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EFFECT OF STAGGERED NON-CONDUCTIVE PARTITION ON HEAT TRANSFER IN A VARIABLE SHAPE CAVITY

ZIAD M. ALMAKHYOUL, RAED A ALI, MAAN S. ALDABBAGH

Lecturer, Department of Mechanical Engineering- university of Mosul-Iraq

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ABSTRACT

This paper represents the effect of staggered non-conductive partitions on natural convection heat transfer in variable shape cavity (rectangular, trapezoidal and triangular cavities) have been studied . The three partitions mounted on upper and lower walls. The vertical walls of all cavities represents cold walls and the upper wall was adiabatic while the lower wall represents the hot wall. The numerical solution using finite difference technique to solving continuity, momentum and energy equations. The results show that the increasing in the lengths of partitions leads to reduce in Nusselt number and heat transfer for all cavities while at the same length of partition the changes in the cavity from rectangular to trapezoidal to triangular cavities makes decreasing in the Nusselt number.

KEYWORDS: Heat Transfer, Non-conductive partition, Staggered Partitions, Cavities

NOMENCLATURE

L	Cavity width
H	Cavity high
Nu	Nusselt number
P	Fluid pressure
Pr	Prandtl numbers
Ra	Rayleigh number
T	Temperature
U	Velocity in x direction
V	Velocity in y direction
ϕn	New temperature
ϕ	Old temperature
X	Horizontal distance
Y	Vertical distance
$\hat{\psi}$	Dimensionless stream function
ψ	Stream function
ω	Vorticity
Ω	Dimensionless velocity

θ	Dimensionless temperature
ρ	Density
ϑ	Kinematic viscosity
α	Thermal diffusivity
β	Thermal expansion coefficient

Subscript

H	Hot
C	Cold

1. INTRODUCTION:

Many researchers have been showed the temperature distribution, stream function, fluid flow and heat transfer in cavities with and without partitions to study the effect of partitions on the heat transfer inside the cavities. (Türkoğlu and Yücel, 1996) Ref no [1] studied the effect of partitions on natural heat transfer inside divided square enclosure. It concluded the increasing in average (Nu) number with increasing in (Ra) number and decreasing when the partitions number are increasing.

(Dağtekin and Öztop, 2001) Ref no [2] showed the heat transmission in the rectangular cavity using two heated partitions attached to the bottom wall at Ra number from (104 to 106). The research was concerned with varying in length and location partitions

(Altaç and Kurtul, 2007) Ref no [3] studied the effect natural convection inside rectangular enclosures using a hot thin plate at the center of enclosure numerically. The plate was isothermal. The vertical wall in the enclosure was the cooled. The study included the effect of Rayleigh numbers from (105 to 107) and enclosure angles between (0° to 90°). The flow of heat and distribution of temperature were correlated mean Nusselt numbers.

(Abdullatif and Ali, 2006) Ref [4] study laminar flow of natural convection inside inclined enclosure with partitions, Rayleigh number at range from (103 to 106) and the inclination angle for the enclosure was (0° -90°). It was found when the Rayleigh number increased the mean Nusselt numbers increased also the increasing in the partition length leads to decreases in speed of flow in the enclosure.

(Ali L., Ayad F. and Ahmad F.,2010) ref no [5] studied the effect buoyancy heat transfer in square enclosure using staggered partitions numerically. The staggered partitions were located on the upper and lower walls and variable length of the partitions. The horizontal wall at different temperature while the vertical walls were adiabatic. The numerical solution using finite difference technique. The results showed that the increasing in the Rayliegh numbers leads to increases in average Nussult number while the Nusselt number decreases with increasing in length of partitions.

(Abid A, 2012) Ref no [6] studied numerically the natural convection inside rectangular enclosure filled with partitions. The lower and upper walls of enclosure are adiabatic while the vertical walls having different temperatures. Two baffles were attached to horizontal walls in upper and lower walls. The governing equations were solved using finite volume technique. The study included variable values of Rayleigh numbers, baffle locations, baffle lengths . The results showed that the increasing in the Rayliegh numbers leads to increases in average Nussult number while the Nusselt number decreases with increasing in length of baffles.

(Moukalled and Darwish, 2012) Ref no [7] studied the effects of two offset partitions on natural convection of heat transfer . The top wall inclined and bottom horizontal wall of trapezoidal cavities. There are two boundary conditions are concerned. The first, the left short vertical wall represented the source of heat while the right long vertical wall is received (cooled wall). While In the second, the right long vertical wall represented source of heat while the left short vertical wall is received (cooled wall). Results are showed decreases in feat transfer when increasing in prandle number and partition lengths.

