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NATURAL CONVECTION IN A VERTICAL CAVITY WITH HEAT SOURCES SIMULATING ELECTRONIC COMPONENTS

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Abstract. *In this work the effect of natural convection on the fluid flow and heat transfer in a vertical cavity with two protruding heat sources simulating electronic components is studied numerically. The vertical walls are assumed adiabatic and the flow is considered two-dimensional, laminar, in steady state and with constant physical properties. The Boussinesq approximation is adopted to obtain the governing elliptic equations, which are discretized by using the finite control volumes method with the SIMPLE algorithm. The velocity and temperature fields are obtained, as well as the temperature variations along the wall where the sources are mounted and the average Nusselt number as a function of the power dissipated, the distribution of the power in the sources, the spacing between the sources and the cavity aspect ratio. With the results obtained it is possible to define a configuration in which the components (sources) exhibit lower temperature levels thus reducing the possibility of failure.*

Key words: *Natural convection, Electronic components, Thermal cavity.*

1. INTRODUCTION

This work deals with a problem of great technological interest in the area of thermal design and control of electronics equipment. The performance and reliability of the components are strongly dependent on the working temperature. The levels of this temperature must be maintained below a critical limit to avoid superheating, and consequently an increase in the failure rate. However, with the miniaturization of the components, this problem becomes even worse because the volumetric power density increases, thus increasing the local heat fluxes. Therefore the design and also the thermal control of an electronic device must be very efficient because the temperature accounts for about 40% of the failures. Furthermore, the failure rate increases almost exponentially with the increase of the working temperature (Peterson & Ortega, 1990).

Different methods of cooling and different fluids can be utilized to overcome this problem. Usually the thermal behavior is predicted taking into account mostly the air cooling by natural convection due to its simplicity and reliability because it does not depend on additional devices.

There are several review papers in this area of thermal design and control of electronic equipments. See for example Kraus & Bar-Cohen (1983), Incropera (1988), Nakayama & Bergles (1990), Peterson & Ortega (1990), Bergles (1991) and Bar-Cohen (1992), among others.

There are basically two configurations where the majority of the numerical works involving the study of heat sources simulating electronic components subjected to different boundary conditions are concentrated. One of them considers the discrete heat sources flush mounted in the walls or with infinitesimal thickness: Keyhani *et al*. (1988), Wang *et al*. (1988), Chadwick *et al*. (1991), Ho & Chang (1994), Dias Júnior *et al*. (1997) and Ramos *et al*. (1998a), among others. Another configuration, more realistic, considers the heat sources as protruding volumes that partially block the flow: Afrid & Zebib (1987), Shakerin (1987), Chen & Kuo (1988), Shakerin *et al*. (1988) and Ramos *et al*. (1998b), among others.

In the present work a numerical analysis of the natural convection within a vertical cavity with protruding heat sources dissipating heat uniformly by unit volume simulating electronic components, as shown schematically in Figure 1, is performed. The flow velocity and temperature fields are obtained as well as the variation of the temperature along the surface where the sources are mounted and the value of the average Nusselt number in the sources, as a function of the power dissipated, the power distribution in the sources, the spacing between the sources and the cavity aspect ratio.

