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Suppressing natural convection in a differentially heated square cavity with an arc shaped baffle

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Abstract

The problem of laminar natural convection heat transfer in a square cavity with an adiabatic arc shaped baffle is analyzed in this paper. As boundary conditions of the cavity, two vertical opposite walls are kept at constant but different temperatures and the remaining two walls are kept thermally insulated. Results are presented for a range of Rayleigh numbers, arc lengths of the baffle, and shape parameters of the baffle. Finally, parametric results are presented in terms of isothermal lines and streamlines. It is identified that flow and thermal fields are modified by the blockage effect of the baffle. The degree of flow modification due to blockage is enhanced by increasing the shape parameter of the baffle.

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1. Introduction

Modification of heat transfer in cavities due to introduction of obstacles and fins attached to the wall(s) has received some consideration in recent years [1–9]. One of the methods of suppressing natural convection in storage tanks is to add ribs on the inner walls of the tank. The ribs restrict the fluid motion in the storage tank and accordingly decrease the rate of heat transfer.

Natural convection in an air-filled differentially heated inclined square cavity with a diathermal and thin partition placed at the middle of its cold wall was numerically studied [1] for Rayleigh numbers

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Heat transfer reductions of up to 47% in comparison to the cavity with no partition were observed due to the suppression of convection. Natural convection in a square cavity with a conducting partition positioned at the middle of its hot wall and with perfectly conducting horizontal walls was numerically studied [2]. For low values of the partition-to-fluid thermal conductivity ratio, reductions in heat transfer relative to the case of cavity with no partition was observed for Rayleigh numbers $10^4$–$10^5$. Sezai et al. [3] investigated the effect of attaching poorly conducting ribs on a vertical heated plate. They concluded that adding ribs to the surface can reduce the rate of free convection heat transfer by as much as 75% when compared with a bare plate. Nag et al. [4] studied the effect of a horizontal thin partition (both of infinite thermal conductivity and insulated) placed on the hot wall of a differentially heated square cavity. The range of the Rayleigh number was $10^3$–$10^6$ and the three partition lengths placed at three positions were studied. It was concluded that for a partition of infinitely high thermal conductivity, the Nusselt number on the cold wall was greater than the case with no fin irrespective of the position of the partition on the hot wall. Shi and Khodadi [5] performed a numerical study in a square cavity due to a thin fin on the hot wall. They concluded that heat transfer capacity on the anchoring wall was always degraded; however, heat transfer capacity on the cold wall without the fin can be promoted for high Rayleigh numbers and with the fins placed in the vicinity of the insulated walls. Natural convection in a differentially heated slender rectangular cavity (aspect ratio of 15) with multiple conducting fins on the cold wall was reported by Scozia and Frederick [6] for Rayleigh numbers $10^3$–$10^5$. As the inter-fin aspect ratio was varied from 20 to 0.25, the flow patterns evolved considerably and the average Nusselt number exhibited maximum and minimum values whose locations depend on the of $Ra$. Facas [7] reported results of a computational study of natural convection (Grashof number range of $9 \times 10^3$–$10^5$) inside an air filled slender cavity (aspect ratio 15) with fins/baffles—0.1, 0.3, and 0.5 of the cavity width—attached along both the heated and cooled sides of the cavity. For the fin length of 0.1, multicellular flow structure was observed. However, for longer fin lengths, the flow broke down into secondary recirculations, in addition to the primary recirculation. As a result, higher heat transfer rates across the two sides of the cavity were observed. Edwards [8] determined the effect of inserting parallel vertical walls on the onset of convection in a cavity heated from below and cooled from above. It was found that the geometry, (aspect ratio) has significant effect on the onset of convection in cavities. Tasnim and Collins [9] performed a numerical analysis to investigate the effect of attaching a highly conducting thin baffle on the hot wall of a differentially heated square cavity. They concluded that adding baffle on the hot wall can increase the rate of heat transfer by as much as 31% compared with a wall without a baffle. By anchoring a baffle, the heat transfer capacity on the hot wall is always increased for $Ra=10^5$, irrespective of the position of the baffle. But for $Ra=10^4$, the heat transfer can be promoted with the baffle placed closer to the insulated (horizontal) walls.

