**HEFAT2007**
**5th International Conference on Heat Transfer, Fluid Mechanics and Thermodynamics**
**Sun City, South Africa**
**Paper number: BA4**

# TRANSIENT NATURAL CONVECTION IN CLOSED AND INCLINED CUBICAL ENCLOSURES. APPLICATION TO ELECTRONIC EQUIPMENT THERMAL REGULATION

A. Baïri (1)*, N. Laraqi (1), Z. Zouaoui(2), J.M. García de María(3), N. Alilat (1) and H. Ouslimani(4)

(1)University of Paris 10, GTE, Department of Heat transfer;
50, rue de Sèvres, F-92410 Ville d'Avray, France.
(2)North East Wales Institute, Mold Road Wrexham LL11 2AW, UK.
(3)Universidad Politécnica de Madrid, Department of Applied physics,
Ronda de Valencia, 3, E-28012 Madrid, Spain.
(4)University of Paris 10, GEA; 50, rue de Sèvres, F-92410 Ville d'Avray, France.
*E-mail: abairi@u-paris10.fr, bairi.a@gmail.com

## ABSTRACT

Certain electronic components are very sensitive to temperature variations during their operations. This is true particularly for electronic systems containing high density integrated circuits, which are increasingly a characteristic of today's equipment due to the ever evolving process of miniaturisation. The correct operation of such electronic systems is related to their thermal state. Therefore, thermal regulation is required for their proper dynamic operation. This regulation is complex in the case of a high level of integration and confinement to small spaces. The proper sizing of the closed or limited opening casing is, therefore, necessary in order to ensure the correct operation of the equipment in confined spaces. This paper presents a treatment of the particular case of the transient natural 2D convection in a cubical enclosure filled with air. The active walls are differentially heated and maintained isothermic. These walls can remain vertical, but can also be inclined with respect to the gravity field. The dynamic and thermal fields within the case are analysed numerically using the finite volume method. An experimental test rig is developed for the thermal analysis by examining the temperature distribution in the immediate vicinity of the hot wall, which simulates the electronic equipment.

## INTRODUCTION

The natural convection is an excellent mean of thermal exchanges combining simplicity, effectiveness, energy efficiency and, therefore, environment preservation. It is also of particular interest for the limitation of undesirable noise, which is often induced by external systems such as cooling fans. Natural convection in closed spaces is dependent on a number of geometric and thermal parameters. The research work presented here focuses on two of them: the temperature difference between the active hot and cold walls and the inclination angle of the case. A number of research studies dealt with the permanent natural convection for the geometry under consideration in this paper. These include the work of Ostrach [1], Catton [2], Baïri [3] and more recently Khalifa [4,5]. This paper is concerned with the transient aspect of natural convection, which has only been addressed by fewer research studies. For convection in enclosures with different conditions, a detailed research review has been carried out by Fusegi et al. [6], considering spatial and temporal variations of thermal boundary conditions, variable properties effects and multi-dimensionalities. The case of transient natural convection in rectangular vertical cavities was treated numerically by Bae et al [7]. They considered the case where the hot walls are composed of discrete elements. It was shown in this work that the temperatures of the hot elements can reach maximum values in transient regime higher that those reached in the steady state regime, which is important for the correct operation of electronic circuits.

In the present work the 2D flow in the cubic case with the active hot and cold walls being either vertical or inclined, is examined. The channel of the case is considered to be adiabatic. The phenomenon is analysed with the average Nusselt number on the hot wall. Only the temperature and velocity distributions in the cavity obtained numerically at representative times within the transient phase are presented here. The thermal aspect at the walls was examined experimentally by means of a specific test bench. The discrepancies between the numerical and experimental results for high values of the Rayleigh number is small and the results found from the analysis of the Nusselt number are in good agreement with previous studies published in the litterature.

