CFD Analysis of Constructal Liquid-Cooled Heat Sinks for High-Power Electronics

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Abstract:

The thermal management of new generation integrated-circuit chips demands cost-effective techniques for enhanced device performance, where the higher density of transistors has pushed to the limit the conventional air-cooling technologies. Liquid-cooling has proved to be an efficient solution with relatively high heat transfer coefficients. A common flow-pattern configuration used for cold plates is the parallel channels heat sink due to its manufacturing ease; however, this configuration has the main disadvantage of flow mal-distribution leading to temperature non-uniformity on the heated surface. The present investigation proposes a constructal flow-pattern for liquid-cooled heat sinks in order to improve heat transfer and flow distribution based on optimized designs. The numerical results show that the hot-spots are minimized with the present configuration, therefore allowing the dissipation of higher heat fluxes with a reduced pressure drop due to the bifurcating nature that allows a superior accessibility to the coolant.

Keywords:
Electronics cooling, CFD, constructal design, liquid-cooling.

1. Introduction

Electronic devices are the keystone of current and near future generations.. The trend of miniaturization and larger densities for these components requires the development and implementation of more effective cooling techniques capable to remove relatively high heat fluxes. Air cooling technology has reached its limits for thermal management of high-end electronics, whereas liquid cooling represents a promising opportunity to design, develop and produce cooling devices with much larger heat transfer coefficient [1]. In 1981, Tuckerman and Pease [2] introduced the concept of microchannels by demonstrating that these flow passages are able to remove up to 790 W/cm^2 with a thermal resistance of 0.09 cm^{2}K/W. After this seminal paper, a large number of investigations have been conducted considering conventional parallel microchannels [3, 4]. Although a large part of the results shows very good thermal performance, there are drawbacks. The convective coefficient is enormously enhanced, but with a strong penalty in the pressure drop. Another disadvantage of these conventional arrangements is the non-uniform temperature distribution under constant heat flux conditions which can damage seriously the electronic device. Thus, it is desirable to increase temperature uniformity while decreasing the pressure drop.

Following the evolution of natural systems, their structures can provide ideas to design new microchannel cooling systems. Systems such as trees, leaves, plant, roots, river basins, circulatory and respiratory systems, etc. can shed light on optimal solutions of many practical problems. These organisms have common shapes: a main channel which is bifurcated in two or more channels at a distance, repeating it n-times; these shapes are known as branching networks.

Several publications have investigated the hydrodynamic and heat transfer features in deterministic branching networks, which have been suggested as efficient alternatives for electronics cooling applications.

Pence et al. [5] proposed a one-dimensional model for fractal-like branching network and computed the pressure and temperature distributions. Their results indicated robustness compared to conventional parallel microchannels. Chen and Cheng [6] studied theoretically and experimentally a fractal-like branching network of rectangular shape, indicating that this network can increase the total convective heat transfer and reduce the total pressure drop in the fluid compared with the conventional parallel channels. Zimparov et al. [7] optimized the thermodynamic performance of simple T- and Y-shape flow systems under fixed temperature of the channel walls. Beserni et al. [8] used the constructal theory to optimize the geometry of an H-shaped cavity. They demonstrated that the geometrical complexity should evolve gradually for the global system performance to be improved.

Fan et al. [9] reported an experimental investigation of the flow uniformity and pressure drop in a fractal distributor inspired in a respiratory system; the constructal theory was used to optimize the hydrodynamic performance of the distributor. Escher et al. [10] employed a one-dimensional model to study the cooling performance of a bifurcating tree-like network very similar to the one proposed by Fan et al. [10], demonstrating an inferior thermal performance when compared to a geometry of conventional parallel microchannels. In the present investigation, similar constructal geometries to those proposed by References [9, 10] are analyzed by means of computational fluid dynamics (CFD) for the specific application of liquid-cooling of commercial processors with copper cold plates using minichannels and the specified flow field configurations.

