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AN EXPERIMENTAL AND NUMERICAL STUDY OF LAMINAR NATURAL CONVECTION IN A DIFFERENTIALLY-HEATED CUBICAL ENCLOSURE

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Abstract: Natural convection of air in a cubical cavity was investigated experimentally and numerically. A cubical enclosure of 200x200x200 mm^3 dimensions was considered. One vertical wall of the cavity was hot, the opposite one was cold and the rest of the walls were adiabatic. Three walls were made of aluminum, whereas the remaining three walls were made of heat-resistant glass. Temperature of the hot wall was kept constant by means of an electrical heater and the cold wall was cooled by ambient air in the experiments. Particle Image Velocimetry (PIV) was used for velocity measurements and Fluent CFD software package was employed for the numerical solution of three-dimensional laminar flow equations at a Rayleigh number of $1.3x10^7$. The numerical and experimental results were compared and found to be in general agreement with each other. Suitability of a characteristic velocity for normalizing velocity data obtained was demonstrated.
Keywords: Natural convection, Enclosure, PIV, CFD, Laminar flow.

FARKLI SICAKLIKTA KÜBİK KAPALI HACİM İÇİNDE LAMİNER DOĞAL TAŞINIMIN DENEYSEL VE SAYISAL İNCELENMESİ

Özet: Kubik kapalı hacim içerisinde havanın doğal taşınım hareketleri deneysel ve sayısal olarak incelenmiştir. 200x200x200mm^3 boyutlarında kübik bir hacim ele alınmıştır. Hacmin dikey duvarlarından biri sıcak, karşı taraftaki soğuk ve diğer duvarları adyabatik olarak kabul edilmiştir. Üç duvar alüminyumdan geri kalan üç duvar ısıya dayanıklı camdan imal edilmiştir. Deneylerde sıcak duvarın sıcaklığı elektrikli rezistans ile sabit tutulmuştur. Soğuk duvarın sıcaklığı çevre havası ile soğutularak sabit tutulmuştur. Hız ölçümleri için Parçacık Görüntülemeli Hız ölçümü (PIV) kullanılmış, laminer akımda Rayleigh sayısı $1.3x10^7$ için üç boyutlu nümerik çözümlemede Fluent CFD paket programı kullanılmıştır. Sayısal ve deneysel sonuçlar karşılaştırılmış ve sonuçların uyum içinde olduğu görülmüştür. Normalize edilmiş hız karakterleri gösterilmiştir.
Anahtar Kelimler: Doğal taşınım, Kapalı hacim, PIV, CFD, Laminer akış.

NOMENCLATURE

h_{eff}	heat transfer coefficient
g	gravitational acceleration, [m/s^2]
Gr	Grashof number, dimensionless
H	dimension of cubical cavity, [m]
k	thermal conductivity, [W/m K]
Nu	Nusselt number, dimensionless
p	pressure [Pa]
p´	dimensionless pressure
Pr	Prandtl number, dimensionless
Ra	Rayleigh number (Eq.1), dimensionless
t	time [s]
t´	dimensionless time
T	temperature, [˚C]
u	horizontal velocity, [m/s]
u´	dimensionless horizontal velocity
v	vertical velocity, [m/s]
v´	dimensionless vertical velocity
v´´	vertical velocity normalized by V_R
V_R	reference velocity, [m/s]
V_{MAX}	maximum vertical velocity, [m/s]
x,y,z	Cartesian coordinates, [m]
X,Y,Z	Cartesian coordinates normalized by H

Greek symbols

α	thermal diffusivity, [m^2/s]
β	thermal expansion coefficient, [1/K]
Θ	dimensionless temperature
υ	kinematic viscosity, [m^2/s]

Subscripts

r	reference
c	cold wall
h	hot wall

INTRODUCTION

Natural convection is encountered frequently in many engineering applications such as ventilation and air-conditioning systems, electronic equipment cooling, solar collectors, ovens and refrigerators (Calcagni et al., 2005). A prismatic or cubical cavity is used as a popular model for experimental and numerical natural convection studies.

