Numerical Study of Convection Air Currents Around a Hot Cylinder Inside a Triangular Cavity

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Abstract

A numerical study was performed of natural laminar convective heat transfer to its concentrated triangular enclosure around a horizontal circular cylinder. The air-filled enclosure kept the inner and outer cylinders at uniform temperatures. The Boussinesq density approximation to the momentum problem and the control volume approach iteratively resolved the governing equations to explain buoyancy. CFD results show that the velocity behavior increases by increasing Ra, so the stream lines becomes more sluggish and less uniform behavior and vortices gets less circulated pattern. The rotation angle α has significant effect on vortices, at 90o gives the higher range of velocity zones of free convection with higher range. The thermal boundary layer seems to be larger in rr=0.455 as compared with rr=0.345 and decreases by increasing α. The larger variation of isotherms and thermal boundary layer appears at lower α because the higher heat transfer rate occurs at higher α and becomes maximum at 90o. Eight correlations of average Nusselt number have been deduced as a function of Rayleigh number for the taken values of aspect ratio and enclosure angles of rotation and inclination.

Keywords: Natural Convection, Horizontal Triangular Enclosure, Inclination Angle, Circular Cylinder.

دراسة عددية ديناميكية حول التيارات الهوائية الحرارية حول أسطوانة ساخنة داخل تجويف مثلثي

عقيل عبدالله محمد ، انسام عادل محمد ، Shylesha V Channapattanac

الخلاصة:

تم إجراء دراسة عددية لانتقال الحرارة بالحمل الطبقي الطبيعي إلى غلافه المثلثي المركز حول أسطوانة دائرية أفقية. تم ملء العلبة بالهواء وتم الحفاظ على الأسطوانات الداخلية والخارجية عند درجات حرارة موحدة. تم نمذجة تأثير الطفو باستخدام تقريب كثافة Boussinesq لمعادلة الزخم وباستخدام نهج التحكم في الحجم وحل المعادلات الحاكمة بشكل تكراري. تظهر نتائج CFD أن سلوك السرعة يزداد بزيادة Ra ، وبالتالي تصبح خطوط التدفق أكثر تباطؤًا وسلوكًا أقل انتظامًا وتقل حركة الدوامات. زاوية الدوران α لها تأثير كبير على الدوامات ، عند 90 درجة تعطي نطاقًا أعلى من مناطق السرعة للحمل الحراري الحر مع نطاق أعلى. تبدو طبقة الحدود الحرارية أكبر في rr = 0.455 مقارنةً بـ rr = 0.345 وتنخفض بزيادة α. يظهر التباين الأكبر في متساوي الحرارة والطبقة الحدودية الحرارية عند أدنى α لأن معدل نقل الحرارة الأعلى يحدث عند أعلى α ويصبح الحد الأقصى عند ٩٠ درجة. تم استنتاج ثمانية ارتباطات لمتوسط عدد نسلت كدالة لرقم رايلي للقيم المأخوذة لنسبة العرض إلى الارتفاع وزوايا العلبة للدوران والميل

1. Introduction

The study of natural convection in rectangular enclosure is the simplest, therefore; the rectangular enclosure is the most common test section for natural convection studies. The application of study free convection in rectangular enclosure is building heating [1]. The previous work investigations of rectangular enclosure are:

Janusz T. Cie'sli 'nsk [2] examined the effect of aspect ratio, temperature difference, and internal body length on the natural flow of a body immersed in a horizontal cylinder enclosure. The results demonstrated that the Nu value increases as the aspect ratio and Ra value rise. Using CFD software, the stream lines, victor's distribution, and isotherms were generated. Ben-Nakhi and Chamkha [3] utilized fins within a rectangular enclosure, with the heater located on the left wall and the other three walls cooled. The researchers discovered that the circular trajectory streamlines grew more slow and distorted due to the ensuing eddies caused by the fin present at $Ra=10^4$. This inquiry provides an understanding of the effect of the interior structure's dimensions. Varol et al. [4] opted for an inclined enclosure with a square interior body. The temperature of slanted surfaces is lower than the temperature of horizontal surfaces. By increasing Ra, increased streamlines and isothermal distribution were observed. Moreover, the direction of the stream line motion depended on the body-related process; for heating, the stream line motion was clockwise, but for cooling, it was counterclockwise. Oztop and Abu-Nada [5] used nanofluid to improve free convection within a rectangular container. The results demonstrated that the nanofluid improves the performance of free convection by modifying its physical properties (a change in Pr, Gr, Nu, and Ra). The greater the Ra number, the greater the Nu, and the more distorted the fluid stream lines. In addition, increased nanoparticle concentrations result in a decrease in enhancement because wall depositions increase heat transfer resistance via conduction. Varol et al. [6] studied the influence of tilt, heater length, and Pr on a square enclosure with insulated walls and a heater situated on a corner. With inclination, the heat transfer coefficient can rise or fall, but the Pr has a considerable effect on the heat transfer rate. The range of the Prandtl number was 0.07 to 70, and the range of the Ra number was 10^3 to 10^6. A. K. Hussein and S. H. Hussein [7] used Ra and inclination to figure out the stream and isothermal lines for a square enclosure. Optimal conditions were reached when $Ra = 10^5$ and 30^o were applied to the Rayleigh number and angle of inclination, respectively. Nardini et al. [8] explored natural laminar convection in a two-dimensional square cavity with side length H based on experimental and numerical analyses of vertical side walls. The discovery of the simple fluid flow and heat transfer properties of natural convection in two-dimensional square enclosures was aided by the use of isothermal forms, flow curves, velocity maps, and average Nusselt numbers to illustrate the effect of the varied heating sizes of the side walls. Chowdhury et al. [9] looked at the natural convection of a heated sphere inside a triangle.The results demonstrated that increasing heat flux and circular diameter with the same Ra (105) and Pr (0.71), increased stream lines. Altaee et al. [10] looked at how a hot triangular body's tilt and Ra affect natural convection inside a cold rectangular enclosure with air inside. The highest values were recorded at 30 degrees and $Ra=10^6$. The stream line counters of varying degrees of inclination and Ra were attained. Sarper et al. [11] investigated spontaneous convection in the vertical channel. Two flush-mounted heaters discreetly warm one of the canal's walls, while the other is insulated. The effects of distance between heaters on heat transfer and hot spot temperature were tested, despite the fact that the total duration of the heaters remained unchanged. The results were presented as differences in surface temperature, hot spot temperature, and Nusselt number together with the corrected Grashof number, distance between heaters, and temperature. Ali et al. [12] explored free convection heat transfer in square enclosures filled with shallow water. The bottom surface was subjected to a boundary layer of constant thermal flux, while the top surface was subjected to ambient air pressure. The results demonstrated that the coefficient of heat transfer increased with the modified Rayleigh number and the overlap of the two aspect ratios' fields. Moreover, as the aspect ratio grows for the corrected Rayleigh number range, the Nusselt number drops in the overlapping region. To examine the influence of heat dissipation on the heat transfer process, Saglam et al. [13] placed discrete heat sources inside a rectangular enclosure with insulated side walls and an isothermal other side wall. For various adjusted Rayleigh numbers and heat dissipation ratios, variations in surface temperatures, the temperature of the hot spot, and the average number of Nusselts were determined. The results were characterized by streamlining, isothermal, and heat lines. Saglam et al. [14] explored natural convection in a rectangular container with two flush-mounted heat sources quietly heating the walls. The lateral sidewall maintained a consistent temperature, while the horizontal sides remained unheated. The results indicate that the two-dimensional measurements conveyed the general heterogeneity of the problem, that the conduction and heat transfer of radiation were not negligible, that the surface temperatures increased with the modified Rayleigh number, and that the nearby heat transfer was an effective method for studying free convective heat transfer. Chowdhury et al. [15] investigated the natural convection of a heated circular body encased in a triangle enclosure system. The results show that increasing heat flux and circular diameter with the same Ra (10^5) and Pr (0.71) increases the length of the stream lines.. Akeel et al. [16-21] studied theoretically different parameters affect the behavior of natural convection heat transfer and fluid flow in an enclosure. Some of these parameters were: porous media, corrugated shape of enclosure and inner cylinder, and nanofluid.

