Natural Convection in Equilateral Triangular Enclosure for Three Situations with Heating Element Applied at Different Positions on the Inclined Side Wall

Tahseen Ali Hussein Alhattab * Ali Safa Nouri Alsaegh **
Farooq Hassan Ali Alinnawi ***

College of Engineering-Mechanical Engineering Department
Babylon University, Babylon, Iraq.
*** E-mail: alhattab.t@gmail.com**
**** E-mail: ali_alsaeg@yahoo.co.uk**
***** E-mail: farooq_hassan77@ yahoo.com**

ABSTRUCT

Laminar natural convection heat transfer and fluid flow due to heating element inserted at different positions over the inclined side wall of equilateral triangle enclosure has been analyzed numerically in this study. The enclosure has filled with air as a working fluid with Pr=0.71. The walls of the enclosure are insulated except an element on one side where the heat flux are applied. The value of heat flux (q'') are 10, 30, 70 and 100 Watt with corresponding Rayleigh number of 10^5, 10^6, 10^7, 10^8 respectively. The length of the heating element was taken as 10% of the triangular enclosure side wall length where the length of the wall equals to 10 cm. The governing mass, momentum and energy equations are applied to the enclosure and solved by finite element using commercial code (ANSYS 5.4) based on TDMA method. Three different situations (normal, side and inverse) of the triangular enclosure with different positions of heating element are examined in this study to obtain the maximum stream function. The results show that the maximum value of stream function and high rate of heat transfer occur for normal situation of the enclosure and when the heating element is applied at position2 (Po.2 is a distance over the inclined side wall which takes a position limited between 0.2L to 0.3L from the edge of the triangle enclosure).

Keywords: **Laminar Flow, Natural convection , Equilateral Triangle, Finite Element, Stream Function.**

الحمل الحر في حيز مثلث متساوي الأضلاع لثلاث أوضاع بوجود شريحة حرارية مسلطة على مواضع مختلفة من ضلعه المائل

تحسين علي حسين الحطاب علي صفاء نوري الصائغ
فاروق حسن علي العناوي

قسم الهندسة الميكانيكية – كلية الهندسة – جامعة بابل – بابل – العراق

الخلاصة:

في هذه الدراسة،اجري التحليل العددي لانتقال الحرارة الطباقي وجريان المائع بوجود شريحة حرارية مسلطة على مواضع مختلفة من الضلع الجانبي المائل لحيز مثلث متساوي الأضلاع. الحيز مليء بالهواء كمائع تشغيل برقم برانتل =٠.٧١ . كانت جدران الحيز معزولة ماعدا موقع الشريحة حيث يسلط الفيض الحراري. إن قيمة الفيض

الحراري المسلط كانت ١٠، ٣٠، ٧٠ و ١٠٠ واط والمناظرة لرقم رالي ١٠^{٥}، ١٠^{٦}، ١٠^{٧} و١٠^{٨}على التوالي. إن طول الشريحة الحرارية المسلطة تساوي ١٠% من طول ضلع الحيز المثلث حيث أن طول كل ضلع يساوي ١٠ سم. لحل المعادلات الحاكمة تم استخدم برنامج جاهز (ANSYS 5.4) والذي يعتمد طريقة الشرائح المحددة ويستند إلى طريقة TDMA. درست ثلاث أوضاع مختلفة للحيز المثلث (العادي، الجانبي و المعكوس) مع تغيير مواضع الشريحة الحرارية على جدار الحيز لإيجاد أعلى قيمة لدالة الجريان. أظهرت النتائج أن أعلى قيمة لدالة الجريان و أعلى معدل لانتقال الحرارة يحدث عندما يكون الحيز المثلث في الوضع العادي والشريحة الحرارية في الموقع الثاني (Po.2) وهو الموقع الذي يقع على طول الضلع المائل للمثلث ويتحدد بالمسافة التي تتراوح بين 0.2L إلى 0.3L من حافة الحيز المثلث.

1. INTRODUCTION:

The phenomenon of natural convection heat transfer plays an important role, both in nature and in engineering systems. Many investigations have been performed for cavities both theoretically and experimentally for a wide range of Gr (Grashof number). Natural convection in an air filled closed space has been studied extensively in recent years in response to energy-related applications, such as thermal insulation of building, solar energy collector, furnaces and fire control in buildings, differential heating and cooling in lakes, pollutants diffusion in sea and natural convection in reservoir sidearm etc. Natural convection heat transfer mostly analyzed in differentially heated square or rectangular enclosures due to simplicity of numerical solution, in order to obtain temperature and flow field. However, the number of studies about partially heated enclosure is very limited and they investigated mostly square cross sectional geometries[1]. A few studies on flow and thermal investigation in triangular enclosure were carried for various applications. Hajri *et al.*[2] studied the natural convection and mass transfer in an equilateral triangular cavity. The buoyancy ratio and the Lewis number value have a profound influence on the thermal, concentration and dynamic fields. The numerical results for steady and laminar two-dimensional convection show that for small values of this parameter, there is little increase in the heat and mass transfer over that due to conduction. For the higher values, the convection mode dominates. Many researchers investigated natural convection in triangular cross-sectional roofs using different numerical techniques. Yasin *et al.*[3], Yasin *et al.*[4], Ahmet *et al.*[5], Ahmet *et al.*[6] and Hakan *et al.*[7] studied numerically the laminar natural convection inside a different shape of roofs, saltbox roof, gable (triangular) and gambrel roofs. They are studied the flow fields and isothermal lines inside the roofs, The results show that a single cell was formed for gambrel and gable (triangular) roofs with opposite direction but double circulation cell was formed for saltbox roof. Yadan Maa *et al.*[8] analyzed numerical simulation of natural convection in a triangular enclosure induced by solar radiation. The results show the variation of the dominant mode of heat and the flow status with the horizontal position along the wedge. In the near tip region, conduction dominated, and the thermal layer indistinct. In the middle region, convection dominated and the thermal boundary layer was distinct and stable. In deep region, the thermal boundary layer was unstable, and heat transfer was dominated by unstable convection. Ahmet Koca *et al.*[9] studied the natural convection heat transfer and fluid flow numerically to obtain effects of Prandtl number in a triangular enclosure which was partially heated from the bottom wall. Temperature of inclined wall is lower than the temperature of the heater while remaining walls are adiabatic. The size and location of the heater, Rayleigh number and Prandtl number were taken as governing parameters on heat transfer and fluid flow. The results show that the heat transfer increase when heater length, Prandtl number, Rayleigh number increase and when heater was located near the right corner. Yasin Varol *et al.*[10] studied numerically the natural convection heat transfer from a protruding heater located in triangular enclosure. Temperature of inclined wall of triangle was lower than the temperature of the heater, while the remaining wall were insulated. Various Rayleigh number, height of heater, heater width, heater location, aspect ratio of the triangular enclosure were examined. Goutam Saha *et al.*[11] solved numerically, two dimensional, steady, natural convection flow in a tilted isosceles triangular enclosure, partially subjected to constant heat flux at the bottom while the upper wall are cooled at constant temperature. They showed that average Nusselt number decreases as the heated strip enlarge, and increase along with inclination angle. Roy *et al.*[13] conducted a steady for natural convection flow in an isosceles triangular enclosure due to uniform and non-uniform bottom heating using penalty finite element analysis with biquadratic elements. They found that the average Nusselt number for the bottom wall was $\sqrt{2}$ times that of inclined wall. Tanmay *et al.*[14] investigated the laminar natural convection in an isosceles triangular enclosure with uniform and non-uniform heating at the side wall for various Rayleigh numbers (Ra) ($10^3 \leq Ra \leq 10^6$) and Prandtl numbers (Pr) ($0.026 \leq Pr \leq 100$). They found that at small Prandtl numbers, geometry does not have much influence on flow structure while at Pr=100, the stream function contours were nearly triangular showing that geometry has considerable effect on the flow pattern. Xu Xu *et al.*[15] carried out a numerical investigation of steady laminar natural convective heat transfer from a heated cylinder to its concentric air filled triangular enclosure. The effect of the Rayleigh number, aspect ratio, inclination angle, and cross-sectional geometry of the inner cylinder on the flow and heat transfer were systematically studied. The computed results indicated that at constant aspect ratio, both the inclination angle and cross-section geometry had insignificant effects on the overall heat transfer rates though the flow pattern were significantly modified.

The main objective of this work is studying the flow behavior and temperature distribution due to the natural convection inside triangular enclosure and finding the maximum strength of stream flow which is important in many applications such as electronics cooling, electricity generation and so on.

