Natural Convection Inside Square Enclosure Containing Equilateral Triangle with Different Orientations

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Abstract

Natural convection heat transfer phenomenon in a square enclosure containing equilateral triangle cylinder on its centroid is numerically investigated. Two dimensional for air (Pr=0.71) with outer cold walls of square enclosure and hot inner walls of the triangle cylinder by using CFD techniques is implemented. (Ansys Fluent 16) is used to simulate this problem. TheRayleigh number varied from 10^4 to 10^6 and orientation angles from 0^0 to 105^0 step 15^0 for each case. The laminar boussinesq approximation is used with the convergence criteria less than 10^{-5} for continuity, momentum and for energy equation 10^{-8}. Fluid and temperature are exhibited by stream and isothermal lines while heat transfer presented by the average Nusselt number. According to orientation angle, results can be divided into two groups; (0^o-45^o) and (60^o-105^o). Maximum heat transfer obtained at orientation angle of 30^0 and Ra=10^6.

Keywords: Natural Convection, Square Cavity, Equilateral Triangle with Different Orientation, Rayleigh Number, Stream Method.

الخلاصة

في هذا البحث تم اعداد دراسة نظرية لظاهرة انتقال الحرارة بالحمل الحر لتجويف مربع الشكل يحتوي داخله على اسطوانة ذات مقطع مثلث متساوي الاضلاع. تم اجراء تحليل ثنائي البعد باستخدام الهواء كمائع يملء الفراغ بين جدران التجويف الباردة و الاسطوانة المثلثة الساخنة . تم اجراء المحاكاة بأستخدام تقنية CFD بواسطة برنامج (Ansys 16) Fluent لمعادلات الاستمرارية والزخم والطاقة وكان مقدار عدد برانتل (Pr=0.71). خلال البحث تم تغيير عدد رايلي من 10^4 الى 10^6 و كانت قيم زاوية تدوير الاسطوانة الداخلية تتغيير من 0^o الى 105^o بمقدار 15^o لكل خطوة تدوير. تم استخدام تقريب (boussinesq) الطباقي مع معيار تقارب اقل 10^{-5} لمعادلة الاستمرارية و 10^{-8} لمعادلتي حفظ الزخم والطاقة. تم بيان خواص المائع و درجة حرارته بواسطة خطوط دالة الانسياب و خطوط درجات الحرارة الكنتورية بينما تم تمثيل انتقال الحرارة عن طريق معدل عدد نسلت. يمكن تقسيم النتائج التي تم الحصول عليها حسب زاوية التدوير الى مجموعتين (0^o–45^o) و(60^o–105^o). اظهرت النتائج ان اقصى انتقال للحرارة يكون عند زاوية توجيه 30^o و عدد رايلي مقداره 10^6.بسب زيادة تاثير انتقال الحرارة بالحمل.

الكلمات المفتاحية: الحمل الحر، تجويف مربع، مثلث متساوي الاضلاع بزوايا توجيه مختلفة، عدد رايلي، طريقة الانسياب.

1- Introduction:

The steady state enhancement and depression of convection heat transfer characteristics in a differentially heated enclosures or cavities has long been researchers and has received an attention due to it has a wide relevancy to engineering applications such as heat exchangers, solar collector design , heating and ventilation of building, nuclear , chemical reactors and cooling of electronic equipment. Many studies have been devoted to solving the problems of the natural convection in enclosures containing commonly geometries encountered in many fields of industrials by different configurations (i.e) (flat and corrugated) and locations, likely (squares, rectangles and rhombic…etc). However, some experimentally and numerical techniques of investigators was concerned of enhancing heat transfer in a square enclosure with triangular**(Altac *et.al.*, 2009),** have considered the problem of natural convection in rectangular enclosure having a thin horizontal isothermal plate and air has been assumed as the working fluid. The results were obtained for the thin plate and the Rayleigh number ranging from (10^5) to ($5*10^7$) show the rate of the heat transfer is increases when the Rayleigh number increased. It was obvious from the results that the increasing plate length the Nusselt number reduced by 25%.The natural convection heat transfer around a tilted heated square cylinder placed in an enclosure was investigated by (**Arnab and Amaresh 2006).** These authors proved that the Stream-function vorticity calculations of the set of governing Navier-stockes equations can be solved by using a finite difference technique in non-orthogonal body-fitted coordinate system .(**Costa *et.al.*,2003)** in their work presented the problem of natural convection in a square shape enclosure with internal solid body inserts located at the corner position, to control the heat transfer within the enclosure. The result expected that the appropriately selected of inserts bodies leading to a significant effect in thermal characteristics of the enclosure and was found to be its influence as heat transfer enhancers are used as insulators. **(Roslan *et.al.*, 2015)** analyzed numerical using (COMSOL Multiphysics software) the conjugate natural convection heat transfer in a differentially heated square enclosure with a conducting polygon .The enclosure left wall was kept heated and the right wall was assumed cooled while the remaining horizontal walls were kept thermally insulated. The Rayleigh number varying range was ($10^3 \leq Ra \leq 10^6$). They concluded that the heat transfer rate increases with the increase the size of the solid polygon, until it reaches its maximum value. In addition, the size of the solid polygon reaches its critical value further, beyond this critical size of the solid polygon, will decrease the rate of heat transfer. They concluded that the effect of by partition caused by decreases Nusselt number by (12%). A combined forced and natural convection heat transfer of air assumed at low speed flow across cylinders located horizontally engrossed in Newtonian fluids studied by (**Hatton *et.al.*, 1970;Hojat and Seyed 2012)** used with numerical finite volume technique to study concerned with two-dimensional solution of natural convection for two different geometries (i.e. circular and square). It was found from the results of total surfaces-averaged Nusselt numbers, in all cases, the rate of heat transfer from the enclosure for a circular cylinder located inside is better for the same Rayleigh number.A numerical studied were presented by (**Karayiannis *et.al.*, 1992)** for two cases of convective heat transfer of air in rectangular cavities. The first case estimated without partition, while second arrangement the vertical partition was fitted and the opposite walls assumed at different constant temperatures. The study has been conducted at a different aspect ratio (i.e (0.1) to (16)) and Rayleigh number ($3.5*10^3 \leq Ra \leq 3.5*10^7$) case one and ($10^5 \leq Ra \leq 1.6*10^8$) for the other case.(**Habibis *et.al.*, 2014)** studied numerical simulation of natural convection in a polygonal enclosure that contains circular cylinder. The governing equations were solved using the built-in finite element method of (COMSOL). The influence of the number of polygonal sides, aspect ratio, radiation parameter, and Rayleigh number on the resulting natural convection problem was investigated. It was found that the number of contra-rotative cells depended on polygonal shapes. Also, results illustrated constant convection heat transfer at (L/D=0.77, i.e (L=gap between inner cylinder and outer polygonal; D= diameter of inner cylinder) and no effected of the polygonal configuration on profile of Nusselt number.(**Xing *et.al.*, 2015)** investigated numerically the natural convection heat transfer for horizontal concentric annuli with cylindrical cross section cylinder and elliptical and square and triangular cross sections, respectively, the inner and outer surfaces were maintained at a fixed temperature. Numerical results were presented for fluid properties reported in table (1). The profiles of flow velocity and thermal fields were presented and discussed by means of streamlines and isotherms .In this paper, the local Nusselt number included both convection and radiation contributed (i.e $Nu=Nu_c + Nu_r$) where Nu_c and Nu_r are local convective and radiative Nusselt numbers, respectively. The surface radiation and presence of corners as well as a larger top space of the respective systems shows the rate of heat transfer significant enhancement, also the increases in reference temperature, and the surface radiation is clearly observed act in the overall heat transfer performance.