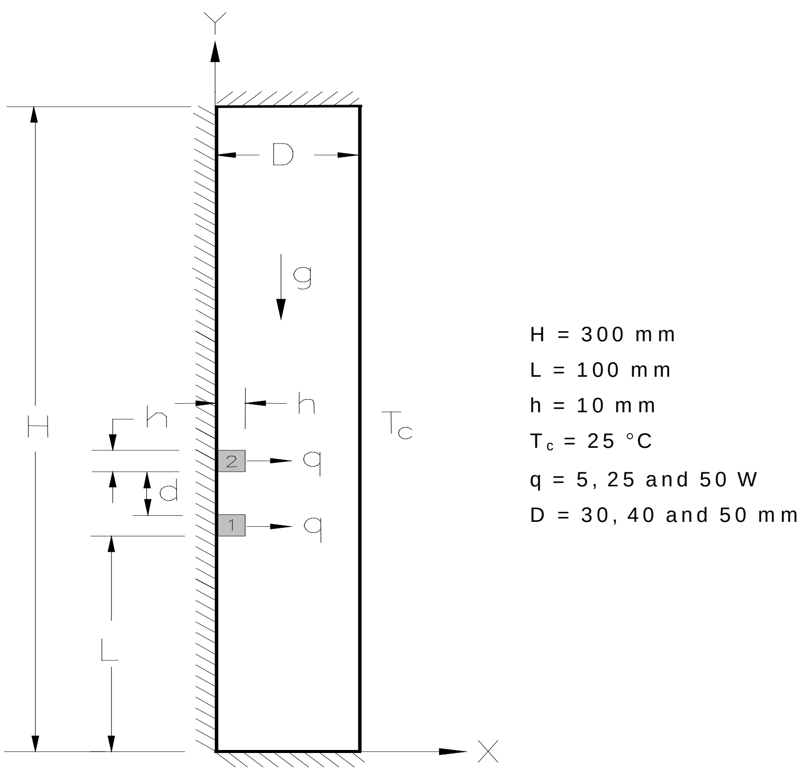

Figure 1: Vertical cavity with heat sources simulating electronic components.

2. MATHEMATICAL MODELING

2.1 Considerations

The flow is assumed as laminar, two-dimensional and in steady state. The physical properties do not vary with temperature, the thermal radiation effects are neglected and the Boussinesq approximation is adopted.

2.2 Dimensionless Parameters

The following dimensionless variables were used in this study, for describing the governing equations:

Distances: $x^* = x / L$ and $y^* = y / L$

Velocities: $u^* = u / (\alpha / L)$ and $v^* = v / (\alpha / L)$

Pressure: $P^* = P / [\rho(\alpha / L)^2]$

Temperature: $T^* = (T - T_\infty) / (q_1 L^2 / k_f)$

Modified Rayleigh number: $Ra_q = g\beta q_1 L^5 / (k_f \alpha \nu)$

Source term: $S^* = k_s / k_f$ (sources) or $S^* = 0$ (fluid)

Properties: $k^* = k_s / k_f$ and $\mu^* = \infty$ (sources) or $k^* = 1$ and $\mu^* = 1$ (fluid)

2.3 Formulation

The governing equations of the problem can be written in the following dimensionless form:

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

$$u^* \frac{\partial u^*}{\partial x^*} + v^* \frac{\partial u^*}{\partial y^*} = -\frac{\partial P^*}{\partial x^*} + \mu^* Pr \left(\frac{\partial^2 u^*}{\partial x^{*2}} + \frac{\partial^2 u^*}{\partial y^{*2}} \right) \tag{2}$$

$$u^* \frac{\partial v^*}{\partial x^*} + v^* \frac{\partial v^*}{\partial y^*} = -\frac{\partial P^*}{\partial y^*} + \mu^* Pr\left(\frac{\partial^2 v^*}{\partial x^{*2}} + \frac{\partial^2 v^*}{\partial y^{*2}}\right) + Ra_q PrT^* \tag{3}$$

$$u^* \frac{\partial T^*}{\partial x^*} + v^* \frac{\partial T^*}{\partial y^*} = k^*\left(\frac{\partial^2 T^*}{\partial x^{*2}} + \frac{\partial^2 T^*}{\partial y^{*2}}\right) + S^* \tag{4}$$

2.4 Boundary Conditions

The problem boundary conditions can be expressed in the form:

Left wall ($x^* = 0$): $u^* = v^* = \dfrac{\partial T^*}{\partial x^*} = 0$ (5)

Right wall ($x^* = D/L$): $u^* = v^* = T^* = 0$ (6)

At the base ($y^* = 0$) and at the top ($y^* = H/L$): $u^* = v^* = \dfrac{\partial T^*}{\partial y^*} = 0$ (7)

2.5 Average Nusselt Number

The average Nusselt number in the source faces can be determined by the coupling of the conduction-convection equations at the interface (i), resulting:

$$\overline{Nu_i} = -\frac{L}{P}\int_0^P \frac{1}{T_i^*}\left(\frac{\partial T^*}{\partial \eta^*}\right) d\varphi^* \tag{8}$$

where:

φ^* : generalized coordinate (direction x^* or y^*);
η^* : generalized coordinate (x^* for direction y or y^* for direction x);
P : source perimeter (3h/L);
i : refers to the interface.

