CONJUGATE NATURAL CONVECTION APPLIED TO THE ELECTRONIC COMPONENT COOLING

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Abstract. *The present study reports the conjugate laminar natural convection in a discretely heated cavity. The cavity has three protuberant discrete heat sources, equally space, mounted to one vertical wall. The opposite vertical wall and the horizontal faces are assumed to be isothermal and adiabatic, respectively. The conduction in the solid (electronic components) and the convection in the fluid region are solved using an only domain by a coupled way (conjugate problem). The partial differential equations system is solved applying the Galerkin finite element method with an unstructured mesh. To check the numerical results, the Nusselt number values for natural convection in a square cavity (with differentially heated walls) with benchmark solutions and presented a good agreement. The effect of the solid to fluid conductivity ratio on the velocity and temperature profiles was analyzed. To improve the electronic component cooling design, a proposed critical dimensionless temperature difference was calculated as a function of the heat source internal energy generation. Results showed that lower temperature values can be reached depending on the location of the heat source with higher heat dissipation rate. This is extremely important to guarantee that the electronic component operational temperature limits are not been exceed.*

Keywords*: Natural convection, conjugate problem, discrete heat source, finite element method.*

1. Introduction

Natural convection is an advantageous method to cool electronic equipment due to its passive character, that is, a self-controlled device. It allows the thermal control eliminating the fan or exhauster and provides a noise and vibration-free environment. Krauss and Bar-Cohen (1995) present an extensive study of both natural and forced convection flow through parallel-plate channels, fins conduction and radiation heat transfer applied to the design and optimization of cooling devices.

The decrease of the heat dissipation capacity due to the reduction in the heat exchange area represents a restriction in the evolution of the electronic components compactness. Most heat sinks are cooled by forced convection but the role of the natural convection is important due to the failure possibility in the commercial coolers (Lorenzetti, 1989).

The thermal design of these heat-sinks equipments must ensure that electronic components temperatures are maintained below a safe operational limit. Due to its inherent passive characteristic, the natural convection heat-sink hasn't an adaptive response to the equipment operation, resulting in a challenging design. Several studies have investigated the natural convection problem in enclosures and vertical plate arrays applied to the electronic equipment cooling. Ganzarolli and Milanez (1995) reported the performance of electronic components cooling packages with equidistant discrete heat sources. The cavity was heated from below and symmetrically cooled from the sides.

Heindel *et al.* (1996) studied the enhancement of natural convection heat transfer from an array of discrete heat sources. The authors obtained experimental data and numerical predictions of a cavity filled with a dielectric liquid (FC-77). Dense parallel plate fin arrays were considered for both vertical and horizontal cavity orientations and the finned surface were found to enhance the heat transfer by as much as a factor of 24 when compared with the unfinned cavity conditions.

Daloglu and Ayhan (1999) presented experimental results of free convection in a rectangular cross-sectional vertical channel which walls are maintained at uniform heat flux. Fins are periodically attached along the both channel plates. Their results showed that the Nusselt number values for finned channel are less than those obtained for smooth configuration for the studied Rayleigh number range.

Sezai and Mohamad (2000) performed a detailed numerical analysis of horizontal discrete heaters of different length/width ratios, flush-mounted with the bottom wall of a cavity. The enclosure was cooled from above and insulated from the bottom. Using the multigrid technique, it was found that the rate of heat transfer is not very sensitive to the vertical wall boundary conditions and the limit of the maximum Rayleigh number to obtain a convergent solution decreases as the aspect ratio of the source is increased.

Zamora and Hernández (2001) analyzed the influence of upstream conduction on the thermal optimization of the spacing of isothermal, natural convection-cooled vertical plate arrays. They obtained correlations for the average Nusselt number using an elliptic numerical model. Results showed that the upstream conduction effects were found to be more relevant for low Prandtl number values and tend to make the heat transfer rate less dependent on the selection of the plate spacing.

