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ANALYSIS OF NATURAL CONVECTION OF AIR IN TRIANGULAR ENCLOSURES OF DIFFERENT INCLINATION ANGLES

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Abstract

In this paper, a numerical investigation has been conducted in triangular enclosures with air inside it to study the effect of enclosure size, aspect ratio of the enclosure (height of the enclosure for a given base length) on the flow, temperature distribution and heat transfer characteristics. The triangular enclosures are isosceles in nature, the two inclined walls are considered to be at higher temperature and the base at lower temperature. The governing equations are solved using the commercial software package ANSYS Fluent. Assuming constant properties of air, the Rayleigh number depends on the size of the enclosure for fixed high and low temperatures. The Rayleigh number is varied from 10^5 to 10^7. The flow and temperature distribution also depends on the aspect ratio of the enclosure. Keeping base length fixed, the increase in the aspect ratio implies more height of the enclosure. Aspect ratios in the range from 0.5 to 1.5 are taken. Results are presented in the form of contours of isotherm and stream function. The heat transfer at the walls and the flow inside the enclosure are analyzed. The motion of fluid is almost uniform and the temperature gradient is more for lower Rayleigh numbers. For enclosures with

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lower aspect ratios, the motion is almost uniform and temperature gradient is more at the active walls. Heat transfer rate from hot wall to cold wall is more for higher Rayleigh numbers and lower aspect ratios.

1. Introduction

Nowadays, it is a primary requirement for the buildings to be energy efficient and we must have control on energy consumption which we use mainly for heating or air-conditioning. In the past many years, researchers have carried out lots of work to provide an efficient thermal comfort to habitats of inclined shaped roof. The effect of size of a triangular enclosure and its height on the flow and heat transfer characteristics can be studied by modelling this problem. Flack and Brun [1] conducted experiments to study natural convection in triangular enclosures with the Grashof number in the range from 1.89×10^6 to 10.3×10^6 and angles varying as 30°, 45° and 60°. Asan and Namli [2] have analyzed the laminar natural convection in a pitched roof of triangular cross-section for Rayleigh number of 10^5 and aspect ratios of $0.125 \leq L \leq 1$. Varol et al. [3] have taken a triangular enclosure with flush mounted heater on the vertical wall to analyze the natural convection. Basak et al. [4] have worked on natural convection flow in an isosceles triangular enclosure with heating at the side walls. The results are presented for the range of Rayleigh numbers from 10^3 to 10^6 and the Prandtl numbers from 0.026 to 1000. Fuad Kent [5] has worked on numerical analysis of laminar natural convection in isosceles triangular enclosures for cold base and hot inclined walls. The angle formed between the inclined wall and bottom wall is varied as 15°, 30°, 45°, 60° and 75° and Rayleigh numbers as $10^3, 10^4$ and 10^5. Kaluri et al. [6] have worked on heatline analysis of natural convection right-angled triangular cavities. The effects of height-base ratio and thermal boundary conditions with various top angles of 15°, 30°, 45°, Prandtl number $Pr = 7.2, 1000, 0.015$ and Rayleigh number $Ra = 10^3$ to 10^5 were studied. Yesiloz and Aydin [7] have worked on laminar natural convection in right-angled triangular cavities heated and cooled on adjacent walls. The Rayleigh number was taken in the range of 10^3 to 10^7. Sojoudia et al. [8] have worked on numerical analysis of free convection heat transfer in an attic shaped enclosure with differentially heated two inclined walls and filled with air. There is a heat source placed on the right side of the bottom surface. Effect of various flow parameters on fluid flow and heat transfer has been discussed. In the present work, the effect of size and aspect ratio of triangular enclosures on the flow and heat transfer for Rayleigh numbers 10^5 to 10^7 and aspect ratios 0.5, 1 and 1.5 are studied.

2. Analysis

The details of the geometry for the configurations considered are shown in Figure 1(a). The triangular enclosure composed of two inclined walls on the top maintained at constant temperature of 318K and a horizontal bottom wall maintained at constant temperature of 298K. Structured meshing method is used for meshing the geometry. Due to symmetry of the problem, half of the physical domain is considered, with the plain of symmetry considered as insulated boundary. The triangular enclosure with structured mesh is shown in Figure 1(b). The properties of air are taken at the mean temperature.

