Heat transfer in inclined air rectangular cavities with two localized heat sources

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Abstract The present work investigates numerically the effects of cavities’ aspect ratio and tilt angle on laminar natural convection of air in inclined rectangular cavities with two localized heat sources. A mathematical model was constructed where the conservation equations governing the mass, momentum and thermal energy together with their boundary conditions were solved. The calculation grid is investigated to determine the best grid spacing, number of iterations, and other parameters which affect the accuracy of the solutions. The numerical method and computer program were tested for pure conduction and convection with full heating (\( \varepsilon = 1 \)) to assure validity and accuracy of the numerical method.

The investigation used rectangular enclosures with position ratios of the heaters, \( B_1 = 0.25, B_2 = 0.75 \), size ratio, \( \varepsilon = 0.25 \), and covered Rayleigh numbers based on scale length, \( s/A \) ranging from \( 10^3 \) to \( 10^6 \). The tilt angle from the horizontal was changed from \( 0^\circ \) to \( 180^\circ \), and the aspect ratio was taken as \( A = 1, 5, \) and \( 10 \). The results are presented graphically in the form of streamlines and isotherm contour plots. The heat transfer characteristics, and average Nusselt numbers were also presented. A correlation for \( Nu \) is also given.

1. Introduction

Natural convection in enclosures has been extensively studied in the past both analytically and experimentally. Chu et al. [1] reported an extensive survey on natural convection from a discrete heater in an enclosure. They examined the effect of heater location in the enclosure with \( A = 0.4–5, Pr = 0.72, \Phi = 90^\circ \), and for a range of Rayleigh number, \( Ra_H \) up to \( 10^5 \). They found that the Nusselt number, \( Nu_H \) was proportional to \( Ra_H \) for any location of the discrete heat source. Turner and Flack [2,3] have experimentally examined the heat transfer in geometry similar to that used by Chu et al. [1] for Grashof numbers, \( Gr_H \) up to \( 9 \times 10^6 \). They obtained the same form of correlation as \( Nu_H = C_1 \cdot Gr_H^{C_2} \), where \( C_1 \) and \( C_2 \) are functions of size ratio, \( \varepsilon \). Yovanovich [4] reported an expression for the thermal constrictive resistance, \( r_c \) of a discrete heat source on a rectangular solid region with the heat sink on the opposite side and other sides are insulated. The expression is given as \( r_c = (1/\pi k) ln [(1/sin (\pi \varepsilon/2) \cos\pi (\beta–0.5))] \).

Elsherbiny et al. [5] determined experimentally the effect of heater location using three heaters on the hot wall. They measured \( Nu \) for each heater and found that \( Nu \) for upper heater was decreased up to a certain \( Ra \) and then increased.

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Markatos and Pericleous [6] have experimentally examined the heat transfer in a square enclosure (A = 1), full contact heated wall (e = 1), Pr = 0.71, Φ = 90°, B = 0.50, and the range of Ra from 10^3 to 10^6. They obtained the velocity distribution and a correlation of Nusselt numbers as a function of Ra_M as Nu_M = C1 (Ra_M)^0.1805, where C1, C2 and C3 are constants depending on heater arrangement. Heindel et al. [10,11] have experimentally studied the heat transfer in an enclosure filled with ethylene glycol. The hot wall consisted of 11 discrete iso-flux heaters where A = 15.5, Pr = 150, and the local modified Rayleigh number was in the range of 9.3 × 10^11 to 1.9 × 10^12. They correlated the local Nusselt number as Nu_L = 1.009 Ra_L^0.1805, where “x” is the local height, measured from bottom of cavity to mid-height of the heated section. Chadwick et al. [8] have experimentally examined the heat transfer in a rectangular enclosure with an iso-flux heating mode with A = 5, Pr = 0.71, e = 0.133, Φ = 90°, and Gr = Gr* (Gr* = gβ(T_h - T_c)(s/A)^3/νκ) ranged from 10^4 to 5 × 10^5. They obtained the value of average Nusselt number as a function of Gr* based on heater length. The average Nusselt number was obtained as Nu_A = 0.153, where C1 and C2 are functions of e. Also, they obtained the value of local Nusselt number as a function of Gr* and the local modified Grashof number based on distance from leading edge of heater (Gr* = gβ(T_h - T_c)(s/A)^3/νκ) and “x” is the distance from the leading edge of the heater to any point at surface heater. The local Nusselt number, Nu_L was correlated as Nu_L = C1s(Gr*^2), where C1 and C2 are functions of B.