2. Geometry description

Three models with different flow distributions were simulated based on the proposals by Fan et al. [9] and Escher et al. [10] in order to analyze their cooling performance and pressure drop features for the possible application of liquid-cooled plates for commercial processors with a thermal design power of 150 W. Since the trend of using liquid cooling loops inside high-end workstations and servers is gaining a lot of interest and application due to the inherent advantages and the refinement of reliable uninterrupted operation, it is of interest to explore this kind of constructal geometries for enhanced performance. The typical size of a packaged processor is a square geometry with a lateral length of 37.5 mm, while common cold plates used for liquid cooling are designed for this size. In the present analysis, three different square cold plates (50 mm of lateral length) were analyzed consisting of constructal designs with 16, 64 and 128 outlets. Since the analyzed geometries present symmetric features, just one fourth of the domain was simulated in order to reduce the computation time, those symmetric portions of the cold plate for each analyzed case are depicted in Figure 1.

3. Computational model

3.1. Model assumptions

The numerical model used for the analysis of the conjugate heat transfer relies on the following assumptions:

1) Steady state. 2) Laminar and incompressible flow. 3) Single phase flow.
4) Negligible radiation heat transfer. 5) Constant fluid properties.

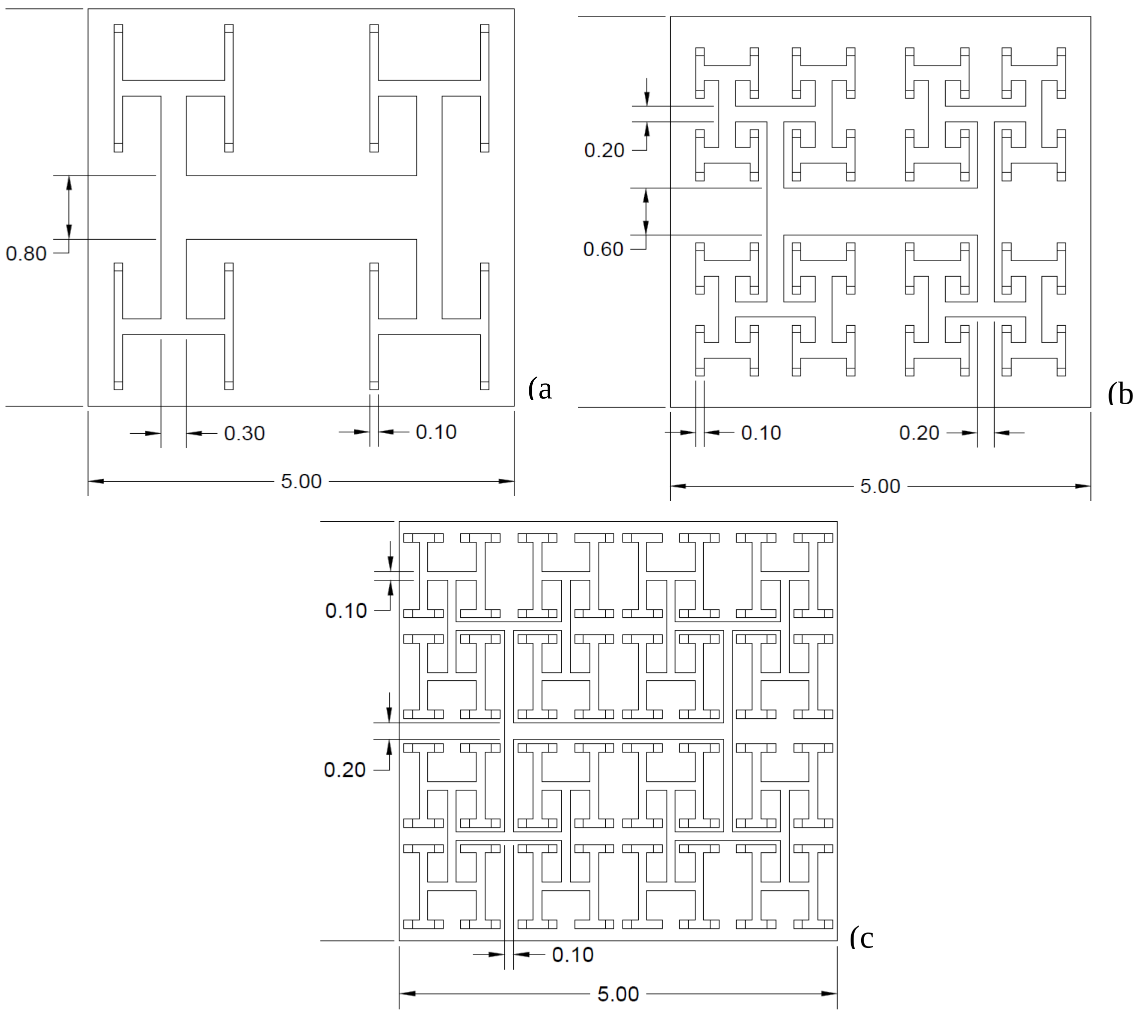

Figure 1. Different constructal geometries used for the simulations (1/4 of actual cold plate): a) 16 outlets, b) 64 outlets, c) 128 outlets.

