

## **NUMERICAL STUDY OF STEADY NATURAL CONVECTION FLOW IN A PRISMATIC ENCLOSURE WITH STRIP HEATER ON BOTTOM WALL USING FLEXPDE**

**Sana Jaafar Yaseen**

Mechanical Engineering / College of Engineering / AL- Basra University/ Iraq

E-mail:sanajaafar@yahoo.com

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**ABSTRACT:-** Laminar natural convection in two-dimensional Prismatic enclosure is studied and analysis numerically. For the enclosure top inclined walls are considered at low temperature, two vertical walls are adiabatic and strip heater at constant high temperature mounted on the bottom enclosure, while the remainder bottom wall kept at low known temperature. The partial differential equations for two dimensional conservation of mass, momentum and energy are solved using finite element software package (FLEXPDE.5). For Rayleigh number varying from  $10^3$  to  $10^5$  and for constant Prandtl number  $Pr=0.7$  the change in temperature and flow fields (stream functions) were investigated for different heater locations and for different number of heaters. The effect of the number and locations of the strip heaters on local and mean Nusselt number were examined. Results were presented by streamlines, isotherms and Nusselt number and it indicates that the Nusselt number is significantly affected by increasing both Ra and number of heaters. A comparison of the streamlines, isotherms curves and average Nusselt at the same boundary conditions was made with that obtained by Tanmay et al.<sup>(7)</sup>, and showed a good agreement.

**Keywords:** Free convection, Prismatic and trapezoidal enclosures, Uniform Wall Temperature

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### **1- INTRODUCTION**

Over the past decades, internal (in enclosure) and external natural convection heat transfer has been important in many engineering applications. Several studies have been achieved in the problem of laminar flow forced convection in ducts (enclosures) of various cross-sections because of its applicability in various fields such as compact heat exchangers, air conditioners, nuclear reactors, solar collectors, solar stills, electronic cooling etc <sup>(1)</sup>. Natural convection is formed in some parts of buildings due to temperature difference

between surface and fluid <sup>(2)</sup>. The roof is an important part of the building because it protects the building from the environmental effects <sup>(3)</sup>. The geometry of roofs may be different. For instance, it can be Gable roof (triangular), Gambrel (pentagon) or saltbox (trapezoidal) <sup>(2)</sup>. Natural convection in an enclosure is as varied as the enclosure geometry, its orientation and thermal boundary conditions <sup>(3)</sup>. A large number of literature is available which deal with the study of natural convection in enclosures with different shapes. Ahmet. K. and Yasin.V. <sup>(4)</sup> study the natural convection heat transfer and flow field inside a saltbox roof with eave in winter day conditions. The governing equations of natural convection in stream function-vorticity form were solved using central difference method to obtain flow and temperature fields inside the enclosure. Parametric study for the wide range of Rayleigh number, aspect ratio of the roof and length of eave has done. The results indicated that both eave length and aspect ratio could used as control parameter for heat transfer. Tanmay Basak et al <sup>(5)</sup>. study the natural convection within trapezoidal enclosures for uniformly and non-uniformly heated bottom wall, insulated top wall and isothermal side walls with inclination angle  $\phi$ . Numerical simulations were performed using Galerkin finite element method with penalty parameter to solve the nonlinear coupled partial differential equations for flow and temperature fields. Parametric study for the wide range of Rayleigh number  $Ra$   $10^3$  to  $10^6$  and Prandtl number  $0.026 < Pr < 1000$  with various tilt angles. The local Nusselt numbers has also been shown for side and bottom walls. It was founded that average heat transfer rate does not vary significantly with  $\phi$  for non-uniform heating of bottom wall. Tanmay B, et al <sup>(6)</sup>. reported the numerical results of natural convection in trapezoidal enclosures for uniformly heated bottom wall, linearly heated vertical wall and of insulated top wall. Parametric studies for the wide range of Rayleigh numbers  $Ra$  ( $10^3$ - $10^5$ ) and Prandtl numbers  $Pr$  ( $0.7$  - $1000$ ) with various tilt angles of side walls  $\phi$  have been performed. It was found that, higher heat transfer rates for  $\phi=0$  and the overall heat transfer rates at the bottom wall was larger for the linearly heated left wall. Tanmay Basak et al <sup>(7)</sup> investigated the effects of uniform and non-uniform heating of inclined walls on natural convection flows within a isosceles triangular enclosure using a penalty finite element analysis. The numerical solution of the problem is presented for various Rayleigh numbers ( $Ra$ )  $10^3$  to  $10^6$  and Prandtl number  $0.026 < Pr < 1000$ . Numerical results are obtained to display the circulations and temperature distributions within the triangle. It has been found that at small Prandtl numbers, geometry does not have much influence on flow structure while at  $Pr = 1000$ , the geometry has considerable effect on the flow pattern. It is observed that non-uniform heating produces greater heat transfer rates at the center of the walls than the uniform heating.

In the present study, a Prismatic enclosure with a strip heater mounted on the bottom wall was studied. The effect of Rayleigh number, location and number of strip heaters on the flow and thermal fields were studied in detail.

## 2- BOUNDARY CONDITIONS AND ASSUMPTIONS

Consider laminar natural convection of (Newtonian fluid) in a Prismatic enclosure. The temperature of the bottom wall ( $T_c$ ) is the same as that of the incline walls. The discrete heater was assumed at constant temperature of ( $T_h$ ) is higher than that of the Prismatic wall ( $T_c$ ) and both are uniform (at constant wall temperature). The lengths of the bottom wall is ( $L$ ) and height is ( $h+w$ ). The dimensionless length of the heater,  $\ell /L$ , is constant at 0.2 while the relative location  $s/L$  changes as 0.2, 0.5 and 0.75. These positions refer to placing the heated section at the bottom corner, the middle and at the last part of the bottom wall. The interest domain was shown in Fig. (1), the origin of the Cartesian coordinates ( $x,y$ ) is positioned at the left bottom side of the Prismatic. The range of Rayleigh number  $10^3 \leq Ra \leq 10^5$  and  $Pr=0.7$  were considered in this study. The fluid was assumed to have constant physical properties but obeys the Bussinesq approximation according to which the compressibility effect everywhere is neglected except for Buoyancy force term. Viscous dissipation and heat generation are absent.

The appropriate boundary conditions are as follows

- Isothermal surfaces i.e.  $\theta=0$  on the incline walls and  $\theta=1$  on the hot strip heater.
- No-slip velocity boundary condition,  $U=V=0$  on all solid walls.
- pressure gradient normal to all surfaces is zero,  $\frac{\partial P}{\partial n} = 0$ , where  $n$  is a normal unit vector.