T_0 [K]	P_0 [atm]	$\alpha*10^{-6}$[kg/(m . s)]	$\mu*10^{-6}$ [kg/(m . s)]	ρ[kg/m 3]	$\beta*10^3$[1/K]
293	1	21.4	18.1	1.205	3.41
573	1	71.6	29.7	0.615	1.75
1273	1	245.9	49.0	0.277	0.79

Table 1. Reference fluid properties ref .(Xing Y.*et.al.*, 2015)

(Xu *et.al.*,2012), performed a numerically by employing control volume method the transient natural convective heat transfer of liquid gallium as a working fluid dictated a coaxial triangular enclosure containing a horizontal heated circular cylinder. The results were presented with two tilted of the triangular cylinder in this study .(**Xu, 2010)**, have investigated the effect of different inclination angles of the enclosure and various cross-section geometries of steady-state laminar natural convective heat transfer around a horizontal cylinder to its concentric triangular. The results shown that at constant aspect ratio, both the oblique angle and cross-section geometry have insignificant effects on the overall heat transfer rates though the flow patterns are significantly modified.

The aim of the present work is to study the problem of natural convection of flow and heat transfer inside a square enclosure and its wall kept at a constant cold temperature T_c and filled by an air (Pr= 0.71) with a heated equilateral triangle, kept at a constant hot temperature T_h and located at the center line of the enclosure. The study is performed for a Rayleigh number varied range ($10^4 \leq Ra \leq 10^6$) and by changing the angles of orientations, which are changed between 0° to 105°.

2- Physical Model and Coordinate System:

The representation of the problem having outer cold walls of square enclosure and hot inner walls of equilateral triangle cylinder as in fig.(1). The equilateral triangle cylinder is placed in the center of enclosure and changes its orientation with clockwise direction in step 15^0 until 105^0, for more degree of orientation the triangle becomes repeated its shape. Air occupied the enclosure (Pr=0.71), difference in temperature between inner and outer walls with gravitational effect produces natural convection phenomenon. The Boussinesq approximation approach is used to calculate the variation in density due to difference in temperature. The coldest temperature is kept at 300K, all properties of air are taken in table **(Hoja and Seyed 2012).** The length of square enclosure is taken as (H=0.1m) while the length of equilateral triangle cylinder taken as (L=0.2H). The hot temperature (T_h) is calculated from the equation below.

$$"T_h = \frac{Ra*\alpha*\nu}{g*\beta*H^3} + T_c \ " \tag{1}$$

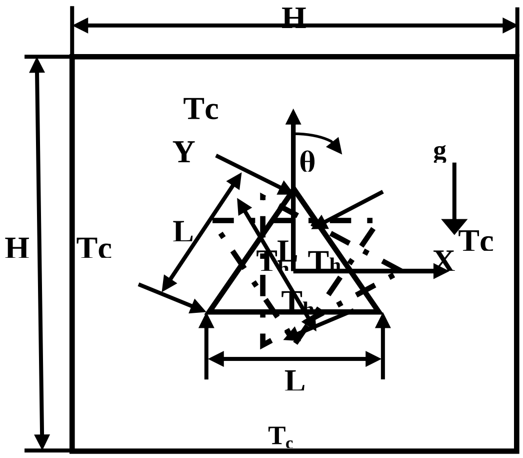


Table. 2 properties of air at 300K	
Variable	Units
ρ	1.1614 [kg/m^3]
C_p	1007 [J/kg.k)
μ	1.846*10^{-5} [N.s/m^2]
ν	1.589*10^{-5} [m^2/s]
K	2.63*10^{-2} [W/m.k]
α	2.25*10^{-5} [m^2/s]

Fig.(1) schematic diagram of physical domain with different orientation.

