Laminar Natural Convection in nonrectangular Enclosure with and without Fins

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Abstract:

In the present work, steady, laminar natural convection in nonrectangular enclosures is analyzed numerically with and without fin. Vertical walls insulated while horizontal walls maintained isothermal at different temperature and the fin was placed on horizontal surface. The length of fin was equal (B/L=0.22, 0.44 and 0.66) and thickness of fin was constant. Various parameters are studied: Rayleigh number (from 10^4 to 10^7), Prandtl number (0.7), number of fin change from (1-3) and aspect ratio (H/L= 0.15 to 0.5). The problem is formulated in terms of the vorticity-stream function procedure. A numerical solution based on program in Fortran 90 with Tec plot program. The finite difference method is used. Streamlines and isotherms are presented for different values of parameters studied. A Nusselt number correlation is derived by using program (DGA v1.00) and mean Nusselt numbers on hot walls are also calculated at different cases. The results show the mean Nusselt numbers decreases with increasing aspect ratio (H/L).Also, predictions reveal a decrease in heat transfer in the presence of fins. The results of the calculations are compared with the previous works and it showed a good agreement.

Key words: Natural convection, fin, square enclosure, partitioned enclosure.

الخلاصة:

يتضمن البحث الحالي دراسة نظرية عددية لانتقال الحرارة بالحمل الحرللجريان المستقر داخل حيز مربع بوجود او عدم وجود الزعانف. الجدران العمودية للحيز معزولة بينما الجدران الافقية ذات درجات حرارة ثابتة لكن مختلفة.ارتفاع الزعانف الموضوعة داخل الحيز (B/L= 0.22,0.44, 0.66) وموضوعة على الجدران الافقية وذات سمك ثابت . الدراسة تغطي المدى لرقم رايلي من10^4 الى 10^7 ولعدد برانتل (Pr=0.7) ونسبة(H/L=0.15-0.5) واعداد مختلفة للزعانف تتراوح من(1- 3). الحل العددي تم باستخدام طريقة دالة الانسياب- الدوامية بأستخدام طريقة الفروق المحددة مع بناء برنامج حاسوبي بلغة(فورتران 90) مع برنامج Tec plot program.تم دراسة تاثير كل من رقم رايلي ، عدد الزعانف ، ونسبة (H/L) على شدة الدوامات الدوارة و انتقال الحرارة داخل الحيزبوجود او عدم وجود الزعانف . بالاضافة الى ذلك تم اشتقاق معادلة التصحيح لرقم نسلت باستخدام برنامج (DGA v1.00) و ايجاد معدل قيم رقم نسلت على طول الجدار المسخن. بينت النتائج ان متوسط نسلت يقل مع زيادة نسبة (H/L) وزيادة عدد الزعانف. وتم مقارنة النتائج العددية مع النتائج العددية المتوفرة وكانت نتائج المقارنة جيدة.

1- Introduction

Natural convection heat transfer inside irregular and complex shaped enclosures has a wide variety of technological applications involving double-wall thermal insulation, underground cable systems, solar-collectors, electric machinery, cooling system of micro-electronic devices, natural circulation in the atmosphere, the molten core of the Earth, etc.

Miomir Raos [2002], investigated numerically using FVM the laminar natural convection phenomena in two-dimensional rectangular enclosure with differentially heated sides and adiabatic horizontal walls, and the effect of rotation of the enclosure is presented in this study too. The study assumed Pr=0.7 and Rayleigh number from 10^3 to 10^6. The results showed complex flow patterns heat transfer rates, with different orientation of the enclosure. Angle of rotation about 65-75 maximize Nu number value for named conditions.

Dias and Milanez [2004] studied numerically a 3-D laminar natural convection in air filled enclosure using finite volume technique .The results were obtained for Ra no. ranges from (10^3 to 10^6) and aspect ratios ranging from (1 to 20).They found that the two dimensional approximation, frequently compared to experimental results, deviates from the three dimensional results as the Rayleigh number increases.

Ben Yedder and Bilgen [1997], studied numerically the effect of laminar natural convection in inclined enclosures bounded by a solid wall for the range $10^3<Ra<10^6$ and the inclination angle (from 30^o to 180^o) with Ar=1 and thermal conductivity ratio K_r was varied (from 1 to 10). Flow, temperature fields and heat transfer rates are examined for these ranges of the Rayleigh number and geometrical parameters of the problem. They found for K_r=10, which correspond to the case with employing a polynomial-base differential quadrature method. The results show that the presence of a vertical partition has a considerable effect on the circulation intensity, and therefore, the heat transfer high wall conductivity, the temperature gradient within the solid wall is very small and the temperature at the internal surface is almost the same as the imposed uniform temperature at its outer boundary. For increasing Ra number, the isotherms show a stratified flow within the enclosure with steep gradients near the vertical boundaries.

Syeda and Shohel [2006] studied numerically the effect of the buoyancy induced flow and heat transfer characteristics inside an inclined L-shaped enclosure. A control volume based Finite-Volume method is applied to discretize the governing equations with collocated variable arrangement. SIMPLE algorithm is used and the system of equations is solved by Stone's SIP solver with full multigrid acceleration. Results are presented in the form of the average Nusselt number for a range of inclination angle =0°–360°; Rayleigh number, Ra=1–10^5; and aspect ratio, A=0.1–0.5.

Shaw et al. [1987], studied natural convection in an enclosure fitted with a partial vertical adiabatic partition. Horizontal walls were adiabatic and vertical walls were maintained at different temperatures. The governing equations were solved with the aid of a cubic spline collocation method. Numerical simulations showed that the height and location of the partition have significant effects on the flow and heat transfer characteristic of the enclosure.

Kamil Kahveci [2007] presented numerical study of laminar natural convection in an enclosure divided by a partition with a finite thickness and conductivity. The enclosure is assumed to be heated using a uniform heat flux on a vertical wall, and cooled to a constant temperature on the opposite wall. The governing equations in the vorticity-stream function formulation are solved by characteristics across the enclosure. The average Nusselt number decreases with an increase of the distance between the hot wall and the partition. With a decrease in the thermal resistance of the partition, the average Nusselt number shows an increasing trend and a peak point is detected. If the thermal resistance of the partition further declines, the average Nusselt number begins to decrease asymptotically to a constant value. The partition thickness has little effect on the average Nusselt number.

Nuri and Hakan [2003], studied natural convection heat transfer in partially divided square enclosures. A finite difference computer program based on control volume approach is developed for the solution. The study assumed Pr=0.7 and Rayleigh number from 10^3 to 10^6. Vertical side walls are kept at different constant temperature, while the horizontal walls are insulated. The effects of Ra number and number of partitions on heat transfer are investigated. They found that , the heat transfer rate increases with increasing Ra number, and at low Ra numbers the conduction is the dominant heat transfer mode, and at Ra=10^3 , the mean Nusselt number remains constant around unity for all numbers of partitions.

Nienchuan and Bejan [1983] experimentally and analytically studied the phenomenon of heat transfer by natural convection in a partially divided enclosure. The nonconducting partition was fitted to the floor of cavity. Heat transfer measurements and flow visualization studies were conducted in Rayleigh number range 10^9–10^{10} for aperture ratios (height of the internal opening above the partition: height of the enclosure) of 1, 1/4, 1/8, 1/16 and 0. It was found that, as the aperture ratio decreases from 1 to 0, the Nusselt number decreases by a factor of 15.

Moukalled and Darwish [2003] Numerical results are reported for natural-convection heat transfer in partially divided trapezoidal cavities representing industrial buildings. Two thermal boundary conditions are considered. In the first, the left short vertical wall is heated while the right long vertical wall is cooled (buoyancy-assisting mode along the upper inclined surface of the cavity). In the second, the right long vertical wall is heated while the left short vertical wall is cooled (buoyancy-opposing mode along the upper inclined surface of the cavity). The effects of Rayleigh number, Prandtl number, baffle height, and baffle location on the heat transfer are investigated. Results are displayed in terms of streamlines, isotherms, and local and average Nusselt number values. For both boundary conditions, predictions reveal a decrease in heat transfer in the presence of baffles, with its rate generally increasing with increasing baffle height and Pr. For a given baffle height, greater decrease in heat transfer is generally obtained with baffles located close to the short vertical wall.