The object of this paper is to describe the numerical form of the natural laminar convective heat transfer in the concentric air-filled triangle around the horizontal cylinder. The impacts on fluid and heat transfer of the various inclination angles and cross-section geometries of the inner cylinder are also studied.

2. Problem Statement

2.1 Governing Equations and Boundary Conditions

Assuming constant thermo physical properties excluding density in the gravitational context. Neglecting the viscous heating effect and, where Bussinesq approximation is applied to the buoyancy effect model. The main transport equations that used in the present work are:

Continuity equation

$$\frac{\partial \rho}{\partial t} = -\nabla(\rho U) \quad \dots (1)$$

Momentum equation

$$\rho(U\nabla)U = \nabla\left(\mu(\nabla U + (\nabla U)^T)\right) + \rho g \quad \dots (2)$$

Energy equation

$$\rho C_p U.\nabla T = \nabla(-k\nabla T) \quad \dots (3)$$

The boundary conditions are :

-The outer surface temperature is constant and equals to 30 oC (cold temperature).

-The values of heat flux of the outer surface of the inner cylinder are 35.16, 58.47, 87.13, 114.76, 149.8, and 202.54 W/m^2

The average Nusselt number can be calculated by finding the heat transfer coefficient contour numerically. The local heat transfer coefficient (h) can be calculated by applying the following expression:

$$h = \frac{q}{(T_w - T)} \quad \dots (4)$$

Where q is constant heat flux, Tw is averaged cylindrical wall temperature and T is the local point temperature. The equation (5-1) can be modified to express the average Nusselt number:

$$Nu = \left(\frac{\bar{h}L}{K}\right) \quad \dots (5)$$

The averaged h is found by applying surface averaged has represented in the following expression:

$$\bar{h} = \frac{1}{A}\int_0^A h_A dA \quad \dots (6)$$

2.2. Numerical solution

The present study includes the solution of momentum and heat transport by linking and solving laminar fluid flow and heat transfer using COMSOL Multiphysics. The geometry is assumed to be 2D triangular with an inside evenly heated cylinder. The program also specifies the physical parameters of air (density, viscosity, thermal conductivity, heat capacity, and gamma). The geometry is adjusted for different angles in X-Y coordinates and aspect ratios. In this study, the coupled equations of heat transfer and momentum transport were utilized to characterize the behavior of natural convection by determining the distribution of stream lines, velocity profiles, and isothermal lines.

2.3. Solution procedure

The model builder in COMSOL Multiphysics is a tree system model. the COMSOL tree consists mainly of the following items:

-Geometry builder: by this item, the geometry of system is generated in which CAD or solid work interfaces are used.

-Material specification: this item is used for physical properties adjusted by using the materials from COMSOL data base or by creating a new blank properties as a new material.

- Physics selection: Through this item, the physics are picked to apply equations 1, 2, and 3 and their coupling or governing, as well as their coupling or governing. To get to the ultimate answer, physics needs the boundary condition.

-Mesh generator: In COMSOL, the mesh is generated automatically according to physics and geometry. The user can adjust the mesh size by selecting mesh mode either be normal, coarse, coarser, extremely coarse, fine, finer and extremely fine.

-Study: by this option, the model solving study is selected either be stationary (steady state) or time dependent (unsteady state. In present work investigation, the study is stationary.

2.4 Geometry and mesh generation

The geometries configurations and mesh contour for various α and rr are illustrated in Figure 1 & 2 The θ is considered 0 degree (horizontal position) because the CFD simulation is 2D study of triangular enclosure with inside heat source.

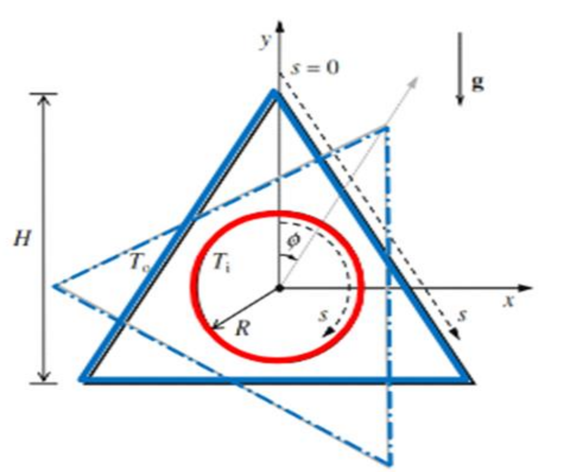


Figure (1): Geometry Configuration

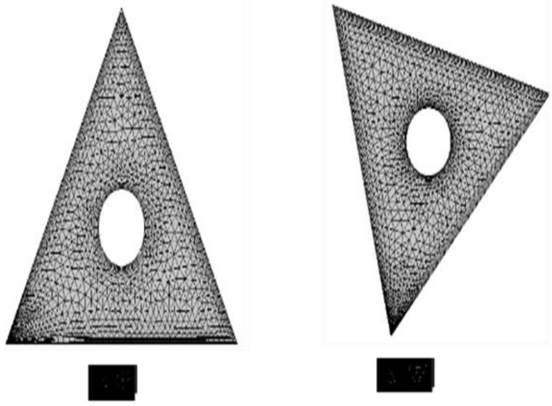


Figure (2): Mesh Contour

2.5 Physics and Multi-physics

In present work, the used physics are fluid flow and heat transfer in fluid. COMSOL uses non-isothermal Multi-physics to couple these physics. The solver builds the matrix of algebraic equations in every single mesh. The algebraic equation of heat transfer (convective heat transfer) depends upon momentum transport equation (fluid flow), the Multi-physics coupler or governor used to govern theses algebraic matrix then it will be solved together to obtain the contours. Gravity option is enabled in fluid flow physics with pressure points on cylinder surface to make solution convergence. For heat transfer case, the two boundary conditions are only required which is the outer surface temperature and the values of heat flux of the outer surface of the inner cylinder

3. Results and discussion

3.1. Stream lines Contour

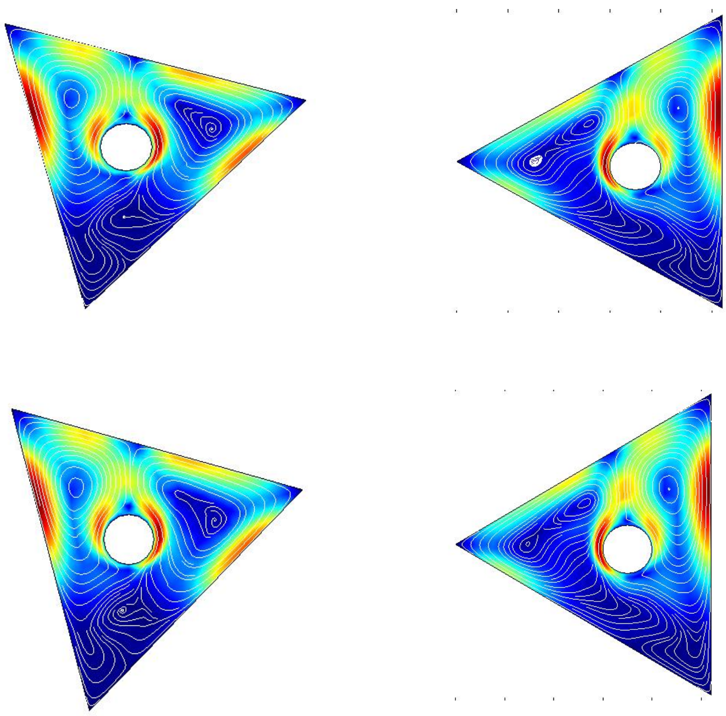
Fig.3 shows the stream lines contour of triangular enclosure of rr=0.345 (large enclosure) for various Ra and α. It seems that the velocity behavior increases by increasing the Ra, so the stream lines becomes more sluggish and less uniform behavior and vortices gets less circulated pattern. The α has significant effect on vortices, at 90o gives the higher range of velocity zones of free convection with higher range. Fig.4 shows the stream lines contour of triangular enclosure of rr=0.455 for various Ra's and α. The rr=0.455 has less sluggish vortices than rr=0.345. The hot air has continuous pattern to circulate upper the cylinder. The sluggish vortices start to appear at higher Ra and become maximum at higher (α). Fig. 5 shows the stream lines contour of triangular enclosure of rr=0.618 for various Ra and α. The vortices at rr=0.618 are less than that at rr=0.455. It seems that the vortices and stream line density in α=0 is more than other angles for various values of Ra. The boundary layers and vortices interactions between the cylinder side and cold side of enclosure wall are observed in higher value of Ra.