2. PHYSICAL DESCRIPTION AND GOVERNING EQUATIONS:

In this study, the model consists of equilateral triangular enclosure with side wall length (L) equal to (10cm) for three different situations of the enclosure (normal, side and inverse situations). The inclined side wall of triangular enclosure is divided to (10) of subdivisions each one represents (0.1) of triangle wall length (L) and then the heating element exposed over these subdivisions. The walls of the triangle are kept at adiabatic boundary conditions except at the heating element where constant heat flux (q'') are applied. From the mentioned details of model, 120 cases are examined to find the maximum stream function strength and the higher rate of heat transfer under the effect of heat element positions and triangular enclosure situations, but only 60 cases are presented in this paper which represents the most important results. Figure(1) indicates the schematic diagram for the used configuration and geometrical details. The two-dimensional governing equations, steady state, laminar incompressible buoyancy-induced flows with one phase are:

Continuity Equation:

$$\frac{\partial(\rho V_x)}{\partial x} + \frac{\partial(\rho V_y)}{\partial y} = 0 \quad \textbf{(1)}$$

Momentum Equation:

$$\frac{\partial\left(\rho V_x V_x\right)}{\partial \mathrm{x}} + \frac{\partial\left(\rho V_y V_x\right)}{\partial \mathrm{y}} = \rho g_x - \frac{\partial \mathrm{P}}{\partial \mathrm{x}} + \frac{\partial}{\partial \mathrm{x}}\left(\mu \frac{\partial V_x}{\partial x}\right) + \frac{\partial}{\partial y}\left(\mu \frac{\partial V_x}{\partial y}\right) \quad \textbf{(2)}$$

$$\frac{\partial\left(\rho V_x V_y\right)}{\partial x} + \frac{\partial\left(\rho V_y V_y\right)}{\partial y} = \rho g_y - \frac{\partial \mathrm{P}}{\partial \mathrm{y}} + \frac{\partial}{\partial \mathrm{x}}\left(\mu \frac{\partial V_y}{\partial x}\right) + \frac{\partial}{\partial y}\left(\mu \frac{\partial V_y}{\partial y}\right) \quad \textbf{(3)}$$

Energy Equation:

$$\frac{\partial}{\partial x}\left(\rho V_x C_p T\right)+\frac{\partial}{\partial y}\left(\rho V_y C_p T\right)=\frac{\partial}{\partial x}\left(k\frac{\partial T}{\partial x}\right)+\frac{\partial}{\partial y}\left(k\frac{\partial T}{\partial y}\right) \quad \textbf{(4)}$$

Where V_x and V_y are the velocity components in the x and y direction respectively, P is the pressure and T is the temperature. The stream function is computed for 2-D structure and is defined in form of its derivatives:

$$\frac{\partial \psi}{\partial X} = -\rho V_Y \quad \text{and} \quad \frac{\partial \psi}{\partial Y} = \rho V_X \quad \textbf{(5)}$$

In this study, the commercial code (ANSYS 5.4) was used to solve the above governing equations in dimensional form. The Rayleigh number (*Ra*) was computed by using the following relations:

$$Ra = Gr * \Pr \quad \textbf{(6)}$$

where

$$Gr = \frac{g\beta \Delta T L^3}{\nu^2} \;,\; \Delta T = T_h - T_c = \frac{q^{\backslash\backslash} L}{k} \;,\; \beta = \frac{1}{T_f} \;,\; T_f = \frac{T_h + T_c}{2} \quad \textbf{(7)}$$

The Boundary Conditions:

The velocity and thermal boundary conditions for the present study are specified as follows:

- Along the walls of triangular enclosure, no-slip velocity boundary condition are imposed, i.e.:

$V_X = V_Y = 0$ for all walls.

- All walls of triangular enclosure are adiabatic except the position of heat flux where the heat flux boundary condition are applied.

The Prandtl number (pr) is taken as 0.71 that corresponds to that of air ,so the effect of Prandtl number are not studied. The effect of Rayleigh number which controls the magnitude of the applied heat flux (q'') are examined in this study. The magnitude of the applied heat flux (q'') are 10, 30, 70 and 100 Watt corresponding to Rayleigh number (*Ra*) of 10^5, 10^6, 10^7, 10^8 respectively. The natural convection flow within this range of Rayleigh number ($10^5<Ra<10^8$) is inherently in laminar regime and thus justified, in general, the steady-state assumption.

In order to investigate the effect of various of parameters on the heat transfer, Nusselt number is selected as an indicator for the heat transfer rate. The local Nusselt number is defined as [12]:

$$Nu = \frac{q''L}{k(T_h - T_c)} \quad \textbf{(8)}$$

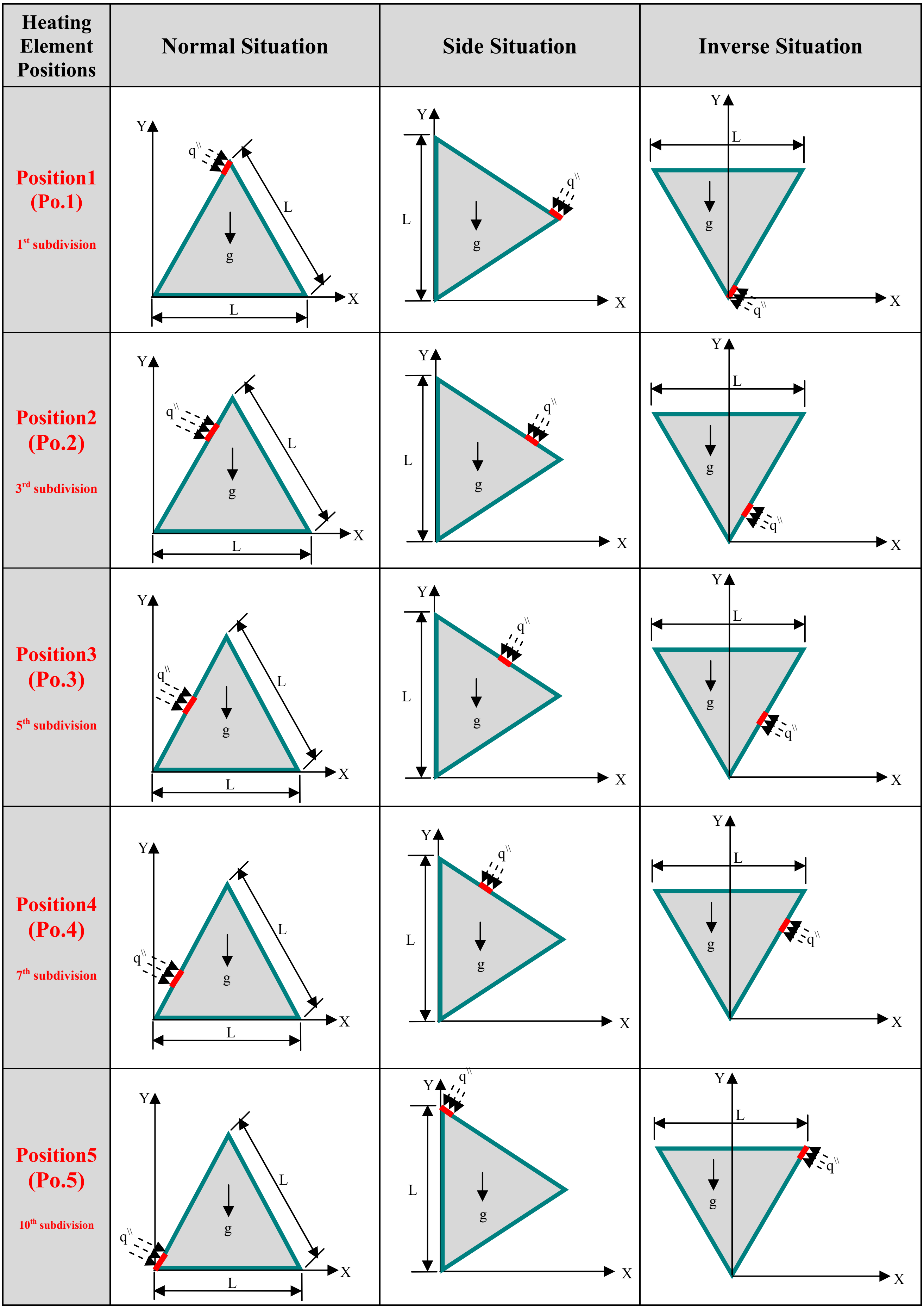

Fig.(1): Schematic diagram of the physical domain and boundary conditions.

3. NUMERICAL SOLUTION:

The grid system over the computational domain is created using unstructured quadratic element, which are unevenly distribution and concentrated near the three corner of the triangular enclosure where higher grid densities are desired as in Figure(2). It provides smooth solution at the interior domain including the corner region. Employing the finite element approach, the governing equations were iteratively solved with the convergence criterion of 10^{-4} for each variable. The set of governing equations is integrated over the domain using exponential interpolation in the mean flow direction inside the finite element. The (ANSYS 5.4) program using the Tri-Diagonal Matrix Algorithm (TDMA) to solve governing equations (continuity, momentum and energy Equations). ANSYS program solves the governing equations in dimensional form. The TDMA method is described in detail in Patanker [16]. The method consists of breaking the problem into a series of tri-diagonal problems where any entries outside the tri-diagonal portion are treated as source terms using the previous values. For a completely unstructured mesh, or an arbitrary numbered system, the method reduces to the Gauss-Seidle iterative method. The set of algebraic equation is solved using Successive Under Relaxation (SUR) technique and 0.1 is taken as under relaxation parameter. A grid independence test is applied to ensure the accuracy of the numerical results and to determine an appropriate grid density. Comparison of the maximum and minimum of stream function among a variety number of elements on the wall are listed in table1 and Figure(3).

In order to validate the numerical code, the results were compared with this obtained by Roy *et al.*[13]. Figure(4) illustrate the comparison of streamline and isothermal body contours of the present study with that obtained by Roy *et.,al.*,[12]. Very good agreement is found to be excellent which verification the commercial CFD code (ANSYS 5.4).

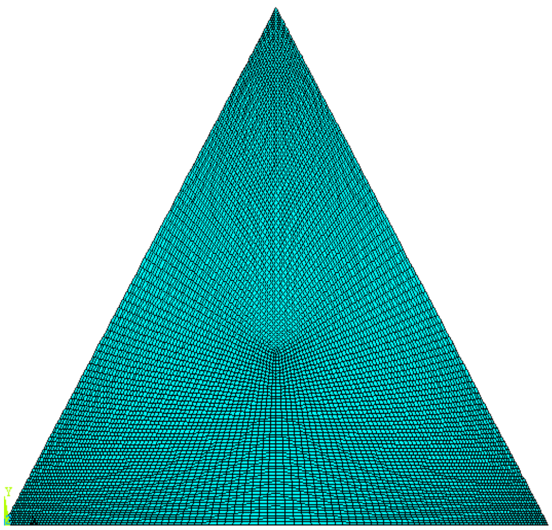

Fig.(2): A typical grid distribution with quadratic (110) elements.