In the present investigation, deionized water was chosen as the coolant through the minichannels embedded in the cooper cold plate, using the thermophysical properties listed in Table 1.

Table 1. Thermophysical properties of liquid water and copper cold plate (at 297 K) [11].

Property	Value	Units
Water		
Viscosity, μ	$2.414\times10^{-5}(10^{247.8/(T-140)})$	kg/(m-s)
Thermal conductivity, k_f	0.6	W/(m-K)
Density, ρ_f	998.3	kg/m^3
Specific heat, c_{pf}	4.183	kJ/(kg-K)
Copper		
Thermal conductivity, k_{cu}	387.6	W/(m-K)
Density, ρ_{cu}	8978	kg/m^3
Specific heat, c_{cu}	0.381	kJ/(kg-K)

3.2. Governing equations

Considering the previously introduced model assumptions, the conservation equations for mass and momentum of fluid in Cartesian coordinates are:

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

$$\rho_{\mathrm{f}}\left(u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}+w\frac{\partial u}{\partial z}\right)=-\frac{\partial p}{\partial x}+\mu\nabla^{2}u \tag{2}$$

$$\rho_{\mathrm{f}}\left(u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}+w\frac{\partial v}{\partial z}\right)=-\frac{\partial p}{\partial y}+\mu\nabla^{2}v \tag{3}$$

$$\rho_{\mathrm{f}}\left(u\frac{\partial w}{\partial x}+v\frac{\partial w}{\partial y}+w\frac{\partial w}{\partial z}\right)=-\frac{\partial p}{\partial z}+\mu\nabla^{2}w \tag{4}$$

The energy conservation equation for the fluid domain is:

$$\rho_{\mathrm{f}}c_{\mathrm{pf}}\left(u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}+w\frac{\partial T}{\partial z}\right)=k_{\mathrm{f}}\nabla^{2}T \tag{5}$$

while the energy conservation equation for the solid domain with temperature-dependent conductivity is:

$$\nabla\cdot\left(k_{\mathrm{cu}}\nabla\cdot T\right)=0 \tag{6}$$

The area specific local thermal resistance is used as a parameter of comparison in the present study, defined as the surface local temperature rise above the fluid inlet temperature divided by the heat flux:

$$R=\frac{T_{\mathrm{w,ave}}-T_{\mathrm{f,in}}}{q} \tag{8}$$

3.3. Boundary conditions

All of the analyzed cases were prescribed with a constant fluid inlet velocity of uniform profile (non-developed condition) corresponding to a Reynolds number of Re = 430 (equivalent to a mass flow rate of 0.095 kg s^{-1}) with an inlet temperature of 293 K. The walls where fluid and solid domains interact where defined as interfaces, while the walls surrounding the heat sink, except the base, are insulated. A constant and uniform heat flux was applied at the cold plate base with a value of q"= 100 kW/m^2. Non-slip boundary conditions were also defined at the internal interface walls.

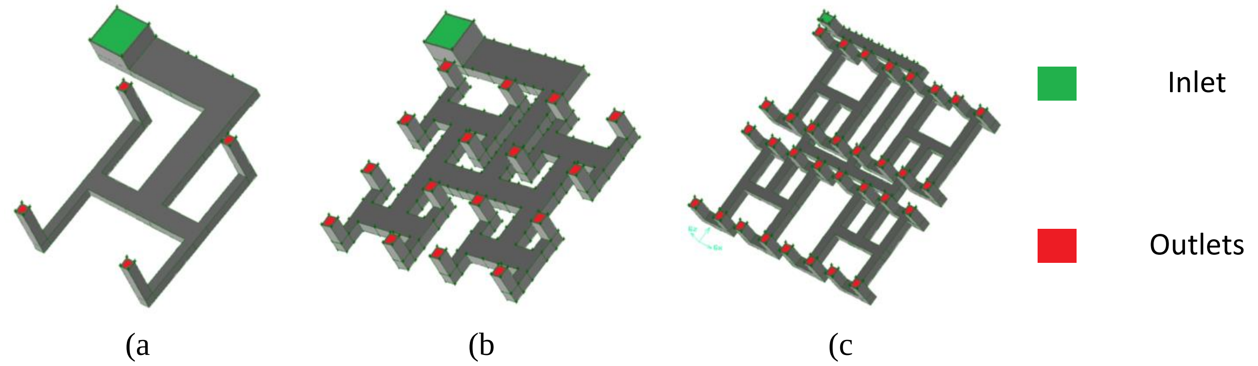

Figure 2. Different constructal geometries used for the simulations (1/4 of actual cold plate): a) 16 outlets, b) 64 outlets, c) 128 outlets.

