Heat Transfer and Flow Characteristics inside Air Filled Cavities

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ABSTRACT

In the present work the heat transfer and flow characteristics for the case of natural convection inside air filled cavities were studied numerically under steady state conditions. Two cases were studied. The first case was of closed cavities, in which the effect of Rayleigh number (Ra), and aspect ratio on heat transfer and flow characteristic inside the cavity was studied for constant temperature boundary conditions on the two opposing vertical walls. The second case was of partially opened vertical walls. The second case was of partially opened cavities, in which the effect of aperture position and size on the flow and heat transfer characteristics was studied where constant heat flux on the vertical opposing wall to the aperture was applied. The continuity, momentum, and energy equations for two dimensional laminar steady flow were solved numerically using the Ansys finite element computer code. For the first case, the results showed that the contribution of the natural convection mechanism in the heat transfer process increased with respect to conduction with either the increase in Ra or the decrease in the aspect ratio, hence the heat transfer was enhanced. For the second case, better heat transfer was obtained for the bottom position and maximum size of the aperture.

Key words: Air filled cavities/ Heat transfer characteristics/ Flow characteristics.

INTRODUCTION

Natural convection flow in macro scale enclosures has many thermal engineering applications such as in double-glazed windows, solar energy collectors, cooling devices for electronic instruments, gas-filled cavities around nuclear reactor cores and building insulation, cooling of radioactive waste containers, ventilation of rooms, fire prevention, dispersion of waste heat in estuaries and crystal growth in liquid metals, and electronic chips. So, macro scale free convection has been profoundly studied in last several decades (1-4). Over the past twenty years, a revolution in electronics has taken place. With the emergence of large-scale integrated circuits, the micro scale free convection plays more and more important role in engineering applications such as microelectronic components cooling and micro machined convective accelerometer (5). A review of the literature was given by Ostrach (6) and Raithby and Hollands (7).

Zeng-Yuan Guo et al. (8) studied numerically the size effect on free convection in a square cavity. The relative importance of three control forces of free convection, inertial force, viscous force and buoyancy force was discussed. It was noted that when Ra is smaller than $10^3$, the heat conduction prevails over convection for the case of very small Ra, in other words, very small scale. It is also found that Nu increased more quickly with Ra in the range of $10^3$ to $10^5$ than that in the range of $10^5$ to $10^7$. They attributed this to the variation of the relative importance of the three kinds of control forces of free convection.
The laminar natural convection in shallow inclined air rectangular cavities was studied both numerically and experimentally by El-Sherbiny et al. \(^{(9)}\). The two-dimensional governing equations for mass, momentum and energy conservation were solved using an implicit finite-difference scheme. He reported that: For \(Ra = 10^3\), the temperature distribution was almost linear indicating the conduction regime. For \(Ra \geq 10^4\), temperature was constant in the core of the cavity and a higher temperature slope occurred near the isothermal surfaces indicating the laminar boundary layer regime. As \(Ra\) increased, the temperature slope increased indicating higher heat transfer.

Barakos and Mitsoulis \(^{(10)}\) studied the laminar and turbulent flow of air inside a rectangular cavity. They used the finite volume approach to solve the mathematical model. They presented the isotherms, streamlines and velocity vectors for different values of \(Ra\). They showed that for lower values of \(Ra\), the isotherms were vertical which indicated the heat transfer by conduction. By increasing \(Ra\), the mechanism of heat transfer converted from conduction to convection.

Dillon et al. \(^{(11)}\) studied the tall differentially heated rectangular cavity by the commercial code Comsol. In this cavity, air is circulated by the aid of natural circulation. Four aspect ratios \(A\) were studied namely: 8,15,20,33. They compared their results with the literature. They reported that: at small \(Ra\) the flow was dominated by conduction. As \(Ra\) was increased the flow became unstable, first resulting multicellular secondary flow pattern, and then if \(Ra\) was further increased the flow became turbulent. They reported that for \(A=15\) the critical \(Ra_c = 1 \times 10^5\), for \(A=20\) \(Ra_c = 3.2 \times 10^4\), for \(A=33\) \(Ra_c = 5.8 \times 10^3\). For \(A=8\) no critical value for \(Ra\) could be determined. Another several studies were carried out on cavities with aspect ratio \(A > 1\) \(^{(12-16)}\).

Dogan et al. \(^{(17)}\) studied numerically the heat transfer by natural convection inside partially opened cavities with one wall heated under different opening ratios, tilt angles, and different aspect ratios. Their results showed that the average heat transfer coefficient increases and the average wall temperature decreases with the increase in opening ratio and decrease in tilt angel. Best heat transfer was obtained for opening ratio of 0.75, tilt angel of 10\(^o\), and aspect ratio of one.