3- Governing Equations:

Continuity Equation

$$"\frac{\partial(\rho U)}{\partial X}+\frac{\partial(\rho V)}{\partial Y}=0" \quad (2)$$

Momentum Equations:

$$"\frac{\partial(\rho U^2)}{\partial X}+\frac{\partial(\rho UV)}{\partial Y}=-\frac{\partial P}{\partial X}+\frac{\partial}{\partial X}\left(\mu\frac{\partial U}{\partial X}\right)+\frac{\partial}{\partial Y}(\mu\frac{\partial U}{\partial Y})" \quad (3)$$

$$"\frac{\partial(\rho UV)}{\partial X}+\frac{\partial(\rho V^2)}{\partial Y}=\rho g_y-\frac{\partial P}{\partial Y}+\frac{\partial}{\partial X}\left(\mu\frac{\partial V}{\partial X}\right)+\frac{\partial}{\partial Y}(\mu\frac{\partial V}{\partial Y})" \quad (4)$$

Energy Equation:

$$"\frac{\partial}{\partial X}\left(\rho UC_pT\right)+\frac{\partial}{\partial Y}\left(\rho VC_pT\right)=\frac{\partial}{\partial X}\left(k\frac{\partial T}{\partial X}\right)+\frac{\partial}{\partial Y}(k\frac{\partial T}{\partial Y})" \quad (5)$$

Where **U** means velocity in x-direction while **V** means velocity in y-direction, **P** represent pressure, **T** represent temperature and thermo-physical properties are ρ: density, μ: viscosity, C_p: specific heat, α: thermal deffusivity and k: thermal conductivity. The above equations are solved numerically by using (Ansys Fluent) which used finite element technigues. The dimensionless numbers Rayliegh and Prandtl are defined as

$$"Ra=\frac{g\beta(T_h-T_c)H^3}{\alpha\vartheta}, Pr=\frac{\vartheta}{\alpha}" \quad (6)$$

The physical dynamic and thermal boundary conditions for the present problem are assummed as follows: At outer and inner walls **U=V=0** **(7)**

At outer walls **$T=T_c$=300K**, inner walls **$T=T_h$** are computed from eq.(1) **(8)**

The heat transfer is computed by local and average Nusselt number around triangle cylinder as below:

$$"Nu=\frac{\partial T}{\partial n}\Big|_{inner\ wall}, \overline{Nu}=\frac{1}{l}\int_0^l Nu\,dl" \quad (9)$$

Where ***n*** is the normal direction reference to the wall and ***l*** is the length of the inner wall.

4- Numerical mode ling and Verification:

Figure 2a-b represents the mesh generation and grid arrangement in the computational domain. Two dimensional quadratic elements in finite element approach have been achieved. A smooth grid is required to obtain a good solution with 10970 elements and 11448 nodes after four different grid generation test as shown in table 3 , the results are obtained at Ra=10^5 and θ=0^o.

Steady laminar natural convection flow was considered in this study. The setup of programming is the simplest method for pressure-velocity coupling with least squares cell based for gradient, second order for pressure and second order upwind for both momentum and energy.

A comparison with another study for two Rayleigh number was achieved to perform validation for Ansys Workbench Fluent are shown in figure (3). The validation appears good congruity with **(Hojatand Seyed 2012)**

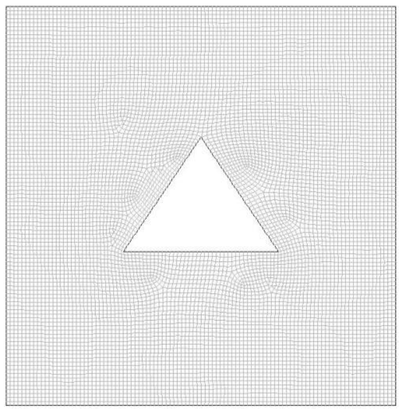


Fig.2a. Mesh Generation

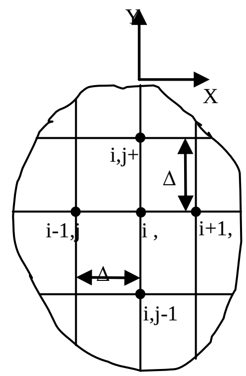


Fig.2b. Grid arrangement

Table3: Grid validation				
No. of elements	8810	8841	9608	10970
Average Nusselt number	7.152	7.402	7.9	7.9039