Meyer et al. (2002) investigated turbulent natural convection flow in a cubical cavity heated from the bottom by means of PIV at a Rayleigh number of $Ra=1.4 \times 10^{10}$. Ahmed and Yovanovich (1992), obtained numerical and experimental data for $0 \leq Ra \leq 10^6$ and Prandtl number of Pr= 0.72 in a cube shaped box with discreet heat sources. Gelfgat et al. (1999) investigated various stable phases of natural convection in heated confined volumes and prepared stability diagrams for aspect ratios (Length/Height) between 1 and 11 and Pr= 0-0.015. Schmidt (1996) and Kwak et al. (1996) obtained two-dimensional numerical solutions for a differentially-heated square prism for $Ra=10^7$. Dol and Hanjalic (2000) obtained numerical solutions using adiabatic and isothermal boundary conditions for turbulent natural convection in a confined volume at $Ra= 4.9 \times 10^{10}$. Leal et al. (2000) developed codes using partial Boussinesq approximation for $Ra=10^3$-10^5 and Pr=0.71. Ha and Jung (2000) solved three-dimensional, steady, conjugate heat transfer of natural convection and conduction in a cubical volume for $Ra=10^3$,10^4 and 10^5. Frederick and Quiroz (2000) analyzed convection flows in a cubical cavity and focused on the transition from conduction to convection regime as the Rayleigh number is increased from 10^5 to 10^7. Corvaro and Paroncini (2007) compared stream functions, isothermal lines, velocity maps for the natural convection heat transfer in a square cavity filled with air in an numerical and experimental study for $Ra=9.08 \times 10^4$-2.51×10^5. Mamun et al. (2004) studied natural convection in a cubical cavity by using PIV at Rayleigh numbers of 10^6 and 6×10^6. Ramesh and Venkateshan (2000) conducted laminar natural convection heat transfer experiments in a square enclosure filled with air using a differential interferometer. Ganguli et al. (2009) performed CFD simulations to predict heat transfer coefficient for tall slender geometries (100mm<H<1000mm) with varying gap widths (5mm<L<84.7mm) and temperature differences (5K<T<90K). Amara et al. (2008) studied the flow motions depending on the boundary conditions applied on the vertical walls. Laguerre et al. (2007, 2008) conducted experiments in order to observe circular air flow in a refrigerator model using PIV and investigated velocity profile in the boundary layers of an empty and a full refrigerator model. They also studied heat transfer by natural convection in domestic refrigerator without ventilation using experimental and numerical approach.

In the present study, laminar natural convection flow of air inside a cubical cavity was investigated both experimentally and computationally at $Ra=1.3 \times 10^7$. One vertical wall of the cavity was hot, the opposite one was cold and the rests of the walls were adiabatic. Surface and air temperatures were measured by thermocouples bonded onto cavity surfaces and placed on a grid inside the confined volume. PIV (Particle Image Velocimetry) was used to measure two components of the mean velocity. Numerical results were obtained by using Fluent CFD software. Experimental and numerical results are presented and discussed in a comparative manner.

Experimental and Numerical Procedure

Experiments were conducted in the Research and Development Laboratory of household appliances company Arçelik by employing a cubical cavity of height H=200 mm. Three surfaces of this cavity were made of 5 mm thick glass and the other three surfaces were made of 12 mm thick aluminum. At least two surfaces of the cavity must be glass in order to have optical access for PIV measurements. A third surface was also made of glass in order to create symmetry. The wall material was Borofloat glass capable of withstanding temperatures as high as 500˚C. A plate type resistance (heat source) placed outside a vertical aluminum wall was used to keep the hot wall at a constant temperature of 69˚C. Five surfaces of the cavity were covered with 100 mm thick glass wool insulation material. The cold vertical wall facing the hot one was not insulated and remained at a constant temperature of 41˚C. Insulation material was also covered with aluminum foil. 100 pieces of J-Type thermocouples calibrated by a Fluke 5500A thermometer were employed for temperature measurements. 60 pieces of thermocouples were mounted on a planar wire mesh which could be placed at any desired location in the cavity. These thermocouples were connected to a HP 3852A data collecting unit connected to a PC. Temperatures were measured in 10 to 30 second intervals using HP-VEE Pro 6.01 software. Experiments were conducted in an air-conditioned room at 21˚C. All measurements were made for the steady state which was obtained after a sufficiently long operation (a few hours) of the heat source. Experimental uncertainties are estimated as 2% and 3% for temperature and velocity data, respectively.