Ho and Chang [9] studied numerically and experimentally the influence of aspect ratio on heat transfer in an enclosure which has 4 iso-flux heaters where A changed from 1 to 10, Pr = 0.71, Φ = 90° and a range of Ra_M from 10^4 to 10^6. They obtained the value of Nusselt number, Nu_M as Nu_M = C1s (Ra_M)^0.1805, where C1, C2 and C3 are constants depending on heater arrangement. Heindel et al. [10,11] have experimentally studied the heat transfer in a rectangular enclosure. The heat wall consisted of 3 × 3 array of heaters where Φ = 90°, A ranged from 2.5 to 7.5, Pr ranged from 5 to 25, and Ra_A ranged from 10^3 to 10^6. They obtained the value of Nu for each row of heaters as Nu = C5Ra^0.25, where C5 is a function of row arrangement.

Ahmed and Yovanovich [12] studied numerically the influence of discrete heat source location on natural convection heat transfer in a range of Ra*, based on scale length, s/A from 0 to 10^6, Pr = 0.72, A = 1, B = 0.5 and 0 ≤ e ≤ 1.0. They obtained analytical correlations for this problem for either isothermal heat source or iso-flux heat source.

Al-Bahi et al. [13] studied numerically the effect of heater location on local and average heat transfer rates in an enclosure which has a single iso-flux heater where A = 1, Pr = 0.71, Φ = 90°, e = 0.125, Ra_A ranges from 10^3 to 10^6 and B = 0.25, 0.50 and 0.75. They obtained isotherms, streamlines and temperature distribution characteristics at high and low Ra*. Al-Bahi et al. [14] studied numerically the effect of tilt angle from horizontal on local and average heat transfer rates in an enclosure which has a single iso-flux heater where A = 5.
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$B = 0.50$, $\varepsilon = 0.125$. $\Phi$ is varied from $0^\circ$ to $180^\circ$, $Pr = 0.71$ and $Ra^*$ ranges from $10^2$ to $10^6$. They obtained isotherms and streamlines plots for different $Ra^*$.

Sezai and Mohamad [15] studied numerically the effect of heater size ratio on average heat transfer rates in a rectangular enclosure which has a single heater where, $\Phi = 0^\circ$, $0 \leq \varepsilon \leq 1$, $Pr = 0.71$, and Rayleigh numbers based on enclosure height, $Ra_L \geq 10^3$. They found that $Nu$ was decreased when $\varepsilon$ increased. Aydin and Pop [16] studied numerically the effect of heater size, Rayleigh number, and Prandtl number on heat transfer rate in an enclosure where $\Phi = 90^\circ$, and $B = 0.50$. They presented the results in the form of isotherms and streamlines plots as well as the variation of the local Nusselt number through the discrete heater. Saeid and Pop [17] studied numerically the effect of heater size, location and aspect ratio on heat transfer rate in a porous cavity saturated with water which possesses a density maximum in the vicinity of $3.98^\circ$C where $\Phi = 90^\circ$, and $50 \leq Ra \leq 1000$. They found that for long heater and low Ra, the maximum heat transfer occurs when $B = 0.50$. For short heater and high Ra, fixing the heater in the upper half of the vertical wall leads to an enhancement of the heat transfer. Also, increasing $A$ more than 0.50, $Nu$ will be increased. Saeid [18] studied numerically the influence of discrete heat source location on natural convection heat transfer in a range of $Ra$ from $10^3$ to $10^5$, $A = 1$ and $0.1 \leq \varepsilon \leq 0.50$. He found that the maximum average Nusselt number takes place when the heater is placed near the bottom of the vertical wall for high values of $Ra$ and relatively higher location for relatively low values of $Ra$ for both isothermal or isoflux heat source.