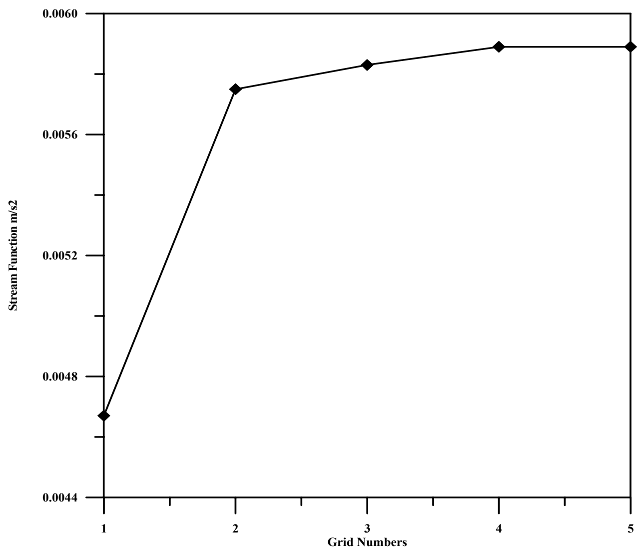

Fig.(3): Variation of Max. Stream Function with Number of elements.

Table 1
Variation of Max. and Min. Stream Function with Number of Elements

Grid No.	No. of elements on line	Ψ_{max}
1	**30**	**0.467e-03**
2	**60**	**0.575e-03**
3	**90**	**0.583e-03**
4	**100**	**0.589e-03**
5	**110**	**0.589e-03**

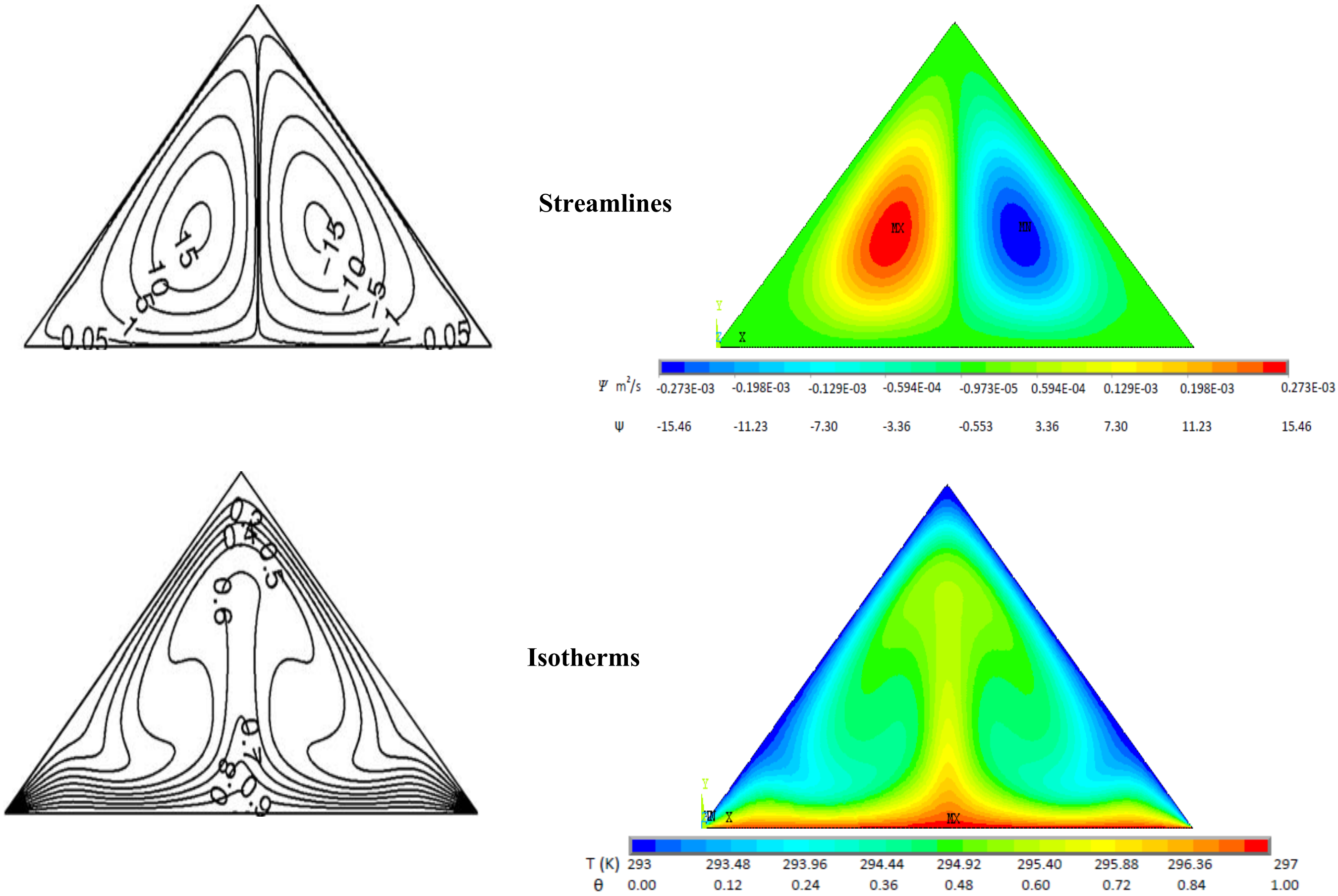

Fig.(4): Comparison of Streamlines and Isotherms for Verification with Results of Roy *et., al.,* [13] .

4. RESULTS AND DISCUSSION:

In order to investigate the effect of position, triangle situation and the amount of heat flux of the heating element on the flow and temperature fields and heat transfer characteristics of the typical enclosure, 120 cases are examined. As mentioned before, only 60 cases are presented in this work which are the most interested results.

4.1. Flow and Temperature Fields:

Stream function and isothermal body contours for the typical cases with heating element at five positions and for three situations(normal , side and inverse)are presented in Figures(5, 6, 7, 8 and 9). The plots are arranged going from left to right with the ascending of heat flux (10,30,70 and 100 Watt) and from top to bottom. The first three rows describes stream function (Part A) and the next three rows describe isothermal contours for different situations (Part B). The values of stream function and temperature for each case is also presented in each Figure.

4.1.1 Flow Fields (Part A)

Figure(5-Part A) represents the flow structures for the typical case with heating element at position1 (Po.1) for three situations. The first row of Figure(5-part A) represents the stream function for normal situation when the heating element at the top edge angle and for different heat fluxes value. When heat flux equals to 10 Watt, two small vortex appear in the enclosure with opposite direction since the conduction effect was dominating the heat transfer more than the convection and the heat flux is too weak to make a large vortex. At heat flux equal to 30, 70 and 100 Watt, the stream function body contour are similitude and consist of a main vortex having clock wise direction in the center of triangle and anti-clock wise direction near the walls. The second row of Figure(5-part A) represents the stream function body contour when heating element at the head angle of the triangular enclosure and for different heat fluxes in side situation. The flow consists of main vortex appears near the center of the triangle with clock wise direction. The value of absolute stream function increases with increasing the heat flux because the temperature difference increase and buoyancy force increase. Third row of Figure(5-part A) represents the stream function for the inverse situation. The flow contains two small cells with opposite direction of flow at two top angles. The maximum value of stream function (ψ) equals to ($0.66\ m^2/s$) and occurs when the triangular enclosure at normal situation and heat flux equals to 100Watt because the air near the heating element becomes light in weight and moves speedy to the bottom wall of the enclosure and replacing by cold air.

Figure(6-Part A) describes the stream function contour when the heating element at position2 (Po.2). The stream function for normal situation is more uniform and having higher value of stream function when it compared with the previous position (Po.1) of heating element. The value of stream function increases with the increasing in heat flux value. It is seen that the core of the enclosure has higher strength of stream function with anti-clock wise direction and this strength decreases near the walls of the enclosure because the velocity at the walls is equal to zero. For the case of side situation, the enclosure occupies by one main vortex and has longitudinal shape with clock wise direction. Same behaviors appear for all values of heat flux with unequal strength of stream function. For the case of inverse situation, the flow field having triple non-uniform shape vortex at three angles of enclosure. With the increasing in the value of heat flux, the top left vortex vanish.

Figure(7-Part A) shows the stream and isothermal body contours when the heating element at position3 (Po.3),where it located at the mid of the inclined wall. For normal situation, main vortex occupied the triangular enclosure and the center of vortex shift to the left bottom angle due to the position of heating element. It is also seen that the strength of stream function is lower than that when the heating element at position2 (Po.2). For the side situation flow consist of main vortex with clock wise direction at the center of the enclosure. For inverse situation, hot vortex becomes large in size and the cold vortex centered at lower angle since the temperature distribution becomes more spread in the triangular space. The values of absolute stream function for normal situation are lower than that of the previous cases (Po.1 & Po.2). Figures(8 & 9-Part A) show the contours when heating element at position4 (Po.4) and position5 (Po.5), respectively. For the normal situation, the main vortex becomes near the bottom wall and spread along it because the heating element located near the base wall. For the side situation, the main vortex becomes opposite in direction due to the vicinity of heating element to the top angle which pushing the hot air to flow in clockwise direction near the walls and with anti-clock wise at the center of enclosure. For inverse situation, the flow consists of main circulation cell appears at the center of triangular enclosure with anti-clock wise direction near the walls and clock wise direction at the center of enclosure because that the heating element at the right tip angle. Also it can be deduce that the strength of stream function of the last positions (Po.4) & (Po.5) are lower than that of the previous positions for all enclosure situations.