3.4. Numerical procedure

A commercial computational fluid dynamics code (ANSYS®) on an Intel core i7 4770k was used for the numerical solution of the governing equations with the finite volume method, where the second-order upwind scheme was employed for the discretization. The SIMPLE algorithm was employed for the coupling of the velocity and pressure, and convergence of the solution was considered when the residuals of mass and momentum were below 10^{-6} and below 10^{-8} for the energy equations. Several models with different number of elements were analyzed. Table 2 presents a summary of the results obtained by varying the mesh density. The parameter used for the comparison made is the pressure drop along the heat sink.

Table 2. Grid independence analysis.

Model with 16 outlets		
Mesh elements	ΔP (kPa)	$\frac{\Delta P^j - \Delta P^{j+1}}{\Delta P^j}$
597395	180.2456	$7.947\text{x}10^{-2}$
1018441	174.2516	$3.461\text{x}10^{-4}$
2495384	172.82	--------
Model with 64 outlets		
Mesh elements	ΔP (kPa)	$\frac{\Delta P^j - \Delta P^{j+1}}{\Delta P^j}$
561846	80.2564	$1.8954\text{x}10^{-2}$
985437	68.6822	$2.4687\ \text{x}10^{-5}$
2365143	65.94	--------
Model with 128 outlets		
Mesh elements	ΔP (kPa)	$\frac{\Delta P^j - \Delta P^{j+1}}{\Delta P^j}$
493841	15.6841	$1.5496\text{x}10^{-1}$
1436785	11.9498	$1.1264\text{x}10^{-4}$
2956421	10.91	--------

4. Results and discussion

4.1. Surface temperature profile for the analysed configurations

The surface temperature is of course one of the main parameters of interest, since the main objective is to maintain it at low and uniform values for an enhanced performance. As expected, the temperature profiles are significantly influenced and determined by the shape of the flow channels and the results for all the analyzed cases are summarized in Table 3.

In the present work the main mechanisms of heat transfer involved are conduction and convection with the working fluid. For these types of cooling devices, convection is predominant and the most important mechanism studied. It is widely known that the convective heat transfer coefficient is highly depending on the velocity of the fluid and other properties. The flow patterns proposed and analyzed present different velocities through the flow pattern due the channel configuration of each model. For the model with 16 outlets the covered area with the flow field is lower than the present in other configuration in addition to a lower exhaust area. A larger exhaust area provides lower pressure drop decreasing the velocity of the fluid along the distribution channels.