**PRESENT STUDY AND CODE VALIDATION**

The present study consists of two parts:

First the effect of \(Ra\) and aspect ratio was studied independently on flow and heat transfer characteristics inside air filled cavities. Using the results of the first part, the aim of the second part was to determine the best design for partially opened cavities from the point of view of aperture position and size. Ansys finite element computer code was used to accomplish this purpose. So, the importance of this study lies in two points namely:

**First**: is to show the effect of aspect ratio \(A\) on heat transfer and flow characteristics without changing \(Ra\), i.e independent of \(Ra\). Some of the previous researches did not take into account the change in \(Ra\) with changing \(A\) due to the fact that they changed \(A\) by changing the characteristic length. In the present study changing \(Ra\) was done by changing the length of a square cavity, hence \(A\) was constant and equal to one. Changing \(A\) was done by changing the height \(H\) of rectangular cavity, keeping the width \(W\), which is the characteristic length, and hence \(Ra\) constant. In both of the two cases of changing \(Ra\) or \(A\) constant temperature boundary conditions was used.

**Second**: is to show the optimal design for partially opened cavity from the point of view of heat transfer using the results of the first part. In this study \(Pr\) is kept at the standard value of 0.71. During calculations the ambient temperature was selected to be constant and equal to 293K.

The computer code was validated by comparison with the previously obtained numerical results for the natural flow in a square cavity Barakos and Mitsoulis \(^{(10)}\) and with the results of Dogan et al.
(17) for partially opened cavities. Good agreement was obtained between these results and the results of the present study.

**NUMERICAL ANALYSIS**

Figure (1) shows the geometry and boundary conditions of the case study. The flow domain is the interior of a 2D rectangular cavity of height $H$ and width $W$. The horizontal walls of the cavity are assumed to be perfectly adiabatic ($q = 0$), while the vertical walls are subjected to constant temperature values of 320 K and 280 K. The case study is considered to be 2 D with the following assumptions:
- Viscous dissipation is negligible.
- Gravity acts in the vertical direction $Y$.
- Properties are constant, whereas density changes with temperature.
- Radiation heat transfer is assumed to be negligible.

Under the previous assumptions, the governing equations for two dimensional laminar flow at steady state conditions will be:

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

$$
u \frac{\partial \rho U}{\partial X} + v \frac{\partial \rho U}{\partial Y} = -\frac{\partial P}{\partial X} + \mu \left( \frac{\partial^2 U}{\partial Y^2} + \frac{\partial^2 U}{\partial X^2} \right) \quad (\text{X Momentum})
$$

$$\nu \frac{\partial \rho V}{\partial X} + v \frac{\partial \rho V}{\partial Y} = -\frac{\partial P}{\partial Y} + \mu \left( \frac{\partial^2 V}{\partial Y^2} + \frac{\partial^2 V}{\partial X^2} \right) + \rho g \beta(T - T_e) \quad (\text{Y Momentum})
$$

$$\frac{\partial \rho CT}{\partial X} + \frac{\partial \rho CT}{\partial Y} = K \left( \frac{\partial^2 T}{\partial X^2} + \frac{\partial^2 T}{\partial Y^2} \right) \quad (\text{Energy})
$$

In the calculation, the ambient temperature was selected to be constant and equal to 20°C. Velocity values were considered zero at all of the four sides for no slip condition. The previous equations were solved by the ANSYS commercial finite element code. The convergence criterion used was that the relative error should be less than $10^{-4}$.

**Fig.(1): Schematic representation of the closed cavity.**
RESULTS AND DISCUSSION

Effect of Ra:

Figure (2) shows the temperature contours for the case of square cavity at different Ra. Ra was altered by altering the dimension of the cavity. Three values of Ra of $10^3$, $10^4$, and $10^5$ were used. From this figure it can be shown that the temperature contours are symmetric with respect to the vertical centreline for all of the values of Ra. It can be shown also that the borders of temperature regions tend to be vertical inside almost the whole of the cavity for the case of small Ra = $10^3$.

This indicates that heat is transferred by conduction. As Ra increases to $10^4$, the shape of the temperature contours deviates from being vertical and this indicates that heat transfer mechanism converts from conduction to convection. Further increase of Ra to $10^5$ causes the temperature contours to be horizontal in the center of the cavity and to be vertical only inside the very thin boundary layers near the two vertical walls. For the purpose of validation of the results of the computer code, the previous results were compared with the results of Barakos and Mitsoulis (10) for the natural flow in a square cavity (10). The results of Barakos and Mitsoulis (10) are shown in Fig.(3). This figure shows the isotherms for different values of Ra. Same remarks can be deduced from this figure for the vertical change of the isotherms with the increase in Ra as discussed above.

Fig.(2): Effect of Ra on temperature contours of the flow inside square cavity.