4.1.2 Temperature Fields (Part B)

Figures(5 to 9-Part B) illustrate the temperature contours for the typical cases with different heating element positions and for three enclosure situations.

Figure(5-Part B) presents the temperature distribution body contour for triangular enclosure when the heating element at position1 (Po.1) for three cases of enclosure situations (normal, side and inverse) and for different values of heat flux (10,30,70 and 100 Watt). For normal situation, the temperature distribution when (heat flux=10 Watt) differs from the other cases when (heat flux= 30, 70 and 100 Watt) where the heat spread at the left half of the triangular enclosure while for other cases, when (heat flux= 30, 70 and 100 Watt), the heat spread at the right half of the enclosure due to the effect of conduction and convection mode. For the side situation, the temperature distribution at low half of the enclosure and the area of temperature distribution increase with the increasing in heat flux. For inverse situation, the temperature distribution at low edge of triangular enclosure was very weak because that the conduction mod is dominant. Figure(6-Part B) describes the temperature distribution when heating element at position2 (Po.2). The heat spreading area is larger than that of the previous case (Po.1) for all values of heat flux and for all situations. Also, the heat spreading area increases with the increasing in the value of heat flux because the thermal boundary layer moves freely along the inclined wall of the enclosure.

Figure(7-Part B) represents the temperature contour for three different situation of the different values of heat fluxes when heating element at position3 (Po.3). In this case, the temperature distribution take a large space in triangular enclosure because the heating element lies at the mid of the inclined wall. Figures(8 & 9-Part B) show the temperature contour when the heating element at two positions (Po.4) and (Po.5). The positions of heating element make the spreading area of the temperature distribution less than that of the previous cases because the heating element located near the base wall of the triangular enclosure.

4.2 Heat Transfer field

In order to evaluate how the heat element positions and heat fluxes effects on the heat transfer rate for three different situations, the average Nusselt number for all cases as a function of element position and values of heat flux are presented in Figure(10). For all cases, the average Nusselt number increases with increasing in heat flux because the energy transport increases. Figure(10-a) represents the average Nusselt number for normal situation of triangular enclose while Figure(10-b) for side situation and Figure(10-c) for inverse situation. The maximum heat transfer rate occurs when the triangular enclosure at normal situation and when heating element at position2 (Po.2) because the natural convection becomes more predominant. The minimum Nusselt number appears when the heating element located at two positions (Po.5 & Po.1).

5. CONCLUSIONS

Heat transfer by natural convection in equilateral triangular enclosure with the presence of heating element at different location on the inclined side wall and for different enclosure situations of a triangle shape enclosure have been numerically studied and solved using commercial code (ANSYS 5.4). The TDMA method helps to obtain smooth solutions in term of stream functions and isothermal body contours for different heat fluxes (q"=10,30,70 and 100Watt) and Prandtle number is equivalent to that of air (Pr=0.71). In view of the obtained results, following conclusions can be drawn:

1- The maximum absolute and uniform stream function can be obtained when the triangle at normal situation and heating element at position2 (Po.2).
2- The positions of heat flux are one of the most important parameters on flow and temperature field.
3- At low heat flux, conduction mode is dominant while at high heat flux the advection mode (natural convection & conduction) is dominant.
4- Heat transfer is very week when the heating element at positions (Po.1) & (Po.5) for all different situations of the triangular enclosure when it is compared with other positions (Po.2, Po.3 & Po.4).

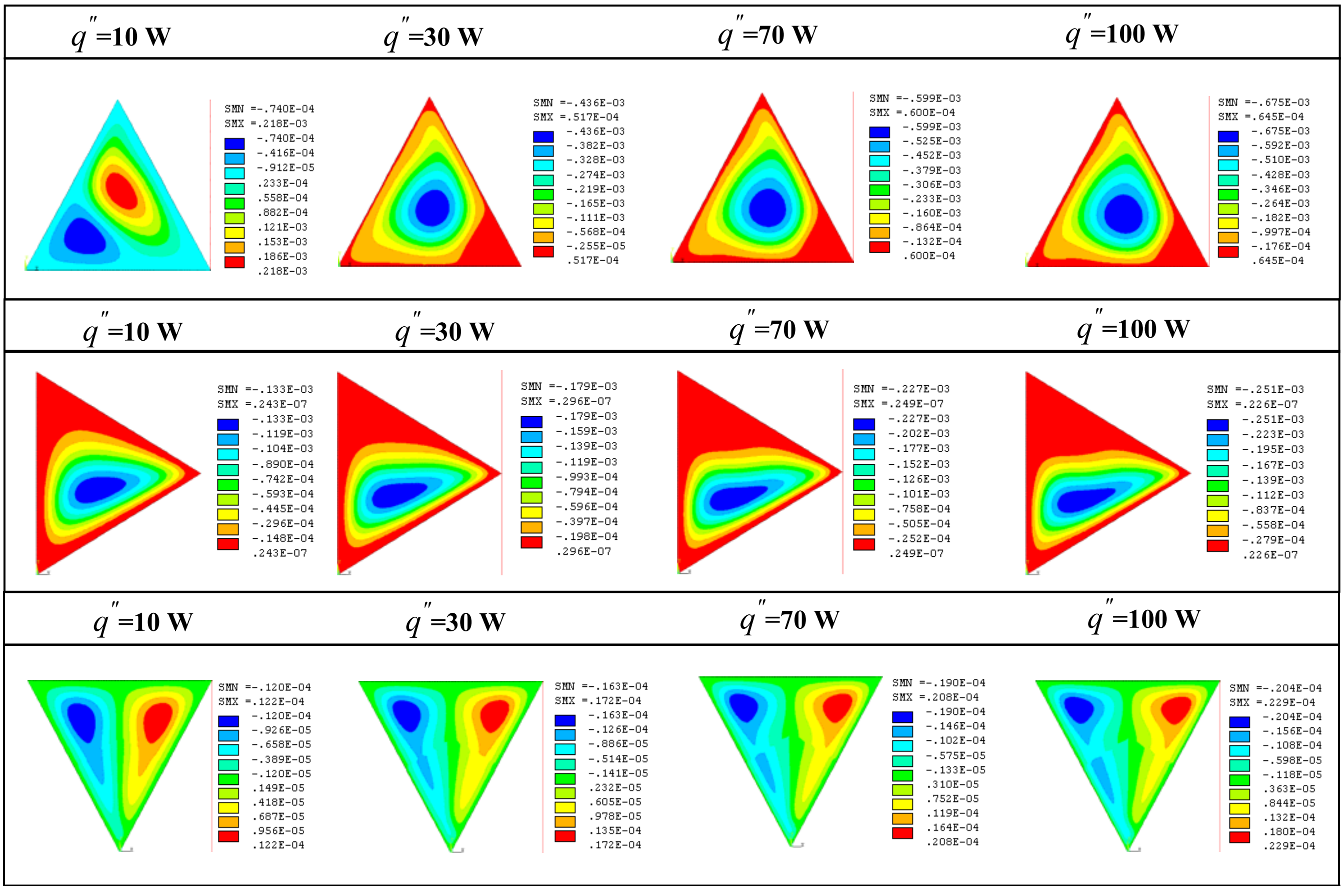

Fig.(5-PartA): Stream function Body Contour (m^2/s)

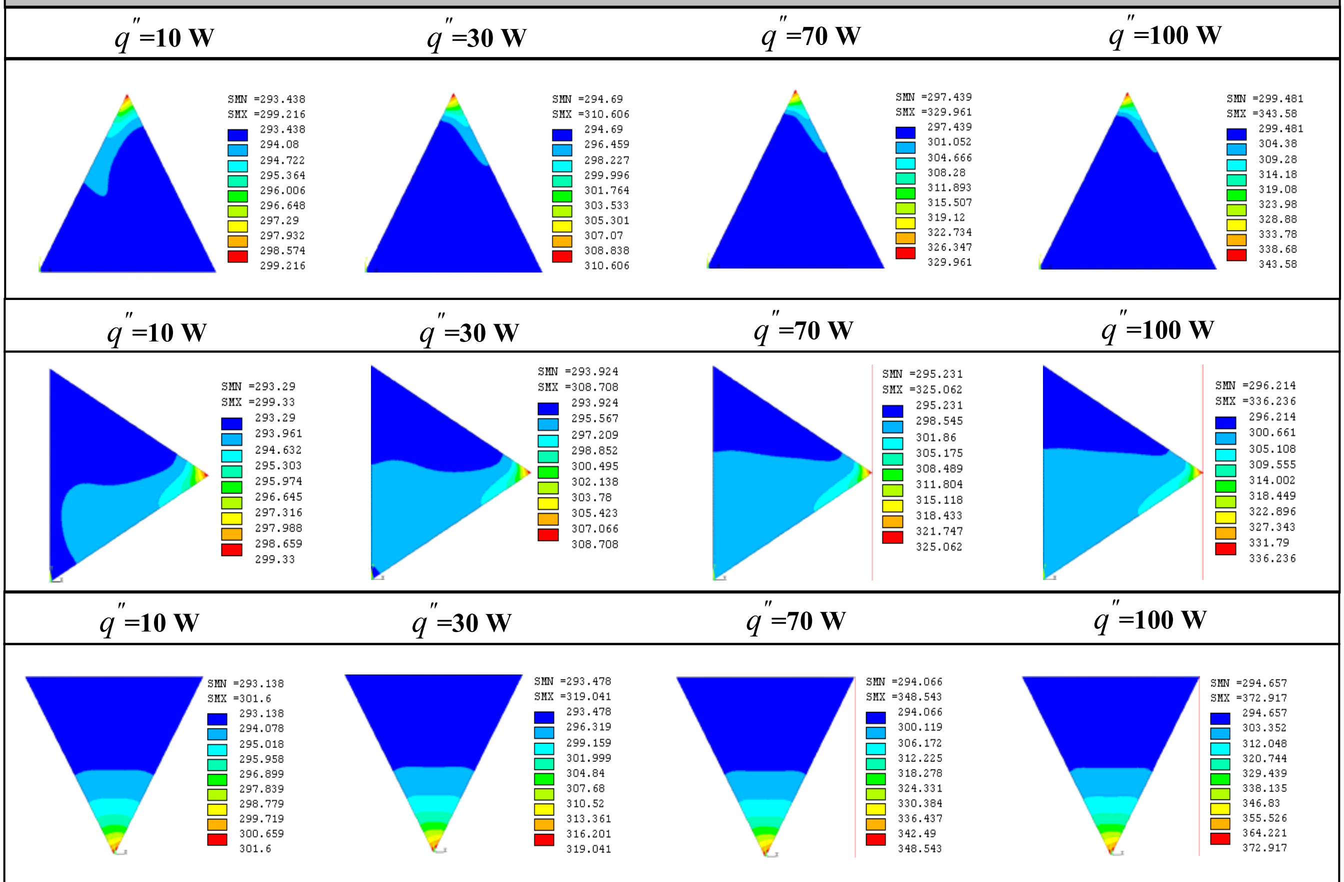

Fig.(5-PartB): Isothermal Body Contour (Kelvin)