DeAndrade and Zaparoli (2003) reported the effect of temperature-dependent thermo-physical properties on the laminar natural convection in a discretely heated cavity. The effect of the variable properties was analyzed and showed that the temperature-dependent properties variations modify the temperature profiles and also affect the heat transfer rate results. To improve the heat sink thermal design, a proposed critical dimensionless temperature difference was calculated as a function of the cavity aspect ratio, showing that a vertical rectangular heat sink configuration provides a better electronic component cooling.

Da Silva *et al.* (2004) studied the optimal distribution of discrete heat sources cooled by natural laminar convection. The authors showed analytically that an optimal arrangement exists: it is not uniform and depend on the Rayleigh number. Besides, the optimal distribution of heat sources leads to a maximal global performance corresponding to a minimal global thermal resistance between the wall and the fluid.

Works previously cited analyze the laminar natural convection in enclosures or vertical channels neglecting the effect of the conduction heat transfer in the solid domains. The conjugate problem, however, involves the coupling of conduction in the solid and convection in the fluid region. One of the first investigations focused on the conjugate effects in free convection was Zinnes (1970). Their work shows that the degree of coupling between conduction heat transfer in a substrate and natural convection in the fluid region is greatly dependent of the substrate/fluid thermal conductivity ratio.

Heindel *et al*. (1995a, 1995b) studied the conjugate problem in a cavity with an array of discrete heat sources using two and three-dimensional models. Numerical results were compared with experimental data, using water and a dielectric liquid (FC-77) as the coolants, and showed that the 2-D model predicted very well general trends and flow patterns experimentally obtained. They concluded that the heat transfer is dominated by the conduction at small values of the Rayleigh number. As this dimensionless parameter increases, the flow pattern becomes more boundary layer-like along the vertical walls, with multiple fluid cells developing in the central fluid region. Besides, this core region becomes more thermally stratified and the substrate conduction mechanism also decreases.

The present study reports the electronic component cooling process by means of air natural convection. The convection in the fluid region and conduction heat transfer in the solid (substrate and heat sources) are studied by a coupled way and solved simultaneously. A range of Rayleigh number values are numerically simulated varying the magnitude of the heat sources energy internal generation. The substrate thermal conductivity (k_s) is maintained constant and the influence of the heat source to fluid thermal conductivity ratio (k_h / k_f) on the airflow patterns and on the heat transfer rate is also studied. Results showed that the critical dimensionless temperature in the electronic components is weakly dependent of the k_s / k_f ratio with a sensible variation only for very low range of this dimensionless parameter.

2. Mathematical formulation

Figure 1 displays a two-dimensional square cavity of height H with three heat sources, equally spaced, mounted to the left vertical wall.

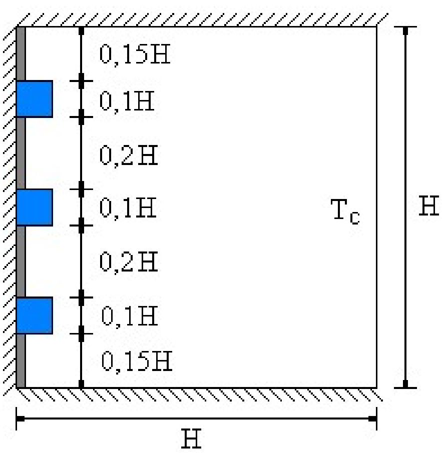


Figure 1 – Problem geometry

The other vertical wall is maintained at a constant temperature T_C and the remaining walls are assumed to be adiabatic. A substrate material with low constant thermal conductivity fills the space among the discrete heat sources. The boundary conditions are:

i) at $x = 0$ and $0 \le y \le H$: $u = 0$, $v = 0$ and $\frac{\partial T}{\partial x} = 0$;

ii) at $x = H$ and $0 \le y \le H$: $u = 0$, $v = 0$ and $T = T_C$;

iii) at $y = 0$ and $0 \le x \le H$: $u = 0$, $v = 0$ and $\frac{\partial T}{\partial y} = 0$;

iv) at $y = H$ and $0 \le x \le H$: $u = 0$, $v = 0$ and $\frac{\partial T}{\partial y} = 0$.