Bae et al. [19] studied numerically the effect of number of heaters (full contact heat source, $\varepsilon = 1$) where $10^3 \leq Ra \leq 10^5$. They found that, at low $Ra$, heat transfer across the enclosure is strengthened as the number of heaters increases. Corvaro and Paroncini [20] studied numerically and experimentally the effect of heater location where $A = 1$, and $Pr = 0.71$ using three heaters on the hot wall. They measured temperature distribution and Nusselt numbers at different Rayleigh numbers on the heated strip. Ramadhyani and Incropera [21] have experimentally determined the heat transfer coefficient for various flow rates, fluids, heater sizes, and enclosure dimensions. Jin et al. [22] studied numerically the effect of slowly rotating an enclosure (about its longitudinal horizontal axis) with 3 rows of discrete heat sources on $Nu$ and heat transfer where $A = 3.75$, and $7.5$, $10^3 \leq Ra^* \leq 10^5$, and $Pr = 0.71$. They found that heat transfer behavior for the heaters of rows 1, and 3 is asymmetrical. Choi and Ortega [23] studied numerically the effect of inlet flow velocity and the inclination angle on $Nu$ and heat transfer where $Pr = 0.71$. They found that $Nu$ strongly depends on the inclination angle in natural and forced regimes. Papanicolaou and Gopalakrishna [24] studied numerically the effect of aspect ratio, size ratio, and Rayleigh number on $Nu$ and heat transfer for isoflux discrete heating where $Pr = 0.71$, and $\Phi = 0$. They found that the value of $Nu$ depends on aspect ratio, size ratio, and Rayleigh number for single and multiple heaters.

Yucel et al. [25] studied numerically the effect of aspect ratio, size ratio, inclination angle, and Rayleigh number on $Nu$ where $0 \leq \Phi \leq 90^\circ$, $0.1 \leq \varepsilon \leq 1.0$, $5 \leq A \leq 20$, and $10^5 \leq Ra \leq 10^6$. They obtained isotherms, streamlines, and temperature distribution characteristics for different $Ra$. Teamah and El-Maghllany [26] investigated the augmentation of natural convection in a square cavity using nanofluids in the presence of a magnetic field and uniform heat generation/absorption. They predicted values for the average Nusselt number for various parametric conditions.

2. Mathematical model

The configuration under consideration is shown schematically in Fig. 1.

The fluid in the enclosure is air ($Pr = 0.71$) with constant properties except for the density in the buoyancy force components existing in the momentum equations. The Boussinesq approximation will relate the variable density to the local temperature. In other words, the two momentum equations will be coupled with the energy equation. The differential equations governing the conservation of mass, momentum, and thermal energy are given as

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

$$\rho \left[ u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} \right] = \rho g \beta (T - T_e) \sin \phi - \frac{\partial p}{\partial x} + \mu \left[ \frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} \right]$$ (2)

$$\rho \left[ u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} \right] = \rho g \beta (T - T_e) \cos \phi - \frac{\partial p}{\partial y} + \mu \left[ \frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} \right]$$ (3)

$$\left[ u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} \right] = \alpha \left[ \frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} \right]$$ (4)

The mathematical set of equations, the continuity Eq. (1), the momentum Eqs. (2) and (3), and the energy Eq. (4) have to be solved numerically. In order to reduce the computational effort, it is necessary to cast the set of equations in a dimensionless form. Introducing the following dimensionless variables
Local Nusselt number is defined as

\[ \text{Nu}_x = \frac{h(s/A)}{k} = \frac{(s/A) \partial T}{T_x - T_s} \]

The average Nusselt number, Nu can be calculated by

\[ \text{Nu} = \frac{h(s/A)}{k} = \frac{1}{A} \sum \frac{\partial T}{\partial Y} \Delta X \ldots \ldots \text{on heated surfaces} \]
Ra = 10^6, the two rolls merged together forming one big cell rotating at a higher velocity. The isothermal lines for Ra = 10^3, are clustered at one heater with higher local heat transfer. For Ra = 10^4, there is a symmetry at the cavity vertical centerline with identical values for both heaters. For Ra = 10^6, the isotherms show the turbulence in the cavity core and they are close at one heater at a time which calls for an oscillating flow since the cavity is horizontal.