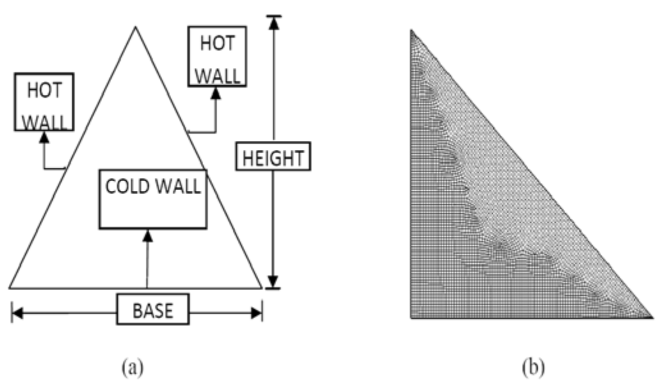


Figure 1. (a) Geometry of the physical domain and (b) meshing of the computational domain.

3. Results

The fluid flow and heat transfer process inside the enclosure depend on the properties of the fluid, size of the enclosure and space between the hot and cold walls. The properties of the fluid and size of the enclosure are taken care of by the Rayleigh number whereas the space between the hot walls and cold wall is determined by the aspect ratio. So, the Rayleigh number and aspect ratio are taken as the parameters to see the effect of their variation on the flow and heat transfer characteristics in the enclosure.

3.1. Influence of enclosure size

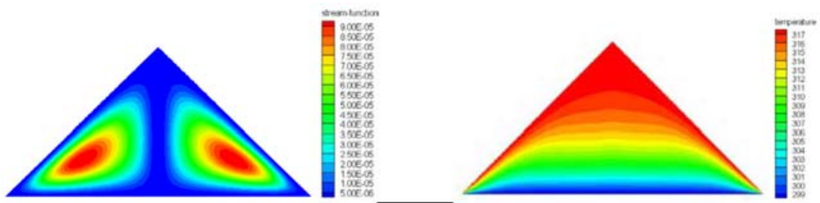
3.1.1. Flow and temperature distribution

The solution is obtained for the computational domain which is half of the physical domain. The isotherms and stream function contours are shown for the entire physical domain whereas the quantities of the horizontal mid-plane are shown for the computational domain only. Due to the symmetric nature of physical domain and boundary conditions about the mid vertical plane (insulated plane), the flow and temperature distribution are also symmetric about that plane.

Keeping the temperature of the walls fixed, increase in Rayleigh number implies increase in size of the enclosure. As the temperature of the inclined walls is higher, the fluid (air) adjacent to the inclined walls is heated. Due to the buoyancy effect, the air moves upward along the inclined wall. While moving upward the fluid particles strike at the inclined wall and the motion is retarded. At lower Rayleigh number, since the size of the enclosure is small, the air moves along the inclined wall up to the top corner and circulates throughout the enclosure (Figure 2(a)). But, for higher Rayleigh number, the air cannot move up to the top corner and circulates much below it with stagnant fluid at the upper portion (Figure 3(a) and Figure 4(a)). The center of the loop is shifted towards the bottom corner as Rayleigh number increases. This implies that the circulation is more near the bottom. The fluid passes through a smaller area at the bottom corner due to the shape of the enclosure. So the velocity is more near this corner. But away from the bottom corner, the fluid gets much more space and the velocity is comparatively smaller.

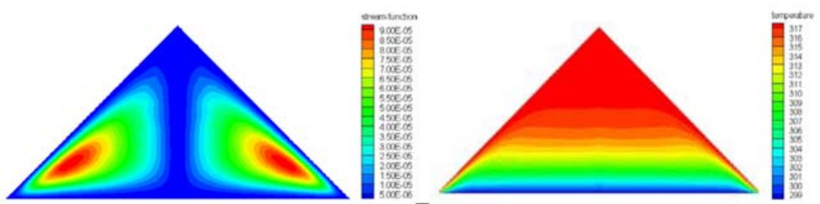
(a) (b)

Figure 2. Contours of (a) stream function and (b) isotherms for $Ra = 10^5$ and $AR = 1$.



(a) (b)

Figure 3. Contours of (a) stream function and (b) isotherms for $Ra = 10^6$ and $AR = 1$.



(a) (b)

Figure 4. Contours of (a) stream function and (b) isotherms for $Ra = 10^7$ and $AR = 1$.

Figure 5 shows the variation of vertical component velocity at horizontal mid-plane of the enclosure. The length is non-dimensionalised to accommodate the results for all Rayleigh numbers. The vertical component velocity is more for a higher Rayleigh number compared to that for a lower Rayleigh number.

