Effect of different positions of the heater on laminar natural convection in right triangular enclosure

Alireza Falahat

Department of Mechanics, Mahshahr Branch, Islamic Azad University, Mahshahr, Iran
a.falahat@mahshariau.ac.ir

***Abstract*—In this work, a numerical analysis of steady state laminar natural convection in a right triangle enclosure has been performed for heater located on horizontal wall and cold inclined wall. the influence of different heater position and Rayleigh number have been studied. The governing equations are solved with finite volume method (FVM) for the range of Rayleigh number as 10^3 to10^6. Results have clearly indicated Heat transfer increases with increasing of Rayleigh number for all of cases. The results show the heat transfer increases with decreasing of distance between hot and cold wall.**

***Keywords*—Natural convection, Heater position, Numerical method, Right triangle enclosure.**

I. INTRODUCTION

Natural convection in enclosures is important in many engineering applications that used in industries. In recent years, triangular enclosures have received a considerable attention because of its used in Building and triangular built-in-storage type solar collectors [1-2].

Several researches have been performed about natural convection in triangular enclosures was a experimental study conducted in by Flack et al. [3]. Akinsete et al. [4]. analysed laminar natural convection in a right triangular enclosure using numerical methods. Steady state solutions were obtained for the conditions in several aspect ratio and Grashof number.

Karyakin et al. [5] studied Transient laminar natural convection in isosceles triangular enclosures. Asan and Namli [6], studied the laminar natural convection heat transfer in triangular shaped roofs with different inclination (aspect ratio) angle and Rayleigh number in wintertime conditions. They used finite volume method and They found that heat transfer decreases with the increasing of aspect ratio.

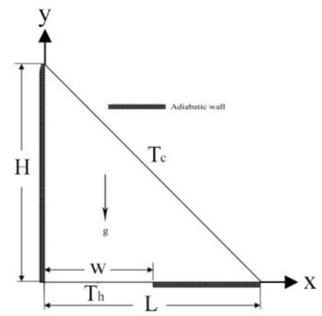
Varol et al. [7] investigated laminar natural convection in a triangle enclosure with flush mounted heater on vertical Wall by Finite difference method. Aminossadati et al [8] studied laminar natural convective heat transfer in the triangular enclosure by numerical method. They have considered the effect of position and dimension of heater and aspect ratio on fluid flow and heat transfer. They found at low Rayleigh numbers, the Nusselt number increases with the aspect ratio and decreases with the distance of the heater from the left vertex of triangle.

The main objective of this investigation is to analyze the effects of different positions of the heater on natural convection in a triangular enclosure with constant partial heating at horizontal walls and cooling at inclined wall with the adiabatic vertical wall is studied numerically using the finite volume method (FVM). In the present study we examine the heat transfer and fluid flow for different positions of the heater and different Rayleigh number in a enclosure.

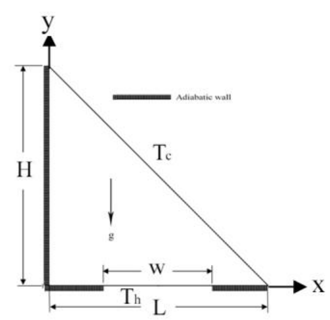
II. FORMULATION OF THE PROBLEM

The physical model with dimensions and boundary conditions according to different positions considered in the investigation are shown in Fig. 1. In these models, heater position can be changed and its dimension $(w = L/3)$ is constant. Length of bottom wall and height of vertical wall are shown by L and H, respectively. Inclined wall of triangle has constant cold temperature (T_c) and heater at bottom wall has constant hot temperature (T_h) and vertical wall is adiabatic.The system was considered to be incompressible, steady-state, Newtonian and the Boussinesq approximation was applied for fluid with constant physical properties.

a)



b)



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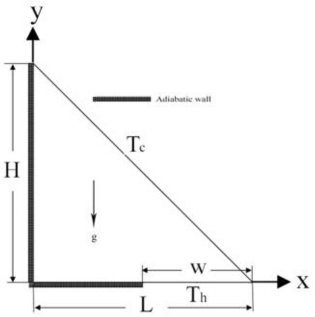