(Costa V.A.F., 2012) Ref no [8] studied the natural convection in partitioned square enclosures. The enclosure filled with air. Two partitions are following arrangement in the enclosure. Rayleigh number is used against variable partition location, partition length and thermal conductivity. The results showed that the increasing in the Rayleigh numbers leads to increases in Nussult number while the Nusselt number decreases with increasing in partition length.

(Ziad M. Al-Makhyoul,2014) ref no [9] studied numerically the effect of baffles on natural convection heat transfer inside trapezoidal cavity. The baffles are fixed to active surface. The vertical and inclined walls of the cavity having different temperature. The upper and lower horizontal surfaces are insulated. In the first case one baffle is fixed and two baffles in the second case and three baffles in the third case. The results showed that the increasing in (Ra) number leads to increasing in (Nu) number while increasing in the baffle length and number of baffles leads to decreasing in Nusselt number.

This present work concerned with study the natural convection of heat transfer using staggered partitions arranged one in upper and two in lower surfaces of enclosure. Three shapes of enclosures have been studied such as rectangular, trapezoidal and triangular enclosures shapes. The vertical walls are to be maintained lower temperature and the horizontal bottom surface is assumed to be a higher temperature while the horizontal top surface is adiabatic. The effect of lengths of partitions and Rayleigh number on Nusselt number, fluid flow and temperature distribution will be considered.

2.THE MATHEMATICAL FORMULATION:

Consider the shapes of enclosures (rectangular, trapezoidal and triangular) as shown in Figure (1). The upper wall of the rectangular and trapezoidal enclosures is adiabatic while the vertical and bottom walls for all enclosures at different temperatures. the vertical walls represented the cold wall while the lower wall represented the active wall. Three insulated staggered baffles are arranged to the upper and lower walls. The positions of the baffles are constant in the three shapes of enclosures while the baffles length is different .

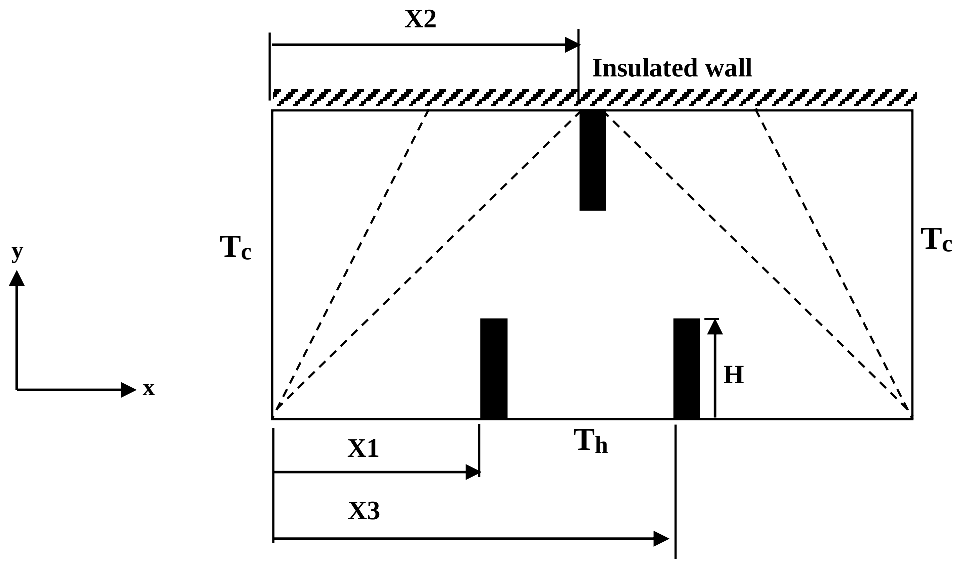

Figure (1) Rectangular, trapezoidal and triangular cavities with staggered non-convective partitions

The system was considered to be two-dimensional, steady-state, Newtonian, incompressible and the Boussinesq approximation was applied for fluid with constant properties. It is assumed that the radiation effect can be taken neglected. The dimensionless equations of stream functions form can be written as:

$$X=\frac{x}{L},Y=\frac{y}{L},,\phi=\frac{\psi \Pr}{\vartheta},\theta=\frac{T-T_c}{T_h-T_c},\Omega=\frac{\omega L^2 \Pr}{\vartheta},Ra=\frac{\beta g(T_h-T_c)L^3 \Pr}{\vartheta^2},\quad \Pr=\frac{\vartheta}{\alpha}$$

Based on dimensionless variables above equations can be obtained as :

