Thermoregulation of electronics inside diode enclosures. Viscous shear stress in 2D natural convection generated by isothermal active walls

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Abstract: - Correct operation of electronic assemblies is subject to their regulation in the temperature range recommended by manufacturers. It is therefore necessary to control the heat exchange phenomena that affect them. When thermoregulation of these assemblies is based on thermocouples, the effects of fluid flow on the junctions of these sensors must be considered, particularly the thermal and velocity gradients. This prevents measurement errors that can be detrimental to the proper functioning of the equipment. The main objective of this study concerns the examination of the viscous stresses that occur by natural convection in parallelogram-shaped cavities containing electronic equipments. These enclosures, called diode cavities in the convective heat transfer sense, lead to very different flows depending on the geometry. Many geometrical configurations are treated while varying the inclination angle of the top and bottom passive adiabatic walls. A detailed study of the boundary layer is used to find the distribution of viscous shear stresses and the heat exchanged by natural convection through the Nusselt number. The numerical approach is made using the finite volume method. The results are confirmed by previous numerical and experimental works, and allow to better control thermoregulation of electronic assemblies by means of thermocouples.

Key-Words: - Electronic equipments, Natural convection, Thermal viscous shear stress, Isothermal heat sources, Thermal engineering applications, Energy saving, Diode cavity, Finite volume method

Nomenclature

\( a \) \quad \text{thermal diffusivity of the air (m}^2\text{s}^{-1})
\( C \) \quad \text{specific heat (J kg}^{-1}\text{K}^{-1})
\( g \) \quad \text{acceleration of the gravity (m s}^{-2})
\( h \) \quad \text{local convection coefficient on the hot wall for a given angle} \ \alpha \ (W \text{ m}^{-2} \text{K}^{-1})
\( h \| \alpha \) \quad \text{mean convection coefficient on the hot wall for a given angle} \ \alpha \ (W \text{ m}^{-2} \text{K}^{-1})
\( H \) \quad \text{height of the cavity; distance between the hot and cold walls (m)}
\( n^* \) \quad \text{dimensionless local normal to the top and bottom passive walls (-)}
\( \frac{Nu}{\alpha} \) \quad \text{local Nusselt Number on the hot wall for a given angle} \ \alpha \ (-)
\( \overline{Nu}_{\alpha} \) \quad \text{mean Nusselt Number on the hot wall for a given angle} \ \alpha \ (-)
\( p \) \quad \text{pressure (Pa)}
\( p^* \) \quad \text{dimensionless pressure (-)}
Greek symbols

\[ \begin{align*}
\alpha & \quad \text{inclination angle of the cavity (°)} \\
\beta & \quad \text{expansion coefficient of the air (K}^{-1}) \\
\lambda & \quad \text{thermal conductivity of the air (Wm}^{-1}\text{K}^{-1}) \\
\mu & \quad \text{dynamic viscosity of the air (Pa s)} \\
\rho & \quad \text{density of the air (kg m}^{-3}) \\
\tau & \quad \text{viscous shear stress at the hot wall (Pa)} \\
\tau^* & \quad \text{dimensionless shear stress at the hot wall (-)} \\
\overline{\tau} & \quad \text{mean dimensionless shear stress at the hot wall (-)}
\end{align*} \]