3.2. Isotherms contour

Fig. 6 shows the isothermal lines contour of triangular enclosure of rr=0.345 for various Ra andα. The temperature varies by gravity and the Ra of cylinder. In general, the isothermal lines become more sluggish in α=90o and higher Ra. In the larger enclosure (rr=0.345), there are significant changes in temperature at various α. Fig. 7 shows the isothermal lines contour of triangular enclosure of rr=0.455 for various Ra and α. The medium enclosure (rr=0.455) shows the behavior of temperature distribution inside the enclosure is affected by Ra. The thermal boundary layer seems to be larger in rr=0.455 as compare with rr=0.345 and decreases by increasing α. The larger range variation appears at lower α because the higher heat transfer rate appears at higher α and becomes maximum at 90o. Fig. 8 shows the isothermal lines contour of triangular enclosure of rr=0.618 for various Ra and α. The minimum heat transfer rate is observed at the small enclosure (rr=0.618). The isothermal temperature contour decreases by increasing the Ra and has irregular behavior with α angle because of the interaction between the various vortices in various parametric conditions. In CFD simulation; generally, it's obvious that the temperature behavior is related to velocity behavior by the action of momentum and heat transfer coupling.

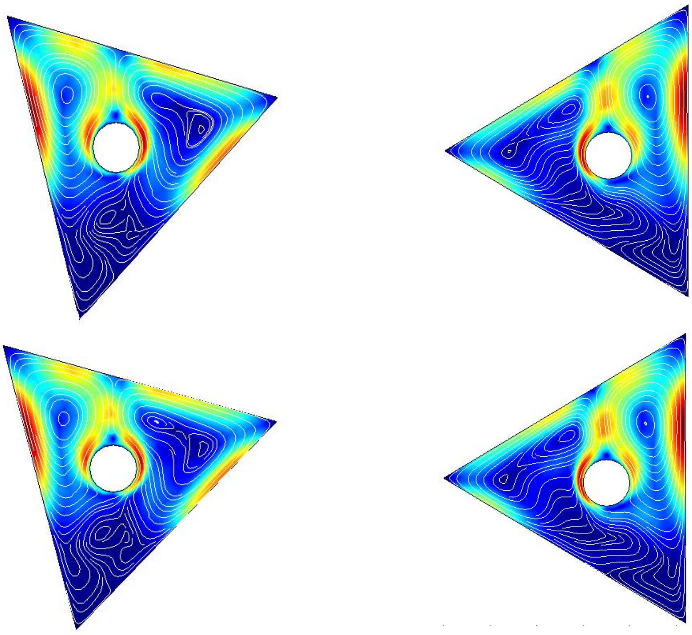
Ra=5.4x10^7

Ra=8.6x10^7



$Ra=1.6x10^8$

$Ra=2.8x10^8$



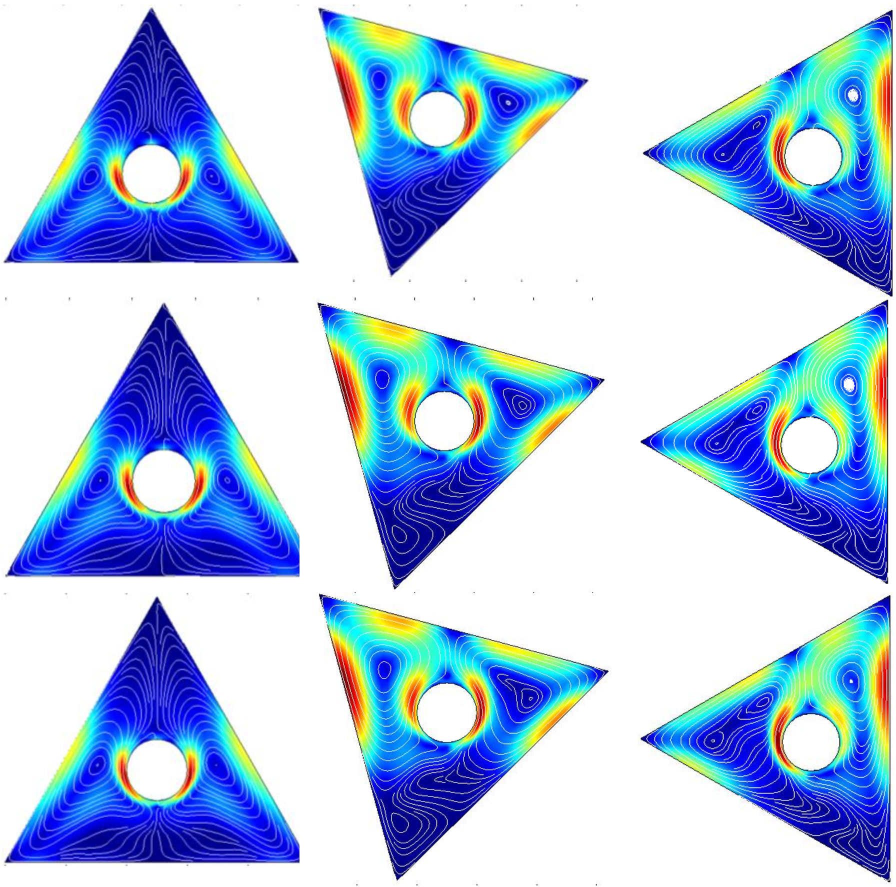
α=45° **α=90°**

Figure (3): Stream lines contour at rr = 0.345, α=45° and 90° with different Rayleigh numbers.