Energy equation:

$$\frac{\partial^2\theta}{\partial X^2}+\frac{\partial^2\theta}{\partial Y^2}=\left(\frac{\partial\phi}{\partial Y}\frac{\partial\theta}{\partial X}-\frac{\partial\phi}{\partial X}\frac{\partial\theta}{\partial Y}\right) \qquad \ldots 1$$

Momentum equation:

$$\frac{\partial^2\Omega}{\partial X^2}+\frac{\partial^2\Omega}{\partial Y^2}=\frac{1}{\Pr}\left(\frac{\partial\phi}{\partial Y}\frac{\partial\Omega}{\partial X}-\frac{\partial\phi}{\partial X}\frac{\partial\Omega}{\partial Y}\right)-Ra\frac{\partial\theta}{\partial x} \qquad \ldots 2$$

Continuity equation:

$$\frac{\partial^2\phi}{\partial X^2}+\frac{\partial^2\phi}{\partial Y^2}=-\Omega \qquad \ldots 3$$

boundary conditions on the lower surface written as:

$$\phi=\Omega=0,\quad \theta=1 \qquad \ldots 4$$

And On the upper surface:

$$\phi=\Omega=0,\qquad \frac{\partial\theta}{\partial Y}=0 \qquad \ldots 5$$

While the boundary conditions on inclined walls:

$$\phi=\Omega=0,\qquad \theta=0$$

3. THE NUMERICAL ANALYSIS:

The equations which used above for steady state, laminar and two dimensional natural convection heat transfer in variable shape cavity with thin baffles attached to the upper and lower walls are solved using the finite difference technique (Taylor series) being effective and simple in formulation with numerical accuracy in results.

For unknown variable nodes ϕ (x,y), the continuous first and second order derivatives at mesh nodes (I,j), which are $(\partial\phi/\partial x)_{i,j}$ and $(\partial^2\phi/\partial x^2)_{i,j}$ respectively may be expressed in three ways:

The first and second derivatives for the approximation of forward difference for the one side are:

$$\left(\frac{\partial\phi}{\partial x}\right)_{i,j} = \frac{-3\phi_{i,j} + 4\phi_{i+1,j} - \phi_{i+2,j}}{2\Delta x} + O(\Delta x)^2 \quad \ldots 6$$

$$\left(\frac{\partial^2\phi}{\partial x^2}\right)_{i,j} = \frac{\phi_{i,j} - 2\phi_{i+1,j} + \phi_{i+2,j}}{(\Delta x)^2} + O(\Delta x)^2 \quad \ldots 7$$

While for the backward derivatives:

$$\left(\frac{\partial\phi}{\partial x}\right)_{i,j} = \frac{-3\phi_{i,j} + 4\phi_{i-1,j} - \phi_{i-2,j}}{2\Delta x} + O(\Delta x)^2 \quad \ldots 8$$

$$\left(\frac{\partial^2\phi}{\partial x^2}\right)_{i,j} = \frac{\phi_{i,j} - 2\phi_{i-1,j} + \phi_{i-2,j}}{(\Delta x)^2} + O(\Delta x)^2 \quad \ldots 9$$

The central difference for the internal meshes can obtained as:

$$\left(\frac{\partial\phi}{\partial x}\right)_{i,j} = \frac{\phi_{i+1,j} - \phi_{i-1,j}}{2\Delta x} + O(\Delta x)^2 \quad \ldots 10$$

$$\left(\frac{\partial^2\phi}{\partial x^2}\right)_{i,j} = \frac{\phi_{i-1,j} - 2\phi_{i,j} + \phi_{i+1,j}}{(\Delta x)^2} + O(\Delta x)^2 \quad \ldots 11$$

Where $O(\Delta x)^2$ is second order of truncation error.

The backward difference equations for the inclined wall can be yields as:

$$\left(\frac{\partial\phi}{\partial x}\right)_{i,j} = \frac{-3\phi_{i,j} + 4\phi_{i-1,j} - \phi_{i-2,j}}{(1+a)\Delta x} \quad \ldots 12$$

$$\left(\frac{\partial\phi}{\partial x}\right)_{i,j} = \frac{-3\phi_{i,j} + 4\phi_{i,j-1} - \phi_{i,j} - 2}{(1+b)\Delta x} \quad \ldots 13$$

$$\left(\frac{\partial^2\phi}{\partial x^2}\right)_{i,j} = \frac{2}{\Delta x(1+a)}\left(\frac{\phi_{i+1,j} - \phi_{i,j}}{a\Delta x} - \frac{\phi_{i,j} - \phi_{i-1,j}}{\Delta x}\right) \quad \ldots 14$$

$$\left(\frac{\partial^2\phi}{\partial y^2}\right)_{i,j} = \frac{2}{\Delta y(1+b)}\left(\frac{\phi_{i,j+1} - \phi_{i,j}}{b\Delta y} - \frac{\phi_{i,j} - \phi_{i,j-1}}{\Delta y}\right) \quad \ldots 15$$

The mean Nusselt number is defined as follows: $Nu = -\frac{\partial\theta}{\partial n}$

where (n) represents the normal direction on plane.