Fig.(5): Stream and Isothermal Body Contour for Three Different Enclosure Situations and Different Heat Fluxes When Heating Element Located at Position1 (Po.1).

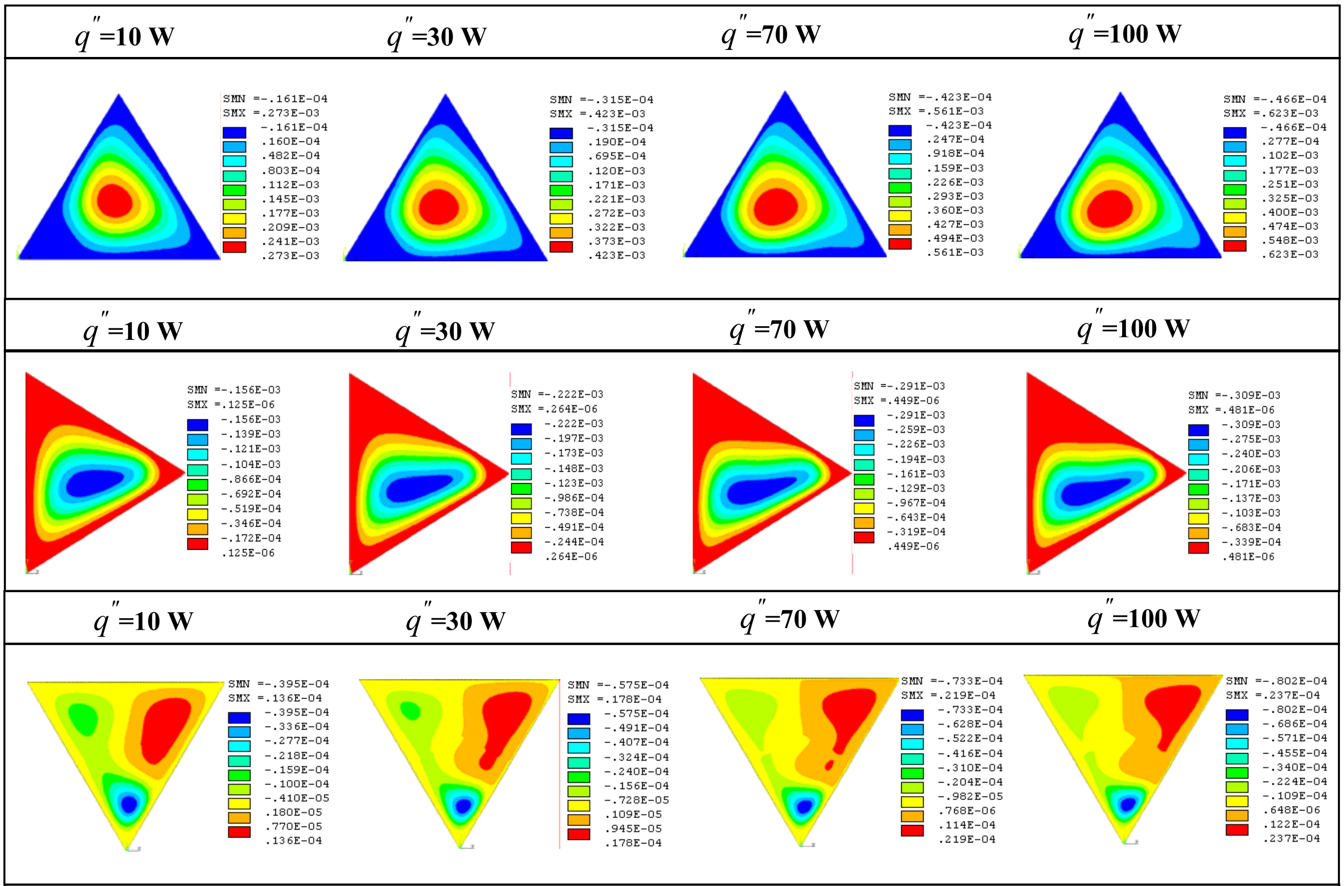

Fig.(6-PartA): Stream function Body Contour (m^2/s)

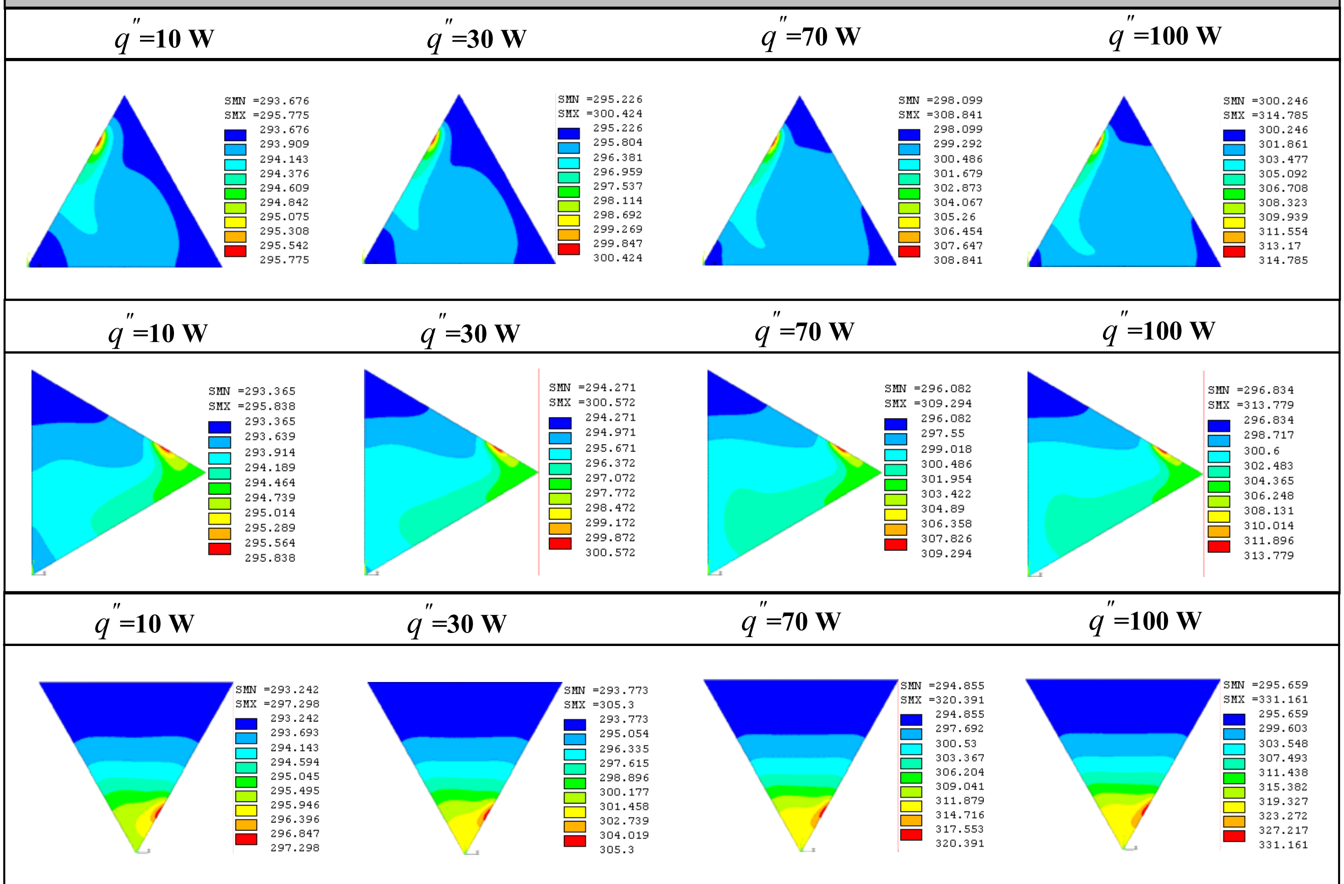

Fig.(6-PartB): Isothermal Body Contour (Kelvin)

Fig.(6): Stream and Isothermal Body Contour for Three Different Enclosure Situations and Different Heat Fluxes When Heating Element Located at Position2 (Po.2).

