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Forced Convection Cooling of a 3D Protruding Heaters Array with Laminar Flow Of Dielectric Fluid

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Abstract — ***In the present work a numerical analysis was performed to investigate the forced convection heat transfer with laminar flow of the dielectric fluid Novec™ 7500 from a 3D protruding heaters array mounted in cross-stream direction on an adiabatic substrate of a horizontal rectangular channel using the ANSYS/Fluent™ 15.0 software. A uniform heat generation rate was assumed in the 3D protruding heaters and the cooling was performed by means of a forced fluid flow with constant properties in the laminar regime under steady state conditions. At the channel inlet, the flow velocity and temperature profiles were assumed uniform. The governing equations and their boundary conditions were numerically solved in a single domain through a coupled procedure using the Control Volumes Method. The SIMPLE algorithm was used to solve the pressure-velocity couple. The discretization of the convective-diffusive terms was performed using the Second-order upwind scheme. Due to the non-linearity of the momentum equation, the correction of the velocity components and the pressure were under-relaxed to prevent instability and divergence. After a computational mesh independence analysis, the numerical simulations were obtained and displayed as a 3D non-uniform mesh with 212,670 control volumes. This computational mesh was more concentrated near the solid-fluid interface regions due to the larger primitive variable gradients in these regions. To obtain the numerical results, typical properties values and geometry dimensions found in forced convection cooling with dielectric fluid of electronics components mounted in a printed circuit board were used. An investigation was done on the effects of the Reynolds numbers ranging from 100 to 300. The thermal parameters of interest, such as, temperature distribution, local and average adiabatic Nusselt numbers, local and average heat transfer coefficients, convective and overall thermal conductance, were found and compared, when possible, with the available results in the literature for the air as the cooling fluid.***

Index Terms—**Array of 3D Protruding Heaters, Dielectric Fluid, Forced Convection, Laminar Flow,**

I. INTRODUCTION

In the last decade, academic researches and scientific-technological efforts were developed in order to enhance the cooling technologies of electronic equipment because, with the innovation of the modern electronic technology, it became faster, smaller and incorporated more functions, resulting in a significant unavoidable increase in the volumetric heat generation rate. That is the case of smart phones, notebooks, tablets and computers [1]. The failure factor of the electronic devices in general increase almost exponentially with the work temperature that should not exceed a value between 85°C and 100°C [2]. The possible causes of the failures are the diffusion of the semiconductor material, the chemical reactions, the movement of the glued materials and the thermal tensions [3]. In special applications, e.g., supercomputers where the heat generation is excessive and the space used for heat transfer is limited, the use of non-conventional and high cost cooling techniques is required. Dielectric fluids are utilized for the proper thermal control of the electronic packaging in question. In a dielectric fluid cooling system, one problematic factor that causes concern is the maintenance, because of the importance of the fluid's discard and the risk of intoxication as a result of handling it. Therefore, the selection of a heat transfer fluid for semiconductor processing equipment and electronics cannot be treated with minor importance anymore, because environmental problems became a critical factor in the decisions of manufacture operations and project of computers. There is the need for high performance and long term solutions, aiming for a low maintenance necessity and this way causing a smaller environment impact [4]. In the present work, problems motivated by the Level 2 of electronic packaging, associated with the thermal control of an array of 3D protruding heaters mounted on a printed circuit board (PCB) were considered, as shown in Fig. 1 [5]. A dielectric fluid was considered as the cooling fluid. The available space for the heaters can be limited and the cooling process must be done through forced convection with moderate velocities due to operational limitations and noise reduction. Under such conditions, there may not be enough space to work with heat sinks in these concentrate heat dissipation components. These components can be simulated by 3D protruding blocks mounted on a substrate [6].

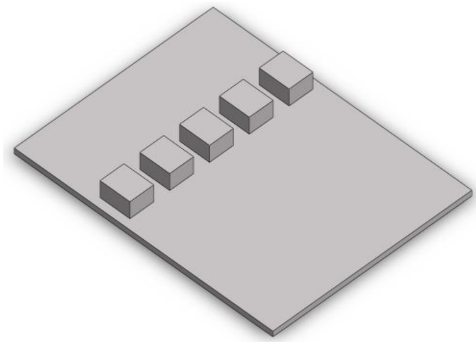

Fig. 1. An array of 3D protruding heaters mounted on a PCB.

The dielectric fluid selected to perform this work was the *Novec*™ 7500 that is utilized for thermal tests and immersion

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cooling of electronics, sold by the brand 3M™ *Novec Engineered Fluids*. This fluid was chosen due to the adequacy with the temperature range used and the environmentally friendly properties, assuming, nowadays, the position of one of the dielectric fluids that causes the least environmental impact. The fluids *Novec*™ are a group of materials with low Global Warming Potential (GWP) and have excellent properties for heat transfer applications, such as, dielectric properties, wide range of boiling points and good materials compatibility, in addition to demand little maintenance and to offer safe performance. They have high resistivity and will not damage electronic equipment or integrated circuits in the event of a leak or other failure. Further information about the *Novec*™ 7500 is presented in the manufacturer's catalog [7].