A PIV (Particle Image Velocimetry) system was used to measure the velocity of air inside the cavity. PIV is a measurement technique used to obtain the instantaneous velocity field in a cross section of flow Raffel et al., (1998). Atomized paraffin oil was used to seed the flow field in this study. Figure1 gives a schematic of the experimental setup and also the (x,y,z) Cartesian coordinate system used in presenting the results. A laser source (Dantec Dynamics New Wave Solo III 15 Hz Laser) placed under the cavity was employed to generate a vertical laser sheet. A CCD camera (Dantec Dynamics Hi Sense MK II Camera) viewed the laser sheet normally from a distance of 150cm. Experimental data were obtained only in the span wise center plane (z/H=0.5). The PIV system was controlled by a DANTEC PIV 2100 processor and data were processed by Dantec Flow manager (V4.71.105). The field of view of the camera was approximately 0.27x0.21 m^2. Vector maps contained 83x63 vectors. The data were processed in 32x32 interrogation areas with no overlap.

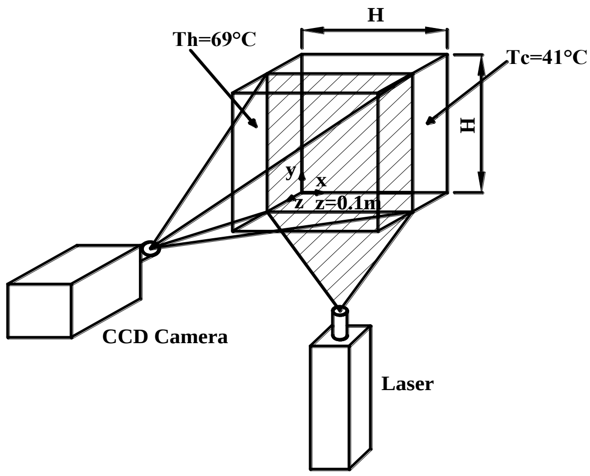

Figure 1. Schematic of test section.

The Fluent program uses the finite-volume method to solve the governing equations for a fluid. A detailed description of the program can be found in the Fluent User's Guide (2005). Three-dimensional, laminar flow equations were solved by using the segregated solver algorithm and second order upwind implicit scheme. The segregated algorithm solves the temperature field after prediction and correction of the velocity field. The PRESTO scheme and the SIMPLE algorithm are employed by Fluent for pressure interpolation and pressure-velocity coupling, respectively. Two opposite walls had constant wall temperatures specified above and the remaining four walls were specified as adiabatic. Solutions were obtained for a mesh size of 61x61x61 which corresponded to 226981 finite volumes (cells) in the cavity. Increasing the mesh size from 61x61x61 to 71x71x71 changed the calculated surface-averaged Nusselt number by only 0.27%. For a mesh size 61x61x61 solution was obtained with 2000 iterations in 12 hours and for 71x71x71 with 2500 iterations in 24 hours. This was accepted as sufficient evidence for grid independence and the solution obtained for 226981 cells was presented here. Convergence criteria were selected as 10^{-3} for the continuity and momentum equations and 10^{-6} for the energy equation. The solution corresponds to a Rayleigh number of Ra=1.3×10^7 which is defined by