## 3. GOVERNING EQUATIONS

The dimensionless form of the two-dimensional, incompressible, steady state governing equations using conservation of mass, momentum and energy can be written as: <sup>(8, 9, and 10)</sup>

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

$$U \frac{\partial U}{\partial X} + V \frac{\partial U}{\partial Y} = -\frac{\partial P}{\partial X} + Pr \left( \frac{\partial^2 U}{\partial X^2} + \frac{\partial^2 U}{\partial Y^2} \right) \quad (2)$$

$$U \frac{\partial V}{\partial X} + V \frac{\partial V}{\partial Y} = -\frac{\partial P}{\partial Y} + Pr \left( \frac{\partial^2 V}{\partial X^2} + \frac{\partial^2 V}{\partial Y^2} \right) + Ra \cdot Pr \cdot \theta \quad (3)$$

$$U \frac{\partial \theta}{\partial X} + V \frac{\partial \theta}{\partial Y} = \frac{\partial^2 \theta}{\partial X^2} + \frac{\partial^2 \theta}{\partial Y^2} \quad (4)$$

Non-dimensional parameters can be given as follows:

$$W = \frac{w}{L}, \quad H = \frac{h}{L}, \quad U = \frac{uL}{\alpha}, \quad V = \frac{vL}{\alpha}, \quad \theta = \frac{T - T_c}{T_h - T_c}, \quad \text{and } P = \frac{(p + \rho g y)H^2}{\rho \alpha^2}$$

$$\text{Rayleigh number } Ra = \frac{g\beta(T_h - T_c)H^3}{\nu\alpha}, \quad \text{and Prandtl } Pr = \frac{\nu}{\alpha}$$

## 4. SOLUTION PROCEDURE

It is well known in the numerical solution field that the set of equations (Eqs. (1) to (3)) may be highly oscillatory or even sometimes undetermined because of inclusion of the pressure term in the momentum equations. In the present study, a finite element software package (Flexpde.5) is relied on in solution of the nonlinear system of equations. In finite element method there is a derived approach with purpose of stabilizing pressure oscillations and allowing standard grids and elements. This approach enforces the continuity equation and the pressure to give the following, what called penalty approach <sup>(9)</sup>

$$\nabla^2 P = \lambda \left( \frac{\partial U}{\partial X} + \frac{\partial V}{\partial Y} \right) \quad (5)$$

The continuity equation (1) is automatically satisfied for large values of  $\lambda$  which called the penalty parameter and its value given as constant in our study taken equal to  $10^7$  as in reference<sup>(7)</sup>. Using Eq.(5), the momentum balance Eqs. (3) and (4) reduce to<sup>(7,9)</sup>:

$$U \frac{\partial U}{\partial X} + V \frac{\partial V}{\partial Y} = -\lambda \frac{\partial}{\partial X} \left( \frac{\partial U}{\partial X} + \frac{\partial U}{\partial Y} \right) + \frac{\partial^2 U}{\partial X^2} + \frac{\partial^2 U}{\partial Y^2} \quad (6)$$

$$U \frac{\partial V}{\partial X} + V \frac{\partial V}{\partial Y} = -\lambda \frac{\partial}{\partial Y} \left( \frac{\partial U}{\partial X} + \frac{\partial U}{\partial Y} \right) + \frac{\partial^2 V}{\partial X^2} + \frac{\partial^2 V}{\partial Y^2} + (Ra Pr)\theta \quad (7)$$

## 5. EVALUATION OF STREAM FUNCTION AND NUSSELT NUMBER

### 5.1. Stream function

Analysis of the flow is displayed by mean of stream function  $\psi$  obtained from velocity components  $U$  and  $V$ . The relationships between stream function  $\psi$  and velocity components for two dimensional flows are <sup>(7)</sup>:  $U = \frac{\partial\psi}{\partial Y}$ ,  $V = -\frac{\partial\psi}{\partial X}$  which yield a single equation:

$$\frac{\partial^2\psi}{\partial X^2} + \frac{\partial^2\psi}{\partial Y^2} = \frac{\partial U}{\partial Y} - \frac{\partial V}{\partial X} \quad (8)$$

$$\frac{\partial^2\psi}{\partial X^2} + \frac{\partial^2\psi}{\partial Y^2} = \frac{\partial U}{\partial Y} - \frac{\partial V}{\partial X} \quad (9)$$

Hence, the continuity equation (1) was excluded from solution system and replaced by equation (8). The no-slip condition is valid at all boundaries as there is no cross flow, hence  $\psi = 0$  on all surfaces.

## 5.2. Nusselt Number

Heat transfer rate is measured by local and average Nusslet number. The following equations are used to calculate the local Nusselt number ( $Nu_L$ ) along the bottom wall is defined by Eq.(8):

$$Nu_L = -\frac{\partial\theta}{\partial n} \quad (10)$$

where  $n$  denotes the normal direction on a plane and  $(\frac{\partial\theta}{\partial n})$  is the temperature gradient normal to the hot wall. The average Nusselt number  $\overline{Nu}$  can be evaluated along the strip heater as (9,11):

$$\overline{Nu} = \frac{1}{\ell} \int_{\ell} Nu_{\ell} dL \quad (11)$$

## 6. VALIDATION

### 6.1 Software Validation

To check the validation of software, the grid dependency and the distribution values of  $(\partial U/\partial X + \partial V/\partial Y)$  over the domain  $Ra=10^3$  with accuracy  $10^{-4}$  are presented in Fig.2, a and b. It is clear from that figure that the continuity equation is exactly validated.

### 6.2 Validation of numerical results

To validate the numerical analysis, this code is used in the same geometry, with the same boundary conditions used in Tanmay et al <sup>(7)</sup>. This geometry is an isosceles triangular cavity heated from below and cooled at the inclined walls. The profile of the dimensionless

temperature and stream function at the bottom in the present study and in Tanmay et al.<sup>(7)</sup> was compared for ( $Ra=10^5$ ,  $Pr=0.026$ ) and satisfactory agreement was observed as shown in Fig.(3). The same code was tested against the results obtained by the same reference comparing the average Nusselt number along the bottom of the triangular enclosure. Excellent agreement observed as reported in Table (2).

## 7. RESULTS AND DISCUSSION

The study considers buoyancy driven motion of air ( $Pr = 0.7$ ) created by a strip heater at constant temperature mounted on the bottom wall of the enclosure. In this numerical work, there are some governing parameters, which describe different physical behavior of the flow and heat transfer by natural convection in a prismatic enclosure such as Rayleigh number value and the number and location of strip heaters mounted on the bottom wall. In the next subsections, we will discuss the effects of these parameters on the flow and heat transfer characteristics with the help of isotherms and the streamline patterns. Local and average Nusselt numbers on the isothermal surfaces are plotted to evaluate the local and overall heat transfer rates.

### 7.1 Flow Fields and Temperature Fields

The isothermal lines and stream functions as Rayleigh number varies from  $10^3$  to  $10^5$  for  $s/L= 0.1$  are shown in Fig.(4). Heat dissipated from the strip heater develops a fluid layer that moves upward the hot fluid loses part of its energy to the cold section of the same wall and moves along the isothermal incline wall then down along the right side wall forming a single central cell with a center shifted to the left from that of the enclosure. The cold wall above the heater, suppresses this buoyancy effect. The isotherms are clustered near the hot section, in nearly parallel lines, indicating domination of diffusion heat transfer mode close to the bottom section of the heater. It is noticed that the pattern of the streamlines changes with increasing Rayleigh number the effect is pronounced at  $Ra =10^5$  where the main cell is distorted and prolonged to plume distribution shape in the lower part.