II. LITERATURE REVIEW

Nakamura *et al.* [8] performed experimental studies to investigate the airflow and the heat transfer around a 3D protruding cubic heater mounted on a substrate. The different temperatures at the heater surface and at the adiabatic substrate were measured under the constant heat flux condition. The behavior of the flow around the heater surface and the substrate was presented qualitatively. The pressure distributions and the local *Nusselt* number in the region close to the cubic heater and to the adiabatic substrate were presented for different *Reynolds* numbers. The characteristics of the heat transfer at the heater and at the substrate were correlated with the characteristics of the flow around them. Within the investigation range, a correlation for the average *Nusselt* number at the 3D protruding heater was expressed by $Nu_m = 0.137\ Re^{0.68}$. Nakajima *et al.* [9] presented numerical results of the laminar flow and the heat transfer of rectangular 3D protruding heaters mounted on the surface of a channel. They studied the case of three rows of 3D protruding heaters. A comparison between the numerical and the experimental results of the laminar flow streamlines was shown. The numerical investigation was executed for *Reynolds* numbers that ranged from 100 to 500, considering the *Prandtl* number equal to 0.7. The main characteristics of the flow around the 3D protruding heaters were the formation of horseshoe vortices and the recirculation regions. The local friction coefficient distributions at the bottom plate upstream the heaters and at the heater walls were presented. The temperature distribution at the heater surfaces was also presented. The heat transfer coefficient varied noticeably at the different surfaces of the heaters and with the change of *Re*. The average *Nusselt* increased with the increase in the *Reynolds* number. Yaghoubi & Velayati [10] numerically studied the heat transfer and the developing flow around a row of cubes in the transversal direction to the airflow, representing 3D protruding heaters mounted on a plate (configuration similar to the Fig. 1). The main characteristics of the flow around the heaters were presented. The velocity profiles and pressure distribution were shown upstream and downstream the cubic heater. The friction coefficient behavior was presented in function of the *Reynolds* number for different heights of the parallel plate channel. The average friction coefficient decreased with *Re*. The numerical results found for the average *Nusselt* number in a 3D protruding heater were compared with the experimental results of [8] and [11]. Nishida & Alves [12] performed a numerical analysis of the forced convection heat transfer of a row of 3D protruding heaters mounted on the bottom wall of horizontal rectangular channel utilizing the air as the work fluid. An investigation was done on the effects of the *Reynolds* numbers ranging from 100 to 300. The behavior of the laminar airflow around the protruding heaters was showed through the streamlines. The temperature distribution, local and average adiabatic *Nusselt* numbers, local and average heat transfer coefficients, convective and overall thermal conductance, were shown. The average adiabatic *Nusselt* number was correlated with deviations not greater than 0.5% through $Nu_{ad} = 1.247Re^{0.426}$. Other works available in the consulted literature that contributed with the air forced convection heat transfer from protruding heater(s) were [13]-[41].

III. MODEL DESCRIPTION

The basic configuration representing the treated problem for one of the 3D protruding heaters is indicated in Fig. 2. In this case, the channel has a height, H, length, L, and width, W. The substrate has the same length and width as the channel with a thickness, t, and thermal conductivity k_s. The heater has a length, L_h, height, H_h, width, W_h and it is located at a distance, L_u, from the channel entrance. The space between the heaters is $2W_s$. A uniform heat generation rate was assumed for each of the protruding heaters and the cooling process occurred through a forced laminar flow with constant properties under steady state conditions. In the channel inlet, the velocity profile (u_0) and the temperature profile (T_0) of the flow were considered uniform. Both top and bottom channel surfaces were adiabatic.

A. *Problem Formulation*

The mathematical model of the present problem was performed for a single domain: the solid regions (protruding heater and substrate) and the fluid flow in the channel. Due to the problem symmetries, the conservation equations were formulated for the domain with length, L, width, $W/2$ and height, $(H + t)$. The governing equations of the considered domain cover the principles of mass, *momentum* and energy conservation, Esq. (1), (2) and (3), respectively, under steady state conditions, constant properties and negligible viscous dissipation. The occasional effects of the natural convection, radiation, oscillation in the flow are not being considered in this modeling, a typical procedure adopted in similar problems, e.g., [37], [42]-[44].

- Mass Conservation (Continuity Equation)

$$\nabla \cdot \boldsymbol{u} = 0 \,. \tag{1}$$

- *Momentum* Conservation (*Navier-Stokes* Equation)

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$$\rho(\boldsymbol{u}\cdot\nabla)\boldsymbol{u} = -\nabla p + \mu\nabla^2\boldsymbol{u}. \tag{2}$$

- Energy Conservation (Energy Equation)

$$\rho c_p(\boldsymbol{u}\cdot\nabla)T = k\nabla^2 T + \delta S. \tag{3}$$

In the energy equation, δ = 1 in the 3D protruding heater region and δ = 0 in the substrate and the fluid regions.

