu}$ is observed for A1=0.75 for all Gr values.
Figs.(9 α 10) show the heat transfer for different values of Gr and different values of aspect ratio of heated thin plat for horizontal and vertical situations respectively at A1=0.5. It is found that heat transfer rate increases as Gr increase. The heat transfer rate increases when the aspect ratio of the plate increases. Fig.(11) comparing the horizontal and vertical situation with each other for A=0.75 and A1=0.25 it can be seen the heat transfer is very higher for vertical situation than horizontal for high values of Grashof number(Gr>10^5). This means that in vertical situation heat transfer is more enhanced. Fig.(12) compare the mean Nusselt of the present work with results of [2] for vertical and horizontal situation of heated thin plate for (A=0.5,A1=0.5).The comparison shows good agreement

Conclusions

A numerical investigation on natural convection in a square cavity in the presence of heated thin plate placed vertically/horizontally for a wide range of Gr($10^3 - 10^8$) using finite difference method. The main conclusions of the present study are:

1-When the plate located horizontally the flow pattern is symmetric about the plate for higher values of Grashof number(Gr>10^3) and

2-when the plate is located vertically the flow is not symmetric about the plate for all values of Gr except when the plate is located at A1=0.5.

3-For increasing Gr the boundary layers get thinner which in turn yield higher heat transfer rates in both vertical and horizontal situation.

4-As the aspect ratio of heated thin plate is increased the average Nusselt number also increases.

5- For horizontal situation maximum value of $\overline{Nu}$ occurs at A1=0.25 and for vertical case it occurs at A1=0.5.

6-Heat transfers becomes more enhanced in vertical situation than in horizontal situation.

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h_1
h_1

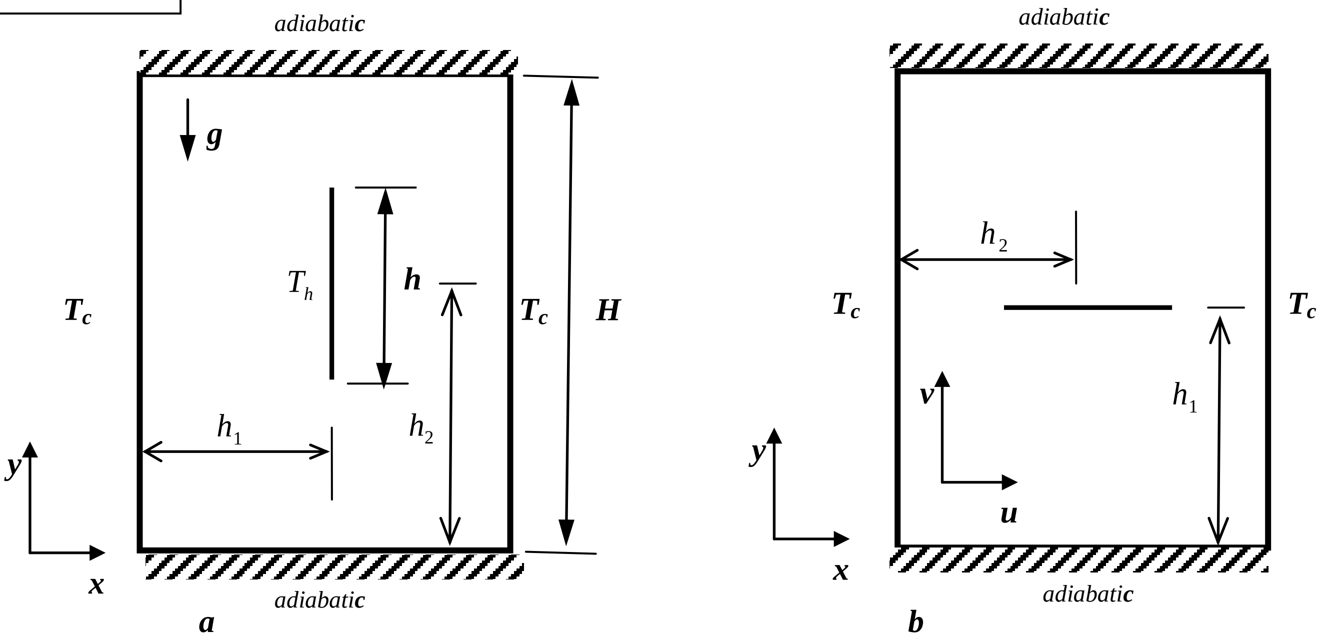
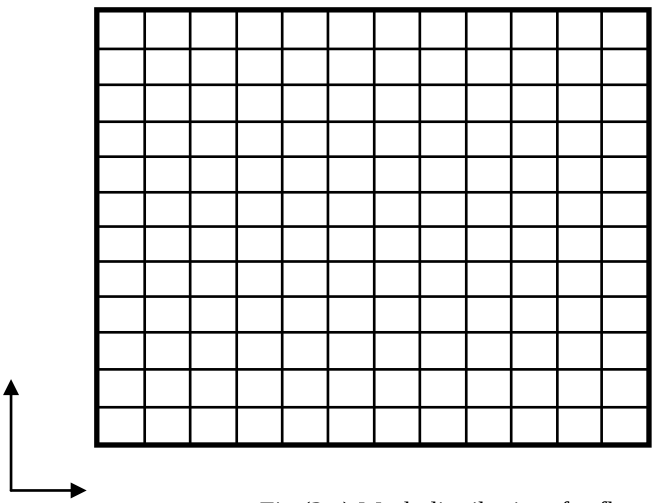


Fig.(1) Physical model and coordinate system:(a)vertical position of heated plate,(b)horizontal position of heated plate



x

Fig.(2.a) Mesh distribution for flow field

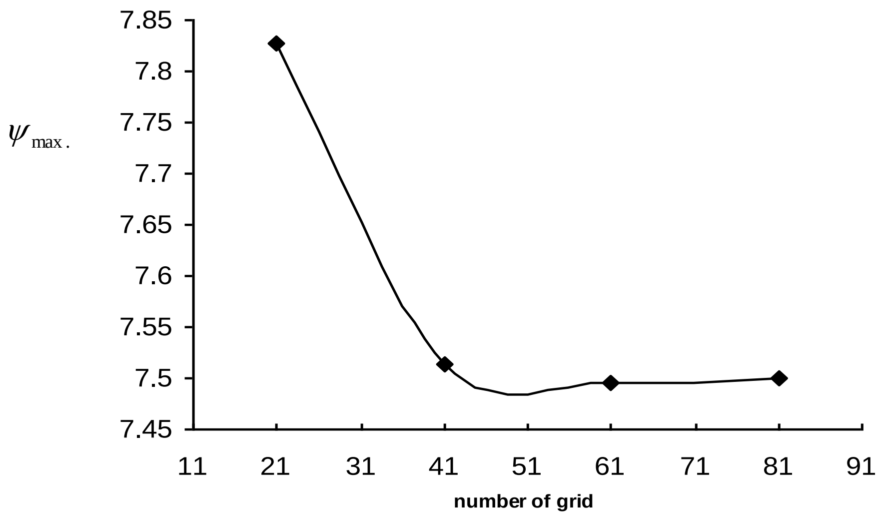


Fig.(2.b) Variation of maximum stream function with the numberof grid points at Gr=10^4