The streamlines and isotherms when the strip heater is placed at the center of the bottom wall,  $s/L= 0.5$  are presented in Fig. (5) for different Rayleigh numbers. As expected, due to the cold incline walls, fluids rise up from middle portion of the bottom wall and flow down along the two vertical walls forming two symmetric rolls with clockwise and anti-clockwise rotations inside the cavity. At  $Ra =10^3$  the magnitudes of stream function are very

low and the heat transfer is primarily due to diffusion. It enhances with increasing Ra number due to domination of convection mode of heat transfer.

When the heater is placed near the end of the bottom wall,  $s/L=0.7$ , this can be seen in Fig.(6) at  $Ra=10^3$  and  $10^4$  two circulation cells appear a big one at the left side and a small one eddy circulation cell appears in the right side. An increase in Rayleigh number to  $10^5$  causes the stream function increases four times than that at  $Ra=10^3$ . Increase in Rayleigh number to  $10^5$ , causes sharp temperature gradients along the bottom wall, due to the buoyant effects caused by the temperature difference between the heater and outer walls, recirculation vortices is formed which are clearly demonstrated by the closed stream lines the plume start to growth to upward due to reducing the density of fluid which lead to growth the thermal boundary layer. Distortion of the isotherms where the heat propagates more towards the cold wall opposite to the heater.

Two symmetrical big cells and two small can be seen in Fig.(7) when using two heaters, at equal distance on the bottom wall of the enclosure ( $s/L=0.25,0.75$ ). As Rayleigh number increases from  $10^3$  to  $10^5$  the boundary layers build-up near the heated section and along portion of the opposite cold wall and flow down along the two vertical walls forming two symmetric rolls with clockwise and anti-clockwise rotations inside the cavity with their center shifted to the bottom of the enclosure.

At  $Ra=10^3$  the magnitudes of stream function are very low and the heat transfer is primarily due to diffusion. It enhances with increasing Rayleigh number due to domination of convection mode of heat transfer. Further increase in Rayleigh number to  $10^5$ , causes sharp temperature gradients along the hot strip.

Fig.(8) shows streamlines and isotherms for different values of Rayleigh number when three heaters mounted on the bottom wall. There are two symmetric circulation cells formed inside the enclosure with their center shifted to the left from that of enclosure. An increase in Rayleigh number causes a distortion of the isotherms where the heat propagates more towards the cold wall.

Fig.(9) is plotted for local Nusslet number along the strip heater for different locations ( $s/L=0.1,0.5$  and  $0.7$ ). As expected, at constant location the maximum value of local Nu number becomes greater as Ra number increased due to the increment of the heat transfer rate. On the other hand, the value of Nu number reduces at the middle of each strip heater and increase at the start and the end of heater due to the large change in temperature between heater and the cold wall surrounded the heater.

The average Nu number is utilized to represent the overall heat transfer rate within the domain of interest. The overall effect of Ra number on the average Nu number for different

location of heater is display in Fig.(10). From that figure, it is obvious that the average Nu number along the strip heater is smoothly increased as Ra number increased, the increment of Ra gives increasing in Nu, it may be attributed to increase the temperature difference and bouncy effecting with increasing Ra. It is observed that the maximum value of  $\overline{Nu}$  occurs at  $s/L=0.5$  (at the middle of the bottom wall).

The value of mean Nu numbers when using two heaters at locations ( $s/L=0.25, 0.75$ ), can be shown Fig.(11). As can be seen from that figure value of  $\overline{Nu}$  for each heater nearly equal, this because the heaters locate at the same distance from the sides of the enclosure, so the heat transient from each heater equal. The increment of Ra gives increasing in the Nu number, it may be attributed to the increase in the temperature difference and bouncy effecting with increasing Ra number. There is no direct relation between Nu number and Ra, but for this case and for a specific range of Ra ( $Ra>2100$ ) Nu number is proportional directly with Ra.

On the other hand when using three heaters, putting at ( $s/L=0.2, 0.5, 0.8$ ) the value of  $\overline{Nu}$  on each heater can be shown in Fig.(12). It is observed from that figure as Ra number increase the value of  $\overline{Nu}$  also increase. On the other hand, the value of  $\overline{Nu}$  is the same for the heater location of ( $s/L=0.2$ ) and ( $s/L=0.8$ ), since its locate at the same distance from the sides of the bottom wall. and the lowest value of  $\overline{Nu}$  occurs at ( $s/L=0.5$ )

Summary of the overall Nusselt number among those five cases is listed in Table (3).

## 8. CONCLUSIONS

In this paper natural convection in an air filled Prismatic cavity with a strip discrete heater is studied. A detailed analysis for the distribution of streamlines, isotherms and Nusselt number was carried out to investigate the effect of the locations of the strip heater at constant temperature mounted on a bottom wall on the fluid flow and heat transfer in the enclosure for Rayleigh numbers in the range of  $10^3 \leq Ra \leq 10^5$ . The conclusions can be drawn as follows:

1. Flow fields and isotherms are strongly affected by changing Rayleigh number, the location of strip heater ( $s/L$ ), and the number of heaters.
2. The value of mean Nusselt number is significantly enhanced with increasing Rayleigh number due to more contribution from natural convection.
3. For the single heater, the optimal location that gives the higher heat transfer rate, maximum value of average Nu number, is  $s/L = 0.5$ , compared to other locations, for all values of Ra.
4. The average Nusselt number could be significantly increased by using more than one heater due to increase the surface area of the hot heater so the heat transfer from the heater to the cold area surrounding the heater increased.

5. When using two heaters putting at ( $s/L=0.25, 0.75$ ) the value of  $\overline{Nu}$  for each one nearly equal.
6. When using three heaters, putting at ( $s/L=0.2, 0.5, 0.8$ ) the value of  $\overline{Nu}$  is the same for the heater which has a location of ( $s/L=0.2$  and  $s/L=0.8$ ), and larger than heater at ( $s/L=0.5$ ).

## 9. REFERENCES

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**Table (1):** Nomenclature

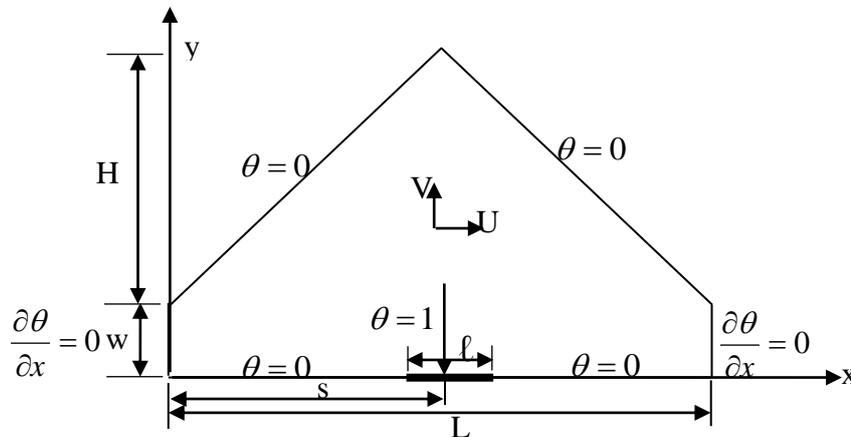
L	Length of the bottom wall (m)	U,V	Dimensionless velocity component
$\ell$	Heater length, (m)	x, y	Cartesian coordinate
s	The location of the strip heater	X,Y	Non-dimensional coordinates
g	Gravitational acceleration ( $m/s^2$ )	<u>Greek symbols</u>	
w, h	Height of the bottom wall (m)	$\alpha$	Thermal diffusivity ( $m^2/s$ )
W, H	Dimensional height of the bottom wall	$\beta$	Thermal expansion coefficient (1/K)
Nu	Nusselt number	$\theta$	Dimensionless temperature
$\overline{Nu}$	Average Nusselt number	$\mu$	Dynamic viscosity ( $N/m^2.s$ )
p,P	Pressure, dimensional( $N/m^2$ ) and dimensionless	v	kinematics viscosity ( $m^2/s$ )
Pr	Prandtl number	$\rho$	Fluid density ( $kg/m^3$ )
Ra	Rayleigh number	$\psi$	Stream function
T	Temperature (K)	$\lambda$	penalty parameter
u, v	Dimensional horizontal & vertical velocity component (m/s)	<u>Subscripts</u>	
		c	cold
		h	hot
		L	Local
		b	bottom