5- Results and Discussion

5-1 Flow structure and Temperature Fields

The figures (4,5,6, and 7) were prepared directed from left to right for different inclination angles of equilateral heated triangle cylinder and going down with development Rayleigh number. The results are happened when the heated triangle is tuned at different orientation angles and ascending Rayleigh number. Because the temperature difference between the hot(inner) and outer(cold) walls the bouncy effects will by appear , It is observed that recirculating vortices are formed in the cavity which are exactly displayed by the closed streamlines, these vortices can be arranged into two parts, first part when the triangle tuned angle from (**θ**=0^o to **θ**=45^o) and the second group when (**θ**=60^o to **θ**=105^o).As manifestation in figure (4 A,B,C), the heated fluid moves up a middle the hot inner triangle wall and the vertical correspondence line until it impinge to the cold enclosure wall. Also, the air becomes moderately cold and the fluid density increasers while it motion goes horizontally in touch with the cold enclosure wall. Therefore, the denser air comes down besides the vertical boundary of cold wall. When the triangle tuned angle(θ=0^o), and corresponding Rayleigh numbers(10^4) the binary symmetric eddies almost equal strength arrangement are clearly demonstrated with two inner vortices into the left and right and the eddies movement with opposite path of motion ,because the conduction mode was dominated in the enclosure regime at lower Rayleigh number . As exhibited in the figure (4) with an increase of Rayleigh number from (10^5 to 10^6) the stream lines change significantly, which indicates the increases convection intensity and the circulating slightly pulled up further, the core region of eddies shapes , which are appearing as an ellipse , that evidence of a advection is the dominating on the rate of heat transfer mechanism in the enclosure. At (**θ**=15^0 ,45^0) and (Ra =10^4) the size between the inner heated triangle and vertical boundary increasing from left and diminishes from right and the streamlines becomes to a greater extent distorted until the left eddy stated to separate into two dissimilar vortex ,and symmetric when oblique angle (**θ**=30^0) , because the closed zone around the triangle is increases and becomes symmetric in left side of enclosure and thus generated enough space then the two identify vortices will be circulated .In addition, a single vortex in left eddy moving downward. The figure (4, C) explained the contours plots for different variation of inclination angles (θ=15 ^{o},30^o , 45^o)and (Ra=10^6) . It is noticed at high Rayleigh number the two cells filled the enclosure and surrounds the cylinder. This arrangement manifested since the boundary conditions and angles of orientations . As expected when (θ=15^o) the right cell seems large in size and penetrates, while the size of the right cell becomes small , when (θ =30^o and 45^o) as mention the space between the heated triangle and vertical boundary increasing from left and diminishes from right thus, the left cell appears as more dominated and the right cell squeeze and expanded besides the cold wall ,due to the rises in the Rayleigh number the convection effect in heat transfer becomes stronger also, the configuration of the thermal boundary layer clearly demonstrated thinner on the inner surface of the tilted body. Figure (5) shows the inversion position of the triangle with respect to its originally state (i.e θ=60^o). The flow plots are presented here with the same arrangement as in figure (4). When (**θ** = 60°and Ra = 10^4) ,the streamlines recirculation seems as the dual boundary layers merge with opposite direction in the lower half of the enclosure, forming the kidney-designed core or cellular structure ,due to dominance of the conduction mechanism on the convection heat transfer. As inclination angle increases the streamlines converts to asymmetric and distorted also, the space between the right side of the enclosure and trapped cylinder walls are becomes bigger, hence the right eddies separated in two cells. Generally, the influence of the triangle rotation squeezed the streamlines between the enclosure walls and the and triangle sides created a dense velocity boundary layer close to triangle. It is observed that at (**θ**=60^o,Ra= 10^5 and 10^6) , four vortices exists , the two small newest similar cells formulation at the top surface of the triangle, except for the sense of cycle however ,increase (Ra) and heated triangle position prevent the air to appears as two facing cells and compels to formulation the four cells. With (**θ**=75^0) the triangle position will be change and the size of enclosure in the right side becomes greater than in left side as shown in figure, hence the newest two cells disappear and the flow has enough space to consist of two eddies (with one cell in left ,and with two cell in the right) since, the ascending of the skew angle of triangle compress the stream lines between its sides and enclosure walls and forced shifted upward the cell in the left which appear later as squeezed thinner. When (**θ** =105^0) two larger rotation cells exist and occupied the enclosure at the upper of the heated triangle, the two rising circulating vortices stretched, where the top of the enclosure noted the presence increases thermal gradient there .In other word ,as (Ra=10^6) the main flow down the triangle come to be weak, and flow up become more strongly also, the flow contours marked a thick boundary layer furthermore, clearly seen increases of cells motion, due to the influence of convection mode. Also, the streamline will appear irregular shapes, mainly in the upper half region that is clearly appear, when (Ra = 10^6).The comparisons between isotherms for these four different tilted angles are shown in figures (6,7) for ($0^o \leq \theta \leq 45^o$, and $60^o \leq \theta \leq 105^o$). In general temperature decreases from hot triangle to cold walls of enclosure, where the perimeter of the hot triangle illustrated the presence of a strong temperature gradient there, at low (Ra) number the triangular inner shape shows a large distortion and denser as expected, at the triangle corners due to the presence of the sharp edges and decreases with increaser in (Ra) .With ($\theta=0^o$,$Ra=10^4$) the temperature fields are symmetrical about the hot triangle due to pure conduction subsequently the strength of the buoyancy-driven convection is actually unaffected. As rotations angle of the equilateral triangle ascending the hot fluid zone and the cold zone remain an approximately equal strength, also the isothermal curves move with hot body rotation. The isothermal contour when ($\theta=60^o$,$Ra=10^4$) as in figure (7,A), take the same behavior as in figure (6,A) but this habit depends on the size of space between the triangle and enclosure. With increasing of the (Ra) number (i.e($\theta=0^0$,$Ra=10^5$ and 10^6)) as indicated in figure (6,B and C) the isothermal pattern characteristics displayed earlier symmetrically, and cold zone magnitude smaller and becomes more thinner near the top wall as the Rayleigh number goes higher (i.e $Ra=10^6$).In addition, the thermal plumes are starts to appears upwelling enriched vertically and impacted to the upper wall, however the development of thermal plumes come to be more concentrated form ($Ra = 10^6$). Generally, the distorted isotherms and thinner boundary layers resulting in strongly dominated convection heat transfer. As skew angles increase the pattern of isotherms at the upper partial of the enclosure are slightly distorted, due to the expansion in the gap above the upper of the putted inner cylinder. When ($\theta =60^o$, $Ra=10^5$,and10^6) as proved in figures (7 ,B and C) thermal rating reflect the effect of increase in the Rayleigh number, as predicted the convection mode becomes stronger, and the contours of temperature show more denser near the top cold wall also, the isotherms beside the heated cylinder separate away starting from its top region and formation impinging a thermal plume at upper of the enclosure. This behavior frequent itself here as discuss above in figure (6 B and C) with different shapes if compare with ($\theta =60^o$,$Ra=10^5$ and 10^6) ,due to the narrow space between the triangle upper two edges and enclosure and opposite direction If compared at angle above ($\theta =60^o$, $Ra=10^5$,10^6). It is worth noting in the current study, when ($Ra=10^6$) thermal plumes becomes more concentrated and is gradually elongate with increasing tilted angle. Clearly , as Rayleigh number increases, the diffusion of heat transfer is increased; hence the distorted and crowded temperature fields approach the half down wall decrease, which illustrated that the act of conduction heat transfer vanish, and the behavior change totally to convection.