Thermal conditions at the heater/fluid and substrate/fluid interfaces are not known a priori, but they are calculated intrinsically through the solution process.

The governing equations (mass, momentum and energy) for steady-state laminar flow of a Newtonian fluid with constant properties are:

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

$$\rho_{f,s,sub}\left(u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial x} \right) = -\frac{\partial P}{\partial x} + (\mu_{f,s,sub})\nabla^2 u \tag{2}$$

$$\rho_{f,s,sub}\left(u\frac{\partial v}{\partial x} + v\frac{\partial v}{\partial x} \right) = -\frac{\partial P}{\partial y} + (\mu_{f,s,sub})\nabla^2 v + \rho_b g \ , \tag{3}$$

$$\rho_{f,s,sub}(Cp_{f,s,sub})\left(u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial x} \right) = (k_{f,s,sub})\nabla^2 T - \dot{q} \tag{4}$$

with:

x	horizontal coordinate	T	temperature
y	vertical coordinate	C_P	constant pressure specific heat
u	x-axis velocity component	k	thermal conductivity
v	y-axis velocity component	ρ	density
P	thermodynamic pressure	ρ_b	fluid density in the buoyancy term
g	gravity acceleration	$\dot{q}$	heat source internal energy generation
∇^2	Laplacian operator	μ	viscosity

At the present study the conjugate problem is solved by a coupled way, resulting that the solution for the differential equations system in the solid and fluid domains is obtained simultaneously. The subscripts "f", "s" and "sub" in Eq. (1) to (4) refers to the fluid, heat source and substrate material properties, respectively. The natural convection problem is solved using the Boussinesq approximation. Therefore, the fluid density is constant, except in the buoyancy-force term of the y-momentum equation with the following linear temperature dependence, Eq. (3):

$$\rho_b = \rho_c\left[1 - \beta(T - T_c)\right] \tag{5}$$

where ρ_c is the density value evaluated at the cold wall reference temperature, T_C and β is the coefficient of volumetric expansion calculated as:

$$\beta = -\left(\frac{1}{\rho_f}\right)\left(\frac{\partial \rho_f}{\partial T}\right) \tag{6}$$

Results will be presented as a dimensionless temperature defined by the following expression:

$$\phi = k_f \frac{T - T_c}{s^2 \sum \dot{q}} \qquad (7)$$

where s is the heat source height.

Each electronic component dissipates a heat amount that should be transferred to the air. To guarantee a stable and reliable operation, the maximum electronic component temperature must be maintained below a manufacturer required critical temperature value ($T_{\max}$). To compare different air-cooled heat sinks devices, a critical dimensionless parameter, ϕ_c, is defined as:

$$\phi_c = k_f \frac{T_{\max} - T_c}{s^2 \sum \dot{q}} \qquad (8)$$

For the coolant fluid, it can be defined the following dimensionless parameters:

$$r = \frac{s}{H} \text{ = geometric aspect ratio} \qquad (9)$$

$$\Pr = \frac{(\mu_f)(Cp_f)}{(k_f)} \text{ = Prandtl number} \qquad (10)$$

$$Gr = \left[\frac{g \beta \dot{q} H^5}{k_f (\mu_f / \rho_f)^2} \right] \text{ = Grashof number} \qquad (11)$$

$$Ra = Gr \Pr \text{ = Rayleigh number} \qquad (12)$$

At the present study the numerical simulations were carried out for a constant Prandtl number = 1.0, constant geometric aspect ratio $r = 0.1$ and the substrate thermal conductivity was maintained constant with $k_{sub} / k_f = 2$.

3. Solution Methodology

At the present study the airflow patterns and temperature fields inside the discretely heated cavity are obtained by a numerical solution. Firstly, a pressure Poisson equation was derived combining Eq. (2) with Eq. (3) and imposing the mass conservation restriction, Eq. (1). After, the resultant partial differential equations system was discretized applying the Galerkin finite elements technique. An unstructured mesh with triangular elements of six nodes and second-degree interpolation polynomials was employed. The obtained algebraic equations system was solved by an iterative procedure in a coupled (no-segregated) way using the conjugate gradient scheme.