Fig.(3): Results of Barakos and Mitsoulis (10) for the effect of Ra on the isotherms of the flow of air inside square cavity.
Figure (4) shows the velocity vectors for the three previously mentioned values of Ra. For low value of Ra, i.e Ra = $10^3$, the flow is characterized with a single vortex in the center of the cavity (Fig.4.a). As Ra increases to $10^4$, the vortex shape changes to be approximately elliptic (Fig.4.b). Further increase of Ra to $10^5$ causes the vortex to change to be in the form of two elliptical vortex attached to each other as shown in Fig.4.c). Increasing Ra causes also the vortex region in the middle of the cavity to widen towards the walls.

Figure (5) shows the temperature distribution in the mid plane for the three previously mentioned values of Ra. For Ra = $10^3$, the temperature profile is linear throughout the whole of the cavity which indicates the heat transfer by conduction. For Ra > $10^3$, almost constant temperature is obtained in the core of the cavity, whereas higher temperature slope is obtained near the isothermal surface due to the laminar boundary layer. As Ra increases, the region of constant temperature becomes wider and higher temperature slope is obtained indicating higher heat transfer.

Figure (6) shows the velocity component U in y direction in the mid plane of the cavity. For all of the Ra values, the velocity increases from zero at the hot surface to a maximum value, then decreases to zero and remains constant in the core of the cavity, then decreases to a minimum value near the cold wall and again becomes zero at the cold wall for no slip condition. By increasing Ra, the local maximum and minimum increases in values and shifts towards the hot and cold walls respectively.
Effect of Aspect Ratio

Figure (7) shows the temperature contours for three different values of aspect ratio which are 1, 5, and 10. Ra is kept constant for the three values of the aspect ratio by the fact that same temperature values are applied on the two sides of the cavity and the characteristic length W is the same for all of the aspect ratios, i.e. the aspect ratio is changed by varying the height H. Figure 7 shows the increasing of the aspect ratio, causes the temperature contour to change from almost horizontal to almost vertical. This indicates that the heat transfer mechanism changes from convection to conduction by increasing the aspect ratio and finally conduction will be dominant for A=10.

(a): A = 1  
(b): A = 5  
(c): A = 10

Fig.(7): Effect of aspect ratio on temperature contours of the flow inside rectangular cavity.
Figure (8) shows the temperature distribution in the mid plane for the three previously mentioned values of A. From this figure, it can be shown that almost constant temperature is obtained in the core of the cavity, whereas higher temperature slope is obtained near the isothermal surface due to the laminar boundary layer for all of the cases. As the aspect ratio A decreases, the region of constant temperature becomes wider and higher temperature slope is obtained indicating higher heat transfer.

Fig. (8): Temperature profiles in the mid-plane of the cavity for different values of A.

Figure (9) shows the velocity component U in y direction in the mid plane of the cavity for the previously mentioned different values of aspect ratio. This figure shows that (as discussed before in the case of the effect of Ra); the velocity has a maximum value near the hot wall and minimum value near the cold one, and a region of almost zero value in the core region of the cavity. By decreasing A, the maximum and minimum velocity increases in value, shifts towards the hot and cold wall respectively, and the core region of zero value gets wider.

Fig. (9): Velocity profiles in the mid-plane of the cavity for different values of A.
Optimal Design for Partially Opened Cavity:

Figure (10) shows Schematic representation of the partially opened cavity problem. The vertical left side is subjected to constant heat flux, whereas the opposite vertical one has an aperture for which the heat size and position from the point of view of heat transfer are to be determined. The other two horizontal sides are considered to be adiabatic.

To get the optimal design for partially opened cavities from heat transfer point of view, obtained results are to be used. With respect to the aspect ratio, it has been shown that for the studied values, best heat transfer was for $A = 1$ as heat transfer by convection is the dominant in the case of square cavity. Maximum $Ra$ and hence maximum heat transfer rate can be obtained by assigning the maximum length in the design of the square cavity in view of the allowable space which depends on the application for which optimal design of the cavity is to be determined.

For the purpose of validation of the computer code, grid independence was verified at first. Then comparison between the present results and results of Dogan et al. \(^{17}\) was made for the case of top opening of aperture ratio (aperture length/cavity length) of 0.5 as shown in Fig. (11). In this figure, the vertical axis represents the ratio of Nusselt number value at a certain heat flux divided by Nusselt number value ($Nu_1$) at heat flux of 40 $W/m^2$. This is done to eliminate the effect of the difference between the used characteristic length in the present study and that which was used by Dogan et al. \(^{17}\). As shown from this figure, good agreement between the two results was obtained where the maximum difference between the present results and the numerical results of Dogan et al. \(^{17}\) did not exceed 13.8%. This was considered to be a sufficient verification of the numerical procedure applied in the present study.
To determine the optimal size and position of the aperture which gives the best heat transfer between the cavity and surrounding air, three different aperture ratios AR were used, namely AR of 0.25, 0.5, and 0.75. Numerical analysis was conducted under different heat fluxes ranging from 40 W/m² up to 320 W/m², with increments of 40 W/m². Comparison was made between the top and bottom positions of the aperture.