Fig. 1 Schematic diagram of the physical model
a) case 1, b) case 2, c) case 3

Under the above approximations, the governing equations for steady natural convection flow using conservation of mass, momentum and energy can be written

Continuity equation

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}=0 \tag{1}$$

Momentum equation

$$u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}=\frac{1}{\rho}\left[\mu\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}\right)-\frac{\partial p}{\partial x}\right] \tag{2a}$$

$$u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}=\frac{1}{\rho}\begin{bmatrix}-\frac{\partial p}{\partial y}+\mu\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}\right)+\\ (\rho\beta)\,g\,(T-T_c)\end{bmatrix} \tag{2b}$$

Energy equation

$$u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}=\alpha\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}\right) \tag{3}$$

where u and v are the velocity components in the x and y directions, respectively. p is the pressure, T is the temperature and β is the thermal expansion coefficient.
The variables can be transformed by using

$$X=\frac{x}{L}\qquad Y=\frac{y}{L}$$
$$U=\frac{uL}{\alpha}\qquad V=\frac{vL}{\alpha}$$
$$P=\frac{pL^2}{\rho\alpha^2}\qquad \theta=\frac{T-T_c}{T_h-T_c}$$
$$Ra=\frac{g\,\beta L^3(T_h-T_c)}{\alpha\nu}\qquad \Pr=\frac{\nu}{\alpha} \tag{4}$$

The non-dimensional forms of the governing equations are:

$$\frac{\partial U}{\partial X}+\frac{\partial V}{\partial Y}=0 \tag{5}$$

$$U\frac{\partial U}{\partial X}+V\frac{\partial U}{\partial Y}=-\frac{\partial P}{\partial X}+\frac{\mu}{\rho\alpha}\left(\frac{\partial^2 U}{\partial X^2}+\frac{\partial^2 U}{\partial Y^2}\right) \tag{6a}$$

$$U\frac{\partial V}{\partial X}+V\frac{\partial V}{\partial Y}=-\frac{\partial P}{\partial Y}+\frac{\mu}{\rho\alpha}\left(\frac{\partial^2 V}{\partial X^2}+\frac{\partial^2 V}{\partial Y^2}\right)+Ra\Pr\theta \tag{6b}$$

$$U\frac{\partial \theta}{\partial X}+V\frac{\partial \theta}{\partial Y}=\left(\frac{\partial^2 \theta}{\partial X^2}+\frac{\partial^2 \theta}{\partial Y^2}\right) \tag{7}$$

The dimensionless boundary conditions of the problem under consideration can be written as

- Bottom wall: On the heater $T=T_h$, On unheated wall $\frac{\partial T}{\partial n}=0$
- Vertical wall: $\frac{\partial T}{\partial n}=0$
- Inclined wall: $T=T_c$

For calculation of the local Nusselt number was performed over partially heated active wall by

$$Nu_x=-\frac{\partial\theta}{\partial Y}\Big|_{Y=0} \tag{8}$$

The average Nusselt number (Nu) at bottom wall is determined by integrating Nu along the hot wall.

$$Nu=\int_0^w Nu_x\,dX \tag{9}$$

The numerical solution is obtained using commercial computational fluid dynamics software, Fluent 6.3.0, which employs a finite volume method for the discretization of the continuity, momentum and energy equations. The SIMPLE algorithm is used to couple the pressure and velocity terms. Discretization of the momentum and energy equations is performed by a second order upwind scheme and pressure interpolation is provided by PRESTO scheme. Convergence criterion considered as residuals is admitted 10^{-3} for momentum and continuity equations and for the energy equation it is lower than 10^{-6}.