Eric and Mohamed [2005] investigated natural convection heat transfer and fluid flow characteristics from a horizontal fluid layer with finned bottom surface numerically and experimentally. The effects of fin height and fin spacing have been investigated for a sufficiently wide range of Rayleigh number. Quantitative comparisons of heat transfer rates and finned surface effectiveness have been reported. The insertion of heat conducting fins has been found to induce an upward fluid motion along the fin walls. For a given value of fin spacing, the number of convection cells between two adjacent fins is function of the values of fin height and Rayleigh number. In comparison with a bare plate, the heat transfer rates for low values of fin height may be decreased by the insertion of fins. For high values of fin height, the finned surface effectiveness is greater than one for a wide range of fin spacing. For low values of Rayleigh number and high values of fin height, the finned surface effectiveness increases linearly with the decrease of fin spacing. Useful guidelines have been suggested to enhance the heat transfer rates from the finned surface.

There are several experimental and numerical studies on natural convection heat transfer in enclosures; however, studies about the nonrectangular enclosure partially divided are rare .In this study, natural convection of air with the Prandtl

number of (0.7) has been analyzed for different cases considered given below:
Case-1- without fin and change value of aspect ratio (H/L).
Case-2- with different numbers of fin.
Case-3- with different height of fin at lower wall.

2-Physical model

The problem studied in this paper is a two dimensional laminar flow and heat transfer in a nonrectangular enclosure with and without fins. The considered Rayleigh numbers was ranged from (10^4 to 10^7), The working fluid was air with (Pr=0.70) .Vertical walls insulated while horizontal walls and the fins were placed on horizontal surface are maintained isothermal at different temperature as shown in Fig.(1). The length of fins was taken equal (B/L=0.22, 0.44, and 0.66) and number of fin change from (1 - 3).

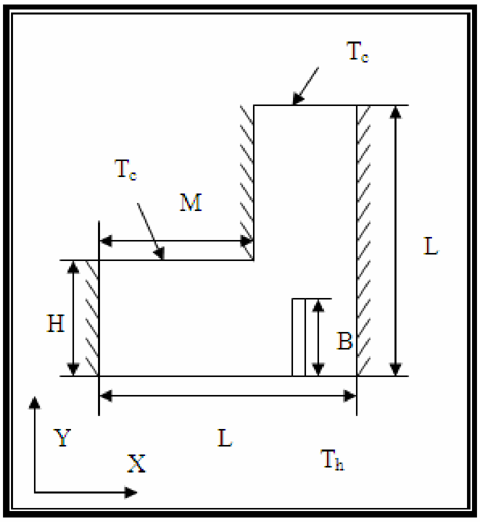

Fig. (1) Shematic of problem and the coordinate system

3- Mathematical model

The fluid is Newtonian and the flow is steady, laminar, two dimensions and incompressible. The non-dimensional governing equations for the conservation of mass, momentum and energy are [Eric and Mohamed 2005]:

Continuity:

$$\frac{\partial U}{\partial X}+\frac{\partial V}{\partial Y}=0 \qquad (1)$$

X- momentum:

$$\frac{\partial}{\partial X}(UU)+\frac{\partial}{\partial Y}(VU)=-\frac{\partial P}{\partial X}+\Pr\left[\frac{\partial^2 U}{\partial X^2}+\frac{\partial^2 U}{\partial Y^2}\right] \qquad (2)$$

Y- momentum:

$$\frac{\partial}{\partial X}(UV)+\frac{\partial}{\partial Y}(VV)=-\frac{\partial P}{\partial Y}+\Pr\left[\frac{\partial^2 V}{\partial X^2}+\frac{\partial^2 V}{\partial Y^2}\right]+Ra\Pr\theta \qquad (3)$$

Energy:

$$U\frac{\partial\theta}{\partial X}+V\frac{\partial\theta}{\partial Y}=\left[\frac{\partial^2\theta}{\partial X^2}+\frac{\partial^2\theta}{\partial Y^2}\right] \qquad (4)$$

With: $U=\frac{uL}{\alpha}$, $V=\frac{vL}{\alpha}$, $X=\frac{x}{L}$, $Y=\frac{y}{L}$,

$P=\frac{pL^2}{\rho\alpha^2}$, $\theta=\frac{T-T_c}{T_h-T_c}$,

$\Pr=\frac{\nu}{\alpha}$, $Ra=\frac{g\beta(T_h-T_c)L^3}{\nu^2}$, Ra=Gr*Pr

The governing equations can be written in dimensionless stream function–vorticity form as:

$$\frac{\partial}{\partial X}(U\omega)+\frac{\partial}{\partial Y}(V\omega)=\Pr\left[\frac{\partial^2\omega}{\partial X^2}+\frac{\partial^2\omega}{\partial Y^2}\right]+(Ra\Pr)\frac{\partial\theta}{\partial X} \qquad (5)$$

$$U\frac{\partial\theta}{\partial X}+V\frac{\partial\theta}{\partial Y}=\frac{\partial^2\theta}{\partial X^2}+\frac{\partial^2\theta}{\partial Y^2} \qquad (6)$$

The only non- zero component of the vorticity is:

$$\omega=\frac{\partial V}{\partial X}-\frac{\partial U}{\partial Y} \qquad (7)$$

From the definition of stream function which verify the continuity equation, vertical and horizontal components can be written as:

$$V=-\frac{\partial\psi}{\partial X} \qquad (8)$$

$$U=\frac{\partial\psi}{\partial Y} \qquad (9)$$

By substituting equation (8) and (9) into equation (7) to obtain the following stream equation:

$$-\omega = \frac{\partial^2 \psi}{\partial X^2} + \frac{\partial^2 \psi}{\partial Y^2} = \Delta^2 \psi \qquad (10)$$

For the solid region (in the fins), the energy equation for the heat transfer by conduction becomes [Kamil Kahveci 2007]:

$$\frac{\partial^2 \theta}{\partial X^2} + \frac{\partial^2 \theta}{\partial Y^2} = 0 \qquad (11)$$

The following equation was defined for the interface between the fin and fluid [Nogotov 1978]:

$$K_f \frac{\partial \theta}{\partial n} = K_{fin} \frac{\partial \theta}{\partial n} \qquad (12)$$

Where:
(K_f) and (K_{fin}) is the thermal conductivity of fluid and fin respectively, (n) represent the normal distance.

4- Boundary conditions

The boundary conditions for the problem can be written in dimensionless form as [Nuri and Ahmed 2003] :

1. The left wall (Insulated wall) (X=0):

$$U = 0, V = 0, \psi = 0, \frac{\partial \theta}{\partial n} = 0 \qquad (13)$$

2.The right wall (Insulated wall) (X=L):

$$U = 0, V = 0, \psi = 0, \frac{\partial \theta}{\partial n} = 0 \qquad (14)$$

3.The lower wall (hot wall) (Y=0) :

$$U = 0, V = 0, \psi = 0, \theta = 1 \qquad (15)$$

4.The upperr wall (cold wall) (Y=L):

$$U = 0, V = 0, \psi = 0, \theta = 0 \qquad (16)$$

5- For the all surface walls the vorticity on the rigid walls is calculated by using the Woods formulae (Woods 1954):

$$\omega_o = \frac{3}{\Delta n^2}(\psi_o - \psi_1) + \frac{1}{2}\omega_1 \qquad (17)$$

5- Numerical Solution

The numerical solution of this system may be obtained by solving its difference system with some available iteration procedure. Consider the second order conservative, monotonic finite difference scheme approximating system of equations (5), (6) (10) and (11). The scheme has been used for steady convective problems involving wide ranges of process parameters and has given good results. It employes the integrointerpolation method. A system of difference equations is obtained by integrating the original system (5), (6),(10)and (11) [Nogotov 1978].