The authors are not aware of any work that has been performed in a square cavity with an arc shaped baffle. In the present study a detailed analysis was performed to check the effect of arc length ($s$), and shape parameter ($\theta$), of an adiabatic baffle of finite thickness (1% of $W$), on heat transfer that is placed in the middle of a square cavity. A range of Rayleigh numbers ($10^5 \leq Ra \leq 10^5$) is considered for air as the working fluid.

2. Problem formulation

Fig. 1 shows the schematic diagram of the problem under consideration and the coordinate system. The problem is considered to be two dimensional. The two horizontal walls (i.e., top and bottom walls)
are insulated, whereas the right wall is maintained at a higher temperature \( T_h \) relative to the left wall \( T_c \). An arc shaped baffle is placed midway between the vertical walls. \( W \) is the width of the cavity and \( r_0 \) is the radius of the arc. \( s \) and \( \theta \) are representing the arc length and the shape parameter of the baffle, respectively. The flow field is considered to be steady and the fluid is incompressible. Thermophysical properties of the fluid are assumed constant, except the density variation in the buoyancy term, i.e., Boussinesq approximation is valid. The dimensionless form of the governing equations can be obtained via introducing dimensionless variables. These are defined as follows:

\[
X = \frac{x}{W}, \quad Y = \frac{y}{W}, \quad U = \frac{uW}{\alpha}, \quad V = \frac{vW}{\alpha}, \quad P = \frac{pW^2}{\rho \alpha^2}, \quad \Theta = \frac{T - T_c}{T_h - T_c}, \quad S = \frac{s}{W}. \tag{1}
\]

Variables \( u, v \) and \( T \) are the velocity components in the \( x, y \) direction and temperature, respectively. Quantities \( \rho \) and \( \alpha \) are the density and thermal diffusivity of the fluid, respectively. Based on the dimensionless variables defined in Eq. (1), the non-dimensional equations for the conservation of mass, momentum and energy equations are

\[
\frac{\partial U}{\partial X} + \frac{\partial V}{\partial Y} = 0 \tag{2}
\]

\[
U \frac{\partial U}{\partial X} + V \frac{\partial U}{\partial Y} = - \frac{\partial P}{\partial X} + Pr \left( \frac{\partial^2 U}{\partial X^2} + \frac{\partial^2 U}{\partial Y^2} \right) \tag{3}
\]

\[
U \frac{\partial V}{\partial X} + V \frac{\partial V}{\partial Y} = - \frac{\partial P}{\partial Y} + Pr \left( \frac{\partial^2 V}{\partial X^2} + \frac{\partial^2 V}{\partial Y^2} \right) + Ra Pr \Theta \tag{4}
\]

\[
U \frac{\partial \Theta}{\partial X} + V \frac{\partial \Theta}{\partial Y} = \frac{\partial^2 \Theta}{\partial X^2} + \frac{\partial^2 \Theta}{\partial Y^2} \tag{5}
\]
In the energy equation, the viscous dissipation terms are neglected and the reference temperature in the buoyancy term was taken to be equal to $T_c$. Parameters $Ra$ and $Pr$ are the Rayleigh and Prandtl numbers, respectively. These are defined as

$$Ra = \frac{g\beta(T_h - T_c)}{\nu x}, \quad Pr = \frac{\nu}{\alpha}.$$  \hspace{1cm} (6)

Quantities $\nu$ and $\beta$ are the kinematic viscosity and thermal expansion coefficient of the fluid, respectively. The dimensionless forms of the boundary conditions are

On the left wall ($X = 0$): $U = V = 0$, $\theta = 0$

On the right wall ($X = 1$): $U = V = 0$, $\theta = 1$

On the bottom of the wall ($Y = 0$): $U = V = 0$, $\partial \theta / \partial Y = 0$