III. RESULTS

A numerical investigation has been performed to obtain laminar natural convection heat transfer in a right triangular enclosure with constant partial heating at horizontal walls and cooling at inclined wall with the adiabatic vertical wall. Finite volume method was used to solve governing equations of Continuity, momentum and energy. The effect of heater position and Rayleigh number are analyzed.

Fig. 2 shows the effect of Rayleigh number on streamlines (2.a) and isotherms (2.b). In this case, heater was located in middle position of horizontal wall (case 2). As shown in the figure, single cell is obtained in a clockwise circulation at all Rayleigh numbers except $Ra = 10^6$ that is made one small vortex at corner of cavity. The cell goes through the bottom corner of triangle with the increasing Rayleigh number. The flow moves through the bottom corner of the triangle and rises up along vertical wall and flows down along inclined wall. Temperature is distributed with this motion of flow and temperature boundary layer starts to develop from hot to cooled wall of enclosure. At the smallest value of Rayleigh number it shows a semi-circle shaped distribution and as Rayleigh number increases, the isotherms are deformed, this distortion of the temperature field is due to the increase in speed of the counter clockwise rotating cell.

Fig. 3 shows streamlines (on the left) and isotherms (on the right) for three different position of heater. As it can be seen from streamlines, single cell is obtained in a clockwise circulation at all position of heater except case 3 (near right bottom corner) that is made one small vortex at corner of cavity. The cell goes through the bottom corner of triangle when the heater becomes closer to inclined wall. When heater Becomes closer to inclined wall, distance of heater to cold wall (inclined wall) is short and heat transfer increases.

Fig. 4 shows variations of local Nusselt number for heater wall for different position of heater and $Ra = 10^5$. It is observed that position of heater is an important parameter on natural convection. When the heater is located on the left of horizontal wall (Case 1), the Local Nusselt number increases. In case 1 the value of local Nusselt number is small because distance between hot and cold wall is long. When the heater is located on the middle of horizontal wall(case 2), the local Nusselt number increases similar case 1. If the heater is located on the right, of horizontal wall (case 3), in this case, temperature difference is high and U-shaped distribution of local Nusselt number like U-shaped.

Fig. 5 shows effect of heater position on average Nusselt number for different Rayleigh number. The value of average Nusselt number increase with decreasing distance between hot and cold wall for all Rayleigh number . In the case 3, the higher Nusselt number is obtained due to short distance between hot and cold wall but in other case the smallest Nusselt number obtained because distance between hot and cold wall is long.