4.RESULTS AND DISCUSSION:

In order to comprehend the natural convection heat transfer inside the cavity, the temperature (isotherm lines) and flow field (stream functions) are plotted for the flow and heat transfer in the variable shape cavities (rectangular, trapezoidal and triangular cavities) with staggered non-conductive partitions were studied using numerical technique in the present research. It is assumed that the vertical walls of cavity represented the cold walls, and the upper wall was insulated while the lower wall represented the hot wall. The air represented the working media in the three cavities with Prandtl number of 0.71. Staggered non-convective partitions are fixed one on the upper surface and two on lower surface. The dimensionless lengths of non-conductive partitions are variable at (0.1, 0.3 and 0.5)

Figure (1) show the effect of Rayleigh number on natural convection inside rectangular cavity on isotherm lines and on the stream functions. It's clear the increasing in the Rayleigh number leads to increasing in gradient of temperature along the vertical side and subsequently heat transfer by convection increasing. When the non-conductive staggered partitions are mounting in upper and lower cavity at (0.1, 0.3 and 0.5) the flow of fluid will be more slow because the partitions cause obstruction therefore, the speed of flow decreases, which cause decreases in the heat transfer by convection. The increasing in the Rayleigh number leads to increasing in the buoyancy force thus the heat transfer by convection enhanced. This analysis also applying on the trapezoidal and triangular cavities in figures (3) and (4). When the cavity is small volume, the force being fast sufficiently to rotate fluid.

Figure (5) represented the comparison among the rectangular, trapezoidal and triangular cavities. It is show the change in cavities shapes from rectangular to trapezoidal and then to triangular cavities leads to increasing in Nusselt number because the minimum cavity makes flow moving faster compare with other cavities while the flow moving slowly in the trapezoidal cavity and more slow in the rectangular cavity. Figure (6) show the effect of increasing in the dimensionless length of non-conductive staggered non-convective partitions on the Nusselt number for the three shapes, rectangular, trapezoidal and triangular cavities. The increasing in the dimensionless lengths of non-conductive partitions leads to more obstruction for the fluid flow inside the cavity and that causes decreasing in the temperature gradient along the cold side of cavity and heat transfer and then decreasing in the Nusselt number.

5. The conclusions:

The present study included the heat transfer inside three shapes of cavities (rectangular, trapezoidal and triangular) as shown in Figure (1). The upper surface of the rectangular and trapezoidal enclosures is adiabatic while the vertical walls and bottom surface for all enclosures at different temperatures. the vertical walls represented the cold wall while the lower surface represented the hot wall. Three non-conductive partitions arranged staggered on the upper and lower surfaces. The positions of the partitions are constant in the three shapes of enclosures while the partition lengths are different at (0.1, 0.3 and 0.3) . the conclusions from this study can be written as:

1-Generally, the increasing in the Rayleigh numbers leads to increasing in Nusselt numbers

2-The increases in the non-conductive partitions leads to decreasing in the Nusselt numbers.

3-The changing in the shape of cavity from rectangular cavity to trapezoidal cavity and then to triangular cavity leads to increases in the Nusselt number.