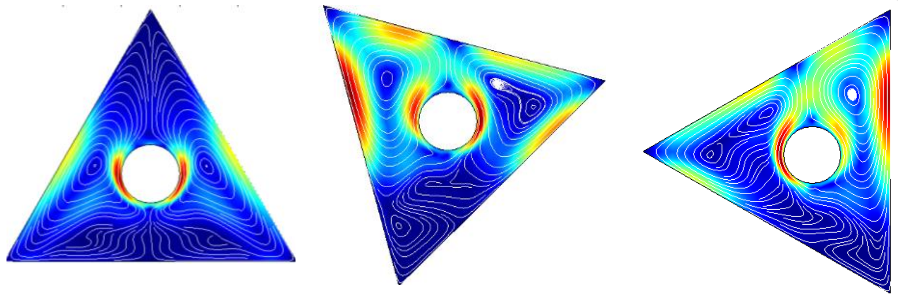
$Ra=1.7\times10^7$

$Ra=2.6\times10^7$

$Ra=5\times10^7$



Ra=9×10^{7}



α=0° **α=45°** **α=90°**

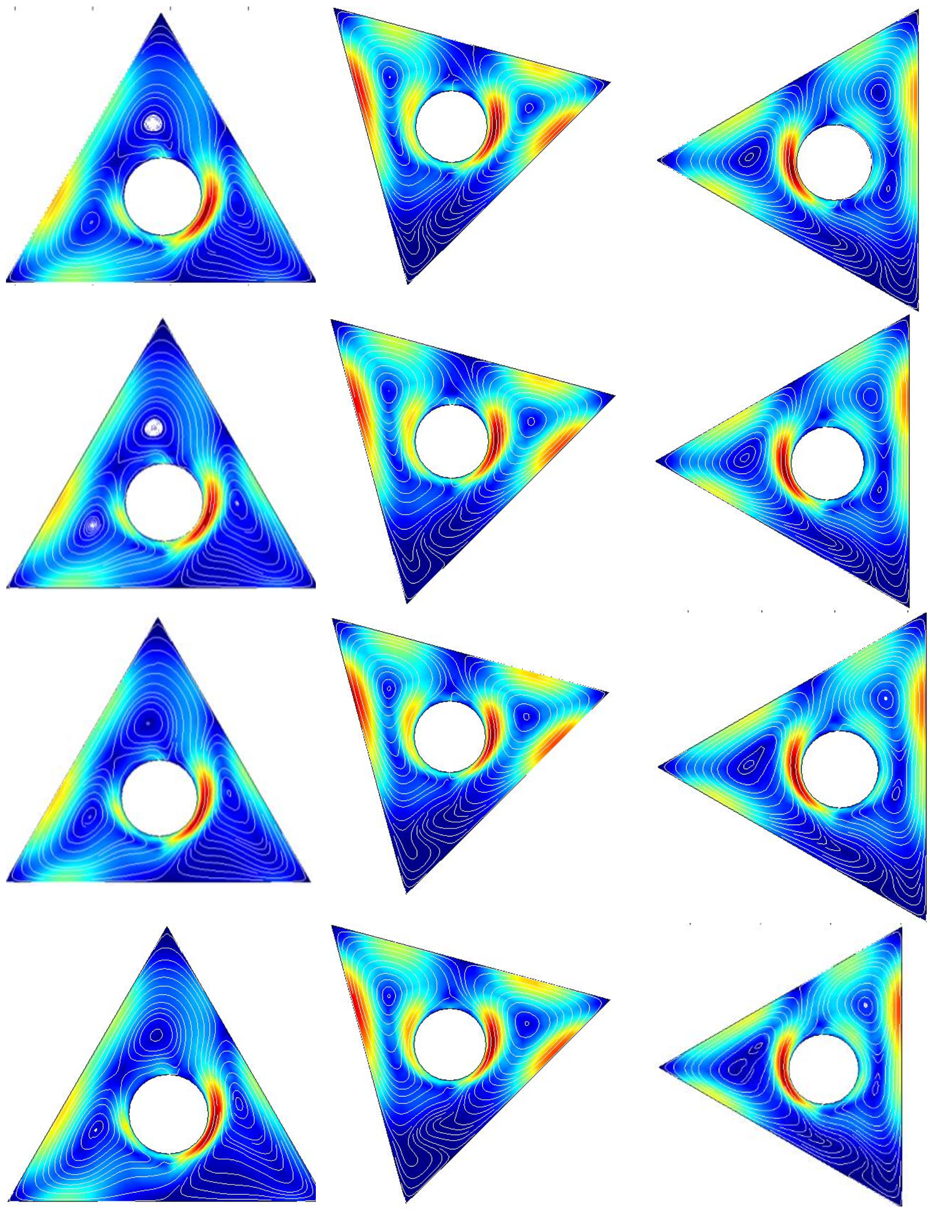
Figure (4): Stream lines contour atrr=0.455, α=0°, 45° and 90° with different Rayleigh numbers

Ra=4.3x10^{6}

Ra=7.2x10^{6}

Ra=1.37x10^{7}

Ra=3.4x10^{7}



α=0° **α=45°** **α=90°**

Figure (5): Stream lines contour at rr =0.618, α=0°, 45° and 90° with different Rayleigh numbers.