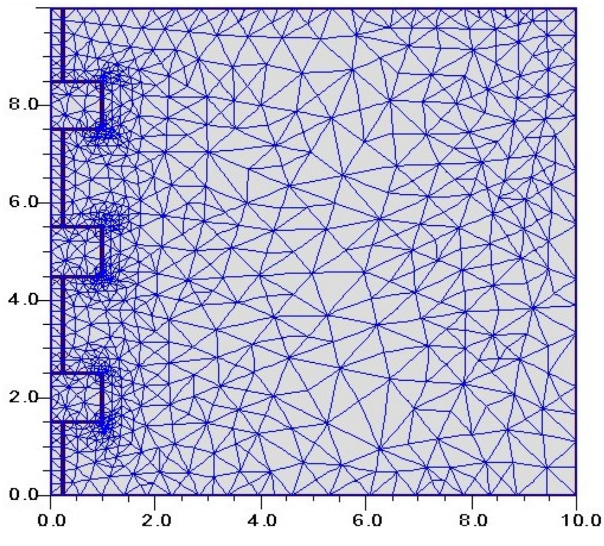


Figure 2 – Example of the computational mesh utilized ($Ra = 3.6 \cdot 10^5$ and $k_s / k_f = 500$).

An adaptive procedure was used with successive mesh refinement in the more intense dependent variable gradients, mainly close to the discrete heat sources as shown in Fig. (2), for a computational mesh in the solution process at $Ra = 3.6 \cdot 10^5$ and $k_s / k_f = 500$ (a typical metallic solid/air thermal conductivity ratio).

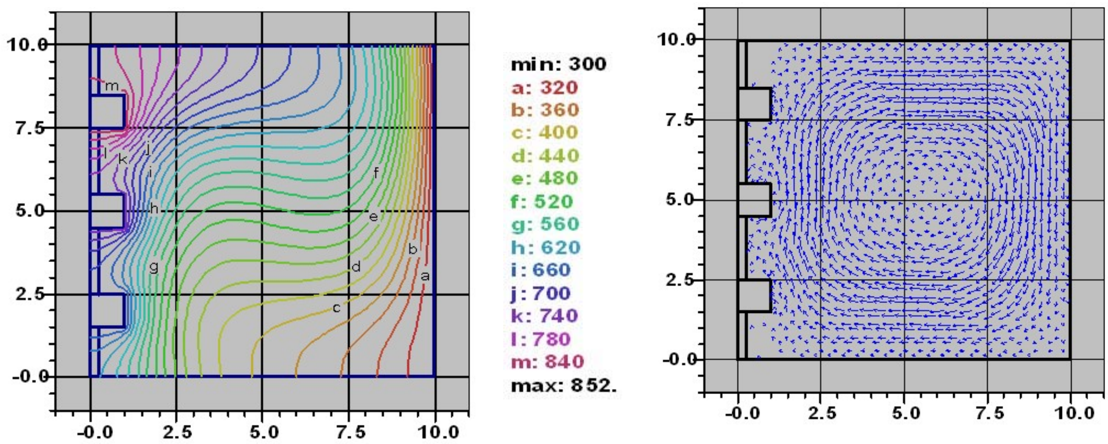
It can be seen a more intense mesh refinement around the upper and lower three heat sources corners. In the central region of the cavity, the mesh is coarse with a lower nodes number, as a result of the weak flow pattern verified in Fig. (3b).

To validate the numerical tool, the natural convection differentially heated square cavity Nusselt number results were obtained with the Boussinesq approximation. These results were presented in DeAndrade and Zaparoli (2003) and showed a good agreement with the literature correlations.

4. Results

Numerical simulations were performed for the Rayleigh number in the range $1 \cdot 10^3 \le Ra \le 1 \cdot 10^5$ and the heat source to fluid thermal conductivity ratio in the range $1 \le k_s / k_f \le 1 \cdot 10^4$. The internal energy generation value is equal for all heat sources mounted on the left vertical cavity wall, varying in the range $1 \le \dot{q} \le 500$.