**Table (2):** Comparison between mean Nu number for the present work and that of Tanmay et al.<sup>(7)</sup> for Pr=0.7

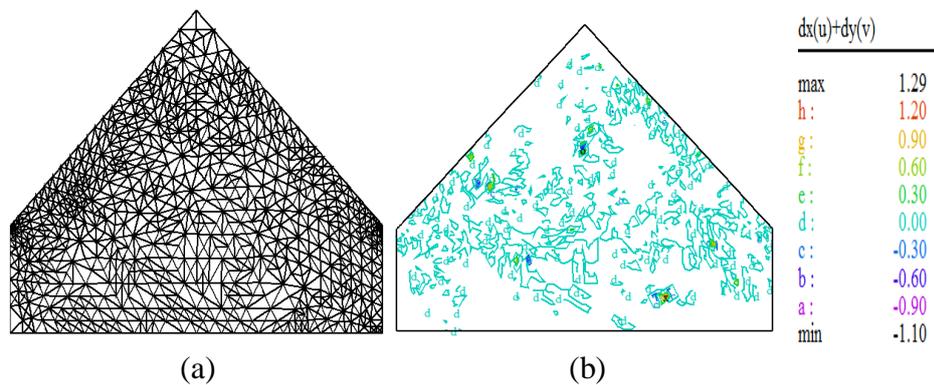
Ra	$\overline{Nu}$ (present)	$\overline{Nu}$ <u>Tanmay et al.<sup>(7)</sup></u>	Error %
$10^3$	5.282	5.5	0.39
$10^4$	5.292	5.55	0.46
$10^5$	5.316	5.73	0.72

**Table (3):** Summary of the present work mean Nu number for different locations and different number of heaters.

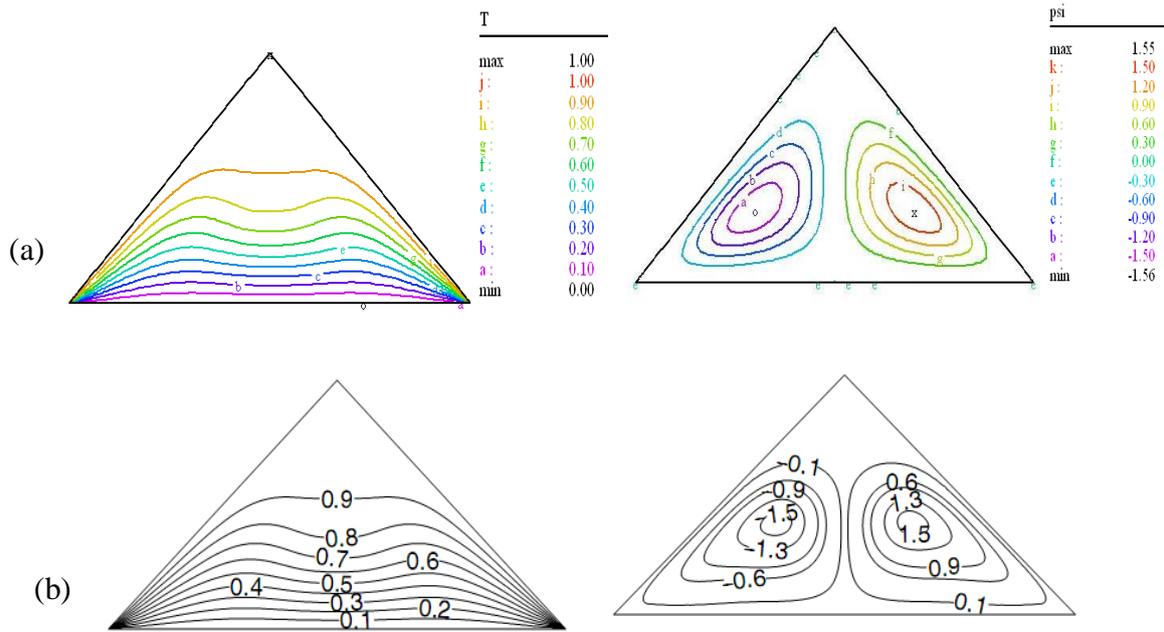
manner	s/L=0.1	s/L=0.5	s/L=0.7	Two heaters	Three heaters
Ra=10 <sup>3</sup>	7.62	11.727	11.5	11.54	10.784
Ra=10 <sup>4</sup>	8.339	11.952	11.9	12.033	11.063
Ra=10 <sup>5</sup>	12.036	13.96	13.4	15.3	13.006