## NOMENCLATURE

| | | |
|---|---|---|
| $a$ | [m$^2$.s$^{-1}$] | thermal diffusivity of the air |
| $C$ | [J.kg$^{-1}$K$^{-1}$] | specific heat |
| $g$ | [m.s$^{-2}$] | acceleration of gravity |
| $h$ | [Wm$^{-2}$K$^{-1}$] | mean convection coefficient |
| $H$ | [Wm$^{-2}$K$^{-1}$] | height of the cavity; distance hot-cold walls |
| $\overline{Nu_c}$ | [-] | calculated mean Nusselt number over the hot wall |
| $\overline{Nu_m}$ | [-] | measured mean Nusselt number over the hot wall |
| $p$ | [Pa] | pressure (Pa) |
| $p*$ | [-] | dimensionless pressure |
| $Pr$ | [-] | Prandtl number |
| $Ra$ | [-] | Rayleigh number |
| $t$ | [s] | time |
| $t*$ | [-] | dimensionless time |
| $T$ | [K] | local temperature of the fluid |
| $T_c$ | [K] | temperature of the cold wall |
| $T_h$ | [K] | temperature of the hot wall |
| $T*$ | [-] | dimensionless temperature |
| $u$, $v$ | [m.s$^{-1}$] | flow velocity components in $x$ and $y$ directions respectively |
| $u*$, $v*$ | [-] | dimensionless flow velocity components $x$ and $y$ directions |
| $x$, $y$ | [m] | cartesian coordinates |
| $x*$, $y*$ | [-] | dimensionless cartesian coordinates |
| Greek symbols | | |
| $\alpha$ | [deg] | angle of inclination of the cavity |
| $\beta$ | [K$^{-1}$] | expansion coefficient of the air |
| $\Delta T$ | [K] | difference of temperatures $\Delta T=T_h-T_c$ |
| $\varepsilon_h$, $\varepsilon_c$, $\varepsilon_p$ | [-] | global infrared emissivity of the hot, cold and passive wall |
| $\varphi$ | [W.m$^{-2}$] | heat flux density |
| $\lambda_i$ | [Wm$^{-1}$K$^{-1}$] | thermal conductivity of the passive walls |
| $\lambda$ | [$Wm^{-1}K^{-1}$] | thermal conductivity of the air |
| $\mu$ | [Pa.s] | dynamic viscosity of the air |
| $\theta$ | [-] | reduced temperature of the fluid |
| $\rho$ | [kg.m$^{-3}$] | density of the air |

## CASE STUDY

The case of the natural convection taking place in cubic air-filled cavities is examined The hot and cold active walls generating the flow inside the enclosure are facing each other and are situated at different angles $\alpha$ with respect to the horizontal. The cold wall is maintained isothermal at $T_c$ while the hot wall is heated at temperature $T_h$. The four other walls of the cavity are considered to be adiabatic. Under such conditions the flow taking place can be considered two-dimensional. This leads to the treatment of the case shown in Figure 1.

## GOVERNING TIME-DEPENDANT EQUATIONS

These equations for the treated problem are

Continuity equation

$$\frac{\partial u^*}{\partial x^*}+\frac{\partial v^*}{\partial y^*}=0 \tag{1}$$

where $u*$ and $v*$ are defined as

$$x^*=\frac{x}{H};\quad y^*=\frac{y}{H};\quad u^*=\frac{u}{a/H};\quad v^*=\frac{v}{a/H} \tag{2}$$

Momentum equations

$$\frac{\partial u^*}{\partial t^*}+u^*\frac{\partial u^*}{\partial x^*}+v^*\frac{\partial u^*}{\partial y^*}=Ra\,Pr\,T^*\cos\alpha+A$$

where $A=-\frac{H^3 g}{a^2}\left(\frac{\partial p^*}{\partial x^*}+\cos\alpha\right)+Pr\nabla^2 u^*$

$$\frac{\partial v^*}{\partial t^*}+u^*\frac{\partial v^*}{\partial x^*}+v^*\frac{\partial v^*}{\partial y^*}=-Ra\,Pr\,T^*\sin\alpha+B \tag{3}$$

where $B=-\frac{H^3 g}{a^2}\left(\frac{\partial p^*}{\partial y^*}-\sin\alpha\right)+Pr\nabla^2 v^*$

Energy equation

$$\frac{\partial T^*}{\partial t^*}+u^*\frac{\partial T^*}{\partial x^*}+v^*\frac{\partial T^*}{\partial y^*}=\nabla^2 T^* \tag{4}$$