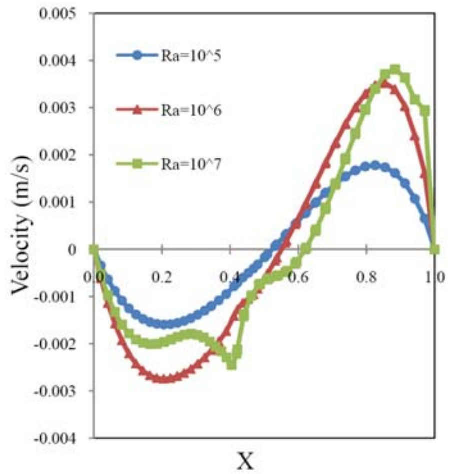


Figure 5. Variation of vertical component of velocity at mid-plane for $AR = 1$.

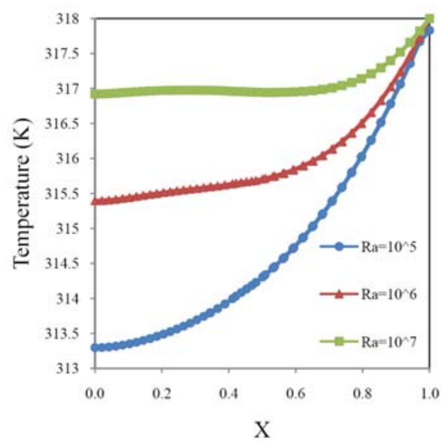


Figure 6. Variation of temperature at mid-plane for $AR = 1$.

From isotherm contours for different Rayleigh numbers (Figures 1(b), 2(b) and 3(b)), it is observed that the isotherms are closely spaced near the walls, especially at the bottom corner. This is because at the bottom corner the adjacent walls are at extreme thermal conditions and the distance between them is very less. As the fluid circulates throughout the enclosure at low Rayleigh number, the temperature varies more. But at higher Rayleigh number, the temperature variation is mainly in lower portion.

Figure 6 shows the mid-plane temperature variation for different Rayleigh numbers. The temperature gradient mainly exists near the inclined wall and the gradient is more for higher Rayleigh number.

3.1.2. Heat transfer characteristics

From Figure 7, it is observed that the heat flux from the hot wall is very large near the bottom compared to the rest of the walls due to high temperature gradient. The heat flux from the hot wall is less for higher Rayleigh number compared because of less temperature gradient at any point compared to that at a corresponding point for a lower Rayleigh number. Similar result is also obtained for the cold wall (Figure 8).

From Table 1, the total heat transfer rate from hot and cold walls is found to be more as Rayleigh number increases. Though the hot and cold walls are having less heat flux for higher Rayleigh number, the heat transfer rates at the walls are more because of higher area of the walls.

Table 1. Total heat transfer rate from the walls at different Rayleigh numbers

Ra	*AR*	*Q* (Hot wall) (*W*)	*Q* (Cold wall) (*W*)	*Q* (Insulated wall) (*W*)
10^5	1	3.957	–3.945	0
10^6	1	4.051	–4.037	0
10^7	1	4.306	–4.285	0

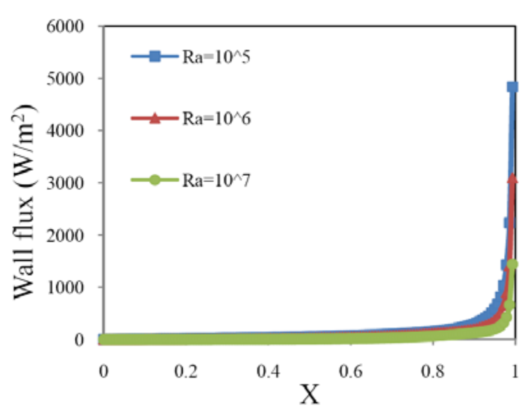


Figure 7. Variation of wall flux at hot wall for $AR = 1$.

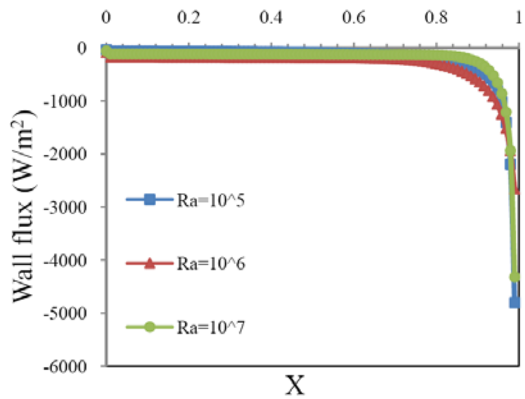


Figure 8. Variation of wall flux at cold wall for $AR = 1$.