On the top wall ($Y = 1$): $U = V = 0$, $\partial \theta / \partial Y = 0$

On the baffle: $U = V = 0$, $\partial \theta / \partial Y = 0$  \hspace{1cm} (7)

![Streamlines](image)

$S=0.25$, $\Psi_{\text{max}}=0.00177$

$S=0.25$, $\Psi_{\text{max}}=0.00173$

$S=0.75$, $\Psi_{\text{max}}=0.00171$

![Streamlines](image)

$S=0.25$, $\Psi_{\text{max}}=0.00205$

$S=0.5$, $\Psi_{\text{max}}=0.00196$

$S=0.75$, $\Psi_{\text{max}}=0.00193$

Fig. 2. (a) Streamlines for $Ra=10^3$ and $\theta=45^\circ$. (b) Streamlines for $Ra=10^3$ and $\theta=180^\circ$. 
In the analysis the local Nusselt number is defined as

\[ Nu_l = -\frac{\partial \Theta}{\partial X} \bigg|_{X=1}, \]

and the average Nusselt number as

\[ Nu_{av} = \int_0^1 Nu_l dY. \]

3. Method of solution

The steady state governing equations were solved iteratively by the finite-volume method with collocated variable arrangement. Solution procedures with detail algorithm are available in Refs. [10,11]. Four grid sizes (32x32, 64x64, 100x100, and 128x128) are chosen for the present analysis. Average Nusselt number for all grid sizes are monitored at \( Ra=10^4 \), \( S=0.5 \), and \( \theta=120^\circ \). The magnitude of average Nusselt number at 128x128 grids shows a very little difference with the result obtained at 100x100 grids. For rest of the calculation in this paper

![Streamlines](image-url)


Fig. 3. (a) Streamlines for \( Ra=10^4 \) and \( \theta=45^\circ \). (b) Streamlines for \( Ra=10^4 \) and \( \theta=180^\circ \).
we chose a grid size of 128×128 for better accuracy. For code validation tests, see Tasnim and Collins [9].

4. Result and discussions

4.1. Flow and thermal field

Streamlines inside the cavity \((Ra=10^3)\) are shown in Fig. 2(a) and (b) for three different arc lengths, \(S=0.25, 0.5,\) and 0.75, and two different shape parameters, \(\theta=45^\circ\) and 180°, respectively. The plots are arranged going from left to right with the ascending value of \(S\). For \(S=0.25\), a weak counterclockwise circulation appears due to the effect of upward moving fluid near the hot wall and downward moving fluid near the cold wall. This is the primary circulation and is similar for both \(\theta=45^\circ\) and 180°. The presence of an adiabatic baffle introduces three secondary circulations at \(\theta=45^\circ\). A small stream of fluid rotates clockwise around the baffle followed by two counterclockwise secondary circulation over and below it. However, for \(\theta=180^\circ\), no circulation appears between the primary circulation below the baffle and the fluid stream rotates around the baffle. An

![Streamlines](image)

Fig. 4. (a) Streamlines for \(Ra=10^5\) and \(\theta=45^\circ\). (b) Streamlines for \(Ra=10^5\) and \(\theta=180^\circ\).
increase in the baffle length gradually decreases the space between the isothermal wall and the baffle and causes the primary stream of fluid to thin out. A combination of higher arc length (S=0.75) and lower shape parameter (θ=45°) divide the flow pattern inside the cavity into two counterclockwise vortices. At θ=180°, the stream of fluid below the baffle circulates in a small region confined by the curvature of the baffle (S=0.75). A considerable portion of the fluid below the baffle remains stagnant for all S and θ=180°. Pattern of the flow remains unchanged at Ra=10^3 (see Fig. 3(a) and (b)), but the strength of the circulation is higher (see value of Ψ_{max}) than the previous case (Ra=10^3). Flow pattern is characterised by asymmetric secondary vortices at Ra=10^5 (see Fig. 4(a) and (b)) due to the high convection current inside the cavity. For θ=45° and S=0.5, a large stagnant region is observed over and below the baffle. However, for θ=180°, a different type of flow pattern is observed. Primary circulation occupies a large portion of the cavity for all S and the core of the cavity shift towards the upper adiabatic wall with increasing arc length.