Following is the procedure in [Nogotov 1978], the governing finite difference equations for (ω, θ, and ψ) can be written in the standard five point formula form. These finite difference equations which subject to appropriate boundary conditions are solved by an iterative method known as successive substitution. If (ω^s, θ^s, and ψ^s) denote functional values at the end of sth iteration, the value of (ω,θ, and ψ) at (s+1)th iteration level are calculated from the following expressions:

$$\theta_{i,j}^{s+1} = (1-F_\theta)\theta_{i,j}^{s} + \frac{F_\theta}{A_\theta}(a_\theta \theta_{i+1,j}^{s} + b_\theta \theta_{i-1,j}^{s+1} + c_\theta \theta_{i,j+1}^{s} + d_\theta \theta_{i,j-1}^{s+1}) \qquad (18)$$

$$\omega_{i,j}^{s+1} = (1-F_\omega)\omega_{i,j}^{s+1} + \frac{F_\omega}{A_\omega}\left[\begin{array}{l}(a_\omega \omega_{i+1,j}^{s} + b_\omega \omega_{i-1,j}^{s+1} + c_\omega \omega_{i,j+1}^{s} + d_\omega \theta_{i,j-1}^{s+1}) \\ + 0.5(Ra/\Pr)h\left(\theta_{i+1,j}^{s+1} - \theta_{i-1,j}^{s+1}\right)\end{array}\right] \qquad (19)$$

$$\psi_{i,j}^{s+1} = (1-F_\psi)\psi_{i,j}^{s+1} + \frac{F_\psi}{4}(\psi_{i+1,j}^{s} + \psi_{i-1,j}^{s+1} + \psi_{i,j+1}^{s} + \psi_{i,j-1}^{s+1} + h^2 \omega_{i,j-1}^{s+1}) \qquad (20)$$

$$\theta_{i,j}^{s+1} = (1-F_{\theta\theta})\theta_{i,j}^{s+1} + \frac{F_{\theta\theta}}{4}(\theta_{i+1,j}^{s} + \theta_{i-1,j}^{s+1} + \theta_{i,j+1}^{s} + \theta_{i,j-1}^{s+1}) \qquad (21)$$

Where (s) is the iteration number, and $F_\theta, F_\omega, F_\psi$ and $F_{\theta\theta}$ are the relaxation parameters which depend on the mesh size and fluid mechanical parameters.

For the interface region, the equation (12) is solved forward or backward according to the location of the partition. The final form of equation (12) is:

$$\theta_{int} = (K_r\,\theta_{fin} + \theta_f)/(1+K_r) \qquad (22)$$

Where:

$$K_r = \frac{K_{fin}}{K_f} \qquad (23)$$

A converged solution was defined as one that meet the following criterion for all dependent variables.

$$\max\left|\frac{\phi^{n+1}-\phi^{n}}{\phi^{n+1}}\right| \le 10^{-6} \qquad (24)$$

6- Calculation of Mean Nusselt Number

The mean Nusselt number can be calculated from equation below:

$$Nu = \frac{hL}{k} \qquad (25)$$

The above equation can be written in dimensionless form as follows:

$$Nu = \int_0^1 \left.\frac{\partial \theta}{\partial Y}\right|_{Y=0} dX \qquad (26)$$

The integration can be evaluated by using numerical integration (Simpson's rule) to obtain an overall Nusselt number as below [Nogotov 1978]:

$$Nu = \frac{h}{3}\left[Nu_{L(1)} + 4\sum_{i=2}^{m-2} Nu_{L(i)} + 2\sum_{i=3}^{m-1} Nu_{L(i)} + Nu_{L(m)} \right] \qquad (27)$$

7- Correlations

A correlation equations had been written to show the aspect ratio, number of fins, and height of fin effects on the rate of heat transfer.

Several analyses where performed, by using program (DGA v1.00 and Excel) to find correlation of Nusselt number for different cases considered given below:

1- Effect of aspect ratio (H/L)

Nusselt number for any aspect ratio (H/L) can be written in the following form:-

$$Nu = cRa^{n1}\left(\frac{H}{L}\right)^{n2} \qquad (28)$$

The above equation is valid for:

$10^4 \le Ra \le 10^7$, $0.15 \le (H/L) \le 0.5$, with out fins.

The final form :

$$Nu = 0.231Ra^{0.273}\left(\frac{H}{L}\right)^{-0.267} \qquad (29)$$

2- Effect number of fins

Nusselt number correlasion can be written in the following form:-

$$Nu = cRa^{n1}D^{n2} \qquad (30)$$

D= number of fin from (1-3)

The above equation is valid for:

$10^4 \le Ra \le 10^7$

The final form :

$$Nu = 0.111Ra^{0.313}D^{-0.137} \qquad (31)$$

3-Effect of height of fin (B/L)

Nusselt number for any value of height of fin (B/L) can be written in the following form:-

$$Nu = cRa^{n1}\left(\frac{B}{L}\right)^{n2} \qquad (32)$$

The above equation is valid for :

$10^4 \le Ra \le 10^7$, $0.22 \le (B/L) \le 0.66$, number of fin=1.

The final form :

$$Nu = 0.115Ra^{0.325}\left(\frac{B}{L}\right)^{0.010} \qquad (33)$$

8- Results and Discussions

Numerical results for the laminar and natural convection in nonrectangular enclosures with and without fin is considered for various values of the Rayleigh number.

*** Effect of aspect ratio (H/L)**

Figs.(2),(3),(4) and (5) respectively, show effects of Ra number on the stream function and temperature contours for number of fins (0) ande for aspect ratio change from(H/L=0.15 to 0.5). For small Rayleigh numbers ($Ra=10^4$), the free convection currents are small and the values of stream function in enclosure increases with the increasing Rayleigh number. When the Rayleigh number increases, the temperature gradient next to hot and cold wall increases. Also development of the plume effect can be seen as the Rayleigh number increases.

*** Effect number of fins**

In Fig.(6) the streamline and isotherm contours for different Rayleigh numbers are plotted for number of fin (1) and fin height (B/L= 0.22).The value of stream function (the strength of flow) in enclosure increases with the increasing Rayleigh number. At low Rayleigh numbers there is two circulating cells forms inside the enclosure; with further increase in Rayleigh numbers, the number of circulating cells formed in the enclosure becomes three or more. To analyze the effects of the fin on the temperature contours, It can be observed that the convection cells move outward away from the fin surface with the increase of Rayleigh numbers. Also, the temperature gradient at the cold wall above the fin tip increases with the increase of Rayleigh numbers.As the number of fins increases, the strength of flow decreases and the secondary cells rotating opposite to the direction of main cell can be seen Figs. (7) and (8).

***Effect of height of fin (B/L)**

To analyze the effects of the fin height on the flow and heat transfer characteristics, the computations are performed for different fin heights at different Rayleigh numbers. Figs.(6),(9) and (10) , it is seen that, the increase of the height of fin lowers the fluid flow in the enclosure and retards the heat transfer from the warm wall to the cold wall efficiently. As a result, the stream function values and the mean Nusselt number decrease with increasing fin heights. Temperature gradient at the fin tip increases with the increase of fin height, no significant differences can be seen on temperature contours for different fin heights.

***Nusselt numbers.**

Fig. (11) Shows the relation between the Nu numbers with Ra number for various aspect ratios (H/L). Results showed that the Nu number increases as aspect ratio decreases.

The variation of mean Nu number with Ra number at different number of fin is shown in Fig.(12). As seen the mean Nusselt number increases with increasing Rayleigh number. The decrease in heat transfer in the presence of fin can easily be depicted from the profiles presented in Figure. As the fluid moves up along the hot wall, its temperature increases and the temperature difference between the fluid and the hot wall decreases.This results in a decrease in Nu values as depicted. It is also seen that variation of mean Nu number decreases with increasing number of fins. As the number of fins increases, fin partially block the motion of fluid flow and as a result, the effect of convection decreases.

In Fig. (13), the variation of the mean Nu number with Ra number is shown for different height of fin. It is seen in this figure, the mean Nusselt number decreases with increasing heights of fin.

***Comparison of present work**

The numerical result of the present work is compared with available numerical result of [Syeda and Shohel 2006]. As displayed in Figs. (14) and (15), the present simulation shows good agreement with the numerical data.

The values of numerically computed Nu are compared with that of Nu calculated from correlation equation (31) for case study effect number of fins. Excellent agreement is observed as is evident from Fig. (16).

9- Conclusion

1- The flow structure is strongly affected by increasing the Rayleigh number.

2- The values of stream function (the strength of flow) in enclosure increases with the increasing Rayleigh number and decrease with increase aspect ratio (H/L).

3- Mean Nusselt numbers decreases with:

- Increasing aspect ratio (H/L)
- In the presence of fins
- For a given fin height, greater value.

