EXPERIMENTAL INVESTIGATION OF NATURAL CONVECTION IN AN ENCLOSURE WITH PARTIAL PARTITIONS AT DIFFERENT ANGLES

by

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Original scientific paper
DOI: 10.2298/TSCI120526017G

Natural convection heat transfer in a partially partitioned enclosure has been investigated experimentally using Mach-Zehnder Interferometry technique. The top and bottom of the enclosure are insulated while one of the vertical walls is heated isothermally. The partitions are made of wood fiber and are attached to the heated wall with angles changing from 30° to 150° in different experiments. The length of each partition is equal to the width of the enclosure, therefore dividing the enclosure to isolated cells only at 90°. At other angles the cells are interconnected near the cold wall. Rayleigh number based on the enclosure width is changed from 3500 to 32000. Results for the local and the average Nusselt numbers at the heated wall of the enclosure are presented and discussed for various partition angles and Rayleigh numbers. It is found that, at each Rayleigh number, there exists an optimum inclination angle which minimizes the average Nusselt number.

Key words: *free convection, enclosure, interferometry, Mach-Zehnder*

Introduction

Natural convection in enclosures is an important topic in heat transfer and has been widely used in engineering applications such as double-pane windows, solar heating, cooling of electronic packages, heat transfer in rooms and buildings, and nuclear reactor cooling. Most of the previous works have focused on the position of the partitions, but in this paper the effect of the partition angle on the heat transfer performance of the cavity has been considered both experimentally and numerically. Van de Pol and Tierney [1] presented a numerical study on the effect of inserting a folded sheet inside the cavities. It was found that the heat flux through the cavity decreases as the number of partitions increases. Arnold *et al*. [2] presented an investigation into natural convection in trapezoidal cavities considering laminar conditions and a two dimensional system. Prediction revealed that as the height of the baffle rises, the heat transfer drops drastically. The laminar natural convection in an air-filled square cavity with a partition on the heated vertical wall was experimentally investigated by Nasteel and Greif [3]. Experiments were performed with an aluminum partition. Bajorek and Lloyd [4] numerically investigated natural convection heat transfer from a stack of horizontal open cavities formed due to fins attached to equipment for a wide range of Rayleigh number, cavity spacing and number of cavities in the stack. The dimensionless optimum fin spacing which yields the maximum heat trans-

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fer was found to be 2.5. In another experimental study, Bilski *et al*. [5] investigated natural convection of air in square enclosures with partially active side walls using holographic interferometry in order to obtain the average Nusselt numbers at different Rayleigh numbers. Tong and Gerner [6] numerically studied the effect of partition position on the overall heat transfer rate. It was found that placing the partition in the middle of the enclosure results in the greatest reduction in heat transfer and can be compared with an enclosure fully filled with porous medium. Shiraishi *et al*. [7] studied natural convection of air in a rectangular cavity divided by multiple partitions, both experimentally and numerically. It was shown that the Nusselt number is inversely proportional to $N+1$, where N is the number of partitions. Acharya and Jetli [8] studied the square cavity with the conducting fin attached to the top or bottom wall. Three divider positions and height have been considered. It was found that the divider position has a negligible effect on heat transfer inside the cavity. Ampofo [9] experimentally studied natural convection of air in a partitioned and non-partitioned cavity by laser Doppler anemometer and a micro diameter thermocouple. In his study the temperature of the hot and cold walls were fixed and so the Rayleigh number and the partitions had a thermal conductivity higher than the walls of the cavity. The results showed a reduction of local and average Nusselt number along the hot wall in partitioned cavity. In a recent study by Ben-Nakhi and Chamkha [10], the effect of the inclination angle of a heated thin fin on natural convection in a square enclosure was numerically investigated. It was found that the average Nusselt number decreases with an increase in the fin length.