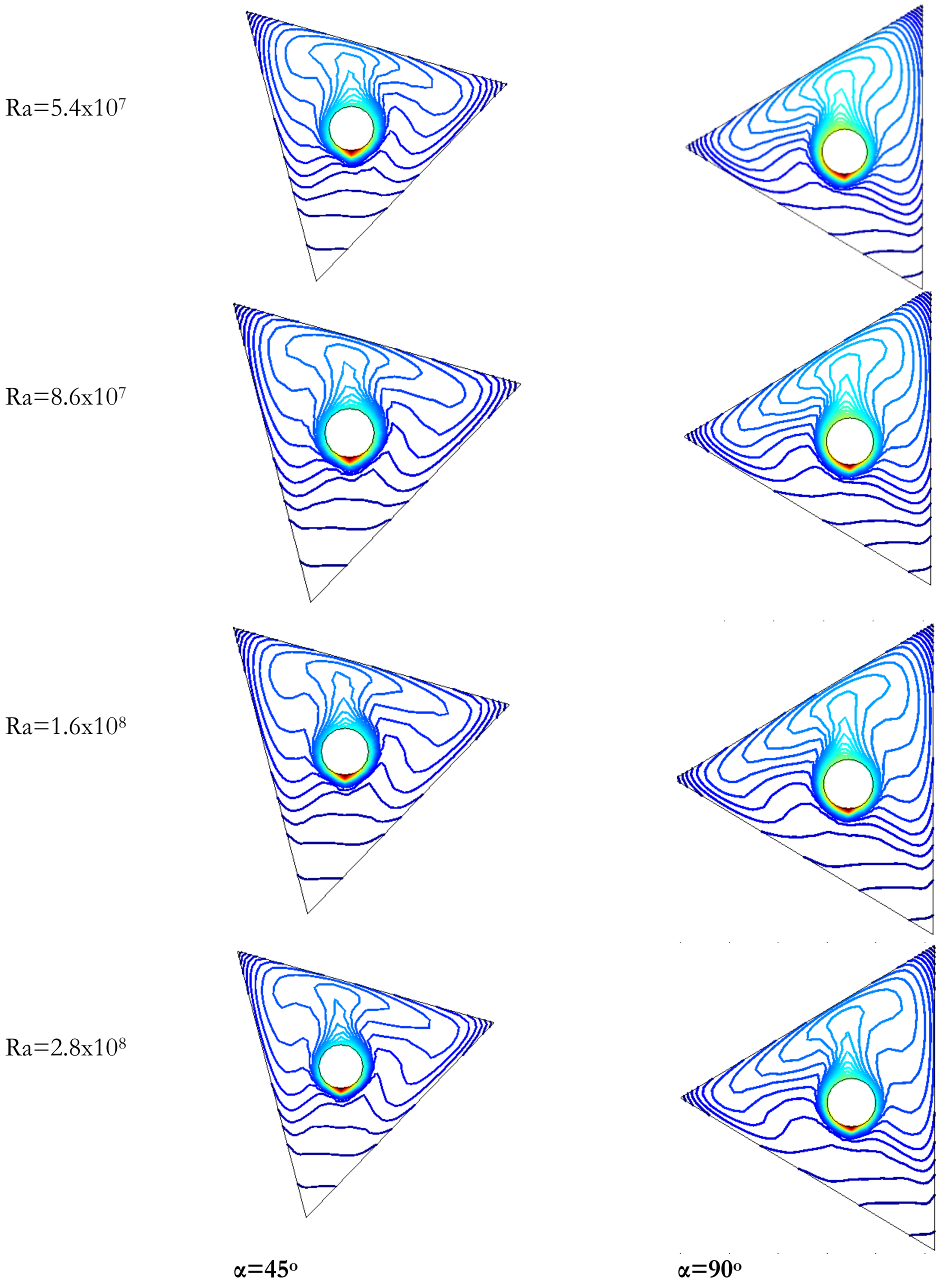
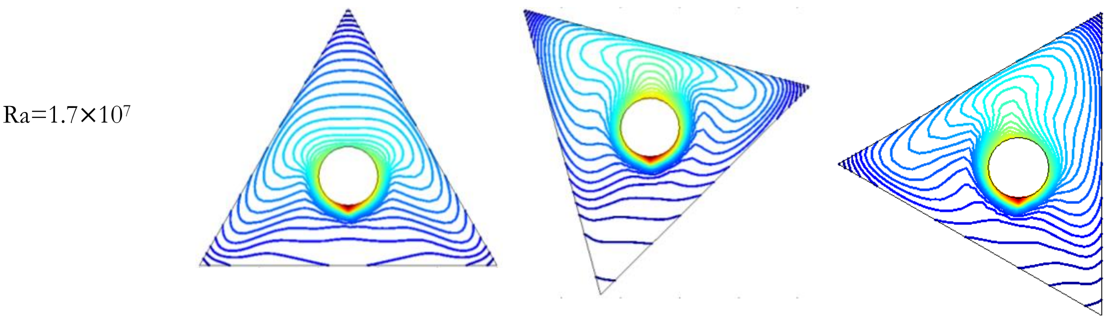
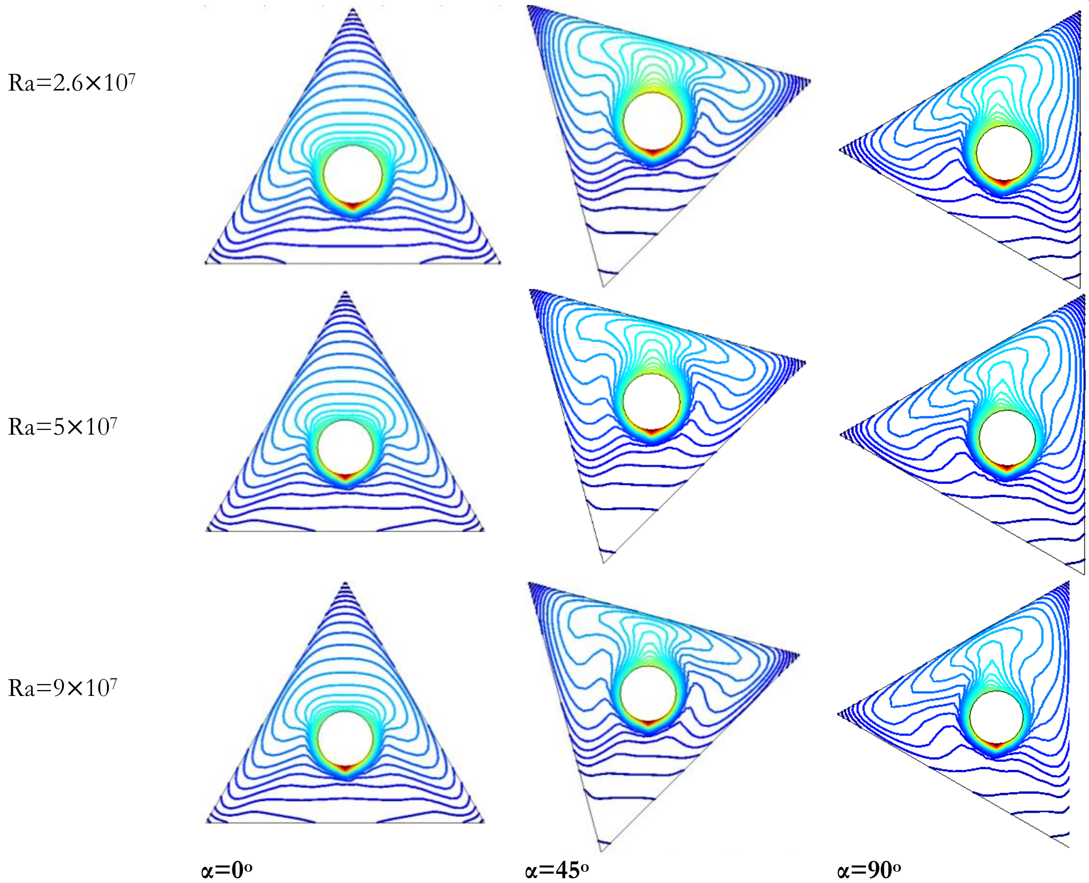
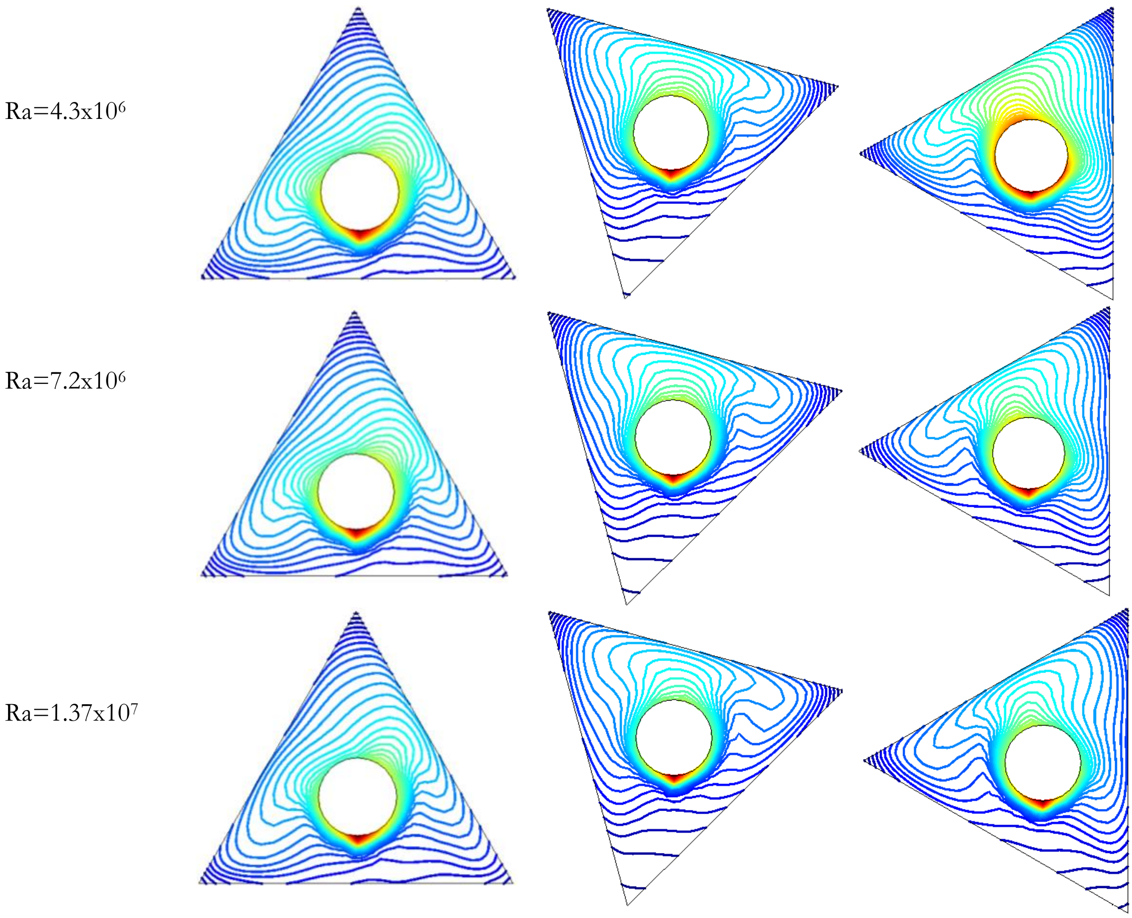


Figure (6): Isotherms contour at rr=0.345, α=45° and 90° with different Rayleigh numbers.