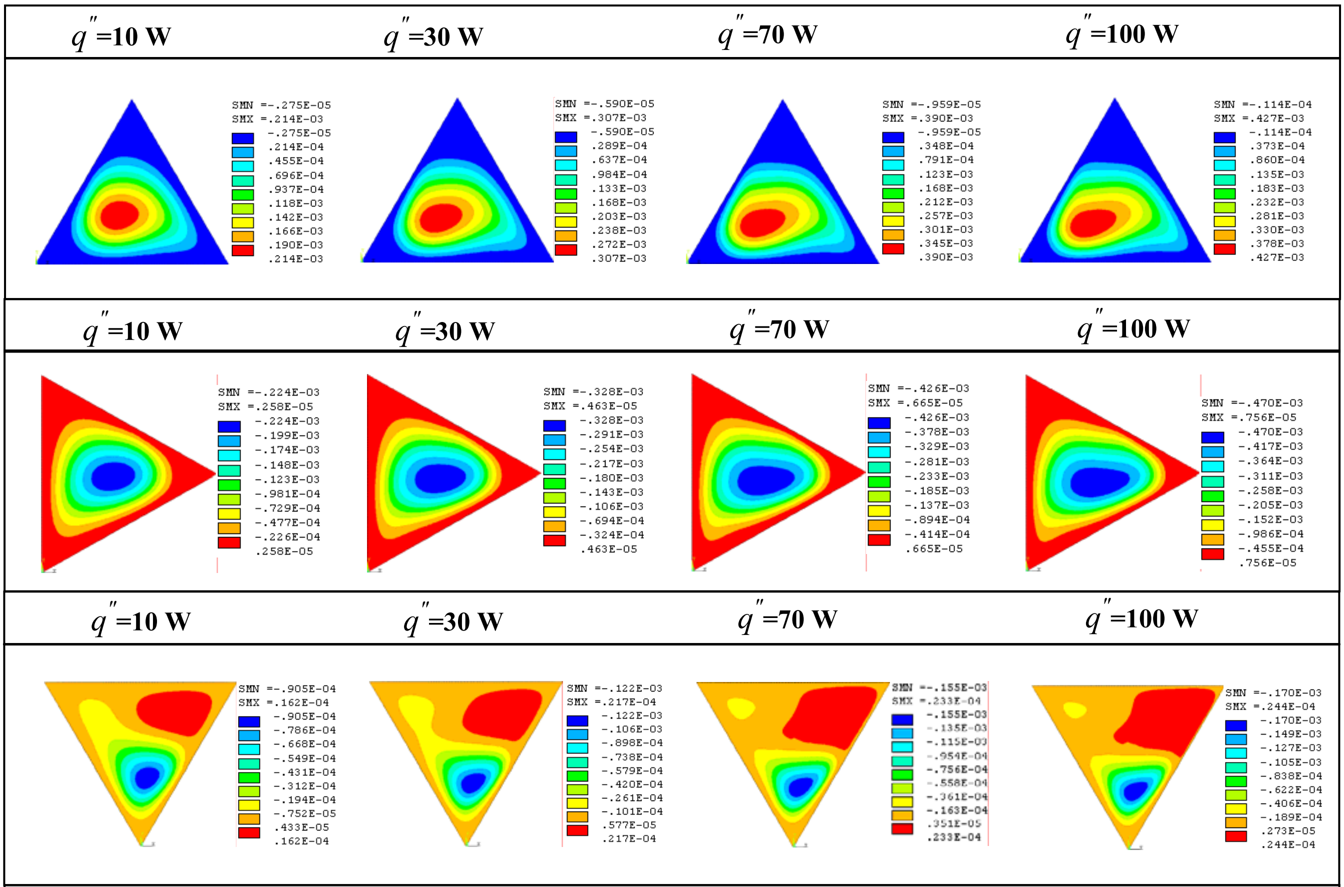

Fig.(7-PartA): Stream function Body Contour (m^2/s)

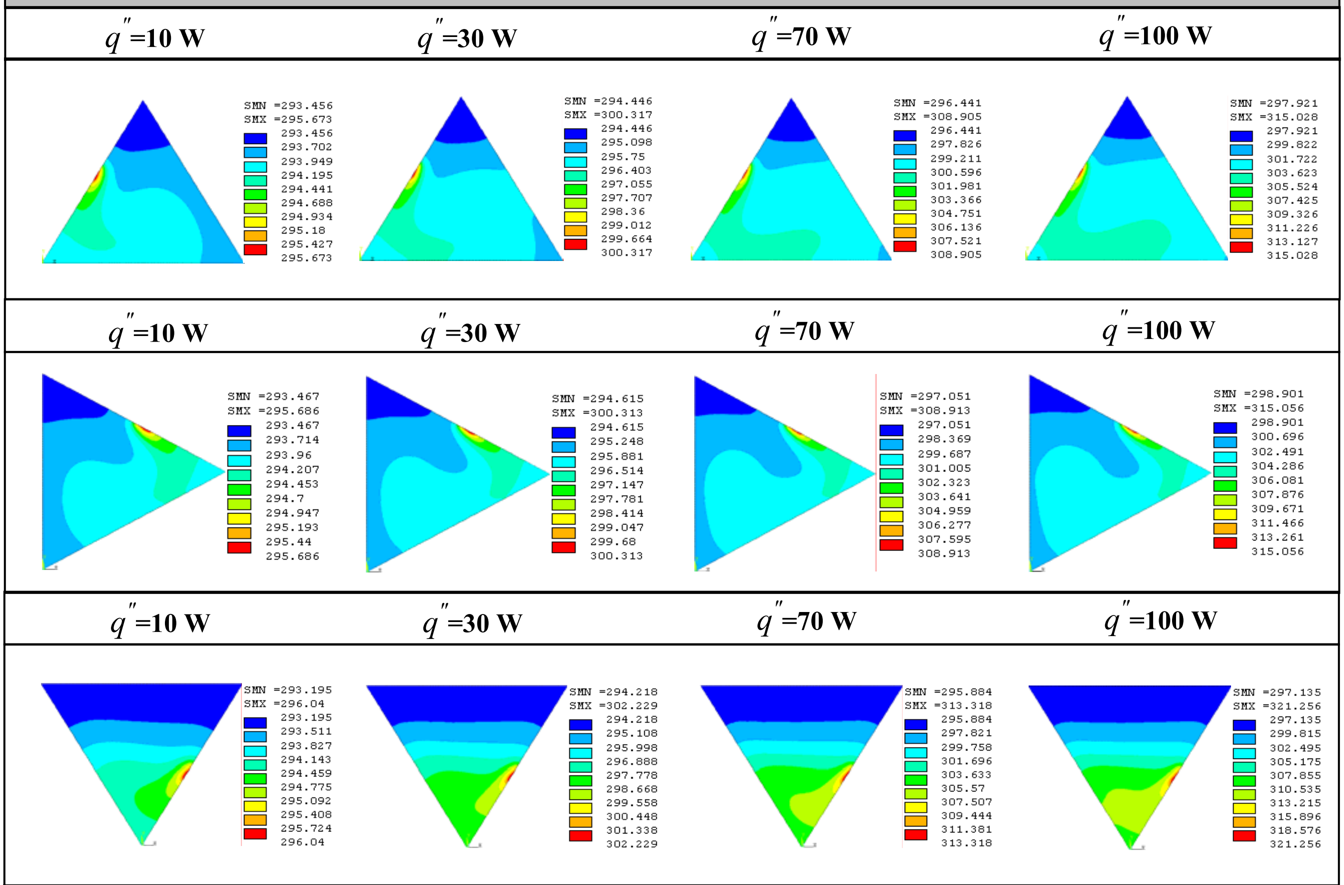

Fig.(7-PartB): Isothermal Body Contour (Kelvin)

Fig.(7): Stream and Isothermal Body Contour for Three Different Enclosure Situations and Different Heat Fluxes When Heating Element Located at Position3 (Po.3).