A large number of studies focused on locating the partitions on active (hot and cold) walls of the cavity. Frederick and Valencia [11] performed numerical simulations for a cavity with a conducting partition attached to its hot wall. The partition length and the partition to fluid thermal conductivity ratio were varied. They found that the short partition enhances the circulation compared with the non-partitioned cavity at Rayleigh numbers ranging from 10^5 to 10^6. At high Rayleigh numbers the partition causes asymmetry in the flow field and therefore different Nusselt numbers along the hot wall in comparison with cold wall are observed. Frederick [12] studied an inclined square cavity with the diathermal partition on its cold wall at Rayleigh numbers of 10^3-10^5, numerically. Heat transfer reduction up to 47% relative to undivided cavity was reported. In another study with multiple fins in tall systems, Facas [13] has focused on the effect of partitions length on natural convection. It was found that the higher length of partitions enhances heat transfer rate across the two sides of the cavity. Nowak and Novak [14] studied natural convection heat transfer in slender cavities with two vertical partitions attached to horizontal walls numerically. It was observed that Nusselt number reduces up to 7% to 26%. Lakhal *et al*. [15] numerically studied the laminar natural convection in vertical tall enclosures with perfectly conducting multi-partitions attached to the heated wall. A numerical study has been carried out in square cavities with adiabatic horizontal walls and isothermal vertical walls by Bilgen [16]. Mezrhab *et al*. [17] numerically studied the effect of partitions in an inclined cavity attached to its cold wall. It was observed that for Rayleigh numbers lower than 10^5 the average Nusselt number of the inclined cavity is higher than in the vertical one, while for the higher Rayleigh numbers, it has an inverse effect. Jami *et al*. [18] presented a numerical investigation of natural convection heat transfer in an enclosure with inclined partitions attached to its hot wall. Effects of partitions length and angle on heat transfer were studied in two different cavity inclination angles of 45° and 90°.

In a recent study by Varol *et al*. [19], natural convection heat transfer in an inclined fin attached square enclosure is studied both experimentally and numerically. Inclination angle of

$30° \leq \theta \leq 120°$ of a single fin was considered in their study. A correlation was developed including all effective parameters on heat transfer and fluid flow. Triangular enclosures have been the subject of many recent studies as [20-22]. Oztop *et al*. [20] investigated the buoyancy-induced heat transfer and fluid flow in a triangular enclosure both numerically and experimentally. It was observed that inclination angle can be used as a control parameter for heat transfer. Heat transfer in a triangular enclosure filled with a fluid-saturated porous medium with a conducting thin fin on the hot vertical wall has been numerically analysed by Varol *et al*. [21]. It was also found that heat transfer was reduced when the thin fin was attached onto the wall. Varol [22] also performed a numerical study to examine the heat transfer in a porous triangular enclosure with a centered conducting body. It was observed that both height and width of the body and thermal conductivity ratio play an important role on heat and fluid flow inside the cavity. Chebyshev spectral collocation method [23] and lattice Boltzmann equation method [24, 25] have also been used to numerically investigate the heat transfer in square cavities. The methods were validated with the benchmark results.

According to what has been done before, there has not been any experimental investigation for the array of fins inside the cavity. The objective of this study is to investigate the effect of the inclination of the partitions on natural convection heat transfer in a rectangular enclosure filled with air. A schematic of the problem is shown in fig. 1. The enclosure dimensions are 9 cm high × 20 cm deep which is formed in two different widths. Two adiabatic partitions of the thickness of 2 mm are installed on the hot wall. The hot wall is maintained at constant temperature with an electrical heater attached to its back. The enclosure height, wood fiber layer thickness, magnesium oxide layer thickness and the thickness of the hot and cold walls are shown in fig. 1.

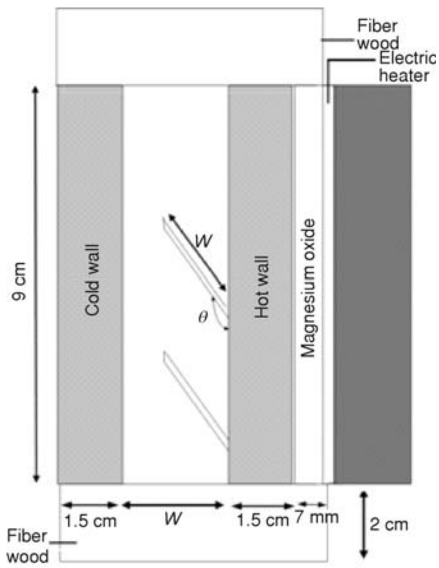

Figure 1. Model geometry

Experimental set-up

Interferometer

Optical techniques are known as non-intrusive methods for studying natural convection heat transfer and obtaining full measurements of the temperature. The Mach-Zehnder interferometer set-up (MZI) used in this experimental study has many applications in engineering especially in investigating of natural convection heat transfer problems. Figure 2 shows