4.2. Temperature fields

Fig. 5(a) and (b), Fig. 6(a) and (b), and Fig. 7(a) and (b) show the contours of dimensionless temperature (θ) for Ra=10^3,10^5, and 10^5 with a baffle of three different arc lengths S=0.5, 0.5,
and 0.75, and two different shape parameters $\theta=45^\circ$ and $180^\circ$, respectively. Since the Boussinesq approximation was adopted in this study, the energy equation does affect the momentum equation. Weaker circulation inside the cavity shows a very little effect on thermal field for $\theta=45^\circ$ (Fig. 5(a) and (b)). Isothermal lines are nearly parallel to the hot (or cold) wall, which is similar to conduction-like distribution. Some deviation of isothermal lines are observed at $S=0.75$ for $\theta=180^\circ$ due to the appearance of a localised circulatory vortex (see Fig. 2(b)) below the baffle.

As natural convection is strengthened (Fig. 6(a) and (b)), temperature contours show slight deviation from the pure conduction case with the contour lines becoming skewed. Under high Rayleigh number condition (Fig. 7(a) and (b)), the degree of distortion from the pure conduction case is very marked and the isotherms are more packed next to the side walls. Two thermal spots one at the bottom of the hot wall and another at the top of the cold wall appear in the isothermal lines. Magnitude of the heat flux is higher at these two spots due to higher temperature gradient. The distortion of isothermal lines appears due to the high convective current inside the cavity. As mentioned before, a baffle can weaken the fluid motion in the area around the baffle, and thus, decreased heat transfer capability is expected. The temperature contours under the baffle are less dense than those above the baffle. This implies less heat transfer on the bottom of the baffle than on the top surface for all the cases as mentioned above.

![Fig. 6. (a) Isothermal lines for $Ra=10^4$ and $\theta=45^\circ$. (b) Isothermal lines for $Ra=10^4$ and $\theta=180^\circ$.](image-url)
4.3. Variation of energy transfer

To understand the energy transport mechanisms inside the channel, according to Landau and Lifshitz [12], the energy flux density vector is calculated as follows:

\[ \mathbf{E} = e_x \hat{i} + e_y \hat{j} = \rho u \left( \frac{u^2}{2} + C_p T \right) - k \nabla T - \mathbf{u} \cdot \Sigma \]

(10)

where \( \Sigma \) is the viscous stress tensor, \( e_x \) and \( e_y \) are the components of \( \mathbf{E} \). The magnitude of the energy flux density vector is the amount of energy passing, in unit time, through a unit area perpendicular to the

Fig. 7. (a) Isothermal lines for \( Ra=10^5 \) and \( \theta=45^\circ \). (b) Isothermal lines for \( Ra=10^5 \) and \( \theta=180^\circ \).

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Fig. 8. Energy streamlines at \( Ra=10^3 \).
direction of fluid velocity (Landau and Lifshitz [12]). At any particular space location (for example, \( P(x_0,y_0) \)) inside the channel, \( \mathbf{E} \) of that location is tangent to the energy streamline passing through the point \( P(x_0,y_0) \).

Fig. 8(a) and (b), Fig. 9(a) and (b), and Fig. 10(a) and (b) show the energy streamline inside the cavity for \( \theta=45^\circ \) and \( 180^\circ \), for a single arc length (\( S=0.5 \)), and for \( Ra=10^3, 10^4, \) and \( 10^5 \), respectively. Energy streamlines originate at the hot wall, pass horizontally along the cavity, and terminate at the cold wall. For \( Ra=10^3 \) (see Fig. 8(a) and (b)) the presence of a baffle has no influence on the energy streamlines. At higher \( Ra=10^4 \), a portion of energy is seen to be trapped above and below the baffle at \( \theta=45^\circ \), which changes dramatically for \( \theta=180^\circ \). Below the baffle, no trapped energy is found at this shape parameter. A thin corridor just below the top wall of the cavity is identified as a passage of free energy transport. This passage becomes thinner and closer to the top wall at high \( Ra \) (see Fig. 10(a) and (b)).