5-2 Heat Transfer and Nusselt Number

Figure(8) shows the distribution of Variation of average Nusselt number around hot equilateral triangle with different tilted angles ,were arranged going up for ($Ra=10^6$, 10^5 and 10^4). As noticed in the figure, the situation of average Nusselt number is almost trend as wave form . For each curve when compared with each other, clearly understood that, Nusselt number has decreased when skew angles change from (0^0) to (15^0) and the wave becomes concave at ($Ra=10^4$) ,while Nu has a local minimum value ($Nu=4.58$) . Again Nusselt number increases dramatically until peak point with increasing of tilted angle (from15^0 to 30^0) for ($Ra=10^4$,10^5, and10^6) as expected, due to the increases of area of heat transfer. After that, the profile seems nearly flat when ($Ra=10^4$). When Rayleigh number increases (i.e) ($Ra=10^5$) the wavy curve is apparently not contribution altitude peaks, the resulting middle peak is lowest that can be seen from the shape , and reduced in local heat transfer between the fluid and the skew body due to separation of the thermal layer .If focus in figure concluded another maximum point confined between(90^0-105^0). In general we note, for all cases, as the Rayleigh number is ascending increased, the average Nusselt numbers are significantly increased for different orientations angles of heated triangle .At ($Ra=10^6$) the average Nusselt number appears by a different behavior, the figure show (Nu) decreasing at (60^0) and has minimum rate then increasing the by same intensity also, it has approximately three times greater than that at ($Ra=10^4$), due to the intensity of convection dominated on regime (i.e., $10^5 \leq Ra \leq 10^6$) and clearly enhancement of the overall heat transfer by natural convection.

6. Conclusions

Two dimensional natural convection heat transfer for steady state condition in a square enclose containing equilateral triangle filled with an air as a working fluid has been numerically investigated by using(ANSYS Fluent 16) for different Rayleigh numbers. The walls of the square enclosure are considered to be cold. While, the triangle walls as consider being hot, the results of the study are existing using many parameters of concern as Rayleigh number, and angle of inclination. Then, the following conclusions can be summarized from the present study.

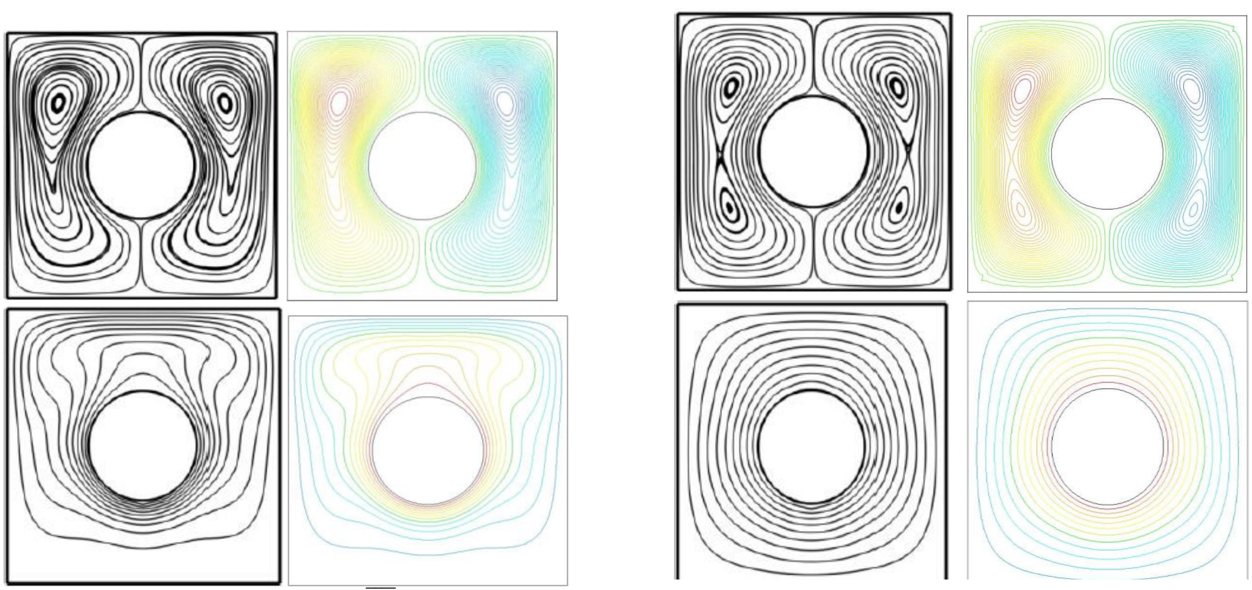
1-As the Rayleigh number goes higher, the flow lines become distorted and the isotherms become denser to the upper region and clearly observed the thermal plumes.

2- Average Nusselt number becomes slightly frequent for the lesser values of Rayleigh numbers due to domination of conductive mode.

3-The flow intensity is pointedly affected by the tilted angle, and the rotation of the heated triangle results in obvious modification on both the streamlines and temperature fields, the presence of two newest identical cells on the top of the triangle when (0^o,$Ra=10^5$,and 10^6) is related to traditional convection.

4-Highest values on average Nusselt number distribution are detected when the angle of inclination change between (15^o -30^o,and 60^o-105^o) for all varying of Rayleigh numbers . This is due to the increases of the size between the triangle and enclosure.

Increased Rayleigh number , as expected, resulting enhancement the rate of heat transfer due to increasing thermal convection phenomena because increased the amount of air velocity.