3.2. Influence of aspect ratio

3.2.1. Flow and temperature distribution

For a given Rayleigh number (fixed enclosure base), different heights of triangular enclosure represent different aspect ratios. As the aspect ratio increases, the space between the inclined wall and bottom wall also increases by the result of which the motion of the fluid decreases with the increase in aspect ratio. More portion of the fluid at the top remains stagnant with increase in aspect ratio. The curvature of the isotherms decreases with the increase in aspect ratio. With the increase in aspect ratio, the isotherms are flattened in the central portion. So the temperature gradient increases at the bottom whereas it decreases near the top (Figures 9-11).

The vertical component of velocity is zero at both the hot wall and insulated wall (Figure 12). The velocity is positive near hot wall and negative near the insulated wall as the fluid moves upward near the hot wall and it moves downward near the insulated wall. The vertical component velocity increases with the increase in aspect ratio up to 1 and then it decreases. The velocity of the fluid is zero at the center of the loop as well as at the wall. So the magnitude of velocity increases from zero to a maximum and then decreases to zero from the hot/insulated wall to the center of the loop. With increase in aspect ratio, the loop is shifted towards the bottom corner. While changing aspect ratio, the horizontal mid-plane crosses the loop at different positions of the loop. The velocity is different depending upon the relative position of the mid-plane with respect to the centre of the loop. So the velocity is the maximum for an intermediate position.

Figure 13 shows the temperature variation at mid-plane for different aspect ratios. As explained earlier, the isotherms are flattened as aspect ratio increases. So the temperature gradient is more near the hot wall for lower aspect ratio and the gradient is reduced towards the insulated wall. For higher aspect ratio, the trend is similar, but the gradient is less compared to that for lower aspect ratio at all locations.

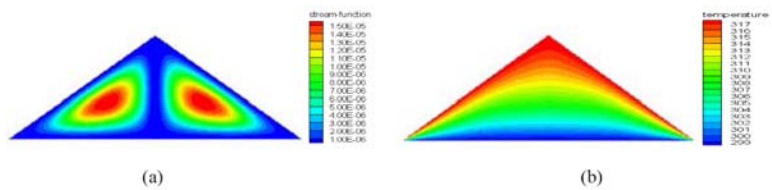


Figure 9. Contours of (a) stream function and (b) isotherms for $Ra = 10^6$ and $AR = 0.5$.

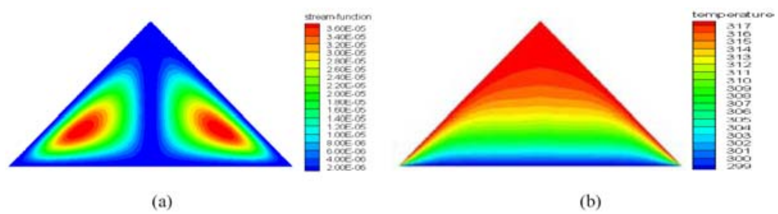


Figure 10. Contours of (a) stream function and (b) isotherms for $Ra = 10^6$ and $AR = 1$.

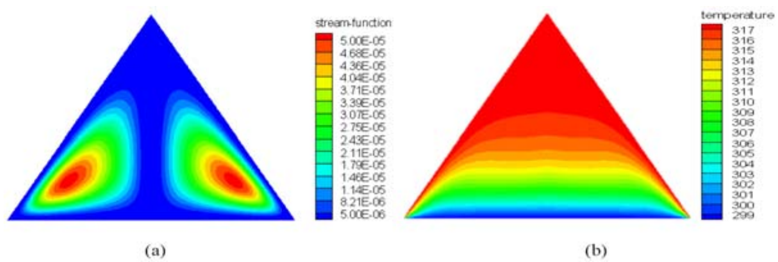


Figure 11. Contours of (a) stream function and (b) isotherms for $Ra = 10^6$ and $AR = 1.5$.

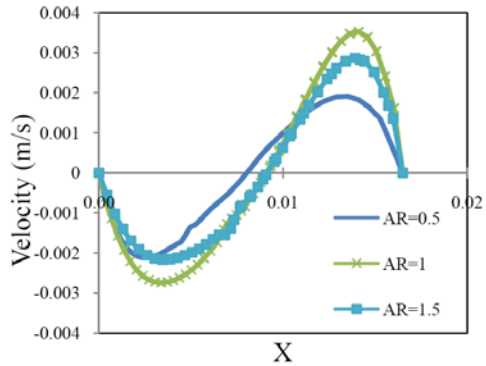


Figure 12. Variation of vertical component of velocity at mid-plane for $Ra = 10^6$.