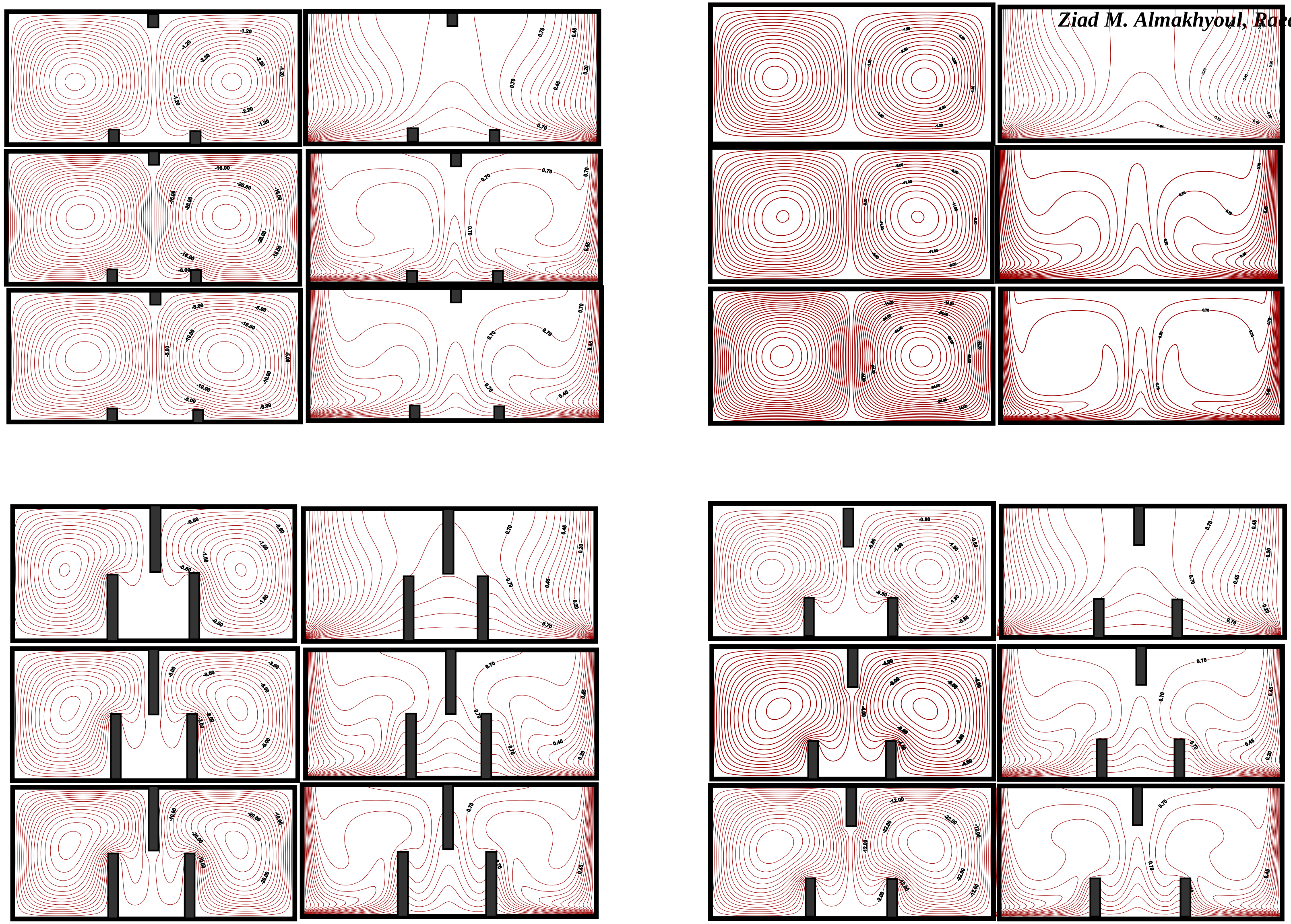

Figure (2) temperature distribution and stream function for rectangular cavity without and with staggered non-conductive partitions