$\overline{Nu} = 2.58$ $\overline{Nu} = 2.60$ **$Ra=10^5$**

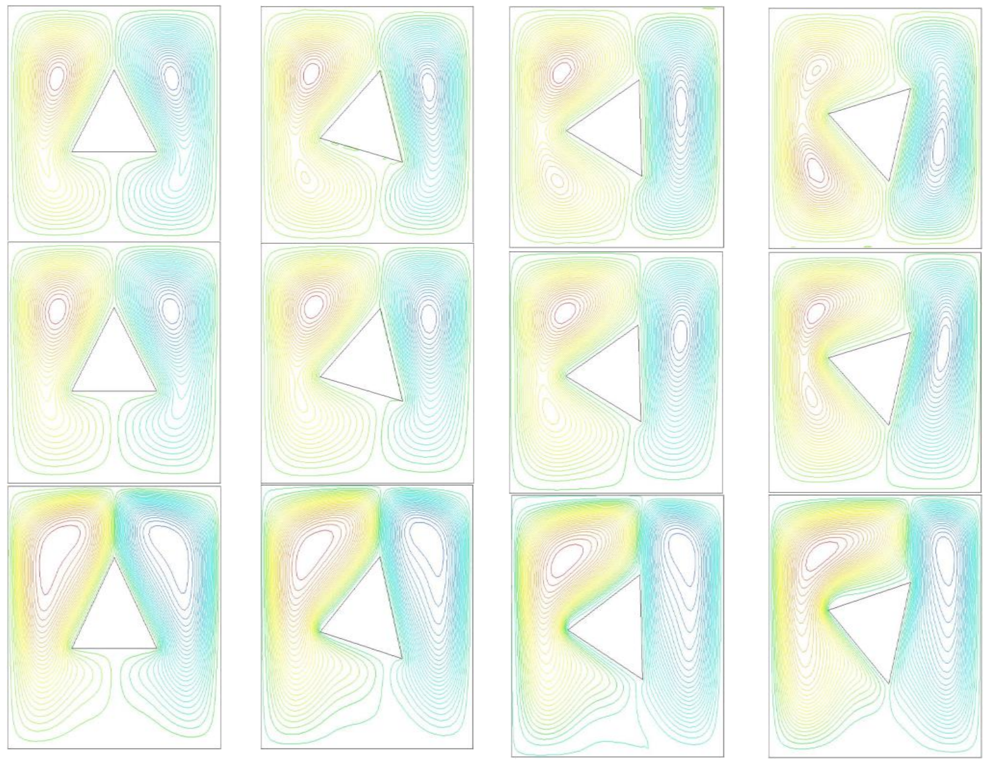
$\overline{Nu} = 1.63$ $\overline{Nu} = 1.68$ **$Ra=10^4$**

Fig(3) Comparison of streamline, isothermal line and heat transfer for present study and ref(6)

A

B

C



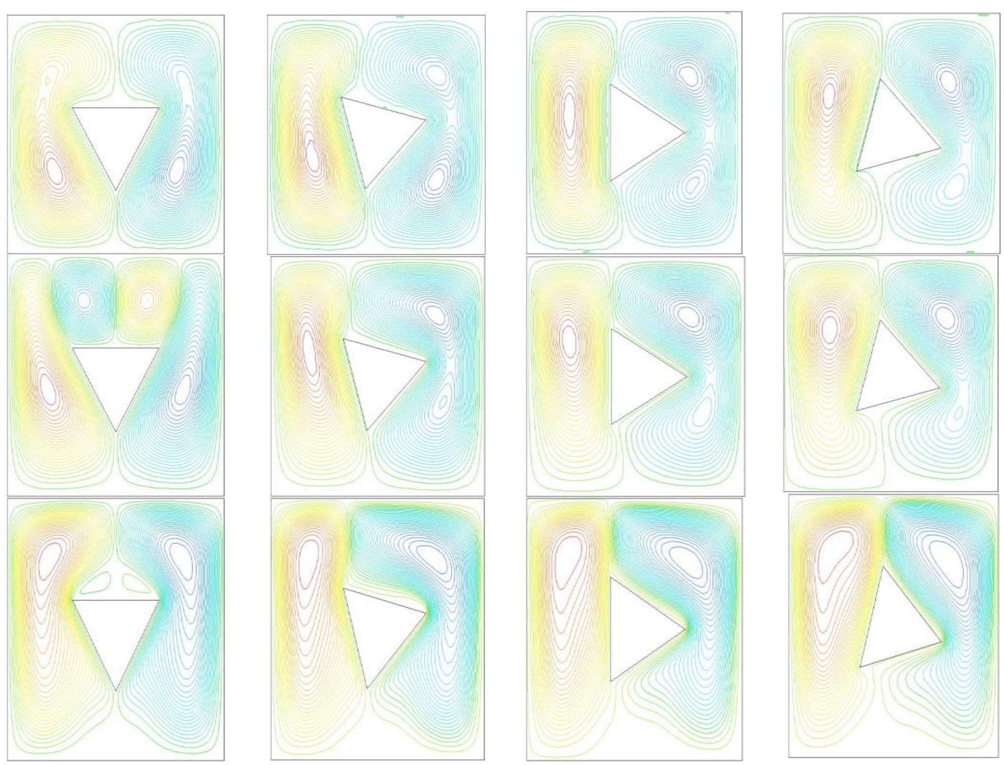
Fig(4)stream lines for (A)Ra=10^4, (B)Ra=10^5,(C)Ra=10^6 at different