Figure (3a) presents the isothermals inside the discretely heated cavity for $Ra = 3.6 \cdot 10^5$ and $k_s / k_f = 500$. The temperature has a boundary layer-like behavior close to the right cold vertical wall and near the left solid/fluid interfaces. The maximum temperature values occur close to the heat source located near the left upper corner of the cavity, as shown in Fig. (3a). In the substrate domain, the temperature profiles present a stratified character which approximates a one-dimensional conduction heat transfer. The conjugate approach employed in the numerical solution allowed capturing these characteristics of the coupled conduction/convection mechanism. In the heat source region, the isothermals are practically constant due to the high thermal conductivity of this material. Figure (3b) illustrates the velocity vectors with a weak flow near the superior and inferior extremes of the wall which the heat sources are mounted and also between each heat source pair.



(a) (b)

Figure 3 – Example of isothermals (a) and velocity vectors (b) for $Ra = 3.6 \cdot 10^5$ and $k_s / k_f = 500$.

The dimensionless temperature profile (ϕ), at $x = 0.1 \cdot H$ (heat sources heads) and along the y-extension, for three Ra numbers values and $k_s / k_f = 500$, is shown in Fig. (4).

It is verified that the dimensionless temperature profiles exhibit a similar behavior: constant values at the solid domain (heat source extension) and an enhancement of the temperature as the fluid is been heated due to the heat dissipation from bottom to top of the cavity.

Figure (5) shows the dimensionless critical temperature ϕ_c as a function of the Ra number. As the Rayleigh number increases, the ϕ_c values decreases due to higher values of the internal energy generation (denominator of the Eq. (8)). For the electronic component cooling process, however, the maximum ϕ_c acceptable value is determined by the component thermal characteristics. For example, the $T_{\max}$ value is an operational restriction of the electronic component known a priori. Using the results presented in Fig. (5), the designer can obtain the Ra number value that corresponds to a suitable heat-sink device design.

The solid (heat sources) to fluid thermal conductivity ratio does not affect significantly the heat-sink performance indicated by ϕ_c, as indicated in Fig. 6. Only for very low k_s / k_f values, the critical dimensionless temperature increases due to the non-isothermal temperature distribution inside the heat source domains.

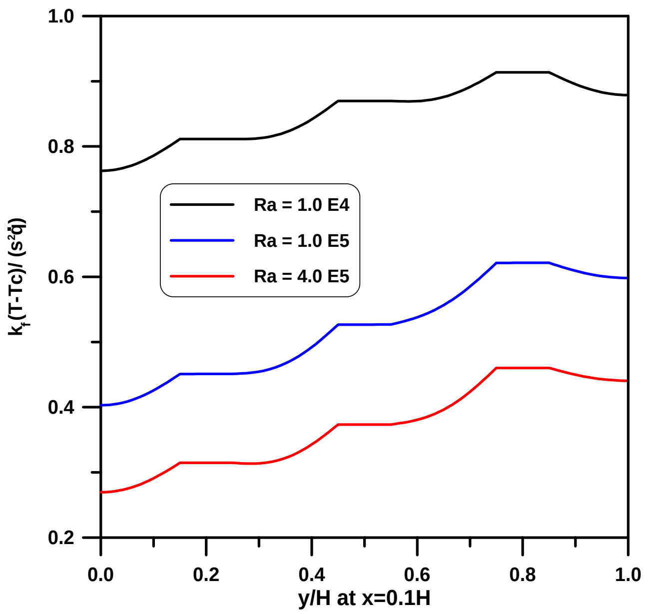


Figure 4 – Dimensionless temperature profile along the y-vertical line touching the heat sources heads.