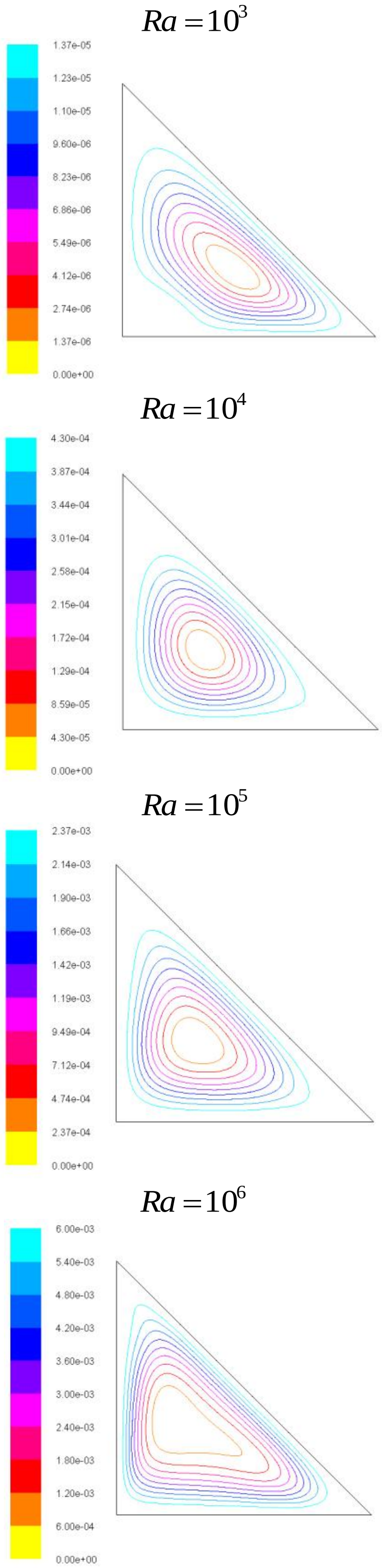

Fig. 2a The streamlines of case 2 for various Rayleigh number

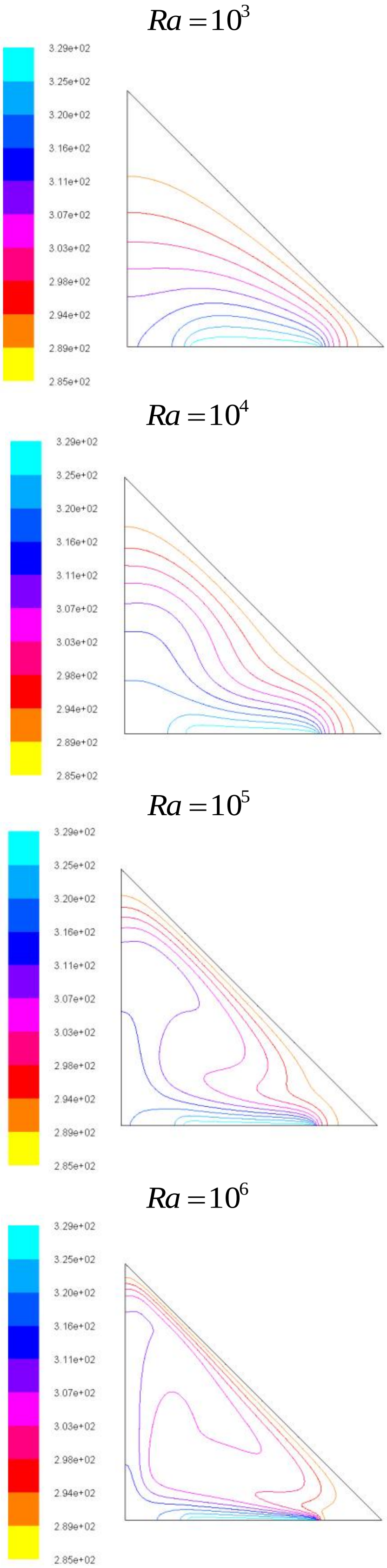

Fig. 2b The isotherms of case 2 for various Rayleigh number

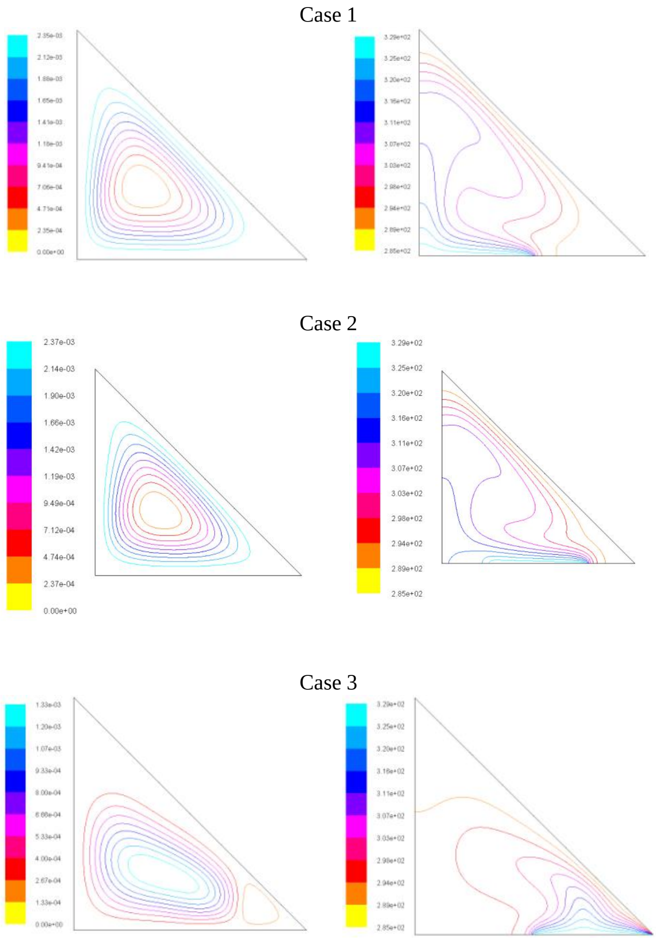

Fig. 3 Comparison of the streamline (on the left) and isotherms (on the right) at $Ra = 10^5$

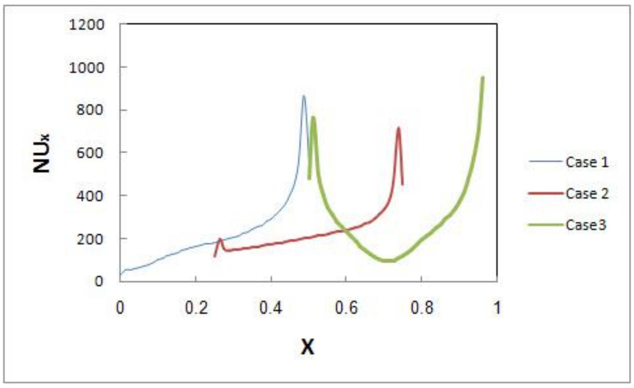

Fig. 4 Local Nusselt number along the horizontal wall at $Ra = 10^5$

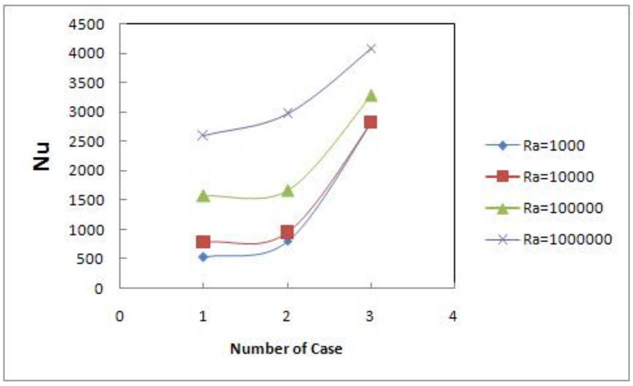

Fig. 5 Variation of average Nusselt number for different cases

IV. Conclusion

In this paper, laminar natural convection in right triangular enclosure is investigated. The influence of different heater position and Rayleigh number have been studied for constant partial heating at horizontal walls and cooling at inclined wall with the adiabatic vertical wall. The results of the numerical analysis lead to the following conclusions:

If Rayleigh number increases, the heat transfer increases in all position of heater (cases 1,2 and 3).

The streamlines and isotherms countor are affected by the position of heater and Rayleigh number.

The heat transfer increase with decreasing distance between hot and cold wall for all Rayleigh number.

Nomenclature

g	Gravitational acceleration, ms^{-2}
H	Height of triangle, m
L	Length of horizontal wall, m
n	Normal direction of coordinate
Nu	Average Nusselt number
p	Fluid pressure, Pa
P	Dimensionless pressure
$\Pr$	Prandtl number
Ra	Rayleigh number
T	Temperature, K
u,v	Velocity component, ms^{-1}
U,V	Dimensionless velocity component
x,y	Dimensional Cartesian coordinates, m
X,Y	Nondimensional cartesian coordinates
w	Width of heater, m

Greek symbols

α	Thermal diffusivity, $m^2 s$
β	Thermal expansion coefficient, K^{-1}
θ	Dimensionless temperature
μ	Dynamic viscosity, $kgm^{-1}s$
ν	Kinematic viscosity, $m^2 s^{-1}$
ρ	Density, kgm^3
ψ	Stream function

Subscripts

c	Cold wall
h	Hot wall

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