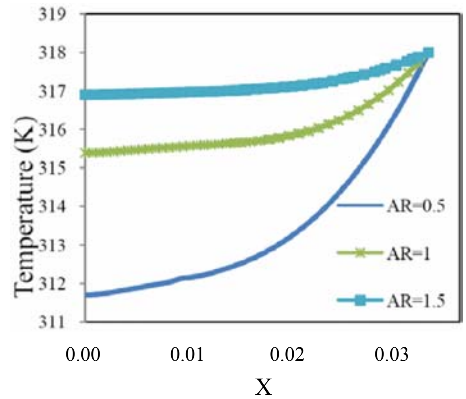


Figure 13. Variation of temperature at mid-plane for $Ra = 10^6$.

3.2.2. Heat transfer characteristics

As explained earlier, the heat flux is very large near the bottom corner compared to the other locations of the active walls. The magnitude of heat flux is less for a higher aspect ratio (Figure 14 and Figure 15) with a given Rayleigh number. Since the base length is same, the height is more for higher aspect ratio. Due to this, the isotherms more widely spread and temperature gradient becomes less leading to less heat flux.

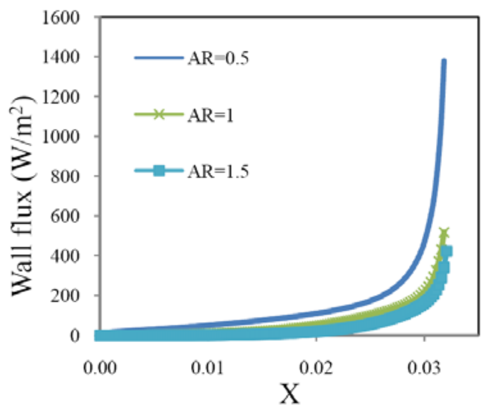


Figure 14. Variation of wall flux at hot wall for $Ra = 10^6$.

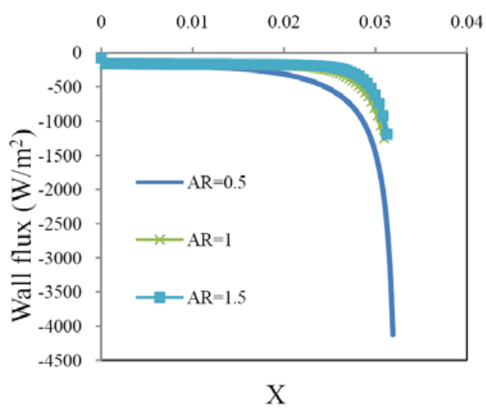


Figure 15. Variation of wall flux at cold wall for $Ra = 10^6$.

Table 2. The heat transfer rate from the walls at different aspect ratios

AR	*Ra*	*Q* (Hot wall) (*W*)	*Q* (Cold wall) (*W*)	*Q* (Insulated wall) (*W*)
0.5	10^6	7.08	−7.0682	0
1	10^6	4.051	−4.037	0
1.5	10^6	3.182	−3.1685	0

The heat transfer rate from the hot and cold walls is also observed to be less for higher aspect ratios (Table 2). The temperature gradient is less as explained earlier. The area of the bottom cold wall is same for all. So the heat transfer rate for the cold wall is obviously reduced. For the inclined hot wall, though the area increases with increase in aspect ratio, the decrease in temperature gradient dominates the effect of increase in area.

4. Conclusion

• At lower Rayleigh number, the air circulates throughout the enclosure whereas for higher Rayleigh number air does not reach to the top corner and the air at the top portion of the enclosure remains almost stagnant.

• For lower Rayleigh number, the motion of the fluid is almost uniform throughout but as the Rayleigh number increases the motion of the fluid is more near the active walls and less in the core region.

• The temperature gradient (hence, the heat flux) at analogous points of different enclosure is less for higher Rayleigh number compared to that for lower Rayleigh number.

• For a given Rayleigh number, with the smaller aspect ratio, the motion is almost uniform within the enclosure. More portion of the fluid at the top remains stagnant with increase in aspect ratio.

• The temperature gradient (hence, the heat flux) is more at the active walls for lower aspect ratio and decreases with increase in aspect ratios.

• The heat transfer rate at hot and cold walls increases as Rayleigh number increases and decreases as aspect ratio increases.

• Heat transfer rate from hot wall to cold wall can be reduced effectively by putting insulation on lower portion of the hot wall.

• Heat can be removed effectively from the enclosed space of higher aspect ratios or higher Rayleigh number by installing an exhaust fan at the top corner of the enclosure.

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