10-Nomenclature

Symbol	Description	Unit
B	Height of the fin	m
H	Height of wall	m
g	Gravitational acceleration	m/sec^2
K_f	Thermal conductivity of fluid (air) $K_f = 0.024W/m.C)$	W/m.K
$K_{f\,in}$	Thermal conductivity of fin(copper) ($K_{fin} = 202$W/m/C)	W/m.K
Kr	Thermal conductivity ratio	
L	Enclosure total length	
Nu	Mean Nusselt number	
Pr	Prandtl number	
Ra	Rayleigh number	
T	Temperature	K
T_c	Cold wall temperature	K
T_h	Hot wall temperature	K
U	Dimensionless velocity component in x-direction	
V	Dimensionless velocity component in y-direction	
x	Horizontal axis	m
X	Dimensionless horizontal axis	
y	Vertical axis	m
Y	Dimensionless vertical axis	
υ	Kinematic viscosity	m^2/sec
θ	Dimensionless temperature	
ψ	Stream function	
ω	Vorticity	

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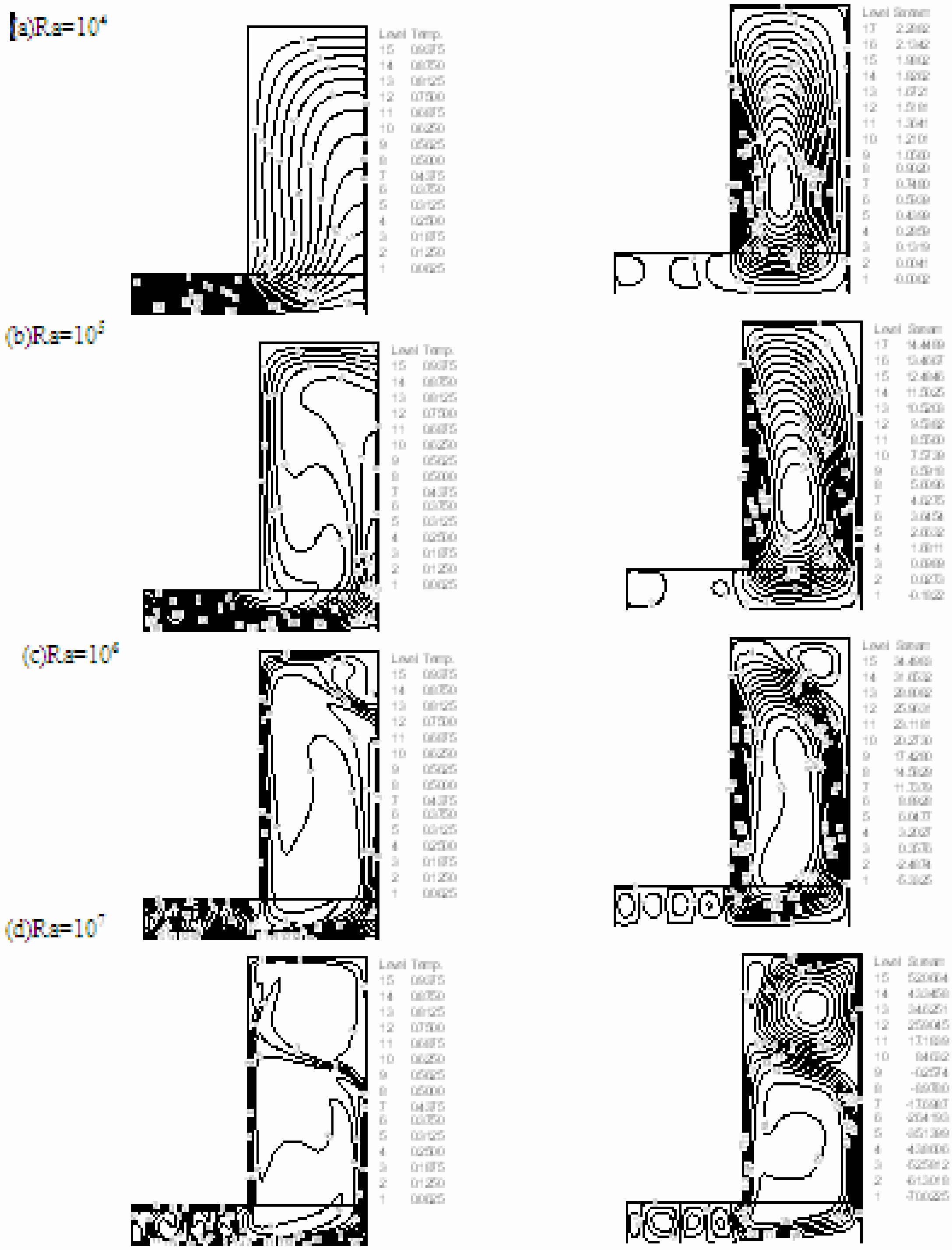

Fig. (2) Effects of Ra number on the temperature contours and stream function without fins and aspect ratio (H/L= 0.15).

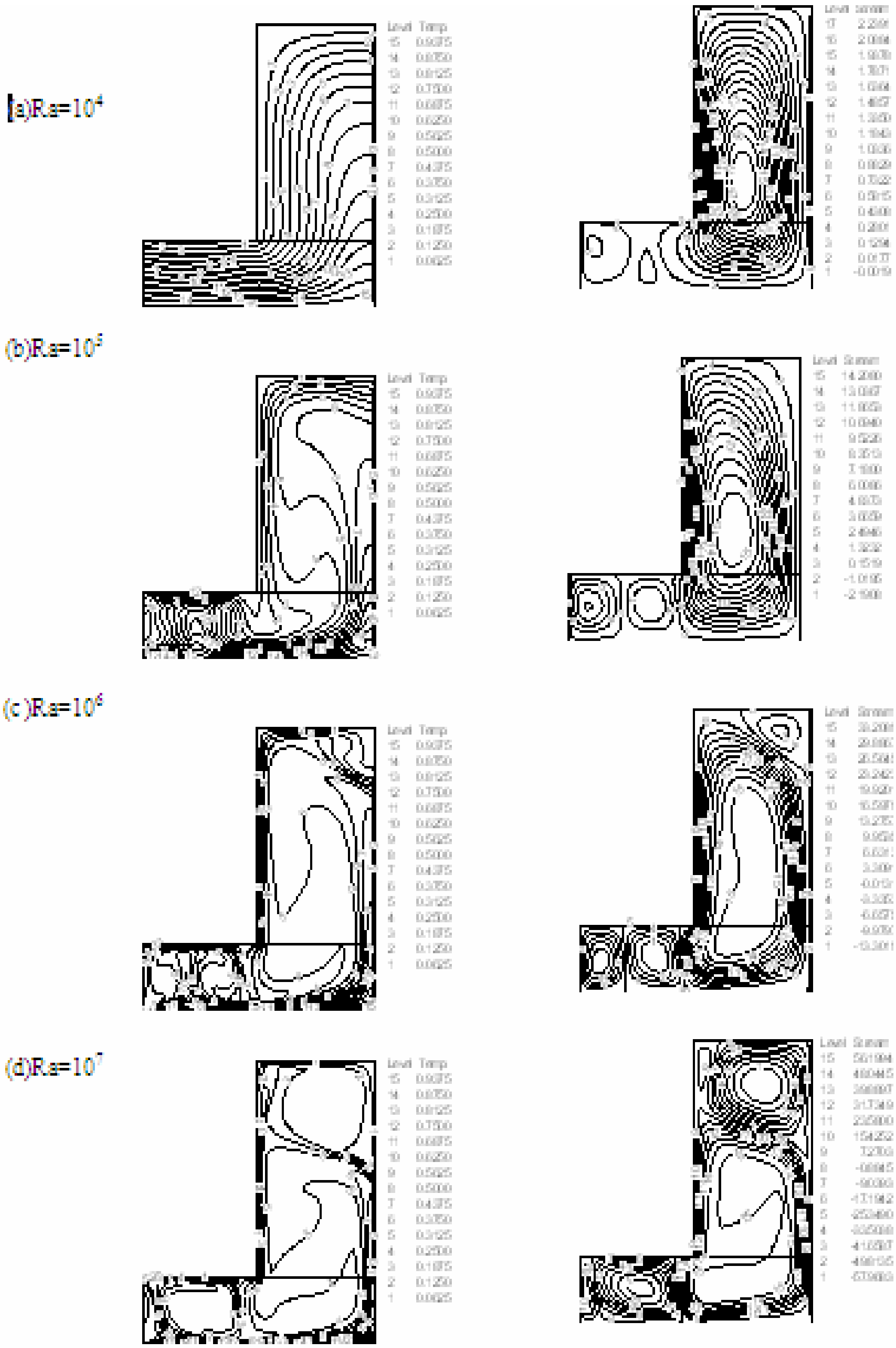

Fig. (3) Effects of Ra number on the temperature contours and stream function without fins and aspect ratio (H/L= 0.25).