### 4.4. Average heat transfer distribution

In order to evaluate how the presence of the baffle affects the heat transfer rate along the hot wall, average Nusselt number is plotted (hot wall) as a function of baffle arc length (\( S \)), in Fig. 11, for \( Ra=10^4 \), and for three different shape parameters (\( \theta=45^\circ, 90^\circ, \) and \( 120^\circ \)). It is observed that average Nusselt number decreases with the increase of \( S \) and \( \theta \). A closer examination of Fig. 11 does reveal that up to a length (\( S \)) of 0.3, average Nusselt number is the same for all three cases (i.e., the shape parameter of the
baffle is insignificant for average heat transfer). But after this, average heat transfer decreases with the increase of baffle length and shape parameter. Similar trends are observed for $Ra=10^3$ and $10^5$ which are not shown here. Figs. 12–14 show the average Nusselt number distribution as a function of shape parameter ($\theta$), for three different Rayleigh numbers ($Ra=10^3$, $10^4$, and $10^5$), and for three different arc lengths ($S=0.25$, 0.5, and 0.75). For $Ra=10^3$, average Nusselt number is almost constant for $S=0.25$, i.e., the shape factor has no effect for this arc length. But for $S=0.5$ and 0.75 average Nusselt number

Fig. 11. Average Nusselt number as a function of arc length.

Fig. 12. Average Nusselt number as a function of shape parameter at $Ra=10^3$. 
decreases with the increase of shape factor. For $Ra=10^4$ average Nusselt number is almost constant for the lowest arc length ($S=0.25$) except $\theta=180^\circ$. Average Nusselt number decreases with the increase of shape parameter for $S=0.5$ and 0.75. Similar patterns are observed for $Ra=10^5$ (convection dominating conduction). The average Nusselt number for the hot wall becomes smaller with the increase of baffle’s arc length and shape parameter, regardless of the Rayleigh number. This is because the baffle can block the flow near the baffle and reduce the corresponding convection in that area.

Fig. 13. Average Nusselt number as a function of shape parameter at $Ra=10^4$.

Fig. 14. Average Nusselt number as a function of shape parameter at $Ra=10^5$. 
5. Conclusion

Buoyancy induced flow and heat transfer inside a square cavity due to a thin arc shaped baffle was investigated numerically. Placing an adiabatic baffle in a differentially heated cavity generally modifies the counterclockwise rotating primary vortex that is established due to natural convection. Flow and thermal field’s modifications are observed due to the blockage effect of the baffle that directly depends on the arc length and shape parameter of the baffle. Viewing the maximum value of the stream function field as a measure of the strength of flow modification, it is shown that for high Rayleigh numbers the flow field is enhanced regardless of the fin’s arc length and shape parameter. By introducing an adiabatic baffle, the heat transfer capacity on the hot wall is always degraded. The deviation of the average Nusselt numbers on the hot wall increases with the increase of baffle’s arc length and shape parameter.

Nomenclature

- \( E \)  energy flux density vector
- \( e_x \) component of \( E \) at \( x \) direction
- \( e_y \) component of \( E \) at \( y \) direction
- \( k \) thermal conductivity of fluid
- \( Nu_l \) local Nusselt number
- \( Nu_{av} \) average Nusselt number
- \( S \) dimensionless arc length of the baffle
- \( T_c \) temperature of the left (cold) wall
- \( T_h \) temperature of the right (hot) wall
- \( U \) dimensionless axial velocity
- \( V \) dimensionless vertical velocity

Greek Symbol

- \( \sum \) viscous stress tensor
- \( \theta \) Shape parameter of the baffle
- \( \Theta \) dimensionless temperature, i.e., \((T-T_c)/(T_h-T_c)\)

References