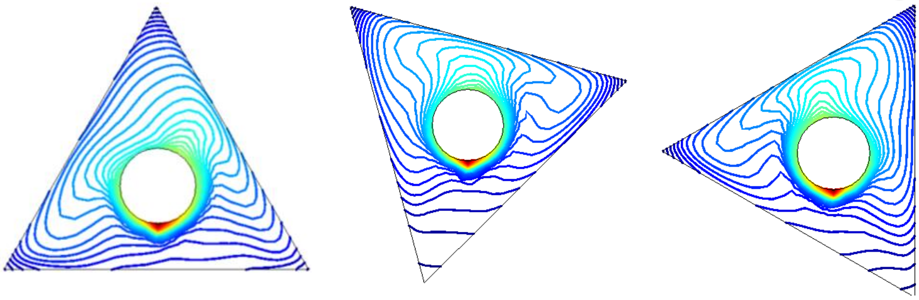




Figure(7): Isotherms contour at rr=0.455, α=45° and 90° with different Rayleigh numbers.



Ra=3.4x10^7



α=0° α=45° α=90°

Figure (8): Isotherms contour at rr=0.618, α=45° and 90° with different Rayleigh numbers.

3.3. The Correlations of Average Nusselt Number

Figures 9, 10, & 11 show the local heat transfer contours for various rr and α at θ=0o. The results show the h increases by Ra, rr and α increasing. In general, the maximum heat transfer coefficient region which is observed at cylinder region. The heat transfer coefficient has changed significantly by changing rr, α and Ra. Figures 12, 13, & 14 show the Logarithmic values of average Nusselt number versus the Logarithmic values of Rayleigh numbers for various angles of triangle orientationα at angle of inclination θ=0o (horizontal position) and rr=0.345, 0.455 & 0.618; respectively. The results show that there is a significant effect for α on Nu values. The correlations of average Nusselt number as a function of Rayleigh number for different values of triangle orientation angles and radius ratio is given in the following form:

$$Nu = cRa^m \qquad ...(7)$$

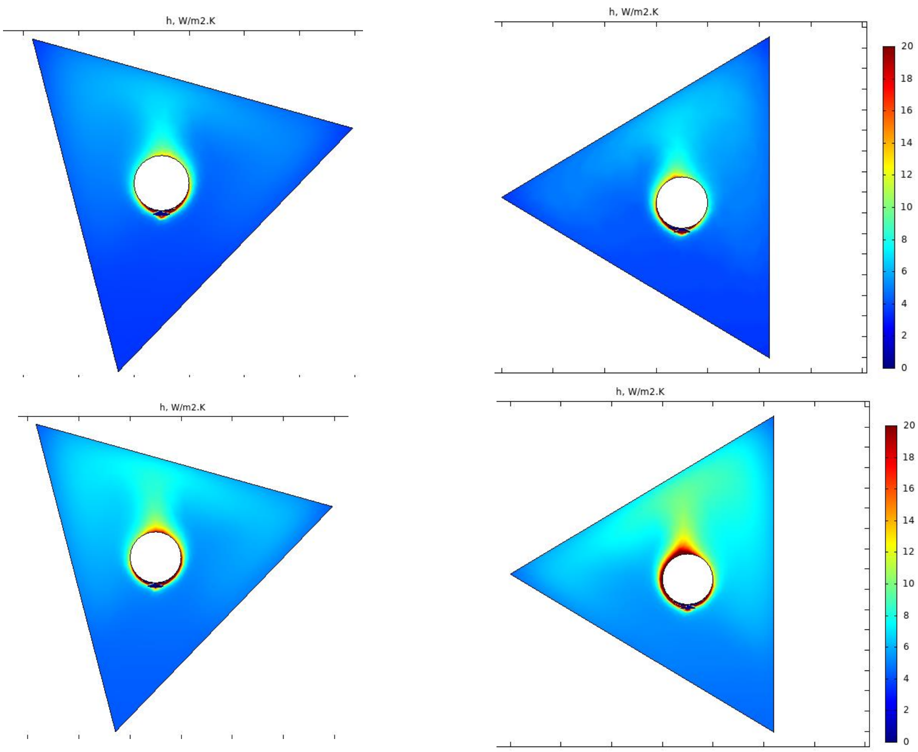
Where values of constants 'c' and 'm' are given in Table 1

Table (1): Constants' values of equation 5-4 for numerical work.

a	*rr*	*C*	*m*
0°	*0.455*	*0.327*	*0.145*
	0.618	*0.662*	*0.11*
45°	*0.345*	*-1.024*	*0.318*
	0.455	*0.37*	*0.145*
	0.618	*0.57*	*0.124*
90°	*0.345*	*-1.544*	*0.358*
	0.455	*0.899*	*0.083*
	0.618	*0.559*	*0.128*