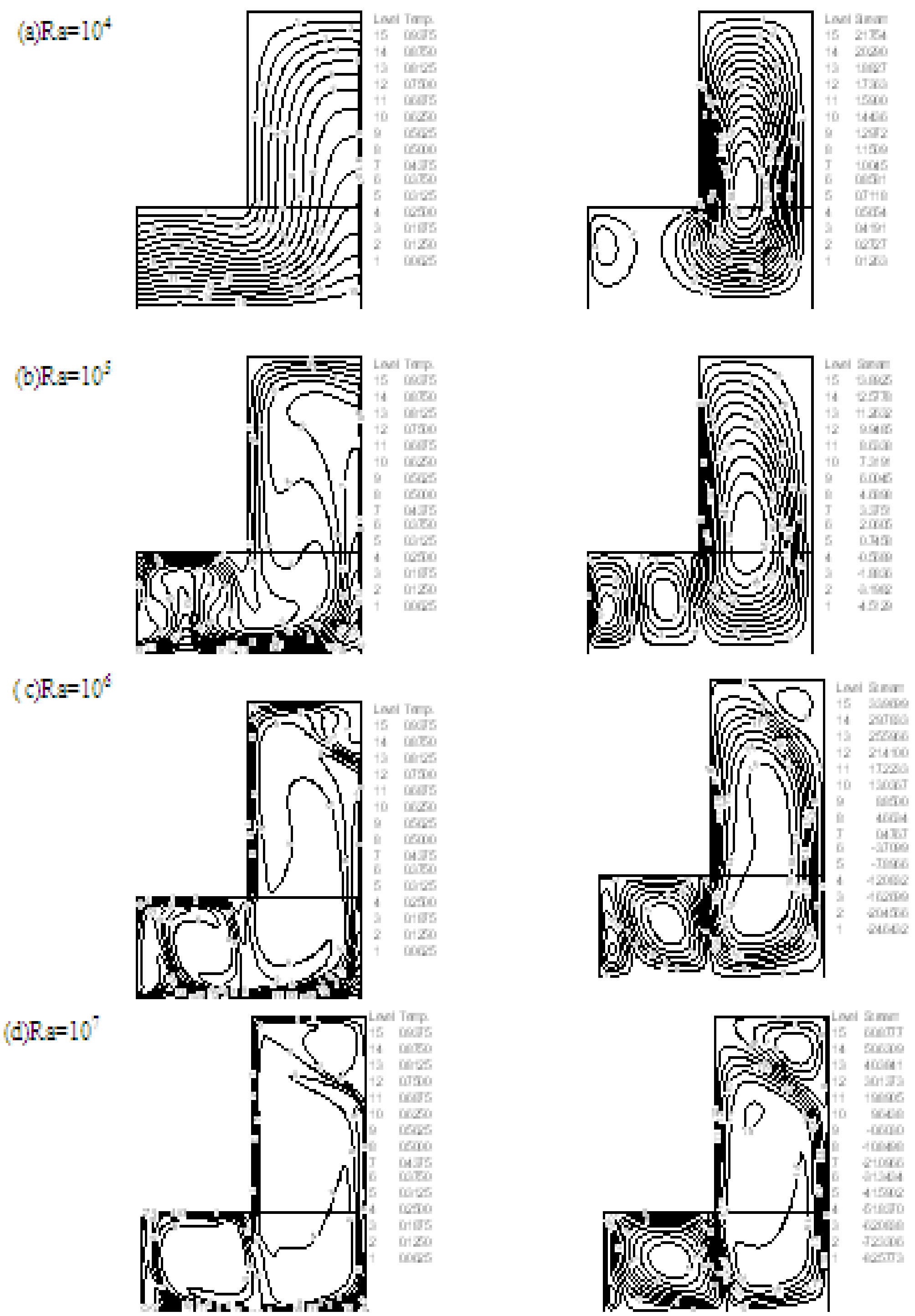

Fig. (4) Effects of Ra number on the temperature contours and stream function without fins and aspect ratio (H/L= 0.35).

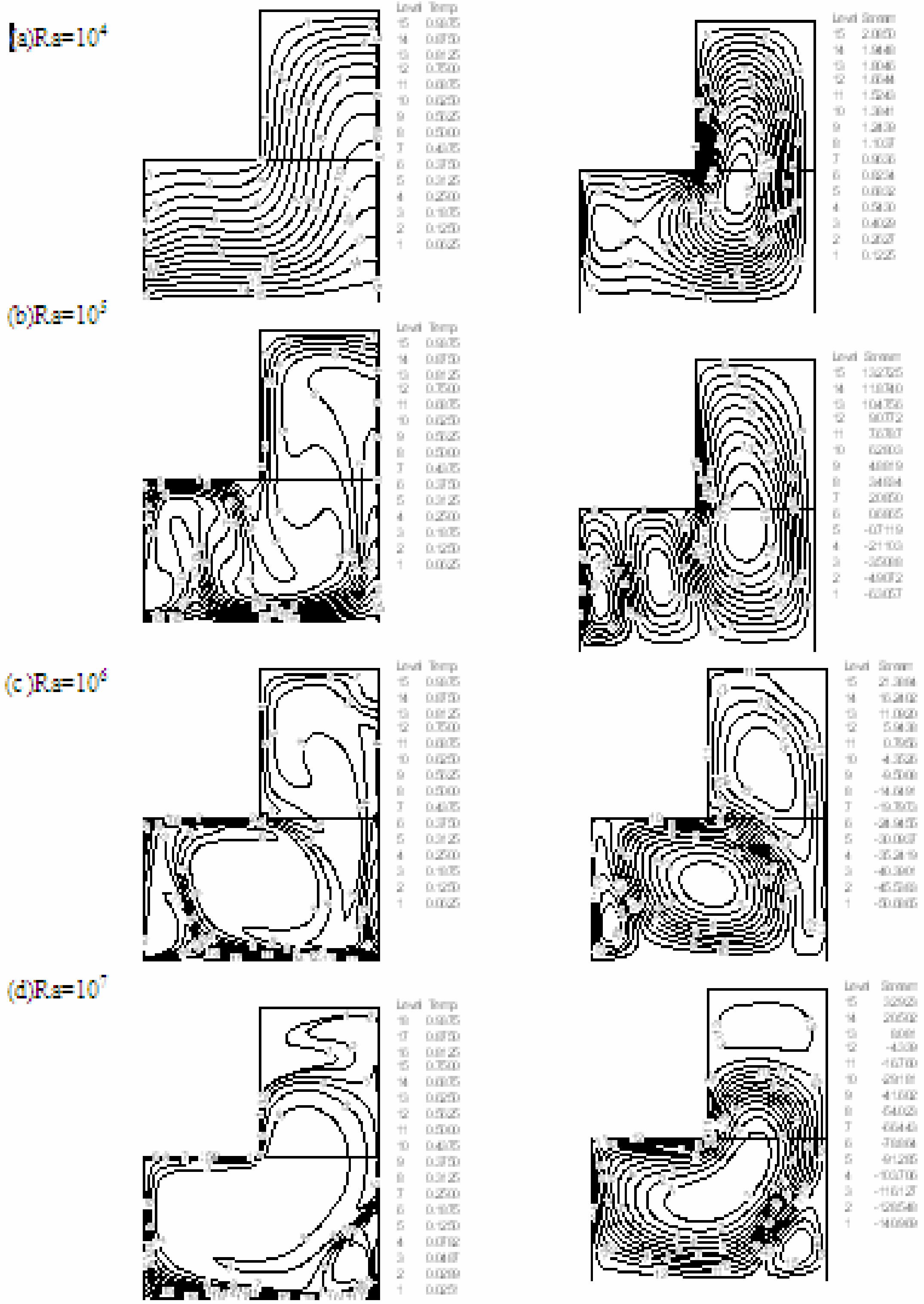

Fig. (5) Effects of Ra number on the temperature contours and stream function without fins and aspect ratio (H/L= 0.5).