3.3.2. For A = 1, and \( \Phi = 30^\circ \)

Fig. 4 shows the streamlines “top” and the isotherms “bottom” [\( A = 1, \Phi = 30^\circ, \varepsilon = 0.25, B_1 = 0.25 \) and \( B_2 = 0.75 \)].
the walls and few cells appeared in the center. The isothermal lines for \( Ra = 10^3 \), are clustered at the lower heater with higher local heat transfer. For \( Ra = 10^4 \), the effect of convection is more pronounced in the isotherms; the isothermal lines are parallel near the cold wall where heat transfer is a transition from conduction to convection. For \( Ra = 10^6 \), the isotherms are clustered near the surface of the two heaters and show the increased effect of convection.

3.3.3. For \( A = 1 \), and \( \Phi = 60^\circ \)

Fig. 5 shows the streamlines “top” and the isotherms “bottom” [\( A = 1, \Phi = 60^\circ, \epsilon = 0.25, B_1 = 0.25, \) and \( B_2 = 0.75 \)].
the center are distorted. The isothermal lines for \( Ra = 10^3 \) are clustered at the lower heater with higher local heat transfer. For \( Ra = 10^4 \), the effect of convection is more pronounced in the isotherms; the isothermal lines are parallel near the cold wall. For \( Ra = 10^6 \), the isotherms are clustered on the two heaters indicating that the effect of convection is increased. The isotherms are horizontal at the center of the cavity.

3.3.4. For \( A = 1 \), and \( \Phi = 90^\circ \)

Fig. 6 shows the streamlines “top” and the isotherms “bottom” of the case \( \{ A = 1, \Phi = 90^\circ, \epsilon = 0.25, B_1 = 0.25, \text{ and } B_2 = 0.75 \} \). For \( Ra = 10^3 \), one big single roll with low rotational speed is shown. The center of the vortex is near the center of the cavity. At \( Ra = 10^4 \), the core of the vortex is distorted and moved slightly downward and near the cold wall. For \( Ra = 10^6 \), the streamlines take a rectangular shape and the vortices at the center are distorted. The isothermal lines for \( Ra = 10^3 \), are clustered at the lower heater with higher local heat transfer. For \( Ra = 10^4 \), the effect of convection is more pronounced in the isotherms. For \( Ra = 10^6 \), the isotherms are still clustered on the lower heater. The isotherms are horizontal at the core of the cavity which indicates a stratified flow.

3.3.5. For \( A = 1 \), and \( \Phi = 120^\circ \)

Fig. 7 shows the streamlines “top” and the isotherms “bottom” of the case \( \{ A = 1, \Phi = 120^\circ, \epsilon = 0.25, B_1 = 0.25, \text{ and } B_2 = 0.75 \} \). For \( Ra = 10^4 \), one big single roll with low rotational speed is shown. At \( Ra = 10^6 \), the core is distorted with two circulations one near the heaters and the other near the cold wall. For \( Ra = 10^6 \), the flow is mainly consisted of two vortices with a dead zone separating the upper vortex from the lower vortex. The isothermal lines for \( Ra = 10^3 \), are parallel to the cold wall which indicates that heat is transferred mainly by conduction. For \( Ra \geq 10^3 \), the lines become horizontal with higher heat transfer close to the top of the cold wall.

3.3.6. For \( A = 1 \), and \( \Phi = 150^\circ \)

Fig. 8 shows the streamlines “top” and the isotherms “bottom” of the case \( \{ A = 1, \Phi = 150^\circ, \epsilon = 0.25, B_1 = 0.25, \text{ and } B_2 = 0.75 \} \). For \( Ra = 10^3 \), one big single roll with low rotational speed is shown. At \( Ra = 10^4 \), the lines become horizontal with higher heat transfer close to the top of the cold wall.