In (3) and (4),

$$\nabla^2=\frac{\partial^2}{\partial x^{*2}}+\frac{\partial^2}{\partial y^{*2}} \tag{5}$$

are the 2D cartesian Laplacian of $u*$, $v*$ and $T*$

The numbers of Rayleigh and Prandtl are defined as

$$Ra=\frac{g\beta H^4\rho}{\mu\lambda a}\varphi;\quad Pr=\frac{\mu C}{\lambda} \tag{6}$$

and the others parameters are

$$p^*=\frac{p}{\rho g H};\quad t^*=\frac{at}{H^2};\quad T^*=\frac{T-T_c}{\dfrac{\varphi}{\lambda/H}}$$

$$\varphi=\frac{\lambda(T-T_c)_{x*=0}}{H};\quad \theta=\frac{T-T_c}{T_h-T_c} \tag{7}$$

The reduced temperature $\theta$ is used in this work for the presentation of the fluid temperature field. The thermal boundary conditions of the treated case are

$$(T^*)_{x*=0}=1;\ (T^*)_{x^*=1}=0;\ \left(\frac{\partial T^*}{\partial y^*}\right)_{y^*=0;y^*=1}=0$$

$$(u^*,v^*)_{x*=0;x*=1;y*=0;y*=1}=0 \tag{8}$$

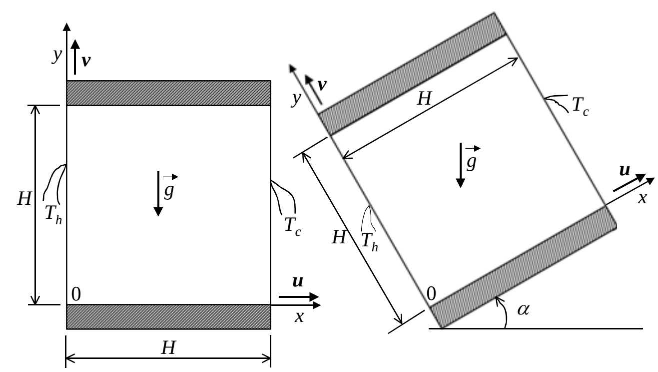


**Figure 1** The studied case

## NUMERICAL METHOD

The air is assumed to be isotropic and all its properties are evaluated at the mean temperature of each volume control. Calculations are carried out by means of the control volume formulation in accordance with the SIMPLE algorithm. The 2D convective flow is assumed as incompressible. A considerable research effort was necessary in order to find a proper under-relaxation factor that prevents the algorithm's divergence. The solutions are assumed to converge when the difference between two successive iterations is less than $10^{-5}$ for the velocities, and $10^{-6}$ for the temperature. Based on the Nusselt number values the optimal mesh is derived. A 195 x 195 discretisation was selected with a refined mesh near the active walls in order to take into consideration the viscous effects and the thermal exchanges in the boundary layer. For inclinations involving strong turbulence intensity, we associate the RNG-kε model to the Navier-Stokes equations for an accurate determination of the temperature gradients and consequently of the thermal transfer at the walls. The radiative exchanges in the cavity were taken into consideration and determined analytically by a calculus based on the radiosity method detailed in [8]. These calculations are based on effective temperature fields measured on the hot walls of the cavity for all the configurations under consideration. The global infrared emissivities of all the walls $\varepsilon_h, \varepsilon_c, \varepsilon_p$ measured in the laboratory are also used.. The fact that the walls are supposed to be adiabatic was taken into account for the experimental work. Accurate calculations of losses through the walls were undertaken in order to validate the mathematical model developed by comparison between the measurements and the calculations.