$$Ra = \frac{g\cdot\beta\cdot(T_h - T_c)\cdot H^3}{\nu_r\cdot\alpha_r} \tag{1}$$

The dimensionless parameters used in data presentation are given by

$$X=\frac{x}{H};\ Y=\frac{y}{H};\ Z=\frac{z}{H};\ u'=\frac{u}{v_{MAX}};\ v'=\frac{v}{v_{MAX}};$$

$$t'=\frac{t}{\frac{H^2}{\upsilon_r}};\ p'=\frac{p}{\rho_r\cdot\frac{\upsilon_r^2}{H^2}};\ \theta=\frac{T-T_c}{T_h-T_c} \tag{2}$$

Dimensionless continuity equation:

$$\frac{\partial u'}{\partial x'}+\frac{\partial v'}{\partial y'}=0 \tag{3}$$

Dimensionless x-momentum equation:

$$\frac{\partial u'}{\partial t'}+u'\cdot\frac{\partial u'}{\partial x'}+v'\cdot\frac{\partial u'}{\partial y'}=-\frac{\partial p'}{\partial x'}+\frac{\partial^2 u'}{\partial x'^2}+\frac{\partial^2 u'}{\partial y'^2} \tag{4}$$

Dimensionless y-momentum equation:

$$\frac{\partial v'}{\partial t'}+u'\cdot\frac{\partial v'}{\partial x'}+v'\cdot\frac{\partial v'}{\partial y'}=Gr\cdot\theta+\frac{\partial^2 v'}{\partial x'^2}+\frac{\partial^2 v'}{\partial y'^2} \tag{5}$$

Dimensionless energy equation:

$$\frac{\partial \theta}{\partial t'}+u'\cdot\frac{\partial \theta}{\partial x'}+v'\cdot\frac{\partial \theta}{\partial y'}=\frac{1}{Pr}\cdot\left(\frac{\partial^2 \theta}{\partial x'^2}+\frac{\partial^2 \theta}{\partial y'^2}\right) \tag{6}$$

Two dimensionless numbers in equations are defined as follows:

$$Gr = \frac{g\cdot\beta\cdot(T_h - T_c)\cdot H^3}{\upsilon_\infty^2} \tag{7}$$

$$Pr=\frac{\upsilon_\infty}{\alpha_\infty} \tag{8}$$

Nusselt number equation used in the numerical study:

$$Nu=\frac{h_{eff}\cdot H_r}{k} \tag{9}$$

The boundary conditions are as follows:

Left wall: $u'=v'=0,\theta=1$ (10)

Right wall: $u'=v'=0,\theta=1$ (11)

Lower wall: $u'=v'=0,\ \frac{\partial\theta}{\partial y'}=0$ (12)

Upper wall: $u'=v'=0,\ \frac{\partial\theta}{\partial y'}=0$ (13)

The boundary conditions and the solution have been input into the program. In order to validate the solution procedure, numerical solutions were also obtained for the Rayleigh numbers of 10^3, 10^4 and 10^5 and the surface-averaged Nusselt numbers were compared with numerical solutions reported in the literature. This comparison showed reasonably good agreement as reported in Kürekci (2006). For example, the Nusselt numbers produced by our solution agree with those of Peng et al. (2003) within 5 percent.

RESULTS AND DISCUSSION

Results of the experiment and computation will be presented and discussed in a comparative fashion. Temperature and velocity fields will be discussed separately. Discussion is limited to the data in the span wise center plane (Z=0.5).

Temperature Field

Figs. 2 and 3 present experimentally and computationally obtained contour plots of the normalized temperature θ, respectively. It is observed that the temperature field near the hot and cold walls has high gradients, whereas the core region of the enclosure shows a linear temperature distribution (stratification). Temperature increases in the vertical direction from bottom to top and the value of the dimensionless stratification $\partial\theta/\partial Y$ is approximately equal to one for both experimental and computational data. A constant value of stratification is suggested by the computational data in the range of 0.1<X< 0.9 and 0.3<Y<0.7.