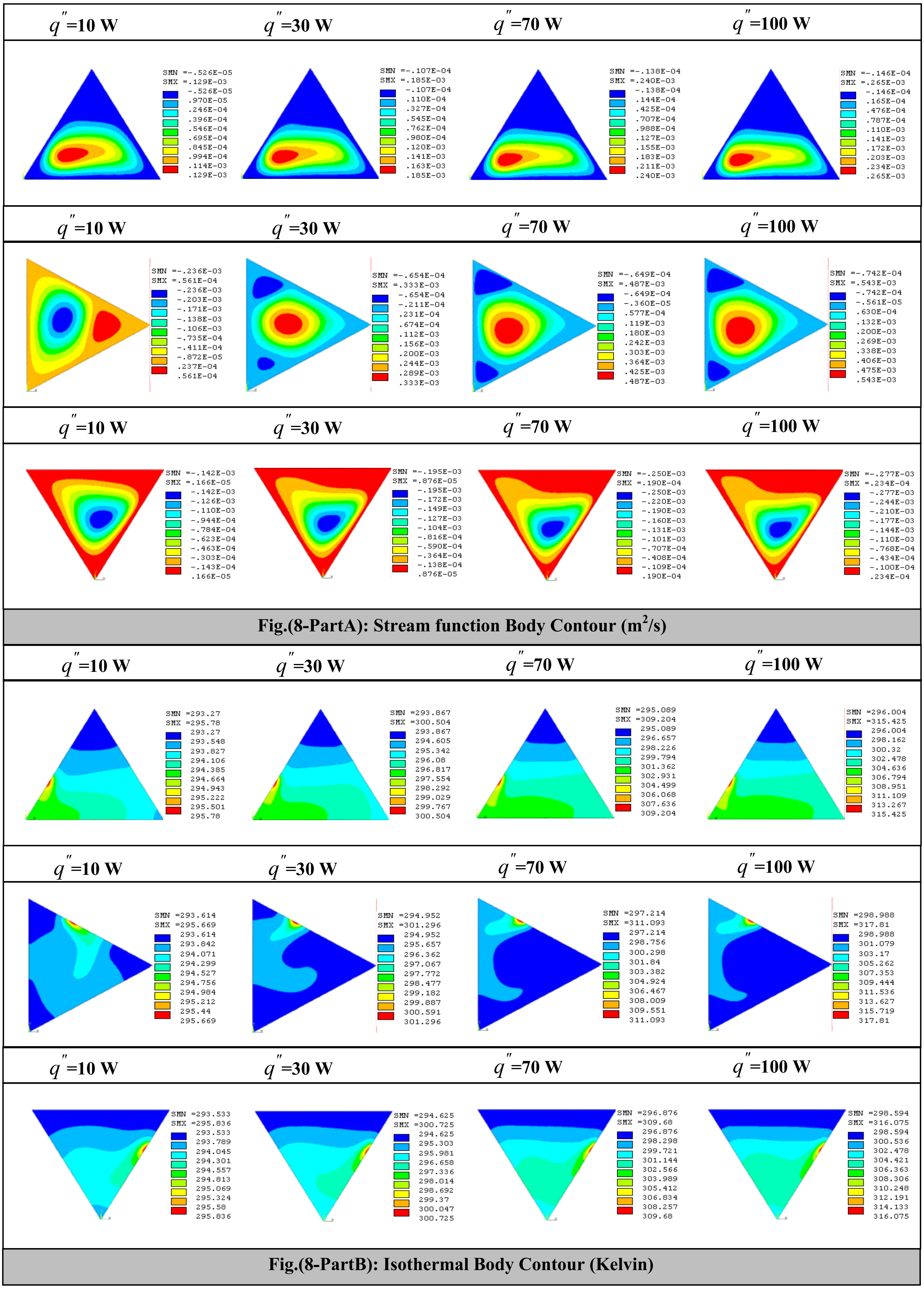

Fig.(8-PartA): Stream function Body Contour (m^2/s)

Fig.(8-PartB): Isothermal Body Contour (Kelvin)

Fig.(8): Stream and Isothermal Body Contour for Three Different Enclosure Situations and Different Heat Fluxes When Heating Element Located at Position4 (Po.4).

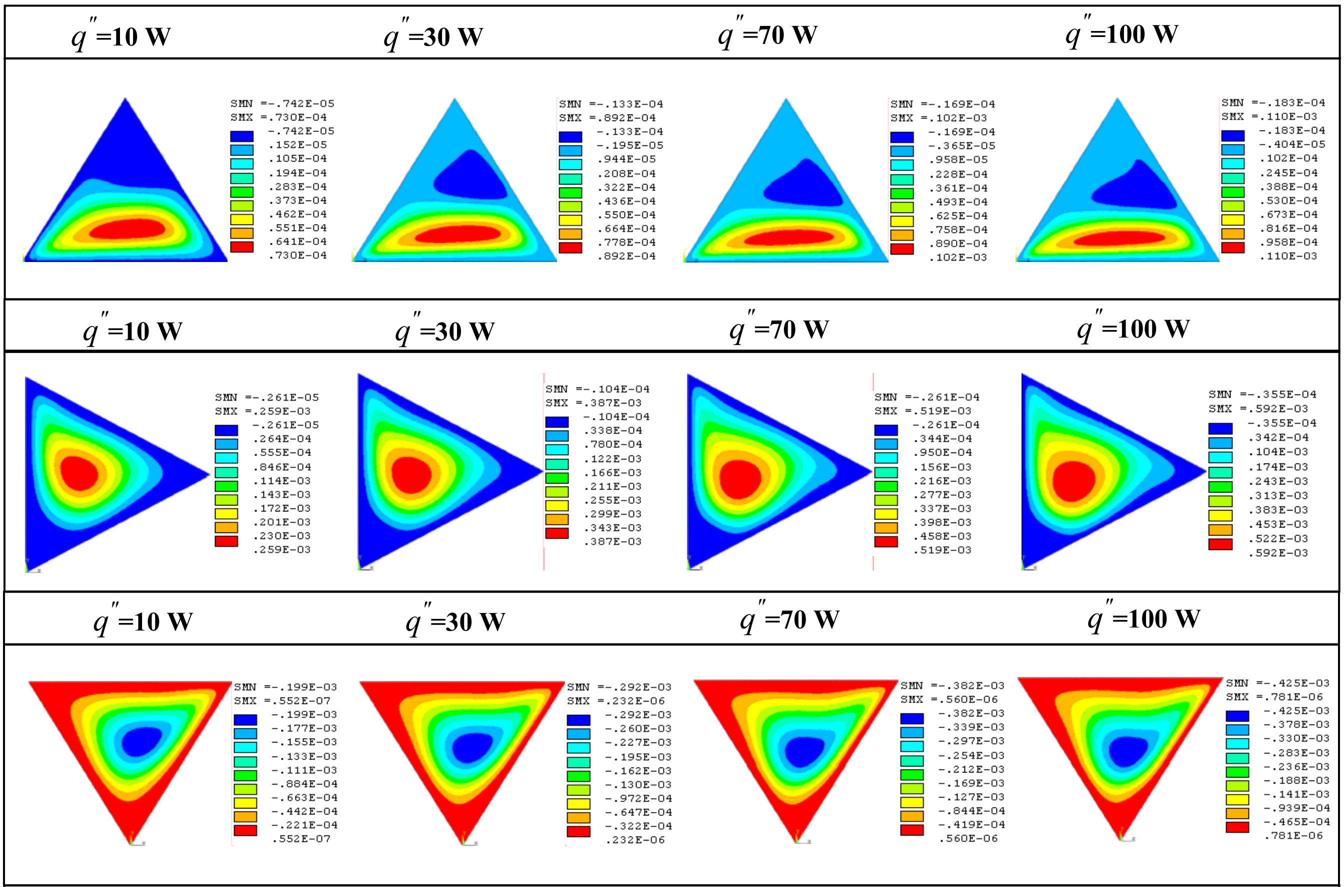

Fig.(9-PartA): Stream function Body Contour (m^2/s)

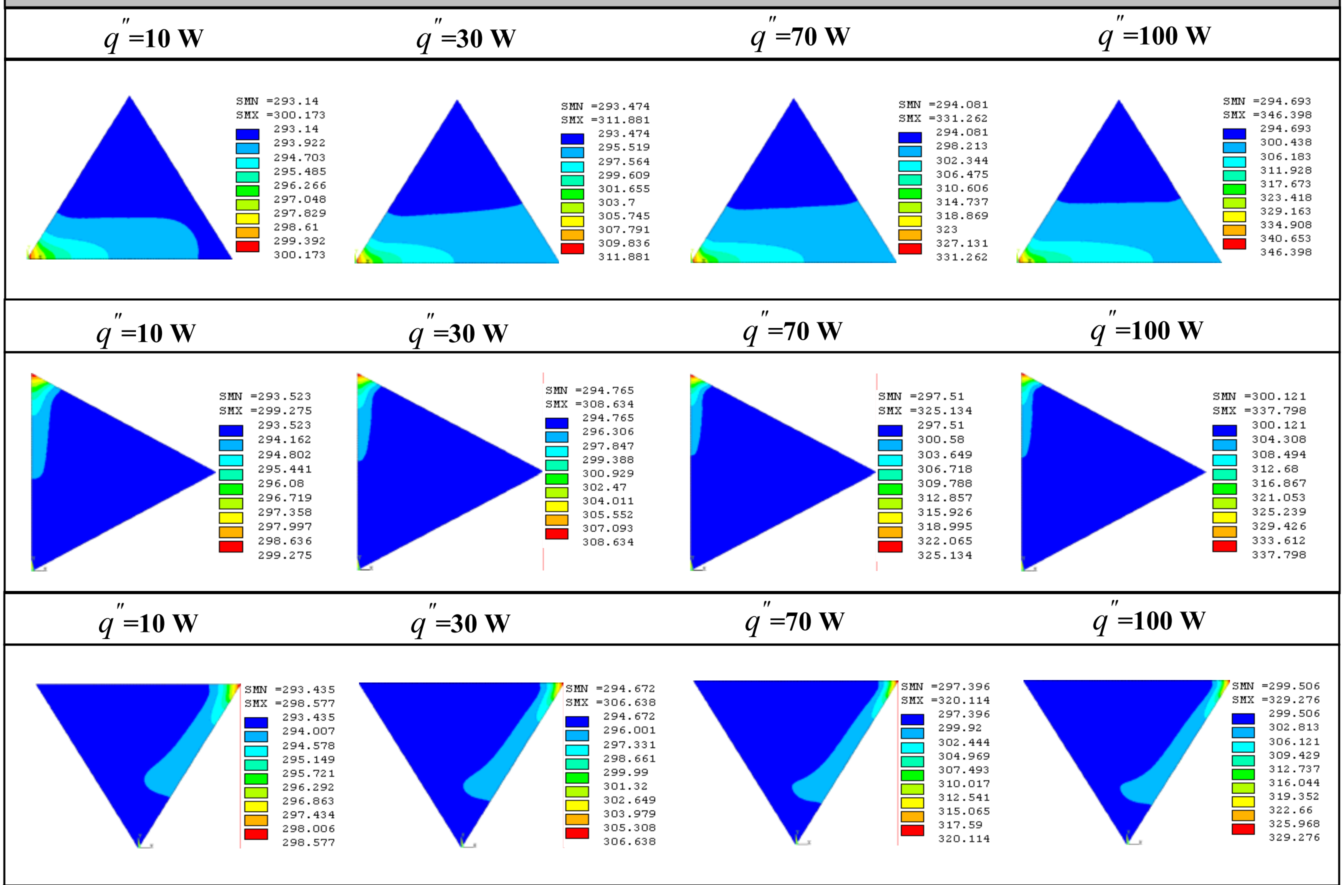

Fig.(9-PartB): Isothermal Body Contour (Kelvin)

Fig.(9): Stream and Isothermal Body Contour for Three Different Enclosure Situations and Different Heat Fluxes When Heating Element Located at Position5 (Po.5).