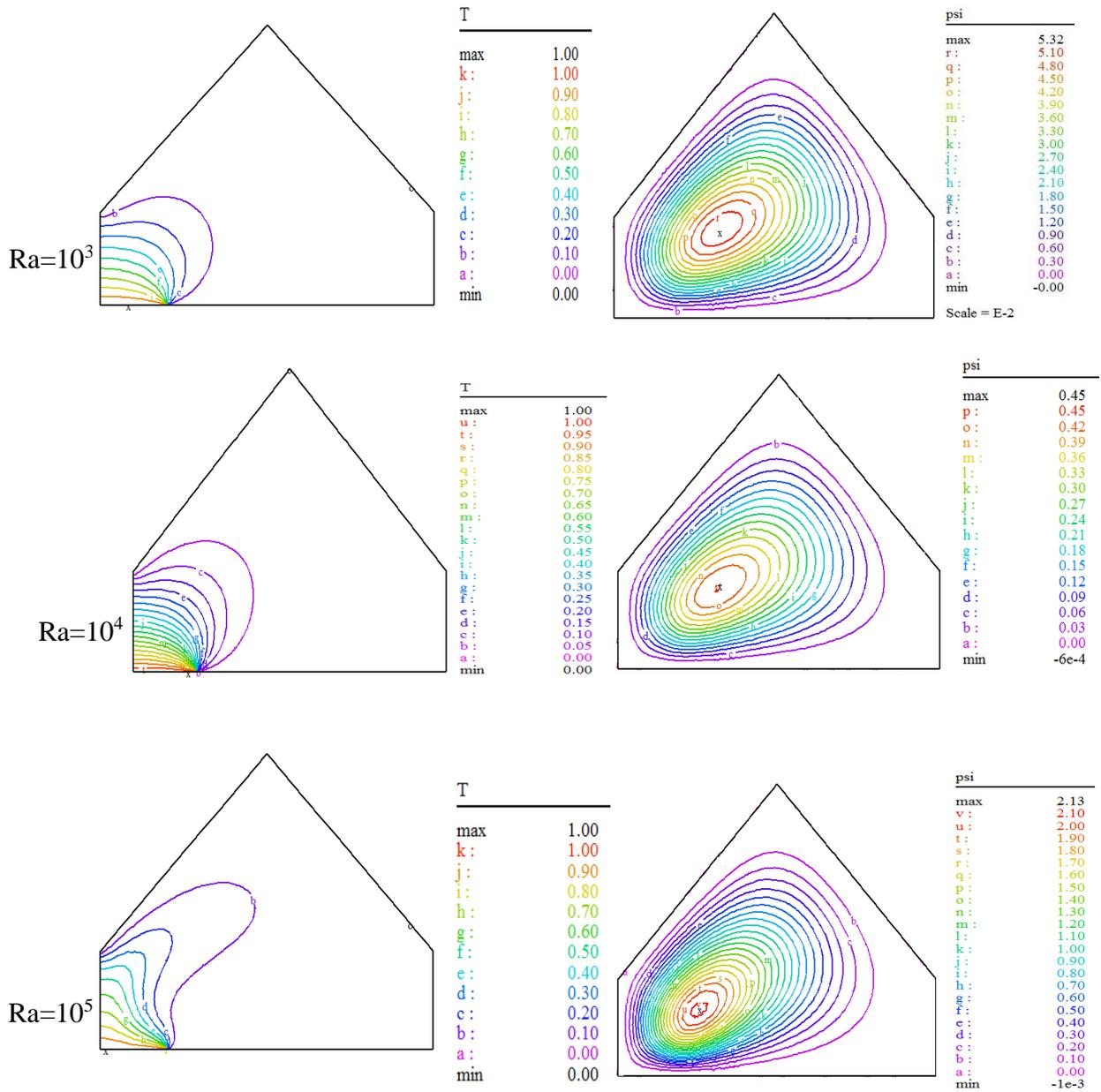
**Fig.(1):** Schematic diagram of the physical system



**Fig.(2):** grid distribution over the domain (b)validation of continuity equation. (for 10<sup>-4</sup> accuracy and Ra=10<sup>3</sup>)



**Fig.(3):** Comparison of streamlines and isotherms for  $Ra=10^5$  and  $Pr=0.026$   
 (a) present study, (b) Tanmay.B et al<sup>(7)</sup>.