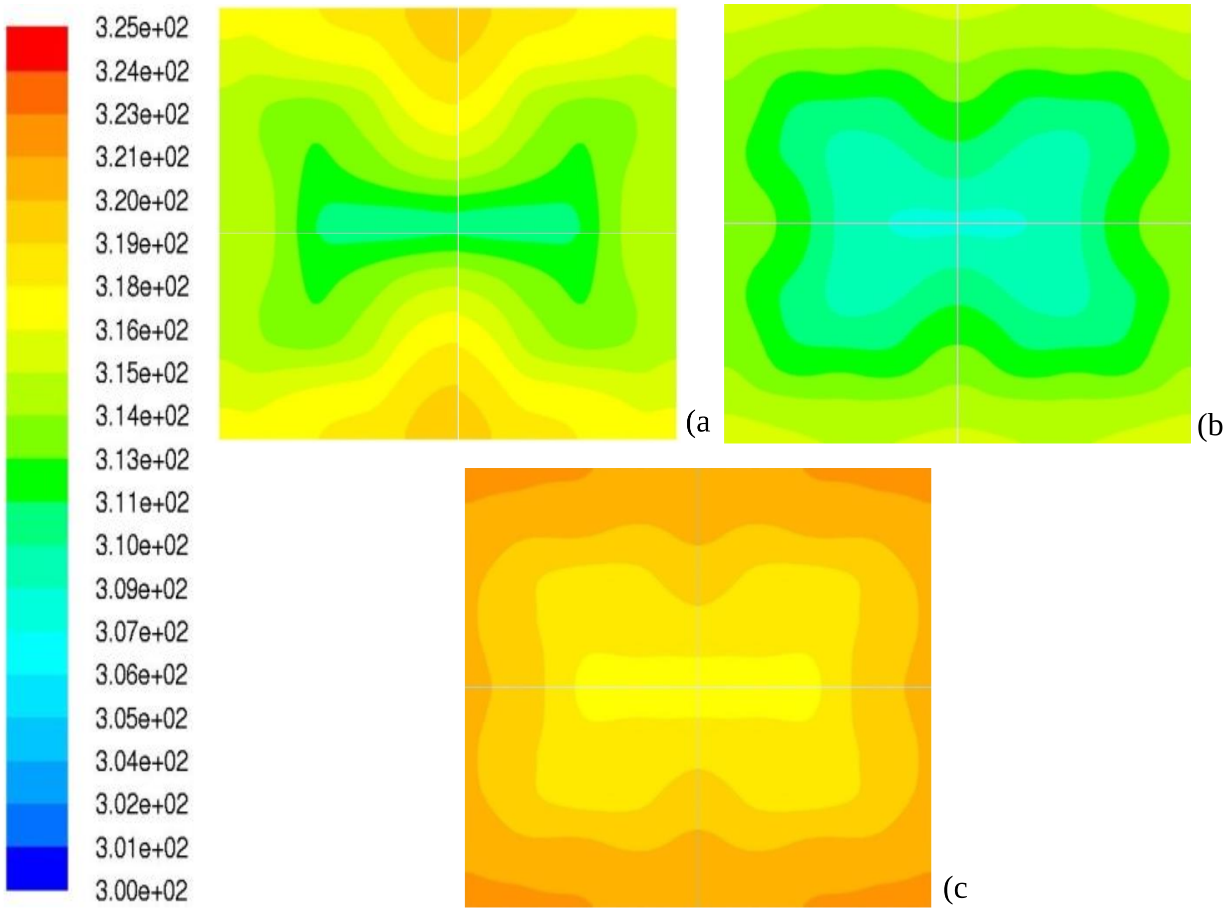

Figure 3. Comparison of the surface temperature for the different configurations (entire cold plate): a) 16 outlets, b) 64 outlets, c) 128 outlets.

For the configuration with 128 outlets the fluid reaches a bigger extension of the cooling plate. However, a big increase in the exhaust area produces inconsistent flow zones. These zones difficult a correct fluid distribution along the cooling plate, interestingly, leading to regions without the presence of fluid. Overall, this phenomenon has an adverse impact in the heat sink's performance. The increase on the number of channels is associated to a decrement in the mass flow rate circulating in each channel. This decrease affects the local heat transfer, leading to the the formation of hot-spots.

The model with 64 outlets provides a good balance between uniformity in the base temperature, low thermal resistance, low pressure drop and pumping power. Figure 3 shows the temperature contour of this configuration. As mentioned before, good uniformity can be observed with a temperature difference of 3.86 K. Table 3 summarizes the results obtained for the three studied models.

Table 3. Summary of results for the three configurations studied.

Number of exits	q'' (kW/m^2)	R_{max} (K/W)	$T_{s,ave}$ (K)	ΔT_s (K)	ΔP (kPa)	$\dot{W}_p$ (W)
16	100	0.077	314.77	4.55	172.82	16.56
64	100	0.061	311.94	3.86	65.94	9.31
128	100	0.086	319.37	2.87	10.91	0.09

Thermal resistance is a parameter that describes in a global way how effective the heat transfer is. A lower value of thermal resistance implies a better heat transfer. Table 3 reports the maximum value of thermal resistance obtained for each configuration. Using this parameter as a reference, the best configuration is the one with 64 outlets reaching a value of 0.061 K/W. The behavior of the thermal resistance is presented in Figure 4. The values of the resistance were taken along the center lines of the cooling flat plane. Figure 4(a) shows the behavior of thermal resistance along the y-axis maintaining a constant value of 2.5 cm in the x-axis, whereas in Figure 4(b) a constant value of 2.5 cm is maintained in the y-axis for showing the value along the x-axis.