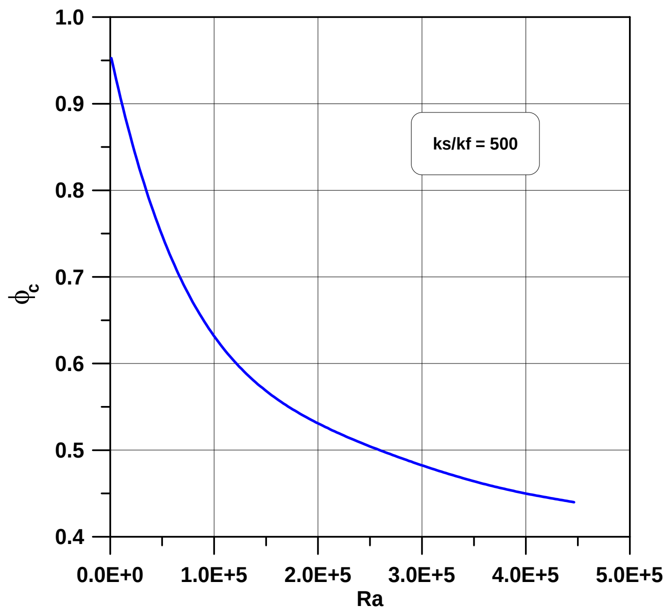


Figure 5 – Dimensionless temperature profile as a function of Rayleigh number.

In all previous simulations, the internal energy generation $\dot{q}$ values were considered equal for the three heat sources. Below, it's presented a study for different $\dot{q}$ values for each heat source, where $\dot{q}_1$ and $\dot{q}_3$ indicates the power of the heat source located at the lower and upper corner cavity, respectively. Table (1) presents results for six different arrays.

Table 1 Heat sources dissipations and ϕ_c values for six different arrays.

$\dot{q}_1$ [W/m^3]	$\dot{q}_2$ [W/m^3]	$\dot{q}_3$ [W/m^3]	ϕ_c
500	250	175	0.04282
500	175	250	0.04534
175	500	250	0.04628
250	500	175	0.04893
250	175	500	0.05391
175	250	500	0.06165

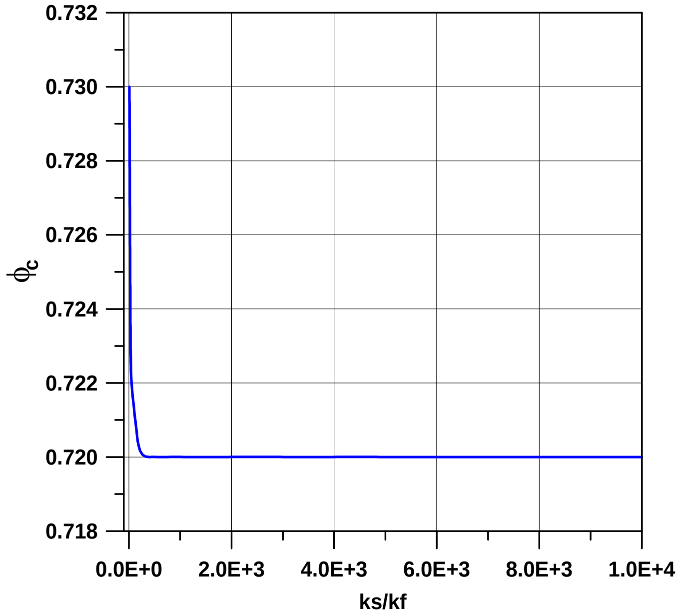


Figure 6 – Dimensionless temperature profile as a function of the solid to fluid thermal conductivity ratio (k_s / k_f).

For theses cases simulated, the heat source sequence for the best design (minimum critical dimensionless temperature) was obtained with the stronger heat source located at the cavity bottom.

5. Conclusions

At the present work, a numerical solution for the conjugated natural convection in a cavity with three protuberant heat sources was obtained using the Galerkin finite element method. The relevance of this study is mainly in the case of failure of the forced convection component (cooler, e.g.). In this case, the natural convection is the only way to guarantee the heat sink cooling. As it was expected, the better array configuration is obtained when the higher power heat source is located in the superior position, avoiding the heat transport upward.

Results showed that the heat source/fluid thermal conductivity ratio does not affect significantly the heat-sink performance when the internal energy generation was constant for the three heat sources. For the cases where the heat sources dissipation was unequal, the critical operational dimensionless temperature of the electronic component was obtained when the heat source power increases downward.

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7. Responsibility notice

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