## EXPERIMENTAL STUDY

An experimental rig was set up to examine this phenomenon, which is limited to the thermal properties at the walls. A view of the experimental assembly used in our work is shown in Figure 2. The studied cubic cavity has an inner side of 200 mm. The hot wall of the cavity is made up of a printed circuit board (PCB) plate, on which two independent resistors are printed. The convection measurements are taken from the central area. Three spaces were selected, from which surface temperatures were measured with thermo-couples. They go through the PCB plate from behind in order not to disturb the flow in the vicinity of the wall. These thermo-couples are used to verify the isothermic property of the surface and to keep the temperaure constant with an error of ±0.1°C using a PID controller implemented on a micro-computer. A second area, knwon as guarding area, lies around the first central one and is controlled independently to avoid lateral losses. The cavity channel is bound to this zone. The temperature is measured at several point in this zone and its isothermal property is ensured at ± 0.1°C in nonstationary permanent regimes. The back face of the plate is well insulated with slabs of extruded polyurethane of 3 x 40 mm in thickness (thermal conductivity measured in the laboratory; $\lambda_i$=0.028 ± 0.001 $Wm^{-1}K^{-1}$). In order to establish the desired temperature $T_h$, total power measurements are carried out on the plate.

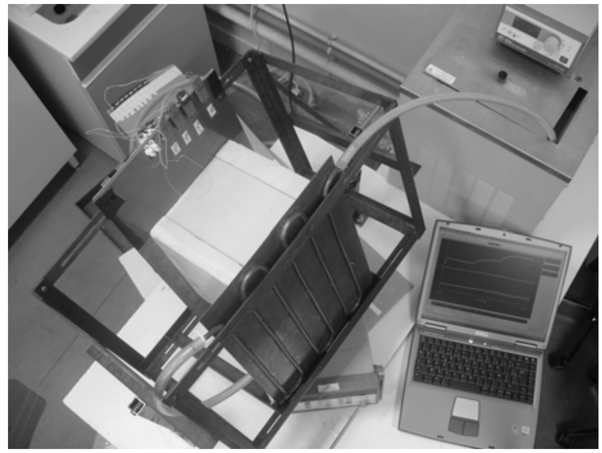

**Figure 2** The experimental assembly

The cold wall consists of a copper plate of 3 mm in thickness with a cooper coil welded onto its back face. The temperature of the cold plate $T_c$ is imposed with a precision of ± 0.1°C by means of a controlled bath using a cryostat in order to have the desired difference of temperatures $\Delta T=T_h-T_c$. The tests are undertaken for different values of $\Delta T$: 10K, 30K and 70K. The useful surface of the cold wall is covered with an aluminized adhesive film. The channel of the cavity (Figure 3) is made of plates of extruded polyurethane of 40 mm thickness.

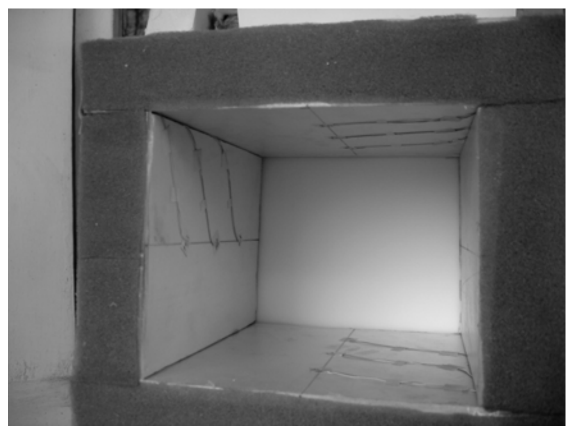

**Figure 3** The channel of the cavity

Preliminary measurements have shown that this thickness is sufficient to ensure the adiabacity of the channel, which is an essential condition in this case study. The temperature is measured every 40 mm over the central lines of the upper side, the lower side and one of the lateral side of the channel on both the internal and external faces.

The system formed by the hot plate, the cold plate and the channel is assembled and laid on a frame, which rotate around a horizontal axis.This allows to change the inclination angle $\alpha$ between 0 and 360 degrees. The air temperature of the environment outside the cavity is also measured. The thermal state of the whole system is determined with a set of 0.1 mm thermocouples, connected to a fast data acquisition system controlled by a computer.