Figure (12) shows the effect of aperture positions on the flow and heat transfer characteristics for AR of 0.5. From Fig. 12 (C,D) it can be shown that for the case of bottom vented, after touching the hot surface air moves through longer distance inside the cavity, and better air circulation is obtained.

This affects the temperature contours as shown in Fig.12 (A,B) where lower values of temperature are obtained. Hence, higher average Nusselt number is expected for the case of lower venting.

Figure (13) shows the influence of heat flux on average Nusselt number (Nu), and comparison is made between top and bottom venting for the case of aperture ratio (AR) of 0.5. From this figure it can be shown that, the average Nusselt number increases with the heat flux increase. The average Nusselt number increases rapidly at lower heat fluxes, then it reaches almost constant value in the two cases. The figure shows also that the average Nusselt number is higher in the case of bottom vented cavity. This can be attributed to the fact that for the case of bottom venting, air moves through longer distance inside the cavity resulting in better heat transfer. To determine the effect of aperture ratio (AR) on heat transfer, Nusselt number is plotted at different values of heat flux and comparison is
made among three different values of AR namely: 0.25, 0.5, and 0.75 for bottom vented cavity. These results are shown in Fig.(14). From this figure it can be shown that the average Nusselt number increases rapidly at low heat fluxes, then it reaches almost constant value for all AR. The figure shows also that the average Nusselt number decreases with decrease of AR. This can be attribute to the fact that when the aperture ratio AR decreases air circulation through the cavity slows down resulting in lower heat transfer rate in comparison with higher AR. Better heat transfer is obtained at higher values of AR.

Fig.(13): Effect of aperture position on Nusselt number for various heat flux.

Fig.(14): Effect of aperture size on Nusselt number for various heat flux.
CONCLUSION

The study Heat transfer and flow characteristics inside air filled cavities is consisted of two parts:

**First part:** the effect of Ra and aspect ratio A on the flow and heat transfer characteristics inside air filled cavity was studied. Three aspect ratios of 1.5, and 10 were used.

**Second part:** using the results of the first part in study of optimal design of partially opened cavity at which the effect of aperture geometry on heat transfer rate was determined. Three opening ratios of 0.25, 0.5, and 0.75 with comparison between top and bottom venting were studied.

The results show that:

1- By either increasing Ra at constant aspect ratio A or decreasing aspect ratio A at constant Ra the mechanism of heat transfer changed from conduction to convection with increase in the heat transfer rate.

2- Either for low Ra or high aspect ratio A, the temperature profile tends to be linear throughout the cavity. By increasing Ra or decreasing A, the slope of temperature profile increases near the walls and tends to be constant in the central region.

3- The velocity profile has a maximum value near the wall. Either with the increase in Ra or the decrease in A, the maximum value increases and its position shifts towards the walls.

4- The flow in the cavity is in single cell with vortex in the center of the cavity for low Ra. With the increase in Ra, the shape of the vortex changes to be elliptic. By further increase of Ra, the elliptic vortex finally splits into two vortices which move towards the corners.

5- Bottom vented cavity resulted in better heat transfer than top vented one, and the increase in aperture ratio resulted in better heat transfer.

6- As a consequence, best heat transfer was obtained for aspect ratio of one with bottom venting, and aperture ratio of 0.75.

**NOMENCLATURE**

- **A**: Aspect ratio of the cavity (height/width)
- **AR**: Aperture ratio = Aperture length/cavity length
- **C**: Specific heat
- **Gr**: Grashof number = gβΔTH³/ν²
- **H**: Height of the cavity
- **Nu**: Nusselt number = hw/γ
- **Nu1**: Nusselt number at minimum heat flux
- **q**: Heat flux
- **Pr**: Prandtl number = 𝜈/𝛼
- **Ra**: Rayleigh number = Gr.Pr
- **TC**: Temperature of the cold side
- **Th**: Temperature of the hot side
- **u**: Velocity component in x direction
- **U**: Dimensionless velocity = 𝑣𝐿/𝛼
- **v**: Velocity component in y direction
- **W**: Width of the cavity
- **x**: Distance in the transverse direction
- **y**: Distance in the longitudinal direction

**Greek letters**

- **β**: Coefficient of expansion = 1/T
- **ΔT**: Temperature difference
- **γ**: Thermal conductivity
- **α**: Thermal diffusivity
- **ν**: Kinematic viscosity

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<th>Symbol</th>
<th>Description</th>
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<tr>
<td>β</td>
<td>Coefficient of expansion</td>
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