**Fig.(4):** Streamlines (right) isotherms (left) for  $s/L=0.1$

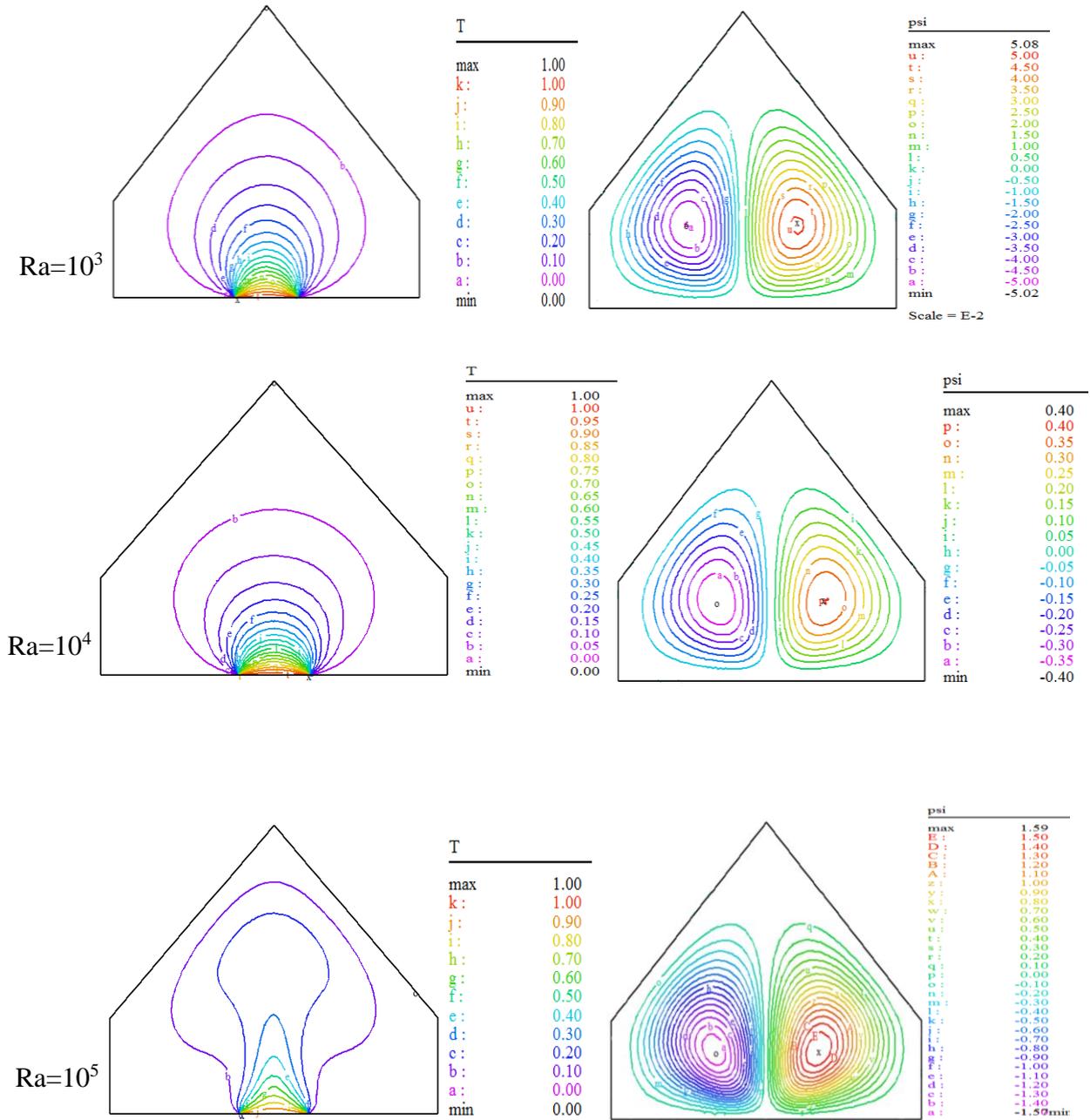


Fig.(5): Streamlines (right) isotherms (left) for  $s/L=0.5$

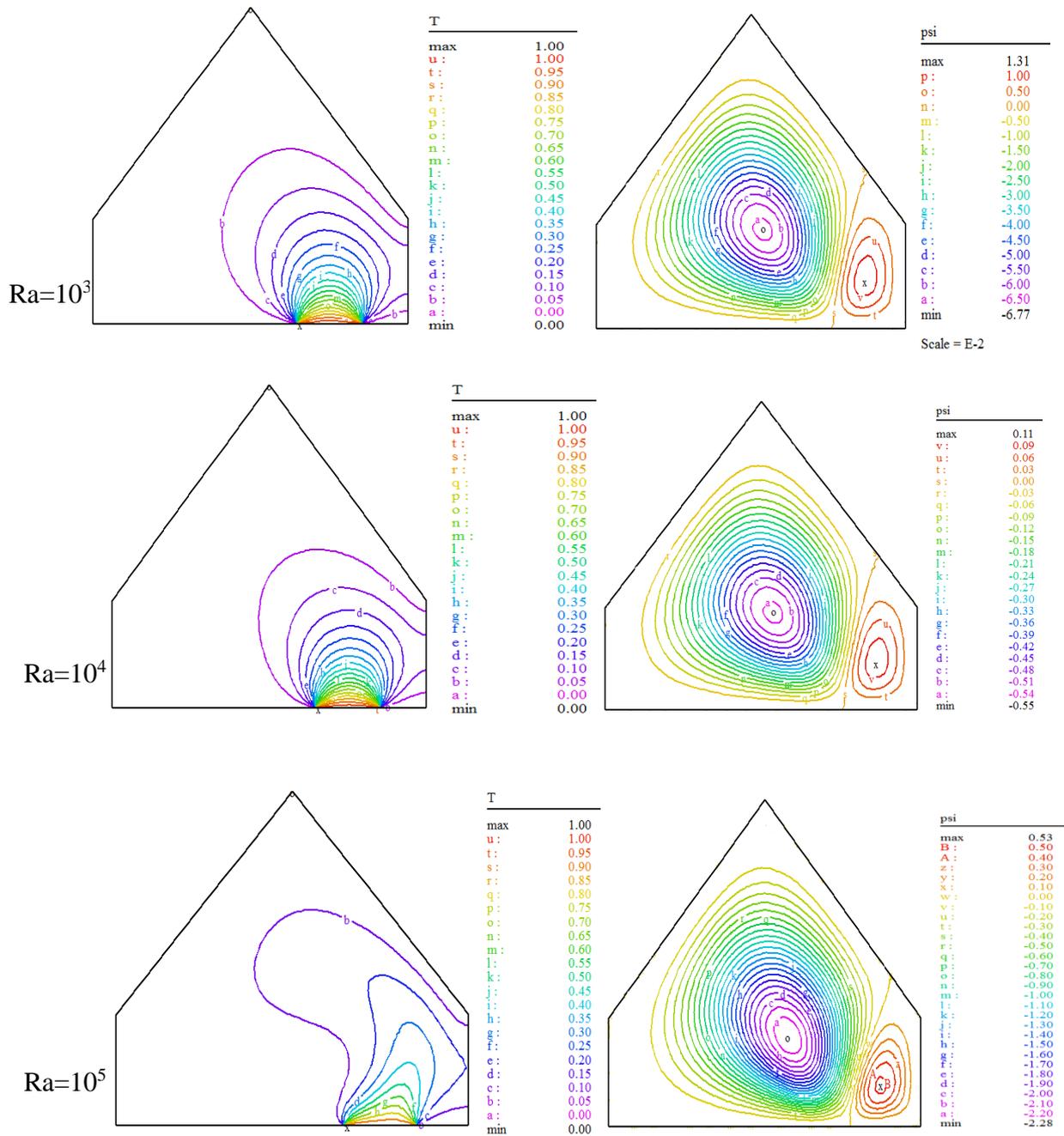


Fig.(6): Streamlines (right) isotherms (left) for  $s/L=0.7$