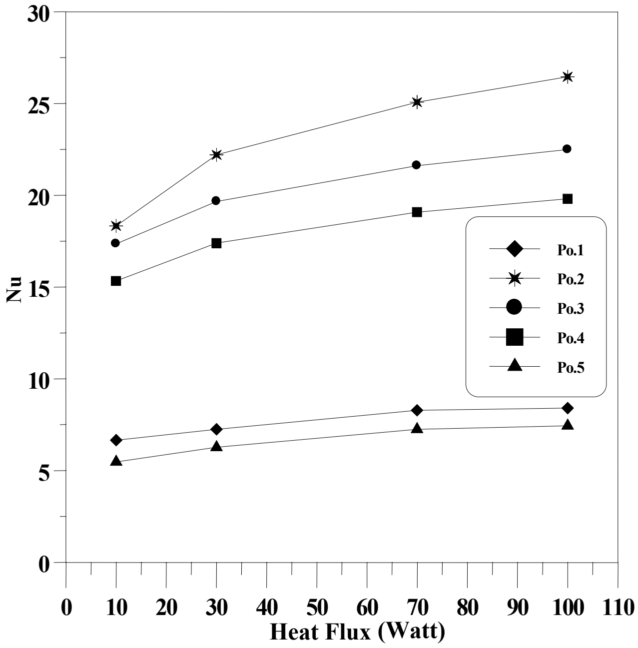

Fig.(10-a) Normal Situation.

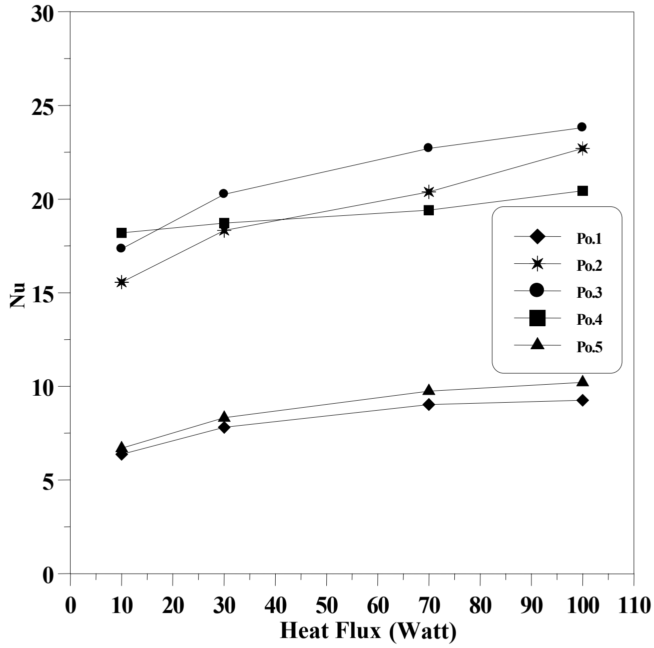

Fig.(10-b) Side Situation.

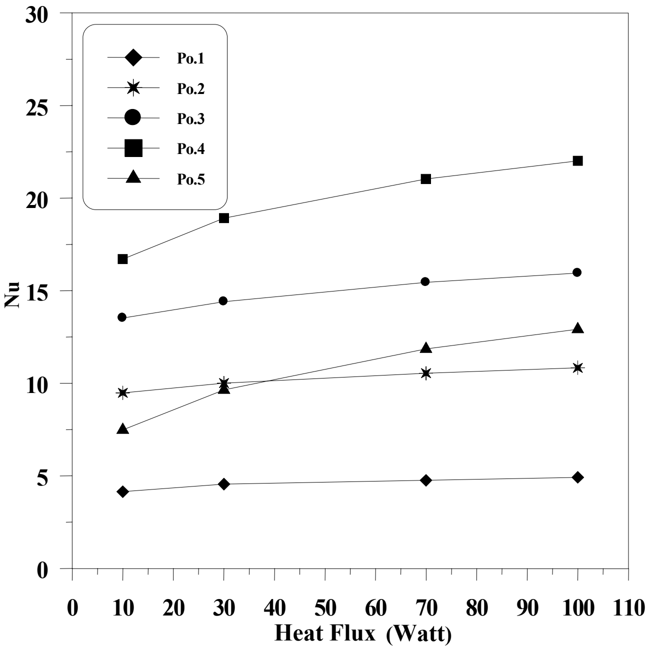

Fig.(10-c) Inverse Situation.

Fig.(10): Variation of average Nusselt number (Nu) with the heat flux values for different heating element positions and triangular situations.

REFERENCES:

[1] O. Aydin, W. J. Yang "Natural convection in enclosures with localized heating from below and symmetrical cooling from sides" Int. J. Numer. Meth. Heat Fluid Flow, Vol. 10, pp: 518-529, 2000.

[2] I. Hajri, A. Omri, S. Ben Nasallah "A Numerical model for the simulation of double-diffusive natural convection in a triangular cavity using equal order and control volume based on the finite element method" Desalination 206, pp:579-588, 2007.

[3] YasinVarol, AhmetKoca, Hakan F. Oztop "Laminar natural convection in saltbox roofs for both summerlike and winterlike boundary conditions" Journal of Applied Science6 (12), pp:2617-2622, 2006.

[4] YasinVarol, AhmetKoca, Hakan F. Oztop "Application of central difference scheme to the solution of natural convection equations for irregular shaped enclosure" Journal of Applied Science7 (4), pp:553-558, 2007.

[5] AhmetKoca, Hakan F. Oztop, YasinVarol "Natural convection heat transfer in a saltbox roof with eave in winter day conditions" Proceeding of clima, WellBeing Indoors, 2007.

[6] AhmetKoca, Hakan F. Oztop, YasinVarol "Effect of geometrical shape of roofs on natural convection for winter conditions" Proceeding of clima, WellBeing Indoors, 2007.

[7] Hakan F. Oztop, YasinVarol, AhmetKoca "Laminar natural convection heat transfer in a shed roof with or without eave for summer season" Applied thermal science (27) pp:2252-2265, 2007.

[8] Yadan Mao, Chengwang Lei, John Patterson "Natural convection in a triangular enclosure induced by solar radiaion" 16thAustralain Fluid Mechanics Conference 2-7 December, 2007.

[9] AhmetKoca, Hakan F. Oztop, YasinVarol "The effects of Prandtle number on natural convection in triangular enclosure with localized heating from below" Int. Communication in Heat and Mass Transfer, Vol.34, pp:511-519, 2007.

[10] YasinVarol, Hakan F. Oztop, Tuncay Yilmaz "Natural convection in triangular enclosures with protruding isothermal heater" International Journal of Heat and mass Transfer, Vol.50, pp:2451-2462, 2007.

[11] GoutamSaha,Md. Tofiqui Islam, SumonSaha, Md. Quamrul Islam "Natural convection in a tilttedisoceles triangular enclosure with discrete bottom heating" Thammasat Int. J. Sc. Tech., Vol.12, No.4, December, 2007.

[12] Sun Joo Kwon, Se Young Oh, Jae Hean Yun, Sei-Young Chung " Parametric investigations on the double diffusive convection triangular cavity" Journal of The GHUNCHEONG MATHEMATICAL SOCIETY, Vol.20, No.4, pp. 420-432, 2007.

[13] S. Roy, TanmayBasak, ch. Thirumalesha, ch.Murali Krishna "Finite elemente simulation on natural convection flow in a triangular enclosure due to uniform and non-uniform bottom heating" Journal of Heat Transfer Vol.130, 2008.

[14] TanmayBasak, S. Roy, S. Krishna Babu, A. R. Balakrishnan "Finite elemente analysis of natural convection flow in an isoceles triangular enclosure due to uniform and non-uniform heating at the side walls" International Journal of Heat and Mass Transfer Vol.51, pp:4496-4505, 2008.

[15] XuXu, Zitao Yu, Yacai Hu, Liwu Fan, Kefa Cen "A numerical study of laminar natural convection heat transfer around a horizontal cylinder inside a concentric air-filled triangular enclosure" International Journal of Heat and mass Transfer, Vol.53, pp:345-355, 2010.

[16] Patankar, S.V., “Numerical Heat Transfer and Fluid Flow” , Hemisphere Publishing Corporation, New York, 1980.

NOMENCLATURE:

Symbols	Descriptions	Units
g	*Gravitational acceleration*	m/s^2
$Gr = \frac{g\beta \Delta T L^3}{\nu^2}$	*Grashof number*	-
k	*Thermal conductivity of fluid*	W/mK
L	*Side Length of triangular enclosure*	m
Nu	*Average Nusselt number*	
p	*Pressure*	N/m^2
$pr = \frac{\nu}{\alpha}$	*Prandtl number*	-
$q^{\backslash\backslash}$	*Heat Flux value*	W
Ra	*Rayleigh number*	-
T	*Temperature*	K
T_f	*Film temperature*	K
T_h	*Hot temperature*	K
T_c	*Cold temperature*	K
V_x	*Velocity component in x-direction*	m/s
V_y	*Velocity component in y-direction*	m/s
x	*Cartesian coordinate in horizontal direction*	m
y	*Cartesian coordinate in vertical direction*	m
c_p	*Specific heat*	$J/kg\ K.$
Greek Symbols		
α	*Thermal diffusivity*	m^2/s
β	*Volumetric coefficient of thermal expansion*	K^{-1}
μ	*Viscosity*	$N.s/m^2$
ν	*Kinematic viscosity of the fluid*	m^2/s
ρ	*Density of the fluid*	kg/m^3
ψ	*Stream Function*	m^2/s
Abbreviation		
Po.	*Position of heating element*	