3. NUMERICAL SOLUTION

3.1 Method

The equations were discretized by the method of the finite control volumes and solved numerically by the SIMPLE (Semi-Implicit Method for Pressure-Linked Equations) algorithm due to Patankar (1980).

3.2 Grid

Non uniform structured grids were used with smaller constant spacing in the region over the sources and between them, where gradients are steeper, with the spacing smoothly increasing from this region at a rate smaller than 10% to avoid numerical instabilities due to a sudden change in the grid size (Roache, 1976).

A grid test was conducted for the basic configuration presented, utilizing meshes of 20×80, 30×120, 40×160, 50×200 and 60×240, in order to verify the maximum temperature in the flow. It was assumed as the converged value the mesh 60×240, with 24×24 points over the sources, this value being limited by the capacity of the machine and by the processing time needed.

3.3 Convergence

The convergence criterion adopted for the iterative process is the following:

$$\xi = \max\left|\frac{\phi_{i,j}^{(n)} - \phi_{i,j}^{(n-1)}}{\phi_{i,j}^{(n)}}\right| \leq 10^{-4} \tag{9}$$

where:

ξ : admitted tolerance;

ϕ : generalized parameter (u^*, v^* or T^*);

n : refers to the n-th iteration;

i,j : position of the point in the domain.

3.4 Procedure

The following parameters were fixed: cavity height, H = 300 mm; sources height and tickness, h = 10 mm; plate width, W = 1000 mm (perpendicular to the plane); and the distance from the cavity base to the first source, L = 100 mm, which is the characteristic length, as defined by Afrid & Zebib (1987). Furthermore, the ratio between the thermal conductivities of the sources (ceramics) and air was taken as k_s/k_f = 1000 and the ambient air temperature as T_c = 25 °C.

Three different values were chosen for the parameters of interest in the simulation: modified Rayleigh number, Ra_q = 1×10^9, 5×10^9 and 1×10^{10}, corresponding to powers dissipated of 5, 25 and 50 W per source, respectively; dissipation rate, q_2/q_1 = 0.5, 1.0 and 2.0; spacing between sources, d/L = 0.1, 0.2 and 0.3; and cavity width, D/L = 0.3, 0.4 and 0.5, corresponding to the following aspect ratio: D/H = 0.100, 0.133 and 0.167.

The parametric analysis was carried out based on the configuration for intermediate values of the above-mentioned range of variation: Ra_q = 5×10^9, d/L = 0.2, q_2/q_1 = 1.0 and D/H = 0.133. Three of these values were always used, with the other varying within the established limits, to make it possible to study the influence of each parameter on the maximum fluid temperature, which is the parameter of interest in the flow, resulting in the cases presented in Table 1. It is worthy mentioning that the bold numbers in this table are the values that are varying respective to the baseline case 2.

Table 1: Description of the cases to be studied in this work.

Cases	Ra_q	q_2/q_1	d/L	D/H
(1)	$\mathbf{1\times10^9}$	1.0	0.2	0.133
(2)	5×10^9	1.0	0.2	0.133
(3)	$\mathbf{1\times10^{10}}$	1.0	0.2	0.133
(4)	5×10^9	**0.5**	0.2	0.133
(5)	5×10^9	**2.0**	0.2	0.133
(6)	5×10^9	1.0	**0.1**	0.133
(7)	5×10^9	1.0	**0.3**	0.133
(8)	5×10^9	1.0	0.2	**0.100**
(9)	5×10^9	1.0	0.2	**0.167**

4. RESULTS

Figure 2 shows the form of the grid utilized in the numerical solution of the problem. The representation of the real grid format is not possible due to large number of lines.

In Figure 3 are presented the qualitative results for the streamlines and isotherms for the baseline case above mentioned.

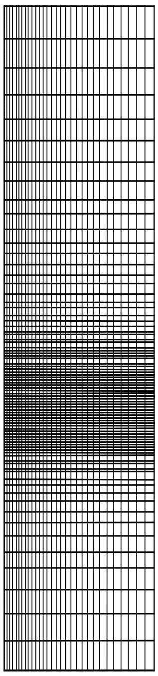

Figure 2: Type of grid used in the numerical solution.