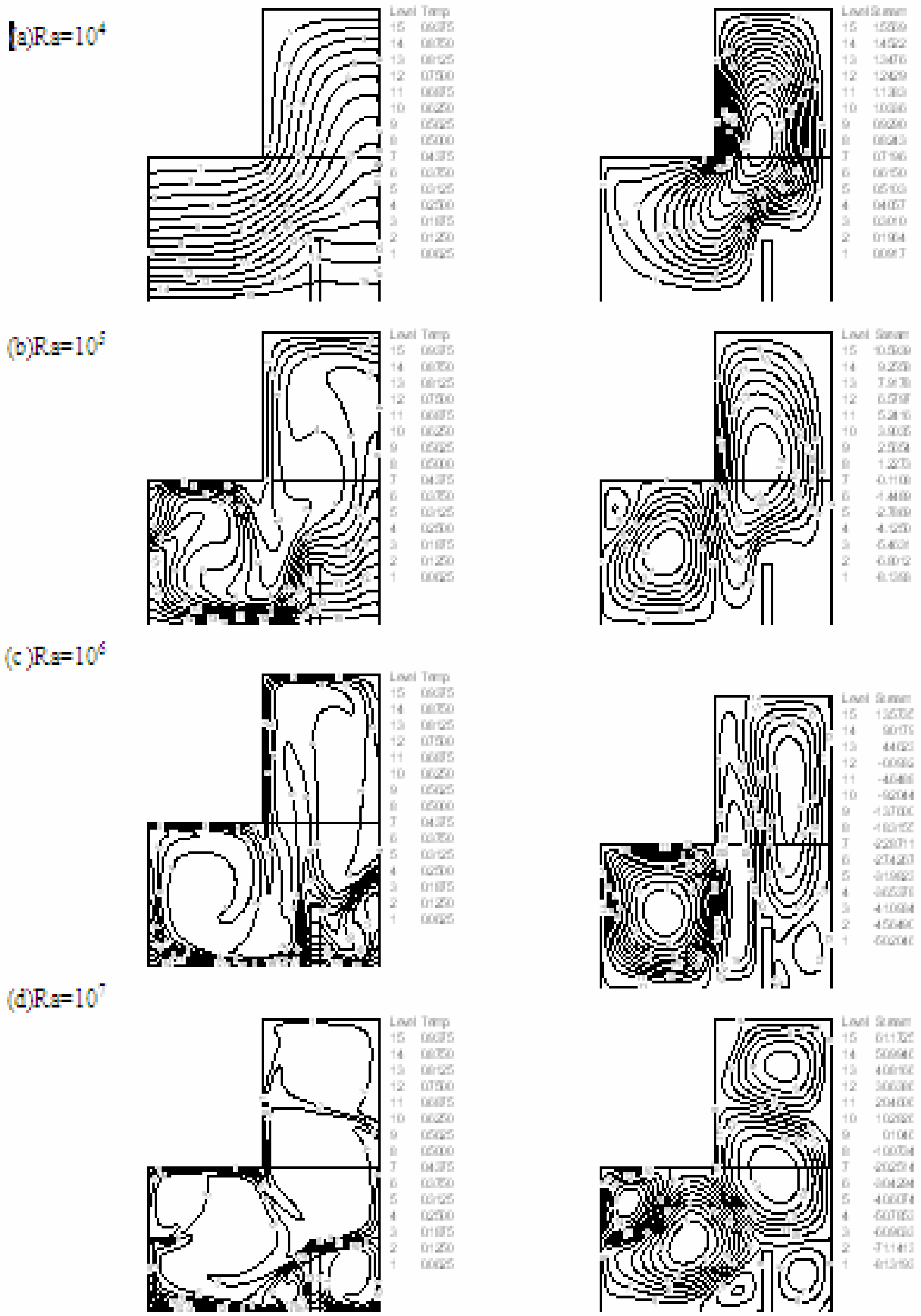

Fig. (6) Effects of Ra number on the temperature contours and stream function for fin at lower wall, height of fin (B/L=0.22) and aspect ratio (H/L= 0.5).

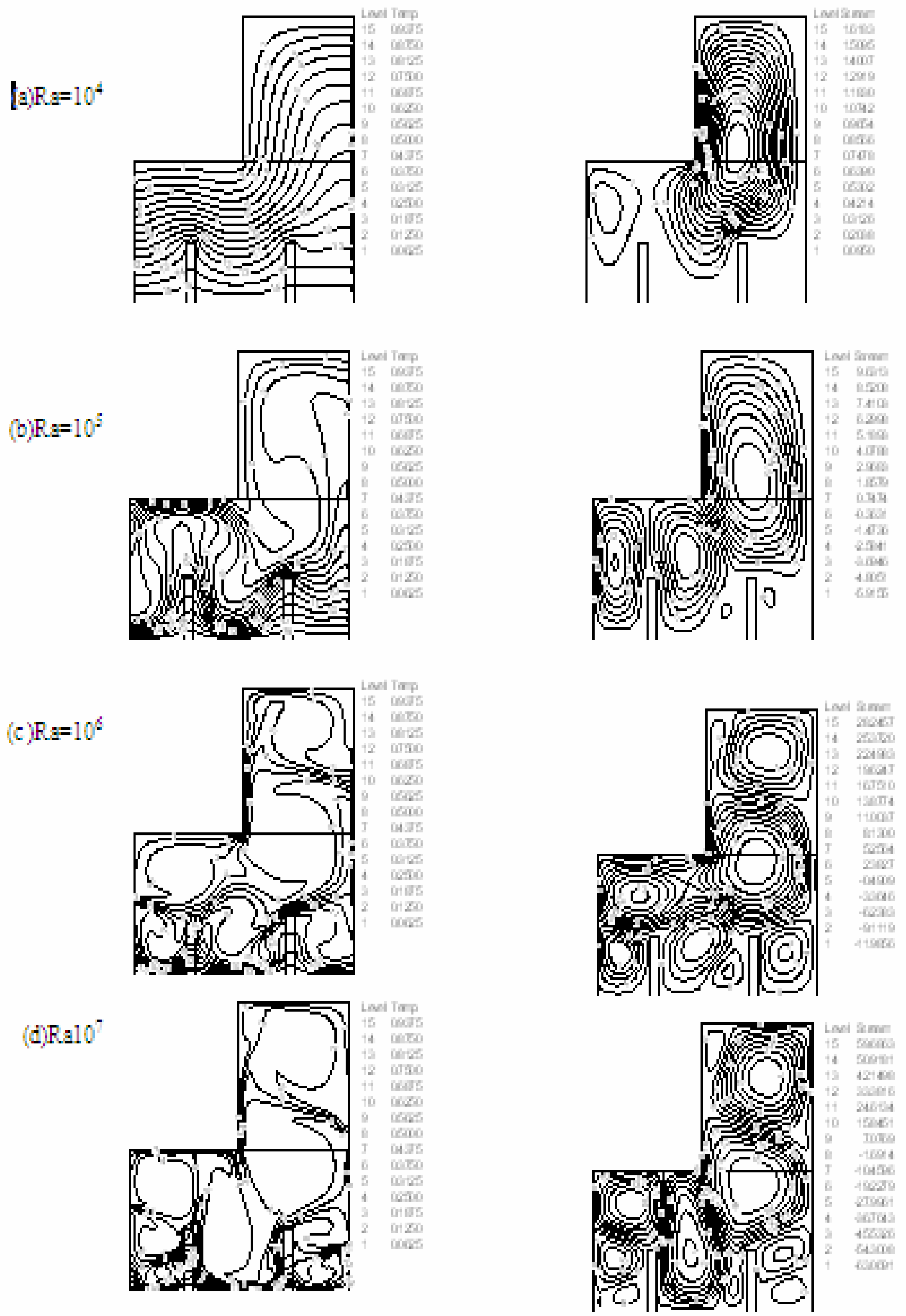

Fig. (7) Effects of Ra number on the temperature contours and stream function for number of fin (2), height of fin (B/L=0.22) and aspect ratio (H/L= 0.5).