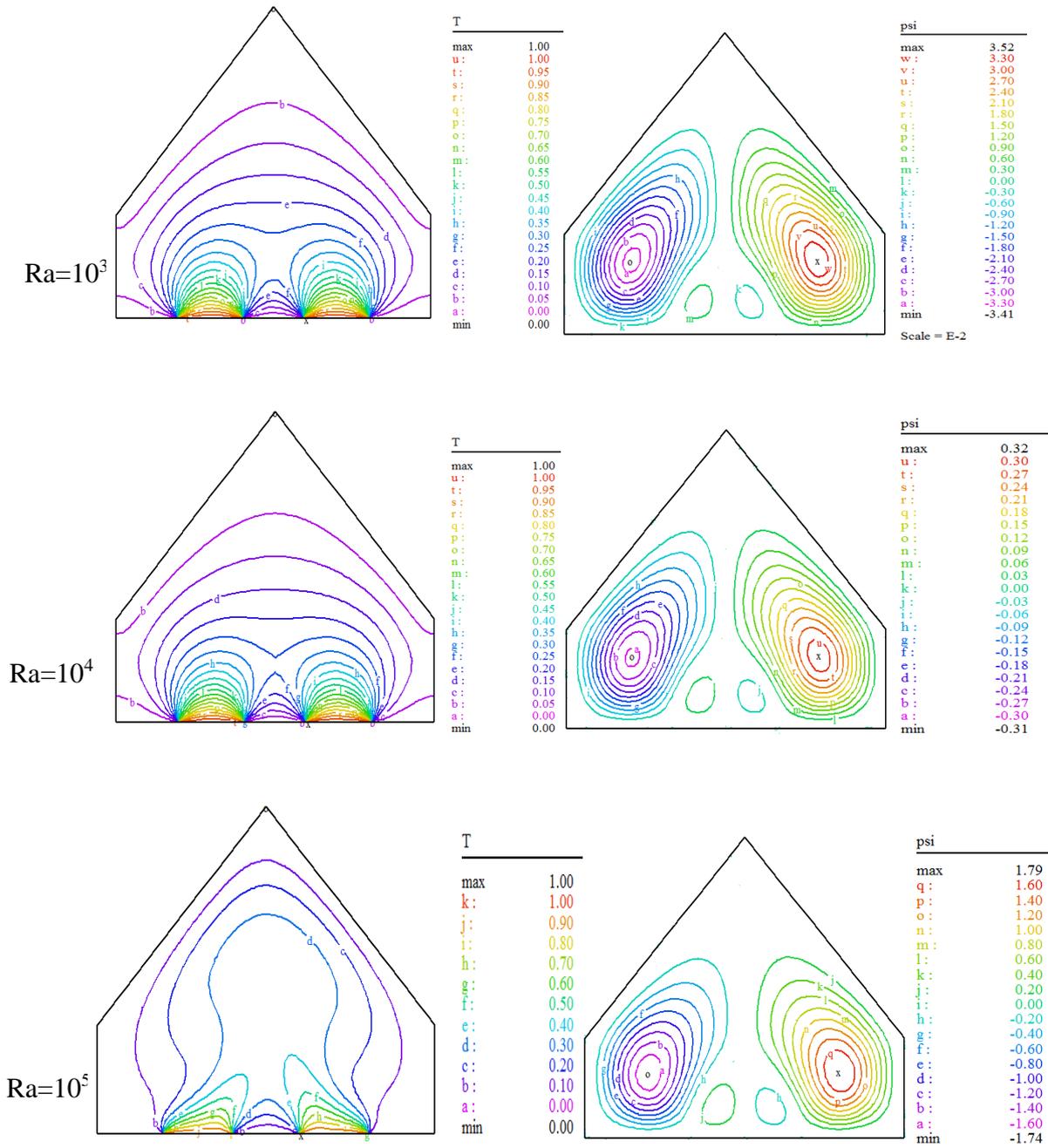


Fig.(7): Streamlines (right) isotherms (left) for two strip heaters

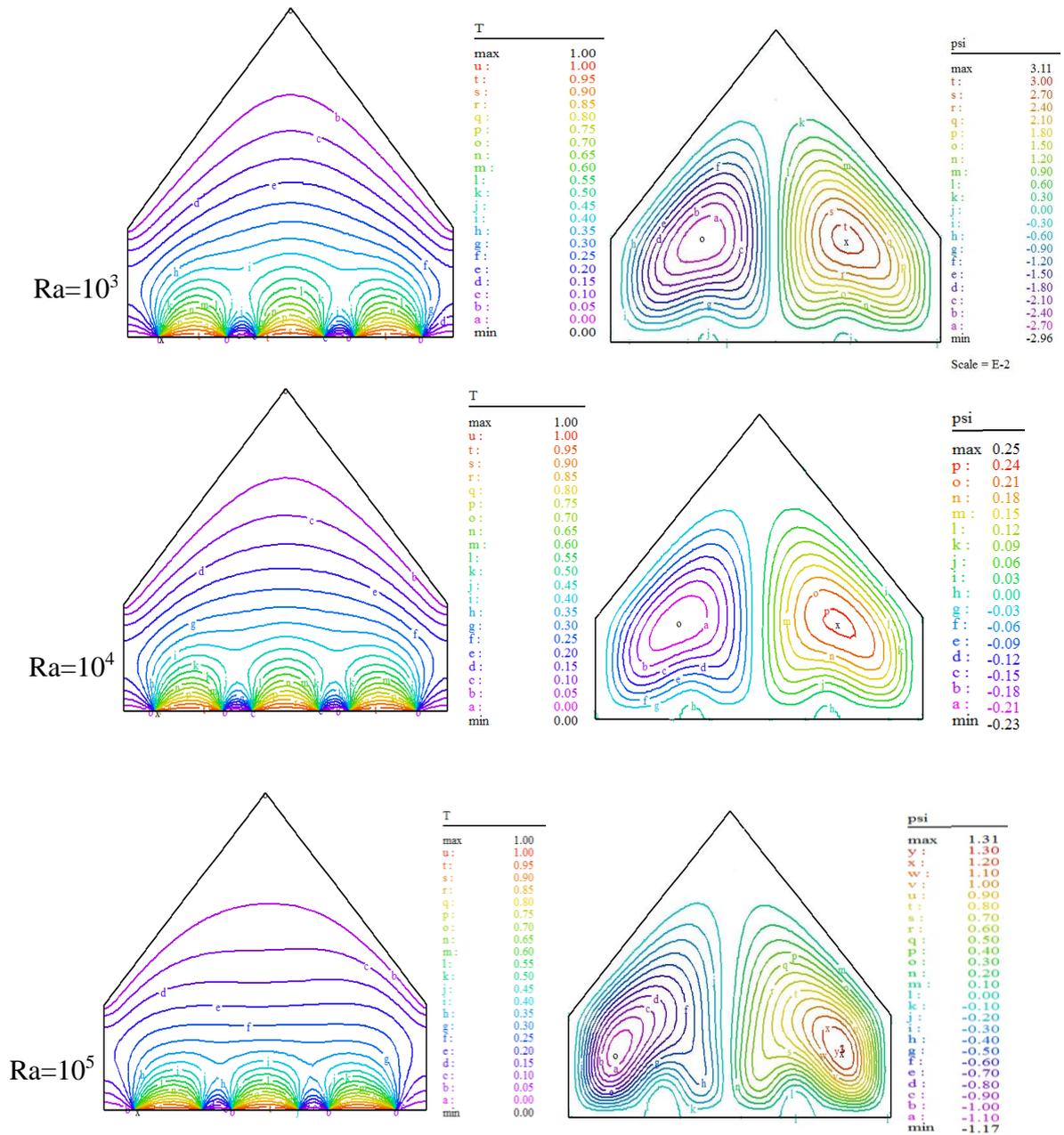
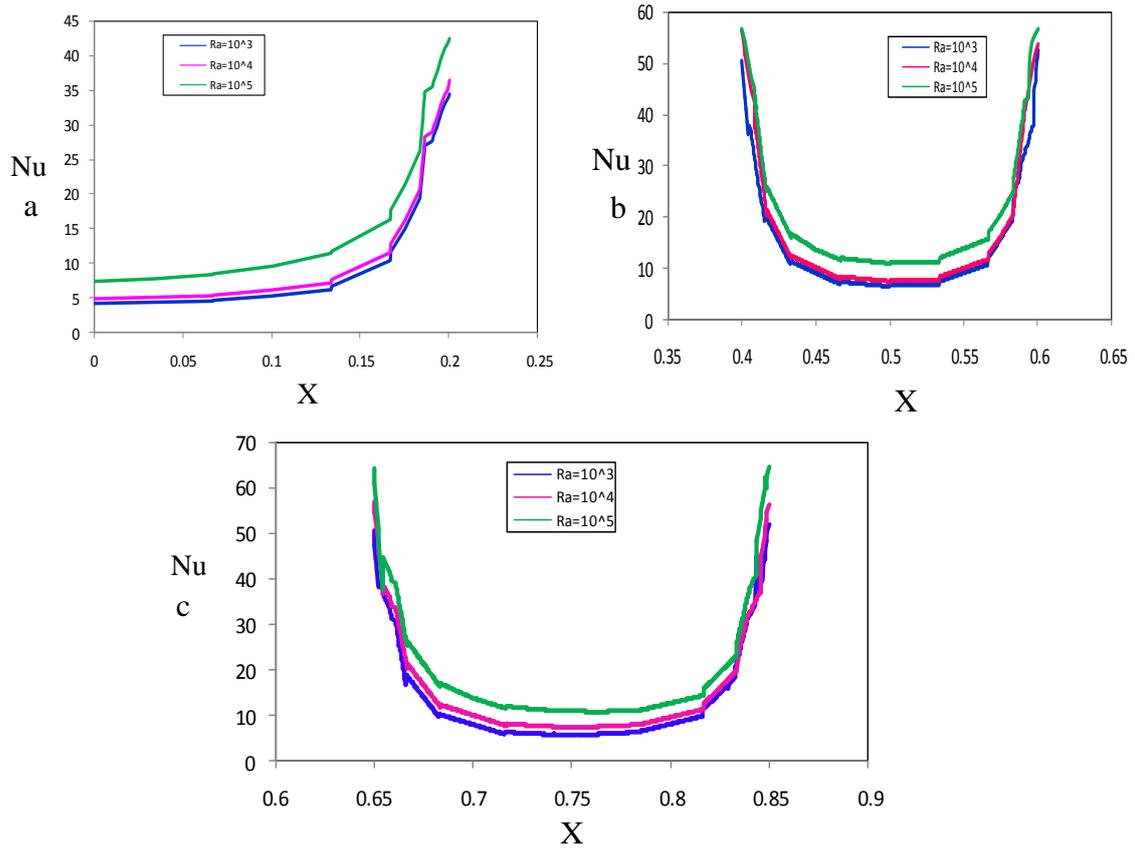
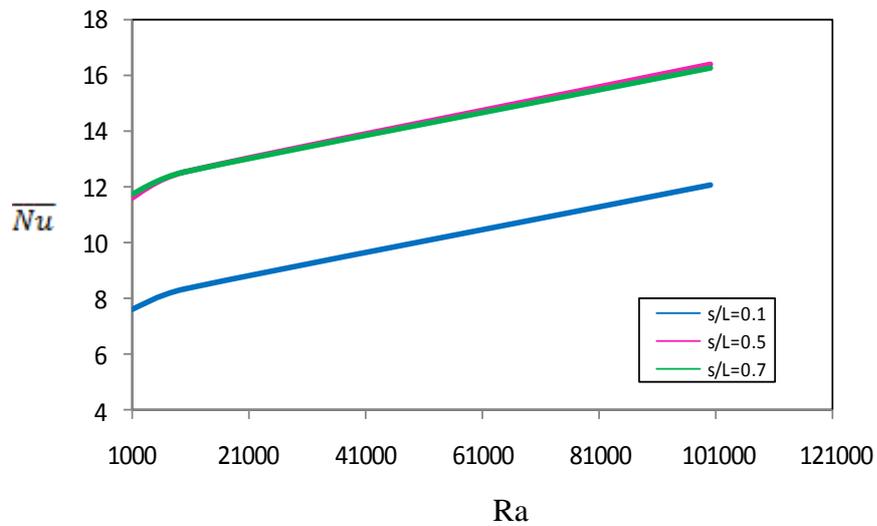