Figs. 4 and 5 present the variation of temperature with X and Y, respectively. Data at X values of 0.25, 0.5 and 0.75 collapse well in Figure 5 for both the computational and experimental data. Therefore, computational data are presented only for X=0.5 in order to avoid overcrowding of curves. Figure 5 shows that temperature increases linearly with Y in the central part of the enclosure. Figure 4 shows the linear stratification in the range of 0.1<X<0.9 and indicates a good agreement between the experimental and computational temperature variations.

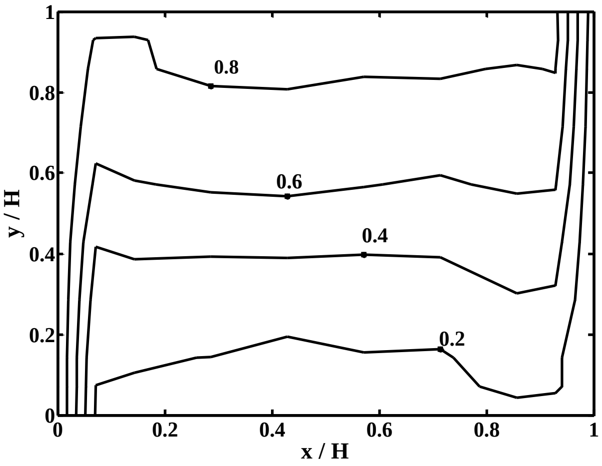

Figure 2. Experimental dimensionless temperature θ contours at z/H=0.5.

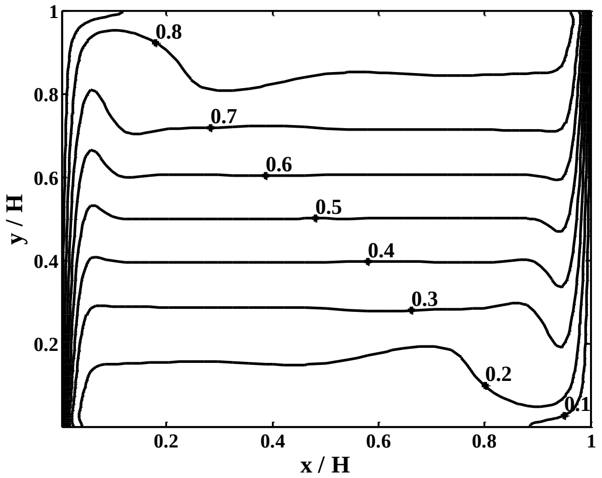

Figure 3. Numerical dimensionless temperature θ contours at z/H=0.5.

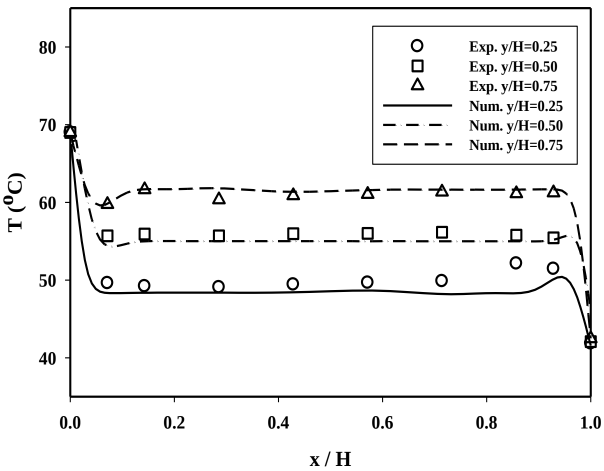

Figure 4 Variation of temperature T with x/H, (at z/H=0.5)