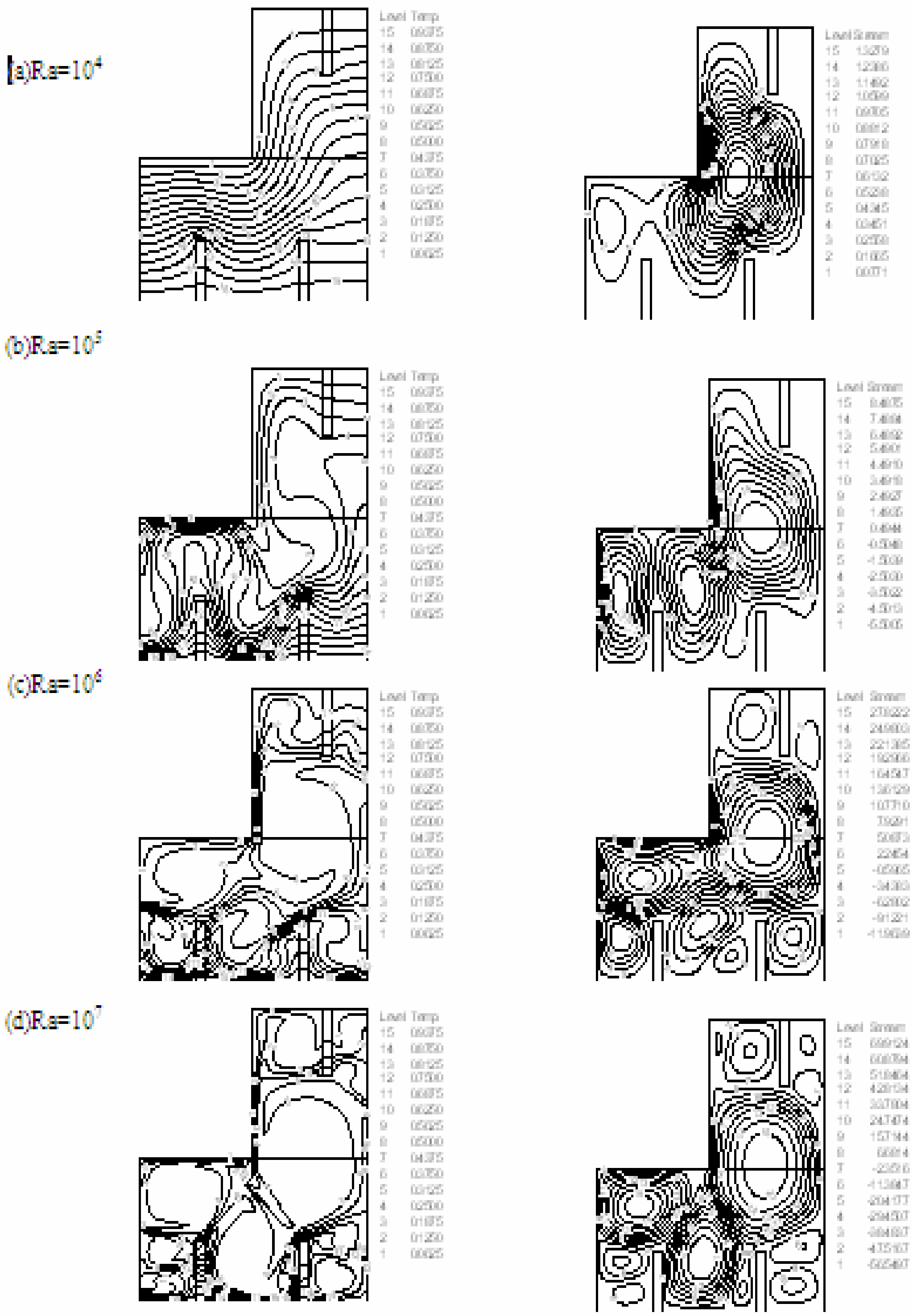

Fig. (8) Effects of Ra number on the temperature contours and stream function for number of fin (3),height of fin (B/L=0.22) and aspect ratio (H/L= 0.5).

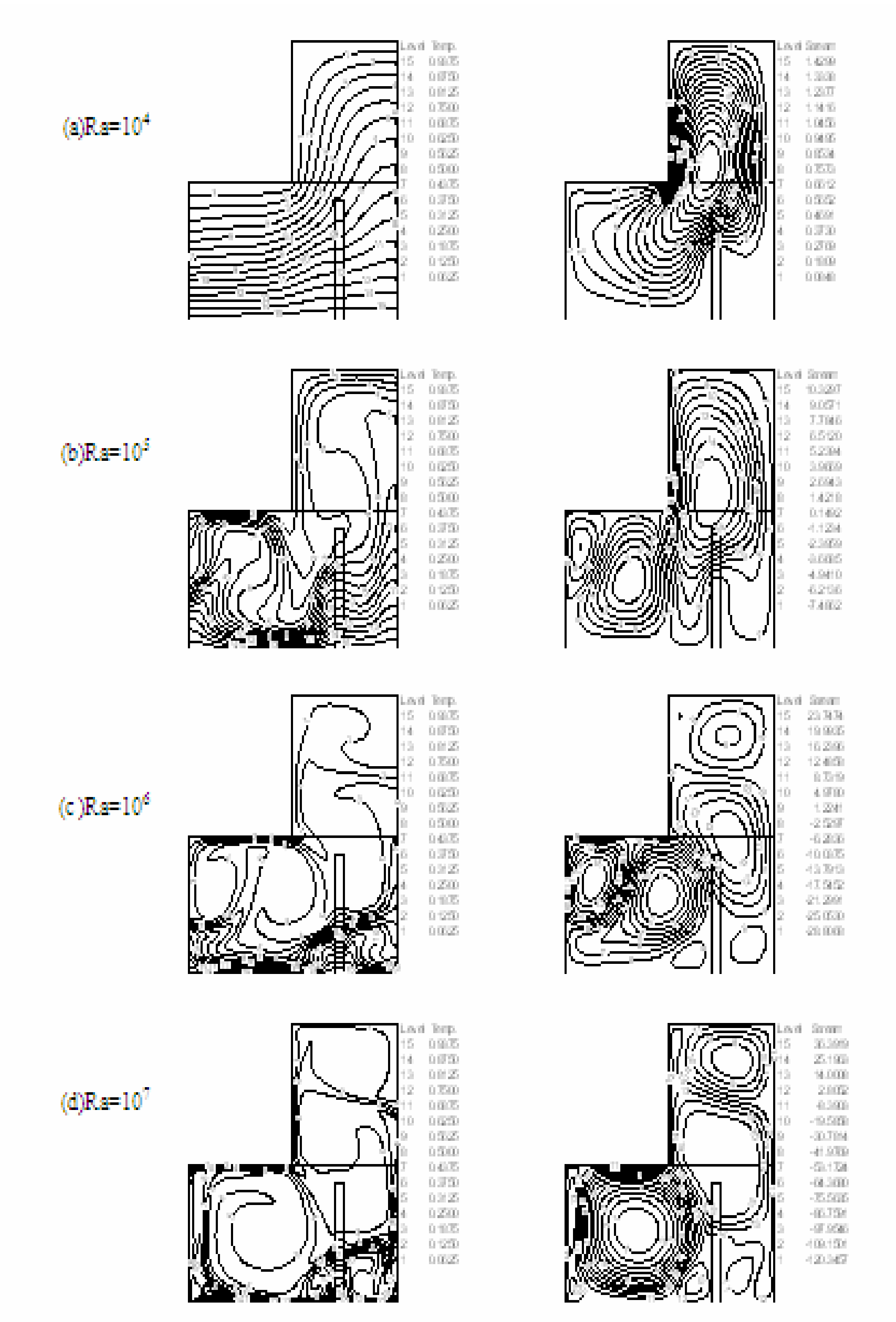

Fig. (9) Effects of Ra number on the temperature contours and stream function for fin at lower wall, height of fin (B/L=0.44) and aspect ratio (H/L= 0.5).

