Journal of Chemical Engineering & Process Technology

Zemani et al., J Chem Eng Process Technol 2014, 5:3
DOI: 10.4172/2157-7048.1000191

Research Article Open Access

Effect of Partial Partitions on Natural Convection in Air Filled Cubical Enclosure with Hot Wavy Surface

Farah Zemani,

Laboratoire d'Energie et Propulsion Navale, Faculté de Génie Mécanique, Université des Sciences et de la Technologie Mohamed, Boudiaf d'Oran, B.P. 1505 Oran El-M'naouar, 3-1000 Oran, Algeria

Abstract

Natural convection in cubical enclosure with hot surface geometry and partial partitions has been analyzed. The geometry is a cube with wavy hot surface (three undulations) and three partitions. The investigation has been performed for different partitions lengths and Rayleigh number while the Prandtl number kept constant. This problem is solved by using the partial differential equations which are the equation of mass, momentum, and energy. The results obtained show that the hot wall geometry with partitions affects the flow and the heat transfer rate in the cavity. It has been found also that the mean Nusselt number decreases compared with the heat transfer in the undulated cubical cavity without partitions.

Keywords: Natural convection; Cubical enclosure; Wavy hot surface; Partial partitions

Nomenclature

g: Gravitational acceleration [m/s^2]

L: Cavity side [m]

Nu: Nusselt number

p: Static pressure [Pa]

P: dimensionless pressure

Pr: Prandtl number, ν/α

Ra: Rayleigh number, $g\beta L^3 \Delta T/a\nu$

T: Temperature [K]

T_0: Average temperature, $(T_c+T_f)/2$ [K]

T_h, T_c: Hot and cold temperature [K]

ΔT: Temperature variation, $T_h - T_c$ [K]

U,V,W: Dimensionless fluid velocities

u,v,w: Velocity components in x,y,z directions [m/s]

X,Y,Z: Dimensionless Cartesian coordinates, x/L, y/L, z/L

x,y,z: Coordinates [m]

Greek symbols

α: Thermal diffusivity of air [m^2s^{-1}]

β: Thermal expansion coefficient [k^{-1}]

θ: Dimensionless temperature

ν: Kinematic viscosity [m^2/s]

ρ: air density [Kg/m^3]

subscripts

a: average

Introduction

Natural convection flow analysis in enclosures has many thermal engineering applications, such as cooling of electronic devices, energy storage systems and fire-safe compartment. In the numerical domain, a big progress has been made on the natural convection in complex geometries [1-4].

For many years a lot of attention has been paid to the problems of natural convection in enclosure. However, the limitations of numerical tools and experimental techniques restrict investigators within the approximation of a two dimensional model even though fluid motion is three dimensional in nature. Bessonov et al. [5] have suggested a benchmark numerical solution for the three dimensional natural convection in cubical enclosure. Three dimensional laminar flow has been also studied by Mallinson et al. [6] and Lee et al. [7] for enclosures of the length aspect ratio A_z varying from 2 to the enclosure with A_z =1 and 2 have been considered by Lankhorst and Hoogendoorn [8] who computed steady flows for numbers of Rayleigh ranging from 10^6 to 10^8. Fusegi et al. [9] made a three dimensional flow analysis on natural convection in a differentially heated cubical enclosure; the detailed structures of the fields were scrutinized by using high-resolution computational results over the range of Rayleigh numbers studied, $10^3 \le Ra \le 10^6$, they clarified three dimensional structures of flow, vorticity and temperature in the cavity; they also compared their numerical results with the experimental measurements as studied by Bilsk et al. [10].