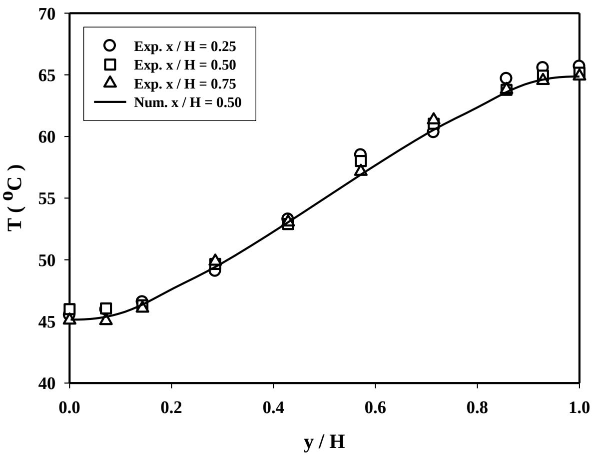

Figure 5. Variation of temperature T with y/H (at z/H=0.5).

Velocity Field

Figs. 6 and 7 present experimentally and computationally obtained contour plots of the normalized velocity u′ =u/V_{MAX} in the z/H=0.5 plane, respectively. V_{MAX} is the maximum value of vertical velocity component. The magnitude of u′ is generally small except near the upper and lower walls where the flow is towards right and left, respectively. This is due to clockwise rotation of flow in the cavity caused by the rise of heated buoyant air on the left wall and the downward collapse of the denser cold air on the right wall. Figure 7 shows that flow development along upper and lower walls is similar (identical in terms of contour shapes and values) in the computation. In contrast, the experimental contours presented in Figure 6 indicate a somewhat different flow development (for example larger velocity magnitude near the upper wall) in the experiment. This can be attributed to the approximate satisfaction of the experimental thermal boundary conditions on the uniform temperature vertical walls and the four remaining adiabatic walls.

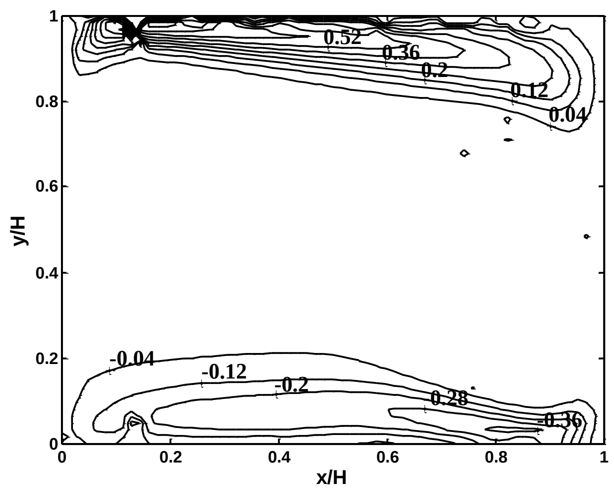

Figure 6. Experimental dimensionless horizontal velocity u′ contours at z/H=0.5.

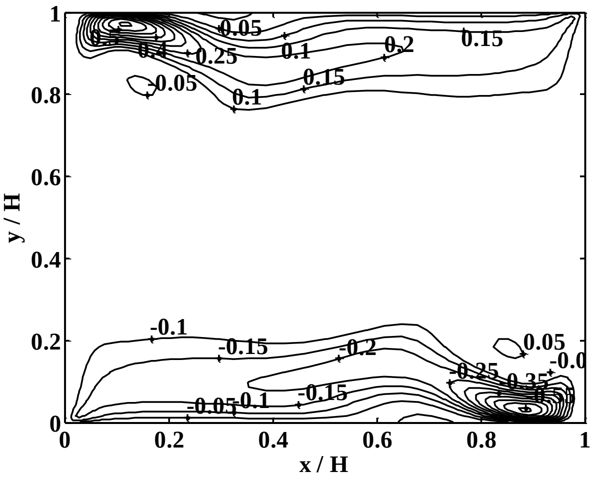

Figure 7. Numerical dimensionless horizontal velocity u′ contours at z/H=0.5.