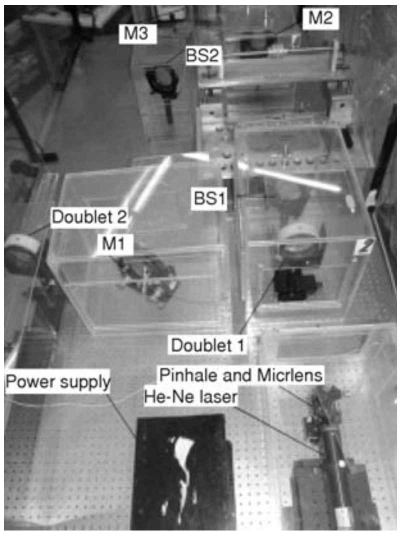

Figure 2. Schematic of the Mach-Zehnder interferometer

the interferometer set-up. The MZI set-up consists of a light source which is here a 10 mW Helium-Neon laser with λ= 632.8 nm. The laser beam passes through a first beam splitter (BS1) which divides the beam into two references and measurement beams. One beam passes through the test section and the other through the undisturbed field. These beams then gather again by mirrors M1 and M2 in a second beam splitter (BS2) and focused on a 3.2 M pixel "Artcam 320 P" CCD camera. The CCD is connected to a computer and a video recorder. In order to prevent the effect of the dust on mirrors and beam splitters, plexiglass sheets are installed on optical components besides; the optical table is enclosed with plexiglass sheets of 1.2 m high to prevent air movement effect of the laboratory on the experiment. Further information about MZI can be found in [26, 27].

Experimental test section

To reduce heat losses from the hot wall, sides of this wall are insulated with wood fiber with a thermal conductivity of k = 0.05 W/mK and thickness of 2 cm. In order to have a two dimensional flow field, the depth of the enclosure is taken 20 cm along the laser beam and also to reduce the radiation, the inner surface of the hot wall is fully polished. A 50 Ω electric heater is placed at the back of the hot wall which is primary filled with magnesium oxide powder with a thickness of 7 mm. Magnesium oxide powder is a material with high conductivity which is used to attain the constant temperature boundary condition at the hot wall. Power supply which is used in this study can produce 30 V-2 A electrical current at its maximum power. Cold wall is in direct contact with the ambient air from outside. The temperature uniformity of the cold and hot walls is checked with three k-type thermocouples placed in different positions of the hot and cold walls near the active surfaces. The drilled holes diameter of the thermocouples is 1.5 mm, with the lengths of 2, 5, and 8 cm.

Maximum variation of 0.2 ºC is observed in steady-state condition. Two other thermocouples of the same type are applied to measure the reference and ambient temperature. The reference thermocouple is placed in the enclosure far enough from the hot surface and its fringe shift (ε) is assumed to be zero. Another thermocouple is placed in the optical table to measure the ambient temperature.

All the temperatures are monitored continuously in a PC by a selector switch and a "TESTO 177 T4" four channel data logger. The pressure and relative humidity of the laboratory is measured during the tests. In order to adjust the enclosure in front of the laser beam, another plate is installed in the back of the hot plate with three holes drilled in it. These holes are connected to three holders via brass bolts. The inclination of the enclosure can be adjusted with these three bolts, providing three degrees of freedom for the enclosure as illustrated in fig. 3(a). In order to let the laser passing through the section and also provide a barrier for the air, two optical windows are installed at the two ends of the test section. As it is shown in fig. 3(b), partitions are primarily fixed at each angle

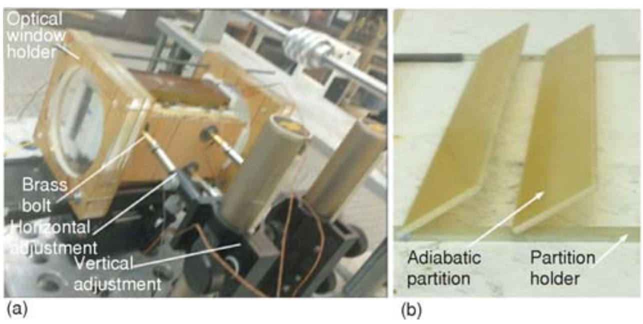

Figure 3. (a) Experimental set-up for mounting the test section, (b) structure of partition walls

on the thin holders made of wood fiber. These holders are installed at the two ends of the enclosure.

Data reduction method

The goal of the data reduction procedure is to determine the local and average Nusselt numbers of the mid cell hot wall at different partition inclination angles and Rayleigh numbers. A channel width of 1.5 cm is employed to provide the Rayleigh numbers ranging from 3500 to 6500, while Rayleigh numbers in the range of 19000 to 31000 can be achieved by using a channel width of 2.5 cm. Interferograms obtained for each position and Rayleigh number are given to the code. The code can calculate the isotherms distance from the hot wall as well as their temperatures based on the difference between hot and cold wall temperature. The mid cell hot wall is divided into 14 equal parts and the Nusselt number is calculated at each point by the method explained in [26, 27]. For a better understanding of data reduction procedure, one of the interferograms is shown in fig. 4.