The complexity of such problems increases when the hot surface of the cubical enclosure becomes wavy. Adjlout et al. [11] have also investigated the two dimensional natural convection in an inclined cavity with hot wavy wall. They simulated the fluid and heat transfer with ADI scheme; one of their interesting results was a decrease of the averaged heat transfer compared with the mean Nusselt number of the square

***Corresponding author:** Amina Sabeur-Bendehina, Laboratoire d'Energie et Propulsion Navale, Faculté de Génie Mécanique, Université des Sciences et de la Technologie Mohamed, Boudiaf d'Oran, B.P. 1505 Oran El-M'naouar, 3-1000 Oran, Algeria, Tel: 00213556714129, E-mail: sabeuramina@hotmail.com

Received January 13, 2014; **Accepted** March 28, 2014; **Published** March 30, 2014

Citation: Zemani F, Sabeur-Bendehina A, Boussoufi M (2014) Effect of Partial Partitions on Natural Convection in Air Filled Cubical Enclosure with Hot Wavy Surface. J Chem Eng Process Technol 5: 191. doi: 10.4172/2157-7048.1000191

Citation: Zemani F, Sabeur-Bendehina A, Boussoufi M (2014) Effect of Partial Partitions on Natural Convection in Air Filled Cubical Enclosure with Hot Wavy Surface. J Chem Eng Process Technol 5: 191. doi: 10.4172/2157-7048.1000191

cavity. They showed that the thermal boundary layer is considerably affected by wavy wall and they recommended investigating the hot wall geometry optimisation. Natural convection in three dimensional rectangular enclosures had been analyzed numerically by Sik Lee et al. [12]; the effect of the Rayleigh number was mainly investigated. The results showed that the temperature disturbance imposed on the end wall reinforced the axial flow and magnified the three dimensional effect. Three dimensional structures in laminar natural convection in a cubic enclosure were investigated experimentally by Hiller et al. [13]. The Rayleigh numbers ranged from 10^4 to 2×10^7 and the Prandtl numbers from 5.8 to 6×10^3. They showed the velocity and vorticity fields and compared the experimental observations with numerical calculations found in the literature. Gunes [14] studied numerically in his thesis three dimensional natural convection cooling in enclosures and vertical channels with spatially periodic flush mounted and protruding heat sources. It was established that convection reduces the maximum operating temperature for all configurations. Yu and Joshi [15] carried out a numerical study of a three dimensional laminar natural convection in a vented enclosure. They presented the local and overall heat transfer from the heat source and the substrate, in terms of Nusselt numbers and the surface temperatures to illustrate the vent effects. Akrour et al. [16] made numerically a three dimensional steady flow on natural convection in a differentially heated cubical enclosure. The variation of Nusselt numbers on the hot and cold walls were also presented to show the overall heat transfer characteristics inside the enclosure. They found that the three dimensional data demonstrate reasonable agreement with the experimental measurement. Natural convection heat transfer associated to fluid dynamics phenomena was studied extensively by Corzo et al. [17]. The results presented revealed a good agreement not only in 2D also in 3D and for a wide range of Ra numbers. Kürekci and Özcan [18] made an experimental and numerical study of laminar natural convection in a differentially heated cubical enclosure. They compared numerical and experimental results and were found to be in general agreement with each other. Ahmadi and Dastgerdi [19] carried out a numerical analysis to study the laminar natural convection in a three dimensional enclosure. Results exposed that the average Nusselt number increases by increasing in Ra, the rate of heat transfer from hot and cold sources and maximum component of velocity increase. Lahoucine et al. [20] carried out the three dimensional numerical study of natural convection in a cubical enclosure with two heated square sections submitted to periodic temperatures, filled with air. The obtained results showed significant changes in terms of heat transfer and flow intensity. Sabeur et al. [21] performed a numerical investigation of the influence of the hot surface geometry on a laminar natural convection in a cubical cavity filled with air differentially heated. The results obtained showed that the hot wall geometry affects the flow and the heat transfer rate in the cavity. The mean Nusselt number decreases compared with the heat transfer in the cubical cavity.