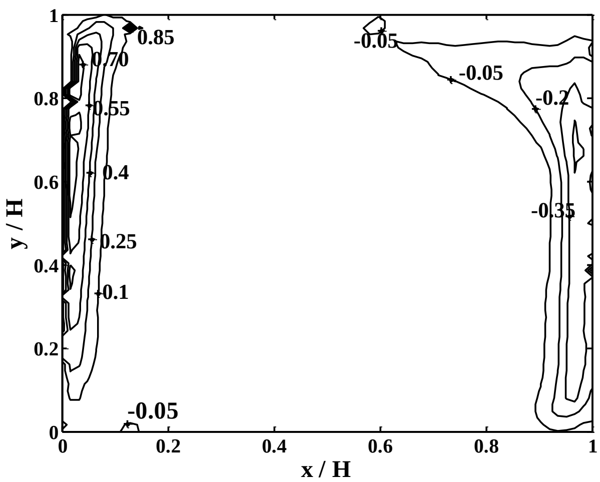

Figure 8. Experimental dimensionless vertical velocity v′ contours at z/H=0.5.

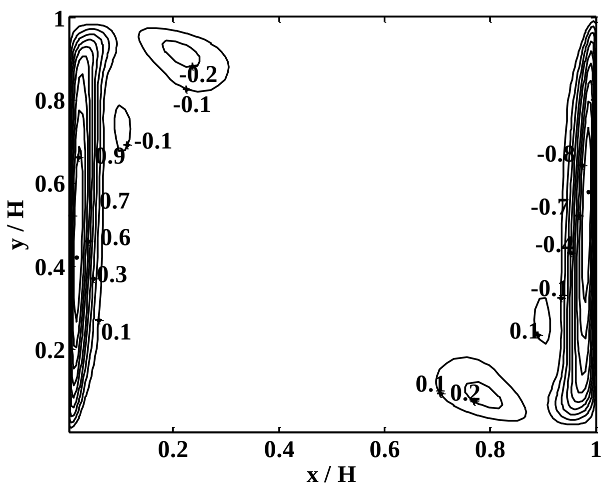

Figure 9. Numerical dimensionless vertical velocity v′ contours at z/H=0.5.

Figs. 8 and 9 present experimentally and computationally obtained contour plots of the normalized velocity v′ = v/V_{MAX} in the z/H=0.5 plane, respectively. Similar to the u′ contours, the magnitude of v′ is negligibly small except near the left and right walls where the flow direction is up and down, respectively. Comparison of Figs. 8 and 9 shows that development of vertical momentum along the left and right walls is identical in the computation but not in the experiment. This effect is more apparent near the corners and is probably due to differences in boundary conditions. Comparison of Figs. 6, 7 and 8, 9 shows that the shear layers on the vertical walls are thinner than those on the upper and lower walls. This is due to the fact that the driving force of the motion (buoyancy) acts in the vertical direction and generates larger speeds on the vertical walls.

Figure 10 presents variation of v velocity profiles with X for Y=0.25, 0.5 and 0.75 (experimental data) and for Y=0.5 (computational data). Similar to the temperature field data, the computational velocity data collapse well in the range of 0.25<Y<0.75. Therefore, computational data are presented only for Y=0.5 in order to avoid overcrowding of curves. Figure 10 shows that the experimental data agree with the computational data except near the cold right wall where computational values of velocity are higher. Approximate satisfaction of the thermal boundary conditions in the experiment is the probable cause of this discrepancy.