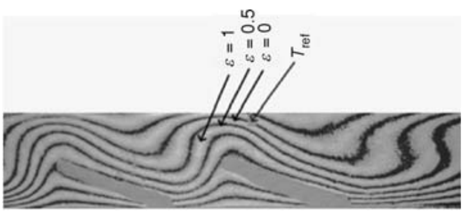

Figure 4. Detailed information used for data reduction procedure

A thermocouple is placed far enough from the hot wall in the reference fringe to measure the reference temperature. This fringe is numbered zero and by getting closer to the hot wall, the fringe number increases. Knowing the temperature of the reference fringe, temperature of other fringes can be calculated from the following relation:

$$T = \frac{3P_\infty N_\mathrm{r} l T_\mathrm{ref}}{3P_\infty N_\mathrm{r} l - 2\lambda \mathrm{R} \varepsilon T_\mathrm{ref}} \tag{1}$$

where R = 287 J/kgK is the gas constant, P_∞ – the ambient pressure, λ – laser wave length, l – the depth of the enclosure, T_ref – the reference temperature, ε – the fringe shift, and $N_r = 1.503 \cdot 10^{-4}$ m^3/kg is the specific refractivity of air. Calculating the temperature gradient depends on the gradient of the fringe shift (ε) with respect to the horizontal distance from the hot wall. Differentiating eq. 1 leads to the temperature gradient as:

$$\left.\frac{\mathrm{d}T}{\mathrm{d}x}\right|_{x=0} = \left.\frac{\mathrm{d}T}{\mathrm{d}\varepsilon}\frac{\mathrm{d}\varepsilon}{\mathrm{d}x}\right|_{x=0} \tag{2}$$

Then $\mathrm{d}T/\mathrm{d}\varepsilon|_{x=0}$ can be calculated from the following equation [26, 27]:

$$\left.\frac{\mathrm{d}T}{\mathrm{d}\varepsilon}\right|_{x=0} = \left.\frac{6N_\mathrm{r} l P_\infty \lambda \mathrm{R} T_\mathrm{ref}^2}{(3N_\mathrm{r} l P_\infty - 2\lambda \mathrm{R} T_\mathrm{ref}\varepsilon)^2}\right|_{x=0} \tag{3}$$

The local heat transfer coefficient can be calculated as:

$$h_\mathrm{y} = -k_\mathrm{s}\left.\frac{\mathrm{d}T}{\mathrm{d}x}\right|_{x=0}\frac{1}{T_\mathrm{H} - T_\mathrm{C}} \tag{4}$$

where k_s is the thermal conductivity of the air at the surface temperature of the hot wall and h_y – the local heat transfer coefficient in different vertical distances measured from the lowest point of middle cell. Therefore, the Nusselt number can be defined as:

$$\mathrm{Nu}_\mathrm{y} = \frac{h_\mathrm{y} W}{k_\mathrm{f}} = \frac{k_\mathrm{s} W}{k_\mathrm{s}(T_\mathrm{H} - T_\mathrm{C})}\left.\frac{\mathrm{d}T}{\mathrm{d}x}\right|_{x=0} \tag{5}$$

where, k_f is the thermal conductivity of the air calculated in the film temperature defined as:

$$T_f = \frac{T_H + T_C}{2} \tag{6}$$

The average Nusselt number can be calculated from the following equation:

$$\mathrm{Nu}_{\mathrm{ave}} = \frac{1}{L}\int_0^L \mathrm{Nu}_{\mathrm{y}}\,\mathrm{d}y \tag{7}$$

Results and discussion

The effect of adiabatic partition angle in natural convection heat transfer in a rectangular enclosure filled with air has been investigated experimentally. Experiments are performed for θ = 30°, 60°, 90°, 120°, and 150° and the Rayleigh number ranging from 3500 to 6500 and 18000 to 32000. This study can be divided into two separate parts. The first part is dedicated to low Rayleigh numbers (W = 1.5 cm) and the other is performed at high Rayleigh numbers (W = = 2.5 cm).

In order to check the accuracy of the experiments, the results of the partitions angle of 90 which are similar to the three isolated cavities case, are compared with the previous works. In this case, the mid cell can be considered as a single enclosure formed with adiabatic up and bottom walls. The early works by Jakob [28], Mull and Reiher [29], and Eckert and Carlson [30] expressed the following correlation with a simple power law relation, based on experimental results:

$$\mathrm{Nu} = d(\mathrm{Gr})^a\left(\frac{L}{W}\right)^b$$
$$\mathrm{Gr} = \frac{g\beta(T_H - T_C)W^3}{\nu^2} \tag{8}$$

where g is the acceleration due to Earth's gravity, β – the volumetric thermal expansion coefficient, and ν is the kinematic viscosity of air calculated at the mean temperature. Figure 5(a) represents the comparison of low Rayleigh numbers (W = 1.5 cm) for θ = 90° and the results for high Rayleigh numbers (W = 2.5 cm) is plotted in fig. 5(b). These comparisons show excellent agreement. A maximum difference of 6% is observed at θ = 90° with the work of Eckert and Carlson [30] as shown in fig. 5.