Fusegi et al. [22] studied numerically a transient three dimensional natural convection in a differentially heated cubical enclosure at Rayleigh number of 10^6. They showed that the behavior of the heat transfer rate in the enclosure was considerably influenced by the presence of the internal gravity wave motion. They also [23] carried out a three dimensional numerical simulation of periodic natural convection in a differentially heated cubical enclosure; at the Rayleigh number of 8.5×10^6, they found that the period of the oscillations was consistent with the experimental measurements. In another study, the same authors [24] made numerical simulations of natural convection in a differentially heated cubical enclosure with a partition. They scrutinized the effects of the partition geometry on the three dimensional flow properties. Silva and Gosselin [25] evaluated the effect of the aspect ratio and horizontal length of a high conductivity rectangular fin attached to the hot wall of a three dimensional differentially heated cubic enclosure in laminar natural convection, a scale analysis was used to predict the domain in which the fin geometry played a significant role. Frederick [26] made a numerical study of natural convection of air in a differentially heated cubical enclosure with a thick fin placed vertically in the middle of the hot wall; he investigated the variation of overall Nusselt number with Rayleigh number and thermal conductivity ratio. He [27] studied also a numerically natural convection heat transfer in a cubical enclosure with two active sectors on one vertical wall over a wide range of Rayleigh number. He described the flow patterns and temperature distribution and proposed an expression for overall heat transfer. In another study, Frederick and Moraga [28] investigated numerically three dimensional natural convection of air in a cubical enclosure with a fin on the hot wall. It was concluded that for 10^5 Ra 10^6, a fin of partial width is more effective in promoting heat transfer than a fin of full width.

Bocu and Altac [29] studied numerically laminar natural convection heat transfer in 3D rectangular air filled enclosures, with pins attached to the active wall. The Rayleigh numbers considered in their study ranges from 10^5 to 10^7; they concluded that the mean Nusselt number increases with increasing Rayleigh number for a fixed case. The present investigation is an extension of the work already established by Belkadi et al. [30] which they treated similar cavity in two dimensional study. The most part of our simulation is to show the three dimensional effects and the influence of the hot wavy wall with partial partitions on the laminar natural convection.

Analysis

The geometry and coordinate system are illustrated in Figure 1. The geometry is simply represented by a cubical cavity. This latter has a hot surface geometry wavy with partitions introduced at the crests with the constant temperature, T_h. The opposite wall is straight and has a constant colder temperature Tc. The others are thermally insulated. The study has been carried out for different partitions lengths and for two Rayleigh numbers (10^5 and 10^6); the Prandtl number is equal to 0.71.

Governing equations

The viscous incompressible flow inside a cubical enclosure and a temperature distribution are described by the Navier-Stokes and the energy equations. The governing equations for this case are for laminar and three dimensional flow. In the model development, the following assumptions are adopted:

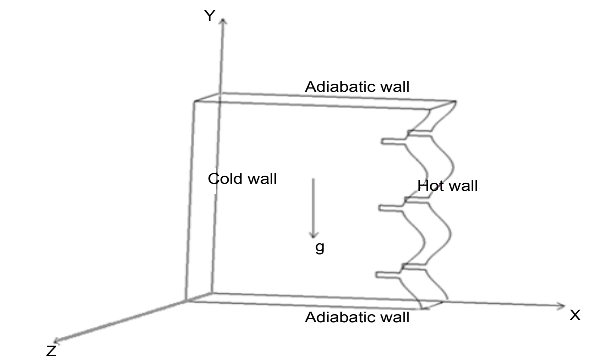

Figure 1: Physical domain.

Citation: Zemani F, Sabeur-Bendehina A, Boussoufi M (2014) Effect of Partial Partitions on Natural Convection in Air Filled Cubical Enclosure with Hot Wavy Surface. J Chem Eng Process Technol 5: 191. doi: 10.4172/2157-7048.1000191

- The process is in steady state.
- The Boussinesq approximation applies, which implies that except for the density in the gravitational term, all other properties in the governing equations are kept constant.
- There is no source or sink in the system.

Once above assumptions are employed into the conservation equation of mass, momentum and energy and the following dimensionless variable are introduced:

$$X=\frac{x}{L}\quad Y=\frac{y}{L}\quad Z=\frac{z}{L} \tag{1}$$

$$U=\frac{u}{\nu/L}\quad V=\frac{v}{\nu/L}\quad W=\frac{w}{\nu/L} \tag{2}$$

$$P=\frac{p+\rho g y}{\rho(\nu/L)^2} \tag{3}$$

$$\theta=\frac{T_0-T_c}{T_h-T_c}\quad T_0=\frac{T_h+T_c}{2} \tag{4}$$

The dimensionless governing equations in Cartesian coordinates for the present study then take the following forms:

Continuity

$$\frac{\partial U}{\partial X}+\frac{\partial V}{\partial Y}+\frac{\partial W}{\partial Z}=0 \tag{5}$$

X Momentum

$$U\frac{\partial U}{\partial X}+V\frac{\partial U}{\partial Y}+W\frac{\partial U}{\partial Z}=-\frac{\partial P}{\partial X}+Pr\left(\frac{\partial^2 U}{\partial X^2}+\frac{\partial^2 U}{\partial Y^2}+\frac{\partial^2 U}{\partial Z^2}\right) \tag{6}$$

Y Momentum

$$U\frac{\partial V}{\partial X}+V\frac{\partial V}{\partial Y}+W\frac{\partial V}{\partial Z}=-\frac{\partial P}{\partial Y}+Pr\left(\frac{\partial^2 V}{\partial X^2}+\frac{\partial^2 V}{\partial Y^2}+\frac{\partial^2 V}{\partial Z^2}\right)+Ra\,Pr\,\theta \tag{7}$$

Z Momentum

$$U\frac{\partial U}{\partial X}+V\frac{\partial W}{\partial Y}+W\frac{\partial W}{\partial Z}=-\frac{\partial P}{\partial Z}+Pr\left(\frac{\partial^2 W}{\partial X^2}+\frac{\partial^2 W}{\partial Y^2}+\frac{\partial^2 W}{\partial Z^2}\right) \tag{8}$$

Energy

$$U\frac{\partial \theta}{\partial X}+V\frac{\partial \theta}{\partial Y}+W\frac{\partial \theta}{\partial Z}=\left(\frac{\partial^2 \theta}{\partial X^2}+\frac{\partial^2 \theta}{\partial Y^2}+\frac{\partial^2 \theta}{\partial Z^2}\right) \tag{9}$$

Knowing that $0 \le Y \le 1$ and $0 \le X \le f(Y)$ with $f(Y)=[1-amp+amp(\cos 2nY)]$ and $0 \le Z \le 1$ with n and amp are respectively, the number of undulations and amplitude as proposed by Adjlout et al. [11]. The geometry of the tested cavity is shown in Figure 1.

The study is completed with the definition of the following boundary conditions:

$\theta = 0.5$ U=V=W=0 on the hot wall

$\theta = -0.5$ U=V=W=0 on the cold wall

$\frac{\partial \theta}{\partial Y}=0$ U=V=W=0 on the adiabatic walls

It can be seen from the above dimensionless equations that the Rayleigh number and Prandtl number are two of the important model parameters for a given flow geometry. The overall heat transfer characteristics are described by the average Nusselt number which is defined over the hot wall as follows:

$$\overline{Nu}=\frac{1}{L}\int_{X=0}^{1}\int_{Z=0}^{1}\frac{\partial \theta}{\partial Y}dXdZ \tag{10}$$

Numerical method and model validation

A 3D uniform and staggered grid is used with a control volume formulation for the discretization. The central difference scheme is employed for the convective diffusive transport variables. Pressure correction and velocity correction are implemented in accordance with the SIMPLE algorithm to achieve a converged solution. The discretized algebraic equations are solved by the tri-diagonal matrix algorithm (TDMA). Relaxation factors of about 0.2-0.7 are used for the velocity components, while relaxation factors of about 0.5-0.8 are adopted for the temperature and pressure corrections. The adiabatic boundary condition is treated by the additional source term method. In the present study, typically 2000-8000 outer iterations are required to achieve convergence. For the convergence criteria, the relative variations of the temperature and velocity between two successive iterations are imposed to be smaller than the specified accuracy levels of 10^{-6}. A grid independency test was carried out, and the results are indicated in Table 1. Three sets of grids 54 × 54 × 54, 60 × 60 × 60 and 67 × 67 × 67 were employed; the case with 67×67×67 grids (Figure 2) was used for taking both the accuracy and convergence rate into account.