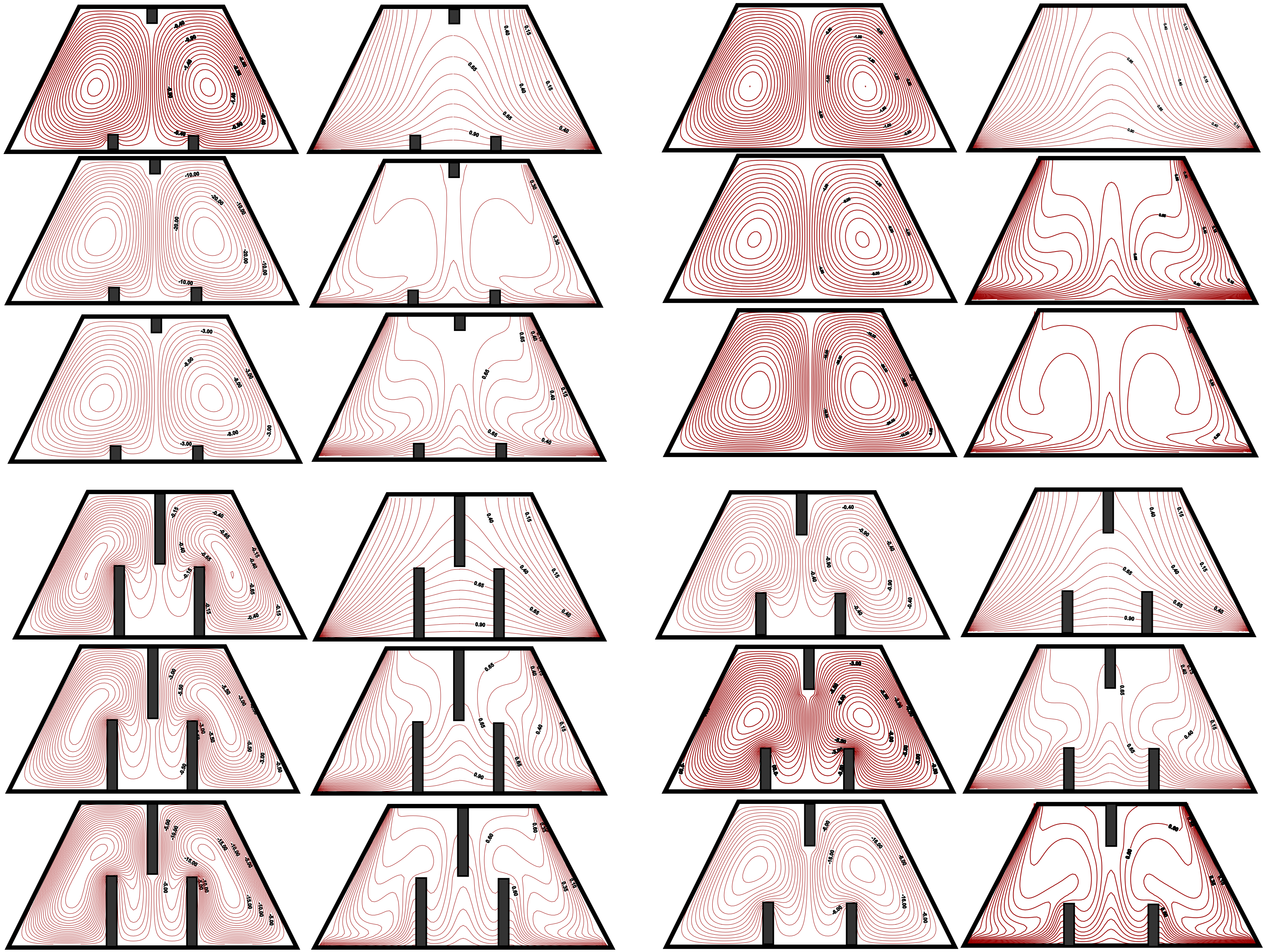

Figure (3) temperature distribution and stream function for trapezoidal cavity without and with staggered non-conductive partitions