The boundary conditions of the flow were uniform velocity (u_0) at the channel inlet, and null velocity at the solid-fluid interfaces (no-slip condition). At the channel outlet, the flow had its diffusion neglected in the *x*-direction for three velocity components. The thermal boundary conditions considered were uniform temperature (T_0) at the channel inlet and negligible thermal diffusion in the *x*-direction at the channel outlet. The top and bottom surfaces were adiabatic. Perfect thermal contact condition was considered at the interface 3D protruding heater-substrate. Symmetry boundary condition (periodic condition) was applied for the velocity and temperature fields at the lateral boundaries of the solution domain (same geometry and heat dissipation for all the 3D protruding heaters).

B. Thermal Parameters of Interest

The solution of the governing equations output the velocity and pressure distributions in the considered domain. The numerical solutions of the primary variables distribution (*u*, *v*, *w*, *p*) were utilized to define the derived quantities. The *Reynolds* number in the channel was based on the 3D protruding heater height (H_h) expressed by

$$Re = \frac{\rho u_0 H_h}{\mu} = \frac{u_0 H_h}{\upsilon}. \tag{4}$$

The local heat transfer coefficient, $h(\xi)$, was defined based in the difference between the local temperature of the heater surface, $T_h(\xi)$, and the inlet temperature of the fluid in the channel T_0,

$$h_0(\xi) = \frac{q_f''(\xi)}{T_h(\xi) - T_0}, \tag{5}$$

where, $q_f''(\xi)$ represents the local heat flux at the heater surface for the fluid flow.

With the definition of the local heat transfer coefficient, Eq. (5), the heater length L_h was selected as the characteristic length for the local *Nusselt* number at the heater.

$$Nu_0(\xi) = \frac{h_0(\xi)\, L_h}{k}. \tag{6}$$

The average heat transfer coefficient and the average *Nusselt* number of the heaters were respectively defined as

$$\bar{h}_0 = \frac{q_f}{A_{cv}\left(\bar{T}_h - T_0\right)}, \tag{7}$$

$$\overline{Nu}_0 = \frac{\bar{h}_0\, L_h}{k}, \tag{8}$$

where, A_{cv} is the heater surface area in contact with the fluid flow.

The convective thermal conductance $(UA)_{cv}$ is defined as

$$(UA)_{cv} = \frac{q_f}{\left(\bar{T}_h - T_0\right)} \tag{9}$$

where in this case, U_{cv} coincident with the heat transfer coefficient (Eq. (7)).

The overall thermal conductance (UA) is expressed by

$$(UA) = \frac{q}{\left(\bar{T}_h - T_0\right)} \tag{10}$$

where, *A* is the total area of the 3D protruding heater.

C. Numerical Solution

The governing equations and their boundary conditions were numerically solved utilizing the Control Volume Method [45] through the *ANSYS/Fluent*™ 15.0 software. The *SIMPLE* (Semi-Implicit Method for Pressure Linked Equations) algorithm was used to treat the pressure-velocity couple. The discretization of the diffusive-convective terms was done through a Second-order *upwind* scheme. The boundary conditions for the laminar flow and the heat transfer were applied at the boundaries of the analyzed domain. The numerical procedures assumed were verified through a comparison with the numerical results of the thermo-fluid-dynamic parameters presented by [46]. After a mesh independency study, the numerical results were obtained with a 3D non-uniform mesh containing 212,670 control volumes. This mesh was more concentrated in the regions near the solid-fluid interfaces due to the larger gradients in the primitive variables of these regions, as shown in Fig. 3. Due to the non-linearity in the *momentum* equation, the velocity components and the pressure correction were under-relaxed to prevent instability and divergence. The numerical computations were made with the use of a microcomputer equipped with an *Intel*™ *Core i*7 3.6 GHz processor and 16 GB of RAM. The processing time of a typical solution was approximately 15 (fifteen) minutes.

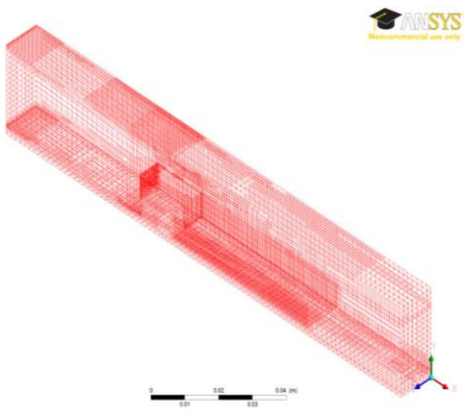

Fig. 3. 3D non-uniform mesh (3D perspective view).