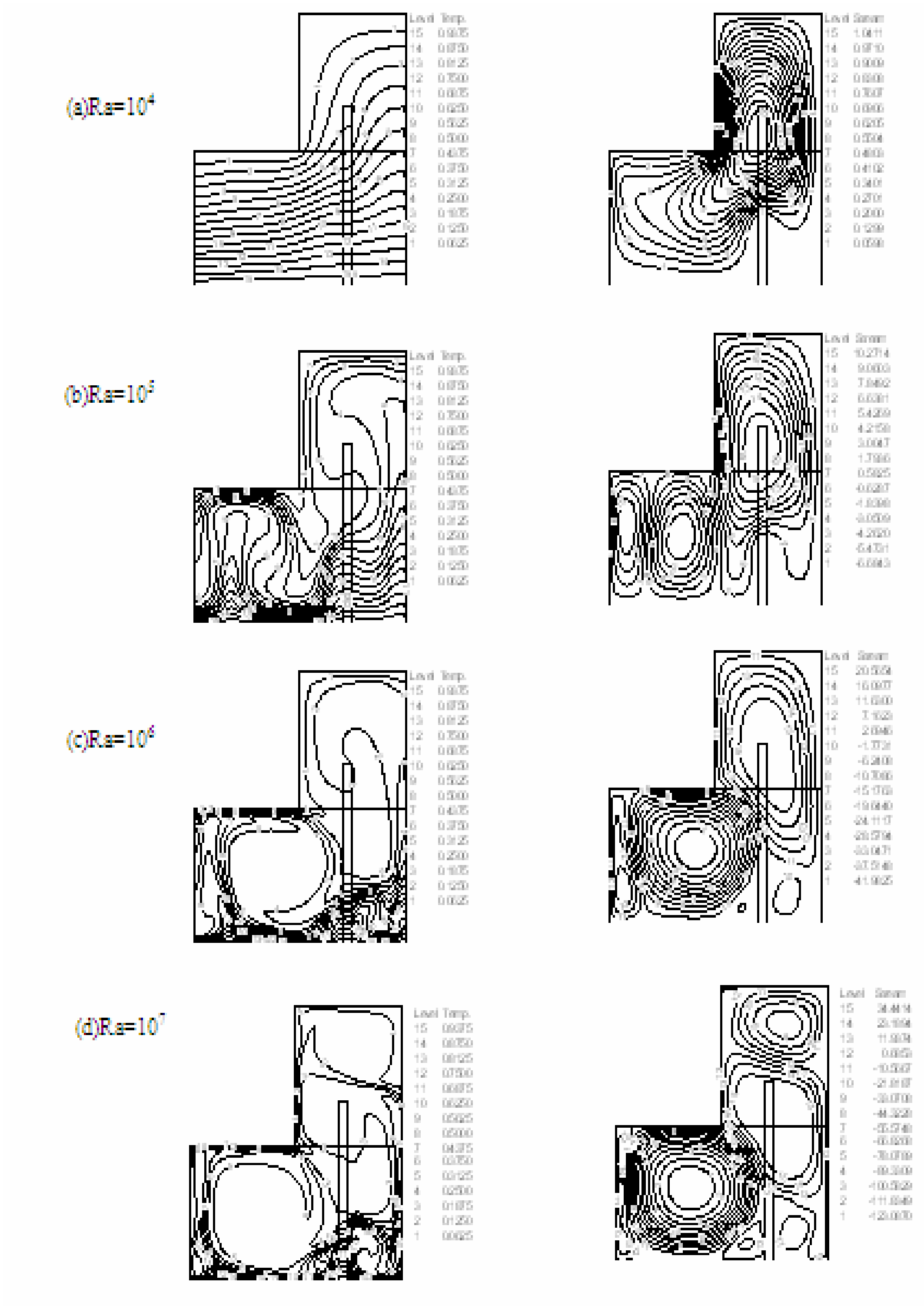

Fig. (10) Effects of Ra number on the temperature contours and stream function for fin at lower wall, height of fin(B/L=0.66) and aspect ratio (H/L= 0.5)

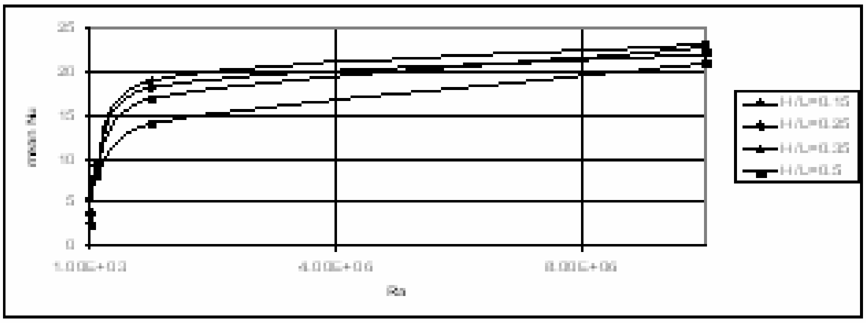

Fig.(11)Variations of mean Nusselt number with the Rayleigh number at different number of aspect ratio (H/L).

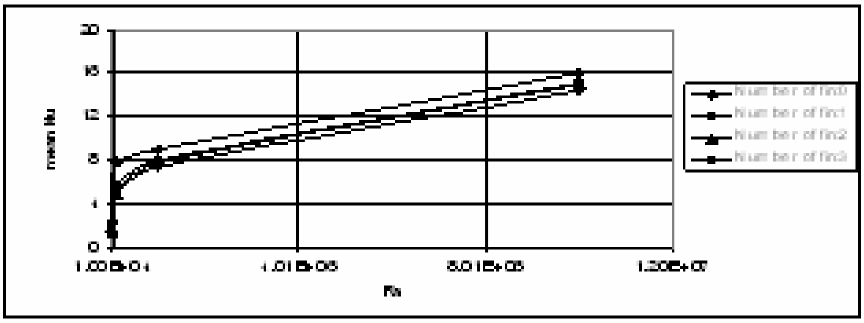

Fig.(12)Variations of mean Nusselt number with the Rayleigh number at different number of fins for (H/L = 0.50).

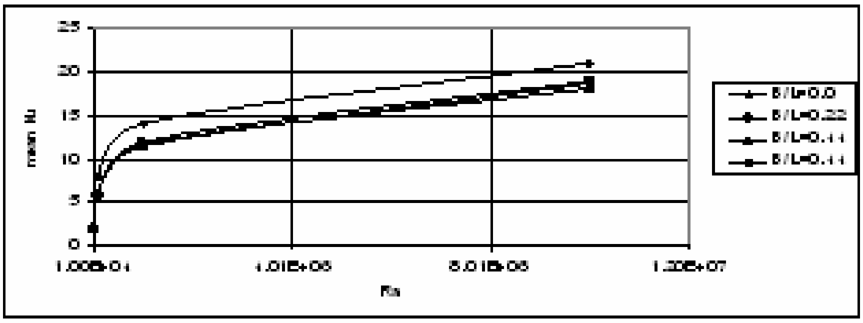

Fig.(13)Variations of mean Nusselt number with the Rayleigh number at different value height of fin (B/L).

(a) Present study (b) Present study (c) Syeda and Shohel

Fig. (14) Comparison between present Study and [Syeda and Shohel 2006] for Streamlines (solid lines) and isothermal lines (dashed lines) for $Ra=10^4$.

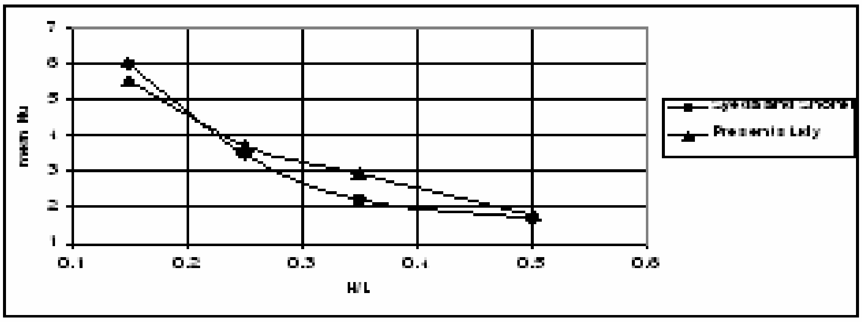

Fig. (15) Comparison between present Study and [Syeda and Shohel 2006] for mean Nusselt number with aspect ratio (H/L) at $Ra=10^4$.

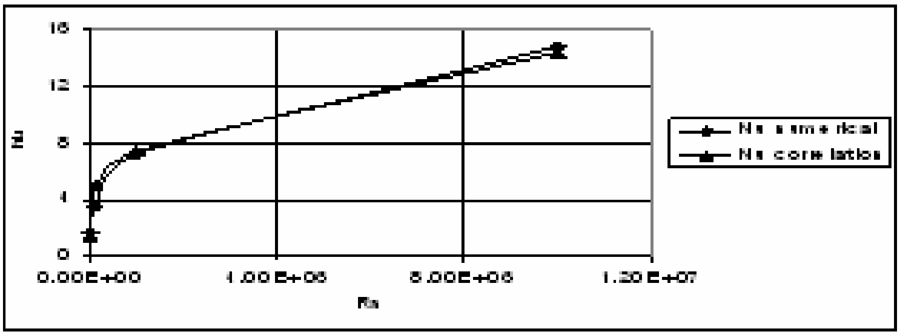

Fig. (16) Comparison of Nu computed numerically with Nu calculated from correlation Eq. (31) for case effect number of fins.