3.3.7. For \( A = 1 \), and \( \Phi = 180^\circ \)

Fig. 9 shows the streamlines “top” and the isotherms “bottom” of the case \( \{ A = 1, \Phi = 180^\circ, \epsilon = 0.25, B_1 = 0.25, \text{ and } B_2 = 0.75 \} \). For \( Ra = 10^3 \), four small vortices are formed in the upper portion of the cavity near the heater plates. The lower portion is a dead zone. There is symmetry about the vertical axis of the cavity. At \( Ra = 10^4 \), the size of the vortices shrinks and the rotational speed gets higher. For \( Ra = 10^6 \), the dead zone dominates the space of the cavity and the flow is restricted to a small area near the heater plates. As \( Ra \) increases from \( 10^3 \) to \( 10^6 \), the isothermal lines do not change. They consist of horizontal lines parallel to the cold wall which indicates that conduction is dominant.

3.3.8. For \( A = 5 \), and \( \Phi = 0^\circ \)

Fig. 10 shows the streamlines “top” and the isotherms “bottom” of case \( \{ A = 5, \Phi = 0^\circ, \epsilon = 0.25, B_1 = 0.25, \text{ and } \} \).
$B_2 = 0.75$. For $Ra = 10^3$, the flow shows what is known as “Benard cells” which consist of four vortices distributed equally along the space of the cavity. As $Ra$ increases the rotational speed of the vortices gets higher but the streamlines stay the same. The isothermal lines for $Ra = 10^3$, show two plumes above each heater with symmetry along the vertical centerline of the cavity. For $Ra = 10^6$, the plumes are distorted and the isotherms at the top of the cavity are parallel to the cold wall.

3.3.9. For $A = 10$, and $\Phi = 0^\circ$

Fig. 11 shows the streamlines “top” and the isotherms “bottom” of case $[A = 10, \Phi = 0^\circ, \epsilon = 0.25, B_1 = 0.25 ~ and ~ B_2 = 0.75]$. For $Ra = 10^3$, the flow consists of four vortices distributed equally along the space of the cavity. As $Ra$ increases the cores of the vortices are elongated in the horizontal direction. The isothermal lines for $Ra = 10^3$, show two plumes above each heater with symmetry along the vertical centerline of the cavity. For $Ra = 10^6$, the plumes are
distorted and the isotherms at the top of the cavity are parallel to the cold wall.

3.2. Average Nusselt number

Table 2 shows the average Nusselt number, Nu for different cases. The Rayleigh number changes in the range $10^3 \leq \text{Ra} \leq 10^6$. The tilt angle from the horizontal, $\Phi$ changes in the range $0^\circ \leq \Phi \leq 180^\circ$. The aspect ratio, $A$ takes the values of 1, 5 and 10. Other parameters are kept constant at $B_1 = 0.25$, $B_2 = 0.75$, and $\varepsilon = 0.25$.

Table 2 Average Nusselt numbers for different cases.

<table>
<thead>
<tr>
<th>$A$</th>
<th>$B_1$</th>
<th>$B_2$</th>
<th>$\varepsilon$</th>
<th>$\Phi$</th>
<th>Avg. Nu for Ra showed</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>$10^3$ $10^4$ $10^5$ $10^6$</td>
</tr>
<tr>
<td>1</td>
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<td>0.75</td>
<td>0.25</td>
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<td></td>
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<td>1.365 2.197 3.37 5.24</td>
</tr>
</tbody>
</table>

Fig. 12 shows the effect of changing the tilt angle, $\Phi$ from $0^\circ$ to $180^\circ$ on the average Nusselt number, Nu at both $\text{Ra} = 10^3$, and $\text{Ra} = 10^4$. The average Nusselt increases with increasing $\Phi$ from $0^\circ$ up to $45^\circ$ then decreases to about unity at $\Phi = 180^\circ$.