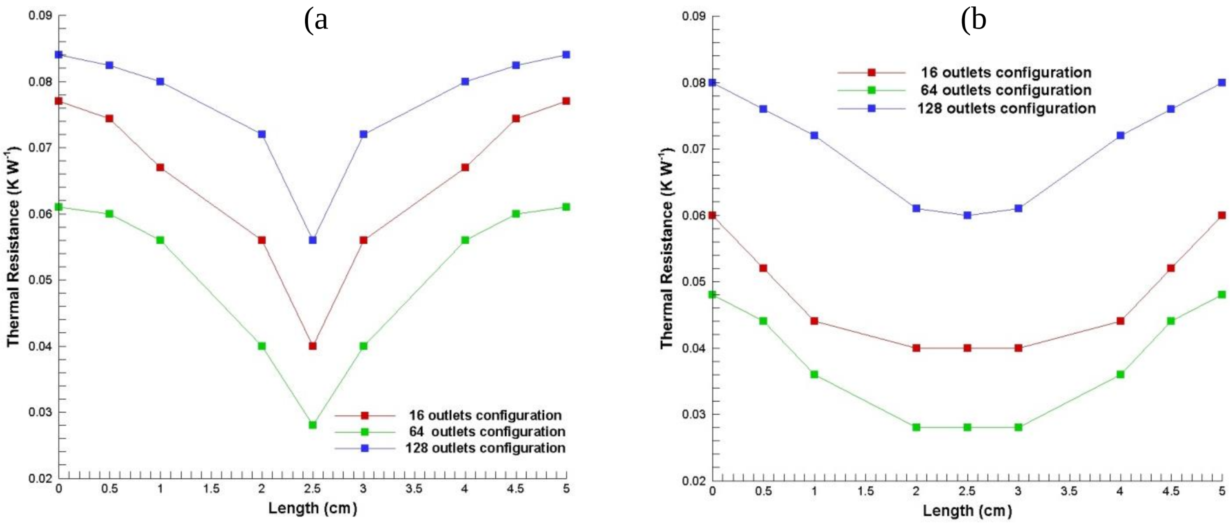

Figure 4. Comparison of thermal resistance along the entire cold plate: a) maintaining x=2.5 cm constant, b) maintaining y=2.5 cm constant.

4.2. Pressure drop

The pressure drop is an important parameter for designing liquid cooling heat sinks. This factor is directly related to the pumping power requirements. An optimal design in this type of cooling devices is the one that presents good temperature uniformity, high heat transfer rate and low pressure drop. Increasing the number of exhausts the pressure drop decreases dramatically; for the 64 outlets model the pressure drop is less than a half of the pressure drop shown by the 16 outlets model. According to the obtained results, the 128 outlets model presents the lowest pressure drop in the study. However, this configuration presents bad temperature uniformity and it is highly exceed on performance by the 64 outlets configuration.