## RESULTS

Figures 4 and 5 show temperature and velocity fields for 3 inclination angles ($\alpha$=0, 25 et 45 degrees), with $T_c$=0°C and $T_h$=30°C, that is $\Delta T$=30K. The field are presented for three different periods of time, that is $t$=3, 9 et 15 mn. A stratification and a concentrated flow in the vicinity of the active walls are noticed, whereas the centre of the cavity remains practically still. This remains valid as long as the temperature difference remains small leading to Rayleigh numbers $Ra$ below $10^3$. The conductive regime tends to become convective when $Ra$ increases above the critical number, which is between 1.3 x $10^3$ and 2.4 x $10^3$.

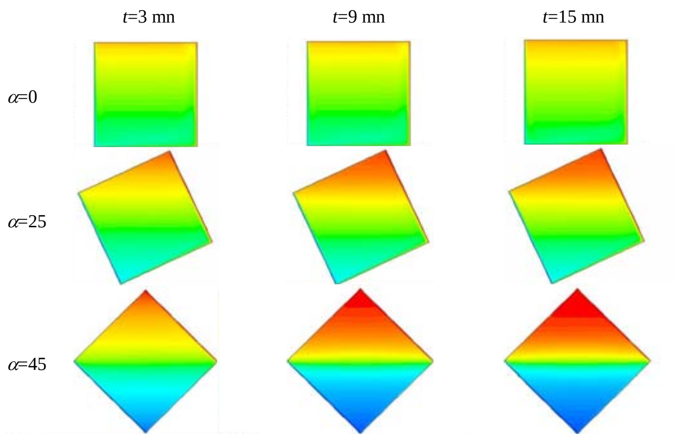


**Figure 4** Temperature fields in the cavity for $\alpha$=0, 25 and 45 degrees, with $t$=3, 9 et 15 mn. $T_c$=0°C et $T_h$=30°C

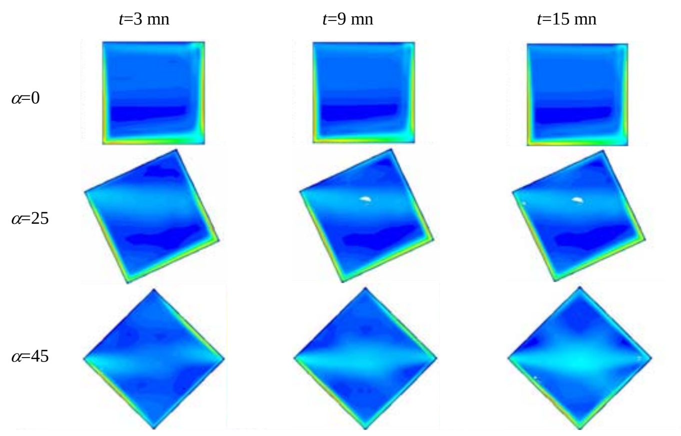


**Figure 5** Velocity fields in the cavity for $\alpha$=0, 25 and 45 degrees, with $t$=3, 9 et 15 mn. $T_c$=0°C et $T_h$=30°C

The Nusselt number evolution with time on the hot and cold walls at $\alpha$=0, 25 and 45 degrees, is shown in figure 6. Note from figure 6 that the setling time for the thermal regime increases with an increase in the inclination angle. The increase is of the order of 50% when $\alpha$ increases from 0 à 45°. Figure 7 shows a comparison between the measured and calculated Nusselt number values at particular times on the hot wall of the cavity. Data dispersion remains bounded and limited. These results correlate well in permanent regime with other research studies.

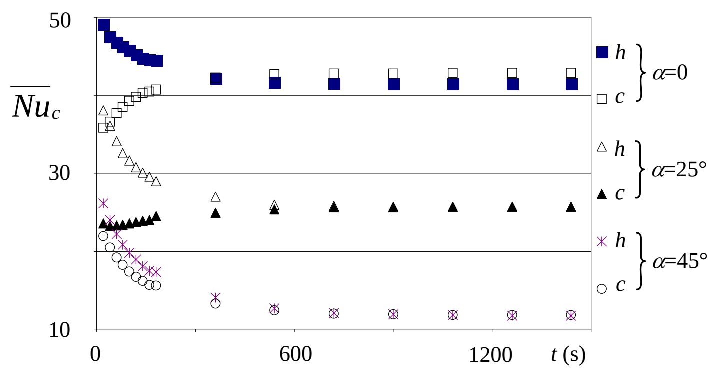


**Figure 6** Calculated Nusselt number $\overline{Nu_c}$ on the hot ($h$) and cold ($c$) walls with respect to time for $\alpha$=0, 25 and 45° .