θ =0^0 **θ** =15^0 **θ** =30^0 **θ** =45^0

A

B

C



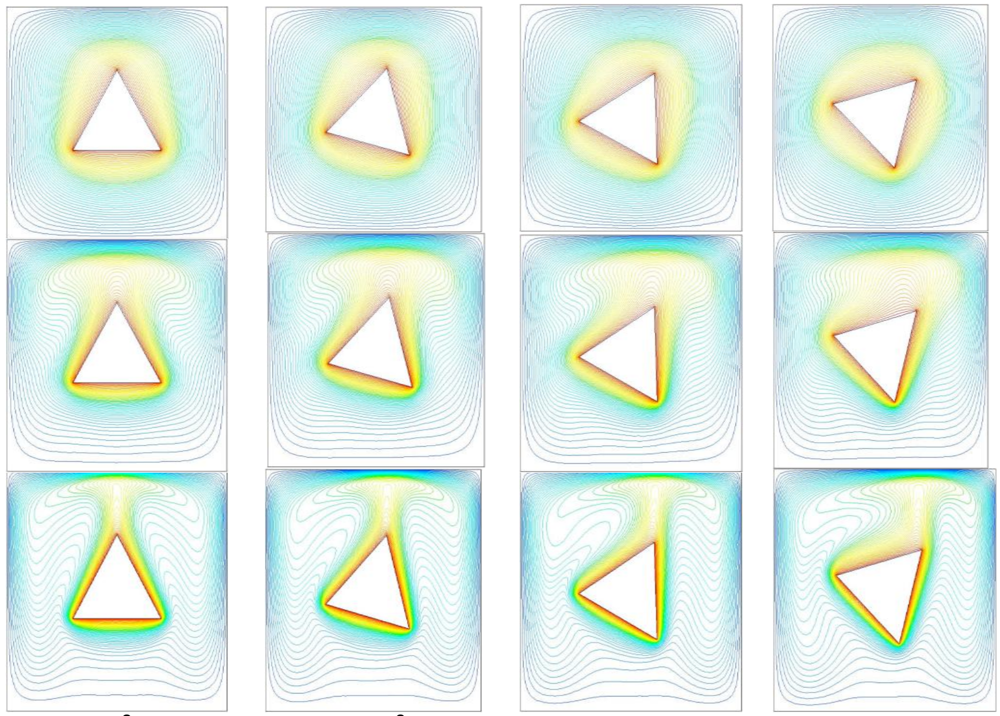
θ=60° **θ=75°** **θ=90°** **θ=105°**

Fig. 5. Streamlines (A)Ra=10^4, (B)Ra=10^5,(C)Ra=10^6 at different angle of triangle cylinder

A

B

C



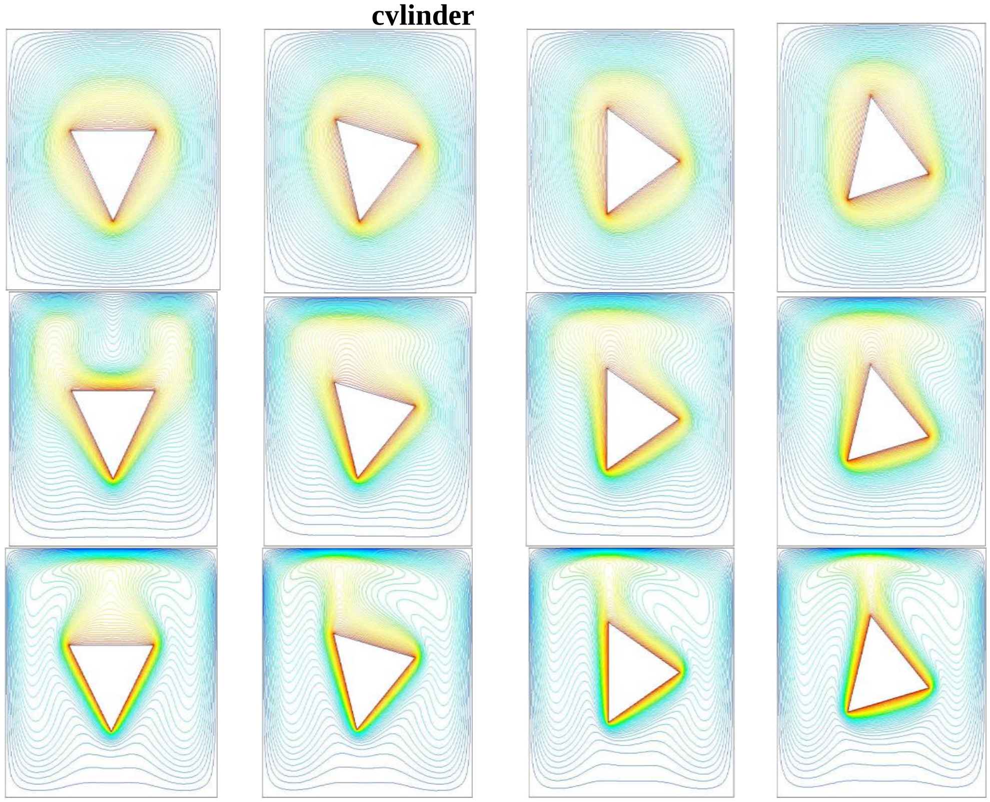
θ=0° θ=15° θ=30° θ=45°

Fig. 6. Variation in isotherms (A)Ra=10^4, (B)Ra=10^5,(C)Ra=10^6 at different angle of triangle cylinder

A

B

C



θ=60° θ=75° θ=90° θ=105°

Fig. 7. Variation in isotherms (A)Ra=10^4, (B)Ra=10^5,(C)Ra=10^6 at different angle of triangle cylinder