1 Introduction

Modern electronic assemblies dissipate greater and greater heat flux densities. They are installed inside of enclosures of specific form according to the application, and the volume is often small. This complicates the work of the engineer who must consider all the aspects of the system design. Natural convection is often preferred in electronic engineering. This heat transfer phenomenon, which occurs spontaneously, saves energy and eliminates mechanical devices used as temperature control systems. Elimination of these mechanisms diminishes the risks of failure increasing the reliability of the systems. The absence of rotating elements such as fans or pumps eliminates also the associated electromagnetic pollution, a major source of malfunction. The correct operation of electronic assemblies is always linked to a good temperature control according to the recommendations of the manufacturers. A great number of works have been published on thermoregulation of electronics installed in most industrial devices. In the electronics industry, the design is based on dynamic and thermal simulation. Today there are several codes more or less efficient. Joshi et al [1] remark the challenges facing the thermal community for the design and optimization of electronic packages. Measurements on a prototype are nevertheless inevitable before manufacturing. It is also an opportunity to refine and readjust the mathematical models used for thermal simulation. When thermocouples are used for thermoregulation of electronics, it is necessary to take into account the effects of the fluid flow on the junctions of these sensors. Velocity gradients and temperatures generated by the flow in which these junctions are immersed may lead to wrong temperature indications and then to an improper operation of the control system. This topic has been examined by several researchers. Zhu et al [2] studied the effects of the flow on the sensors for the development of a micromachined thermal gas inertial sensor. The study of thermal and dynamic boundary layers near the active hot wall is necessary for an efficient design of equipments. Surveys dealing with natural convection in cavities are numerous, showing the great interest of its applications. Several geometries are treated. Cubical and parallelepipedic ones are the most studied given their simple form and easy implementation. Many studies dedicated to them are available in the literature, as [3,4,5,6]. The present work concerns a particular air-filled configuration. Both hot and cold active walls of the cavity are opposite and remain vertical. They have the same dimensions and are separated by a horizontal distance equal to their height. They are maintained isothermal with an imposed temperature difference corresponding to the real operation of the equipment. The top and bottom walls of the cavity are inclined with respect to the horizontal, so right sections have a parallelogrammic shape. When the angle is positive, the hot wall is lower than the cold one. The air flow is favored in comparison to the case of the right cavity (zero angle). The cavity is then qualified as "conductive" in the convective heat transfer sense of the word. When the channel is narrowed too much, what happens from a certain angle (around 30°), the flow is reduced and convection decreases. When the angle is negative, the hot wall is higher than the cold one. The flow is then hindered by the geometry of the enclosure, causing a significant decrease in the convective exchange. In this sense, the cavity is considered "insulating". Being conductive in one direction and insulating in the other one, the enclosure is called "diode cavity" in the heat transfer convective sense of the word. Values of inclination angles considered in the present work are 0° (square cavity), ±30° and ±60°. Naylor et al [7] studied 2D air-filled parallelogram-shape enclosures with various aspect ratios and large Rayleigh range. Baïri et al [8] applied it to the thermoregulation of on-board equipments and Costa [9] for a window,
presenting many interesting data. Hyun et al [10] consider also high Rayleigh numbers, several inclination angles, many Prandtl numbers and several aspect ratios. The works of Maekawa et al [11] and Seki et al [12] provide important data concerning the Nusselt number. Radiative exchanges have been examined by Sieres et al [13] and Baïri et al [14]. García de María et al. [15] propose correlations of the type Nu-Ra-α for a hot wall consisting on discrete isothermal bands separated by adiabatic ones. Correlations concerning the hot wall maintained isothermal treated in this work are proposed by Baïri et al. [16] for inclination angles varying between –60° to +60° step 15° and a large range of Ra numbers. These previous studies [11,12,15,16] are used to compare some results of the present work. From another point of view, the study of boundary layers has been done by Ostrach [17] and Gebhart [18] through the well known similarity method taking into account the viscous dissipation. Nadim et al. [19] used the homotopy analysis method to solve the nonlinear differential equations of the free convection boundary layer on a vertical plate. The survey of Sillapää et al. [20] based on the theory of similarity variables completed by measurements, examine the effects of the variable shear stress on cylindrical surfaces. All this studies confirm the importance of taking into account the local phenomena for a correct disposition of the sensors used for thermal regulation of electronic assemblies.

2 Governing equations and calculation procedure
The parallelogram-shape air-filled cavity is represented in Fig. 1.

![Diode cavity](image)

Fig. 1: Diode cavity (a) the channel (b) the 2D considered model

Both hot and cold active walls (x=0; x=H) remain vertical, maintained isothermal at $T_h$ and $T_c$ respectively. The channel of the cavity is thermally insulated. Flow inside the cavity is examined on its median plane, considered as 2D.

Governing equations of this problem are

* continuity equation

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

* momentum equations

$$
\begin{align*}
\frac{\partial u^*}{\partial x^*} + \frac{\partial v^*}{\partial y^*} &= Pr \nabla^2 u^* \\
\frac{\partial u^*}{\partial x^*} + \frac{\partial v^*}{\partial y^*} &= Pr \nabla^2 v^* + Ra Pr T^* - \frac{H g}{\alpha^2} \left( 1 + \frac{\partial^2 T^*}{\partial y^*} \right)
\end{align*}
$$

* Energy equation

$$u^* \frac{\partial T^*}{\partial x^*} + v^* \frac{\partial T^*}{\partial y^*} = \nabla^2 T^* + \frac{g \beta H}{C} \frac{T^*}{Ra}$$

In these equations, $x^*, y^*, u^*, v^*, p^*, T^*$ are the dimensionless Cartesian coordinates, velocity components, pressure and temperature respectively.