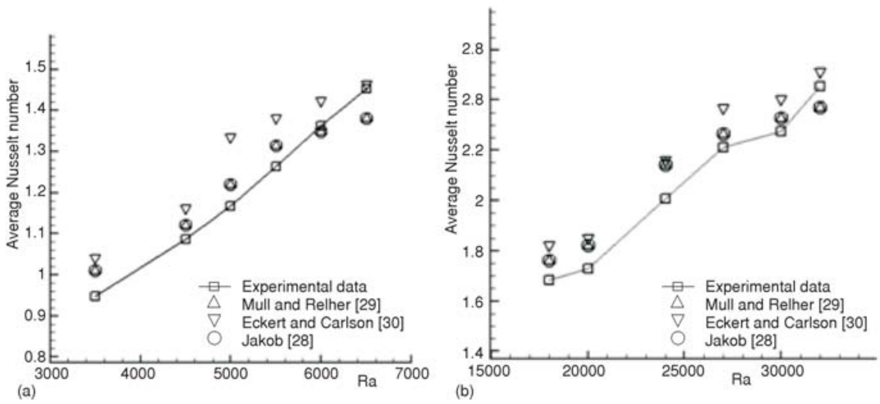

Figure 5. Comparison between the experimental data and empirical previous works for θ = 90° and (a) W = 1.5 cm and (b) W = 2.5 cm

For each of the above works, the range of the aspect ratio (L/W), Grashof Number (Gr), and values of the correlation constants are given in tab. 1. It is observed that the average Nusselt number increases with the increase of Rayleigh number in each angle. Moreover there is an optimum angle at which maximizes the Nusselt number at each investigated Rayleigh number. The experimental results indicate that partitions suppress natural convection heat transfer as their angles decreases from 90°. This suppression occurs due to the blockage of the air movement which is circulating counter clockwise inside the enclosure. As the angle of the partitions increases from 90º, the average Nusselt number begins to decrease. In this case, the partitions are not also block the air movement, but guide it towards each other. It can be deduced that when partitions angle increases from 90º, the interface between the air and the length of the enclosure increases which produces a shear layer against the air movement leading to a decrease in convective heat transfer.

Table1. Empirical constants and range of L/W and Gr for eq. (8)

Investigator	*d*	*a*	*b*	Range of L/W	Range of Gr
Mull and Reiher [29]	0.18	1/4	–1/9	3.12 to 42.2	$2\cdot10^4$ to $2\cdot10^5$
Eckert and Carlson [30]	0.119	0.3	–0.1	10	$8\cdot10^4$ to $2\cdot10^5$
Jakob [28]	0.18	0.25	–0.111	3.12 to 42.2	$2\cdot10^4$ to $2\cdot10^5$

Figure 6 indicates the infinite fringe interferograms for the Rayleigh numbers of 20,000, 27,000, and 32,000 with θ= 30º and θ= 120º. Partitions cause the breakdown of the density stratification in the fluid layers near the hot wall, which has a direct effect on temperature distribution across the enclosure by forming lines circulating from right to left in the enclosure.

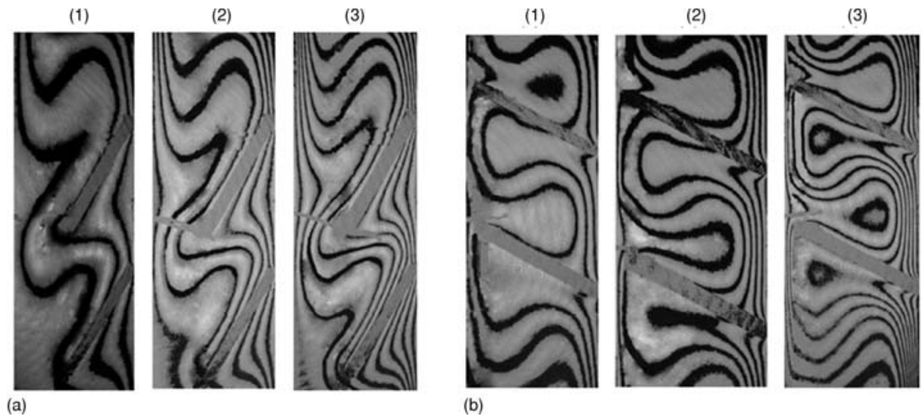

Figure 6. Interferograms captured by CCD camera for (a) θ = 30°, (b) θ = 120° at (1) Ra = 20000, (2) Ra = 27000, and (3) Ra = 32000