Results and Discussion

The streamlines and Isotherms in the computation domain for the case mentioned above and for Ra= 10^5 and 10^6 and different partitions length and in different planes XYZ. It was noticed in Figures 3 and 4 that near the wavy geometry, the thermal boundary layer thickness increases and decreases just before the partitions or just after a trough resulting in a big decrease in the global heat transfer along the hot geometry. When Compared with the cube and the undulated cavity without partitions, the thermal boundary thickness appears to enhance with the presence of partitions. Every partition takes part to the thickening of the thermal

Grid	54 × 54 × 54	60 × 60 × 60	67 × 67 × 67	Bessov [5]
Nu_a	4.458	4.325	4.287	4.339

Table 1: Comparison of average Nusselt Number at several grids for cubical cavity Ra= 10^5.

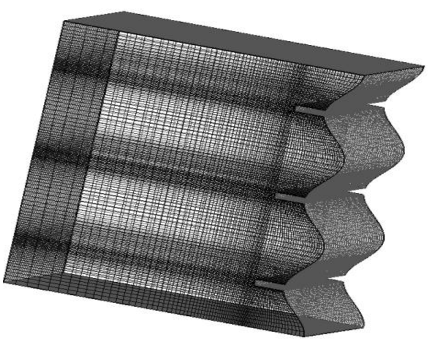

Figure 2: Grid of the geometry.

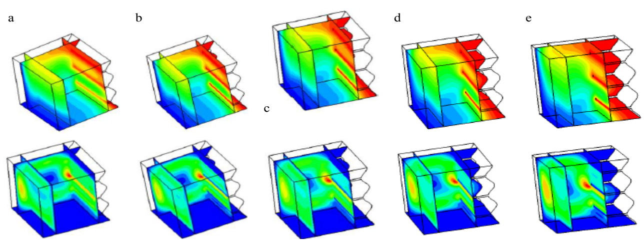

Figure 3: Temperature field and Isosurfaces of the absolute values of the vorticity at Ra=10^5 for all partitions Length and at different planes: X=0.84 Y=0.2 and Y=0.8 a-1 amp, b-2 amp, c-3amp, d-4 amp, e-5 amp.

Citation: Zemani F, Sabeur-Bendehina A, Boussoufi M (2014) Effect of Partial Partitions on Natural Convection in Air Filled Cubical Enclosure with Hot Wavy Surface. J Chem Eng Process Technol 5: 191. doi: 10.4172/2157-7048.1000191

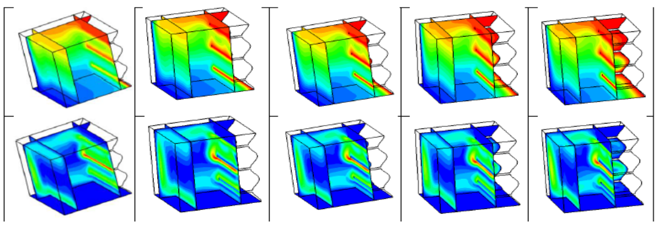

Figure 4: Temperature field and Isosurfaces of the absolute values of the vorticity at Ra=10^6 for all partitions Length and at different planes: X=0.84 Y=0.2 and Y=0.8 a-1 amp, b-2 amp, c-3amp, d-4 amp, e-5 amp.

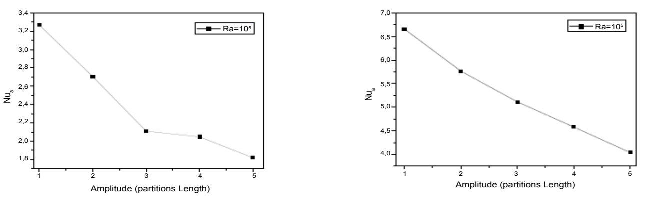

Figure 5: Average Nusselt number variation according to amplitude(partition length) for Ra=10^5 and Ra=10^6.

Figure 6: Local Nusselt Number distributions at the heated wall for a-Cube ,b-undulated cavity without partitions c-undulated cavity with partitions (1amp) d- undulated cavity with partitions (2amp) e-undulated cavity with partitions (3amp) for Ra=10^5.

Geometry (2D)	square	Undulated cavity without partitions	Undulated cavity With partitions
Nu_a	4.52	3.51	3.36
Geometry (3D)	Cube	Undulated cavity without partitions	Undulated cavity With partitions
Nu_a	4.32	3.48	3.26

Table 2: Comparison of the Mean Nusselt Number between the cubical and undulated enclosure (Ra=10^5).