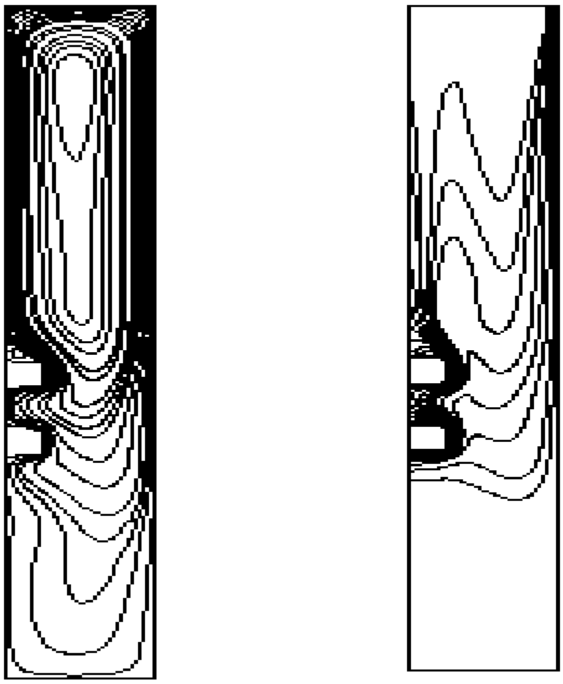

Figure 3: Streamlines and Isotherms for baseline case (case 2).

Figures 4 to 7 present quantitative results of the temperature along the wall where the sources are mounted (*Tp*), as a function of the modified Rayleigh number, of the power distribution, of the spacing between them and of the aspect ratio of the cavity.

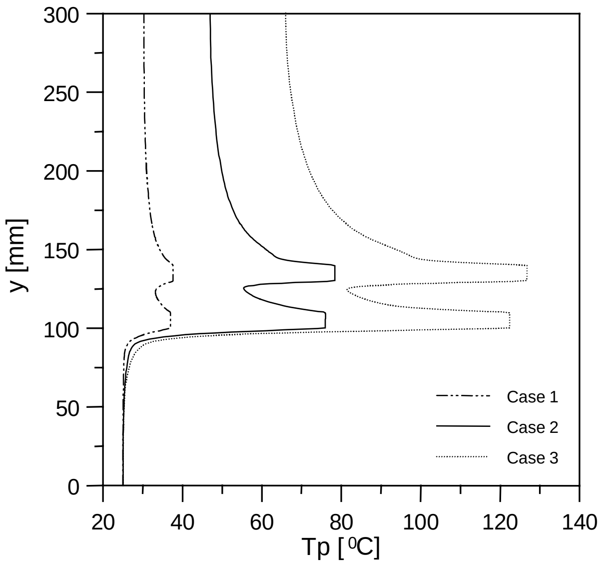

Figure 4: Wall temperature as a function of the modified Rayleigh number.

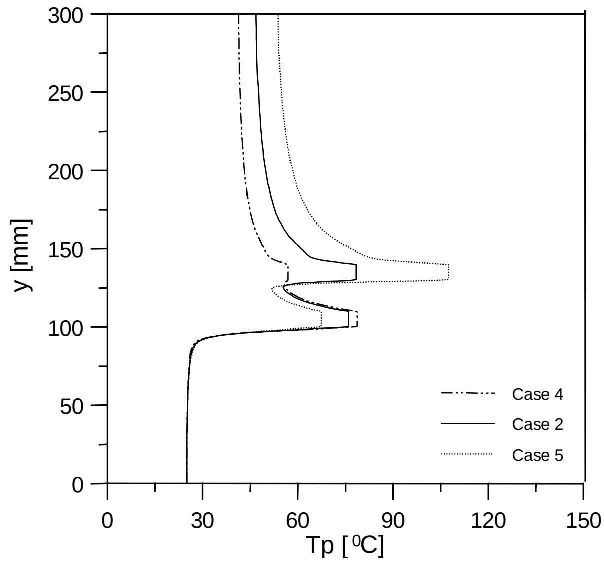

Figure 5: Wall temperature as a function of the power distribution in the sources.