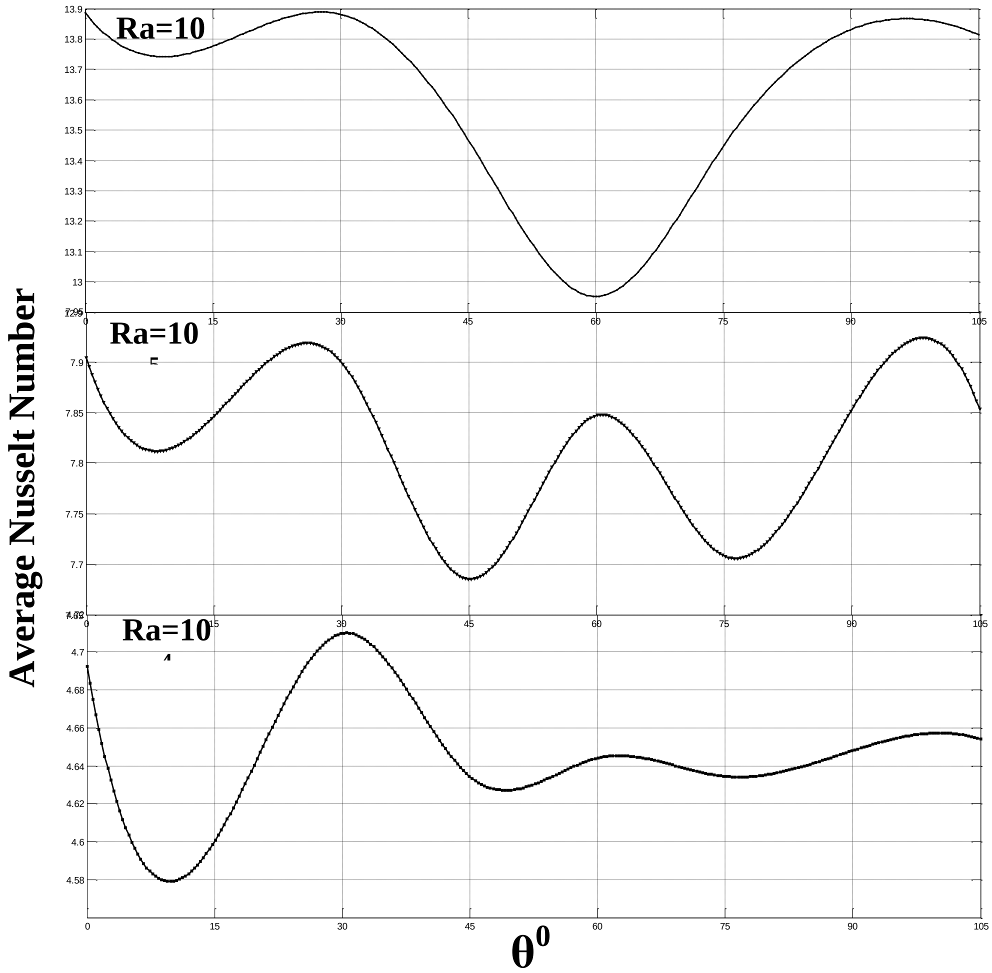


Fig. 8. Variation of average Nusselt number (Nu) Around Hot Equilateral Triangle with Different Orientations for Ra=10^4, 10^5 and 10^6.

References

Altac, Z. and Konrat,S.; 2009, "Natural Convection Heat Transfer From A thin Horizontal Isothermal Plate in Air-Filled Rectangular Enclosure", J. of Thermal Science and Technology, 29, No. 1, 55-56.

Arnab, K., Amaresh, D.; 2006"A numericalstudy of natural convection around a square horizontal heated cylinder placed in an enclosure",Int. J. Heat and Mass Transfer, vol.49, pp.4608-4623.

Costa, V.A., Oliviera, M. and Sousa; 2003, "Control Of Laminar Natural Convection In differentially Heated Square Enclosure Using Solid Inserts At The Corner", International Journal of Heat and Mass Transfer, Vol. 46, 3529–3537.

Habibis S., Ammar ,I. A. ,and Ishak, H.; 2015" Natural Convection in a Differentially Heated Square Enclosure with A Solid Polygon". Advances in Mechanical Engineering ,Vol. 7(12),1-10.

Hatton ,A. , James , D. and Swire, H.; 1970 "Combined Forced And Natural Convection With Low-Speed Air Flow Over Horizontal Cylinders" , Journal of Fluid Mechanics, Vol. 42, 17–31.

Hojat, K., and Seyed, A. M.; 2012, "Comparison of Natural Convection A Round A Circular Cylinder With A Square Cylinder Inside A Square Enclosure", Journal of Mechanical Engineering and Automation, 2(6) 176-183.

Karayiannis, T. G., Ciofalo,M.,and Barbaro,G.; 1992" On Natural Convection In A Single And Two zone Rectangular Enclosure" Int.J. Heat Mass Transfer, Vol. 35,No. 7, 1645-1657.

Roslan, R., Saleh, H., and Hashim,I.; 2014" Natural convection in polygonal enclosures with inner circular cylinder", The Scientific World Journal Volume ,Article ID 617492, 11 pages

Xing, Y., Fatemeh ,T., and Kambiz ,V.; 2015 "Analysis Of Natural Convection In Horizontal Concentric Annuli Of varying Inner Shape Numerical Heat Transfer", Part A, 68, 1155–1174.

Xu, X., Yu,Z., Hub,Y., Fan,L.,and Cen, K.; 2012 "Transient Natural Convective Heat Transfer Of A Low-Prandtl-Number Fluid From A Heated Horizontal Circular Cylinder To Its Coaxial Triangular Enclosure." International Journal of Heat and MassTransfer,vol.55, 995–1003.

Xu, X., Zitao ,Y., Yacai ,H., Liwu, F.,and Kefa ,C.; 2010 " A numerical Study Of Laminar Natural Convective Heat Transfer A Round A Horizontal Cylinder Inside A Concentric Air-Filled Triangular Enclosure", International Journal of Heat and Mass Transfer, 53, 345–355.

Nomenclature

d Dimensional cylinder length (m)
D Dimensionless cylinder length
Gr Grashof number
g Gravitational acceleration (m/s^2)
k Thermal conductivity of fluid (W/m.K)
L Length of the enclosure (m)
n Normal direction
Nu Nusselt number
p Pressure (N/m^2)
P Dimensionless pressure
Pr Prandtl number
r Relaxation function
R Radius of circular cylinder (m)
Re Reynolds number
Ri Richardson number
T Temperature (K)
T_c Cold temperature (K)
T_h Hot temperature (K)
u ,v Cartesian velocity components (m/s)
X, Y Cartesian coordinate

Greek Symbols

α Thermal diffusivity (m^2/s)
β Thermal expansion coefficient (1/K)
ξ Vorticity function
ζ Dimensionless vorticity function
υ Kinematic viscosity (m^2/s)
φ angle of circular cylinder (degree)
θ Dimensionless temperature
ρ Density of the fluid (kg/m^3)
ψ Stream function
Ψ Dimensionless stream function
δ Distance from bottom wall to the circular cylinder center
ω angular velocity (rad/sec)

Superscripts

$*$ dimensionless value