Geometry (2D)	square	Undulated cavity without partitions	Undulated cavity With partitions
Nu_a	8.82	7.55	6.97
Geometry (3D)	Cube	Undulated cavity without	Undulated cavity With

Table 3: Comparison of the Mean Nusselt Number between the cubical and undulated enclosure (Ra=10^6).

Geometry	1 Amp	2 Amp	3 Amp	4 Amp	5 Amp
Nu_a	3.26	2.7	2.11	2.04	1.82

Table 4: Mean Nusselt Number for tested enclosures for (Ra=10^5).

Geometry	1 Amp	2 Amp	3 Amp	4 Amp	5 Amp
Nu_a	6.65	5.76	5.11	4.58	4.03

Table 5: Mean Nusselt Number for tested enclosure for (Ra=10^6).

boundary layer. That shows the induction of the partition length on isotherms and streamlines distributions. It was clearly seen that an increase in partitions length seems to increase in the thermal boundary thickness.

Figure 6 presents the local Nusselt Number distributions at the heated wall for undulated cavity with wavy hot geometry with partitions, undulated cavity without partitions and cube (x=1) for Ra=10^5. The wavy tendency of the local Nusselt number is describing the cavities with corrugated wall. It was observed also that for the undulated cavity, the heat transfer decreases notably compared with the undulated cavity without partition and the cube as illustrated in Table 2. The relative decrease of the global heat transfer for the cavity with hot surface geometry and partitions reaches 40% for Ra=10^6 compared with a cube.

The distribution of average Nusselt number for different partition length are represented in Figure 5 for all Rayleigh numbers investigated (Ra=10^5 and 10^6 respectively). It was clearly seen that there is an influence of the partition length on the mean Nusselt number so; the trend of the curve is notably decreasing with an increase in the partition length (Tables 3-5).

Conclusion

The present investigation deals with the effect of the presence of a partition on the hot wall geometry of undulated cubical cavity, air filled, and differentially heated enclosure. These partitions caused decrease of up 40% in the heat transfer with respect to the case at the same Rayleigh number. As Ra grew, boundary layer got progressively thinner, and the effects of the partitions on the temperature field became more and more localized around the partitions. This was confirmed by the results, the percentage heat transfer reduction grows with the partitions length which is a function of amplitude of the waviness hot geometry. As the insulating effect was related to a local distortion of the temperature field, it can be concluded that multiple partitions would be required in such cavities to reach the levels of heat transfer reduction.

References

1. De Vahl Davis G (1983) Natural Convection of air in a square cavity: a benchmark numerical solution, International Journal of Numerical Methods in Fluids 3: 249-264.
2. Catton I (1978) Natural Convection in enclosures. In: Proceeding of the sixth international Heat Transfer Conferences 6: 13-31.
3. Ostrach S (1972) Natural convection in enclosures. Advances in Heat Transfer 8: 161-227.
4. Yang KT (1988) Transitions and bifurcations in laminar buoyant flows in confined enclosures. J Heat Transfer 110: 1191-1204.
5. Bessonov OA, Brailovskaya VA, Nikitin SA, Polezhaev VI (1997) Three dimensional natural convection in a cubical enclosure: A Bench Mark numerical solution. Proceeding of computational heat transfer symposium.
6. Mallinson JD, de Vahl Davis G (1977) Three-dimensional natural convection in a box: a numerical study. J Fluid Mech 83: 1-31.
7. Lee TS, Son GH, Lee JS (1988) Numerical predictions of three-dimensional natural convection in a box. Proc 1st KSME-JSME Thermal and Fluids Engng Conf. 2: 278-283.
8. Lankhorst AM, Hoogendoorn CJ (1988) Three-dimensional numerical calculations of high Rayleigh number natural convective flows in enclosed cavities. Proc Natn Heat Transfer Conf ASME HTD-96, 3: 463-470.
9. Fusegi T, Hyin JM, Kuwahara K (1991) A numerical study of 3D natural convection in a differently heated cubical enclosure. Fluid Dynamics Research 8: 5-6.
10. Bilski SM, Lloyd JR, Yang KT (1986) An experimental investigation of the laminar natural convection velocity in square and partitioned enclosure. In: Proceeding of the eight international Heat Transfer Conference 4: 1531-1518.
11. Adjlout L, Imine O, Azzi A, Belkadi A (2002) Laminar natural convection in an inclined cavity with a wavy wall. International Journal of Heat and Mass Transfer 45: 2141-2152.
12. Lee TS, Son GH, Lee JS (1989) Numerical study on natural convection in three-dimensional rectangular enclosures. KSME Journal 3: 50-55.