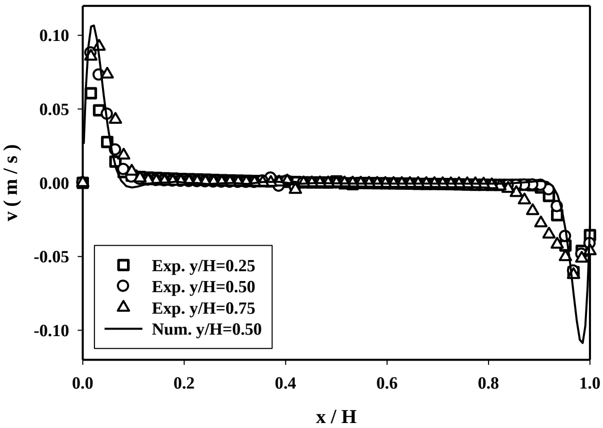

Figure 10. Dimensional vertical velocity v profiles at z/H=0.5.

Figure 11 presents variation of experimentally determined vertical velocity profile $v'' = v/V_R$ with X at Y=Z=0.5 where V_R is a characteristic velocity given by

$$V_R = \sqrt{g \cdot \beta \cdot (T_h - T_c) \cdot H} \tag{14}$$

The experimental v″ velocity profile obtained by Mamun (2003) at the same location of a cubical cavity for $Ra=6x10^6$ is also presented in Figure 11. The fact that the two profiles agree with each other reasonably well indicates suitability of V_R as a normalization parameter for this flow. The cubical cavity of Mamun (2003) has linear variations of temperature between the heated and cooled vertical walls (as opposed to the adiabatic walls of our study). This suggests that the vertical velocity profile in the mid-planes of this cubical enclosure (Y=Z=0.5) is not overly sensitive to the boundary conditions on the walls connecting the heated and cooled vertical walls.

Numerical results for hot-wall Nusselt numbers were calculated and compared with the studies in the literature.

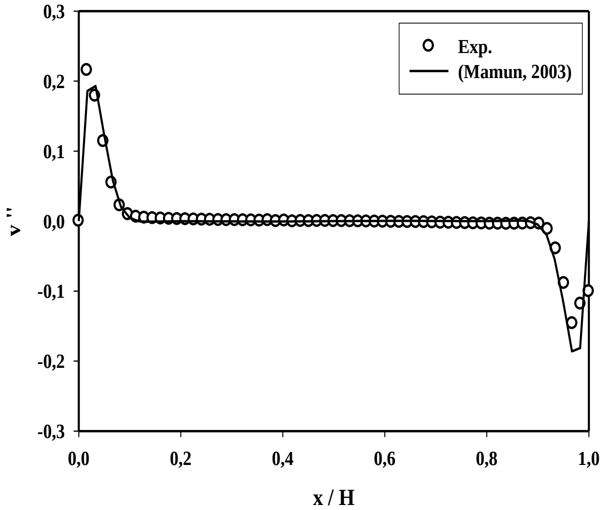

Figure 11. Dimensionless vertical velocity v″ profile at z/H=x/H=0.5.

Table1. Calculated Nusselt numbers: comparison with other works for $Ra=10^7$.

Present work	Leong and Hollands, (1999)	Bairi, (2008)	Dixit and Babu, (2006)	Bilgen, (2005)
16.21	15.549	16.073	16.79	16.629
% Deviation	4.07	0.85	-3.58	-2.58

CONCLUSIONS

Temperature and velocity fields driven by buoyancy forces in a cubic enclosure heated from a vertical side-wall were investigated numerically and experimentally at $Ra=1.3x10^7$. Numerical and experimental results for the mid-plane of the enclosure were compared and found to be in general agreement with each other. Some discrepancies have been observed between the numerical and experimental results. These discrepancies may be attributed to the differences in boundary conditions. Vertical temperature stratification had a dimensionless value of approximately one in the central part of the enclosure. Nusselt numbers for hot wall were calculated and compared with data in the literature. As a result of analysis, the results are found to be in a good agreement with the studies in the literature. By comparison with the experimental data of Mamun (2003), it was demonstrated that the characteristic velocity given by Eq. (14) is a suitable parameter for normalizing velocity data obtained at different Rayleigh numbers. This comparison also suggests that the velocity field near the heated and cooled vertical walls is not overly sensitive to the boundary conditions on the remaining four walls.

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