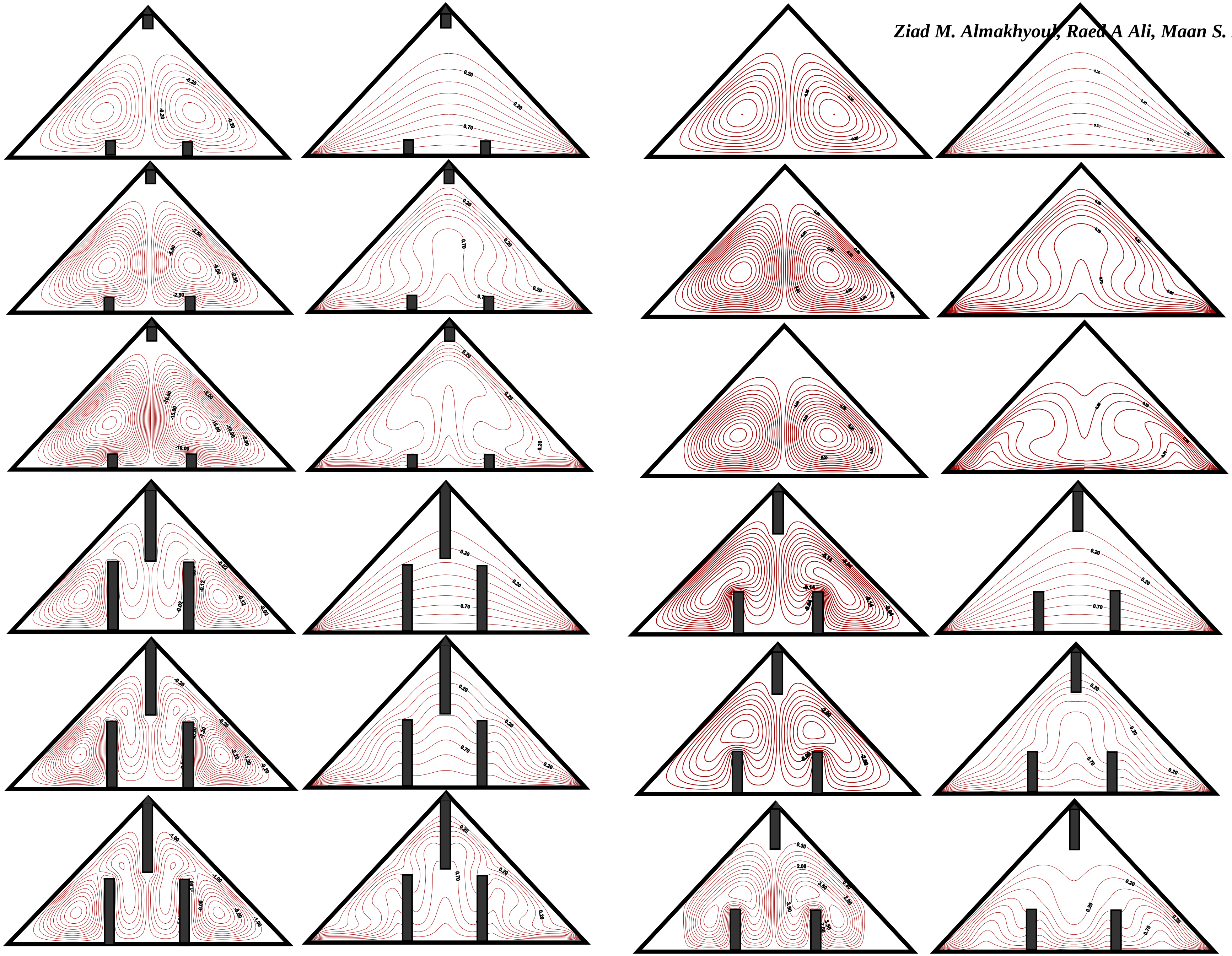

Figure (4) temperature distribution and stream function for triangular cavity without and with staggered non-conductive partitions