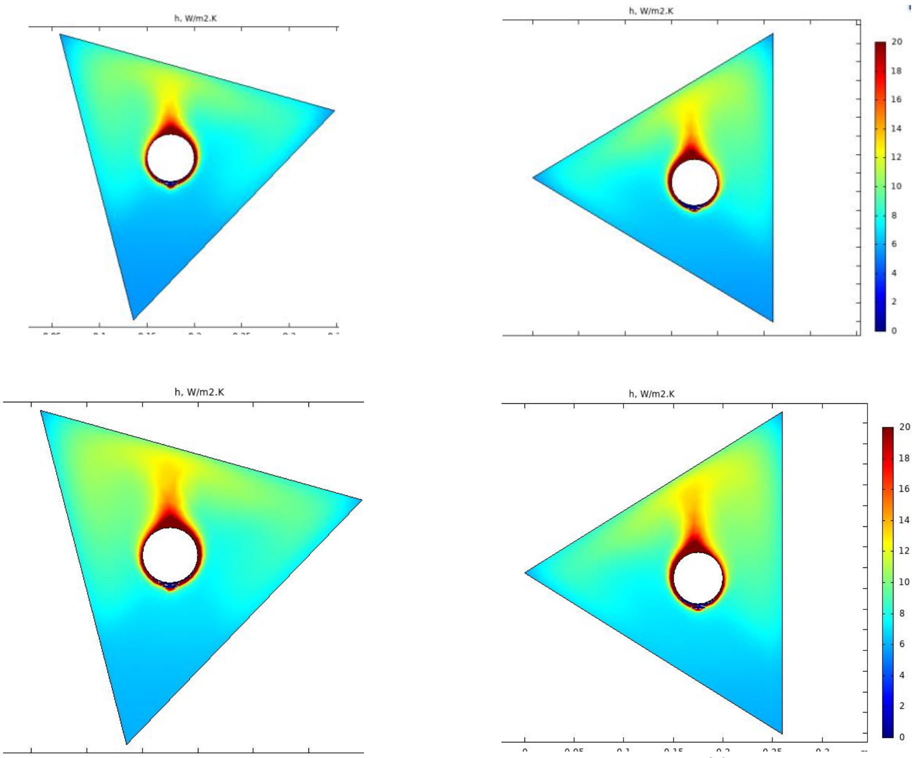
Ra=5.6x10^7

Ra=1.22x10^8



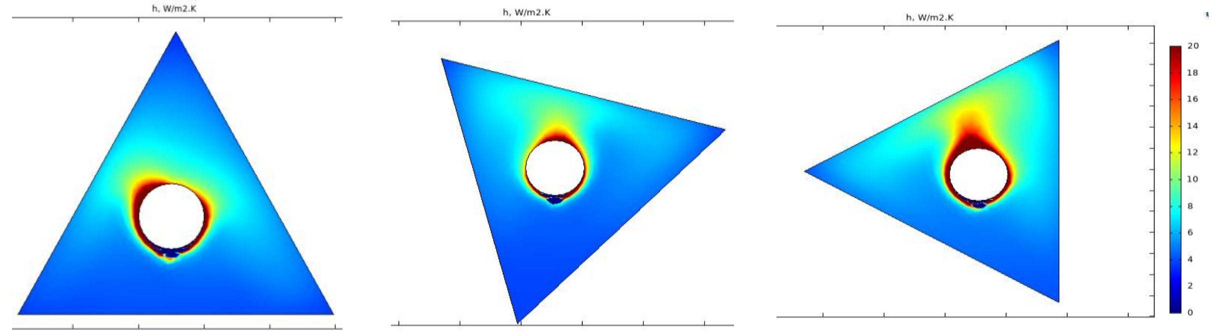
Ra=2.19x10^8

Ra=2.88x10^8

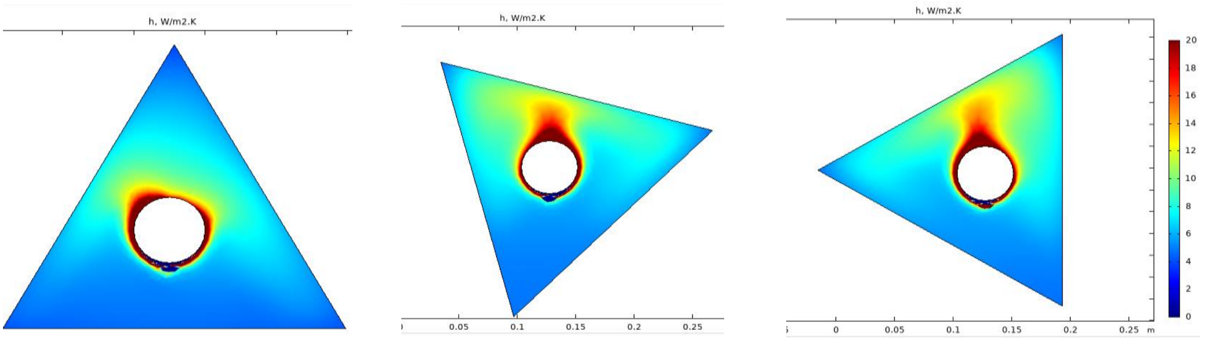


α=45°   α=90°

Figure (9): Isotherms contour at rr=0.345, α=0°, 45° and 90° with different Rayleigh numbers.



Ra=2.37x10^7



Ra=5.3x10^7

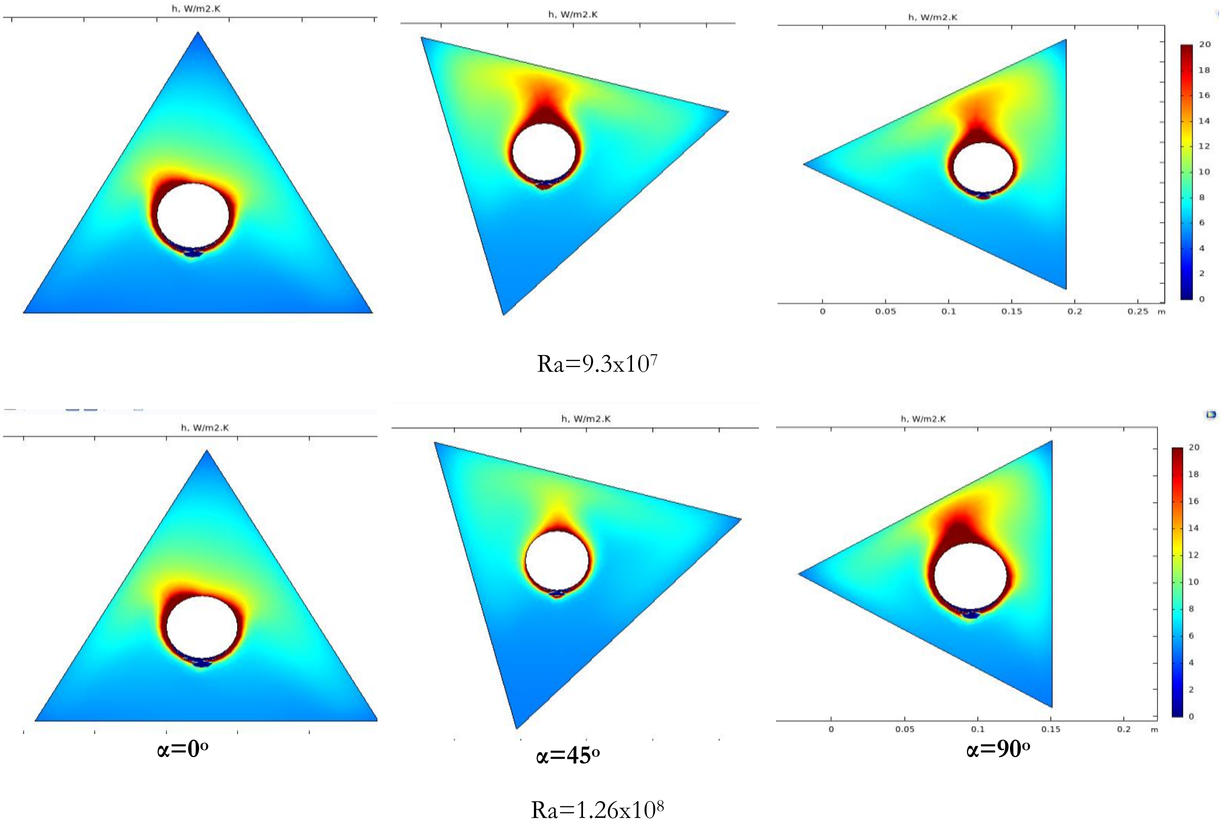
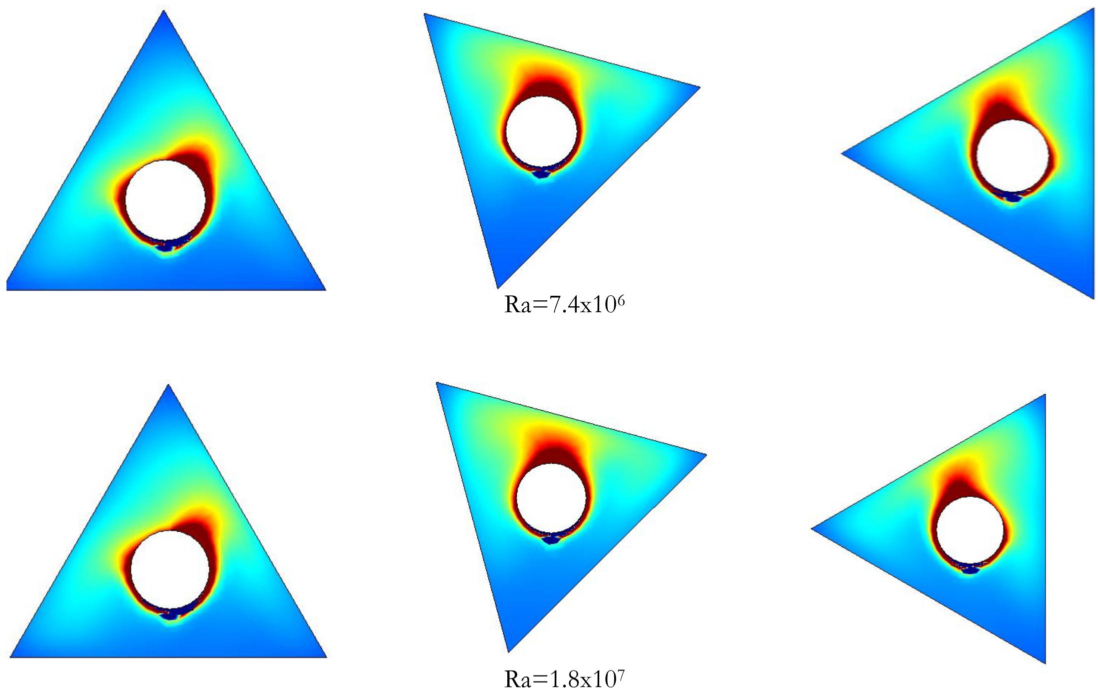


Figure (10): Isotherms contour at rr=0.455, α=0°, 45° and 90° with different Rayleigh numbers.