13. Hiller WJ, Koch S, Kowalewski TA (1989) Three-Dimensional Structures in Laminar Natural Convection in a Cubic Enclosure. Experimental Thermal and Fluid Science 2: 34-44.

14. Gunes H (1992) Three dimensional thermal convection in enclosures and vertical channels: applications to cooling of electronic equipment. Theses and Dissertations 148.

15. Yu E, Joshi Y (1997) A numerical study of three-dimensional laminar natural convection in a vented enclosure. Int J Heat and Fluid Flow 18: 600-612.

16. Akrour D, Bennacer R, Kalahe D (2000) Natural convection of air in a square cavity. Laboratoire de Mécanique des Fluides, Faculté de Physique.

17. Corzo SF, Damián SM, Ramajo D, Nigro NM (2011) Numerical simulation of natural convection phenomena, Mecánica Computacional 30: 277-296.

18. Kürekci NA, Özcan O (2012) An experimental and numerical study of laminar natural convection in a differentially-heated cubical enclosure. Isi Billimi ve Teknigi Dergisi 32: 1-8.

19. Ahmadi M, Dastgerdi BB (2012) Fluid flow and heat transfer characteristics of natural convection in square 3-D enclosure due to discrete sources. Proc. of the International Conference on Future Trends in Automation and Robotics - FTAR -114.

20. Belarche L, Abourida B, Mediouni T (2013) Three-dimensional Numerical Study of Natural Convection in a Cubical Enclosure with Two Heated Square Sections Submitted to Periodic Temperatures. IJCSI International Journal of Computer Science Issues 10: 8-16.

21. Sabeur-Bendehina A, Imine O, Adjlout L, Imine B (2007) Effect of the hot surface geometry on laminar Natural Convection in Cubical Air Filled Enclosures. Proceedings of the Third International Conference on Thermal Engineering: Theory and Applications.

22. Fusegi T, Hyun JM, Kuwahara K (1991) Transient three-dimensional natural convection in a differentially heated cubical enclosure. Int J Heat Mass Transfer 34: 1559-1564.

23. Fusegi T, Hyun JM, Kuwahara K (1992) Three-dimensional numerical simulation of periodic natural convection in a differentially heated cubical enclosure. Applied Scientific Research 49: 271-282.

24. Fusegi T (1992) Numerical simulations of natural convection in a differentially heated cubical enclosure with a partition. Int J Heat and Fluid Flow 13: 176-183.

25. da Silva AK, Gosselin L (2005) On the thermal performance of an internally finned three-dimensional cubic enclosure in natural convection. International Journal of Thermal Sciences 44: 540-546.

26. Frederick RL (1997) Natural convection heat transfer in a cubical enclosure with two active sectors on one vertical wall. Int Comm Heat Mass Transfer 24: 507-520.

27. Frederick RL (2007) Heat transfer enhancement in cubical enclosures with vertical fins. Applied Thermal Engineering 27: 1585-1592.

28. Frederick RL, Moraga SG (2007) Three-dimensional natural convection in finned cubical enclosures, International Journal of Heat and Fluid Flow 28: 289-298.

29. Bocu Z, Altac Z (2011) Laminar natural convection heat transfer and air flow in three-dimensional rectangular enclosures with pin arrays attached to hot wall. Applied Thermal Engineering 31: 3189-3195.

30. Belkadi M, Aounallah M, Imine O, Adjlout L, Addad Y (2006) Free convection in an inclined square cavity with partial partitions on a wavy hot wall. Progress in Computational Fluid Dynamics 6: 428-434.