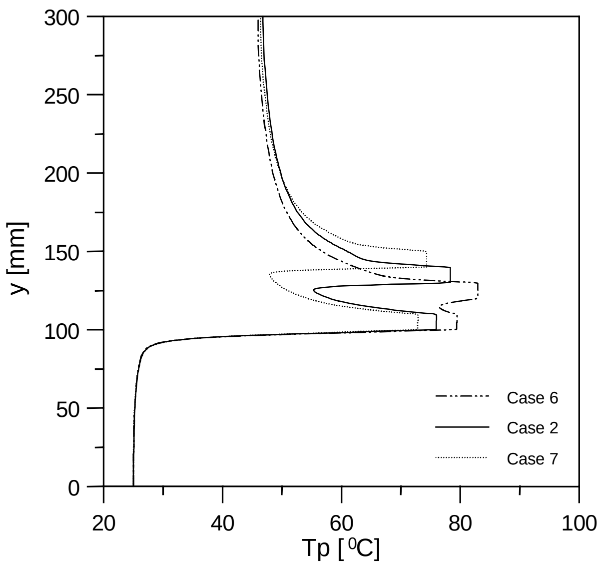

Figure 6: Wall temperature as a function of the spacing between the sources.

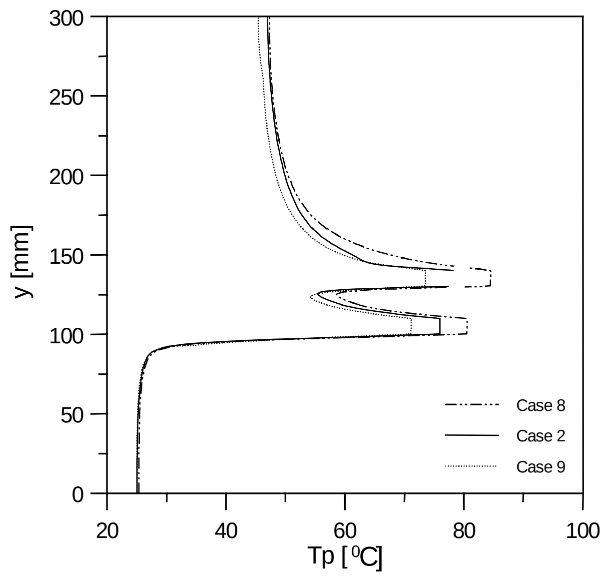

Figure 7: Wall temperature as a function of the aspect ratio of the cavity.

Table 2 shows the values of the average Nusselt number in the faces of the sources and the maximum temperature in the cavity wall where the sources are mounted (Tp_{max}), for each case considered.

Table 2: Average Nusselt number in the sources and maximum temperature in the cavity wall.

Cases	$\overline{Nu}_1$	$\overline{Nu}_2$	Tp_{max}[°C]
(1)	39.76	28.03	37.7
(2)	52.66	43.38	78.4
(3)	60.77	48.82	126.8
(4)	54.83	18.07	78.7
(5)	48.22	59.14	107.6
(6)	49.61	35.88	83.0
(7)	53.33	45.37	73.4
(8)	52.93	37.63	86.3
(9)	52.28	40.58	74.2

5. CONCLUSIONS

The streamlines show that the air circulates clockwise within the cavity, with a recirculation zone near the top of the cavity. The obstruction imposed by the protuberance of the sources, cause distortions in these lines in this region.

The isotherms exhibit a boundary layer behavior near the wall where the sources are mounted. As the value of the Rayleigh number is increased, the air penetrates more between the sources and the boundary layer thickness decreases.

The temperature of the vertical wall where the sources are mounted increase steeply in the vicinity of the sources. The levels of these temperatures are proportional to the Rayleigh number. The upper source is affected by the dissipation in the lower source. For the same heat dissipation in both sources, the temperature of the upper source is always higher. As the spacing between the sources is increased, the temperature levels are reduced. The temperature levels are reduced when the cavity aspect ratio is increased.

As for the average Nusselt number in the sources, it increases considerably with the Rayleigh number. For larger sources spacing, a smooth increase in the average Nusselt number in each source is observed. With respect to the cavity width, it may be noticed that when the aspect ratio is increased, the values of the average Nusselt number in the sources are also increased.

By analyzing the obtained results we may conclude that the more adequate configuration, exhibiting the lower temperature levels, and therefore lower failure rates, would be achieved with the utilization of lower power dissipations, locating the source with higher power dissipation in the upper position (downstream of the other source) and with the higher spacing between the sources and with the higher aspect ratio possible.

Eventually, the multigrid technique will be implemented and the variation of the properties with temperature will be considered to compare the numerical results with the results of experimental tests being conducted.

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