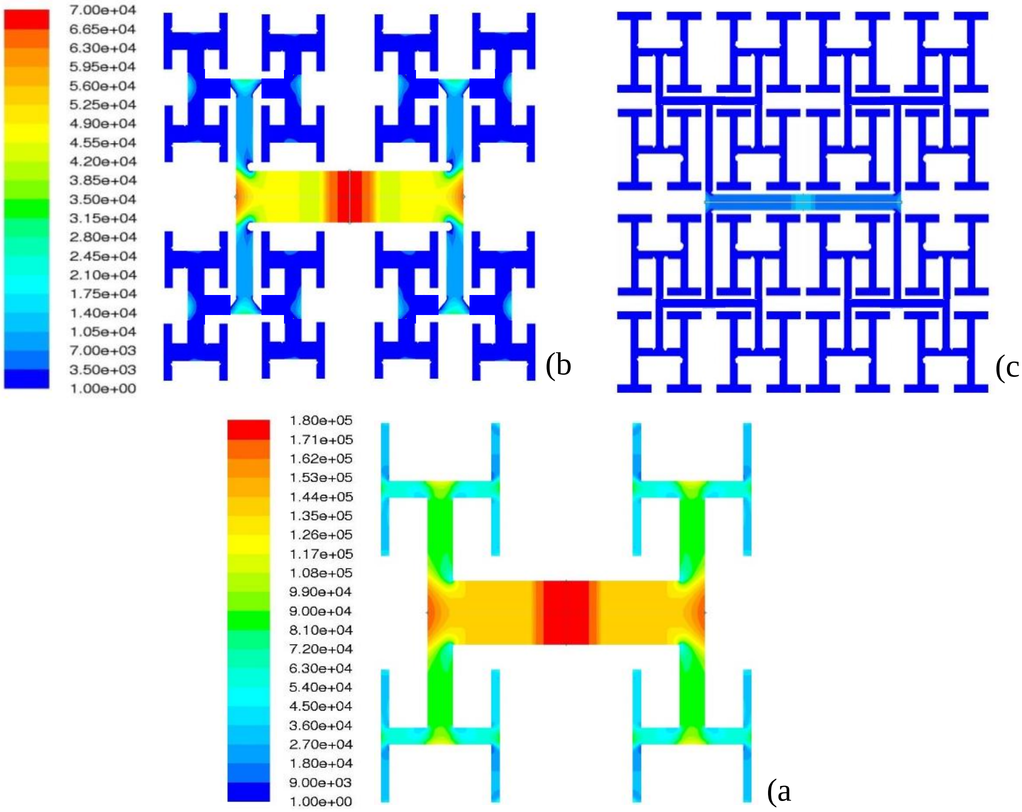

Figure 5. Comparison of the surface pressure drop (Pa) for the different configurations (entire cold plate): a) 16 outlets, b) 64 outlets, c) 128 outlets.

With the addition of exits, the working fluid has fewer restrictions to flow through the channels. The mass flow rate is divided and distributed by the multiple channels; for these reason the velocity in the channels decreases with every bifurcation and exit. The configuration with 128 outlets shows poor flow distribution due the existence of low opposition to flow path-lines. These path-lines are present near the inlet, generating zones with lower mass flow rate.

The 64 outlets configuration presents good balance between covered area, number of exits, pressure drop, and base temperature uniformity. Low pressure drops implies lower pumping power consumption, as mentioned before; this is also an important parameter for designing liquid cooling devices. The model with 64 outlets offers the best relation between all the parameters expected in this type of devices.

5. Conclusions

A CFD analysis for the flow and heat transfer for a non- conventional flow pattern heat sink was carried out in the present paper. The flow regime was fitted to be laminar flow a commercial CFD code was used.

To accomplish a correct comparison between the three different flow distributors proposed in this work, the main parameters analyzed were: the results of temperature uniformity on the flat plates, the maximum and average temperatures at the interface surface, the pressure drop and the pumping power. A suitable study of these parameters will provide the guide for the adequate selection of the best device.

The configuration with 64 outlets presented good flow uniformity in the flat plate heat sink achieving a low maximum temperature difference of 3.86 K in addition to a remarkable low pressure drop compared with the 16 outlets configuration. The 64 outlets configuration showed excellent temperature uniformity in comparison to the rest of the analyzed configurations. The main reason for these results is the excellent flow distribution obtained with the addition of several outlets. The inclusion of several outlets leads to a larger area extension with the channels. The increase on the number of exists provides a notable reduction in pressure drop.

Other important reference parameter in the study of liquid cooling based heat sinks is the pumping power consumption. In order to point an optimal result out of the present analysis, a good relation between the pumping power and the parameters listed before (temperature uniformity and low maximum temperature difference) is sought. The configuration with 64 outlets presents low pumping power per unit of heat flux. As such, the conclusion of this study is that the basic concept of a flow distributor with symmetric flow bifurcations is highly recommended for use in liquid-cooled heat sinks, leading the selection of the 64 outlets heat sink configuration as the best of the 3 studied.

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Nomenclature

K	Thermal conductivity, (W/(m-K))
c_p	Specific heat, (kJ/(kg-K))
q''	Heat flux, (W/m^2)
$\dot{m}$	Mass flow rate, (kg/s)
R	Thermal resistance, (K/W)
T	Temperature, (K)
ΔP	Pressure drop, (Pa)
$\dot{W}_p$	Pumping work, (W)
U	Velocity component in x direction, (m/s)
V	Velocity component in y direction, (m/s)
W	Velocity component in z direction, (m/s)
	Greek symbols
ρ	Density, (kg/m^3)
μ	Dynamic viscosity, (kg/(m-s))
	Subscript
F	Fluid
Cu	Cooper
Ave	Average
In	Inlet
S	Surface
P	Pump

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