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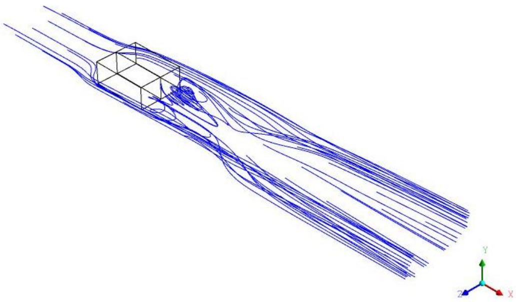
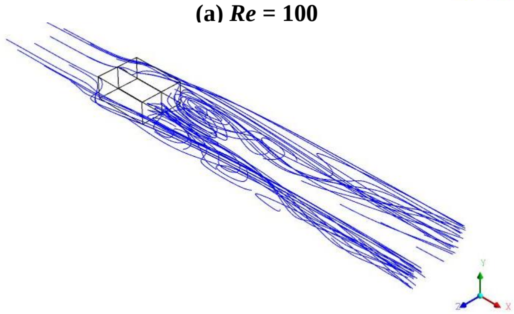
IV. RESULTS AND DISCUSSION

In order to obtain the numerical results, typical design and properties values found in cooling applications of electronic components mounted on a circuit printed board [47]. The geometric configurations showed in Fig. 2 were assumed considering a space of H = 0.0254 m between the parallel plates. The cooling fluid considered in the current study was the dielectric fluid *Novec*^{TM} 7500. The 3D protruding heaters were considered to be made of pure aluminum and the substrate, adiabatic. The properties of the fluids and solids were considered constants at 300 K [11]. The thermo-physical properties of the dielectric fluid according to the manufacture's catalog [7] were equal to c_p = 1,128 J/kg.K, k = 0.065 W/m.K, μ = 0.00124 Pa.s, ρ = 1,614 kg/m^3 and Pr = 21.519. The dissipation rate in each heater was 2W which corresponds to a volumetric heat generation rate of 904,055.5 W/m^3. The effects of the *Reynolds* numbers *Re*=100,150, 200, 250, and 300 were investigated. According to Morris & Garimella [48], the flow is laminar in the channel for this range of *Re*. In Fig. 4, the streamlines around a 3D protruding heater, in a perspective view, are presented for *Reynolds* numbers of 100 and 300, and in Fig. 5, these streamlines are presented in more detail for the region upstream the protruding heater. The main characteristics of the laminar flow are the horseshoe vortices which start upstream the heater and develop around the heater lateral surfaces; a small recirculation upstream the 3D protruding heater; the detachment of the fluid boundary layer at the top of the heater causing a recirculation; and a large recirculation region downstream the heater due to the flow reattachment. It is interesting to state that the fluid flow development around the 3D protruding heater lateral surfaces does not freely happen due to the small space between the heaters.

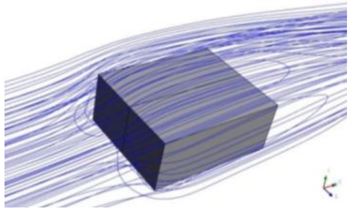


(a) *Re* = 100

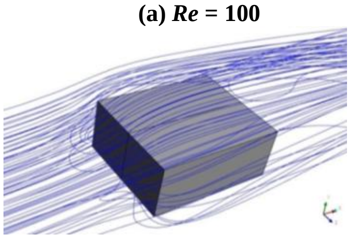


(b) *Re* = 300

Fig. 4. Streamlines around a 3D protruding heater (in a perspective 3D view) [49].



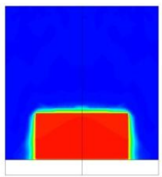
(a) *Re* = 100



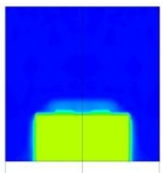
(b) *Re* = 300

Fig. 5. Streamlines around a 3D protruding heater (in a perspective 3D view – detail) [49].

A numerical study of the laminar flow of the dielectric fluid *Novec*^{TM} 7500 around the 3D protruding heaters was presented in detail in [49]. In this work, the authors analyzed the streamlines, the velocity profile, the mean friction coefficient, the pressure distribution and total pressure drop in the channel, the required pumping power, and the *Darcy-Weisbach* friction factor. Considering the adiabatic substrate, the heat transfer at the surfaces of the 3D protruding heater to the dielectric fluid flow happens only through forced convection characterizing a convective cooling process. The isothermal maps for *Re* = 100 and 300 are shown in Figs. 6, 7, and 8 for the *xy*, *xz* and *yz*-planes, respectively. The protruding heater can be considered with a uniform temperature due to its high thermal conductivity. Furthermore, the influence of the laminar flow around the 3D heater in the temperature distribution can be clearly noticed.

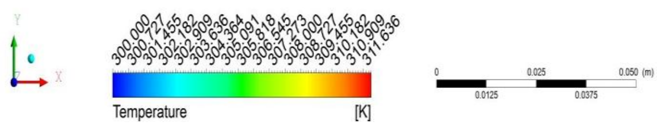


(a) *Re* = 100



(b) *Re* = 300

Fig. 8. Isothermal map at the *yz* plane with *x* = 2.375*H*.