Figure 7(a) represents the variation of the average Nusselt number of the hot wall with respect to θ for different low Rayleigh numbers. As it can be observed, the maximum average Nusselt number occurs at the partitions inclination angle of 90º. The 90º angle resembles the con-

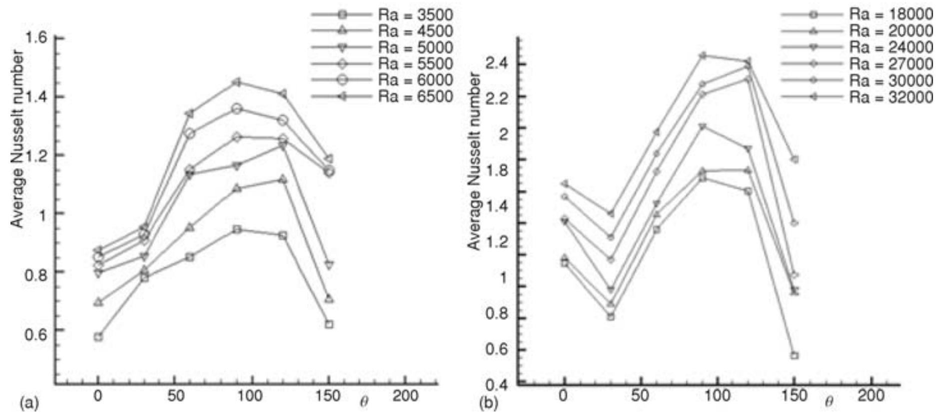

Figure 7. Average Nusselt Number *vs*. inclination angle at low Rayleigh numbers for (a) *W* = 1.5 cm and (b) *W* = 2.5 cm

dition having three isolated rectangular cavities. It can be deduced that inserting partitions causes the average Nusselt number to decrease compared with the 90º degree inclination angle. Average Nusselt number *vs*. inclination angle has been illustrated in fig. 7(b) for different Rayleigh numbers. It is found that at $\theta = 30º$ and $\theta = 150º$ the average Nusselt number is lower than non-partitioned ($\theta = 0$) enclosure and other inclination angles have positive effect on natural convection heat transfer. Maximum average Nusselt number occurs at $\theta = 90º$. It can be deduced that the variation of the average Nusselt number with inclination angle strongly depends on the Rayleigh number. At low Rayleigh numbers ($W = 1.5$ cm), applying partitions causes the average Nusselt number to be increased in comparison with the non-partitioned enclosure. Maximum average Nusselt numbers which occur at 90º indicates an increase of about 60% in comparison with non-partitioned enclosure. At high Rayleigh numbers ($W = 2.5$ cm), and $\theta = 30º$, and $\theta = 150º$ inclination angles, the average Nusselt number is lower than non-partitioned ($\theta = 0º$) enclosure, while other inclination angles have positive effect on natural convection heat transfer. However, in both cases the maximum average Nusselt number occurs at $\theta = 90º$. At low Rayleigh numbers, the partitions prevent from developing of the boundary layer on the hot wall and cause the fluid to mix horizontally. At a sufficiently high Rayleigh numbers, the boundary layer formed at the lower half of the hot wall continues to develop along the partition.

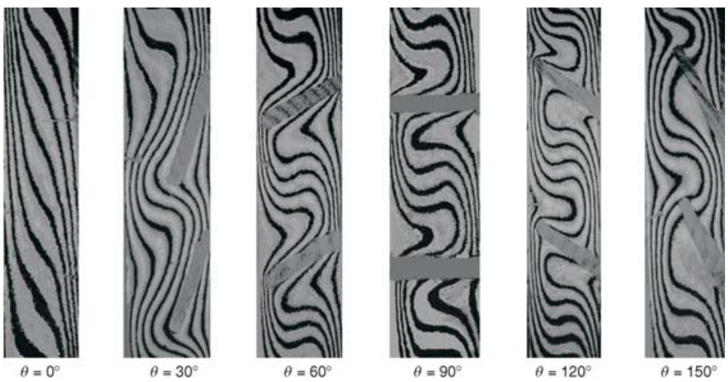

Figure 8. Interferograms of different partition angles at Ra = 5500 (*W* =1.5 cm)

To investigate the effect of partition angle on heat transfer, the interferograms for the Rayleigh numbers of 5500 and 30000 are shown in figs. 8 and 9, respectively. In general, as the fin inclination angle θ increases, the flow movement below the fin is enhanced while the flow above it slows down, causing the average Nusselt number in the mid cell to be changed.

The local Nusselt number variation is shown in fig. 10 according to the angle of inclination. It can be observed that the local Nusselt number decreases along the mid cell hot wall for the angles of 0º, 30º, 60º, and 90º.