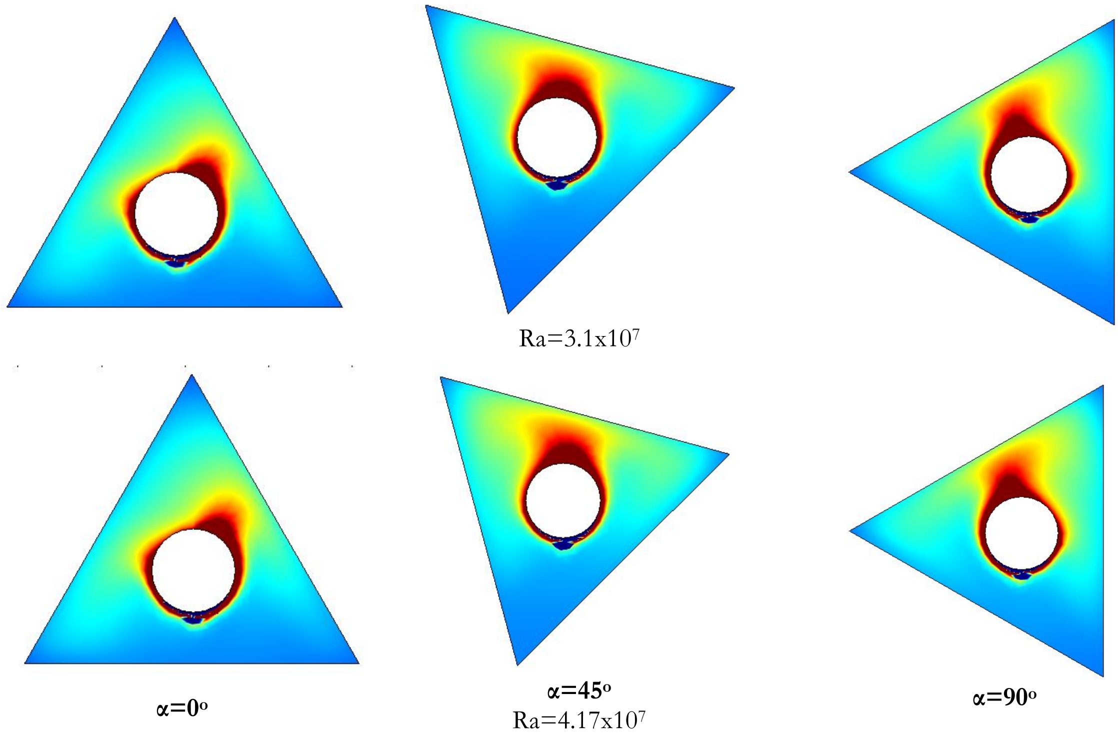


Figure (11): Isotherms contour at rr=0.618, α=0°, 45° and 90° with different Rayleigh numbers.

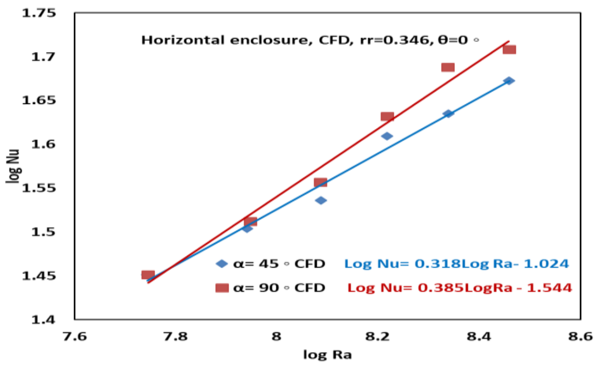


Figure (12): Logarithmic Nusselt number versus Logarithmic Rayleigh number for various α at θ=0° and rr=0.35, CFD simulation.

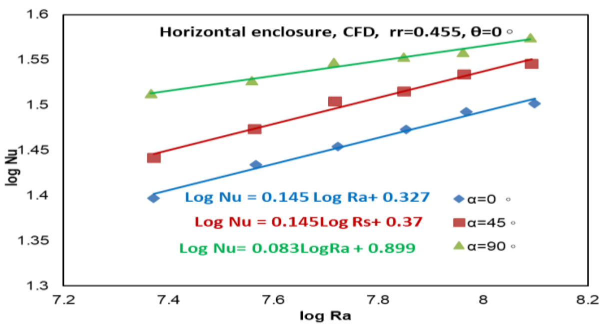


Figure (13): Logarithmic Nusselt number versus Logarithmic Rayleigh number for various α at θ=0 and rr=0.455, CFD simulation.

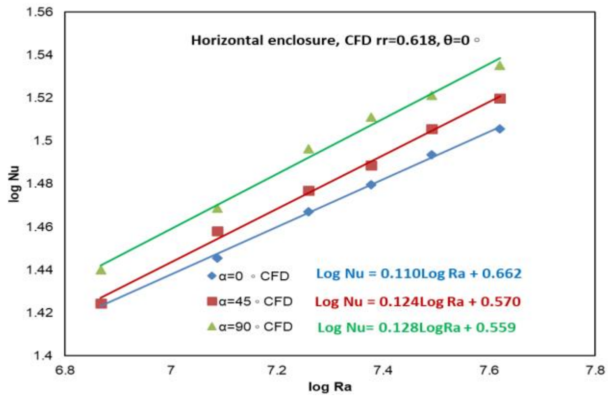


Figure (14): Logarithmic Nusselt number versus Logarithmic Rayleigh number for various α at θ=0° and rr=0.618, CFD simulation.

4. Conclusions

A computational study was conducted to ensure a steady natural laminar convective heat transfer from a heated cylinder to a condensed air-filled triangular enclosure. Effects on the flow and heat transfer of Rayleigh number, inclination angle and cross-section geometry of the internal cylinders have been studied. There are some conclusions on the basis of the numerical data discussed and the debate.:

1. Due to an increase in natural convection contributions, the flow strength and overall heat transfer increase dramatically as the Rayleigh number rises.
2. The symmetries of flow patterns will therefore be broken down at the highest Rayleigh number studied for lower rr ratios.
3. Turn the enclosure induces obvious changes in both thermal and flow fields. The flow rate is greatly determined by the inclination angle at the maximum

rr ratio, although the cumulative heat transfer is almost unchanged.

4. The flow patterns at lower rr rates are almost unimpacted by the geometry of the inner cylinder's cross-section and the influence increases with the increase in the rr ratio. However, the transversal geometry has negligible impacts on the heat transfer overall.

5. Eight correlations of average Nusselt number vs Rayleigh number were found for varied triangle orientation angles and radius ratios.

Nomenclature

A	Surface area (m^2)
Cp	Heat capacity (J/Kg. K)
g	gravity acceleration (m/s^2)
Gr	Grashof number
h	individual heat transfer coefficient ($W/m^2.K$)
L	characteristics length (m)
K	Thermal conductivity (W/m. K)
Nu	Nusselt number
q	heat flux (w/m^2)
T	Temperature (°C)
Ra	Rayleigh number
Ra*	modified Rayleigh number
rr	Aspect ratio
Pr	Prandtle number
U	Velocity components in x and y direction (m/s)
W	uncertainty

Greek letters

ν	kinematic viscosity (m^2/s)
β	thermal expansively (1/K)
α	radial angle of enclosure
ϱ	fluid density (kg/m^3)

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