Figs. 9(a), 9(b), and 9(c) show the local adiabatic *Nusselt* number distributions along the lines ABCD, EFGH and IJKL at the 3D protruding heater surfaces, respectively, in function of the *Reynolds* number. Nu_{ad} (ξ) increases with *Reynolds* number. The results of the average heater temperature, average adiabatic *Nusselt* number, convective thermal conductance, heat transfer coefficient, overall thermal conductance and overall heat transfer coefficient are shown in Table 1 in function of the *Reynolds* number considering the forced convection cooling process with laminar flow of the dielectric fluid *Novec*^{TM} 7500. In order to associate the

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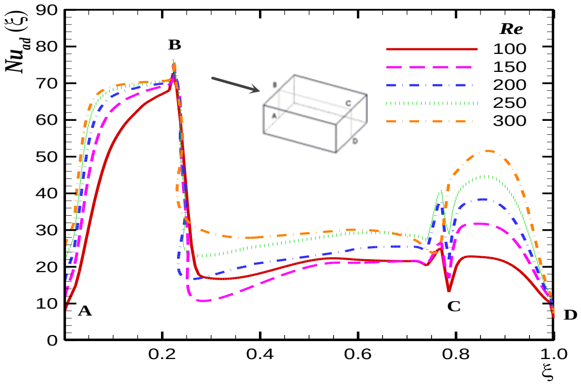
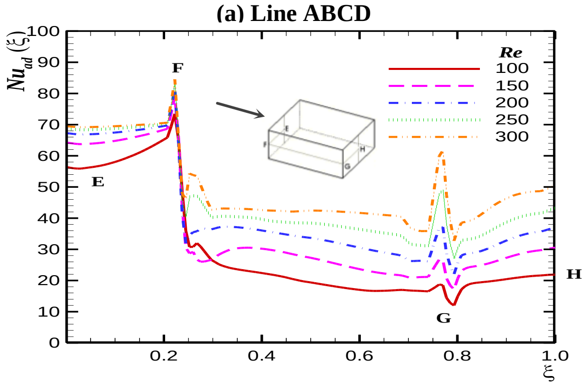
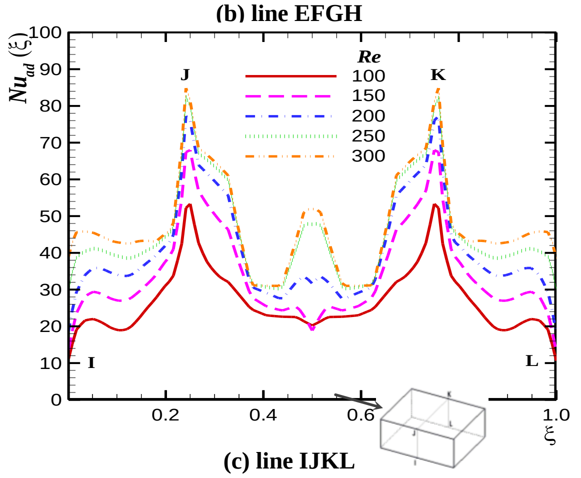
numerical values, the results considering air as the work fluid are also presented in Table 1 [12]. The properties of the air were considered constant, obtained at 300K [11].



(a) Line ABCD



(b) line EFGH



(c) line IJKL

Fig. 9. Local adiabatic Nusselt number distribution

The magnitudes of the average heater temperatures, considering the convective cooling process of the dielectric fluid *Novec*™ 7500, are smaller than the ones obtained when considering the air forced cooling process due to the fact that the thermal conductance (heat transfer coefficients), that are associated with the *Novec*™ 7500 (dielectric liquid), are greater than the ones related to the air. In Fig. 10 the average heater temperature distribution is presented in function of the *Reynolds* number parameterized in the work fluids. As expected, the average temperature decreases with the increasing *Reynolds* independent of the cooling fluid. Fig. 11 illustrates the behavior of the average *Nusselt* number with the *Reynolds* number parameterized in the work fluids. This important thermal parameter increases with *Reynolds* independent of the fluid considered, indicating the drop in the average heater temperature. Furthermore, the magnitudes related to the *Novec*™ 7500 (dielectric liquid) are greater than the ones related to the air.

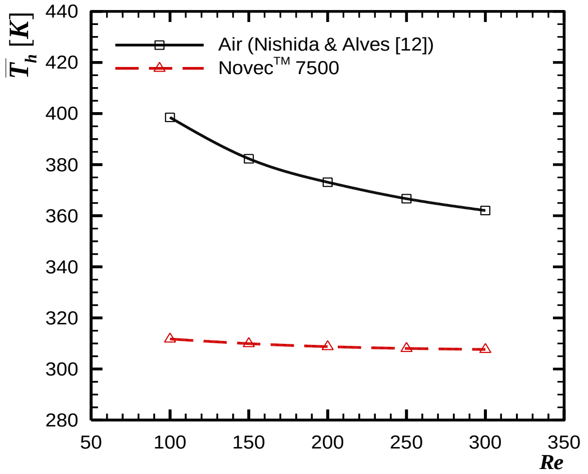


Fig. 10. Average heater temperature.