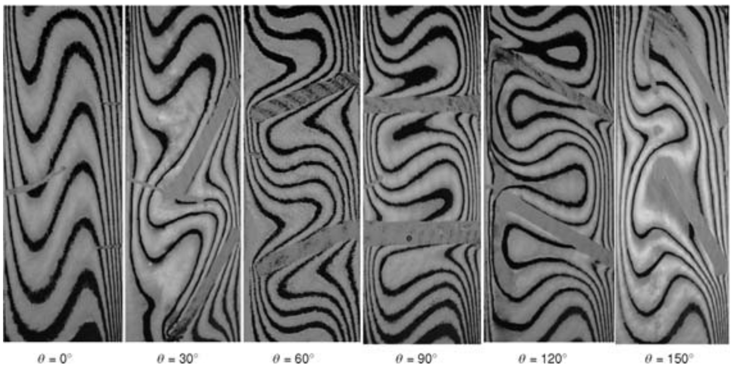

Figure 9. Interferograms of different angles at Ra = 30000 (W = 2.5 cm)

Angles of 120º and 150º cause Nusselt number variation to have an optimum along the hot wall. Due to the vortex creation in these angles, the local Nusselt number, as it is clear in interferograms of figs. 8 and 9, increases along the got wall to reach its maximum value and then begins to decrease. As θ increases, isotherms non-linearity in the core region happens and these changes in the temperature distributions causes the wall heat transfer to be changed.

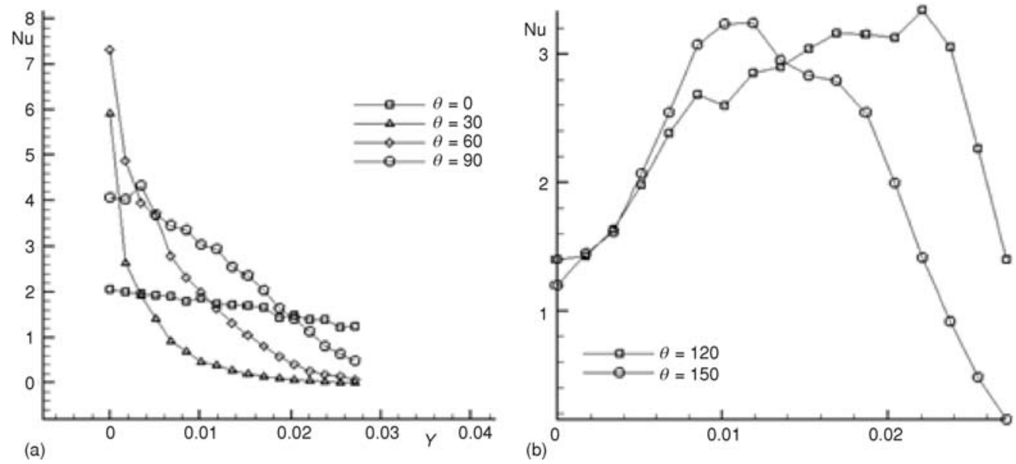

Figure 10. Local Nusselt number for acute angles at Ra = 32000 and W = 2.5 cm for (a) acute angles and (b) obtuse angles

Numerical simulation and comparison with experiments

The 2-D numerical simulations were performed using the FLUENT commercial package. The governing equations for the steady-state, two-dimensional laminar flow with consideration of the temperature-dependent property variations and buoyancy can be found in [19]. The incompressible ideal gas Navier-Stokes equations were solved by the finite volume method with structured meshes and the coupling between the velocity and pressure given by the SIMPLE algorithm. Grid independence studies resulted in meshes having about 15000 elements for each

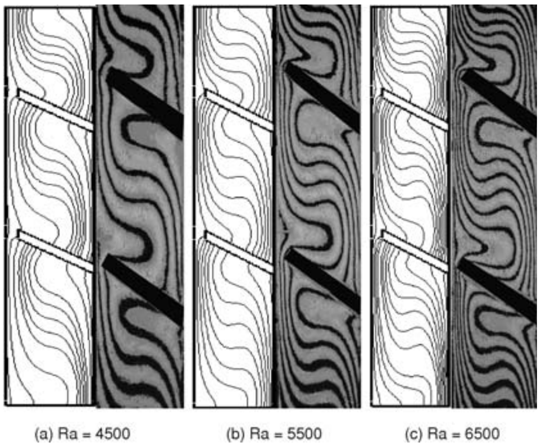

Figure 11. Comparison between the experimental data and numerical simulation for θ = 120° at (a) Ra = 4500 and (b) Ra = 5500, (c) Ra = 6500

case. Figure 11 compares the temperature contours obtained from interferometry method with the numerical results for θ = 60º for different Rayleigh numbers. The numerical results are in good agreement with the experimental data.