$$
\begin{align*}
x^* &= \frac{x}{H} \\
y^* &= \frac{y}{H} \\
u^* &= \frac{u}{a/H} \\
v^* &= \frac{v}{a/H}
\end{align*}
$$

$$p^* = \frac{p}{\rho g H} ; \frac{T^* - T_c}{T_h - T_c} = \frac{T - T_c}{T_h - T_c}$$

The Rayleigh and Prandtl numbers are defined as

$$Ra = \frac{g \beta (T_h - T_c) H^3 \rho}{\mu a} ; Pr = \frac{\mu C}{\lambda}$$

The treated temperatures $T_h$ and $T_c$, associated to the height $H$ lead to a large range of Rayleigh numbers $1 \times 10^3 \leq Ra \leq 5 \times 10^5$, corresponding to real applications. The second term on the right-hand side of Eq (3) is the viscous dissipation term and represents the heat generated by fluid friction.

The boundary conditions of the considered problem are

* adherence of the fluid to the walls of the cavity,

$$
\begin{align*}
x^* &= 0 ; 0 \leq y^* \leq 1 \\
x^* &= 1 ; \tan \alpha y^* - \tan \alpha \\
0 \leq x^* \leq 1 ; y^* &= x^* \tan \alpha \\
0 \leq x^* \leq 1 ; y^* &= 1 + x^* \tan \alpha
\end{align*}
$$

$$u^* = v^* = 0$$

* top and bottom passive walls are adiabatic
\[ \frac{\partial T}{\partial n^*} \mid_{[0 \leq x' \leq 1 ; y'=x' \tan \alpha]} = 0 \]  
\[ \text{with } n^* = n / H^* \]

\[ T^*_h = \frac{T_h}{T^*_0} \]

\[ T^*_c = \frac{T_c}{T^*_0} \]

\[ h_{\alpha} = \frac{-\lambda}{T_h - T_c} \left( \frac{\partial T}{\partial x} \right)_{x=0, \alpha} \]

**3 Results**

Thermal and dynamical aspects of the problem have been determined for inclination angles \( \alpha = 0, \pm 30^\circ \) and \( \pm 60^\circ \), and a wide range of \( Ra \) numbers \( 1 \times 10^3 \leq Ra \leq 5 \times 10^9 \).

Fig. 2 represents \( T^* \) and the ratio \( V^* = v / v_{max} \) at \( Ra = 4.12 \times 10^7 \). The diode effect is clearly shown in this figure: convection is present for “conducting” configurations while thermal stratification is observed for the “insulating” ones. Dynamic and thermal aspects are also consistent. This is confirmed from the distributions of the Nusselt numbers on the active wall, represented in Fig. 3 for the same values of \( \alpha \) and \( Ra \) number. Values concerning the insulating and conducting cavities are represented separately, and compared with those of the right cavity (\( \alpha = 0 \)).
Fig. 2: Fields $T^*$ and $V^* = v/v_{max}$ for $\alpha=0, \pm 30^\circ$ and $\pm 60^\circ$, at $Ra=4.12 \times 10^7$.

Nu always decreases with decreasing angles for the insulating enclosures, which is not the case for the conductive ones. Average values concerning the entire wall $\overline{Nu}$ are represented by vertical lines on the horizontal axis of Fig 3. These results are consistent with the correlations $Nu-Ra-\alpha$ proposed in [16]. The maximum deviation observed is 1.5% for $\alpha=-60^\circ$.

Mean values for the entire wall $\overline{\tau}$ are represented by vertical lines on the horizontal axis. Local values of the shear stress $\tau^*$ are systematically lower in the case of insulating cavities when compared with the right ones. Shear stress regularly decreases with increasing $y^*$ values and becomes zero at the top of cavity, as expected. Values are however relatively large at the bottom of the cavity. The velocity gradient causes greater shear stresses in the upper half of the hot wall, and smaller ones in the lower part than in the case of insulating cavities.

4 Conclusion

When thermocouples are used for thermoregulation of electronic assemblies, it is necessary to take into account the effects of the fluid flow on their junctions. Velocity gradients and temperature generated by the flow in which they are immersed may lead to wrong temperature indications and then to an improper operation of the equipment. These effects are evident in the case of the diode cavities considered in this work. Local distribution of viscous shear stress clearly shows the influence of the angle of inclination. The graphs provide information on the proper locations where the thermocouples must be placed, according to the angle and Ra number combination, when used for thermoregulation of electronic equipment (here represented by the hot wall of the cavity).

References:


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