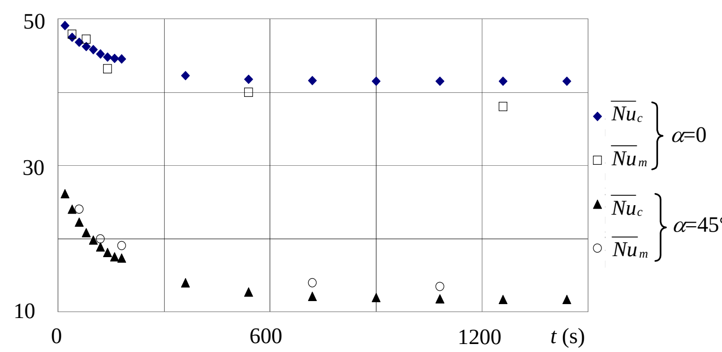


**Figure 7** Comparison between the calculated $\overline{Nu_c}$ and the measured $\overline{Nu_m}$ Nusselt numbers on the hot wall with respect to time for $\alpha$=0 and 45 degrees.

In all case one can notice that the convective exchanges are well correlated with the thermal and dynamics characteristics. The Nusselt number varies with the Rayleigh number and dimensionless time $t^*$ according to a law of type

$$Nu = aRa^n \left(t^*\right)^m \tag{9}$$

The coefficient $a$ and the exponents $m$ and $n$ depend obviously on the configuration and the range of the Rayleigh number $Ra$. These particular laws have already been proposed by the authors in previous work for the steady state regime and research work is presently undertaken to establish these correlations in transient regime. Preliminary results suggest an exponent $n$ between 0.22 and 0.32. However, a definite value has not yet been found for all the configurations dealt with in this research work. The same thing could be said for the exponent $m$, which should logically have a negative value.

## CONCLUSION

The two parameters $\alpha$ et $\Delta T$ have a considerable influence on the natural convection flow setting in the cavity. Such an influence has to be taken into account in the dimensioning of cases containing power electronics components, which are sensitive to the temperature changes during their operation. The work presented here is presently being extended to all inclination angles between 0 and 360 degrees. Correlations of the type $Nu = aRa^n\left(t*\right)^m$ will be proposed.

## REFERENCES


[1] Ostrach S, Natural convection in enclosures, in: Hartnett, Irvine (Eds.), *Advances in Heat Transfer*, Vol. 8, Academic Press, 1972.

[2] I. Catton, Natural convection in enclosures, in: Proc. 6th Int. Heat Transfer Conf., Vol. 6, 1978.

[3] Baïri A., Contribution à la connaissance actuelle des écoulements de convection naturelle dans les cavités parallélépipédiques. Ecoulements laminaires, *Université de Poitiers*, Ecole Nationale Supérieure de Mécanique et d'Aérotechnique, 1983.

[4] Khalifa A-J.N., Natural convective heat transfer coefficient - a review. I. Isolated vertical and horizontal surfaces. *Energy Conversion and Management* 42, 491-504, 2001.

[5] Khalifa A-J.N., Natural convective heat transfer coefficient - a review. II. Surfaces in two- and three-dimensional enclosures, *Energy Conversion and Management* 42, 505-517, 2001.

[6] Fusegi T. and Hyun J. M., Laminar and transitional natural convection in an enclosure with complex and realistic conditions, *Int. J. Heat Fluid Flow*, 15, 258-268, 1994.

[7] Bae J.H., Hyun J.M., Time dependent buoyant convection in an enclosure with discrete heat sources, *International Journal of Thermal Sciences*, vol. 43, 3-11, 2004.

[8] Baïri A., Laraqi N., García de María J.-M., Importance of radiative heat exchange in 2D closed diode cavities applied to solar collectors and building, *International Journal of Sustainable Energy*, vol. 1, 33-44, 2005.