The distributions of the average Nusselt on the hot wall for Ra=3500 are shown in fig. 12(a). It is obvious that there is an angle that maximizes the average Nusselt number from the hot wall. By increasing the partition angle from 0º to 90º, the blockage of the air movement decreases and the average Nusselt number increases monotonically. Increasing the contact angle above 90º results in creation of a shear layer acting against the air movement and leads to a decrease in the average Nusselt number. The comparison between the experimental data and numerical simulation for a higher Rayleigh number (Ra = 18000) is shown in fig. 12(b). By increasing the Rayleigh number, the results are more consistent with the experimental values. Having higher temperature difference between the hot and cold walls, more fringes are created in the enclosure which will improve the data reduction procedure.

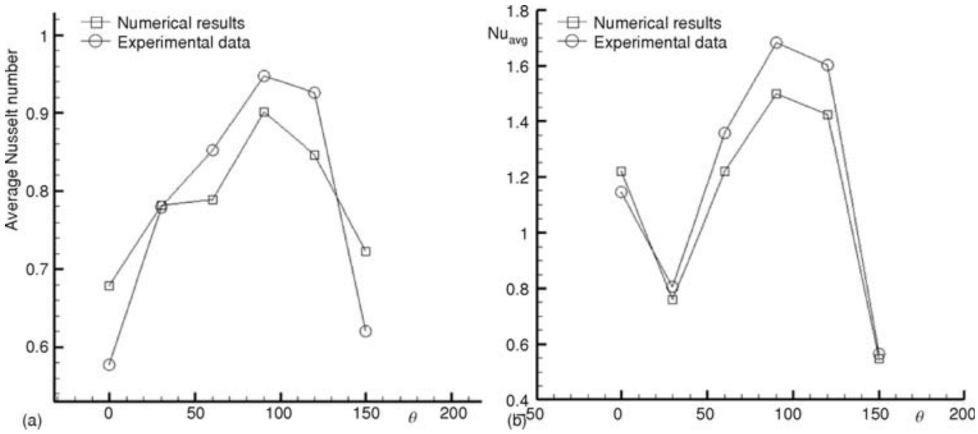

Figure 12. Comparison between the numerical simulation and experimental data for (a) Ra = 3500, W = 1.5 cm and (b) Ra = 18000, W = 2.5 cm

Conclusions

Laminar free convection from the hot wall in a partitioned rectangular enclosure with two isothermal walls is investigated both experimentally and numerically in air. Experiments are carried out using Mach-Zehnder interferometer for different spacings between hot and cold wall, different partitions angle and Rayleigh numbers. The experimental data show that the average Nusselt number increases with increasing Rayleigh number for all the inclination angles and spacing. In addition, at each Rayleigh number there exists an optimum angle that maximizes the average Nusselt number from the hot wall. It is also observed that for acute angles, the local Nusselt number decreases along the hot wall at each Rayleigh number, while there is a location at which the Nusselt number reaches its maximum value and reduces afterwards for obtuse angles.

Nomenclature

a – parameter in eq. 8
b – parameter in eq. 8
d – parameter in eq. 8
Gr – Grashof number, $(= g\beta(T_H - T_C)W^3/\nu^2$, [–]
g – gravity acceleration, [$\mathrm{ms^{-2}}$]
h – heat transfer coefficient, [$\mathrm{Wm^{-2}K^{-1}}$]
k – thermal conductivity of air, [$\mathrm{Wm^{-1}K^{-1}}$]
L – enclosure height, [cm]
l – enclosure depth, [cm]
N_r – specific refractivity, [$\mathrm{m^3\,kg^{-1}}$]
Nu – Nusselt number, $(= h_y\, W/k_f)$, [–]
P – pressure, [Pa]
R – gas constant, [$\mathrm{Jkg^{-1}K^{-1}}$]
Ra – Rayleigh number based on the enclosure width, $(= g\beta(T_H - T_C)\,\mathrm{W}^3/\nu a^2$, [–]
T – temperature, [K]
W – enclosure width, [cm]
X – horizontal co-ordinate
Y – vertical co-ordinate

Greek symbols

β – volumetric thermal expansion coefficient, [1/K]
ε – fringe shift
θ – partitions inclination angle with respect to the hot wall, [°]
λ – laser wavelength, [nm]
ν – kinematic viscosity of air, [$\mathrm{m^2s^{-1}}$]

Subscripts

avg – average
C – cold
f – film condition
H – hot
ref – reference condition
s – surface condition
∞ – ambient condition

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Paper submitted: May 26, 2012
Paper revised: February 11, 2013
Paper accepted: February 11, 2013