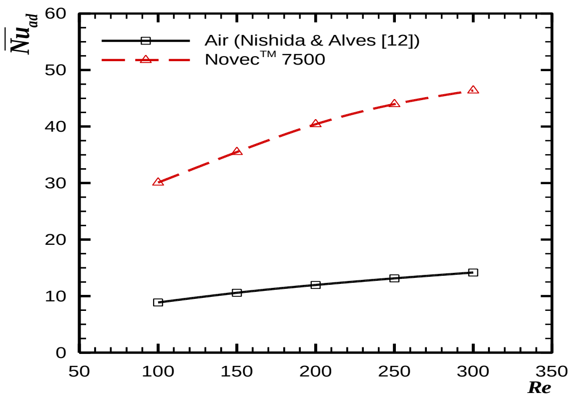


Fig. 11. Average adiabatic *Nusselt* number.

The results of the average adiabatic *Nusselt* number can be correlated for *Novec*™ 7500 with deviations not greater than 1.5% by

$$\overline{Nu}_{ad,\,dielectric\ fluid} = 4.750\,Re^{0.402} \tag{11}$$

A general correlation for the average *Nusselt* number considering the 3D protruding heater mounted on a adiabatic substrate, with deviations not greater than 2.0%, can be expressed by

$$\overline{Nu}_{ad} = 1.504\,Re^{0.414}\,Pr^{0.354} \tag{12}$$

The behavior of the overall thermal conductance, (UA), in function of the *Reynolds* number parameterized in the work fluids is shown in Fig. 12. (UA) increases with Re indicating a greater heat exchange. In this case, since the substrate is adiabatic, the convective thermal conductance and the overall thermal conductance are identical, that way, $(UA) = (UA)_{cv}$. Additionally, the magnitudes related to the *Novec*™ 7500 are greater than the ones related to the air. However, the heat transfer coefficient (Fig. 13) and the overall heat transfer coefficient (Fig. 14) are different. The heat transfer coefficients increases with the *Reynolds* number indicating a greater heat exchange between the protruding heater and the

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fluid flow with the greater mass flow in the channel. Moreover, the *U* values associated with the *Novec*™ 7500 are greater than the ones associated with the air.

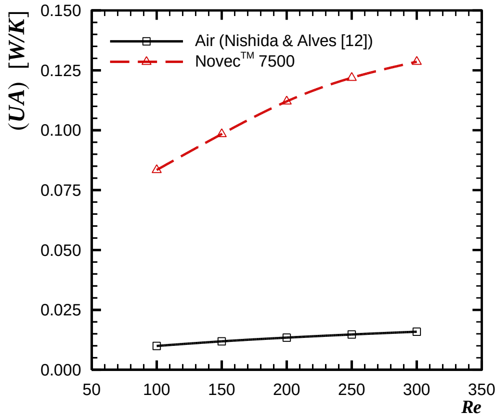

Fig. 12. Overall thermal conductance.

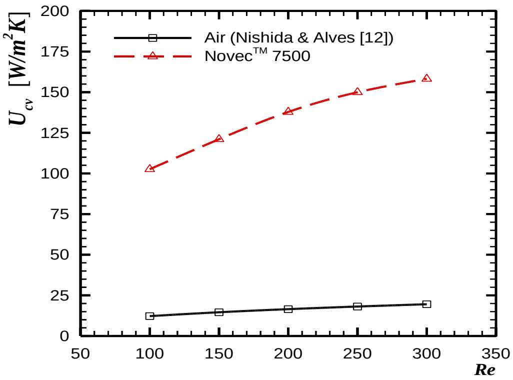

Fig. 13. Heat transfer coefficient

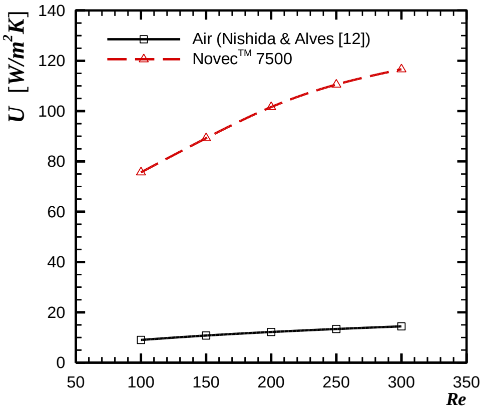

Fig. 14. Overall heat transfer coefficient.