Fig.(8):Streamlines (right) isotherms (left) for three strip heaters



**Fig.(9):** Variation of  $Nu_L$  at different heater location for different  $Ra$  number



**Fig.(10):** Average  $Nu$  dependency on  $Ra$  number for one heater at different locations

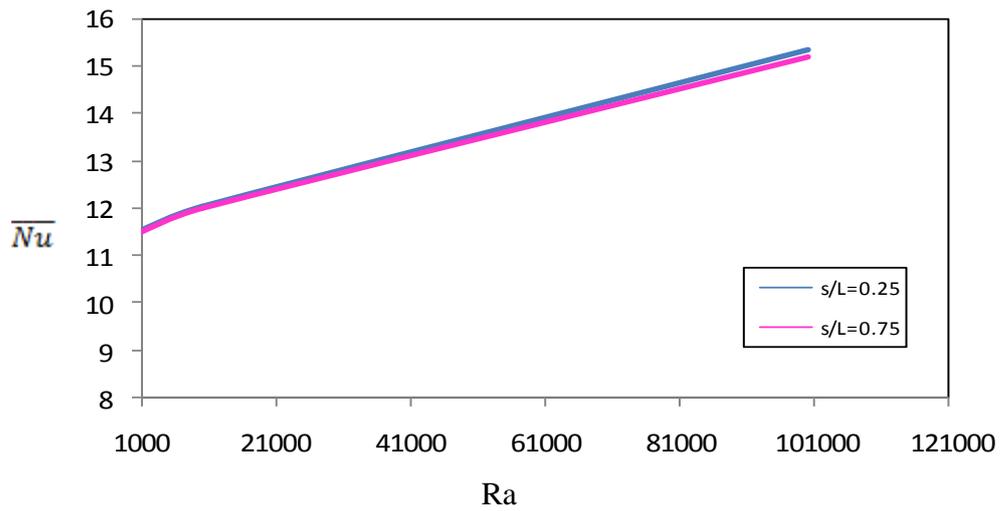


Fig.(11): Variation of mean Nu as a function of Ra number (Two heaters)

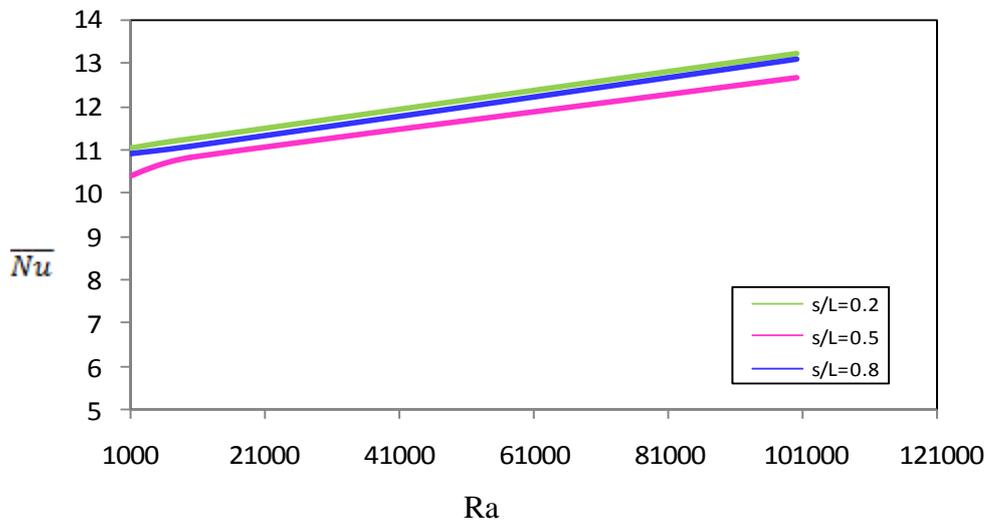


Fig.(12): Average Nu dependency on Ra for three heaters

دراسة عددية لانتقال الحرارة بالحمل الحر في حيز على شكل موشور  
مع مسخنات حرارية مثبتة على جداره السفلي باستخدام حقيبة برمجية تعمل  
بطريقة العناصر المحددة

سناء جعفر ياسين

مدرس مساعد

قسم الهندسة الميكانيكية- كلية الهندسة- جامعة البصرة

الخلاصة

اجريت دراسة وتحليل عددي لعملية انتقال الحرارة بالحمل الحر ثنائي البعد في حيز على شكل موشور مملوء بالهواء. للشكل الموشور الجدران العليا المائلة اعتبرت في درجة حرارة واطئة ثابتة والجدران العموديان معزولان حرارياً والمسخنات في درجة حرارة مرتفعة ثابتة وضعت على الجدار السفلي بينما باقي الجدار في درجة حرارة واطئة معروفة. منظومة المعادلات اللاخطية ثنائية البعد الحاكمة لعملية لحفظ الكتلة، الزخم والطاقة حلت باستعمال حقيبة برمجية تعمل بطريقة العناصر المحددة (FLEXPDE.5).

في هذه الدراسة يتراوح رقم رايلي (Ra (Rayleigh بين  $10^3$  إلى  $10^5$  واعتبر رقم برانتل ثابتاً (0.7) حيث تمت دراسة تأثير كلاً من موقع المسخنات وزيادة رقم Ra لعدة مواقع من الصفائح الساخنة. النتائج مُثلت بواسطة خطوط الجريان (streamlines) و خطوط التحوار (isotherms) ورقم نسلت (Nusselt number). قورنت النتائج المستحصلة لنفس الظروف مع ما هو منشور في المصدر [7] وأظهرت توافق جيد. الكلمات : الحمل الحر , فجوه موشورية و مربعة , درجة حرارة الجدار منتظمة.