Fig. 13 describes the same effect on the individual heaters at Rayleigh number of $\text{Ra} = 10^4$. For the lower heater, Nu increases with $\Phi$ from $0^\circ$ up to $75^\circ$ then decreases to a value

![Figure 10](image10.png) Streamlines “top” and isotherms “bottom” [$A = 5, \Phi = 0^\circ, \varepsilon = 0.25, B_1 = 0.25, \text{and } B_2 = 0.75$].

![Figure 11](image11.png) Streamlines “top” and isotherms “bottom” [$A = 10, \Phi = 0^\circ, \varepsilon = 0.25, B_1 = 0.25 \text{ and } B_2 = 0.75$].
close to unity at $\Phi = 180^\circ$. For the upper heater, $\text{Nu}$ decreased as $\Phi$ was increased from $0^\circ$ to $180^\circ$.

The effect of changing the aspect ratio, $A$ from $A = 1$ to $A = 10$ on the average Nusselt number is shown in Fig. 14 for constant values of Rayleigh number from $10^3$ to $10^6$. Increasing the aspect ratio, $A$ had the effect of decreasing the average Nusselt number $\text{Nu}$.

4. Nusselt number correlation

The average Nusselt number for the natural convection in a differentially heated enclosure with two localized heat sources is a function of “$A$, $B_1$, $B_2$, $\varepsilon$, $\Phi$, Ra and Pr”. It is quite difficult to find a single correlation containing all these parameters, and hence the correlation will be very complicated. For the case of air ($\text{Pr} = 0.71$), $10^3 \leq \text{Ra} \leq 10^6$, aspect ratio, $A = 1, 5, \text{and } 10$, size ratio, $\varepsilon$ of 0.25, $B_1 = 0.25$, $B_2 = 0.75$, and tilt angle, $\Phi$ from $0^\circ$ to $180^\circ$, a correlation was tried in the form

$$\text{Nu} = C_1 \cdot (B_1^2 + B_2^2) + C_2 \cdot \text{Ra}^{C_3}$$

(15)

where $C_1$, $C_2$ and $C_3$ depend on $A$ and $\Phi$ as given in Table 3.
A comparison between the numerical results and the proposed correlation showed that the correlation is suitable with a maximum deviation of 15% and more than 70% of the points are under 1% deviation. Figs. 15 and 16 show examples of this comparison for $B_1 = 0.25$, $B_2 = 0.75$, and $e = 0.25$. The graphs show a good match between the numerical results and the correlation (see Fig. 17).

5. Conclusion

Natural convection in a differentially heated enclosure was studied numerically. The governing equations were discretized using the center differencing method and the equations were solved with its conjugate boundary conditions by using Gauss–Seidel iteration method (line by line elimination technique). A regular grid with a size of $42 \times 42$ was used to illustrate the boundary layer heat transfer. The results are displayed in the form of isotherms and streamlines contours. These contours are drawn by Surfer 10 software.

The tilt angle from the horizontal, $\Phi$ affects Nu significantly. It was shown that the maximum Nu could be obtained by tilting the cavity $60^\circ$ from the horizontal. As $\Phi$ increases beyond $60^\circ$, Nu starts to decrease until it reaches its minimum value at $\Phi = 180^\circ$. It was also shown that at $\Phi = 150^\circ$, the effect of Rayleigh number on the average Nusselt number is very low. At tilt angle $180^\circ$ from the horizontal, the average Nusselt number becomes completely independent of Rayleigh number. The aspect ratio, $A$ affects the average Nusselt number, Nu. It was shown that by increasing the aspect ratio from 1 to 10, Nu decreased by about 23%. The position ratios, $B_1$ and $B_2$ affect Nu as it was shown that moving the heaters away from each other resulted in decreasing Nu by about 15%.

A correlation for the average Nusselt number was obtained to represent the numerical results and a comparison was also made between the numerical results and the proposed correlation. It was found that the correlation is suitable with maximum deviation of 15%.

References