V. CONCLUSION

In the present work a numerical analysis was performed to investigate the forced convection heat transfer with laminar flow of the dielectric fluid *Novec*™ 7500 from a 3D protruding heaters array mounted in cross-stream direction on an adiabatic substrate of a horizontal rectangular channel (Fig. 1) utilizing the *ANSYS/Fluent*™ 15.0 software. A uniform heat generation rate was assumed at the 3D protruding heaters and the cooling process occurred through a forced laminar flow with constant properties under steady state conditions. In the channel entry, the velocity and temperature profiles were uniform. The conservation equations and their boundary conditions were numerically solved within a single domain that includes the solid and fluid regions through a coupled procedure utilizing the Control Volume Method. The occasional effects of the natural convection, radiation and fluid flow oscillation were not considered in the problem modeling. Due to the problem symmetries, the basic configuration of the problem was reduced to the one in Fig. 2. The *SIMPLE* algorithm was used to treat the pressure-velocity couple. The discretization of the diffusive-convective terms was done through a Second-order upwind scheme. Due to the non-linearity of the *momentum* equation, the correction of the velocity components and the pressure were under-relaxed to prevent instability and divergence. The verification of the adopted numerical procedures was performed through a comparison of the thermo-fluid-dynamic parameters of the numerical results with the ones presented in [46]. After a study of the computational mesh independence, the numerical results were obtained, displayed as a 3D non-uniform mesh with 212,670 control volumes (Fig. 3). This computational mesh was more concentrated near the solid-fluid interface regions due to the larger primitive variable gradients in these regions. In order to obtain the numerical results, typical design and properties values found in cooling applications of electronic components mounted on a circuit printed board. The geometric configurations showed in Fig. 2 were assumed considering a space of H = 0.0254 m between the parallel plates. The effects of the *Reynolds* number, based on the protruding heaters height, were inspected for Re = 100, 150, 200, 250, and 300. The flow in the channel was always laminar for the range of Re investigated. The behavior of the laminar flow around the 3D protruding heaters was showed through the streamlines. The streamlines around a protruding heater were presented for *Reynolds* numbers of 100 and 300 (Figs. 4 and 5). The main characteristics of the laminar flow were the horseshoe vortices which start upstream the heater and develop around the heater lateral surfaces; a small recirculation upstream the protruding heater; the fluid boundary layer detachment at the top of the heater causing a recirculation; and a large recirculation region downstream the heater due to the flow reattachment. More information about the laminar flow of the dielectric fluid *Novec*™ 7500 around the 3D protruding heaters can be found in [49]. For the forced convection cooling process (adiabatic substrate), the isothermal maps for Re = 100 and 300 considering the *xy*, *xz* and *yz*- planes were presented (Figs. 6, 7, and 8). The local *Nusselt* number distributions along the lines ABCD, EFGH and IJKL at the surfaces of the 3D protruding heaters in function of the *Reynolds* number were presented in Figs. 9(a), 9(b), and 9(c), respectively. The main thermal parameters of interest, average heater temperature (Fig. 10), average adiabatic *Nusselt* number (Fig. 11), overall thermal conductance (Fig. 12), heat transfer coefficient (Fig. 13) and overall heat transfer coefficient (Fig. 14) were presented in function of the *Reynolds* number in Table 1 considering air and dielectric fluid *Novec*™ 7500. The average adiabatic

Nusselt number was correlated with deviations not greater than 2.0% through Eq. (12). The magnitudes of the average heater temperatures, considering the convective cooling process of the dielectric fluid *Novec*™ 7500, are smaller than the ones obtained when considering the air convective cooling process due to the fact that the thermal conductance (heat transfer coefficients), that are associated with the *Novec*™ 7500 (dielectric liquid), are greater than the ones related to the air. Finally, it is interesting to state that the fluid flow development around the 3D protruding heaters lateral surfaces does not freely happen due to the small space between the protruding heaters. The fluid dynamic symmetry conditions of the blocks were dominant and the corresponding flow and the thermal wake were different than a single 3D protruding heater with free domain in the transversal direction to the flow.