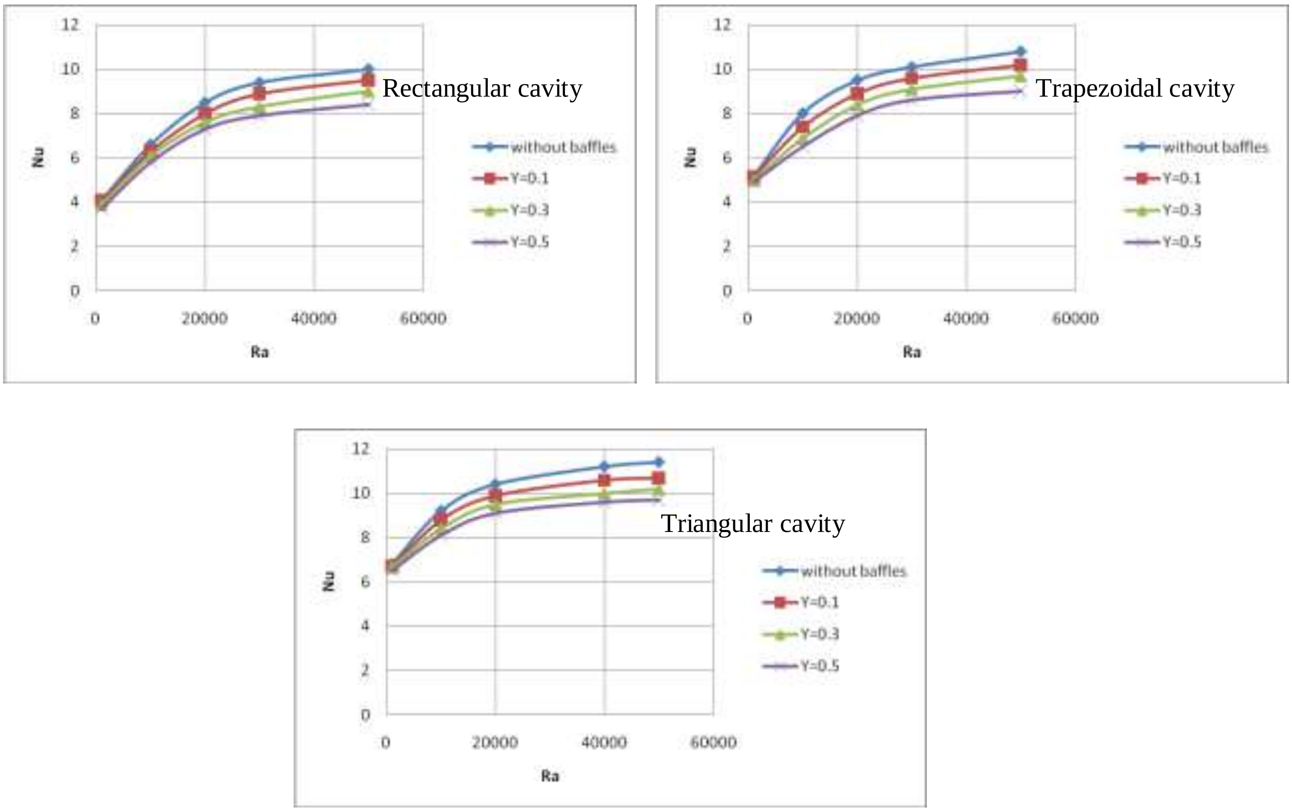

Figure (5) Relation between Rayliegh number and Nusselt number for rectangular, trapezoidal and triangular cavity without and with staggered non-conductive partitions at lengths of (Y=0.1, 0.3 and 0.5)

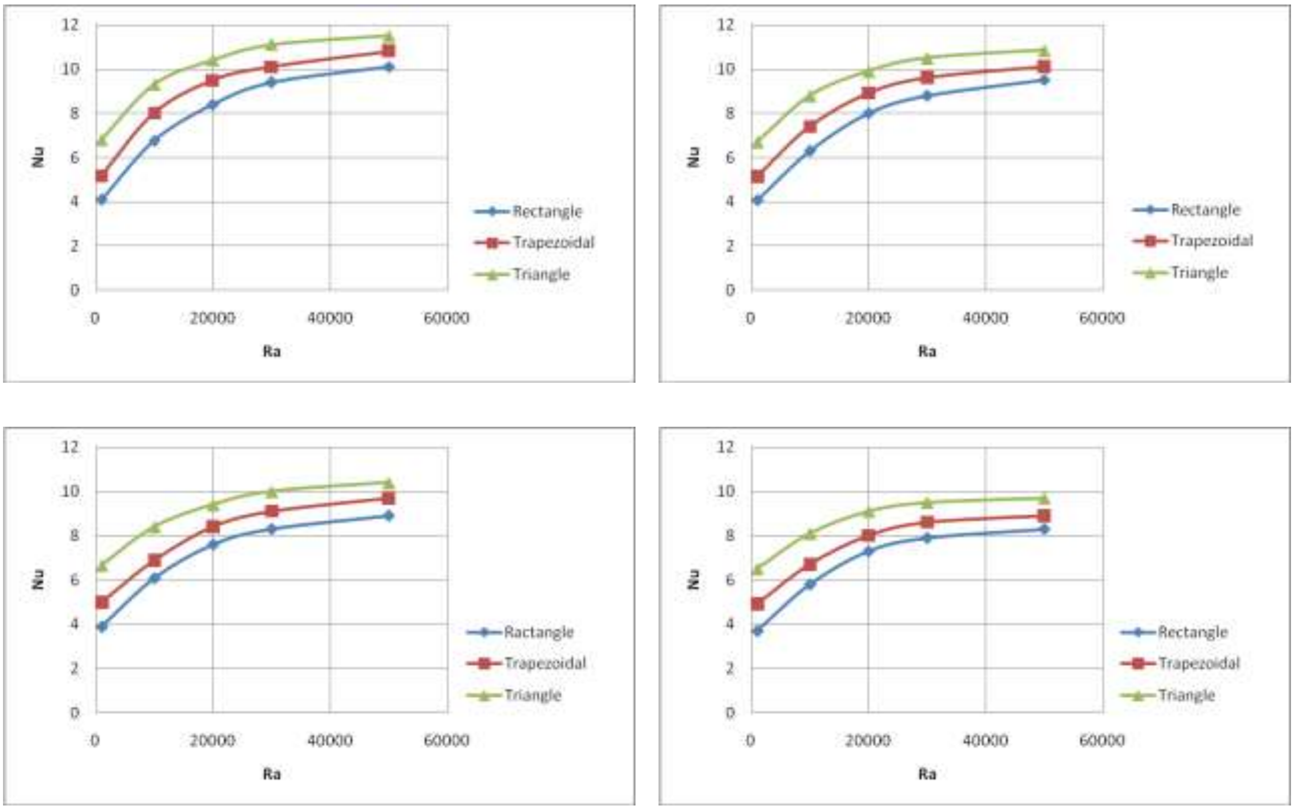

Figure (6) Effect of non-conductive partition lengths on Nusslt number for rectangular , trapezoidal and triangular cavities at (Y=0.0, Y=0.1, Y=0.3 and Y=0.5)

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