VI. ACKNOWLEDGMENT

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APPENDIX

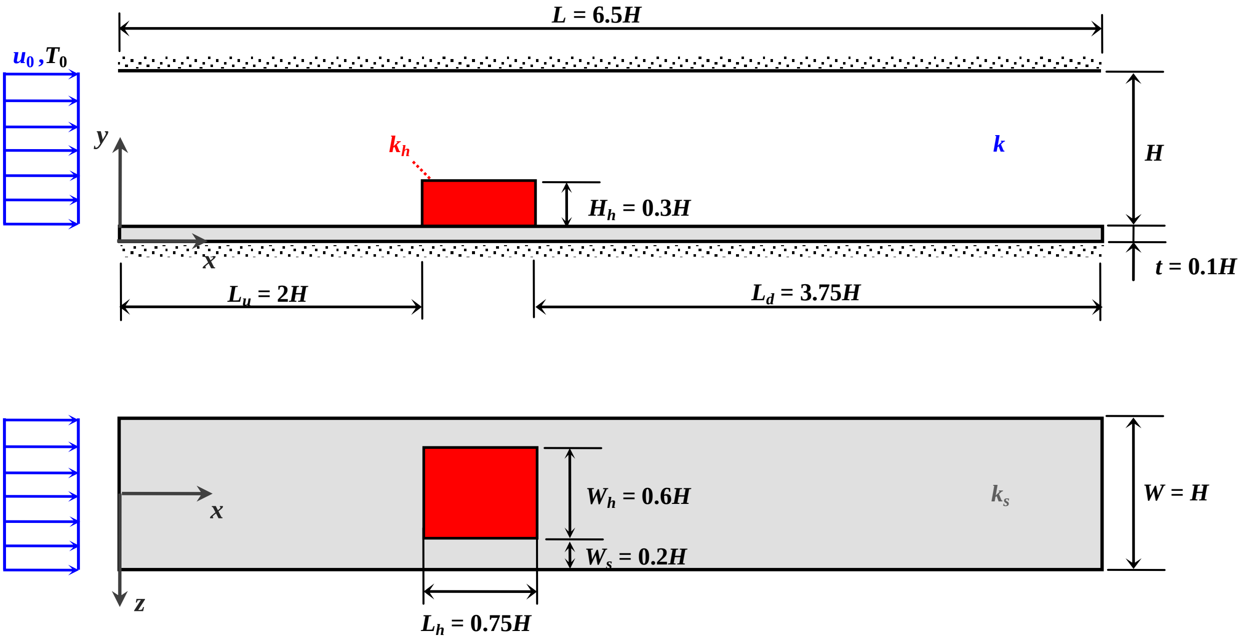
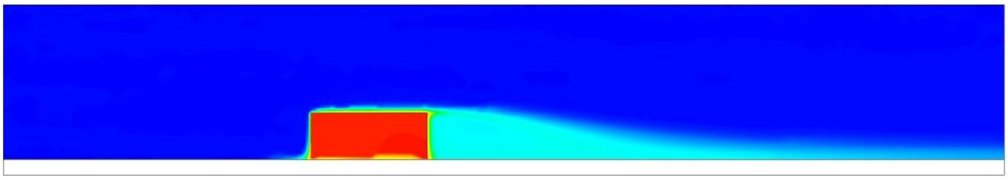
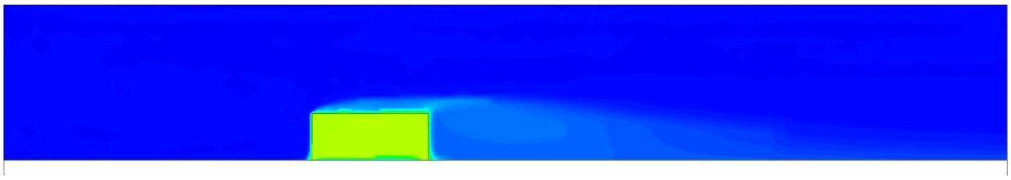


Fig. 2. Basic configuration representing the problem for one of the 3D protruding heaters

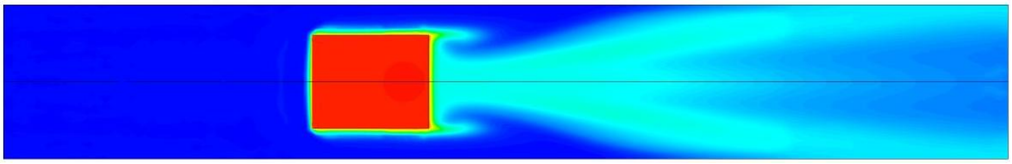


(a) *Re* = 100

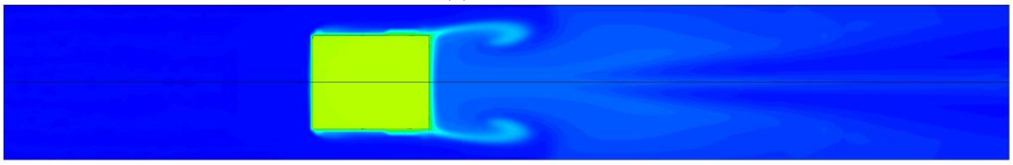


(b) *Re* = 300

Fig. 6. Isothermal map at the *xy*-plane with *z* = 0.



(a) Re = 100



(b) Re = 300

Fig. 7. Isothermal map at the xz plane with y = 0.16H..

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Table 2. Thermal parameters of interest

Re	Novec™ 7500						Air [12]					
	$\bar{T}_h$ [K]	$\overline{Nu}_{ad}$	$(UA)_{cv}$ [W/K]	U_{cv} [W/m²K]	(UA) [W/K]	U [W/m²K]	$\bar{T}_h$ [K]	$\overline{Nu}_{ad}$	$(UA)_{cv}$ [W/K]	U_{cv} [W/m²K]	(UA) [W/K]	U [W/m²K]
100	311.74	30.07	0.0834	102.61	0.0834	75.65	398.50	8.87	0.0100	12.24	0.0100	9.04
150	309.94	35.48	0.0984	121.06	0.0984	89.24	382.32	10.58	0.0119	14.61	0.0119	10.78
200	308.76	40.40	0.1120	137.84	0.1120	101.61	373.12	11.96	0.0134	16.52	0.0134	12.19
250	308.06	43.95	0.1219	149.97	0.1219	110.55	366.66	13.13	0.0147	18.13	0.0148	13.38
300	307.66	46.36	0.1286	158.20	0.1286	116.62	362.05	14.16	0.0159	19.55	0.0159	14.43